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Research Paper

Integrated system of exhaust air heat pump and advanced air distribution for energy-efficient provision of outdoor air

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ABSTRACT

A large outdoor air supply is required to control the airborne infection risk of respiratory diseases (e.g., COVID-19) but causes a high energy penalty. This study proposes a novel integrated system of the exhaust air heat pump and advanced air distribution to energy-efficiently provide outdoor air. The system energy performances are evaluated by the experimentally validated thermodynamic model of heat pump and heat removal efficiency model of advanced air distribution. Results show the exhaust air heat pump with advanced air distribution can save energy because of three mechanisms. First, the exhaust air heat pump reuses the exhaust air to reduce the condensation temperature, thereby improving the coefficient of performance. Second, advanced air distribution reduces ventilation load. Third, advanced air distribution reduces the condensation temperature and enhances the evaporation temperature, thereby improving the coefficient of performance. The exhaust air heat pump saves energy by 18%, advanced air distribution saves energy by 36%, and the integrated system of the exhaust air heat pump and advanced air distribution can save energy by 45%. As a specific application, compared with the conventional system (i.e., the outdoor air heat pump with mixing ventilation), the exhaust air heat pump with stratum ventilation saves energy by 21% – 35% under various outdoor air ratios and outdoor air temperatures. The proposed integrated system of the exhaust air heat pump and advanced air distribution contributes to the development of low-carbon and healthy buildings.

1. Introduction

Infectious respiratory diseases (e.g., COVID-19) pose large threats to human health [1]. The aerosols generated by breathing, speech, and coughing can linger in the air for up to nine hours [2]. The virus exhaled by the infectors of respiratory diseases with an aerodynamic diameter up to 80 – 120 nm, e.g., the virus of coronavirus families [3], can be carried by the aerosols [4]. The virus stays viable in the aerosols for more than 3 h [5]. Due to the inhalation of the aerosols with the infectious virus [6], airborne transmission is one of the main transmission routines of infectious respiratory diseases [7]. More and more evidence suggests that airborne transmission is one of the main transmission routes of COVID-19 [8]. Infectious virus of COVID-19 has been detected in respiratory aerosol samples [9]. Mainly due to the airborne transmission, super-spreading events of COVID-19 have been reported, e.g., the 2-hour bus journey in Ningbo, China with 23 of 68 passengers infected [10] and the 2.5-hours Skagit Valley Chorale choir practice with 53 of 61 attendees infected [11]. To control airborne infection risk of respiratory diseases, ventilation is required to be properly designed and operated by different regulatory guidelines [12].

Large outdoor air supply and advanced air distribution are two main aspects of ventilation for airborne infection risk control [13]. Ventilation replaces the indoor polluted air with outdoor air to dilute the contaminant concentration in indoor air [14]. Ding et al. [2] found that increasing the outdoor air supply from 1 ACH to 9 ACH reduced the half-life of the aerosols from 28 – 40 min to 4 – 6 min. According to the Wells-Riley model, increasing the outdoor air supply effectively reduces the airborne infection risk, e.g., for COVID-19 [15]. Epidemiological evidence, e.g., the study on tuberculosis [16], also confirms the reduced airborne infection risk by the increasing outdoor air supply. To control airborne infection risk, the World Health Organization recommends increasing the outdoor air supply up to 160 L/s per person in infective wards and operating the outdoor air system at its maximal capacity for other buildings [17]. Simultaneously, air distribution of ventilation should be properly designed [18], otherwise, increasing the outdoor air supply...
supply might increase the contaminant concentration in the breathing zone because of poor air distribution [19], thereby increasing the airborne infection risk. Conventional air distribution (i.e., mixing ventilation) aims to uniformly distribute outdoor air in the indoor environment with a uniform contaminant concentration in indoor air [20]. Advanced air distribution aims to distribute outdoor air more in the breathing zone than other zones [21] with lower contaminant concentration. Different studies have confirmed the effectiveness of advanced air distribution, e.g., stratum ventilation [23], displacement ventilation [24], improved mixing ventilation [25], and attachment ventilation [26], in reducing airborne infection risk.

The large outdoor air supply increases energy consumption [27], which is a severe penalty when facing urban heat islands, climate change, and requirements for decarbonization [28]. The building cooling load consists of the space cooling load and the ventilation load [29]. The ventilation load is caused by the outdoor air supply, and increases with the increasing outdoor air supply, particularly at a high outdoor air temperature [30]. For the normal scenario without pandemics of infectious respiratory diseases, Hsieh et al. [31] calculated that to provide outdoor air at the minimal requirement of ASHRAE Standard 62.1 [32], the ventilation load accounted for up to 48% of the building cooling load. For airborne infection risk control of COVID-19, Aviv et al. [30] found that, compared with the normal scenario, the energy consumption increased by up to 215% due to the increasing outdoor air supply.

The exhaust air heat pump (EAHP) is an effective method of reducing the energy consumption of the outdoor air supply [33]. Different from the conventional outdoor air heat pump (OAHP) which uses outdoor air as the cold source to cool the condenser, the EAHP reuses the exhaust air from the indoor space as the cold source. Since the exhaust air temperature is lower than the outdoor air temperature, the EAHP reduces the condensation temperature thereby improving the coefficient of performance (COP). Hsieh et al. [31] experimentally demonstrated that for the outdoor air supply with a window-type air conditioner, compared with the OAHP, the EAHP could enhance the COP by 37.4%. The energy performance of the EAHP can even exceed that of the ground source heat pump [34]. Compared with other energy-efficient methods for the outdoor air supply (e.g., cooling the outdoor air by the earth-air heat exchanger and recovering the cooling energy of the exhaust air by the flat plate heat exchanger), the EAHP has the advantages of no cross-contamination between the exhaust air and the outdoor air supply [34], the capability of controlling the temperature and the relative humidity of the outdoor air supply [35], and simple system configuration with low installation cost [36]. Moreover, the EAHP has good compatibilities with other energy-efficient methods, e.g., the combination of the EAHP with indirect evaporative cooling [37], cross flow countercurrent air-to-air heat exchanger [38], heat pipe [39], and solar collector [40]. Due to these advantages of the EAHP, Schibuola and Tambani [41] recommended the EAHP to energy-efficiently provide outdoor air for airborne infection risk control of COVID-19.

This paper innovatively proposes an integrated system of the EAHP and advanced air distribution for energy-efficient provision of outdoor air. The integrated system has the potential to leverage the advantages of both EAHP and advanced air distribution to reduce the energy consumption of outdoor air supply. Conventional air distribution, i.e., mixing ventilation, targets distributing conditioned air uniformly in the entire indoor space [19]. However, not the entire indoor space is critical for thermal comfort and indoor air quality [18]. Advanced air distribution, e.g., stratum ventilation, displacement ventilation, and wall attachment ventilation, targets distributing conditioned air into the occupied zone for thermal comfort and the breathing zone for inhaled air quality, while leaving the other zone of the indoor space unconditioned/less conditioned for energy saving [21]. The EAHP belongs to the field of energy systems and the advanced air distribution belongs to the field of indoor environments. The novelty and contributions of this study are achieved by integrating the two fields. Particular attention should be given to the compatibility between the EAHP and advanced air distribution. Advanced air distribution enhances the supply air temperature [42] and the exhaust air temperature [43]. The enhanced supply air temperature benefits the compatibility between the EAHP and advanced air distribution since the enhanced supply air temperature increases the evaporation temperature of the EAHP thereby increasing the COP. In contrast, the enhanced exhaust air temperature negatively affects the compatibility between the EAHP and advanced air distribution since the 

### Nomenclature

- $A_c$: heat transfer area of condenser (m$^2$)
- $A_p$: heat transfer area of evaporator (m$^2$)
- $h_1$: enthalpy of refrigerant at state point 1 (kJ/kg)
- $h_2$: enthalpy of refrigerant at state point 2 (kJ/kg)
- $h_{2,\text{is}}$: enthalpy of refrigerant at state point 2 under ideal condition (kJ/kg)
- $h_{\text{e,in}}$: enthalpy of air entering evaporator (kJ/kg)
- $h_{\text{e,out}}$: enthalpy of air leaving evaporator (kJ/kg)
- $m_{\text{a,c}}$: flowrate of air entering condenser (kg/s)
- $m_{\text{a,e}}$: flowrate of air entering evaporator (kg/s)
- $m_{\text{f}}$: flowrate of refrigerant (kg/s)
- $n_f$: fan efficiency
- $n_{\text{e}}$: isentropic efficiency of compressor
- $n_{\text{mech}}$: mechanical efficiency of compressor
- $n_{\text{motor}}$: motor efficiency of compressor
- $N_{\text{TU}_c}$: number of heat transfer units of condenser
- $N_{\text{TU}_e}$: number of heat transfer units of evaporator
- $P_1$: evaporation pressure (Pa)
- $P_2$: condensation pressure (Pa)
- $Q_c$: heat released by condenser (kW)
- $Q_v$: space cooling load (kW)
- $Q_v$: ventilation load (kW)
- $Q_{\text{cl}}$: space cooling load (kW)
- $Q_a$: total cooling load (kW)
- $t_{\text{c,in}}$: temperature of air entering condenser (°C)
- $t_{\text{c,out}}$: temperature of air leaving condenser (°C)
- $t_{\text{e}}$: exhaust air temperature (°C)
- $t_{\text{e,in}}$: temperature of air entering evaporator (°C)
- $t_{\text{e,out}}$: temperature of air leaving evaporator (°C)
- $t_{\text{a,o}}$: outdoor air temperature (°C)
- $t_{\text{a,z}}$: air temperature of occupied zone (°C)
- $t_c$: condensation temperature (°C)
- $u$: constant
- $\rho_a$: density of air (kg/m$^3$)
- $\varepsilon_c$: effectiveness of condenser
- $\varepsilon_e$: effectiveness of evaporator
- $\gamma_0$: outdoor air ratio

### Abbreviation

- COP: coefficient of performance
- EAHP: exhaust air heat pump
- HRE: heat removal efficiency
- OAHP: outdoor air heat pump
enhanced exhaust air temperature increases the condensation temperature of the EAHP thereby decreasing the COP. The objective of the following context is to explain and demonstrate the effectiveness of the integrated system of the EAHP and advanced air distribution in reducing the energy consumption of outdoor air supply. This objective is achieved by compared with the existing OAHP and existing EHAP with conventional air distribution regarding energy performance for outdoor air provision. The energy performance of the OAHP and EAHAP is evaluated by an experimentally validated thermodynamic model and the advanced air distribution is represented by the experimentally validated heat removal efficiency model.

In summary, the research gap in existing studies is that although the EAHAP is a promising technology to provide outdoor air for healthy indoor environments, existing studies on EAHAP do not consider the effect of indoor air distribution, thus cannot leverage advanced air distribution to improve the energy efficiency of EAHAP for outdoor air provision. The novelty and contributions of this study are 1) to propose the integrated system of the EAHAP and advanced air distribution to energy-efficiently supply outdoor air for the first time, and 2) to justify and demonstrate the effectiveness of the integrated system of the EAHAP and advanced air distribution in improving the energy efficiency of outdoor air provision at different outdoor air ratios and outdoor air temperatures.

2. Methodology

Outdoor air provision is required by regulatory guidelines for airborne infection risk control of infectious respiratory diseases like COVID-19 [12]. When providing outdoor air is mandatory, handling processes of outdoor air including contaminant removal like particle matter and VOCs, dehumidification and cooling can be required before the outdoor air is supplied into the indoor environments. The cooling process generally consumes the most energy of outdoor air provision [30]. This study focuses on reducing the cooling energy consumption of outdoor air provision by proposing the integrated system of the EAHAP and advanced air distribution. The proposed integrated system can share the same contaminant removal and dehumidification methods as the existing OAHP and the existing EAHAP with conventional air distribution.

Thus, to supply the same flowrate of outdoor air, compared with the existing OAHP and the existing EAHAP with conventional air distribution, the proposed integrated system does not suffer from any additional energy penalty in outdoor air handling of contaminant removal and dehumidification. The following texts of this study focus on justifying and demonstrating the advantages of the proposed integrated system over the existing OAHP and the existing EAHAP with conventional air distribution regarding the energy performance of outdoor air provision.

2.1. Outdoor air heat pump and exhaust air heat pump

Figs. 1 and 2 show the OAHP and the EAHAP respectively. The two heat pumps consist of the same four main components, including the compressor, the condenser, the throttle valve, and the evaporator. The differences between the two heat pumps are their connections with the ventilation system. For the OAHP (Fig. 1), the outdoor air is used as the cold source to absorb the heat released by the condenser, and the mixing of the outdoor air and the returned air is used as the heat source which is cooled by the evaporator. The cooled air is supplied into the room for indoor air quality and thermal comfort. The exit air from the room is divided into parts: the returned air and the exhaust air. The OAHP releases the exhaust air to the outdoor directly without reusing it. The exhaust air is compensated by the outdoor air supply. The ratio between the outdoor air supply and the total air supply (i.e., the sum of the outdoor air supply and the returned air) is defined as the outdoor air ratio [19]. For the normal scenario without the pandemics of respiratory diseases like COVID-19, the outdoor air ratio is determined to maintain the indoor CO₂ concentration at the desired level, e.g., 800 ppm above the outdoor CO₂ concentration [44]. For the airborne infection risk control of respiratory diseases, the outdoor air ratio is elevated to satisfy higher requirements on outdoor air supply, e.g., an outdoor air ratio of 100% without returned air [17].

For the EAHAP (Fig. 2), the exhaust air is reused as the cold source to cool the condenser. Since the temperature of the exhaust air from the room is lower than the outdoor air temperature, the condensation temperature of the EAHAP is lower than that of the OAHP. It should be noted that the flowrate of air used to cool the condenser is larger than

Fig. 1. Schematic of outdoor air heat pump.
that used to heat the evaporator since more heat exchange takes place in the condenser than that in the evaporator according to the energy conservation law. Thus, besides the exhaust air from the room, outdoor air is also introduced to cool the condenser. For example, Cao et al. [35] recommended that the optimal ratio of the flowrate of the exhaust air to the total flowrate of air used to cool the condenser was between 0.35 and 0.40.

2.2. Model of heat pump

The EAHP and the OAHP share the same heat pump model except for the condensation temperature caused by different cold sources (Figs. 1 and 2). Fig. 3 shows the schematic of the pressure-enthalpy diagram of the heat pump. The compressor sucks the gas at a low temperature and low pressure from the evaporator (at the state point 1) and pressurizes it to the state point 2 with a high temperature and high pressure. The gas out from the compressor is condensed in the condenser (at the state point 3) and decompressed by the throttle valve (at the state point 4). The liquid refrigerant at a low temperature and low pressure is evaporated in the evaporator (at the state point 1). The model of the heat pump consists of these four processes in the four components. The sub-models of the other three components and the fan used to supply and exhaust the air are described as follows.

The compressor (i.e., the process from the state point 1 to the state point 2) is determined by the isentropic efficiency, mechanical efficiency, and motor efficiency (Equation 1) [46]. The mechanical efficiency and the motor efficiency are generally fixed [47], and from different studies, different fixed values of the mechanical efficiency (e.g., 0.93 – 0.96) and the motor efficiency (e.g., 0.85 – 0.95) can be found [48,49]. The ideal condition of the compressor is isentropic and the isentropic efficiency of the real condition is defined as Equation 2. The isentropic efficiency can be modeled as the function of the pressure ratio (defined as the ratio of the condensation pressure to the evaporation pressure) [50]. For example, Kavian et al. [51] and Liu et al. [52] modeled that the isentropic efficiency linearly decreased with the increasing pressure ratio. Wang et al. [49] modeled the isentropic efficiency as a quadratic function of the pressure ratio. According to the data reported by Kinab et al. [53], this study proposes a piecewise model combining a linear function and a quadratic function to quantify the isentropic efficiency. When the pressure ratio is less than (2.5 - α), the isentropic efficiency is modeled as a quadratic function of the pressure ratio (Equation 3) and otherwise, the isentropic efficiency is modeled as a linear function of the pressure ratio (Equation 4). At the pressure ratio of (2.5 - α), the isentropic efficiency achieves its maximal value, and the constant α is determined according to the application condition for a high isentropic efficiency [53]. The power input into the fan is calculated by Equation 5 [19].
Where α is a constant; h₁ and h₂ are the enthalpies of the refrigerant at the state point 1 and the state point 2 under the real condition respectively (kJ/kg); h₂,is is the enthalpy of the refrigerant at the state point 2 under the ideal condition (kJ/kg); m₁ is the flow rate of air through the fan (kg/s); mₑ is the refrigerant flowrate (kg/s); P₁ and P₂ are the pressures of the state point 1 and the state point 2 respectively, i.e., the evaporation pressure and the condensation pressure respectively (Pa); ΔPᵣ is the pressure drop of fan (kPa); Wₑ is the power input to the compressor (kW); Wᵣ is the power output to the compressor (kW); nₑ, nₐ, and nₘₑ are the isentropic efficiency, the mechanical efficiency, and the motor efficiency of the compressor respectively; $\rhoₐ$ is the density of the air through the fan (kg/m³).

The processes in the condenser and the evaporator are modeled by the effectiveness-NTU method (Equations 6 and 7) [54]. The effectiveness refers to the heat transfer effectiveness of the heat exchanger, i.e., the condenser (Equation 8) and the evaporator (Equation 9) [38]. The NTU refers to the number of heat transfer units (Equations 10 and 11) [55]. For simplification, the multiple of the area and the overall heat transfer coefficient of the condenser or the evaporator can be regarded as a fixed value [56–58]. For the OAHP, the temperature of air entering the condenser is the temperature of outdoor air, and for the EAHHP, the temperature of air entering the condenser is the temperature of outdoor air and the temperature of the mixing of exhaust air and outdoor air (Equation 12). For both the OAHP and the EAHHP, the temperature of air entering the evaporator is the temperature of the mixing of outdoor air and returned air (Equation 13).

$$ Wₑ = \frac{mₑ(hₑₐₑ - hₑ)} {nₑₐₑchₐₑ} $$  \hspace{1cm} (1)

$$ nₑ = \frac{hₑₐₑ - hₑ}{h₂ - h₁} $$  \hspace{1cm} (2)

$$ nₑ = -0.3000 \left( \frac{P₂}{P₁} + a \right)^2 + 1.5300 \left( \frac{P₂}{P₁} + a \right) - 1.2000 $$  \hspace{1cm} (3)

$$ nₑ = -0.0333 \left( \frac{P₂}{P₁} + a \right) + 0.8333 $$  \hspace{1cm} (4)

$$ Wᵣ = \frac{mᵣΔP}{nᵣₚᵣ} $$  \hspace{1cm} (5)

Where α is a constant; h₁ and h₂ are the enthalpies of the refrigerant entering the condenser and leaving the condenser respectively (°C); tₑ,is and tₑ,cool are the temperatures of air entering the evaporator and leaving the evaporator respectively (°C); tₑ,o is the outdoor air temperature (°C); tₑ is the exhaust air temperature (°C); $Uₑ$ and $Uᵣ$ are the overall heat transfer coefficients of the condenser and the evaporator respectively (W/(m²·K)); $cₑ$ and $cᵣ$ are the effectiveness of the condenser and the evaporator respectively; $γₑ$ is the outdoor air ratio.

The COP of the heat pump (both the OAHP and the EAHHP) is calculated as the ratio between the total cooling load and the power input (Equation 14). The total cooling load is the same as the heat absorbed by the evaporator (Equation 15). The total cooling load consists of the space cooling load and the ventilation load. The space cooling load is the heat removed by the ventilation from the room (Equation 16). Thus, the ventilation load can be calculated from the total cooling load and the space cooling load by Equation 17.

$$ COP = \frac{Qₑ}{Wₑ} $$  \hspace{1cm} (14)

$$ Qₑ = mₑₐₑ(hₑₐₐₑ - hₑₐₑ) $$  \hspace{1cm} (15)

$$ Qᵥₑ = cₑmₑₐₑ(tₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑₑالة) [19,20,62]. The heat removal efficiency has been widely used for the energy performance evaluation of advanced air distribution like stratum ventilation and displacement ventilation [19,20,42,43,63,64]. Since this study concerns the energy performance, air distribution is modeled by the heat removal efficiency. The 1-D air temperature distribution in Fig. 4 is sufficient for the calculation of the heat removal efficiency [19,20,42,43,63,64].

Different air distributions have different layouts of the inlets and exits, with different air temperature distribution profiles [65]. With the conventional air distribution, i.e., mixing ventilation, the air enters and exits the room at the ceiling level, and the air temperature in the room is designed to be constant at different heights of the room (Fig. 4a) [20]. There are mainly two types of advanced air distribution, i.e., the thermal buoyancy-dominated one and the supply momentum-dominated one. The thermal buoyancy-dominated advanced air distribution is characterized by the thermal stratification with the air temperature increasing with the height, and the supply momentum-dominated advanced air distribution is characterized by the thermal stratification with the lowest air temperature at the head level [21]. Displacement ventilation is a representative of the thermal buoyancy-dominated advanced air distribution. With displacement ventilation, the air enters the room at the floor level and exits the room at the ceiling level, and the air temperature in the room is designed to increase with the height (Fig. 4b) [42]. Stratum ventilation is a representative of the supply momentum-dominated advanced air distribution. With stratum ventilation, the air enters the room at the middle level of the wall and exits the room at the ceiling level, and the air temperature is lowest at the head level of occupants (Fig. 4c) [43].

The heat removal efficiency is used to characterize the air distribution. The heat removal efficiency is defined as the ratio of the difference
between the exhaust air temperature and the supply air temperature to the difference between the air temperature in the occupied zone and the supply air temperature (Equation 18) [43]. The larger the heat removal efficiency, the more advanced the air distribution [20]. A heat removal efficiency of unity indicates that the supply air is uniformly distributed in the room, e.g., mixing ventilation, and a heat removal efficiency of larger than unity indicates that the air is more efficiently supplied into the occupied zone than into other zones, e.g., displacement ventilation and stratum ventilation [21]. Besides personalized ventilation, the heat removal efficiency of existing air distributions generally ranges between 1.0 and 1.5 [20].

$$HRE = \frac{t_{ae, out} - t_{ae, int}}{t_{ae, out} - t_{ae, in}}$$

(18)

Where HRE is the heat removal efficiency; $t_{ae, in}$ is the air temperature in the occupied zone (°C).

As shown in Section 2.2, to model the energy performance of the combination of the heat pump with air distribution, the temperature of air supplied into the room (i.e., the temperature of air leaving the evaporator) and the temperature of exhaust air from the room are required. Combing Equations 16 and 18, the supply air temperature and the exhaust air temperature can be calculated from the heat removal efficiency, the space cooling load, the supply airflow rate, and the air temperature of the occupied zone with Equation 19 and Equation 20 respectively. The air temperature in the occupied zone can be determined according to the thermal comfort requirement, e.g., 25.0 °C [66]. The supply airflow rate is determined for a reasonable air velocity in the occupied zone [60]. The space cooling load can be determined by the internal cooling load and the heat transmission from the ambient through the envelope [67].

$$t_{ae, int} = t_{ae, out} - \frac{Q_{scl}}{c_p \dot{m}_{ae} HRE}$$

(19)

$$t_{ae} = t_{ae, out} + \frac{Q_{scl}}{c_p \dot{m}_{ae}} \left(1 - \frac{1}{HRE}\right)$$

(20)

3. Results

3.1. Model validation

Experimental data reported by Kinab et al. [53] are used to validate the theoretical model in Section 2 because of the detailed information reported and the wide experiment conditions covered. A total of nine experiments are included. The experiment conditions cover the condensation temperature between 26 °C and 54 °C and the evaporation temperature between 6 °C and 15 °C. The details of the experiments can be found in Kinab et al. [53]. Fig. 5 shows the measured COPs and those predicted by the model described in Section 2. The predicted COP varies between 2.3 and 4.6, which accurately captures the variation in the measured COP (Fig. 5). Compared with the measured COP, the error in the predicted COP is within ±5% and randomly distributes around zero (Fig. 6). The mean absolute error and the standard deviation of absolute errors of the predicted COP are 3.2% and 1.6% respectively. Thus, the theoretical model in Section 2 is credible.
3.2. Energy performance

Case studies are conducted according to our previous studies in a classroom with the dimensions of 8.8 m (length) × 6.1 m (width) × 2.4 m (height) [43]. The classroom is located in the interior zone, thus the space cooling load is explained by the internal load (e.g., the lighting and occupants). The air temperature in the occupied zone is set to 25.0 °C for thermal comfort [66]. The supply airflow rate is set to 10 ACH [65]. The space cooling load is set to 2.6 kW with the temperature difference between the supply air and the exit air of 6.0 °C [68]. The relative humidity of the indoor air and the outdoor air is set to 65% and 75% respectively [65]. The outdoor air ratio is set to 100% and the outdoor air temperature is set to 35.0 °C as the design condition [69]. Regarding the parameters of the heat pump, the constant α (Equations 3 and 4) is set to 0.8 with the maximal isentropic efficiency achieved at the pressure ratio of 1.7 [53]. The heat transfer properties of the condenser (UAe) and the evaporator (UAc) are set to 2.2 W/K and 0.73 W/K respectively with the effectiveness of the condenser and evaporator both of around 0.82 (Equations 8 and 9). The ratio of the flowrate of air cooling the condenser to the flowrate of air heating the evaporator is 3 [35]. The refrigerant is R410A [53].

3.2.1. Effects of advanced air distribution on cooling load

Fig. 7 shows the supply air temperatures and exhaust air temperatures for different air distributions characterized by the heat removal efficiency. Advancing the air distribution with the heat removal efficiency increasing from 1.0 to 1.5, the supply air temperature increases from 19.0 °C to 21.0 °C (Equation 19), and the exhaust air temperature increases from 25.0 °C to 27.0 °C (Equation 20). Thus, the more advanced the air distribution, the higher the supply air temperature and the exhaust air temperature. At the outdoor air ratio of 100%, the ventilation load decreases by 19% (from 17.9 kW to 15.1 kW) with the increasing heat removal efficiency, which reduces the total cooling load by 16% (from 20.4 kW to 17.6 kW) (Fig. 8). This is because the increasing heat removal efficiency enhances the supply air temperature, which reduces the sensible load and the latent load of the outdoor air supply. Thus, the more advanced the air distribution, the smaller the ventilation load and the total cooling load. Moreover, the ventilation load accounts for a large proportion of the total cooling load, ranging from 86% to 88%, which emphasizes the importance of improving the energy efficiency of the outdoor air supply.

3.2.2. Effects of advanced air distribution on COP of heat pump

Fig. 9 shows that by advancing the air distribution (i.e., increasing the heat removal efficiency), the condensation temperature decreases for both the OAHP and the EAHP, which is explained as follows.

\[ t_c = t_{x,m} + \frac{Q}{\varepsilon_c \varepsilon_m \dot{m}_{x}} \]  

(21)

Fig. 7. Variations of supply air temperature and exhaust air temperature with heat removal efficiency.

Fig. 8. Variations of total cooling load and ventilation load with heat removal efficiency.

Fig. 9. Variations of condensation temperatures of outdoor air heat pump (OAHP) and exhaust air heat pump (EAHP) with heat removal efficiency.

According to Equation 21, which is derived from Equation 8 and the energy conservation law of the condenser (Equation 22), advancing the air distribution negatively affects the condensation temperature by enhancing the exhaust air temperature (Fig. 7) and positively affects the condensation temperature by decreasing the total cooling load (Fig. 8). The decreased total cooling load due to advancing the air distribution reduces the heat released by the condenser (according to the energy conservation law of the heat pump). The positive effect of the decreased total cooling load overwhelms the negative effect of the enhanced exhaust air temperature thereby decreasing the condensation temperature. Moreover, the condensation temperature of the EAHP (decreasing from 57.1 °C to 53.1 °C) is lower than that of the OAHP (decreasing from 61.7 °C to 56.5 °C) (Fig. 9). This is because the reuse of the exhaust air by the EAHP reduces the temperature of air entering the condenser. It can be observed that the condensation temperature difference between the OAHP and the EAHP decreases from 4.6 °C to 3.4 °C with the heat removal efficiency increasing from 1.0 to 1.5 (Fig. 9). This is because the increasing heat removal efficiency enhances the exhaust air temperature (Fig. 7). With the increasing exhaust air temperature, the reduction in the temperature of air entering the condenser caused by the reuse of the exhaust air is weakened. Anyway, a reduction in the condensation temperature larger than 3.3 °C by the reuse of the exhaust air indicates the effectiveness of the EAHP with advanced air distribution.
Fig. 10 shows that the COPs of both the OAHP and the EAHP increase when advancing the air distribution with an increasing heat removal efficiency. The increase in the COP is partially explained by the decreasing condensation temperature with the increasing heat removal efficiency (Fig. 9). Moreover, the increasing evaporation temperature (Fig. 11) with the increasing heat removal efficiency also contributes to the increase in the COP. The increasing evaporation temperature with the increasing heat removal efficiency is explained as follows. Firstly, the enhanced supply air temperature by advanced air distribution helps to increase the evaporation temperature (Fig. 7). Second, the decreasing total cooling load with the increasing heat removal efficiency (Fig. 8) reduces the temperature difference between the air entering and leaving the evaporator \((t_{a,e,in} - t_{a,e,out})\), which increases the evaporation temperature according to Equation 23 (derived from Equation 9 [54]). Compared with the COP of the OAHP (between 2.2 and 2.9), the COP of the EAHP (between 2.6 and 3.4) is higher because of the lower condensation temperature. The COP difference between the EAHP and the OAHP decreases from 21% to 16% with the heat removal efficiency increasing from 1.0 to 1.5, which is consistent with the variation of the difference between the condensation temperatures of the two heat pumps.

\[
t_e = \frac{t_{a,e,in} - t_{a,e,out}}{t_e} \quad (23)
\]

### 3.2.3. Effects of advanced air distribution on system energy consumption

Due to the decreasing total cooling load (Fig. 8) and the increasing COP (Fig. 10), the power consumption of the OAHP decreases from 9.4 kW to 6.0 kW with the increasing heat removal efficiency from 1.0 to 1.5. Thus, advancing the air distribution by increasing the heat removal efficiency from 1.0 to 1.5 saves energy by 36%. By reusing the exhaust air to reduce the condensation temperature, the EAHP further reduces the power consumption. The power consumption of the EAHP decreases from 7.8 kW to 5.2 kW with the heat removal efficiency increasing from 1.0 to 1.5. Compared with the OAHP, the EAHP further saves energy by 14% – 18%. Using the OAHP with the heat removal efficiency of 1.0 as the benchmark, the EAHP merely (i.e., the EAHP with the heat removal efficiency of 1.0) saves energy by 18% reducing the power consumption from 9.4 kW to 7.8 kW), advanced air distribution with the heat removal efficiency of 1.5 merely (i.e., the OAHP with the heat removal efficiency of 1.5) saves energy by 36% (reducing the power consumption from 9.4 kW to 6.0 kW), and the integrated system of the EAHP and advanced air distribution (i.e., the EAHP with the heat removal efficiency of 1.5) can save energy by 45% (reducing the power consumption from 9.4 kW to 5.2 kW) (see Fig. 12).

### 4. Discussion

While outdoor air supply is essential in airborne infection risk control of infectious respiratory diseases [70], the above analyses demonstrate that the proposed EAHP with advanced air distribution can energy-effectively provide outdoor air with three mechanisms. First, the EAHP reuses the exhaust air to reduce the condensation temperature, thereby improving the COP. Second, advanced air distribution reduces the ventilation load. Third, advanced air distribution reduces the condensation temperature and enhances the evaporation temperature, thereby improving the COP. The above analyses use the heat removal efficiency between 1.0 and 1.5 to represent the effect of air distribution. Since the heat removal efficiency of the existing air distributions is generally between 1.0 and 1.5, the obtained results are general regarding the effect of air distribution [21]. However, when the EAHP is integrated with specific air distribution, the heat removal efficiency of air distribution is defined. The heat removal efficiency of air distribution can be predicted by Equation 24 [43]. For a specific air distribution, the three constants in Equation 24, i.e., \(a\), \(b\), and \(c\), are specific. For example, for the stratum-ventilated classroom located in the City University of Hong Kong (Fig. 13), the three constants are 1.737, –0.070, and 0.766 respectively (Equation 25) [43]. The model of the heat removal efficiency of stratum ventilation has been

\[
Q_c = c_m a_c (t_{c,in} - t_{c,in})
\]

Where \(Q_c\) is the heat released by the condenser (kW).

\[
\text{COP} = \frac{Q_c}{W_c}
\]

\[
\text{Power consumption} = W_c + W_p
\]

Fig. 12. Variations of power consumptions of outdoor air heat pump (OAHP) and exhaust air heat pump (EAHP) with heat removal efficiency.
experimentally validated with a mean absolute error of around 2% (Fig. 14). The detailed validation of the heat removal efficiency model (Equation 25) can be found in Reference [43]. The classroom is in the interior zone of the building with a space cooling load of 2.6 kW and its designed supply airflow rate is 10 ACH. Accordingly, the heat removal efficiency of the stratum-ventilated classroom is around 1.2 (Equation 25).

\[
\frac{1}{HRE} = \frac{V_s(a + b_{\text{strum}})}{Q_{\text{str}}^\text{cool}} + c
\]

\[
\frac{1}{HRE} = \frac{V_s(1.737 - 0.070t_{\text{strum}}) + 0.766}{Q_{\text{str}}^\text{cool}}
\]

Fig. 13. Schematic of stratum-ventilated classroom [43].

The effect of the outdoor air ratio on the proposed EAHP-stratum ventilation is tested. For airborne infection risk control with the pandemics of respiratory diseases like COVID-19, the returned air is recommended to be switched off with an outdoor air ratio of 100% [12]. However, in the post-pandemic era, by making a trade-off between the health effect and energy penalty, the returned air might be used with the outdoor air ratio of less than 100% [30]. Fig. 13 shows the variations of the total cooling load and the ventilation load with the outdoor air ratio when the outdoor air temperature is 35 °C. The minimal outdoor air ratio of 45% is determined by the outdoor air of 10 L/s for each occupant [19] and 16 occupants in the studied classroom [43]. The total cooling load of the EAHP with stratum ventilation increases from 9.6 kW to 19.0 kW with the outdoor air ratio increasing from 45% to 100% because of the increase in the ventilation load (Fig. 15). In contrast, the total cooling load of the OAHP with mixing ventilation increases from 10.3 kW to 20.4 kW. The heat removal efficiency of mixing ventilation is assumed to be 1.0 [43]. Due to the increasing total cooling load, the condensation temperature increases (Equation 21). Moreover, the outdoor air ratio increases the temperature of the air entering the evaporator, thereby reducing the evaporation temperature (Equation 23). Because of the reduced evaporation temperature and the increased condensation temperature, the COP of the EAHP with stratum ventilation decreases from 5.1 to 3.0 with the outdoor air ratio increasing from...
45% to 100% (Fig. 16). In contrast, the COP of the OAHP with mixing ventilation decreases from 4.3 to 2.1 with the outdoor air ratio increasing from 45% to 100% (Fig. 16). However, regardless of the variation of the outdoor air ratio, compared with the OAHP with mixing ventilation, the EAHP with stratum ventilation decreases the total cooling load and enhances the COP, thereby saving energy by 21% – 35% (Fig. 17). The larger the outdoor air ratio, the more energy saved by the EAHP with stratum ventilation (Fig. 17).

The outdoor weather condition also affects the performance of the exhaust air heat pump with stratum ventilation, particularly the outdoor air temperature [33]. Fig. 18 shows, at the outdoor air ratio of 100%, the increasing outdoor air temperature from 31 °C to 35 °C enhances the total cooling load of the EAHP with stratum ventilation from 11.7 kW to 19.0 kW, and enhances the total cooling load of the OAHP with mixing ventilation from 13.1 kW to 20.4 kW. Fig. 19 shows that the increasing outdoor air temperature from 31 °C to 35 °C reduces the COP of the EAHP with stratum ventilation from 5.7 to 3.0 and reduces the COP of the OAHP with mixing ventilation from 4.5 to 2.1. However, regardless of the variation of the outdoor air temperature, compared with the OAHP with mixing ventilation, the EAHP with stratum ventilation reduces the total cooling load and enhances the COP, thereby saving energy by 29% – 35% (Fig. 20). The larger the outdoor air temperature, the more energy saved by the EAHP with stratum ventilation (Fig. 20).

This study focuses on the energy performance of the proposed integrated system of the EAHP and advanced air distribution. Advanced air distribution is used to reduce energy consumption by reducing the cooling load and improving the COP. Thus, both Sections 3 and 4 are organized with the logic that the effects of advanced air distribution on the cooling load (including the total cooling load and ventilation cooling load via the variations of the supply and exhaust air temperatures) are presented first, followed by the effects of advanced air distribution on the COP of heat pump and the effects of advanced air distribution on system energy consumption. The effects of advanced air distribution have been studied with a general method and specific cases in this study. The general method implements the heat removal efficiency of 1 – 1.5 to indicate the different advanced levels of air distribution (Section 3), and the specific cases compare a specific advanced air distribution, i.e., stratum ventilation, and the conventional air distribution, i.e., mixing ventilation (Section 4). Regarding indoor environment quality, the air temperature in the occupied zone is maintained at 25.0 °C for thermal comfort [66], and the outdoor air supply is maintained to be larger than 10 L/s for each occupant for indoor air quality [19]. In future studies, the room air temperature and outdoor air ratio will be optimized (via the optimization of the supply air temperature, supply airflow rate, and outdoor air ratio) for further improvements in energy efficiency while maintaining thermal comfort and indoor air quality.
5. Conclusions

This study proposes to integrate the EAHP and advanced air distribution for energy-efficient provision of outdoor air. The EAHP reuses the exhaust air to reduce the condensation temperature, thereby improving the COP. Advanced air distribution reduces the ventilation load. Moreover, advanced air distribution reduces the condensation temperature and enhances the evaporation temperature, thereby improving the COP. Advanced air distribution reduces the condensation temperature is because the positive effect of the ventilation load reduced by advanced air distribution on the condensation temperature overwhelms the negative effect of the exhaust air temperature enhanced by advanced air distribution on the condensation temperature. Advanced air distribution reduces the evaporation temperature is because of the positive effects of the reduced ventilation load and the enhanced supply air temperature by advanced air distribution on the evaporation temperature. These three mechanisms explain the high energy efficiency of the integrated system of the EAHP and advanced air distribution.

The EAHP saves energy by 18%, advanced air distribution saves energy by 36%, and the integrated system of the EAHP and advanced air distribution has an energy saving potential of 45%. Particularly, the effectiveness of the EAHP with stratum ventilation is demonstrated under different outdoor air ratios and outdoor air temperatures. Compared with the conventional system (i.e., the outdoor air heat pump with mixing ventilation), the EAHP with stratum ventilation saves energy by 21% – 35%.

This study has demonstrated the effectiveness of the proposed integrated system of the EAHP and advanced air distribution for cooling applications. In future studies, the effectiveness of the proposed integrated system for heating applications will be tested. This study has not optimized the operation control of the proposed integrated system (e.g., the supply air temperature, supply airflow rate, and outdoor air rate), while the operation control could affect the energy efficiency of the proposed integrated system. In future studies, the operation control of the proposed integrated system will be optimized for further energy efficiency improvement. Moreover, the proposed integrated system of the EAHP and advanced air distribution is promising in reducing economic cost compared with the existing system of the EAHP and conventional air distribution regarding both the initial cost and operation cost. Advanced air distribution reduces the cooling load thereby reducing the integrated system size. The reduced system size decreases the initial cost. The integrated system has lower energy consumption thereby lowering the operation cost. The economic cost of the proposed integrated system will be evaluated based on the optimized integrated system in future studies.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Data availability

Data will be made available on request.

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