Energy and Buildings 69 (2014) 1–13

Contents lists available at ScienceDirect

Energy and Buildings

journal homepage: www.elsevier.com/locate/enbuild

Optimization and analysis of a multi-functional heat pump system with air source and gray water source in heating mode

Xiaoyu Liu, Siu-Kit Lau [*,1], Haorong Li

The Charles W. Durham School of Architectural Engineering and Construction, University of Nebraska-Lincoln, 1110 South 67th Street, Omaha, NE 68182-0816, United States

ARTICLE INFO

Article history:
Received 15 April 2013
Received in revised form 18 August 2013
Accepted 28 September 2013

Keywords:
Multi-functional heat pump
Space heating
Hot water supply
Energy efficiency

ABSTRACT

A multi-functional heat-pump system utilizing the ambient air and gray water as heat source and sink has been optimized for heating and cooling of residential buildings, respectively. This optimization is made based on an existing prototype of the multi-functional heat pump system in the authors' previous study [1,2]. The optimized prototype consists of a heat pump system and a hot water supply system. The prototype is set in two environmental chambers that they simulate the outdoor and indoor environments, respectively, for performance testing. The system was designed to allow four combinations of a water-source evaporator and an air-source evaporator acted as heat sources in space and/or hot water heating modes. The four combinations consist of (a) air-source-only, (b) water-source-only, (c) air-and-water-sources-in-parallel, and (d) air-and-water-sources-in-series, in the refrigerant cycle. In this paper, the drawbacks of the initial prototype in heating mode are discussed and the modifications are proposed. The results show that the performance of optimized prototype superior to that of initial one. The optimized multi-functional heat pump system, compared with initial prototype, can be more practical and provide significant energy savings in space heating and hot water supply.

1. Introduction

Heat pumps are popular for heating and cooling applications. Ground source, air source, combining solar energy and geothermal heat pump were proposed by numerous researchers [3]. Ambient air is an excellent source in terms of availability, thus air source heat pumps are widely used and comparably low initial cost. Compared with other heat sources, air source generally has two problems: (1) the absolute temperature level compared with most other heat sources is relatively low and (2) the annual temperature variation makes the heat output characteristics for the heat pump become quite opposite to the desired one, i.e. low capacity when it is cold outside [2]. Solar source is green and free energy source. Kara et al. [4] and Kuang and Wang [5] applied the direct expansion solar-assisted heat pump for space heating, space cooling and hot water supply. Solar energy has the advantage of not increasing the heat load for the earth and also be safe for the environment and ecological [6]. Ground source can provide vast geothermal energy that is also green and free. Ozgener and Hepbasli [7–9] developed a multi-function heat pump system by utilizing solar energy and geothermal heat. The major disadvantage of solar source and ground source heat pumps is the high initial cost. Moreover, the heat balance of soil may be a problem for maintaining high operational efficiency of the ground source heat pump. Gray water, that is a type of wastewater, is discharged from buildings which usually have a higher temperature than that of outdoor air and freezing. Wastewater source comes from cold and hot water usages in buildings, and thus it is an intermittent source. Numerous researchers have investigated the utilization of multiple heat sources to overcome some abovementioned weakness of a single heat source and to improve the heat pump system performance. Parallel and serial configurations are two fundamental configurations of the heat exchangers for heat pumps with multiple heat sources. Ito and Miura [10] study the parallel combinations of air source and water source via experiments. They find that the system with dual heat sources may have a higher evaporation temperature and COP than the case of a single heat source. Liu and Bullard [11] developed simulation models of a heat pump with multi evaporators. However, there is a lack of publications found by the authors on the series configuration of heat exchangers using different heat sources.

In the U.S. residential buildings, the largest energy consumption is space heating, followed by the water heating and the space cooling, as shown in Fig. 1. The space heating accounts for about 45%, which is almost half of the total energy consumption in residential buildings. Water heating consumes 18% of the total energy

* Corresponding author. Tel.: +86 21 33386688; fax: +86 21 33386699.
E-mail addresses: elau@armstrong.com, sklau@engineer.com (S.-K. Lau).
[1] Present address: Armstrong (China) Investment Co., Ltd., 10/F Block B, Greenland Center, No. 600 Middle Long Hua Road, Xuhui District, Shanghai 200032, PR China.

0378-7788/$ – see front matter
http://dx.doi.org/10.1016/j.enbuild.2013.09.039

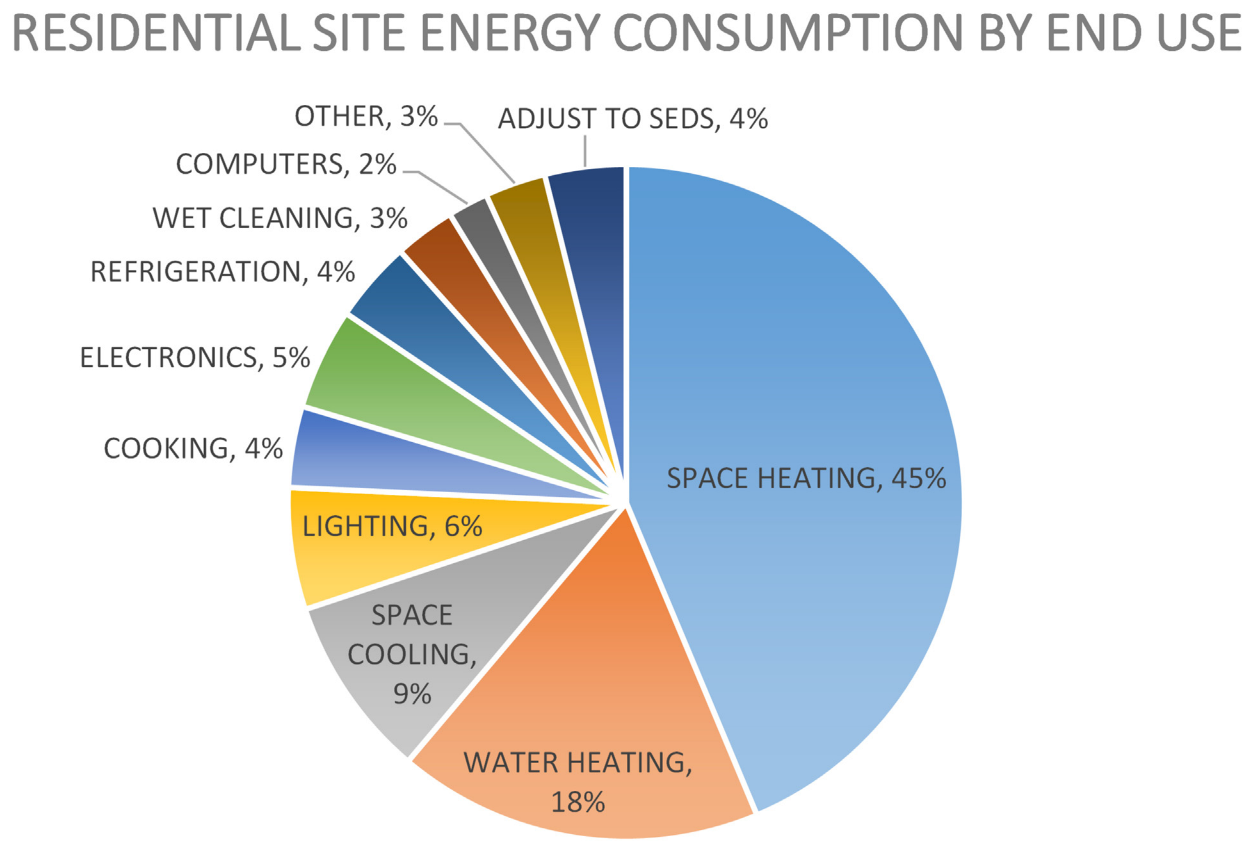

Fig. 1. Residential site energy consumption in the U.S. by end-use [39].

consumption and the percentage of energy consumption for space cooling is half of that for water heating in residential building. Obviously, space heating, water heating and space cooling play essential roles on energy savings. The potential energy sources (for instance, air and water sources) have been investigated [10,12–18]. These research works have been focused on saving energy in supplying hot water, and did not consider the possibility of saving energy using integrating air conditioning and hot water supply systems. Heat pump water heaters have disadvantages of comparably low hot water production rate, large storage tank and the requirement of extra electric backup heater [19].

Ying [20], Cook [21], Goldschmidt [22] and Toh and Chan [23] investigated on recovery of condenser heat from the air conditioner to supply hot water while providing space cooling. From their experimental and numerical results, they observed that the COP of the system could be increased while supplying domestic hot water do not lower down the cooling capacity. Jiang et al. [24] further investigated the recovery of condenser heat in air conditioners via experiments. Substantial energy savings was confirmed by their works. It is noted that air conditioners are only operating at some periods of time over the year in winter and summer; however, the daily demand of hot water is almost stationary throughout all year around. Therefore, a major limitation for the auxiliary condenser heat recovery system is that hot water is not always available. It is because condensing waste heat is only available while air conditioners operate in cooling mode [24]. Moreover, this kind of heat recovery only can reclaim approximately 10–25% condensing heat in total condense heat [25].

To improve the performance of heat pumps, previous studies have investigated the multi-functional heat pump system that not only provides hot water but also space heating and cooling. The research work can be divided into groups with simulations and experiments, respectively. Techarungpaisan et al. [26] built the steady-state model of a split type air conditioner with integrated water heater. Murphy [27] developed the multi-functional air source heat pump system which can provide space heating, space cooling and hot water based on TRNSYS platform. Rad et al. [28] developed a multi-functional heat pump system model that utilizes solar source and ground source. In their study, the energy savings of the system applied in six cities in Canada are presented. Ji et al. [19,29] has developed a prototype system and simulation program for an integrated domestic air-conditioner and water heater. Hong [30] built a prototype of multi-functional solar assisted air source heat pump system. These research works mainly focus on the residential buildings application. Coskun et al. [31] proposed and studied the application in industrial of a novel hybrid district energy system utilizing geothermal and biogas. However, a multi-functional heat pump system utilizing the heat from gray water and with different heat source combinations is not investigated in previous research.

This paper proposes an optimized prototype of a multi-function heat pump system with utilizing air and gray-water heat sources focusing on improving the overall efficiency of the energy utilization in residential buildings. Liu et al. [1,2] has experimentally studied the performance of an initial prototype of the multi-functional heat pump system. An author of this paper has been involved in the development of a gray water treating system consisted of a simple screen, a bio-filter filled with shredded tire chips and a membrane bioreactor for this application [32]. Through optimization of the initial prototype, current prototype shows better performance and stability in use. In the present study, the system performance in different functions with different types of heat source combinations in heating mode will be discussed.

2. Prototype setup

2.1. Prototype description

Fig. 2 shows the schematic diagram of the prototype of the multi-functional heat pump system. This prototype consists of a heat pump system and a hot water supply system. The heat pump system 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, a gray water tank with an immersed heat exchanger and a refrigerant liquid receiver. The hot water supply system consists of a water pump for circulating the water in the pipe, a 30-gallon hot water tank for storage of hot water, and two plate heat exchangers in parallel combination or separately used for heating hot water. This multi-functional heat pump system can realize five functions: space heating only, space heating plus hot water, space cooling, space cooling plus hot water and hot water supply only. Two rooftop units (RTU) are used to control the temperature in the outdoor chamber and the indoor chamber, respectively.

In this system, four types of heat source combinations can be implemented as showed in the following parts:

2.1.1. Air source only

The air-to-refrigerant (or air-source) heat exchanger in the outdoor chamber serves as an evaporator, while the air-to-refrigerant heat exchanger in the indoor chamber works as a condenser. The refrigerant flow directly enters the outdoor heat exchanger, without passing through the immersed water-to-refrigerant heat exchanger in the gray water tank.

2.1.2. Water source only

The heat source is the waste heat of gray water coming from residential buildings. The water-to-refrigerant (or water-source) heat exchanger immersed in the gray water tank works as an evaporator and the air-to-refrigerant heat exchanger in the indoor chamber operates as a condenser. The refrigerant flow passes through the water source heat exchanger without going through the air-source heat exchanger. Gray water typically has a higher temperature than outdoor air in winter. It can improve heat pump system performance.

2.1.3. Air and water sources in parallel (parallel source combination)

The air-to-refrigerant and water-to-refrigerant heat exchangers are arranged in parallel in the refrigeration cycle and both serve as evaporators in heating mode. The heat exchanger in the indoor chamber acts as a condenser. Refrigerant flow passes through two evaporators simultaneously. This arrangement will adjust the amounts of refrigerant flow passing through different heat exchangers to utilize the heat in different heat sources.

2.1.4. Air and water sources in series (series source combination)

The air-source and water-source heat exchangers are arranged in series in the refrigeration cycle. The refrigerant flow passes through the water-to-refrigerant heat exchanger and the air-to-refrigerant heat exchanger sequentially. In this prototype, water-source heat exchanger is ahead of air-source heat exchanger. The discussion of the arrangement of two heat exchangers can refer to Liu et al. [1]. Absorbing heat from gray water is put at the priority position in this arrangement since refrigerant evaporation normally absorbs more heat than sensible heating.

In Table 1, V1 to V6 presents labels of the six solenoid valves, as shown in Fig. 2, installed in the heat pump system to switch the heat source combinations. The default positions of all the valves are closed. When air source is the only heat source, only solenoid valve V1 is opened. When the water source is the only heat source, solenoid valve V5 is opened and the ball valve installed at the outlet of the air source heat exchanger is closed. When air source and water source work in parallel, V1 and V5 valves are opened. While air source and water source work in series, only V3 valve is opened. Pump 1 presents the circulating pump in gray water tank and pump 2 presents the circulating pump in the hot water supply system as shown in Fig. 2.

2.2. Instrumentation and test procedure

In this prototype system, one four-way valve is installed at the outlet of the compressor to switch between heating and cooling modes. Six solenoid valves are used to form different heat source combinations. There are two thermal expansion valves being used for heating mode and cooling mode. The pressure and temperature of the refrigerant flow are measured at the locations as shown in Fig. 2. Six pressure sensors are installed at different positions to measure the pressure distributions of the refrigerant flow. Temperatures are measured with copper-constantan thermocouples and platinum resistance thermometers. The air flow rate and temperature of air flow through the air source heat exchanger are also measured by hot wire flow meters and thermometers. 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 of 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 are recorded at each one second interval. The uncertainty analysis of the experimental test can be found in Appendix.

In heating mode, this system can realize three functions, which are (i) space heating only, (ii) space heating plus hot water supply and (iii) hot water supply only.

2.2.1. Space heating only mode

An air handler is used to dissipate heat into the room air. The circulation pump in the hot water supply system is turned off and thus there is no water circulation for hot water heating. The heat loss can therefore be neglected, when the refrigerant vapor from the outlet of compressor passes through the plate heat exchanger.

At the space heating mode, there is no hot water supply, and the COP of the system at any time instant (t) can be defined as

$$COP_{h,t} = \frac{Q_h(t)}{W(t)}, \tag{1}$$

where $Q_h\,(t)$ is the heat exchange rate in the condenser. $W\,(t)$ is the power of the compressor and fans. Within an operating period of duration τ, the average COP is defined as

$$COP_{h,avg} = \frac{\int_0^{\tau} Q_h(t)\,dt}{\int_0^{\tau} W(t)\,dt}. \tag{2}$$

2.2.2. Space heating plus hot water supply mode

In this mode, hot water heating and space heating are provided simultaneously with space heating supply in priority. The refrigerant vapor first passes through the smaller plate heat exchanger for heating hot water and then goes into the heat exchanger in the air handler for space heating. The smaller plate heat exchanger and heat exchanger in the air handler both work as condensers. The pump in the hot water system is turned on to maintain a constant flow rate (10 L/min) passing through the plate heat exchanger. This mode tries to improve the performance of the system and maximize utilization of the energy, while minimize the effect of supplying hot water for providing space heating. The mode especially fits when the system is oversized or under the partial load condition.

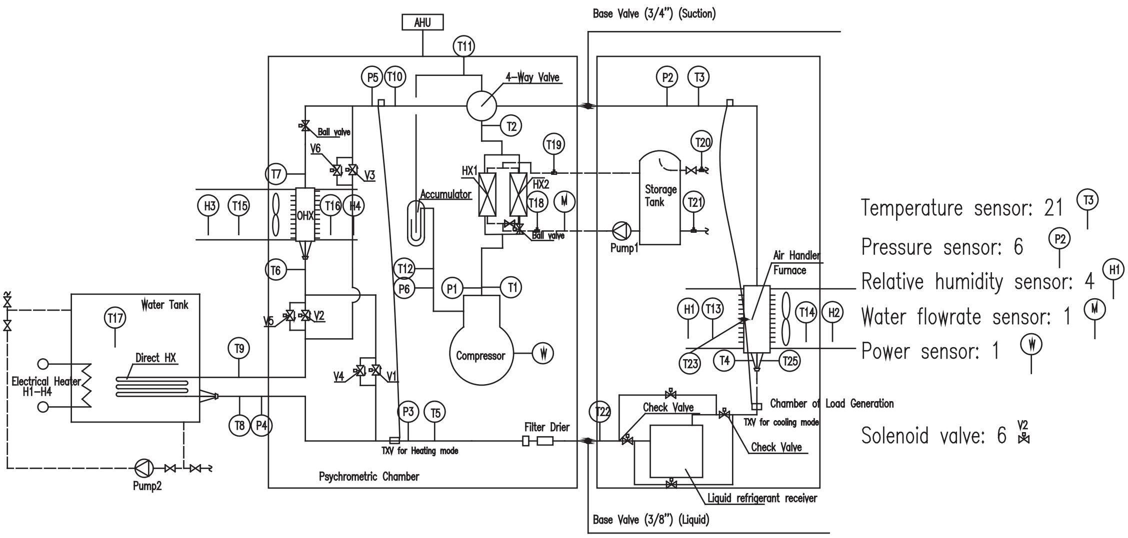

Fig. 2. Schematic diagram of the optimized prototype (multi-functional heat pump system).

Table 1
Heat source combination in heating mode.

Heat source combinations	Opened valves	Space heating only	Space heating plus hot water supply
Air source only	V1	Pump 1 off, pump 2 off	Pump 1 on, pump 2 off
Water source only	V5 and a ball valve	Pump 1 off, pump 2 on	Pump 1 on, pump 2 on
Air and water sources in parallel	V1 and V5	Pump 1 off, pump 2 on	Pump 1 on, pump 2 on
Air and water sources in series	V3	Pump 1 off, pump 2 on	Pump 1 on, pump 2 on

In the present study, the satisfied hot water temperature is set at $48.9 \pm 1\,^{\circ}\mathrm{C}$ according to the Building America Research Benchmark Definition [33].

When the space heating mode and hot water supply are required simultaneously, the system COP at any time instant (t) is given as

$$COP_{h,t} = \frac{Q_h(t) + Q_{h,w}(t)}{W(t)}, \tag{3}$$

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

$$COP_{h,avg} = \frac{\int_0^{\tau} Q_h(t) + Q_{h,w}(t)\,dt}{\int_0^{\tau} W(t)\,dt}. \tag{4}$$

2.2.3. Hot water supply only mode

In this mode, the fan in the air handler located in the room is turned off. The heat loss of refrigerant flow through the heat exchanger of the air handler can be neglected. Only hot water heating is provided. The refrigerant vapor first passes through two plate heat exchangers in parallel combination for heating hot water and then passes the heat exchanger in the air handler. Two plate heat exchangers are acted as condensers. The pump in the hot water system is turned on to maintain a constant flow rate (20 L/min) passing through the plate heat exchanger. This mode tries to provide the maximum ability of hot water production for hot water shortage condition. In the present study, the satisfied hot water temperature is maintained at $48.9 \pm 1\,^{\circ}\mathrm{C}$.

When the only hot water supply is required, the system COP at any time instant (t) is given as

$$COP_{h,t} = \frac{Q_h(t) + Q_{h,w}(t)}{W(t)}, \tag{5}$$

where $Q_{h,w}(t)$ is the heat exchange rate in the plate heat exchanger for heating hot water. $W(t)$ is the power of the heat pump system includes compressor and fans. As mentioned before, the pump used for circulation of hot water is neglected in the calculation due to its small power at 30 W. Within an operating period of duration τ, the average COP is defined as

$$COP_{h,avg} = \frac{\int_0^{\tau} Q_h(t) + Q_{h,w}(t)\,dt}{\int_0^{\tau} W(t)\,dt}. \tag{6}$$

3. Optimization analysis

3.1. Optimization plan for heating mode

This optimized multi-functional heat pump system is developed based on the initial prototype as discussed in Liu et al. [1,2]. Fig. 2 shows the schematic diagram of the optimized prototype and the schematic diagram of the initial prototype can be found in Ref. [1]. There are four modifications in optimized prototype from the initial prototype. First is installing a refrigerant liquid receiver. Second is installing a thermal expansion valve to replace the capillary tube for cooling mode. Third is installing one more plate heat exchanger in the hot water supply system. Forth is that installing a ball valve at the outlet of the air source heat exchanger that works as an evaporator in heating mode. Details of the modification will be provided in this section.

According to the analysis of the system performance of the initial prototype in heating mode [1], there are existing three problems which can degrade system performance in the heating mode.

Firstly, the refrigerant amounts required in heating mode and cooling mode are different. The demand of refrigerant in cooling mode is generally much larger than the demand in heating mode. In conventional air source heat pump system for residential building, the refrigerant amount is kept at a balance point between the demands of the heating mode and cooling mode. However, for multi-functional heat pump system, the system is much more complicated than the conventional air source heat pump system. The refrigerant amount used in the multi-functional heat pump system is larger than that in the air source heat pump. The operation conditions vary often and different operation conditions need different refrigerant amounts. The prototype of the multi-functional heat pump system is developed from a conventional air source heat pump. The suggested refrigerant amount for the originally conventional air-source heat pump system cannot be used. Considering the different demand of refrigerant in heating mode and cooling mode, a balance point is found through the tests. However, the balance point keeps the initial prototype working but also limits the further improvement of the prototype's performance. Moreover, the demand of the refrigerant amount varying sharply and quickly during switching operation conditions and functions was observed in the experimental tests of the initial prototype. This is because different number heat exchangers were used in separate refrigeration cycle. Considering these drawbacks, a refrigerant receiver is proposed to control the demand of the refrigerant amount during system operation.

Secondly, according to the experimental tests of the initial prototype [1], the COP of the initial prototype is very low (i.e. less than 2) under the hot water supply only mode. It is because there are two modes in supplying hot water. They are providing hot water and space heating simultaneously and only supplying hot water, respectively. A plate heat exchanger cannot individually handle these two situations. In space heating plus hot water supply mode, one smaller plate heat exchanger is used to recover the super heat part of the refrigerant discharged by the compressor and minimize the impact on space heating. While in hot water supply only mode, one larger plate heat exchanger plate heat exchanger is installed with the smaller plate heat exchanger in parallel to utilize the whole condenser heat and avoid causing high inlet temperature of the compressor. At the outlet of the larger plate heat exchanger, a ball valve is installed to control the refrigerant flow through the larger plate heat exchanger or not.

Thirdly, the outdoor air temperature has a large influence on the system performance under water source only on the initial prototype [1]. However, the COP and the heating capacity of the system with water source only should not be theoretically affected by the outdoor air temperature, because no heat should be absorbed from the outdoor environment. In the initial prototype design, the system performance is affected by several factors. On the one hand, the heat loss of refrigerant pipes exists. On the other hand, in the heating mode, the outlet of air source evaporator is connected with the outlet of the water source evaporator in the initial prototype. The temperature sensor of the thermal expansion valve is installed after the merge point of the outlets of the air and water source evaporators. When the low pressure and low temperature refrigerant vapor leave the water source evaporator, some of the refrigerant vapor may enter to the air source evaporator and evaporates again due to the low outdoor air temperature and the downstream pipe resistance. It results in the outdoor air temperature affects the temperature measured by the sensor and affects the refrigerant flow rate in the initial prototype. Based on the analysis of the initial prototype, a ball valve is installed at the outlet of air source evaporator when the air source heat exchanger works as an evaporator in heating mode for optimized prototype.

3.2. Comparison of the system performance of initial and optimized prototypes

To verify the optimized system, the performances of two multi-functional heat pump system prototypes have been compared. The effects of the modifications on the optimized prototype's performance are investigated. This comparison is made on typical

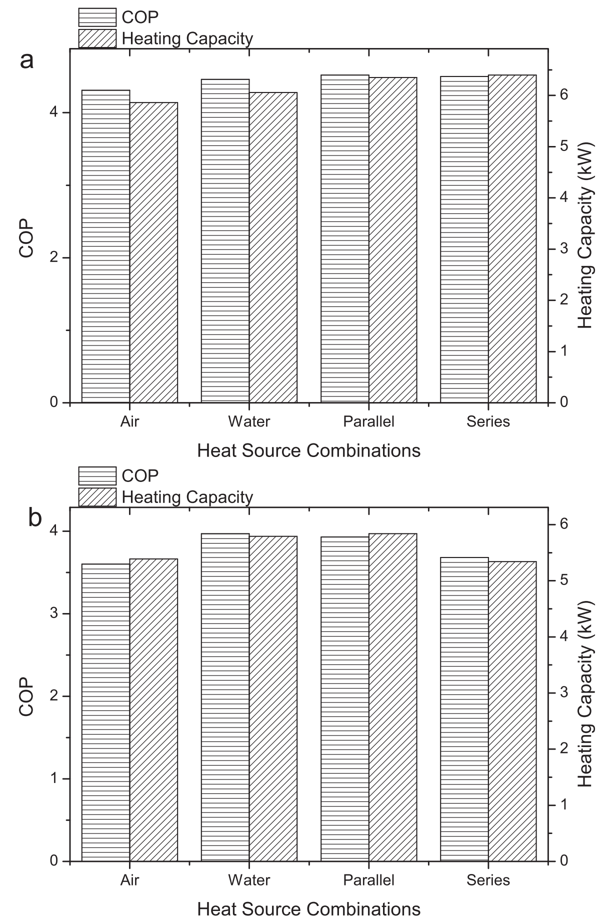

Fig. 3. (a) Optimized prototype in space heating mode at outdoor air temperature 8.3 °C with various heat source combinations. (b) Initial prototype in space heating mode at outdoor air temperature 8.3 °C with various heat source combinations.

operating conditions in space heating mode. In this comparison, the outdoor and indoor temperatures were at 8.3 ± 1 °C and 21.1 ± 1 °C, respectively, referring to the rated operating condition. The gray water temperature was set as 21.1 ± 1 °C according to AHRI 320 [34]. Other parameters are based on the standard ANSI/AHRI Standard 210/240 [36] and ANSI/AHRI Standard 320 [34,35].

Fig. 3(a) illustrates the coefficients of performance (COP) and heating capacity of the optimized multi-functional heat-pump system in space heating mode with various heat source combinations. The "air", "water", "parallel", "series" denote the air-source-only, water-source-only, parallel source combination, and series source combination, respectively. The parallel source combination has the highest COP and heating capacities followed by the series source combination. Fig. 3(b) shows the COP and heating capacity of the initial multi-functional heat pump system. The COP with the parallel source combination is slightly less than that with the water source only. But the heating capacity of the parallel source combination is higher. Compared of the system performance of the optimized prototype and the initial prototype, the COP of optimized prototype increases 19.72%, 12.34%, 15.01% and 22.28% using air, water, and parallel and series sources combinations, respectively. Moreover, heating capacity of optimized prototype increases 8.72%, 4.66%, 8.73% and 19.85% using air, water, sources in parallel and in series, respectively. The COPs and heating capacities with all heat sources combinations are obviously improved.

Fig. 4(a) and (b) illustrates the COP and heating capacity of the optimized multi-functional heat pump system and the initial multi-functional heat pump system in space heating plus hot

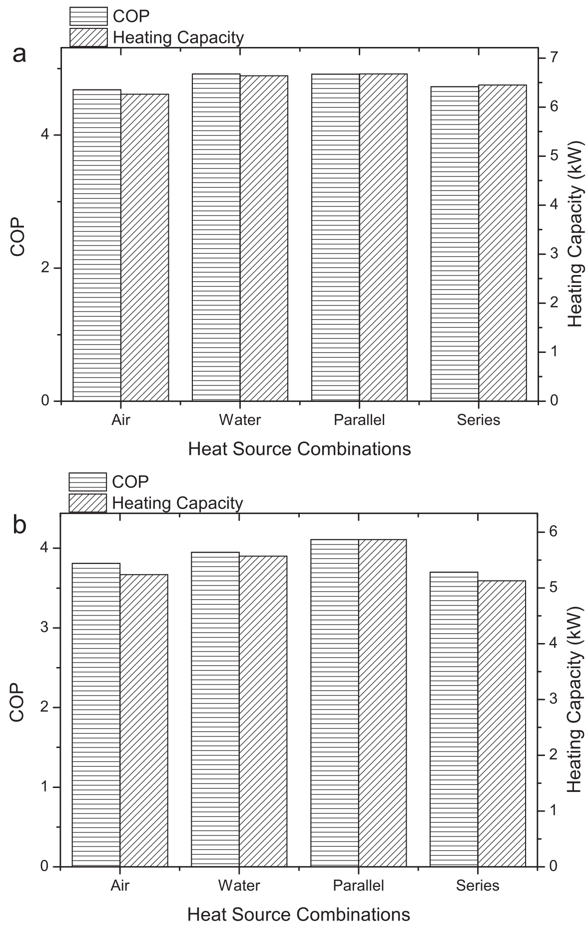

Fig. 4. (a) Optimized prototype in space heating plus hot water supply mode at outdoor air temperature 8.3 °C with various heat source combinations. (b) Initial prototype in space heating plus hot water supply mode at outdoor air temperature 8.3 °C with various heat source combinations

water supply mode. The parallel source combination in both prototypes has the highest COP and heating capacities. Comparing the system performance of the optimized prototype and the initial prototype, the COP of the optimized prototype increases 22.89%, 24.55%, 19.61% and 27.77% using air, water, sources in parallel and in series, respectively. In addition, heating capacity of optimized prototype increases 19.49%, 19.17%, 13.77% and 25.78% using air, water, sources in parallel and in series, respectively. The COPs and heating capacities with all heat sources combination have evidently been improved.

4. Analysis of experimental results

4.1. Space heating mode

4.1.1. Effects of outdoor air temperatures on the performance of the system

Table 2 presents that various operating conditions have been tested with the multi-functional heat pump system in space heating mode. The experiments of four heat source combinations are implemented at three different outdoor air temperatures. A total of twelve experiments (i.e. three conditions and four combinations) have been performed.

Fig. 5 shows the COP and heating capacity of the optimized system with the four heat source combinations at the outdoor air temperature of 15.6 °C. It is observed that the best performance

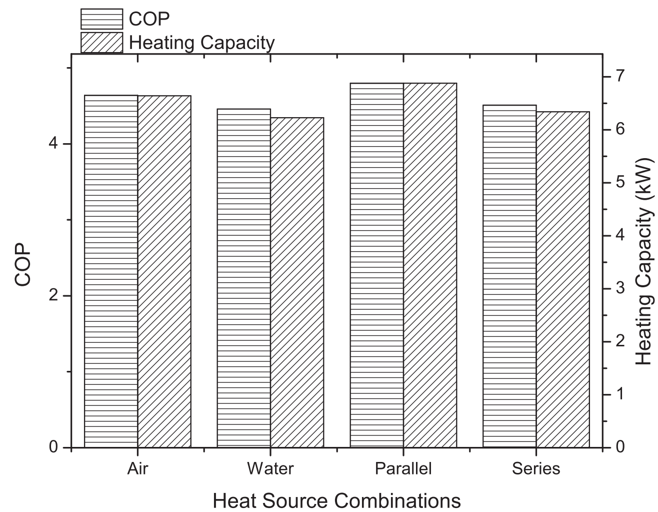

Fig. 5. Heating capacity and COP in space heating mode at outdoor air temperature 15.6 °C with various heat source combinations.

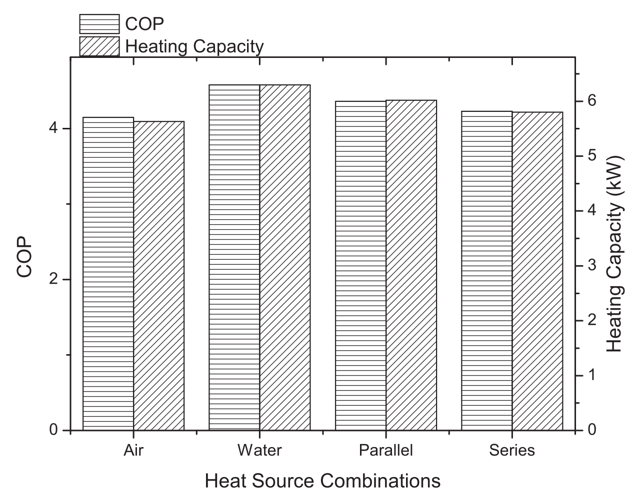

Fig. 6. Heating capacity and COP in space heating mode at outdoor air temperature 1.1 °C with various heat source combinations.

occurs when the heat sources are still parallel. When the heat source is water only or in series combination, the performance of the system becomes the worst. Fig. 6 illustrates the COP and heating capacity of the system at 1.1 °C. The COP and heating capacity with water source only becomes the highest, and those of the system with air source only is the lowest at outdoor air temperature 1.1 °C. In Fig. 3(a), parallel and series sources combinations almost have the same performance. Air source only and water source only have similar performance. Comparing Figs. 3(a) and 5, when the outdoor air temperature decreases from 15.6 °C to 8.3 °C, the COP of the system with air source only drops 7.11% and the COP of the system with water source only keeps the same. The heating capacity of the system with air source only drops 11.75% while that of the system with series source combination increase 0.95%. When the outdoor air temperature is further dropped to 1.1 °C, comparing Figs. 3(a) and 6, the COP of the system with series source combination drops 6% while that of the system with water source only increases 2.69%. The heating capacity of the system with series source combination drops 9.38%, while that of the system with water source only increases 3.96%. Outdoor air temperature change has limited effects on the system performance with water-source-only.

Table 2
Experiment conditions for space heating.

Heat source combinations				Outside Air temperature (°C)			Indoor air temperature (°C)	Gray water tank temperature (°C)
Air	Water	Parallel	Series	15.6	8.3	1.1	21.1	21.1

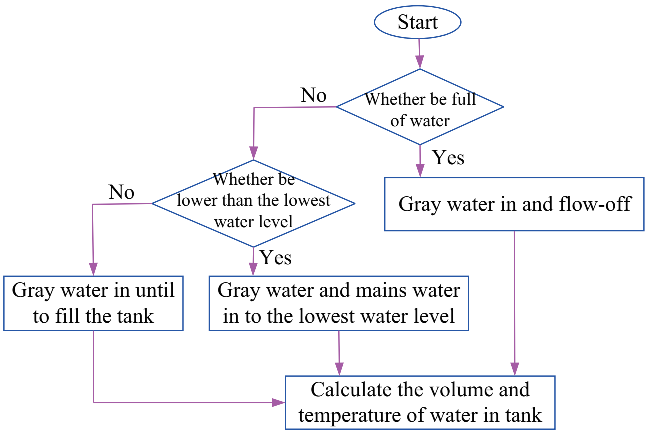

Fig. 7. The operation strategies of gray water tank.

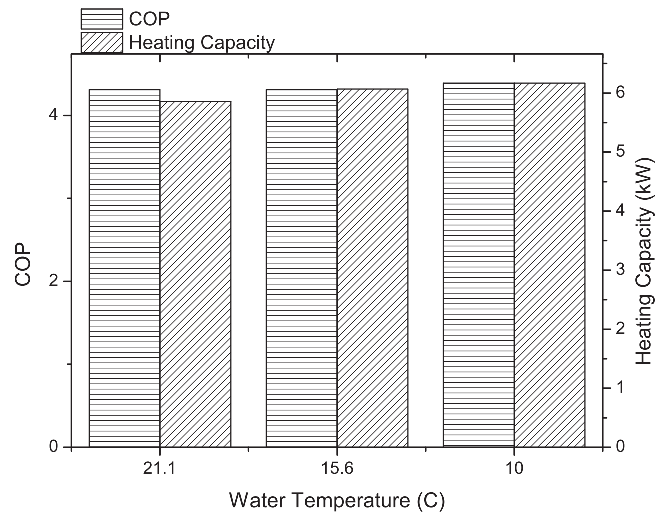

Fig. 8. Heating capacity and COP in space heating mode with air source only at various gray water temperature.

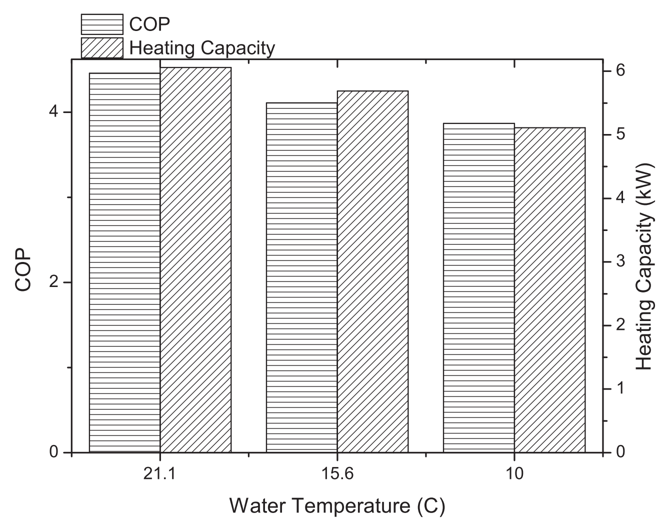

Fig. 9. Heating capacity and COP in space heating mode with water source only at various gray water temperature.

4.1.2. Effects of water temperatures on the performance of the system

In multi-functional heat pump system with air source and gray water source, the gray water temperature is critical in the system performance. Mains water plays a role in controlling and maintaining water volume in the gray water tank as shown in Fig. 7 of Ref. [36]. In some situations, the water temperature in the gray water tank may be below the mains water temperature. Hence the supplemental mains water temperature may raise the resultant temperature of the mixed water inside the tank. In this section, the effects of gray water temperature on the performance of the optimized prototype are discussed.

The mains water temperature in a day varies significantly depending on location and the time of year. It can be calculated by the equation developed by Hendron et al. [37]. In New York, based on the calculation, the daily mains-water temperature is in a range of 9–22.6 °C and the annual mean mains-water temperature is 15.8 °C. In general, mains-water temperature is less than the gray water temperature from indoor during most of the time in winter. Lower bound energy savings have been estimated based on an assumption of using the calculated mean mains-water temperature and the lowest mains-water temperature, respectively, at the gray water tank temperature. Heat pump system is tested at outdoor air temperature 8.3 °C in space heating mode as shown in Table 3.

Figs. 8–11 show the heating capacities and COPs of the optimized heat pump system with different heating source combinations at different gray water temperatures. In Fig. 8, the decrease of gray water temperature has some positive impact on the system performance with air source only. When the gray water temperature changes from 21.1 °C to 15.6 °C, COP of the system does not change, but heating capacity of the system increase about 3.58%. If gray water further drops from 15.6 °C to 10 °C, the COP increases 1.86% and heating capacity increases 1.65%. Gray water temperature change has the impacts on the system performance with air source only may be caused by many factors, such as the pressure distribution in air source and water source heat exchangers, which may affect the refrigerant amount flowing to the air source heat exchanger. Fig. 9 shows that the COP and heating capacity of the system with water source only reduce 7.85% and 6.11% when gray water temperature decreases from 21.1 °C to 15.6 °C. When the gray water further drops to 10 °C, the COP and heating capacity fall 5.84% and 10.19%, respectively. As the gray water temperature decreases, the COP and heating capacity dramatically decrease with water source only. When gray water temperature decreases from 21.1 °C to 15.6 °C, there is no change in the COP while heating capacity of the system slightly increase 1.57% with parallel combination increases (see Fig. 10). When gray water temperature further drops from 15.6 °C to 10 °C, the COP decreases 0.66% and the heating capacity increases 2.02%. It is learned from those results that gray water temperature has limited effect on the system performance with parallel sources combination. Fig. 11 shows that the COP and heating capacity of the system decrease greatly as the gray water temperature decreases with the heat source in series. When the gray water temperature changes from 21.1 °C to 15.6 °C, the COP of the system with series sources combination decreases 7.33% and the heating capacity of the system with series combination

Table 3
Experimental conditions of testing three gray water temperatures.

Outside air temperature (°C)	Indoor air temperature (°C)	Gray water tank temperature (°C)		
8.3	21.1	21.1	15.6	10

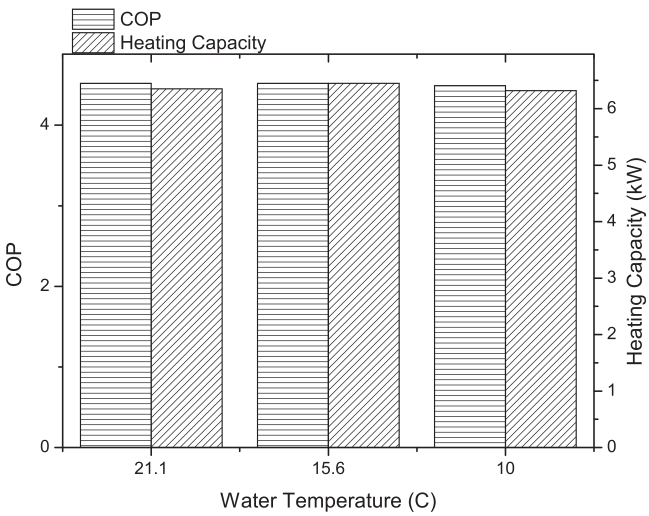

Fig. 10. Heating capacity and COP in space heating mode with heat sources in parallel at various gray water temperature.

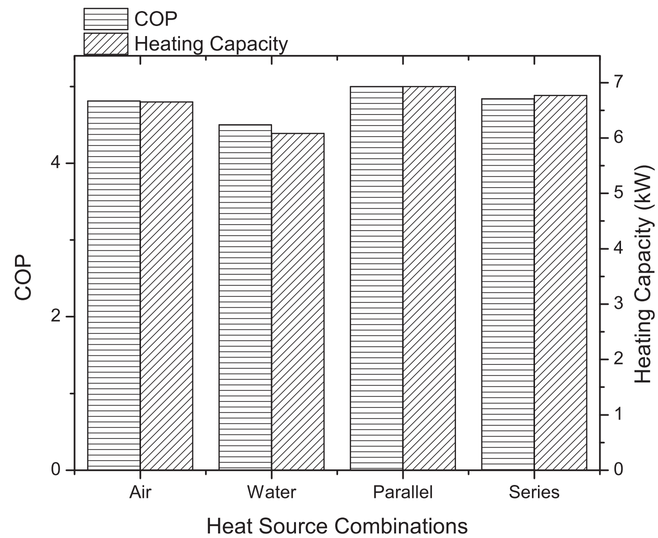

Fig. 12. Heating capacity and COP in space heating plus hot water supply mode at outdoor air temperature 15.6 °C with various heat source combinations.

reduces 9.69%. When the gray water temperature becomes 10 °C, the COP and heating capacity further decrease 3.36% and 5.71%, respectively. According to the Figs. 8–11, the change of gray water temperature has significant effects on water source only and series sources combination, but has very limited impact on air source only and parallel source combinations. This situation is the same as that of the initial prototype. Compared with the performance test data from the initial prototype, the optimized prototype has better performance. Even water temperature in the gray water tank becomes 10 °C, the performances of the optimized system with other three heating source combinations are still better than the manufacture data shown in the paper of Liu et al. [1].

Referring to Figs. 8–11, when the gray water is 21.1 °C, the system with parallel sources combination has the highest COP and heating capacity. The system with series sources combination has the similar performance as that with parallel sources combination does. As gray water drops to 15.6 °C, the system with parallel source combination still has the highest COP and heating capacity while the system with water source only has the lowest COP and heating capacity among those four heat source combinations. The system performance with series sources combination is only better than that with water source only. When the gray water temperature is at 10 °C, the system with parallel sources combination maintain the best performance among all the four heat source combinations. The system performance with water source only is the worst at this low gray water temperature. According to the system performance data shown in Liu et al. [1], the system performance of the optimized prototype in various gray water temperature is better than that of the initial prototype.

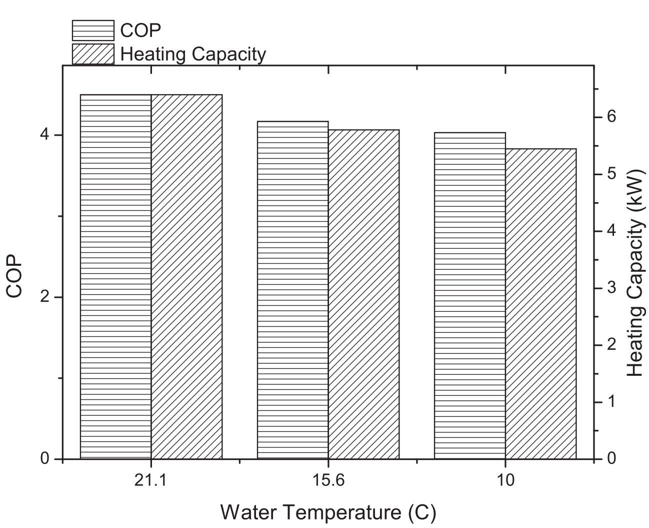

Fig. 11. Heating capacity and COP in space heating mode with heat sources in series at various gray water temperature.

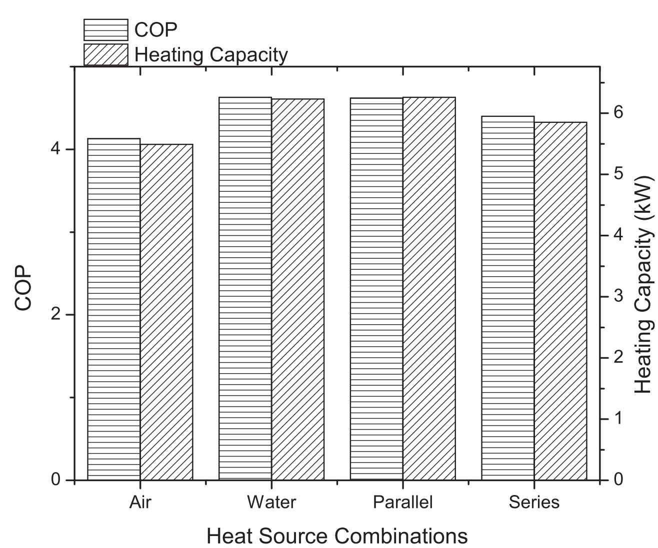

Fig. 13. Heating capacity and COP in space heating plus hot water supply mode at outdoor air temperature 1.1 °C with various heat source combinations.

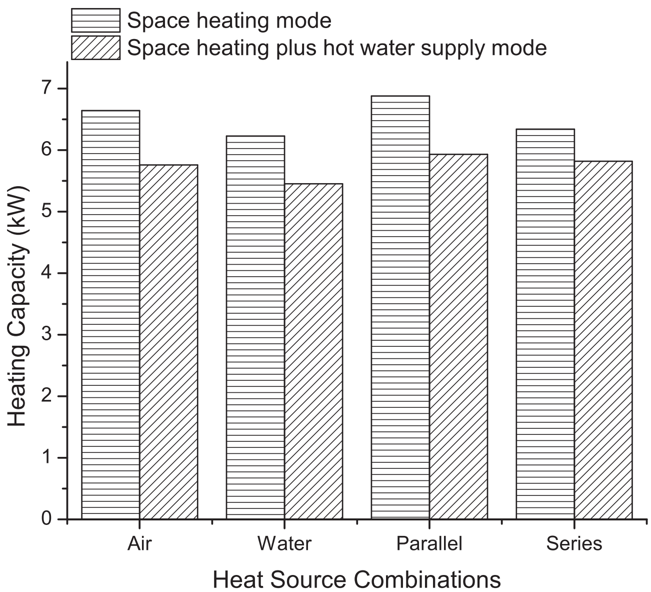

Fig. 14. Heating capacity for space heating only at outdoor air temperature 15.6 °C with various heat source combinations.

4.2. Space heating plus hot water supply mode

Table 4 presents various operating conditions of multi-functional heat pump system in space heating plus hot water supply mode. The experiments of four heat source combinations are implemented at three different outdoor air temperatures. A total of twelve experiments have been carried out.

Figs. 4(a), 12 and 13 show the COPs and heating capacities of the system in space heating plus hot water supply mode with four types of heat source combinations at different outdoor air temperatures. Referring to Fig. 12, the COP of the system with the parallel sources combination is the highest and that of the system with the water source only is the lowest when the outdoor air temperature is 15.6 °C among the four heat source combinations. When the outdoor air temperature is at 8.3 °C, the COP and heating capacity of the optimized system with the parallel sources combination are both the highest. At the same condition, the COP and heating capacity of the system with air source only are both the lowest as shown in Fig. 4(a). As the outdoor air temperature decreases to 1.1 °C as shown in Fig. 13, the COP of the system with the parallel sources combination and that of the system with water source only are almost the same and superior to the system with other heat source combinations. The COP and heat capacity with air source is the lowest at gray water temperature of 1.1 °C. The heating capacity with the parallel sources combination is the highest as shown in Fig. 13.

Compared of Figs. 4(a) and 12, when the outdoor air temperature drops from 15.6 °C to 8.3 °C, the COP of the system with air source only drops 2.66% while that of the system with water source only increases 9.32%. The total heating capacity of the system with air source only drops 5.85% while that of the system with water source only increases 9.18%. When outdoor air temperature becomes 1.1 °C, compared with Figs. 4(a) and 13, the COP of the system with air source only still has the largest decrease of 11.79% and that of the system with water source only has the least which is 5.89%. The heating capacity of the system with air source only drops 12.31% and that of the system with water source only reduces 6.15%. Outdoor air temperature change has limited impacts on the system performance with water source only while has the largest influences on that of the system with air source only based on above-mentioned results with the decrease of outdoor air temperature.

4.2.1. The comparison of heating capacity in space heating

In space heating plus hot water supply mode, the heat exchanger for heating hot water and the heat exchanger in the indoor air-handler work as condensers. Parts of the total heating capacity of the system are used for heating hot water via reclaiming the superheat part of the discharged refrigerant from the compressor.

Figs. 14–16 show the comparison of the space heating capacity in space heating mode and space heating plus hot water mode. In Fig. 14, the heating capacities for space heating only in space heating plus hot water supply mode are reduced up to 13.25% for air source only, compared with the heating capacity in space heating mode. As shown in Fig. 15, when the outdoor air temperature is 8.3 °C, the heating capacity for space heating only in space heating plus hot water supply mode drops up to 9.53% for series sources combination, compared with those in space heating

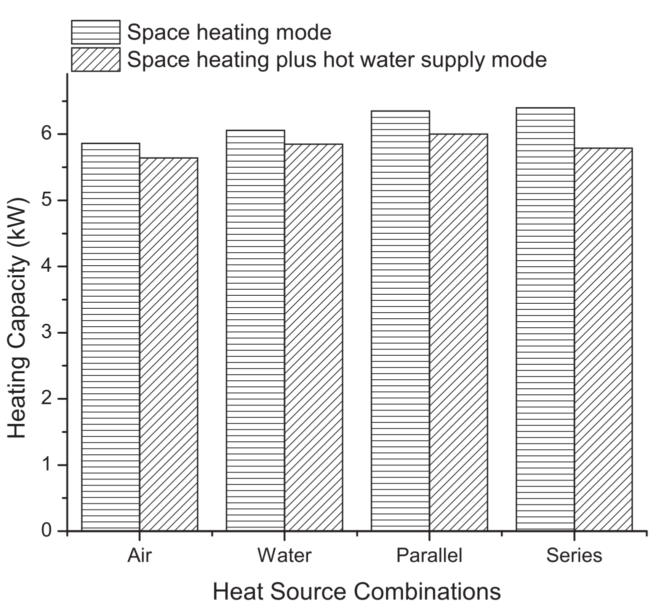

Fig. 15. Heating capacity for space heating only at outdoor air temperature 8.3 °C with various heat source combinations.

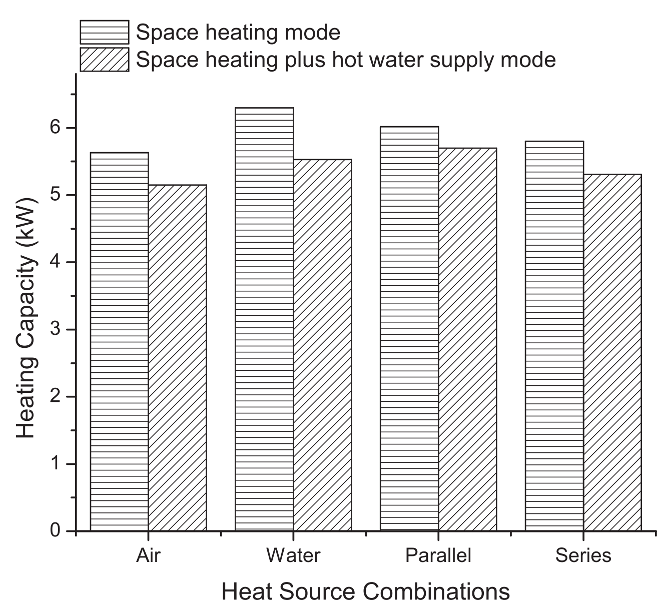

Fig. 16. Heating capacity for space heating only at outdoor air temperature 1.1 °C with various heat source combinations.

Table 4
Space heating plus hot water supply mode experiment conditions.

Heater exchanger choice				Outside air temperature (°C)/average relative humidity of outdoor air (%)			Indoor air temperature (°C)	Gray water tank temperature (°C)	Hotwater tank temperature (°C)
Air	Water	Parallel	Series	15.6	8.3	1.1	21.1	21.1	48.9

mode. In Fig. 16, the heating capacity for space heating only in space heating plus hot water supply mode at outdoor air temperature 1.1 °C drops up to 12.22% for air source only while that with parallel source combination has the least drop range, compared with those in space heating mode. The system still has larger heating capacity for space heating, compared with the sample data provided by the manufacturer against selecting the proper heat source combination.

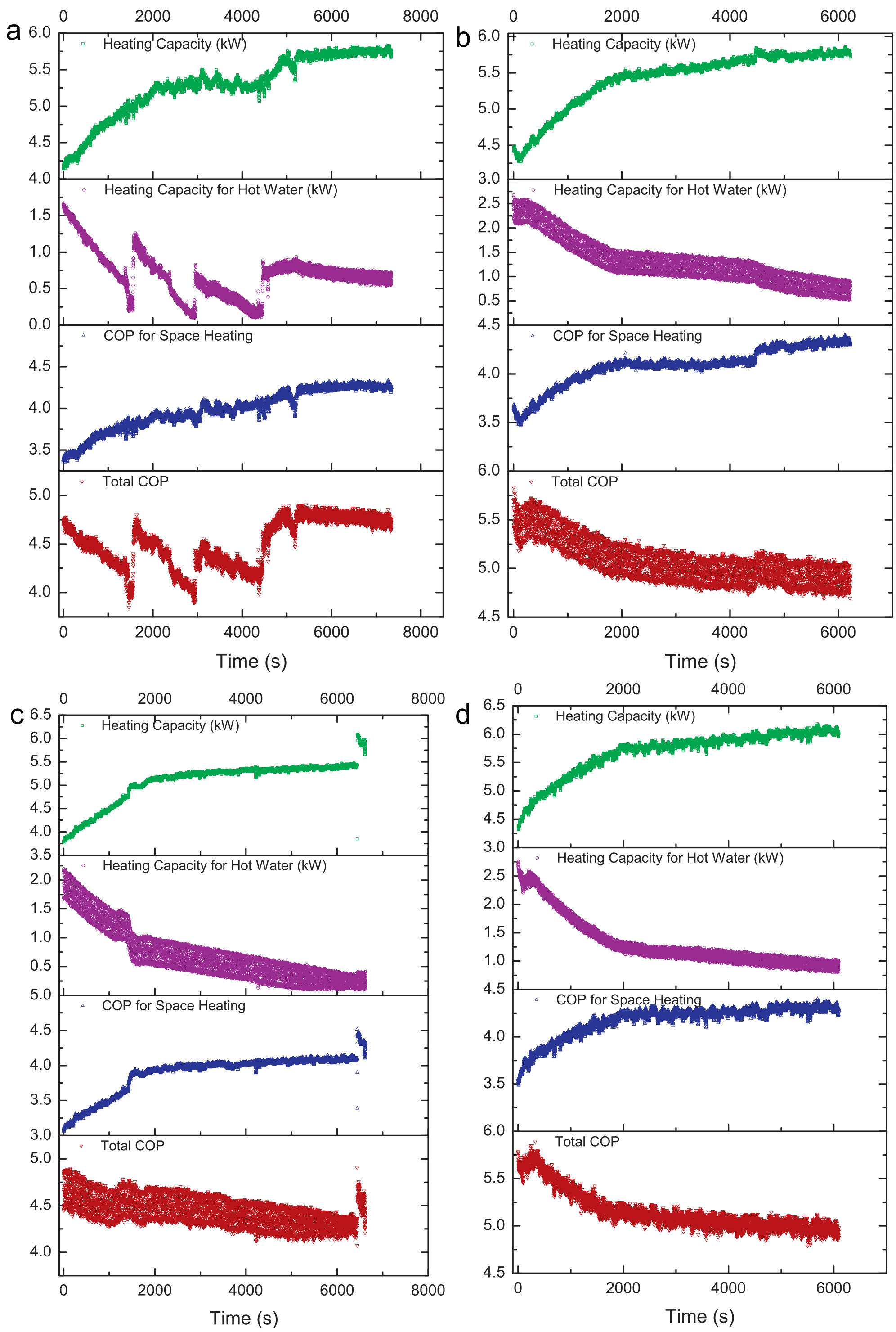

Fig. 17. Dynamic heating capacity and COP at outdoor air temperature 8.3 °C with various source combinations: (a) air, (b) water, (c) parallel, and (d) series

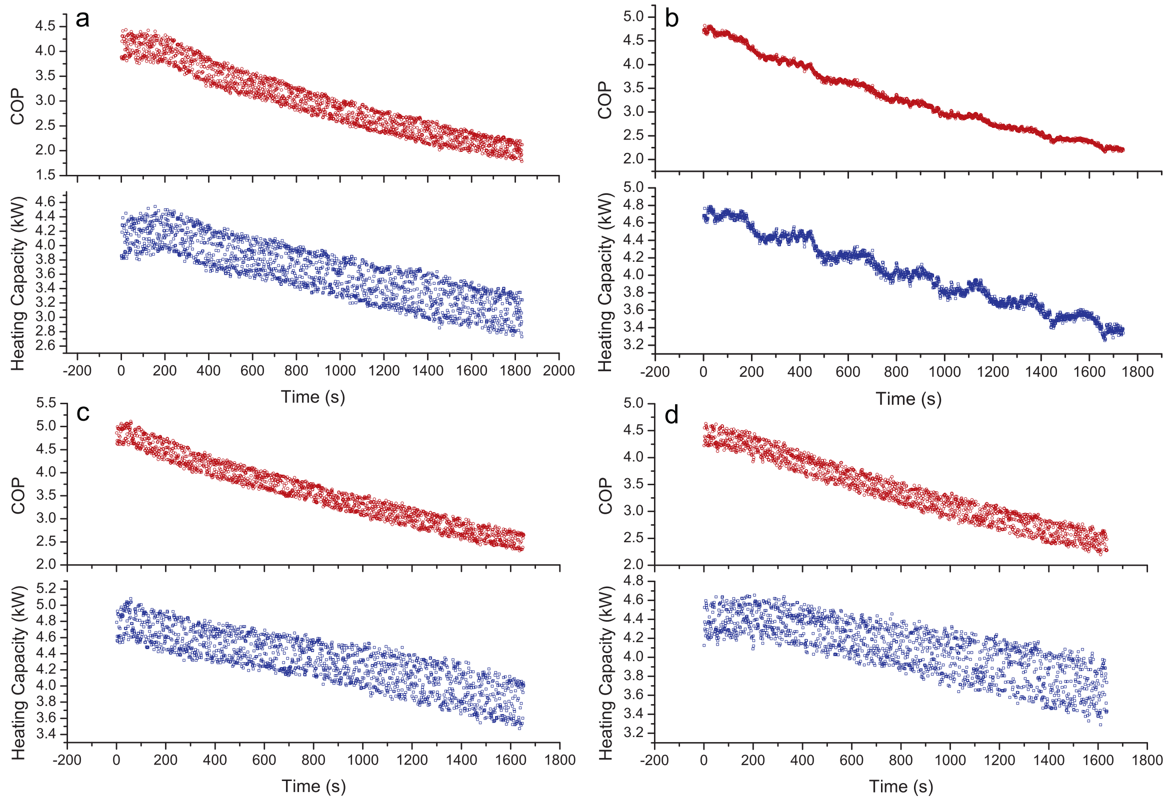

Fig. 18. COP and heating capacity at outdoor air temperature 8.3 °C with various heat source combinations: (a) air, (b) water, (c) parallel, and (d) series

4.2.2. Dynamic COP and heating capacity in space heating plus hot water supply mode

To investigate the impacts of heating water from a low temperature to a high temperature on the heat pump system performance, several experiments are carried out in space heating plus hot water supply mode. Table 5 shows the operating conditions in this system dynamics test. This section will discuss the heating capacity and time usage with different heat source combinations to achieve satisfactory hot water temperature and volume.

Fig. 17 illustrates heating capacity for space heating only and heating capacity for supplying hot water, COP for space heating only, and total COP (from top to bottom plots) at air source only, water source only, sources in parallel and in series, respectively. In the bottom plot of Fig. 17(a), with air source only, the total COP fluctuates between around 4 and 4.8 as the hot water temperature increases. The COP for space heating clearly increases from around 3.5 to 4.5 as shown in the second bottom of Fig. 17(a). The heating capacity for space heating, shown in the top of Fig. 15, increases from around 4.3 kW to 6 kW, while that for supplying hot water decreases from around 1.7 kW to 0.5 kW as the hot water temperature increases. The time span is about 2 h. In Fig. 17(b), when the heat source is the water source only, the total COP decreases from around 5.7 to 4.7 as the hot water temperature increases. The COP for space heating increases from 3.5 to 4.4. The heating capacity for space heating with water source only increases from around 4.5 kW to below 6 kW. The heating capacity for supplying hot water decreases from around 2.7 kW to 0.65 kW over the time span. The time span is about one hour forty-three minutes. Fig. 17(c) shows the total COP change over the time when the air source and the water source are parallel, the COP for space heating increases from around 3 to 4. The decrease rate of the total COP is the slowest among all types of heat source combinations. The heating capacity for space heating increases from around 4 kW to 6 kW. The heating capacity for supplying hot water decreases from around 2 kW to around 0.3 kW. The time span is about one hour fifty minutes. Fig. 17(d) presents the system performance with series source combination. The total COP decreases from around 5.8 to 4.8. The COP for space heating increases from around 3.5 to 4.3. The heating capacity for space heating increases from around 4.3 kW to 6 kW. The heating capacity for supplying hot water decreases from around 2.5 kW to 0.6 kW. This type of heat source combination has the shortest time span, which is about one hour and forty minutes. Comparing Fig. 17(a)–(d), the dynamic change of hot water temperature has the larger impact on the system with water source only and series combinations than that on the system with air source only and parallel combination. The total heating capacity is almost constant in heating hot water from 30 °C to 48.9 °C. Comparing Fig. 17(a)–(d), water source only and series sources combination can take shorter time than others in heating hot water to satisfactory temperature.

4.3. Hot water supply only mode

When the hot water demand is high, the system operation mode will be switched to the hot water supply only mode. This mode will only supply hot water. The purpose of the hot water supply only mode is to provide satisfactory demand for hot water in a short period of time. So, in this section, the ability of heating hot water from low temperature to high temperature will be investigated. Sixteen experiments are made as shown in Table 6 (i.e. four heat sources combination and four outdoor air temperatures). Only the experimental results at outdoor air temperature 8.3 °C will be discussed in this section

Fig. 18 shows the COP and heating capacity for supplying hot water at air source only, water source only, sources in parallel and in series respectively. The COP and heating capacity for supplying hot water decrease as the hot water temperature increases. The system with water source only, sources in parallel and sources in

Table 5
Dynamic COP experiment conditions.

Outside air temperature (°C)	Indoor air temperature (°C)	Gray water tank temperature (°C)	Hot water tank temperature (°C)
8.3	21.1	21.1	30–48.9

Table 6
Dynamic COP and heating capacity experiment conditions.

Heat source combinations				Outdoor air temperature (°C)				Gray water tank temperature (°C)	Outdoor air temperature (°C)
Air	Water	Parallel	Series	35	15.6	8.3	3.4	21.1	30–48.9

Table A.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: 0–100% RH @ 0–50 °C Temperature: 0–100 °C	4–20 mA	24 Vdc	±2.5% RH ±0.3 °C
	0–1000 psi	4–20 mA	24 Vdc	±0.25%
Gauge pressure transducer	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%

series have similar performance while air source only has the worst performance. Considering the COP and heating capacity, the parallel sources combination has a small advantage over the others. The parallel sources combination also uses the shorter time in heating hot water from 30 °C to 48.9 °C than other heat source combinations do. The COP and total heating capacity of hot water supply only mode are obviously lower than those of space heating plus hot water mode. However, the time for heating hot water to satisfactory temperature of hot water supply only mode is quicker than that of space heating plus hot water supply mode. The COP of hot water supply only mode is much larger than that of conventional electrical water heater, which the COP of the conventional electrical water heater is normally assumed to be 1.

5. Conclusion

In this study, the drawbacks of the initial prototype are analyzed. Remedial measures are implemented in optimized prototype. Installing a refrigerant receiver can provide ability in balancing the demand of refrigerant amount in heating mode and cooling mode and in controlling the demand of refrigerant amount in different functions with various heat sources combinations. Two plate heat exchangers in parallel combination for supplying hot water shows better performance compared with conventional electrical water heater. A ball valve installed at the outlet of air-source evaporator can minimize the effect of outdoor air temperature on the system performance with water source only. According to the experimental tests, the system with air and water heat sources in parallel has the best performance in space heating mode and space heating plus hot water supply mode, compared with the system with other heat source combinations. The optimized multi-functional heat pump system shows superior performance in the heating mode compared with the initial multi-functional heat pump system. The performance of the multi-functional heat pump system with air source only, sources in parallel and in series combinations decrease as the outdoor air temperature decreases in space heating mode and space heating plus hot water supply mode. The outdoor air temperature has limited impacts on the system performance change of water source only via installing a ball valve at the outlet of the air source evaporator. The heating capacity for space heating of the multi-functional heat pump system in space heating plus hot water supply mode is smaller than that in space heating mode. However, the COP of the system in space heating plus hot water supply mode is higher than that in space heating mode. In space heating plus hot water supply mode, the total COP of the multi-functional heat pump system decreases as the hot water temperature increases. The heating capacity for heating hot water decreases, and the heating capacity for space heating increases as the hot water temperature increases. The process of heating hot water from 30 °C to 48.9 °C usually takes less than 2 h. The heating time with water source only, sources in parallel or in series are almost the same. The system with air source only takes the longest heating time. With the gray water temperature change, the system performance with air source only has the least variation while that with series source combination and water source only have appreciable variation which decline with gray water temperature decrease. In hot water supply only, the process of heating hot water from 30 °C to 48.9 °C takes the least time, less than half hour. The system with parallel source combination has the best performance.

Acknowledgments

The financial supports of the Nebraska Energy Sciences Research and the U.S. Environmental Protection Agency (CR-83419301) are gratefully acknowledged.

Appendix. Uncertainty analysis

Table A.1 shows the range and precision of the measurement equipment. The uncertainty of the experimental test can be divided into type A and type B standard uncertainty [38]. Take the heating capacity with air source only at outdoor temperature 8.3 °C as an example: the mean value of heating capacity is 5856.95 W. The type A and type B standard uncertainties are 223.4327 W and 168.0546 W, respectively. The combined uncertainty of heating capacity is 279.5792 W.

References

[1] X. Liu, L. Ni, S.-K. Lau, H. Li, Performance analysis of a multi-functional heat pump system in heating mode, Applied Thermal Engineering 51 (1–2) (2013) 698–710.

[2] X. Liu, L. Ni, S.-K. Lau, H. Li, Performance analysis of a multi-functional heat pump system in cooling mode, Applied Thermal Engineering 59 (1–2) (2013) 253–266.
[3] J.A. Pietsch, The unitary heat pump industry – 25 years of progress, ASHRAE Journal 19 (1977) 15–18.
[4] O. Kara, K. Ulgen, A. Hepbasli, Exergetic assessment of direct-expansion solar-assisted heat pump systems: review and modeling, Renewable and Sustainable Energy Reviews 12 (5) (2008) 1383–1401.
[5] Y.H. Kuang, R.Z. Wang, Performance of a multi-functional direct-expansion solar assisted heat pump system, Solar Energy 80 (7) (2006) 795–803.
[6] DOE/EERE, SunShot Vision Study: Solar Power Environmental Impacts and Siting Challenges http://www1.eere.energy.gov/solar/sunshot/vision_study.html
[7] O. Ozgener, A. Hepbasli, A parametrical study on the energetic and exergetic assessment of a solar-assisted vertical ground-source heat pump system used for heating a greenhouse, Building and Environment 42 (1) (2007) 11–24.
[8] O. Ozgener, A. Hepbasli, Modeling and performance evaluation of ground source (geothermal) heat pump systems, Energy and Buildings 39 (1) (2007) 66–75.
[9] O. Ozgener, A. Hepbasli, Performance analysis of a solar-assisted ground-source heat pump system for greenhouse heating: an experimental study, Building and Environment 40 (8) (2005) 1040–1050.
[10] S. Ito, N. Miura, Studies of a heat pump using water and air heat sources in parallel, Heat Transfer – Asian Research 29 (6) (2000) 473–490.
[11] X. Liu, C. Bullard, Multiple Heat Exchangers Simulation Within the Newton–Raphson Framework, 2002.
[12] C. Chaichana, L. Aye, W.W.S. Charters, Natural working fluids for solar-boosted heat pumps, International Journal of Refrigeration 26 (6) (2003) 637–643.
[13] S.K. Chaturvedi, D.T. Chen, A. Kheireddine, Thermal performance of a variable capacity direct expansion solar-assisted heat pump, Energy Conversion and Management 39 (3–4) (1998) 181–191.
[14] Y.W. Li, R.Z. Wang, J.Y. Wu, Y.X. Xu, Experimental performance analysis on a direct-expansion solar-assisted heat pump water heater, Applied Thermal Engineering 27 (17–18) (2007) 2858–2868.
[15] H. Arif, Exergetic modeling and assessment of solar assisted domestic hot water tank integrated ground-source heat pump systems for residences, Energy and Buildings 39 (12) (2007) 1211–1217.
[16] B.J. Huang, J.P. Lee, J.P. Chyng, Heat-pipe enhanced solar-assisted heat pump water heater, Solar Energy 78 (3) (2005) 375–381.
[17] Y.H. Kuang, K. Sumathy, R.Z. Wang, Study on a direct-expansion solar-assisted heat pump water heating system, International Journal of Energy Research 27 (5) (2003) 531–548.
[18] M.N.A. Hawlader, S.K. Chou, K.A. Jahangeer, S.M.A. Rahman, E. Lau, K.W., Solar-assisted heat-pump dryer and water heater. Energex 2002 – new and renewable sources of energy – topic I, Applied Energy 74 (1–2) (2003) 185–193.
[19] J. Ji, T.-T. Chow, G. Pei, J. Dong, W. He, Domestic air-conditioner and integrated water heater for subtropical climate, Applied Thermal Engineering 23 (5) (2003) 581–592.
[20] W. Ying, Performance of room air conditioner user for cooling and hot water heating, ASHRAE Transactions 95 (1989) 441–444.
[21] R. Cook, Water storage tank size requirement for residential heat pump/air-conditioner desuperheater recovery, ASHRAE Transactions 96 (2) (1990) 715–719.
[22] V. Goldschmidt, Desuperheater water-heater and air-to-air heat pump system: representative performance data, ASHRAE Transactions 96 (2) (1990) 417–421.
[23] K. Toh, S. Chan, Thermosiphon heat recovery from an air-conditioner for a domestic hot water system, in: The 1993 Winter Meeting of ASHRAE Transactions. Part 1, Chicago, IL, USA, 01/23–27/93, 1993, pp. 259–264.
[24] H. Jiang, Y. Jiang, Y. Wang, Z. Ma, Y. Yao, An experimental study on a modified air conditioner with a domestic hot water supply (ACDHWS), Energy 31 (12) (2006) 1789–1803.
[25] A.H.W. Lee, J.W. Jones, Thermal performance of a residential desuperheater/water heater system, Energy Conversion and Management 37 (4) (1996) 389–397.
[26] P. Techarungpaisan, S. Theerakulpisut, S. Priprem, Modeling of a split type air conditioner with integrated water heater, Energy Conversion and Management 48 (4) (2007) 1222–1237.
[27] R. Murphy, K. Rice, V. Baxter, W. Craddick, Air Source Integrated Heat Pump for Near Zero Energy Houses: Technology Status Report. ORNL/TM-2007/112, 2007.
[28] F.M. Rad, A.S. Fung, W.H. Leong, Combined solar thermal and ground source heat pump system, in: Proceedings of Building Simulation, 2009, pp. 2297–2305.
[29] J. Ji, G. Pei, T.-T. Chow, W. He, A. Zhang, J. Dong, H. Yi, Performance of multi-functional domestic heat-pump system, Applied Energy 80 (3) (2005) 307–326.
[30] L. Hong, Simulation and Experimental Investigation on Optimum Application of Multi-Functional Solar Assisted Air Source Heat Pump Systems, 2010.
[31] C. Coskun, Z. Oktay, I. Dincer, Performance assessment of a novel hybrid district energy system, Applied Thermal Engineering 48 (2012) 268–274.
[32] M. Hu, Application of Shredded-Tire Biofilter and Membrane Bioreactor for Greywater Reclamation, 2011.
[33] E. Kozubal, J. Woods, J. Burch, A. Boranian, T. Merrigan, Desiccant Enhanced Evaporative Air-Conditioning (DEVap): Evaluation of a New Concept in Ultra Efficient Air Conditioning, 2011, pp. 275–3000 (contract 303).
[34] AHRI, Performance Rating of Unitary Air-Conditioning and Air-Source Heat Pump Equipment (ANSI/AHRI Standard 320), 1998.
[35] AHRI, Performance Rating of Unitary Air-Conditioning and Air-Source Heat Pump Equipment (ANSI/AHRI Standard 210/240), 2008.
[36] L. Ni, S.K. Lau, H. Li, T. Zhang, J.S. Stansbury, J. Shi, J. Neal, Feasibility study of a localized residential grey water energy-recovery system, Applied Thermal Engineering 39 (2012) 53–62.
[37] R. Hendron, R. Anderson, C. Christensen, M. Eastment, P. Reeves, Development of an energy savings benchmark for all residential end-uses, in: Proceedings of SimBuild 2004, IBPSA-USA National Conference, Boulder, CO, USA, 2004.
[38] B.N. Taylor, Guidelines for Evaluating and Expressing the Uncertainty of NIST Measurement Results (rev, DIANE Publishing, 2009.
[39] Doe/Eere, 2010 Buildings Energy Databook, 2011.