

A New Dedicated Outdoor Air System with Exhaust Air Heat Recovery

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Abstract

DOAS (dedicated outdoor air system) has drawn much attention recently. However, it is quite energy-intensive resulted from the big temperature and/or humidity difference between the outdoor air and supply air. This paper proposed a new DOAS with exhaust air heat recovery. In cooling mode, there are one evaporator and three condensers in parallel. The outdoor air is cooled and dehumidified by the evaporator. Besides the outdoor condenser, another two are used for supply air reheat and exhaust air heat recovery. In heating mode, the reheat coil is off and two evaporators work for exhaust air heat recovery and ambient heat absorption, respectively. A pressure regulating valve equipped on the indoor evaporator keeps proper evaporating temperature for a frost free operation. A validated model is applied to fulfill the system design and performance prediction. The simulation results showed that the system cooling COP reaches 3.3 at ambient temperature 35°C/28°C and the heating COP achieves 4.8 at ambient temperature 7°C/6°C, respectively. In winter the proposed system can maintain the desired supply temperature and keep both evaporators frost free (ambient temperature -20°C to 10°C). The proposed system can efficiently recover waste energy from exhaust air in the whole year and has the extensive applicability, especially in the cold climate where considerable energy savings can be found.

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Key words: dedicated outdoor air system; heat pump; exhaust air heat recovery; frost free; numerical simulation

1. Introduction

DOAS (dedicated outdoor air system) has got wide researches and applications [1-3] since Mumma first proposed the concept of DOAS [4]. For a typical DOAS, outdoor air is separately conditioned without mix-up with space return air (RA) and is responsible for the total latent load and partial sensible load [5]. DOAS combined with the device handling sensible heat load (e.g. chilled beams, radiant cooling systems [6]) can achieve higher indoor air quality compared with the traditional air-conditioning systems.

However, DOAS is quite energy-intensive resulted from the big temperature and/or humidity difference between the outdoor air (OA) and supply air (SA). According to open literatures, related researches mostly focused on the dehumidification process in cooling mode, which mainly consists of desiccant dehumidification and vapor-compression systems [7]. Desiccants can be either solid or liquid and have better dehumidification

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performance than vapor-compression systems. But their applications got restricted due to higher cost and larger area occupation [8]. Besides, desiccants need additional heat source to regenerate, which is inconvenient in urban areas [9]. Therefore, the relatively inexpensive, convenient and small space occupied vapor-compression systems are widely used, even though their system COP (coefficient of performance) is relatively lower. There are two major types: chilled-water based system and direct-expansion (DX) system. Since the DX system is less expensive, less complex and more energy efficient than the chilled water based system [10], the DX dehumidification systems are widely used in both residential and commercial buildings.

In cooling mode, the evaporating temperature of a common DX DOAS has to be lower than the supply air dew point temperature to satisfy with the desired space humidity, which usually results in poor system efficiency. Besides, extra energy consumption on the supply air reheating makes the system COP even worse. Many researchers have made efforts to improve efficiency of conventional DX systems. Liang proposed a DX system in combination with a membrane-based total heat exchanger [9], whose COP is one time higher than a conventional system. Another DX system proposed by Liang using an auxiliary condenser to reheat supply air was claimed to have a COP of 6.8 under nominal operating conditions [11]. Li proposed a DOAS using multi-stage dehumidifying coils and a heat pipe to recover exhaust energy. It was revealed that this system could reduce annual energy use by 13.6% compared with a conventional system [12]. Some other energy efficient methods applied in desiccant and water-based dehumidification systems were also proved energy effective. Ham proposed a liquid desiccant system characterized by a membrane enthalpy exchanger, an indirect evaporative cooler, and a sensible heat exchanger, could save 12% of primary energy compared to a conventional system [13]. An experimental study about a small capacity DOAS with sensible and latent energy recovery wheels, revealed that 30% of the total cooling load could be provided by the energy recovery wheel [14]. We could summarize that most methods to improve system performance is to employ heat recovery equipment, such as enthalpy wheels, membrane based heat exchangers, run-around coils [15] and heat pipes. However, some of the methods above have unavoidable drawbacks. For example, wheels have cross contamination issues between exhaust air (EA) and supply air, flat-plate heat exchangers and run-around coils are faced with freezing problem in winter [16]. Therefore, a DOAS with good performance in terms of dehumidification, and able to avoid those problems mentioned above, is definitely desired by the market.

When we turn to heating mode in winter, the DX DOAS is changed into a heat pump then. Low temperature outdoor air is heated to the desired supply temperature. Partial or the entire space heating load is handled by the DOAS. However, low ambient temperature will cause a lower evaporating temperature and easily get the outdoor heat exchanger frosted. Thus it is difficult in stably providing desired volume of fresh air or desired air supply temperature. This tricky problem hasn't got properly addressed to the authors' view. Most frost-related researches focused on the defrosting methods for an ordinary air source heat pump (ASHP), in which the condenser inlet air consists of mostly return air and a smaller fraction of outdoor air. Therefore, the ASHP absorbs less heat from ambient environment and has a relatively higher evaporating temperature thus its frosting risk is lower than a DOAS. For the ASHP, it was proved that widely used reverse cycle defrosting method will cause at least 15% capacity degrade [17] while hot gas bypass method will lead to much longer defrosting time [18]. Recently some efficient defrosting methods was also put forward, such as exhaust and outdoor air mixing methods [19], auxiliary electric heaters before and after outdoor evaporator tubes [20], and solid dehumidification combined with storage energy regeneration methods [21]. Useful as those methods are, a defrosting method or frost free system specifically aimed at DOAS remains to be developed all the same.

Based on those issues to be addressed, in this work we propose a new dedicated outdoor air system with exhaust air heat recovery. An exhaust air heat pump (EAHP) is employed to recover exhaust air energy. It has been proved to have shorter payback period compared with other exhaust air heat recovery methods [22]. Besides, in winter EAHP could delay even avoid frost issue which frequently happens on other heat recovery equipment. Considered that the EAHP alone isn't able to deal with space humid load in summer and heating load in winter, an outdoor heat exchanger is still adopted as heat sink or heat source. In cooling mode, there are one evaporator and three condensers in parallel. The outdoor air is cooled and dehumidified by the evaporator. Besides the outdoor condenser, another two are used for supply air reheat and exhaust air heat recovery, respectively. In heating mode, the reheat coil is off and two evaporators work for exhaust air heat recovery and ambient heat absorption, respectively. Note that the two heat sources here are at different temperatures, thus a pressure regulating valve is equipped on the indoor evaporator to keep proper evaporating temperature for a frost

free operation. In this way, the new dedicated outdoor air system can operate in an efficient way and mitigate frost issues as well.

In terms of the proposed new DOAS, we developed the system simulation model using in-house developed simulation software GREATLAB [23] and well designed the system. The simulation model was validated with experimental data. Moreover, we analyzed the system performance under design and off-design conditions and optimized control strategies based on the validated model.

2. System description and modeling

In this section, we introduce the working principle for the new DOAS both in cooling and heating modes, and develop its simulation model using GREATLAB [23].

2.1. System description

Fig. 1 is the schematic of the new DOAS in cooling mode. The treated outdoor air is cooled by the evaporator to a nearly saturated state to achieve the desired absolute humidity. It is then reheated to a comfort supply temperature by the reheating coil. For the refrigerant circuit, compared with a simple vapor-compression cycle, it is characterized by two auxiliary condensers working in parallel with the outdoor condenser, which are the exhaust air coil and the reheating coil, to recover exhaust air and condensing energy, respectively. The refrigeration system benefits from low inlet air temperature of the two condensers (return air at 27°C and evaporator outlet air temperature at 12°C), therefore, better system performance can be obtained from lower condensing temperature.

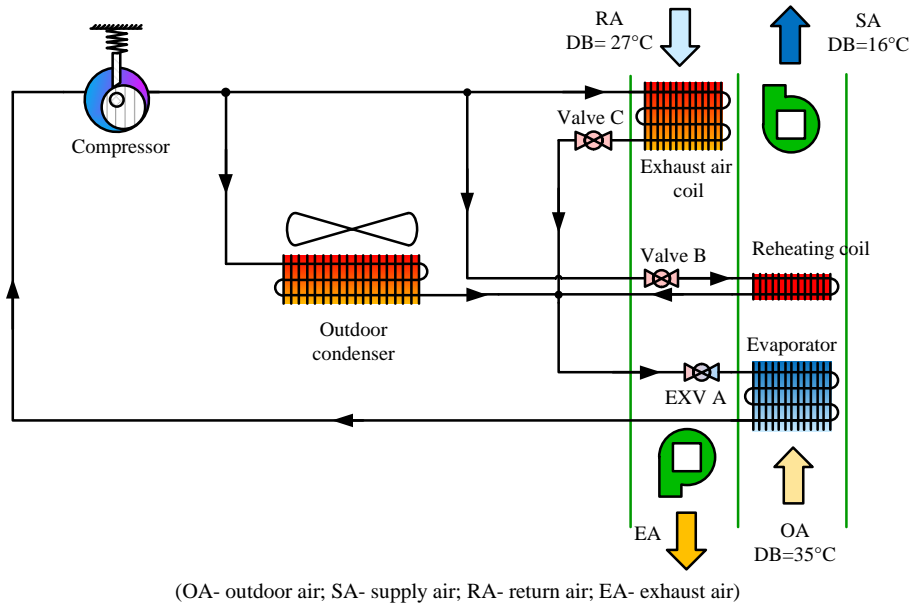


Fig. 1. Schematic of the new DOAS in cooling mode

In terms of the system control strategy, the suction superheat is controlled by EXV (electronic expansion valve) A (refer to Fig. 1). Meanwhile, the desired absolute humidity of supply air can be controlled by the variable-speed compressor and the desired temperature of supply air (reheating coil outlet) can be controlled via Valve B (Fig. 1) by regulating the refrigeration mass flow entering the reheating coil. As for the mass flow allocation between the outdoor condenser and exhaust air coil, Valve C (Fig. 1) is used to tune the mass flow fraction so as to obtain the optimum system performance.

In heating mode, the new DOAS is changed into a heat pump through a reserving valve. As shown in Fig. 2,

the treated outdoor air is heated by the condenser, while outdoor and exhaust coil operate as evaporators to absorb heat from ambient environment and exhaust air. The outlet superheat of each evaporator is independently controlled by an EXV (EXV D and EXV C referred to Fig. 2). The desired temperature of supply air can be controlled by the variable-speed compressor.

Here we introduce the detailed frost free operation method in heating mode. According to Chinese standard [24], when the ambient temperature is higher than -8°C in winter (low ambient temperature conditions), the outdoor evaporator is easy to frost due to relatively higher humidity ratio. Under this condition we operate indoor evaporator (exhaust coil) only to recover both sensible and latent heat from exhaust air, and further simulation results will prove that desired supply air temperature is available in this way. When the indoor exhaust air coil shows insufficient ability to guarantee a comfort supply temperature, in other words, there is slight frost on the indoor coil surface at extra low temperature conditions, we turn on EXV D (Fig. 2) thus outdoor evaporator also works to absorb heat from the ambient environment. Considered that outdoor air humidity ratio is very low under this condition, the outdoor evaporator hardly gets frost [25]. On the other side, the pressure regulating valve (Valve D) equipped on the exhaust air coil outlet line aims to keep its evaporating temperature at the set value to ensure a frost free operation. Therefore, the two evaporators with two different evaporating temperatures can realize the frost free operation under most winter conditions.

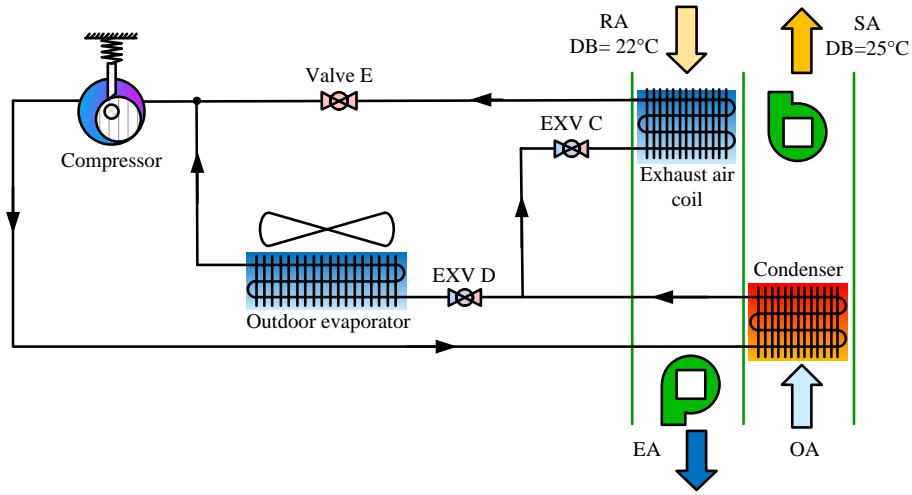


Fig. 2. Schematic of the new DOAS in heating mode

2.2. Modeling

The proposed DOAS was applied in a residential house in Shanghai, according to design requirements and engineering practice, we developed the system simulation model using GREATLAB [23] and fulfilled the optimum system design.

GREATLAB is in-house developed refrigeration and air-conditioning system modeling and analysis software. Incremental tube-by-tube model is adopted for finned-tube evaporator and condenser. Details of the model can be found from [26].

An AHRI like compressor polynomial model [27] is used to simulate the performance of variable-speed compressor. Namely,

$$y = c_1 + c_2 T_e + c_3 T_c + c_4 f_r + c_5 T_e^2 + c_6 T_c^2 + c_7 f_r^2 + c_8 T_e T_c + c_9 T_e f_r + c_{10} T_c f_r + c_{11} T_e^2 f_r + c_{12} T_e f_r^2 + c_{13} T_c^2 f_r + c_{14} T_c f_r^2 + c_{15} T_e T_c f_r \quad (1)$$

Name	Specification	Simulation model type
Refrigerant	R410A	REFPROP 9.0 [28]
Compressor	Rotary, variable-speed	Curve-fitting model
Expansion device	Electronic expansion valves (EXVs)	Manufacturer curve
Outdoor condenser	Finned-tube; three rows; 800mm□57.15mm□528mm	Incremental tube-by-tube model
Exhaust air coil	Finned-tube; eight rows; 200mm□152.4mm□286mm	
Evaporator	Finned-tube; ten rows; 250mm□190.5mm□396mm	
Reheating coil	Finned-tube; one row; 250mm□19.05mm□396mm	
Fan	Two centrifugal fans for EA and OA, variable speed One axial fan for outdoor condenser, variable speed	Manufacturer curve

3. Model validation

In order to validate the prediction accuracy and reliability of the simulation model, we first made a prototype according to the optimum design (Table 2) and tested it in a standard psychrometric room. Then we would give the comparison between the experimental and simulation data.

3.1. Prototype and experimental setup

The prototype keeps in consistent with the optimum design obtained via numerical model as mentioned above. Fig. 4 shows the indoor and outdoor parts of the prototype. The prototype was tested in a standard psychrometric room including one indoor chamber and one outdoor chamber, where the integrated air-conditioning and control system allow managing the testing conditions as shown in Table 1. In order to acquire the operating performance of the experimental system and run the system at desired conditions, the tested prototype was instrumented with several temperature and pressure sensors, power meters and air flow meters. The measurement uncertainties of these sensors are listed in Table 3. Corresponding measured parameters are: refrigeration temperatures and pressures of suction, discharge and liquid lines, temperatures of exhaust air, supply air and outdoor condenser leaving air, air flow rate of exhaust air, supply air, power consumption of fans and the compressor.

Table 3. Uncertainties of measured parameters

Measured parameter	Uncertainty
Dry bulb temperature	±0.1 °C
Wet bulb temperature	±0.5 °C
Refrigerant pressure	±0.25 %
Refrigerant temperature	±0.5 °C
Power consumption	±0.5 %
Air flow rate	±0.5 %

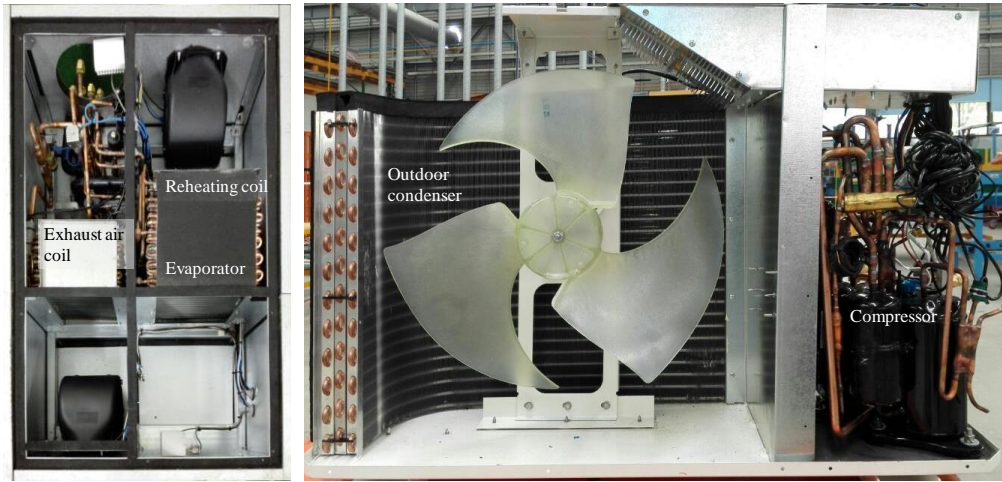


Fig. 4. Components of the prototype

3.2. Model validation

Now that we have obtained the test data at design conditions via experimental investigations in the last section, we would compare it with the simulation results to validate the prediction accuracy of the model. The simulation results show that the system cooling and heating COP is 3.3 and 4.8 at design conditions. Comparisons of some key performance parameters are shown in Table 4, which indicates good agreement between the simulation and test data. Therefore, subsequent system performance based on the simulation model is accurate and reliable.

Table 4. comparisons of simulation and test data

Parameters	Cooling design condition			Heating design condition		
	Simulation	Test	Error	Simulation	Test	Error
Cooling (heating) capacity / kW	7.29	7.18	+1.5%	3.25	3.30	-1.5%
Total power / kW	2.22	2.24	-0.9%	0.68	0.70	-2.8%
COP	3.28	3.21	+2.1%	4.78	4.71	+1.4%
Evaporating temperature / °C	7.9	8.2	-0.3 °C	5.5	5.9	-0.4 °C
Condensing temperature / °C	46.1	47.9	-1.8 °C	30.4	29.9	+0.5 °C

Note: the reheating coil wasn't switched on in cooling test.

4. System performance analysis

4.1. Cooling mode

We calculated the system cooling performance at different outdoor dry-bulb temperatures but the same outdoor wet-bulb temperature 28°C using the validated simulation model. As shown in Fig. 5, system cooling COP drops from 3.9 to 3.1 when the outdoor dry-bulb temperature rises from 30°C to 38°C. Meanwhile, the condensing temperature goes up with the outdoor dry-bulb temperature rise, but the evaporating temperature remains almost constant. The test results are in line with the fact that the evaporator performance under wet condition is largely determined by the entering air wet-bulb temperature not the dry-bulb temperature. Even so, the rise of outdoor dry-bulb temperature increases the condensing temperature via the outdoor coil, which leads to higher compressor work and lower system COP.

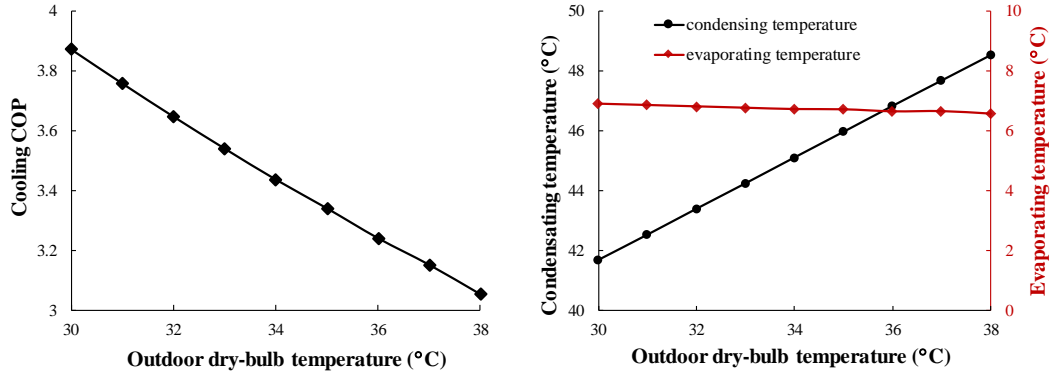


Fig. 5. System cooling performance versus outdoor dry-bulb temperature

In addition, we found there is an optimum refrigeration mass flow allocation between the outdoor condenser and the exhaust air coil, which can be tuned by Valve C (Fig. 1). This results from two heat sinks (return air and outdoor air) here in different grade and two condensers in different size. Note that results show in Fig. 5, Fig. 7 and Fig. 8 are all adjusted to the optimum point.

4.2. Heating mode

As mentioned above, in this part we are going to illustrate whether the DOAS (or heat pump) can maintain the desired supply temperature with the exhaust air coil alone while keeping it frost free at low ambient temperature, and how the pressure regulating valve works at extra-low ambient temperature.

Under low ambient temperature conditions, the outdoor evaporator is shut off and the exhaust air coil works alone. When the outdoor dry-bulb temperature varies from -8°C to 7°C , we calculate the tube outer wall temperature of the exhaust air coil and corresponding compressor frequency at each condition provided that the supply air temperature maintains at desired value 28°C . As shown in Fig. 6, the tube wall temperature is higher than 0°C in the ambient temperature range from -8°C to 7°C . Therefore, condensate from the indoor evaporator won't get frost at low ambient temperature conditions. Accordingly, the compressor running frequency is from 51Hz to 23Hz, relative to the compressor working envelope (20Hz to 120Hz).

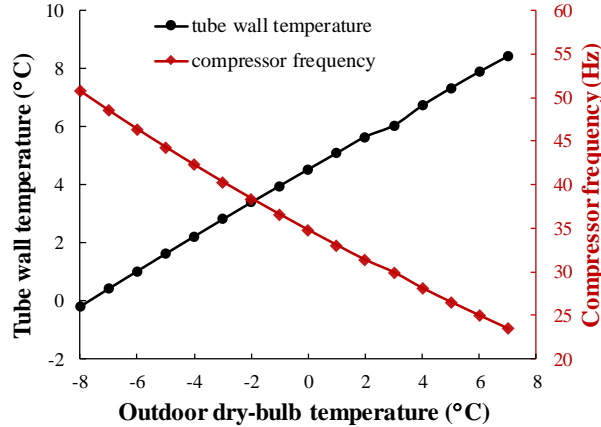


Fig. 6. Tube wall temperature and compressor frequency versus outdoor dry-bulb temperature at low ambient temperatures

Under extra-low ambient temperature conditions, both the outdoor evaporator and the exhaust air coil work together to absorb heat from ambient environment and exhaust air, respectively. Simulation results for outdoor dry bulb temperature from -20°C to -9°C are shown in Fig. 7. We can notice that the indoor evaporator (exhaust air coil) tube wall temperature maintains constant and is not below 0°C under all the conditions. This is because the pressure regulating valve equipped on the exhaust air coil outlet line controls its evaporating temperature at -1°C and the recovered heat is thus almost a certain value. Otherwise, the two evaporator outlet lines are both

directly connected to the compressor suction line. In that way, the only evaporating temperature is pulled down by the extra-low ambient temperature, which can result severe frost on the exhaust air coil. On the other hand, at the extremely low outdoor temperatures, the outdoor air humidity ratio is too low (maximum value 1.5 g/kg-air at -9°C) to frost. Therefore, the proposed DOAS is able to maintain the desired supply temperature while keeping both the outdoor and indoor evaporators frost free at extra-low ambient conditions.

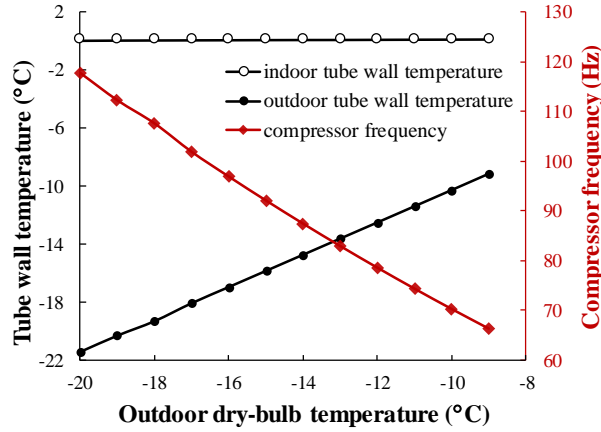


Fig. 7. Tube wall temperatures and compressor frequency versus outdoor dry-bulb temperature at extra-low ambient temperatures

Fig. 8 shows the net heating COP versus the outdoor dry bulb temperature in full range of winter conditions (-20°C to 10°C). Overall, the proposed system is energy efficient in winter operation, even when outdoor temperature drops to -20°C , the heating COP reaches 2.7. This owes to the efficient heat recovery capacity of the new DOAS in the whole winter. The lower limit working temperature of the DOAS is -20°C subject to the compressor frequency and minimum suction pressure, which shows an extensive applicability of the new DOAS. In addition, the sudden drop of heating COP when turning to extra-low ambient temperature conditions (Fig. 8) results from the added work of outdoor fan when the outdoor evaporator turns on. The heating COP trend, which increases at first and then decreases after a peak, is caused by the increased fraction of fan work in the total work.

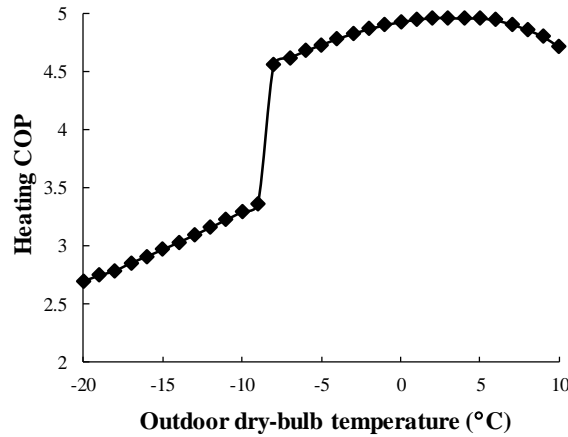


Fig. 8. Heating COP in full range of winter conditions

5. Conclusions

In this study, a new dedicated outdoor air system with exhaust air heat recovery was proposed. It includes two significant energy-saving features. An exhaust air coil in parallel with the ordinary outdoor condenser (evaporator) is used to recover energy from exhaust air in the whole year. A frost free operation method based on a pressure regulating valve equipped on the indoor evaporator can increase the system heating COP and extend

its application range.

In terms of the proposed DOAS, we developed the system simulation model using in-house developed simulation software GREATLAB and fulfilled the optimum system design. The simulation model was validated with experimental data.

The simulation results showed that the system cooling COP reached 3.3 at ambient temperature 35°C/28°C and the heating COP achieved 4.8 at 7°C/6°C, respectively. The system can maintain the desired supply temperature while keeping both the outdoor evaporator and exhaust air coil frost free in heating mode (ambient temperature -20°C to 10°C). The proposed system can efficiently recover waste energy from exhaust air in the whole year and has the extensive applicability, especially in the cold climate where considerable energy savings can be found.

Acknowledgements

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