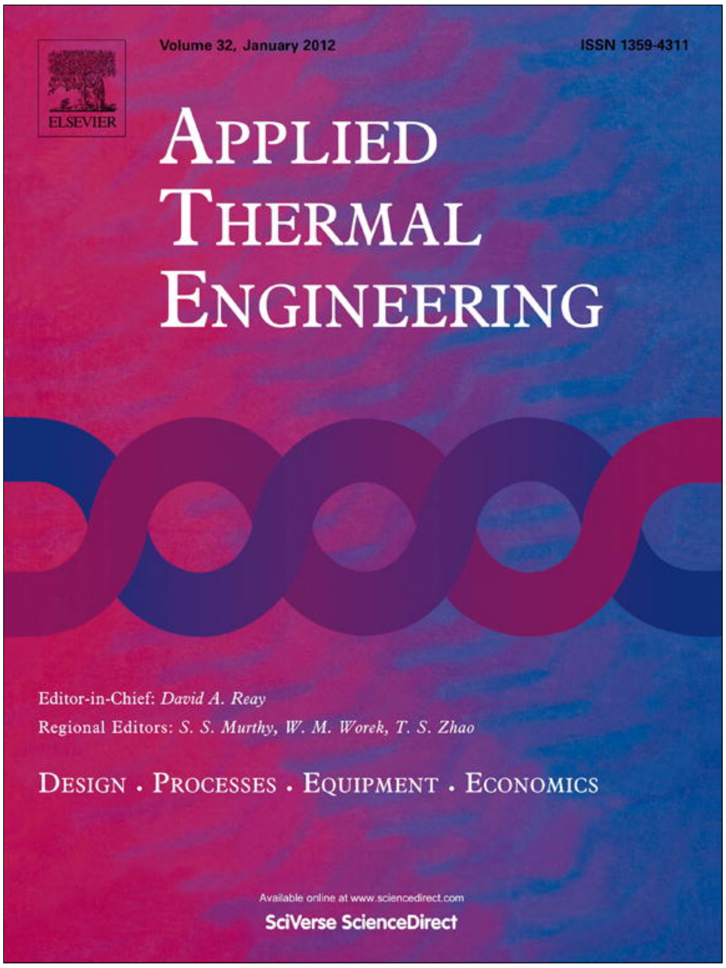

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Feasibility study of a localized residential grey water energy-recovery system

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ABSTRACT

In order to improve the overall efficiency of building energy usage, a grey water energy-recovery system, which adopted a multiple-function heat pump system, was proposed for domestic water heating, and space heating and cooling of buildings. A numerical model has been developed for investigation of annual energy and water consumptions of the proposed system and the conventional building energy system with gas furnace space heating, package air-conditioning, and electricity water heater for hot water heating. Based on a case study of a typical residential house with four family members and three bedrooms in New York, NY, results show that the overall source energy consumption with the proposed system decreases 33.9% for space heating and cooling and hot water heating, and also the potable water consumption reduces 27.2% compared with those of the conventional system. An extended study with fourteen other cities was performed in various climate zones in the U.S. Among these 15 cities, the savings in source energy and potable water consumption have ranges of 17%–57.9% and 15%–34.1%, respectively. The results also show the proposed system can provide substantial energy and potable water savings, particularly with moderate outdoor temperature.

1. Introduction

In the U.S., the residential building sector represented 20% of primary energy and 20% of carbon dioxide emissions in 2006 [1]. The major components dominated the energy consumption in residential buildings are space heating, space cooling, and water heating systems. Statistical results show that the space heating and cooling, and water heating systems consume 51.9% of energy, cost 55.7% of expenditures, and emit 50.4% of carbon dioxide in residential building [1].

Most existing households in the U.S. are designed to use a boiler, a furnace, and an air-conditioner individually and independently to serve the water heating, space heating, and space cooling. The utilization of a heat pump for the space heating and cooling only accounts for 8.3% of total number of households [2]. In addition, use of electric and natural gas boilers for hot water heating dominates the market of residential buildings (38.8% and 52.8%, respectively [2]). Although a great deal of effort has been done to enhance the efficiencies of individual systems and significant progresses have been achieved in the past decade [3–6], it is still challenging to achieve the aggressive, multi-year goal of 40–70% whole-house energy savings by 2020 set by the U.S. Department of Energy's (DOE's) Buildings Technology Program [7].

Residential buildings not only consume energy, but also potable water. A typical residential building with a four-member family consumes about 1300 L/d (i.e. liters/day) of fresh water and generates 800 L/d of wastewater [8]. A huge amount of energy existing in wastewater is discharged to the environment without being properly reclaimed. Some tentative attempts have proposed to utilize the heat of residential wastewater, such as shower-water heat recovery using a simple single-pass counter-flow heat exchanger in high-rise residential buildings [9] and dishwasher waste-heat reuse through a spiral heat exchanger [10]. Some researchers have focused on the energy reclamation from the municipal wastewater [11–13] and wastewater of commercial buildings [14,15]. However, from the best knowledge of the authors, there is no previous research on the recovery of heat from residential wastewater.

Condenser heat recovery is an energy saving technology that reclaims condenser heat from air-conditioners for other heat demands in residential buildings. There are numerous theoretical and experimental studies on this technology found in existing publications. Most of those studies utilized condenser heat for generating domestic hot water from a split type air-cooled, water-

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cooled air-conditioner or air source heat pump with a heat exchanger [16–19]. Moreover, the condenser heat can also be applied clothes drying [20]. However, the potential energy and cost savings for a localized waste-heat recovery system in residential building has not been addressed [21]. This study investigates the potential saving of incorporating condenser heat recovery using heat pump to reclaim heat from residential wastewater for space heating, space cooling, and domestic water heating.

2. System description

The proposed localized wastewater heat recovery system would reclaim heat energy from drainage and serve as a low-temperature heat source for space heating and hot water heating system. Fig. 1 shows the conceptual diagram of the integrated system with an outdoor air heat exchanger coil as the supplementary heat source and sink. Wastewater of a household can be divided into grey water and black water. The grey water sources include sinks, laundry, and bath/shower, while the black water sources include kitchen water and toilet [22]. Grey water is considered for heat recovery in the present study due to its simplicity in water treatment [23]. The treated grey water can be reused for irrigation and other purposes. When the volume of grey water is not sufficient for irrigation, potable water will be drawn into the tank as a supplement. An overflow pipe is used to drain the excess water in the wastewater tank. For the heat pump, if energy of the wastewater in the wastewater tank cannot provide sufficient heat in heating mode, the solenoid valves V4 and V5 in Fig. 1 will be opened and the deficient heat will be extracted from outdoor air by an outdoor coil. In cooling mode, the condenser heat is mainly discharged by the outdoor air heat exchanger except during periods of high outdoor temperature. At high outdoor temperature, if there is a demand for irrigation, potable water can act as a heat sink before irrigation. Utilization of potable water as heat sink at high outdoor temperature can reduce the peak electricity demand by improving the efficiency of the heat pump.

3. Model development

3.1. Design of prototype

The prototype building to be tested is a typical two-storey house (with a basement) for a 4-member family. It has three bedrooms and an approximate size of 220 m^2 excluding the floor area of the basement. The total gross wall area is 380 m^2, and the window-wall ratio is 12.65%. In order to evaluate the energy and economy performance of the proposed wastewater energy-recovery system in various climate conditions, 15 cities are selected based on ASHRAE Standard 90.1-2007 [24] as shown in Table 1. The space heating and cooling loads of the prototype were estimated by EnergyPlus 4.0. Typical weather condition of TMY3 is adopted from the website of EnergyPlus [25]. The indoor air design conditions are to maintain the indoor air temperature to 24.0 °C in any time and permit the relative humidity of air changeable. The statistical results (maximum and minimum) of space heating and cooling loads are also shown in Table 1.

3.2. Water flow rate and temperature models

3.2.1. Domestic hot water

The major end-uses for domestic hot water include shower/bath, sinks, dishwasher, and clothes washer. The average daily water consumptions by different end-use are shown in Table 2, and their typical hourly hot water use-profiles are shown in Fig 2 [26]. For the selected prototype, that is the number of bedrooms $N_{br} = 3$, the total hot water consumption is about 300 L/d based on the given equations in Table 2.

There are two different hot water supply temperatures shown in Table 2; however, the hot water heating facilities usually supply water at a specified temperature and then mix it with potable cold water in the inlets of end-use apparatuses. The hourly supplied hot water flow rate can be estimated by:

$$L_{hot} = \sum L_{end-use,i} \times P_{end-use,i} \times \frac{T_{end-use,i} - T_{mains}}{T_{hot} - T_{mains}} \quad (1)$$

where L_{hot} denotes the hot water supply flow rate (liters/hour) at a temperature of 49 °C. $L_{end-use,i}$ is the water consumption (L/d) of ith end-uses. $P_{end-use,i}$ is the percentage of hourly total hot water supply flow rate at ith end-uses, while $T_{end-use,i}$ is the corresponding end-use hot water temperature (°C). T_{hot} is the hot water supply temperature (i.e. 49 °C in the present study) from water heating facilities. T_{mains} is the cold water supply temperature (°C), which has geographical and temporal variations, and can be found by [7],

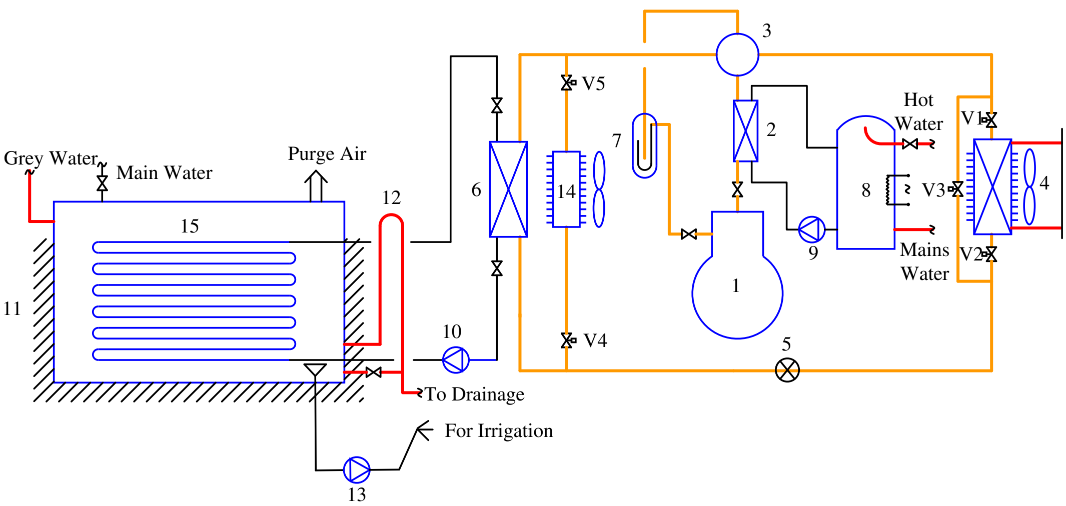

Fig. 1. Principal design of the localized energy-recovery system from residential wastewater assisted by outdoor air heat exchanger. Labels: 1 = compressor; 2 = plate heat exchanger for service water; 3 = four-way reversing valve; 4 = indoor coil; 5 = bi-flow expansion valve; 6 = wastewater heat exchanger; 7 = suction accumulator; 8 = hot water storage tank; 9, 10 = circulation pumps; 11 = wastewater tank; 12 = overflow pipe; 13 = pump for irrigation; 14 = outdoor coil; 15 = submerged-pipe coil; V1-V9 = solenoid valves.

Table 1
The selected cities in different climate zones.

NO.	Zone Name	Typical Cites	Max Heat load (W)	Max cooling load (W)	SECFE
1	Very Hot–Humid	Miami, FL	–	8966	3.057
2	Hot–Humid	Houston, TX	5237	9049	3.154
3	Hot–Dry	Tucson, AZ	4072	11123	3.061
4	Warm–Humid	Atlanta, GA	5789	9434	3.201
5	Warm–Dry	Lancaster, CA	4282	10520	2.446
6	Warm–Marine	San Francisco, CA	2809	6854	2.446
7	Mixed–Humid	New York, NY	7675	8686	2.903
8	Mixed–Dry	Las Vegas, NV	9069	8503	2.927
9	Mixed–Marine	Oklahoma, OK	7335	10027	3.030
10	Cool–Humid	Omaha, NE	9440	9301	3.613
11	Cool–Dry	Denver, CO	7929	9943	3.317
12	Cool–Marine	Bellingham, WA	5667	7947	1.807
13	Cold–Humid	Minneapolis, MN	12704	9222	3.529
14	Cold–Dry	Helena, MT	9341	9293	2.908
15	Very Cold	Jamestown, ND	22511	9656	3.681

$$T_{\text{mains}} = \left(T_{\text{amb,avg}} + \frac{10}{3}\right) + \text{ratio} \times \frac{\Delta T_{\text{amb,max}}}{2} \times \sin\left[\frac{360}{365}(\text{day} - 15 - \text{lag})\right] - 90 \quad (2)$$

where $T_{\text{amb,avg}}$ and $\Delta T_{\text{amb,max}}$ represent the annual average ambient air temperature (°C) and the maximum difference between monthly and average ambient temperatures (°C), respectively. Also, day = Julian day of the year, 1–365 day,

$$\text{ratio} = 0.4 + 0.018\left(T_{\text{amb,avg}} - 6.667\right)$$

$$\text{lag} = 35 - 1.8\left(T_{\text{amb,avg}} - 6.667\right)$$

The hourly heating load of hot water can be calculated based on the flow rate of hot water and the temperature difference between hot water and cold water supply.

3.2.2. Grey water and outdoor water use

The actual flow rate of wastewater depends on the residential water end-use. If the hourly flow rate and temperature of each water end-use is obtained, the projected flow rate and temperature of wastewater can be estimated.

There are several models and studies about residential water consumption [27–29]. However, these models, based only on monthly billing data, cannot provide the water consumptions of individual end-uses. In order to obtain the water consumption of different water end-uses, a statistical prediction models of residential end-use water [30] based on the American Water Works Association (AWWA) is developed and discussed in this subsection.

AWWA analyzed nearly one million individual water use events collected from 1188 residences in the 12 study sites, extensive household level information obtained from the mail survey completed by approximately 6000 households, and historic water

Table 2
The domestic hot water consumption by end-use.

End-use	End-use water temperature/°C	Water usage/L/d
Clothes washer	49.0	$28.4 + 9.46N_{\text{br}}$
Dishwasher	49.0	$9.46 + 3.15N_{\text{br}}$
Shower	40.6	$53 + 17.67N_{\text{br}}$
Bath	40.6	$13.25 + 4.43N_{\text{br}}$
Sinks	40.6	$47.32 + 15.75N_{\text{br}}$

Where: N_{br} = number of bedrooms.

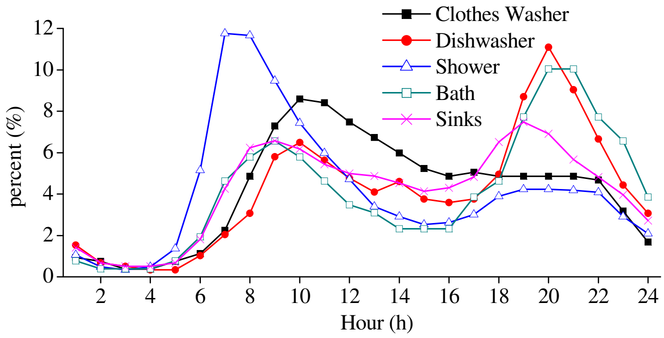

Fig. 2. Hot water use profile.

billing records from 12,000 residences [30]. The information on the end-uses of water in residential buildings provides the basis to estimate the indoor and outdoor uses individually, to identify variations in water consumptions for each fixture or appliance, and to develop prediction models to forecast residential water consumptions. There are two models: the end-use models can be used to predict water consumption of each type of end-use; another extend billing data models can be used to predict average total single-family household water consumption in any given billing periods and locations.

The extended billing data models are derived from the end-use models. For outdoor water consumptions, the billing data models were only applied to 12 sample cities. Only one among these 12 cities was the same as one of the selected cities in the present study (i.e. Denver). Therefore, modification is needed for this model extending to other cities for outdoor water consumption estimation. A parameter of evapotranspiration (ET) for the water requirement of irrigation is well documented and has a strong relationship with outdoor water consumption [31–33]. ET possibly can be used for adjustment of the outdoor water consumptions at various locations. The value of ET is related to two separate processes, which are the water loss (1) from the soil surface by evaporation and (2) from the crop by transpiration. It illustrates the water demand per unit time for a crop in particular meteorological and growing conditions. ET can be estimated by the use of standardized equations developed by the American Society of Civil Engineers (ASCE) based on the FAO Penman–Monteith model [34,35]. The end-use models, billing data models, and proposed extended billing data models are shown in the Appendix A.

The temperature of grey water can be interpolated through the flow rate and temperature of hot water and cold water supply. The method is shown as

$$T_{\text{grey}} = T_{\text{mains}} + \frac{(1 - 0.01\eta_V)L_{\text{hot,grey}}}{L_{\text{grey}}}\left[(1 - 0.01\eta_T)T_{\text{hot}} - T_{\text{mains}}\right] \quad (3)$$

where T_{grey} denotes temperature of grey water (°C). η_V and η_T are the loss coefficients of hot water's flow rate (%) and hot water's temperature (%), respectively. They are set to be 10% and 8%, respectively, in the present study. $L_{\text{hot,grey}}$ represents the flow rate of hot water that turns into grey water (L/h) and L_{grey} is the flow rate of grey water (L/h).

3.2.3. Wastewater tank temperature

The wastewater tank is an important component in this novel system. The tank does not only acts as the container for the submerged-pipe coil for heat transfer but also collects grey water for outdoor water use. Grey water is drawn into the tank to release heat, and then may drain off through the overflow pipe when the tank is full. If there is outdoor water consumption, the water will be delivered to the service points by a pump for irrigation at a specific time of day. In order to maintain the lowest water level set point in the tank, potable cold water will be used if necessary.

The volume and the temperature of water in the wastewater tank interact with the efficiency and heat extraction of the heat pump. Iteration is needed to calculate the temperature of water in the wastewater tank. There are three scenarios according to the water levels in the wastewater tank as discussed below.

In the first scenario, the wastewater tank is full of water, and the water level is just on the highest level as shown in Fig. 3. The grey water flows into the water tank and drains off through the overflow pipe to release the heat to the heat pump system. Using the conservation of energy and mass, the volume of temperature of water in the wastewater tank can be determined by

$$\begin{cases} V = V_0 \\ c_p \rho L_{\text{grey}} (T_{\text{grey}} - T) = Q_{\text{heat}} + c_p \rho V (T - T_0) \end{cases} \quad (4)$$

where T and T_0 are the hourly final temperature (°C) and the hourly start temperature (°C) of the water in the tank, respectively. V and V_0 denote the hourly final volume and hourly start volume of water in the tank (m^3), respectively. Q_{heat} is the extracted heat from the tank (kJ), which is less than zero when heat is rejected to the wastewater. c_p and ρ are the specific heat of water at constant pressure (kJ/(kgK)) and density of water (kg/m^3), respectively.

The resultant temperature of water in the wastewater tank can be obtained through rearrangement of the Eq. (4).

$$T = \frac{L_{\text{grey}} T_{\text{grey}} + V_0 T_0}{L_{\text{grey}} + V_0} - \frac{Q_{\text{heat}}}{c_p \rho (L_{\text{grey}} + V_0)} \quad (5)$$

The second scenario occurs when the water level is in between the lowest and highest water level as shown in Fig. 4. The grey water flows in, and no water drains off. The final volume and temperature of water in the wastewater tank are given by

$$\begin{cases} V = V_0 + L_{\text{grey}} \\ T = \dfrac{L_{\text{grey}} T_{\text{grey}} + V_0 T_0}{L_{\text{grey}} + V_0} - \dfrac{Q_{\text{heat}}}{c_p \rho (L_{\text{grey}} + V_0)} \end{cases} \quad (6)$$

respectively.

In these two abovementioned scenarios, if the temperature of water in the wastewater tank is set at extreme temperatures determined by the efficiency of heat pump, the maximum extracted or withdrawal heat can be found as

$$Q_{\text{heat,max}} = \begin{cases} c_p \rho [L_{\text{grey}} (T_{\text{grey}} - T_{\text{tank,min}}) + V_0 (T_0 - T_{\text{tank,min}})] & \text{extracted heat} \\ -c_p \rho [L_{\text{grey}} (T_{\text{tank,max}} - T_{\text{grey}}) + V_0 (T_{\text{tank,max}} - T_0)] & \text{withdrawal heat} \end{cases} \quad (7)$$

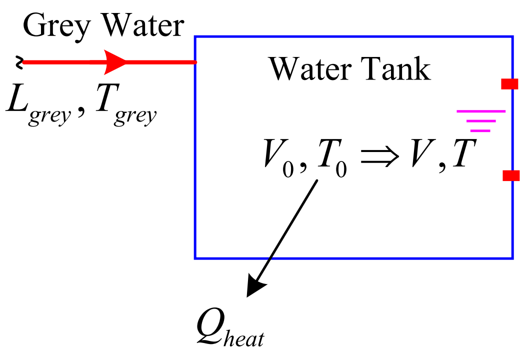

Fig. 4. Wastewater tank scenario 2: water level in between upper and lower limits.

respectively, where $T_{\text{tank,min}}$ and $T_{\text{tank,max}}$ are the minimum temperature set point and maximum temperature set point of water in tank (°C).

The third scenario occurs when the water level is lower than the lowest water level set point as shown in Fig. 5. Therefore, potable cold water supply is required to keep the lowest water level. The required volume of mains water and the temperature of wastewater in the tank are given as

$$\begin{cases} L_{\text{mains}} = V - V_0 \\ T = \dfrac{V_0}{V_{\text{lowest}}} T_0 + \dfrac{V_{\text{lowest}} - V_0}{V_{\text{lowest}}} T_{\text{mains}} \end{cases} \quad (8)$$

where V_{lowest} represents the water volume to keep the lowest water level in the wastewater tank (L), while L_{mains} is the required volume of mains water (L).

3.2.4. Energy consumption and evaluation

The proposed and conventional systems consist of air-conditioner, heat pump, gas furnace, fan, pump, etc. The efficiency and partial load performance of the equipment significantly influence the energy consumption. In this numerical (feasibility) study, the output performance of the compressor provided by the manufacturer was adopted [36], and the simplified heat transfer models of evaporator and condenser were incorporated. The details of the models can be found in Appendix B. In this way, the efficiency of the heat pump, coefficient of performance (COP), could be calculated from the temperatures of water in the wastewater tank and the outdoor air. For a conventional air-conditioner, the COP was determined by the performance data from typical product data sheets. In addition,

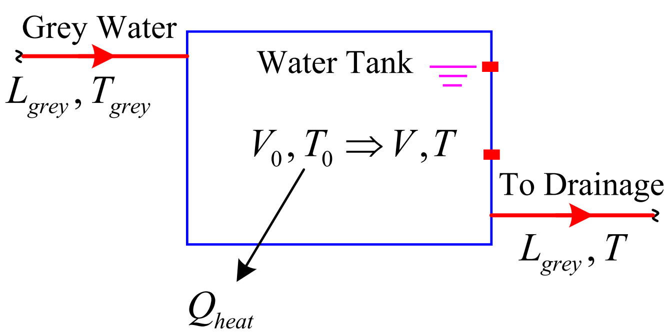

Fig. 3. Wastewater tank scenario 1: full of water. The grey water flows in, drain off from overflow pipe and releases heat.

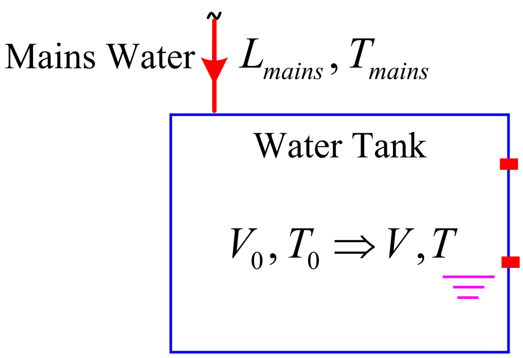

Fig. 5. Wastewater tank scenario 3: water level below the lower set point. The mains water flows in to reach the lowest water level.

annual fuel utilization efficiency (AFUE) is used for estimation of the energy consumption. The AFUE of the gas furnace was 0.80, and the AFUE of the electrical boiler for hot water heating was 0.92. Then the fuel utilization can be derived from heating water load divided by AFUE.

Moreover, the power consumption of pumps and fans also was included in the energy analysis. According to the required flow rate and static pressure, the pumps and fans were selected and the power of pumps and fans were found. The energy consumptions of pumps and fans are equal to the simple product of power and operation time because there is only start–stop control and no speed control in the system.

In order to evaluate the energy consumption of the wastewater energy-recovery system and the conventional space heating and cooling and hot water heating system, the natural gas and electricity were both converted to source energy. Based on 2007 data from EPA *eGRID* version 1.1, the source energy conversion factors of electricity (SECFE) for the 15 cities (states) are shown in the Table 1 [44]. The variations of SECFEs are due to the different percentages of electricity generated by particular fuel types, such as coal, natural gas, fuel oil, nuclear, hydro, or biomass. For natural gas, the source energy conversion factor is 1.088, which considers the process energy efficiency (efficiency of extraction is 97.0%, processing is 96.9%, transportation is 99.0% and distribution is 98.8%) [44].

4. Energy consumption analyses

4.1. Assumptions and strategies

The proposed system has a flexible design that allows several approaches to meet the required loads of space heating, space cooling, and hot water heating. Some operation strategies and control rules are discussed below.

(1) When space cooling load is used, the condenser heat first is reclaimed for domestic hot water heating. If space heating load and hot water load are used simultaneously, the sensible heat of compressor's output is extracted for hot water heating and the latent heat is for space heating.

(2) In space heating or hot water heating mode, a minimum temperature set point of grey water in the tank is 4.4 °C. If the temperature of grey water is lower than the minimum temperature set point, the heat pump will be switched to extract heat from outdoor air. However, if the temperature of the outdoor air is lower than −4.0 °C, the heat pump will be stopped due to inefficient operation of the air source heat pump at such low output temperature. A supplementary heat source will be used to supply heat for the space heating and hot water heating. The control strategies for space heating mode are shown in Fig. 6.

(3) In space cooling mode, if there is a surplus of condenser heat, the outdoor air will be firstly used for as a cooling sink for the heat pump. The grey water may be used as a heat sink only when the outdoor air temperature is higher than 28.0 °C. Since the temperature of grey water may be high in summer, heat can only be rejected in grey water during a particular period of time each day. The time of using grey water as a heat sink is determined based on the distribution of cooling load and the time of irrigation for outdoor water use. For New York, the grey water is used for cooling at 3:00 pm if the outdoor temperature is less than 28 °C. Grey water will be delivered to service points for outdoor water use at 4:00 pm, and the potable cold water will be drawn to the tank if the water volume in the tank is less than the lowest water level set point. The water is used until the temperature of water in the tank is higher than the highest temperature set point, 35.6 °C. The time of 3:00 pm and 4:00 pm are selected because the design cooling load generally occurs at 4:00 pm, and daily maximum cooling loads of most locations (approximately 73%) occur between 4:00 pm and 6:00 pm. The control strategies for space cooling mode are shown in Fig. 7.

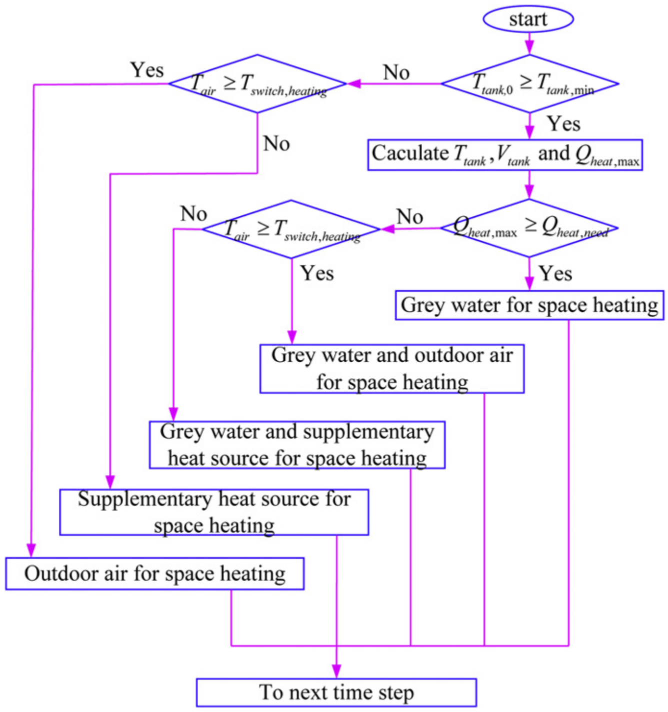

Fig. 6. The control strategies for space heating mode. T_{tank}, $T_{tank,0}$ and $T_{tank,min}$ are the hourly final temperature (°C), the hourly start temperature (°C) and the minimum temperature set point of water in the grey water tank (°C), respectively. V_{tank} denotes the hourly volume water in the tank (m^3). T_{air} and $T_{switch,heating}$ are the outdoor air temperature (°C) and the switch temperature from outdoor air to supplementary heat source in space heating mode (°C). $Q_{heat,max}$ and $Q_{heat,need}$ are the maximum extracted heat from the grey water tank (kJ) and the low-grade heat needed for heat pump (kJ).

(4) Generally speaking, it is beneficial when the system performs multiple-functions (space heating, space cooling and hot water heating) simultaneously. Of course, each function should share the benefit of electricity consumption saving. Therefore, even if the energy of hot water heating comes from condenser heat, the hot water heating also needs to share an equal proportion of energy consumption of the compressor, because the recovery of condenser heat may decrease the COP of the unit.

(5) The equipment for conventional system are a gas furnace for space heating, electrical package air-conditioner for space cooling, and an electric water heater for hot water heating. However, the auxiliary heat sources of the proposed system for space heating and hot water heating, if the outdoor air dry bulb temperature is lower than −4.0 °C, are gas and electricity, respectively.

(6) If the water level is higher than the maximum level, the grey water will drain off through the overflow pipe. Otherwise, if the water level is lower than the minimum level set point, the potable cold water will be drawn. The maximum and minimum effective volumes of the water tank are 1.3 m^3 and 0.83 m^3, respectively.

4.2. Results and discussion

The comparison of daily source energy consumption between the proposed system and the conventional system for the prototype

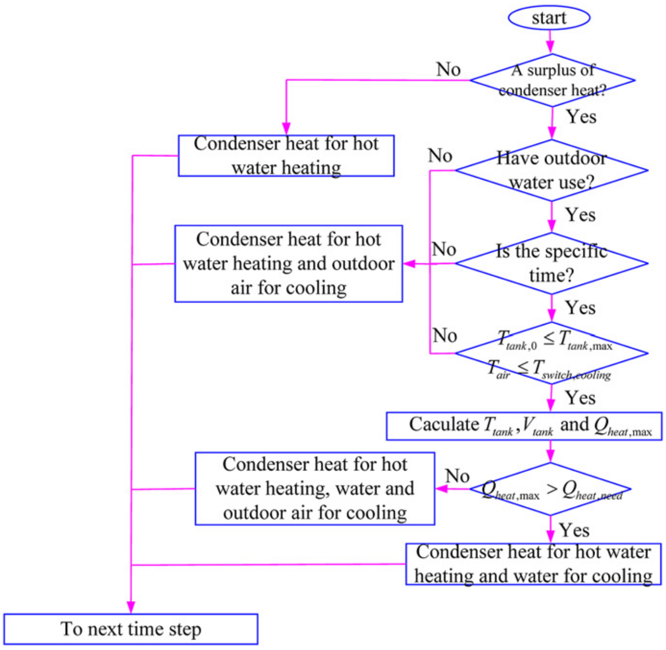

Fig. 7. The control strategies for space cooling mode. T_{tank}, $T_{tank,0}$ and $T_{tank,max}$ are the hourly final temperature (°C), the hourly start temperature (°C) and the maximum temperature set point of water in the grey water tank (°C), respectively. V_{tank} denotes the hourly volume water in the tank (m^3). T_{air} and $T_{switch,cooling}$ are the outdoor air temperature (°C) and the switch temperature from outdoor air to grey water in space cooling mode (°C). $Q_{heat,max}$ and $Q_{heat,need}$ are the maximum withdrawal heat to the grey water tank (kJ) and the withdrawal heat needed for heat pump in space cooling mode (kJ).

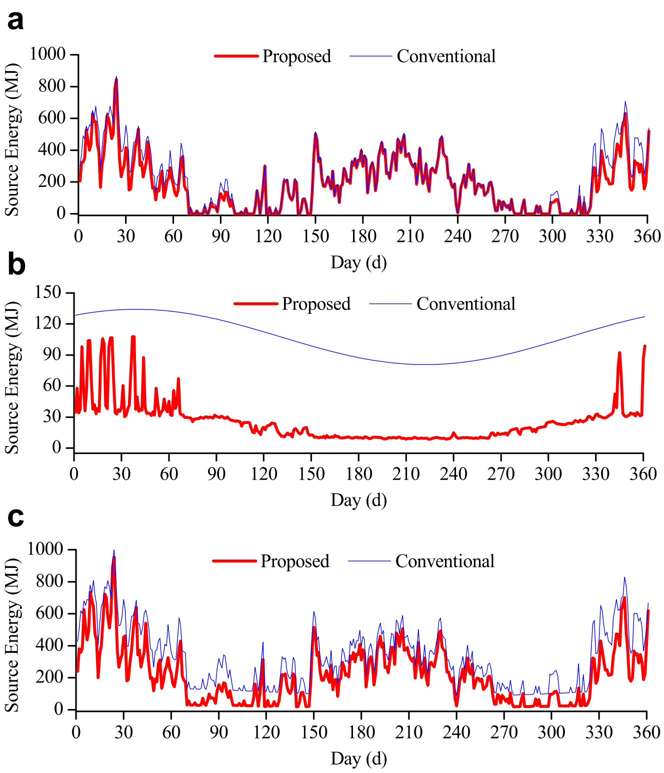

Fig. 8. The comparison of daily source energy consumption between the proposed system and the conventional system. (a) Space heating and air-conditioning; (b) Hot water heating; (c) Total.

in New York, NY is shown in Fig. 8. Compared with conventional systems, the localized residential wastewater energy-recovery system can save 33.9% source energy consumption. For hot water heating, the energy savings is 76.0%. There are at least three reasons for this energy saving:

(1) The efficiency of a heat pump is much higher than a gas furnace for space heating and an electric water heater for hot water heating.
(2) The efficiency of the heat pump may also be higher than an air-conditioner package for space cooling due to a lower temperature of grey water. In New York city, the mean COP of an air-conditioner package is 4.026. However, if the grey water is used for peak cooling load, the mean COP increases to 4.053.
(3) The recovery of condenser heat in space cooling for service water heating also plays an important role in energy savings.

Investigating the distribution of heat source and sink of space heating, hot water heating and cooling, as showed in the Table 3, electricity and supplement source (gas) supply 44.72% of heating load, and the remaining heating load is supplied by the low-grade grey water and air source for space heating. Moreover, for hot water heating, 77.11% of hot water load is supplied by grey water, air and condenser heat. The electricity and supplement only supply 22.89% of hot water load.

In summer, the peak electricity demand of conventional package air-conditioning and electrical water heater is 4.4 kW, but that of the proposed system is only 3.9 kW. The proposed system reduce 10.6% of peak electricity demand and delay the peak demand for an hour, i.e. from 4:00 pm to 5:00 pm.

Fig. 9 gives the results of energy consumptions using the proposed system in different cities. The proposed system shows good energy saving capability. Among these 15 cities, the minimum of total energy savings in percentage is 17.1%, while the maximum is 57.9%.The best energy savings occurs in Bellingham, WA.

The relationships of energy savings in percentage (ESP) with heating degree days (HDD), cooling degree days (CDD) and mean outdoor air temperature (MOAT) are shown in Fig. 10. The energy savings of space heating in percentage exponentially decrease as the number of heating degree days increases. The reason is due to the fact that, when the weather becomes colder and colder, there is longer period of time for the proposed system using the supplemental heating source, i.e. gas furnace other than heat pump, with the increase of heating degree days. These may reduce the energy saving of the proposed system. The ESP of hot water heating gradually decreases as the MOAT increases and has an approximately linear relationship with the MOAT. The energy savings of hot water heating result from the heat pump and recovery of condenser heat. The heat required for hot water heating increases for

Table 3
The distribution of heat source and sink of space heating, hot water heating and cooling.

	Space heating	Space cooling	Hot water heating
Total/GJ	31.5	44.5	12.3
Electricity/%	19.47	–	17.75
Supplement/%	25.25	–	5.14
Subtotal/%	44.72	–	22.89
Grey water and outdoor use/%	10.36	3.03	32.44
Outdoor air/%	44.92	89.07	16.02
Hot water/%	–	7.90	–
Condenser heat/%	–	–	28.65
Subtotal/%	55.28	100	77.11
Total/%	100	100	100

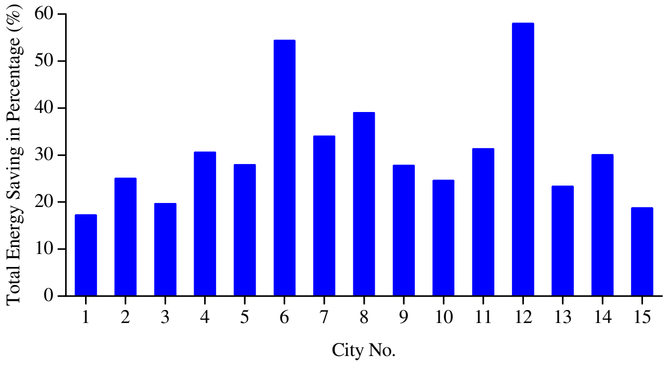

Fig. 9. Total energy saving of energy consumption in percentage (The city no. refer to Table 1).

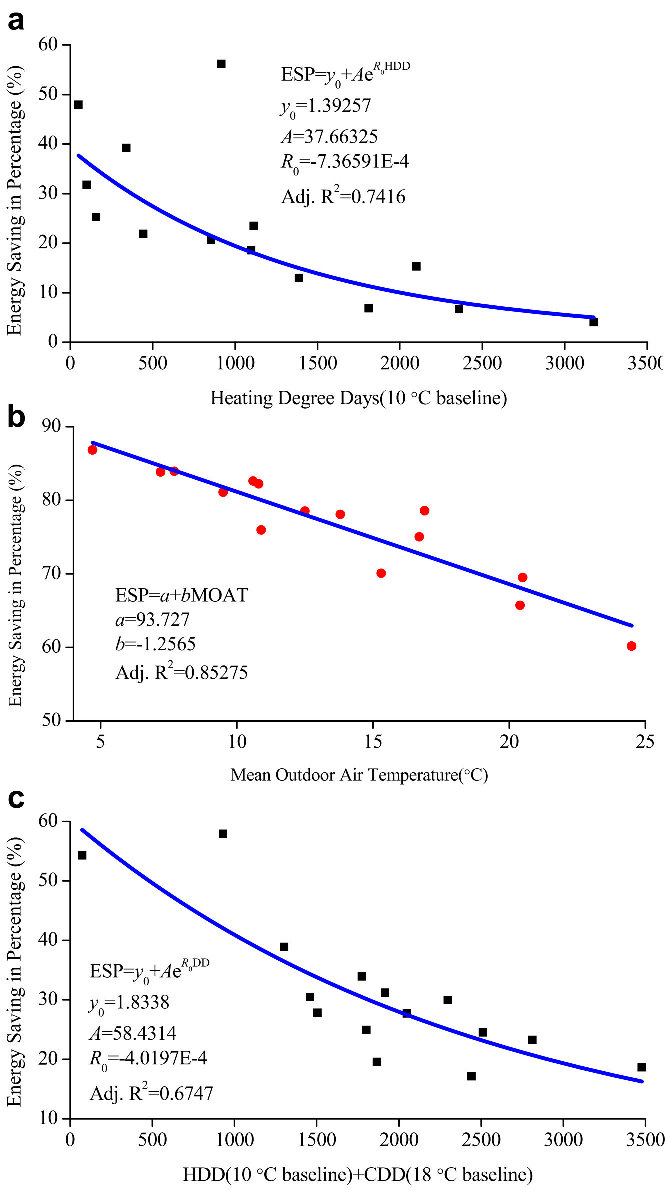

Fig. 10. The relationship of energy saving in percentage with local climate conditions. (a) Space heating and heating degree days; (b) Hot water heating and annual mean outdoor air temperature; (c) Total energy saving in percentage and the sum of heating degree days (HDD) and cooling degree days (CDD).

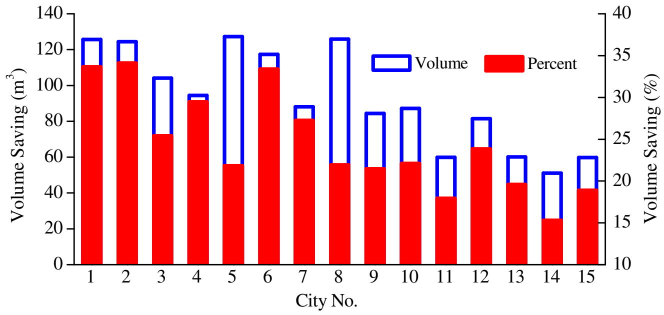

Fig. 11. Potable water savings of the proposed system in percentage and volume.

a decrease in temperature of potable cold water supply, which is related to the outdoor air temperature. The ESP of the proposed system also shows a relationship with the sum of heating degree days and cooling degree days. Generally, the value of the sum of heating degree days and cooling degree days reflects the heterogeneity of outdoor air temperature. If this value is large, the weather of this location is either extremely hot or cold. That is to say, the place with moderate temperature is more suitable for the proposed system and can achieve larger ESP. Otherwise, in very cold climates the advantage of air heat pump cannot be utilized due to low efficiency at low-temperature. In turn, in very hot climates, there is very small or no requirement of space heating. The only advantage of this system is the recovery of condenser heat.

Furthermore, this new system has excellent potable water saving potential because the treated grey water is used for outdoor water use, for example, irrigation. The mains water saving capability is illustrated in Fig. 11. For a 4-member single-family, the new system can save at least 50 m^3 of potable water per year, and some locations can save as much as 130 m^3 per year. The percentages of potable water savings have a range of 15%–34.1%. In general, the potable water savings are more significant in hot climates.

5. Conclusion

An innovative grey water energy-recovery system has been developed for domestic water heating and space heating and cooling of buildings in order to improve the overall efficiency and total energy input of building energy usage. This system adopted a multiple-function heat pump system to reclaim the energy from grey water in association with outdoor air extraction and rejection. Moreover, a multiple-function water tank was developed. This water tank, which can service as a container for a heat exchanger and grey water, allows the treated grey water to be used for irrigation as required, rather than be aimlessly sprinkled on the ground. A numerical model has been developed for the comparison of annual energy consumption and potable water savings between the innovative water energy-recovery system and the conventional building energy system with gas furnace space heating, package air-conditioning and an electric water heater for hot water heating. From the results, it can be concluded that

(1) The innovative water energy-recovery system has high energy savings and potable water savings. This new system is most suitable for places with moderate outdoor temperature.
(2) From a case study of a typical residential house with four family members and three bedrooms in New York, NY, the simulation results show that source energy consumptions decrease 23.5%, 2.7% and 76.0% for space heating, cooling, and hot water heating, respectively. The statistical results of energy consumption show that the total energy savings are as high as 33.9%

compared with that of the conventional system. The proposed system also reduces potable water consumption by 27.2% (or 72.9% of outdoor water use).

(3) Fourteen other cities were selected for an extended study in different climate zones in the U.S. Among all 15 cities, the total source energy savings have a range of 17%–57.9%. Hot water heating has the most significant energy savings with over 60% reduction. The results demonstrate high energy savings of this proposed system. Furthermore, the potable water savings have a range of 15%–34.1%. The higher water savings can be observed in the hot climate.

Acknowledgements

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Appendix A. Grey water and outdoor water use

The monthly indoor and outdoor use can be found by the following equations [30]:

$$q_{m,\text{in}} = e^{3.004+\beta_m} q_t^{0.372} q_s^{0.012} q_f^{0.124} q_d^{0.066 b_d} q_c^{-0.020 b_c} q_l^{0.085} q_o^{0.006} \tag{A.1}$$

$$q_m = q_{m,\text{in}} q_{\text{out}}^{\beta_{m,L} b_{\text{out}}} \tag{A.2}$$

$$q_{m,\text{out}} = q_m - q_{m,\text{in}} \tag{A.3}$$

where q denotes the daily household indoor water use in units of gal/household/day. b is a binary (0/1) variable denoting the presence of an end-use. β_m is the adjustment for billing month; $\beta_{m,L}$ is the adjustment for billing month and location. The subscripts are defined as: m = the billing month, in = indoor, out = outdoor, t = toilet, f = faucet, d = dishwasher, c = clothes washer, l = leaks and o = other.

The average water consumptions of toilet flushing, shower, faucet, dishwasher, clothes washer, leaks and other/unknown use can be predicted by the end-use models [30]. In the end-use models, socioeconomic factors, such as household sizes, incomes, marginal price of water and wastewater, the home's age, and lot sizes, etc. are considered.

The values of β_m are listed in Table A.1 from the AWWA study. The values of $\beta_{m,L}$ have a strong relationship with the locations due to different climates. Although the AWWA report gave the default values of $\beta_{m,L}$ at selected cities, there is only one city (i.e. Denver) in the present study listed in the AWWA report. Therefore, it is critical to extend the estimation of $\beta_{m,L}$ for the other cities.

Generally, the purposes of outdoor water use are irrigation and swimming pools. The water required for irrigation is dominated and has a strong relationship with evapotranspiration (ET). However, for single-family homes, the outdoor water consumption is affected by many factors including land size, irrigation method, landscape type, landscape quality, and water price. These factors are complicated and interact with each other. ET is one of the factors. Nevertheless, the effects of land size, irrigation method, landscape type, landscape quality and water price are considered in estimating q_{out} [30]. Only the effects of month and location are involved in the parameter $\beta_{m,L}$ for irrigation, which can be determined by meteorological and growing conditions. Therefore, the values of $\beta_{m,L}$ may have a strong relationship with ET and the precipitation. In order to find this relationship, the values of ET, precipitation, and $\beta_{m,L}$ in 11 sample cities in AWWA study [30] (except Waterloo, Ontario, Canada due to lack of meteorological data) were analyzed. A linear regression analysis of ET, precipitation, and $\beta_{m,L}$ is performed.

A linear regression model is built based on the relationship of $\beta_{m,L}$ with ET and precipitation in 8 selected cities in AWWA report [30]. Then the other 3 cities are left to validate the resultant linear regression model. The linear regression model is shown as below:

$$\beta_{m,L} = z_0 + a \times \text{ET} + b \times \text{PS} \tag{A-4}$$

where z_0, a, and b are coefficients for linear regression model, which are found to be $z_0 = 0.17308$, $a = 5.58299 \times 10^{-4}$ and $b = -5.53988 \times 10^{-4}$, respectively. ET is the monthly mean evapotranspiration in a unit of mm/month. PS denotes monthly mean liquid precipitation in a unit of mm/month, which can be obtained by

$$\text{PS} = P + 0.1\,S \tag{A.5}$$

where the P and S are monthly mean precipitation and snow fall, respectively, assuming the density of snow is 100 kg/m^3.

Fig. A.1 shows the validation of the linear regression model using the data of other three selected cities. There is good agreement between sample data and the resultant linear regression model. In the present study, this model is used for estimation of $\beta_{m,L}$ in other locations not shown in the AWWA study. The ET and precipitation can provide approximately acceptable expressions for the outdoor water use.

Appendix B. Models of the refrigeration system in the prototype

B.1 Model of the compressor

In this prototype, the compressor model was founded in map-based method [36] given as:

$$f_i(T_c, T_e) = c_{1i} T_c^2 + c_{2i} T_c + c_{3i} T_e^2 + c_{4i} T_e + c_{5i} T_c T_e + c_{6i} \tag{B.1}$$

where the f_i is the power consumption, refrigerant flow rate or refrigerating capacity of the compressor; c_{1i} to c_{6i} are the fit coefficients. T_c and T_e are the temperatures of condenser and evaporator (°C), respectively.

If the degrees of superheat and subcooling are varied, the amendatory equations are showed as:

$$m_{r,\text{act}} = \left[1 + F_v\left(\frac{v_{r,\text{map}}}{v_{r,\text{act}}} - 1\right)\right] m_{r,\text{map}} \tag{B.2}$$

$$W_{\text{act}} = \left(\frac{m_{r,\text{act}}}{m_{r,\text{map}}}\right)\left(\frac{\Delta h_{\text{com,act}}}{\Delta h_{\text{com,map}}}\right) W_{\text{map}} \tag{B.3}$$

$$Q_{e,\text{act}} = \left(\frac{m_{r,\text{act}}}{m_{r,\text{map}}}\right)\left(\frac{\Delta h_{\text{eva,act}}}{\Delta h_{\text{eva,map}}}\right) Q_{e,\text{map}} \tag{B.4}$$

where m_r denotes the flow rate of refrigerant (kg/s); W is the power consumption of compressor (kW); Q_e is the refrigerating capacity (kW); v_r is the specific volume in the suction inlet of compressor (m^3/kg); F_v is the corrective coefficient of volumetric efficiency; and Δh_{com} and Δh_{eva} are the enthalpy differences between inlet and outlet of compressor and evaporator, respectively. The subscripts are defined as: act = actual condition and map = given condition.

In this paper, the output performance chart of compressor serial 10B1778AL for R410A produced by Copeland Scroll was adopted.

B.2 Model of fin-tube heat exchanger

The fin-tube heat exchanger can be air condenser or air evaporator. For the fin-tube heat exchanger, the three-zone steady-state model was applied. The model assumes that the heat exchanger can be divided into three distinct zones on the refrigerant side: the vapour de-superheating zone, the two-phase zone and the sub-cooled liquid zone. In each zone, one global heat transfer coefficient was calculated.

(1) Heat exchanger equation

For refrigerant side,

$$Q_r = m_r(h_{r,o} - h_{r,i}) = \alpha_r A_r (T_w - T_{r,m}) \tag{B.5}$$

For air side,

$$Q_a = m_a(h_{a,i} - h_{a,o}) = \xi \alpha_a A_a (T_{a,m} - T_w) \tag{B.6}$$

If the heat leakage was considered, the relationship of heat transfer in refrigerant side and air side can be written as:

$$Q_a = \gamma Q_r \tag{B.7}$$

where Q denotes heat (kW); m is mass flow (kg/s); h is enthalpy (kJ/kg); A is surface area (m^2); α is coefficient of convective heat transfer (W/(m^2 °C)); T is temperature (°C); ξ is moisture absorption coefficient; and γ is the coefficient of heat leakage, which is set 0.9 in this study. The subscripts are defined as: r = refrigerant; a = air; i = inlet; o = outlet; w = wall; m = mean.

(2) Convective heat transfer coefficient

In the vapour de-superheating zone and the sub-cooled liquid zone, the refrigerant is in single phase, the convective heat transfer coefficient can be translated as in Wu and Han [37].

$$\alpha_s = 0.023 Re_s^{0.8} Pr_s^{0.4} \frac{\lambda_s}{d_e} \tag{B.8}$$

where Re denotes Reynolds number; Pr denotes Prandtl number; λ is the coefficient of heat conductivity (W/(m°C)); and d_e is diameter of tube (m). The subscript s means single phase.

While in the two-phase zone, when the fin-tube heat exchanger is condenser, the correlation of the convective heat transfer coefficient is [38]:

$$\alpha_{tp} = \alpha_s \left[(1-x)^{0.8} + \frac{3.8x^{0.76}(1-x)^{0.04}}{Pr_{tp}^{0.83}} \right] \tag{B.9}$$

where x is refrigerant dryness (%); and the subscript tp means two-phase.

When the fin-tube heat exchanger is evaporator, the correlation of the convective heat transfer coefficient in two-phase zone can be found by [39];

$$\alpha_{tp} = \begin{cases} \alpha_r(x) & 0.2 < x < x_d \\ \alpha_r(x_d) - \left(\dfrac{x - x_d}{1 - x_d}\right)^2 [\alpha_r(x_d) - \alpha_s] & x \ge x_d \end{cases} \tag{B.10}$$

In addition,

$$\alpha_r(x) = 3.4 \left(\frac{1}{X_{tt}}\right)^{0.45} \alpha_l \tag{B.11}$$

$$\alpha_l = 0.023 Re_l^{0.8} Pr_l^{0.3} \frac{\lambda_s}{d_e} \tag{B.12}$$

$$x_d = 7.943 \left[Re_v \left(2.03 \times 10^4 Re_v^{-0.8} (T_{w,i} - T_e) - 1 \right) \right]^{-0.161} \tag{B.13}$$

$$X_{tt} = \left(\frac{1-x}{x}\right)^{0.9} \left(\frac{\rho_v}{\rho_l}\right)^{0.5} \left(\frac{\mu_l}{\mu_v}\right)^{0.1} \tag{B.14}$$

where ρ is density (kg/m^3); μ is kinetic viscosity (Pa s) and the subscripts v and l mean vapour and liquid, respectively.

The convective heat transfer coefficient in air side is given as [40]:

$$\alpha_a = 0.772 Re^{0.447} \left(\frac{S_1}{d_e}\right)^{-0.363} \left(\frac{NS_2}{d_e}\right)^{-0.217} \frac{\lambda_a}{d_e} \tag{B.15}$$

where S_1 is the fin spacing (m); S_2 is tube spacing along air flow (m); N is the number of tube rows and the subscript a means air.

B.3 Model of plate exchanger

The convective heat transfer coefficient of single phase fluid, including liquid refrigerant, vapour refrigerant and water, is derived by [41]:

$$Nu = 0.2121 Re_f^{0.78} Pr_f^{\frac{1}{3}} \left(\frac{\mu_f}{\mu_w}\right)^{0.14} \tag{B.16}$$

where the subscripts f and w denote fluid and wall of plate exchanger.

In two-phase zone of a plate exchanger condenser, the heat transfer coefficient can be written as [42]:

$$Nu = 0.0143 Re_{vl}^{0.86} Pr_l^{\frac{1}{3}} \frac{1}{2} \left[\sqrt{1 + x_i \left(\frac{\rho_l}{\rho_{tp}} - 1\right)_i} + \sqrt{1 + x_o \left(\frac{\rho_l}{\rho_{tp}} - 1\right)_o} \right] \tag{B.17}$$

B.4 Model of thermodynamics expansion valve

The thermodynamics expansion value model is built up in force balance method. The flow rate of refrigerant is obtained by [43]:

$$m_{exp} = 447.2 C_D A_{exp} \sqrt{\rho_{exp,i} \Delta P_{exp}} \tag{B.18}$$

$$C_D = 0.02005 \sqrt{\rho_{exp,i}} + 0.634 v_{exp,o} \tag{B.19}$$

$$A_{exp} = \pi h (\sin 0.5\theta)(d - 0.5h \sin\theta) \tag{B.20}$$

where m_{exp} denotes flow rate of refrigerant (kg/s); C_D is flow coefficient; A_{exp} is circulation area (m^2); $\rho_{exp,i}$ is refrigerant density in inlet of expansion valve (kg/m^3); ΔP_{exp} is pressure difference between inlet and outlet of expansion valve (10^5 Pa); $v_{exp,o}$ is specific heat capacity in outlet of expansion valve (m^3/kg); h is displacement of valve flap (m); θ is cone angle of valve body; and d is diameter of pinhole in expansion valve (m). The energy equation of expansion valve is

$$h_{exp,i} = h_{exp,o} \tag{B.21}$$

where $h_{exp,i}$ and $h_{exp,o}$ are the enthalpy in inlet and outlet of expansion valve (kJ/kg), respectively.

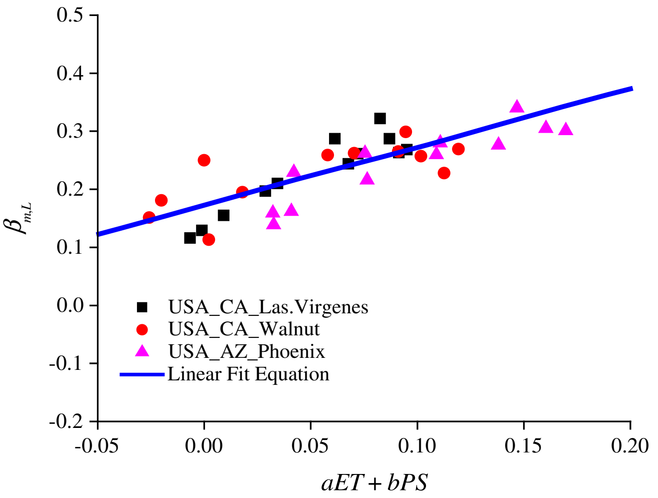

Fig. A.1. The validation of linear regression model using the data of selected 3 cities.

Table A.1
The values of β_m.

Month	Jan	Feb	Mar	Apr	May	Jun
β_m	−0.170	−0.177	−0.211	−0.031	−0.160	0.106
Month	Jul	Aug	Sep	Oct	Nov	Dec
β_m	0.108	0.166	−0.108	−0.162	−0.156	0.000

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