13th IEA Heat Pump Conference
April 26-29, 2021 Jeju, Korea

A residential heat pump system for cooling, heating, dehumidification and outdoor air supply

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Abstract

With growing demands of indoor air quality and thermal comfort, dedicated outdoor air system with dehumidification function is becoming more popular for residential use. However, due to limited air volume of outdoor air, dedicated outdoor air system often needs to supply air at very low humidity ratio for dehumidification, which decreases the system energy efficiency. In this work, by combining dedicated outdoor air system and small air-cooled chiller, a heat pump system for cooling, heating, dehumidification and outdoor air supply is proposed. In the system, a high temperature water coil is applied to precool the outdoor air for decreasing the energy consumption of dehumidification. Besides, dehumidification capacity of outdoor air system with precooling is enough to handle overall latent load. In the meantime, without the demand of dehumidification, the water supplying temperature of chiller can rise to 15°C or above, which is not only beneficial to energy efficiency improvement, but also available for dry terminal. Modeling results of the system indicate that the annual power consumption of the new system can be reduced by 16.9% relative to the combined operation of a dedicated outdoor air system and an individual small chiller.

Keywords: heat pump; air conditioning; chiller; outdoor air system; dehumidification

Nomenclature

A	area (m^2)	SA	supply air
APF	annual performance factor	T	temperature (°C)
c	specific heat($kJ \cdot kg^{-1} \cdot K^{-1}$)	TMY	typical meteorological year
COP	coefficient of performance	v	specific volume ($m^3 \cdot kg^{-1}$)
D	diameter (m)	WB	wet bulb
DB	dry bulb		
DOAS	dedicated outdoor air system	*Greeks*	
EXV	electronic expansion valve	η	efficiency
f	friction factor; frequency (Hz)		
G	mass flux ($kg \cdot s^{-1} \cdot m^{-2}$)	*Subscripts*	
h	specific enthalpy ($kJ \cdot kg^{-1}$)	a	air
Le	Lewis number	c	condensing; cooling season
m	mass flow rate ($kg \cdot s^{-1}$)	e	evaporating
OA	outdoor air	h	heating season
p	pressure (kPa)	r	refrigerant
Q	cooling/heating capacity(kW)	s	surface
RA	return air		

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1. Introduction

Since people pay more attention to indoor air quality today, residential dedicated outdoor air system (DOAS) become more popular in the market, and different types of DOAS have been widely researched, too. The basic type of DOAS is just in charge of fresh air supplying and purification. But considering energy saving and indoor thermal comfort, now DOAS usually has an additional dehumidification unit and can deal with partial or overall latent load.

For dehumidification DOAS, desiccant dehumidification and mechanical dehumidification are the two main approaches[1]. Desiccant dehumidification method transfers water from air to solid or liquid desiccant by adsorption or absorption process. It usually has better dehumidification efficiency, but it also has the problem of high cost, large volume and regeneration[2]. On the contrary, mechanical dehumidification method, which has relative lower dehumidification efficiency but lower cost and small volume, is more popular in residential DOAS market. Up to now, most of improved DOAS types were developed for better performance. Mumma et al. developed a novel DOAS to reduce humidification and reheating load, including a preheat coil, an enthalpy wheel, a deep cooling coil and a sensible heat exchanger[3]. Zhang et al. proposed a novel DOAS with total heat exchanger[4, 5]. Li et al. proposed a novel DOAS with multi-stage direct expansion coil and zero-energy heat pipe[6]. This system could reduce about 15.6% annual energy consumption according to the simulation results. Zhang et al. designed a frost-free exhaust air heat recovery DOAS[7]. Zhang et al. proposed a novel DOAS with two-stage direct-expansion dehumidification, subcooled reheating, and exhaust air heat recovery[8]. Coefficient of performance (COP) of this novel DOAS can reach 5.42 and 26% higher than the conventional one.

With the development of DOAS technology itself, the issue about how DOAS help air conditioning system save energy is also drawing much attention. A conventional solution is DOAS with the sensible heat load handling device. In this solution, a dehumidification DOAS deals with overall latent load, and the indoor terminal only deal with partial sensible load. This solution is a way to achieve temperature and humidity independent control, which can improve indoor thermal comfort effectively. Mumma et al. has shown that the combination of DOAS and radiant cooling system can reduce about 29% energy consumption compared to conventional variable air volume systems[9]. Saber et al. evaluated the performance of a decentralized DOAS with a radiant cooling system in the tropical climate[10]. Yin et al. analyze the energy saving potential of DOAS with radiant cooling system under different climate conditions in China[11]. Zhang et al. discussed the adaptability of the combined system with DOAS and radiant cooling panels in summer moist heat regions[12]. Li et al. developed a radiant air conditioning system with outdoor air dehumidifier for temperature and humidity independent control, and tested the performance of this system under variable operating conditions[13]. Lim et al. compared the performance of DOAS with thermoelectric module radiant cooling panel or hydraulic ceiling radiant cooling panels via a detailed energy simulation[14]. However, although the COP of chiller in this combination solution can be improved due to the high temperature of chilled water, COP of dehumidification DOAS in such solution is still relatively low. COP of the common mechanical dehumidification DOAS in the market is at most 3.5[6]. Due to the limited air volume of outdoor air, dehumidification DOAS often needs to supply air with very low humidity ratio and low temperature, which lead to low evaporating temperature and increase the energy consumption for dehumidification. Therefore, a higher efficiency solution for dehumidification DOAS is still needed to be explored.

In this work, we proposed a novel multi-function residential heat pump system, which included chiller subsystem and outdoor air subsystem. In the system, a high temperature water coil is applied to precool the outdoor air for decreasing the energy consumption of dehumidification. With an ingenious iterative design of two subsystems, the new system can achieve better performance than direct combination of chiller and DOAS. This paper is composed of five sections. Section 2 gives a system description of this new system. Section 3 presents a system model of the new system for performance analysis, and introduces the method of system modeling. Section 4 discusses the performance of new system under variable operating conditions, according to the simulation results. And section 5 gives some brief conclusions of this work.

2. System description

The new system is an integration of conventional chiller system and DOAS. Fig. 1 is the schematic of new system. It has 2 subsystems: chiller subsystems and outdoor air subsystem (OA subsystem), and the two subsystems are connected with each other by water pipes.

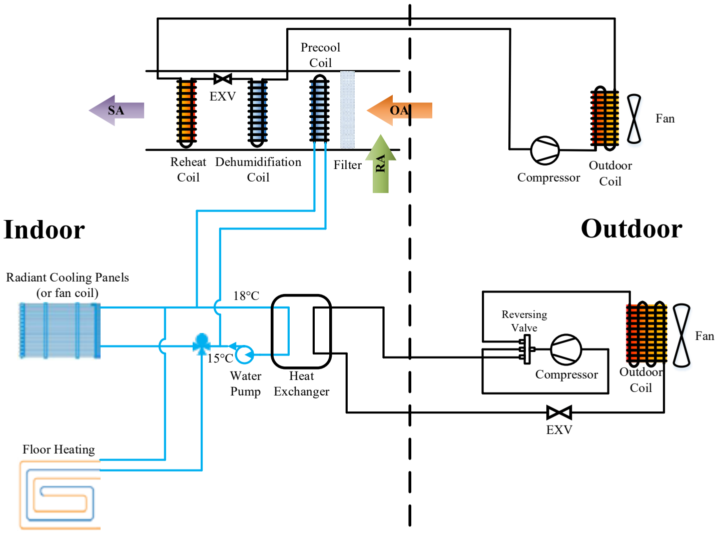

Fig. 1. Schematic of the new system

The chiller subsystem(below) is similar to a conventional residential chiller system. The difference is that water temperature of chilled water is 15°C in cooling season. And types of terminals are radiant cooling panels or dry fan coil. High temperature chilled water means better performance and less energy consumption of the chiller subsystem. But the chiller subsystem only deals with sensible load due to high temperature of chilled water.

The OA subsystem deal with the overall latent load and part of sensible load of rooms in cooling season. In order to increase the air volume, return air is mixed with outdoor air. Then the mixed air flow through three coils: precool coil, dehumidification coil and reheat coil. Precool coil is a water-air coil and water are from chiller subsystem. Dehumidification coil and reheat coil belong to a dedicated refrigeration cycle, driven by a dedicated compressor and an outdoor coil of OA subsystem. The mixed air is cooled, dehumidified and reheated by these three coils and sent to indoor finally.

In terms of the control strategy, temperature and humidity independent control is an important control target of this system. Therefore, compressors in both chiller subsystem and OA subsystem are variable speed compressor. The desired indoor dry bulb temperature is controlled by compressor of chiller subsystem, and the desired absolute humidity of supply air (SA) is controlled by compressor of OA subsystem. The suction superheat of these two refrigeration cycles is controlled by the EXV in respective cycle.

In heat pump mode, the dedicated refrigeration cycle of OA subsystem is turned off, and the precool coil change to preheat coil. To avoid unnecessary energy waste of mixing process, the RA inlet is turned off by an air valve in heat pump mode. Since there is no latent load in heating season, the overall load of OA subsystem is less than in cooling season. Therefore, the heating capacity of preheat coil is enough to heat the outdoor air to about 42°C.

There are three main advantages of this new system:

1) High chilled water temperature of chiller. Performance of chiller and thermal comfort of terminals are improved.
2) Outdoor air precooled by chilled water from chiller. Precooling process can reduce the cooling load of the dedicated refrigeration cycle, which means less energy consumption, smaller compressor and smaller outdoor unit of OA subsystem.
3) In heat pump mode, outdoor air can be effectively heated without dedicated heat pump cycle for OA subsystem.

3. System modeling

In order to analyze the performance of new system, a simulation model is developed using GREATLAB[15, 16]. GREATLAB is a general simulation software of refrigeration and air-conditioning system. The main methods of component and system simulation are briefly introduced below. More details about this model could be found in the references[15, 16].

Fin-tube coil is the main type of heat exchangers in this system. A tube-in-tube incremental model, which can account refrigerant and air distribution of the coil, is applied for fin-tube coil modeling. This model includes four levels: element level, tube level, circuit level and heat exchanger level. And the model is solved from inside to outside. First the equations at element level are solved. Then every tube is calculated by results at element level, and every circuit is calculated by results at tube level. Finally, all circuits are calculated and the mass flow rate is adjusted to balance pressure drop between circuits.

The equations at element level are as follows.

Refrigerant energy equation.

$$Q_{\mathrm{r}} = m_{\mathrm{r,in}}\left(h_{\mathrm{r,in}} - h_{\mathrm{r,out}}\right) \quad (1)$$

Refrigerant mass conservation equation:

$$m_{\mathrm{r,in}} = m_{\mathrm{r,out}} \quad (2)$$

Refrigerant momentum equation.

$$p_{\mathrm{r,in}} - p_{\mathrm{r,out}} = G^2\left(v_{\mathrm{r,out}} - v_{\mathrm{r,in}}\right) + f\frac{G^2 v_{\mathrm{m}}}{2D} \quad (3)$$

Air energy equation.

$$Q_{\mathrm{a}} = m_{\mathrm{a}}\left(h_{\mathrm{a,in}} - h_{\mathrm{a,out}}\right) \quad (4)$$

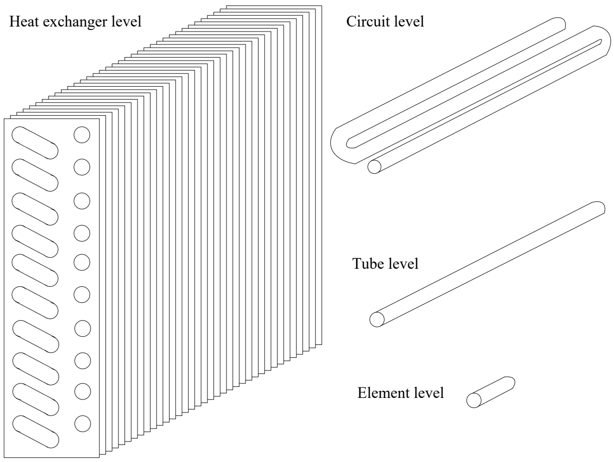

Fig. 2. Structure of the finned-tube heat exchanger model.

Where,

$$Q_{\text{a,sens}} = \frac{A_{\text{s}}}{1/(k_{\text{a}}\eta_{\text{s}}) + (R_{\text{m}} + R_{\text{fr}}) + (A_{\text{s}}/A_{\text{r}})/k_{\text{a}}}(T_{\text{a}} - T_{\text{r}}) \tag{5}$$

$$Q_{\text{a,latent}} = m_{\text{a,d}}\left(W_{\text{a,in}} - W_{\text{a,out}}\right)h_{\text{latent}} \tag{6}$$

Air humidity equation

$$-m_{\text{a}}\,\mathrm{d}W_{\text{a}} = k_{\text{d}}\left(W_{\text{a}} - W_{\text{w}}\right)\mathrm{d}A_{\text{a}} \tag{7}$$

where,

$$k_{\text{d}} = \frac{k_{\text{a}}}{Le \cdot c_{\text{p,a}}} \tag{8}$$

The correlations chosen for heat transfer coefficient and pressure drop are shown in 오류! 참조 원본을 찾을 수 없습니다..

Table 1 Correlations of heat transfer coefficients and friction factors

Air	**Heat transfer**	**Wang et al.[17]**
	Pressure drop	**Wang et al.[17]**
Refrigerant single phase	Heat transfer	Gnielinski[18]
	Pressure drop	Blasius
Refrigerant evaporation	Heat transfer	Gungor and Winterton[19]
	Pressure drop	Choi et al.[20]
Refrigerant condensation	Heat transfer	Cavallini et al.[21]
	Pressure drop	Choi et al.[20]

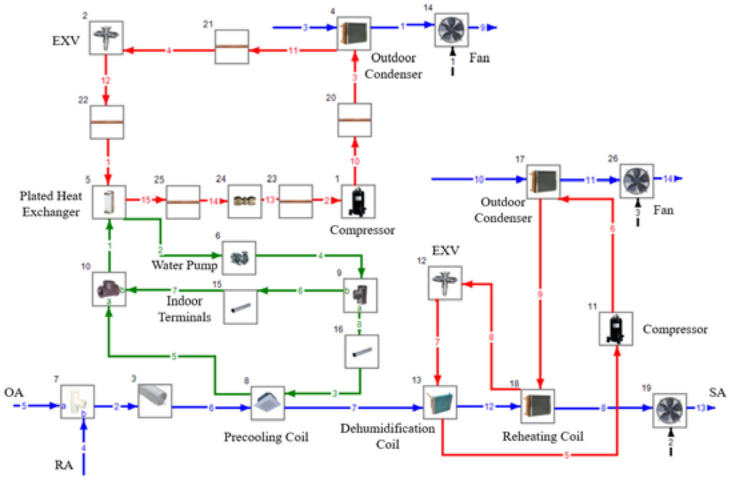

Fig. 3. System model of the new system in GREATLAB (cooling season).

Since the two compressors in this system are both variable-speed compressor, a multi-variable polynomial model[15] is applied for compressor simulation.

$$y = c_1 + c_2T_e + c_3T_c + c_4f_r + c_5T_e^2 + c_6T_c^2 + c_7f_r^2 + c_8T_eT_c + c_9T_ef_r + c_{10}T_cf_r + c_{11}T_e^2f_r + c_{12}T_ef_r^2 + c_{13}T_c^2f_r + c_{14}T_cf_r^2 + c_{15}T_eT_cf_r \quad (9)$$

where y can represent mass flow rate or power consumption of compressor. T_e and T_c represent evaporating temperature and condensing temperature, respectively. f_r represents compressor frequency. Coefficients c_1-c_{10} are curve-fitted according to compressor manufacturer's data.

Performance of expansion valves and fans are calculated according to the performance curves from manufacturers.

Based on all these component models, simulation model of the new system can be set up in GREATLAB, as shown in Fig. 3. System solver of this model is based on the Newton-Raphson algorithm. All fluid properties are calculated by REFPROP 9.0[22]. With the simulation model, we can analyze the performance of the new system under variable operating conditions. And the advantages of this new system will be discussed in detail in the next section.

4. System performance analysis and discussions

In this section, performance of the new system is numerically analyzed under different operating conditions. Firstly, a suitable baseline and some typical operating conditions are chosen in subsection 4.1. Then, an annual performance factor method is developed to analyze the annual performance of the new system in subsection 4.2. Finally, the results are presented and discussed in subsection 4.3.

4.1. Baseline and typical operating conditions.

To analyze performance of the new system, a comparison between the new system and conventional residential air-conditioning solution is necessary. In this study, a combination of chiller system and basic DOAS is chosen as the baseline of comparison and analysis. Schematic of baseline system is shown in Fig. 4.

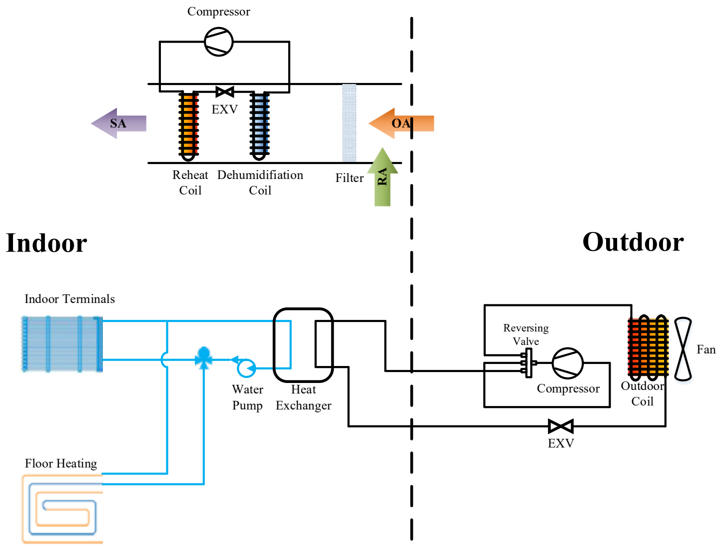

Fig. 4. Schematic of baseline system.

The chiller in both baseline system and improved new system is a practical chiller prototype, which has about 15kW design cooling capacity and is designed for about 150m^2 room. Some important design information of this chiller is shown in **오류! 참조 원본을 찾을 수 없습니다.**. And following the modeling method introduced in section 2, the simulation model of chiller was set up and well validated by experimental data. The comparison of some key performance parameters between simulation data and experimental data is presented in **오류! 참조 원본을 찾을 수 없습니다.**.

Table 2 Specifications of chiller

Name	Specification	Name	Specification
Refrigerant	R410A	Condenser	Fin-tube coil, 2 rows, 108 tubes, Tube length: 1000mm.
Design Cooling Capacity	15kW	Evaporator	Plated heat exchanger
Compressor	Rotary, variable-speed	Fan	Double axial fans, variable speed
Expansion device	EXV		

Table 3 Validation of chiller simulation model

Operating conditions	Cooling(heating) capacity/kW			Power/kW			Mass flow rate (water)/L·s^{-1}		
	Simulation	Test	Error	Simulation	Test	Error	Simulation	Test	Error
Cooling 100% load	15.11	14.93	**1.2%**	5.21	5.20	**0.2%**	0.69	0.68	**1.5%**
Cooling 75% load	11.20	11.29	**0.8%**	3.02	2.90	**4.1%**	0.68	0.68	**0.3%**
Cooling 50% load	8.02	7.58	**5.7%**	1.88	1.89	**0.5%**	0.71	0.68	**4.4%**
Cooling 25% load	5.52	5.29	**4.2%**	1.25	1.19	**5.0%**	0.70	0.68	**2.9%**
Heating 100% load	17.12	17.68	**3.2%**	5.84	5.80	**0.7%**	0.69	0.70	**1.4%**

The DOAS in baseline system is also a typical type of residential DOAS, prototype of this DOAS is shown in Fig. 5. The key design information of this DOAS is shown in Fig. 6. In this study, considering the demand of fresh air supplying and dehumidification for about 150m^2 room, one chiller needs to match 3 DOAS in the baseline system.

Table 4 Specification of DOAS

Name	Specification	Name	Specification
Refrigerant	R407C	Evaporator	Fin-tube coil, 4 rows, 36 tubes, Tube length: 420mm.
Design Dehumidification Capacity	26L/Day	Condenser	Fin-tube coil, 4 rows, 36 tubes, Tube length: 420mm.
Rotary	Rotary, constant speed	Fan	Single centrifugal fans, constant speed
Expansion device	Capillary tube		

As a multi-function heat pump system, it is necessary to analyze its system performance under variable operating conditions in cooling season, heating season and transitional season. Therefore, a variable operating conditions analysis is necessary for the new system. In this section, some typical operating conditions are chosen according to the typical meteorological year (TMY) data of Shanghai, as shown in Fig. 6. The TMY data are from DeST[23].

Fig. 6 shows the chosen 7 typical cooling operating conditions and 4 typical heating operating conditions. Point A is the rated cooling operating conditions (T_{DB}=35°C, T_{WB}=28°C). Point B and C are two 40°C+ extreme cooling operating conditions. Point D-F are three low temperature cooling operating conditions. Point G represents dehumidification operating conditions in transitional season. Point H-J are three different heating operating conditions. And point K represents heating operating conditions in transitional season.

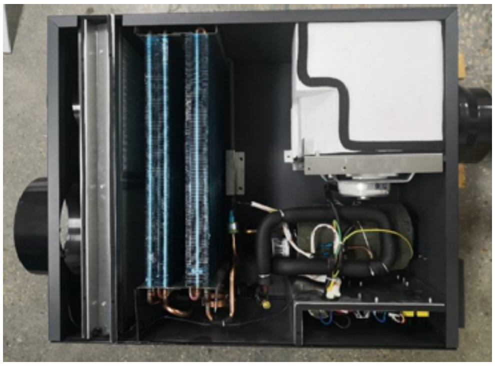

Fig. 5. Photo of DOAS in baseline system.

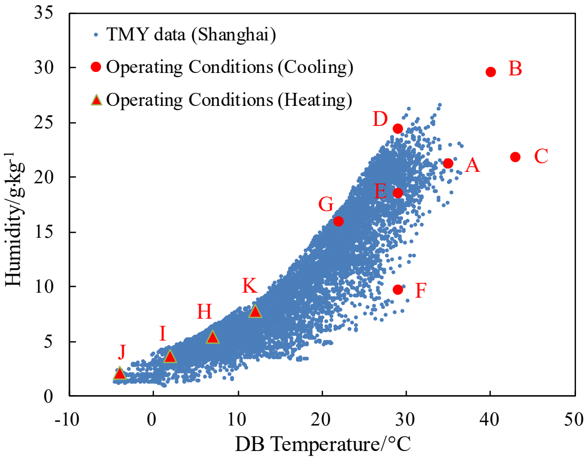

Fig. 6. Typical operating conditions in Shanghai.

4.2. Annual performance factor analysis

For further analysis, the annual performance factor (APF) of the new system is applied. In this study, APF is defined as:

$$\mathrm{APF} = \frac{Q_c + Q_h}{W_c + W_h} \tag{10}$$

where Q_c represents the total cooling capacity in cooling season and Q_h represents the total heating capacity in heating season. W_c and W_h represent the total power consumption in cooling season and heating season respectively.

APF is also calculated with TMY data of Shanghai, which is a typical region with hot moist summer and cold winter. In this work, APF is calculated as follows. Firstly, cooling season and heating season is divided into several temperature zones. The temperature and humidity interval of each zone is 1°C and 1g/kg respectively. Secondly the annual occurred time of each temperature zone is counted according to the TMY data. Then the energy consumption and cooling or heating capacity of system is calculated in every temperature zone, based on the simulation model. Finally, APF can be calculated in terms of the statistics of total energy consumption and total cooling or heating capacity within a year.

4.3. Results and discussions.

Based on the simulation model, performance of baseline system and the improved new system is calculated and compared with each other under variable operating conditions. The results in cooling season are shown in **오류! 참조 원본을 찾을 수 없습니다.**. Under rated operating conditions (A), COP of the new system is 3.94 and 46.5% higher than baseline system. Under the operating conditions B-F, the new system has about 23.7%-45% COP improvement compared to baseline system. Under the operating conditions G, which is a typical dehumidification operating conditions in transitional season, the new system has about 96.2% COP improvement compared to the baseline system. These results indicate better performance of the new system over the common combination of chiller and DOAS, especially under the operating conditions with large latent load.

Table 5 Performance comparison in cooling season

Operating Conditions	Outdoor Temperature (DB/WB)	Cooling Capacity /W	Latent Cooling Capacity /W	COP (Baseline)	COP (New System)	COP Improvement
A	35°C/28°C	14772	2275	2.69	3.94	**46.5%**
B	43°C/30°C	22162	2324	2.04	2.56	**25.5%**
C	40°C/33°C	20738	3654	2.15	3.01	**40.0%**
D	29°C/19°C	7525	522	3.76	4.65	**23.7%**
E	29°C/25°C	8799	1824	3.80	4.78	**25.8%**
F	29°C/28°C	9823	2848	3.31	4.80	**45.0%**
G	22°C/21°C	1898	1388	2.66	5.22	**96.2%**

Table 6 Contribution of different aspects to system performance improvement

System	Operating Conditions A		Operating Conditions F	
	COP	COP Improvement	COP	COP Improvement
Baseline	2.69	-	3.31	-
+ Outdoor Unit of DOAS	3.46	**28.6%**	3.81	**15.1%**
+ Precool coil	3.94	**46.5%**	4.80	**45.0%**

Performance improvement for the new system is mainly from two aspects: precooling process and heat rejecting to outdoor ambient. **오류! 참조 원본을 찾을 수 없습니다.** shows the contribution of different technology under operating conditions A and F. Both two aspects are important to the overall system performance improvement, and these two feature technology have different performance effect under different operating conditions. Under operating conditions A (T_{DB}=35°C, T_{WB}=28°C), adding outdoor unit of DOAS can bring 28.6% COP improvement, and adding precool coil can further increase COP improvement to 46.5%. But under operating conditions F (T_{DB}=29°C, T_{WB}=28°C), which is a low temperature dehumidification operating conditions, COP improvement from outdoor unit of DOAS is merely 15.1% while the overall COP improvement is nearly same to operating conditions A. The results indicate that precooling process is more effective for dehumidification operating conditions while heat rejecting is more important for those operating conditions with high cooling load.

Fig. 7 shows the detailed dehumidification process in OA subsystem under rated operating conditions according to the simulation results. It shows why the precool coil can reduce energy consumption of dehumidification. The precool coil, which handles about half cooling capacity, works with high temperature chilled water. Namely, half of cooling load for dehumidification process can be taken away in a more efficient way. And with this stepped cooling and dehumidification process, the uniformity of temperature difference in the whole duct is improved. Therefore, the overall COP of this system is higher than a conventional combined solution. Moreover, due to the cooling load of the dedicated cycle in OA subsystem is less than conventional DOAS, a smaller outdoor unit can be applied in OA subsystem of this multi-function system.

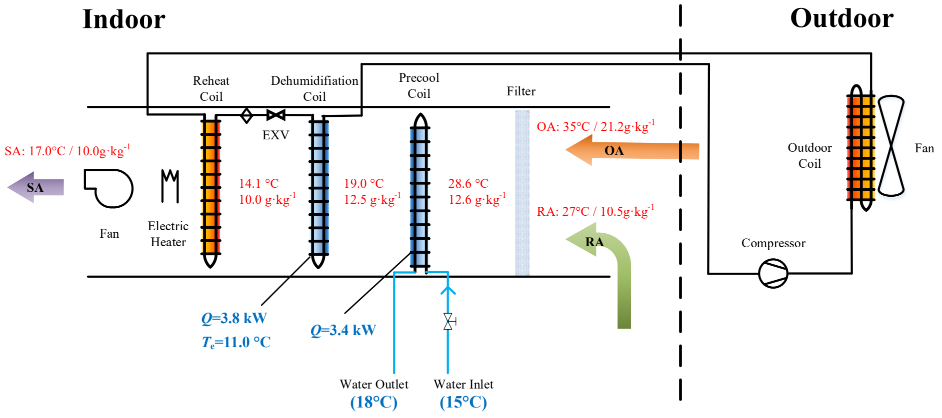

Fig. 7. Detailed schematic of dehumidification process in OA subsystem

The results in heating season are shown in 오류! 참조 원본을 찾을 수 없습니다.. In heat pump mode, compressor of OA subsystem is turned off in both baseline system and the new system. Since the improvement of the new system is mainly for cooling and dehumidification process, COP of the new system has little improvement compared to baseline system.

Fig. 8 shows the supply air temperature of OA subsystem under different heating operating conditions. This system can effectively heat outdoor air without dedicated heat pump cycle for OA subsystem. And the supply air temperature of OA subsystem can keep steadily under different operating conditions. In heat pump mode, the compressor of dedicated cycle in OA subsystem is turned off, only the water coil still works in OA subsystem. Even so, with the 45°C hot water supplied from chiller, the outdoor air can also be stably heated to about 42°C without mixing return air. And the indoor thermal comfort can be easily ensured under different operating conditions in winter.

Table 7 Performance comparison in heating season

Operating Conditions	**Outdoor Temperature (DB/WB)**	**COP (Baseline)**	**COP (New System)**	**COP Improvement**
H	7°C/6°C	2.82	2.76	**-2.1%**
I	2°C/1°C	2.06	2.06	**0.0%**
J	-4°C/-5°C	1.87	1.85	**-0.1%**
K	12°C/11°C	3.21	3.15	**-1.9%**

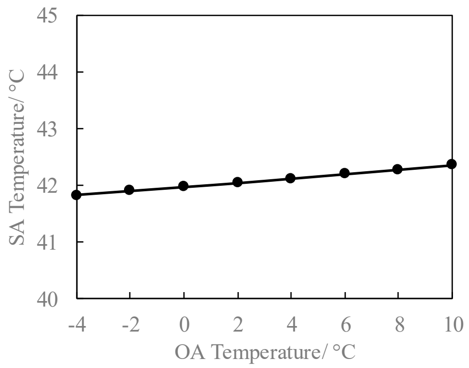

Fig. 8. Supply air temperature under different operating conditions in heating season

Table 8 APF of different system

Baseline	New System	APF Improvement
2.84	3.32	**16.9%**

Results of APF analysis are shown in **오류! 참조 원본을 찾을 수 없습니다.**. According to the simulation results, APF of the new system is 3.32 and 16.9% higher than baseline under the meteorological conditions of Shanghai. The results indicate that the new system has better performance and can reduce considerable energy consumption in annual operation especially in hot moist summer and cold winter region.

5. Conclusions

In this study, a novel multi-function residential heat pump system for cooling, heating, dehumidification and outdoor air supply is proposed. This system is composed by two subsystems: chiller subsystem and OA subsystem. In cooling season, the chiller subsystem only deals with sensible load and supply 15°C high temperature chilled water to sensible heat terminals. The OA subsystem has a dedicated refrigeration cycle and deals with overall latent load and partial sensible load. In order to decreasing the energy consumption of dehumidification and improve the performance of dehumidification, a precool water coil is added to precool outdoor air by high temperature chilled water. In heat pump mode, the outdoor air can be heated to over 40°C only by the water coil in OA subsystem.

A simulation model is developed and performance of this system is numerically analyzed based on modeling results. The overall COP of the new system can reach 3.94 under rated operating conditions, which is 46.5% higher than a conventional residential air-conditioning solution. And in terms of annual performance, APF of the new system is 3.32 and 16.9% higher than the conventional solution, under the meteorological conditions of Shanghai. Since the structure of this new system is complex and interactive, performance analysis in this work mainly focuses on the overall performance of whole system. In order to research the specific sources of energy conservation in this system, some other suitable analysis methods, such as uniformity analysis or second law analysis, are still worth to be applied for further analysis in future works.

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