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Research on Relative Humidity and Energy Savings for Air-Conditioned Spaces without Humidity Control When Adopting Air-to-Air Total Heat Exchangers in Winter

Ming Dong, Jialiang Zhang, Liufeng Zhang *, Lianbo Liu and Xingqiang Zhang

College of Science and Technology, Ningbo University, Ningbo 315300, China; dongming@nbu.edu.cn (M.D.); zhangjialiang@nbu.edu.cn (J.Z.); liulianbo@nbu.edu.cn (L.L.); zxqpeter@126.com (X.Z.)
* Correspondence: zhangliufeng@nbu.edu.cn

Abstract: In view of the problem that the exchange effectiveness is calculated according to a fixed value or only considering the influence of outdoor air parameters when analyzing the suitability of total heat recovery for plate heat recovery equipment in air-conditioned spaces without humidity control, the indoor humidity calculation model and moisture balance equation were established in this research to predict relative indoor humidity. Moreover, the relationship between total heat recovery, effective heat recovery, and the reduction in outdoor air heating load was analyzed using a psychrometric chart of the outdoor air treatment process. Referring to the standard for weather data of building energy efficiency in the Ningbo region, 6 typical days were taken as the calculation conditions. The moisture balance differential equation was solved using MATLAB software to obtain numerical solutions for the hourly indoor air humidity ratio, relative humidity, exchange effectiveness, and effective heat recovery when adopting an air-to-air total heat exchanger in an air-conditioned room of an office, classroom, or commercial building in the winter. The results indicate that, under the calculation conditions, the relative indoor humidity of commercial buildings is relatively high, making it unsuitable for a total heat exchanger. The relative humidity of indoor air in offices and classrooms can be maintained above 30%, and the total exchange effectiveness of a total heat exchanger is between 45% and 100%. The effective total heat recovery was calculated as sensible heat recovery under most calculation conditions.

Keywords: without humidity control; relative humidity; total heat exchanger; dynamic total exchange effectiveness; effective total heat recovery; plate heat recovery equipment

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1. Introduction

In order to reduce the carbon emission intensity of newly built residential buildings and public buildings by an average of 7 $kgCO_2/(m^2 \cdot a)$, on the basis of the design standard for energy efficiency implemented in 2016 [1], it is imperative to reduce the energy consumption of air conditioning systems. In addition to improving the energy utilization efficiency of air conditioning systems, reducing the air conditioning load is particularly important. However, in order to improve indoor air quality, a room needs to receive the necessary volume of outdoor air, which is limited by air conditioning load. Therefore, total heat exchangers are widely used in buildings as heat recovery equipment to reduce outdoor air load [2,3]. The ability to reasonably calculate the heat recovery volume is the key to the suitability analysis of the total heat recovery technology.

Currently, enthalpy wheels [4–6], plate heat recovery equipment, and heat pipe recovery equipment [7] are the main pieces of heat recovery equipment for ventilation systems [8], and scholars have conducted a series of studies on them. Unlike the enthalpy wheels commonly used in centralized air conditioning systems, for decentralized and semi-centralized air conditioning systems, plate heat recovery equipment are widely used in new and renovated buildings due to their simple structure and small space requirements. Given the

extensive utilization of plate heat recovery equipment in building air conditioning systems, the research on their heat exchange performance holds paramount significance. For plate heat recovery equipment, scholars have carried out research on the impact of exhaust heat recovery on the outdoor air load and energy-saving effects of buildings in different regions. Exhaust heat recovery has an impact on the outdoor air heat load of ultra-low energy consumption residential buildings in severe cold and cold regions [9,10], and the energy-saving effect of heat recovery technology gradually increases with an increase in indoor and outdoor temperature difference. Cheng et al. [11] tested the energy-saving effects of heat recovery technology in a residential building during hot summer and cold winter periods. The results showed that the heat recovery system in the summer could reduce air conditioning energy consumption by 14.5%, while there was no significant energy-saving effect in transition seasons. Duan et al. [12] calculated the heat recovery of residential exhaust air at night. The results showed that full heat recovery mode was more suitable for the whole year in cold areas and regions with hot summers and cold winters while utilizing total heat recovery mode in the cooling season and sensible heat recovery mode in the heating season were more appropriate in regions with hot summers and warm winters. Zhou et al. [13] pointed out that, at desired indoor temperatures, the majority of the total heat recovery in Beijing and Shanghai is generated in winter, indicating that heat recovery in winter plays an important role in energy saving for outdoor air systems. Some scholars have studied the energy conservation of exhaust heat recovery based on different building types. For example, Wang et al. [14] analyzed the energy conservation of heat recovery technology in libraries in Beijing and demonstrated that both sensible heat recovery and total heat recovery resulted in significant energy conservation. Yang et al. [15] studied the suitability of the plate–fin total heat exchanger, rotary heat exchanger, and heat pump outdoor air unit with heat recovery in the reconstruction of outdoor air systems. As total heat recovery depends largely on the heat exchange effectiveness, research on the factors affecting heat recovery effectiveness has become more popular, mainly focusing on the influence of the heat and humidity exchange medium [16–19], the influence of airflow inside the total heat exchanger [20–24], and the influence of indoor and outdoor air parameters. The heat exchange efficiency of the total heat exchanger under different outdoor conditions with varied parameters is significantly different from the heat recovery efficiency under nominal conditions [25]. Using theoretical derivation and experimental verification, Wang et al. [26] proposed that the sensible exchange effectiveness of the total heat exchanger is independent of indoor and outdoor air parameters, while the absolute humidity ratio exchange effectiveness and total exchange effectiveness are dependent on indoor and outdoor air parameters. Zhong et al. [27,28] and Zhang et al. [29] also verified this conclusion. In terms of the impact of outdoor air parameters on the total exchange effectiveness, Wang et al. [30] used rated total exchange effectiveness and dynamic total exchange effectiveness to calculate the hourly energy savings of a total heat exchanger and demonstrated that changes in outdoor meteorological parameters should be considered when calculating heat recovery. In addition, Zhang et al. [31] and others proposed that there is no heating load or that the heating load is small in some parts of the rooms in the inner area in winter and that the heat recovered by the air conditioner cannot be counted as energy savings. In the technical and economic analysis of exhaust heat recovery equipment, it is necessary to accurately calculate the effective heat recovered by a heat exchanger.

To sum up, at present, in the calculation of total heat exchanger heat recovery or suitability analysis, heat exchange effectiveness basically adopts a fixed value or the dynamic total heat exchange effectiveness considering only the changes in outdoor meteorological parameters, and the indoor air parameters of air-conditioned spaces are considered fixed values. However, the vast majority of air conditioning systems are mainly comfort air conditioners, mainly controlling indoor air temperature in the winter and generally not the relative humidity. Therefore, the existing research needs to be improved in the following two aspects: (1) when analyzing total heat recovery, the influence of relative indoor humidity on the heat exchange effectiveness of the total heat exchanger caused

by the change in outdoor meteorological parameters and indoor humidity load is not considered and (2) for air-conditioned spaces without humidity control, the relationship between total heat recovery and outdoor air load reduction is worth studying. In view of the above problems, this paper takes Ningbo City as an example to calculate the indoor humidity of air-conditioned spaces of different functional buildings when adopting plate heat recovery equipment under typical daily meteorological parameters. The dynamic total heat exchange effectiveness, considering the changes in indoor and outdoor air parameters, is used to calculate the heat recovery of exhaust air from air conditioning systems in the winter. The heat recovery of the total heat exchanger and the reduction in outdoor air load are compared, which provides a reference for the prediction of relative indoor humidity and the calculation of the energy savings of total heat exchangers.

2. Methods

2.1. Calculation Method for Total Heat Recovery in Winter

The heat recovery of total heat exchangers depends on the outdoor air rate, the temperature and humidity difference between outdoor air and exhaust air, and the heat exchange effectiveness of the total heat exchanger. The basic calculation is as follows: select the typical days during the period when air conditioning is being utilized for heating operations in the winter, fit the outdoor air humidity curve formula according to the hourly meteorological data in typical years, and solve the moisture balance differential equation to determine the hourly temperature and humidity in air-conditioned spaces and the heat exchange effectiveness. The hourly heat recovery of the total heat exchanger during the operation period is thus obtained.

2.1.1. Total Heat Exchange Effectiveness

The exhaust air and outdoor air in the total heat exchanger represent heat and humidity exchange, and the total exchange effectiveness and absolute humidity ratio exchange effectiveness are calculated according to Formula (1) and Formula (2), respectively.

$$\eta_{\mathrm{h}} = \frac{h_{\mathrm{w}} - h_{\mathrm{s}}}{h_{\mathrm{w}} - h_{\mathrm{n}}} \times 100\% = \frac{\eta_{\mathrm{t}}}{1 + 2.48\frac{d_{\mathrm{w}} - d_{\mathrm{s}}}{t_{\mathrm{w}} - t_{\mathrm{s}}}} + \frac{\eta_{\mathrm{d}}}{1 + 0.404\frac{t_{\mathrm{w}} - t_{\mathrm{s}}}{d_{\mathrm{w}} - d_{\mathrm{s}}}} \tag{1}$$

$$\eta_{\mathrm{d}} = \frac{d_{\mathrm{w}} - d_{\mathrm{s}}}{d_{\mathrm{w}} - d_{\mathrm{n}}} \times 100\% \tag{2}$$

where η_{h} is the total exchange effectiveness, %; h_{w} is the enthalpy of outdoor air, kJ/kg; h_{s} is the enthalpy of supply air, kJ/kg; h_{n} is the enthalpy of exhaust air, kJ/kg; η_{t} and η_{d} are the sensible exchange effectiveness and absolute humidity ratio exchange effectiveness, %; d_{w}, d_{s}, and d_{n} are the humidity ratios of the outdoor air, supply air, and exhaust air, respectively, g/kg; and t_{w} and t_{n} are the temperatures of the outdoor air and exhaust air, respectively, °C.

Sensible exchange effectiveness η_{t} and the absolute humidity ratio exchange effectiveness η_{d} depend on the number of heat transfer units NTU and number of mass transfer units NTU_{D} [26] of the total heat exchanger, respectively, and the calculation of absolute humidity ratio exchange effectiveness is shown in Equation (3).

$$\eta_{\mathrm{d}} = \left[1 - \mathrm{e}^{NTU_D{}^{0.22} \cdot (\mathrm{e}^{-NTU_D{}^{0.78}} - 1)}\right] \times 100\% \tag{3}$$

where NTU_{D} is the number of mass transfer units of total heat exchanger and is dimensionless.

The calculation of the number of mass transfer units NTU_{D} is shown in Equation (4).

$$NTU_D = \frac{k_{\mathrm{d}} \mathrm{F}_1}{\mathrm{g}_{\mathrm{f}}} = \frac{\mathrm{F}_1}{\left\{\frac{2}{\alpha_{\mathrm{d}}} + \frac{\delta_{\mathrm{P}}}{D_{\mathrm{wn}}} \frac{10^6[(1-C)\varphi_1 + C] \cdot [(1-C)\varphi_2 + C]}{e^{(5294/T)}\omega_{\max} C}\right\} \mathrm{g}_{\mathrm{f}}} \tag{4}$$

where k_d is the total mass transfer coefficient, kg/(m^2·s); F_1 is the primary heat transfer area of the total heat exchanger, m^2; g_f is the mass flow rate, kg/s; α_d is the convective mass transfer coefficient, kg/(m^2·s); δ_p is the thickness of the wet exchange film, m; D_{wn} is the mass diffusion coefficient, kg/(m·s); C is the calculated value corresponding to the adsorption type of the wet exchange film and is dimensionless; φ_1, φ_2, respectively, refer to the relative humidities of outdoor and indoor air, %; T is the average indoor and outdoor temperature, K; and ω_{max} is the maximum water capacity of the film, kg/kg.

The relative humidity of outdoor air can be found in [32]. According to the indoor air pressure P = 101.3 kPa and the temperature t = 20 °C in air-conditioned spaces in winter, the humidity ratio of the wet air in this state is d_s = 14.88 g/kg [33], and the relative humidity of the indoor air is calculated according to Equation (5).

$$\varphi_2 = \frac{d_n}{d_{sa}} \times 100\% = \frac{d_n}{0.1488} \tag{5}$$

where d_n, d_{sa} are the humidity ratio of wet air and saturated air at the same temperature and pressure, g/kg.

2.1.2. Indoor Humidity Balance Equation

Winter comfort air conditioning systems generally only control indoor temperature and do not have separate humidification or dehumidification devices for relative humidity control. However, relative humidity is not only related to a person's thermal comfort but also, to a certain extent, affects the heat recovery capacity of total heat exchangers. Therefore, the relative humidity of air-conditioned spaces without humidity control should be calculated as follows.

Establish an indoor humidity ratio calculation model, as shown in Figure 1. Assuming the room volume is V, the indoor design temperature is t_n and the humidity gain is d (t) at the desired indoor temperature. After the humidity evaporates, it is immediately mixed with indoor air and outdoor air, and the indoor humidity is in a uniform state. When the air conditioning system starts operating, the indoor air humidity ratio is d_0. At time t, the outdoor air, supply air, and exhaust air humidity ratio are d_w (t), d_s (t), and d_n (t), respectively. Absolute humidity ratio exchange effectiveness is η_d. The ventilation rate is L. The air-conditioned space in winter can be obtained from the humidity balance equation:

$$\rho L d_s(t)\mathrm{d}t + d(t)\mathrm{d}t - \rho L d_n(t)\mathrm{d}t = \rho V \mathrm{d}d_n(t) \tag{6}$$

which is transformed into:

$$\frac{\mathrm{d}d_n(t)}{\mathrm{d}t} = \frac{d(t) + \rho n_1 q(1-\eta_d)[d_w(t) - d_n(t)]}{\rho V} = \frac{n_1 d_p(t)}{\rho V} + n_2(1-\eta_d)[d_w(t) - d_n(t)] \tag{7}$$

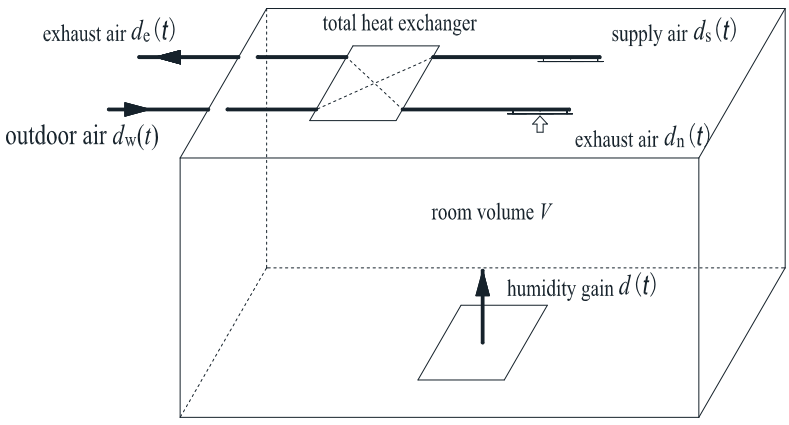


Figure 1. Indoor humidity ratio calculation model.

In the above equation, ρ is the air density, calculated at 1.205 kg/m^3; L is the room ventilation rate, m^3/h; d (t) is the humidity gain, g/h; d_n (t) is the indoor air humidity ratio, g/kg; d_w (t) is the humidity ratio of outdoor air, g/kg; n_1 is the number of people in the room; q is the per capita outdoor air volume, m^3/(h·person); d_p (t) is the per capita humidity gain, g/(h·person); V is the volume of the room, m^3; t is the time, h; and n_2 is the air change rate of the room, calculated as times per hour.

Substituting Equations (3)–(5) into Equation (7) yields:

$$\frac{\mathrm{d}d_n(t)}{\mathrm{d}t} + n_2\left\{1-\left\{1-\mathrm{e}^{-\frac{F_1}{\left\{\frac{2}{\alpha_d}+\frac{\delta_p}{D_{wn}}\frac{10^6[(1-C)\varphi_1+C]\cdot[(1-C)\frac{d_n}{14.88}+C]}{e^{(5294/T)}\omega_{max}C}\right\}g_f}}^{0.22}\cdot\left(\mathrm{e}^{-\frac{F_1}{\left\{\frac{2}{\alpha_d}+\frac{\delta}{D_{wn}}\frac{10^6[(1-C)\varphi_1+C]\cdot[(1-C)\frac{d_n}{14.88}+C]}{e^{(5294/T)}\omega_{max}C}\right\}g_f}}^{0.78}-1\right)\right\}\right\}[d_n(t)-d_w(t)]-\frac{d(t)}{\rho V}=0 \quad (8)$$

Assuming the operating period of the air conditioning system is [t_0, t_i], at time t_0, the indoor air humidity ratio d_n (t_0) is equal to the outdoor air humidity ratio d_w (t_0), that is, the initial conditions meet:

$$d_n(t_0) = d_w(t_0) \quad (9)$$

2.1.3. Outdoor Air Heating Load and Heat Recovery Calculation

The outdoor air treatment method for air-conditioned spaces is shown in Figure 2, which involves the following processes: (a) W→N; (b) W→O; (c) W→S→N; and (d) W→S→O′.

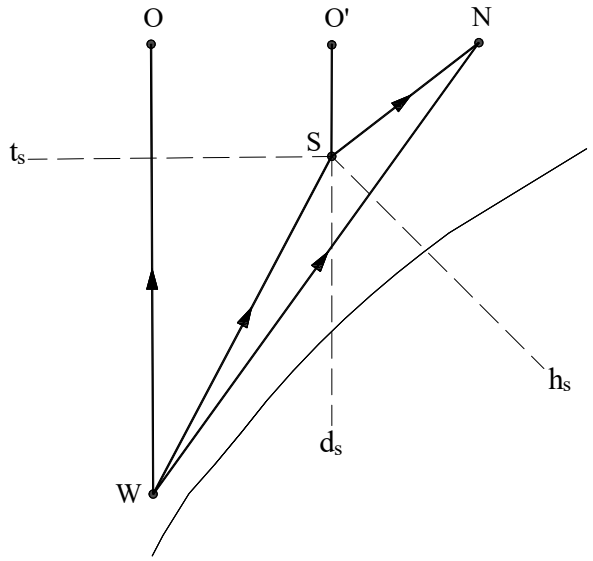


Figure 2. Psychrometric chart for outdoor air treatment.

W→N represents a room without total heat recovery and with strict temperature and humidity control. The indoor air status point is N, while the outdoor air status point for low temperature and low humidity in the winter is W. The unit mass outdoor air heating load is:

$$q_0 = h_n - h_w \quad (10)$$

In the above formula, q_0 is the outdoor air heating load, kw; h_n is the specific enthalpy of the indoor air state point, kJ/kg; and h_w is the outdoor air specific enthalpy, kJ/kg.

W→O refers to a room without total heat recovery, which generally controls temperature and not humidity. The low temperature and low humidity outdoor air in the winter, state W, is heated to point O, which is isothermal with the indoor air. The unit mass outdoor air heating load is:

$$q_0 = C_P(t_o - t_w) = h_o - h_w \quad (11)$$

In the above formula, q_0 is the outdoor air heating load, kw; C_p is the specific heat at constant pressure of air, J/(kg·K); t_o and t_w are the dry-bulb temperature of the final state of outdoor air treatment and outdoor air, °C; and h_o and h_w are the specific enthalpies of the final states of outdoor air treatment and outdoor air, kJ/kg.

W→S→N represents a room with total heat recovery and strict control of temperature and humidity. In winter, outdoor air in the W state is exchanged through heat and humidity to the S state and is then heated and humidified to the indoor air state point N. The heat recovery per unit mass of outdoor air in this process and the heating load after heat and humidity exchange are as follows:

$$q_h = h_s - h_w \tag{12}$$

$$q_{sh} = h_n - h_s \tag{13}$$

In the above formulae, q_h is the recovered energy of the total heat exchanger, kw; h_s is the specific enthalpy of supply air of the total heat exchanger, kJ/kg; q_{sh} is the outdoor air heating load after heat and humidity exchange, kw; and h_n is the specific enthalpy of the indoor air state point, kJ/kg. The meanings of other parameters are the same as before.

W→S→O′ represents an air-conditioned space with total heat recovery that does not control humidity. In winter, the outdoor air in the W state is exchanged through heat and humidity to the S state and is then heated to the same temperature as the indoor state point O′. The heat recovery per unit mass of outdoor air in this process is meticulously calculated using the preceding Equation (12) and the heating load after heat and humidity exchange is:

$$q_{sh} = C_P(t_{o'} - t_s) = h_{o'} - h_s \tag{14}$$

In the above formula, $h_{o'}$ is the final specific enthalpy of outdoor air treatment, kJ/kg. The meanings of the other parameters are the same as in previous equations.

At this time, the reduction value of the unit's mass outdoor air heating load is:

$$q_j = q_o - q_{sh} = C_P(t_o - t_w) - C_P(t_{o'} - t_s) = C_P(t_s - t_w) \tag{15}$$

The meanings of the parameters in the above equation are the same as previously stated.

One study [34] posits that relative indoor humidity should be between 30% and 60%, with the air being dry when it is lower than 30% and wet when it is higher than 60%. Therefore, when the total heat recovery is not adopted and the relative indoor humidity is higher than 30%, the latent heat recovery does not significantly improve the indoor thermal and humidity environment or reduce the air heating load. The effective heat recovery amount of the total heat exchanger is equal to the sensible heat recovery amount, which is the reduction value of the outdoor air heating load q_j. When the total heat recovery is not adopted and the relative indoor humidity is lower than 30%, although the air conditioning system does not control the humidity, latent heat recovery can improve the indoor thermal and humidity environment to a certain extent. When the temperature is 20 °C and the relative humidity is 30%, the indoor humidity ratio is 4.46 g/kg. If the humidity ratio of the supply air of the total heat exchanger is less than 4.46 g/kg, the total heat recovery can be regarded as effective heat recovery. If the humidity ratio of the supply air of the total heat exchanger is greater than 4.46 g/kg, the effective heat recovery can be calculated as the sum of the sensible heat recovery and the latent heat recovery when the humidity ratio of the outdoor air is 4.46 g/kg.

2.2. A Calculation Model for Total Heat Recovery in Air Conditioning Systems

This study uses Ningbo City as an example; it belongs to the north zone regarding building energy efficiency design in Zhejiang Province, and the heating calculation period used ranged from December 5th to February 20th of the following year [35]. Rooms in different functional buildings such as offices, schools, and commercial buildings were

selected as the research objects. The operating hours of indoor total heat exchangers were 8:00 to 18:00, 8:00 to 18:00, and 8:00 to 21:00, respectively.

(1) The desired indoor temperature in winter was 20 °C, and the relative humidity was not separately controlled. The room area, volume, occupant density, and desired outdoor air volume are shown in Table 1.

Table 1. Indoor parameters of air-conditioned rooms.

Functional Buildings	Occupant Density/(person/m^2)	Number of People	Per Capita Outdoor Air Volume/[m^3/(h·person)]	Room Area/m^2	Room Volume/m^3	Air Change Rate/(times/h)
office	0.125	32	30	256	768	1.302
classroom	0.6	40	24	67	201	4.975
commercial	0.25	50	19	200	600	1.667

(2) The indoor latent heat source mainly refers to the moisture from the breath and skin evaporation of occupants. At the desired indoor temperature, the per capita humidity gains, d_p, of air-conditioned spaces such as offices, classrooms, and businesses are 69 g/(h·person), 38 g/(h·person), and 134 g/(h·person), respectively [36]. Before operating an air conditioning system, there should be no people indoors, i.e., there should be no latent heat load.

(3) The outdoor air parameters were selected according to the typical meteorological year (TMY) parameters of Ningbo City [32]. During the heating period of air conditioning systems, a total of six typical days, namely December 6th, December 21st, January 5th, January 20th, February 4th, and February 19th, were selected for this research. The fitting formula for the instantaneous variation of outdoor air humidity ratio d_w (t) from 1:00 to 24:00 is shown in Table 2.

Table 2. Fitting formula for outdoor air humidity ratio d_w (t).

Date	Fitting Formula
December 6th	$d_w(t) = -5.680 \times 10^{-5}t^4 + 2.036 \times 10^{-3}t^3 - 8.603 \times 10^{-3}t^2 - 0.165t + 6.989$, $R^2 = 0.91$
December 21st	$d_w(t) = -7.844 \times 10^{-6}t^4 - 8.085\times10^{-4}t^3 + 4.329 \times 10^{-2}t^2 - 0.419t + 6.158$, $R^2 = 0.98$
January 5th	$d_w(t) = -1.044 \times 10^{-5}t^4 + 4.395\times10^{-4}t^3 - 1.681 \times 10^{-4}t^2 - 0.200t + 6.288$, $R^2 = 0.98$
January 20th	$d_w(t) = 4.211 \times 10^{-5}t^4 - 2.134\times10^{-3}t^3 + 3.639 \times 10^{-2}t^2 - 0.186t + 3.487$, $R^2 = 0.95$
February 4th	$d_w(t) = -8.057 \times 10^{-5}t^4 + 3.606\times10^{-3}t^3 - 4.975 \times 10^{-2}t^2 + 0.278t + 0.962$, $R^2 = 0.92$
February 19th	$d_w(t) = -1.356 \times 10^{-5}t^4 + 2.909\times10^{-4}t^3 + 4.766 \times 10^{-3}t^2 - 0.088t + 4.400$, $R^2 = 0.95$

(4) The geometric dimensions of the plate heat recovery equipment structure and the heat and mass transfer performance coefficients of the heat transfer film [26,30] are shown in Table 3, with a rated outdoor air volume and exhaust air volume of 1000 m^3/h.

Table 3. Structure and material parameters of a plate heat recovery equipment.

l/m	h/m	r/m	δ_p/m	δ_f/m	a/m	b/m	λ/(w·m/K)	D_{wn}/(m^2/s)	ω_{max}/(kg/kg)	C	α_d/(kg·m^{-2}·s)
0.25	1.20	0.002	0.00006	0.0002	0.00023	0.00023	0.1	8.00×10^{-8}	0.70	6.5	0.05

where l is core body side length; h is core thickness; r is film spacing; δ_f is fin thickness; a is the bottom length of the fin channel; b is the waist length of the fin channel; and λ is the thermal conductivity of thin films and fins. The meanings of other parameters are the same as before.

(5) The hourly occupancy rate of people in offices, classrooms, and commercial rooms [1] is shown in Table 4.

Table 4. Hourly occupancy rate of people in different functional rooms.

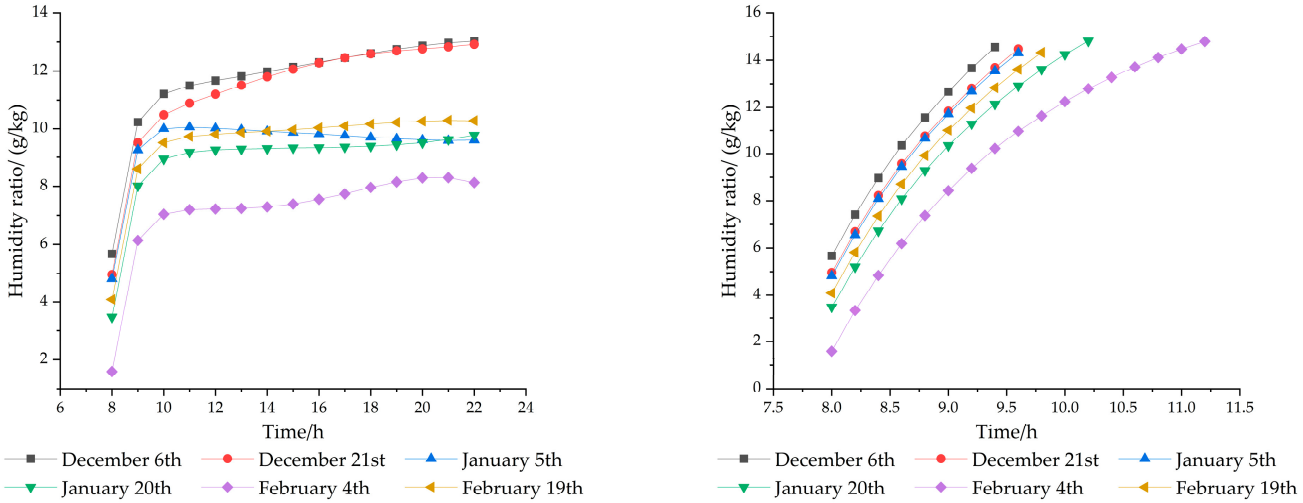
Functional Rooms	Fitting Formula
Office	$u(t) = -5611.99301 + 1884.30070t - 228.87238t^2 + 12.11538t^3 - 0.23601t^4$, $R^2 = 0.92$
Classroom	$u(t) = -5611.99301 + 1884.30070t - 228.87238t^2 + 12.11538t^3 - 0.23601t^4$, $R^2 = 0.92$
Commercial	$u(t) = -1845.35310 + 538.32067t - 55.55477t^2 + 2.51037t^3 - 0.04195t^4$, $R^2 = 0.97$

3. Results and Discussion

Utilizing the pertinent parameters outlined in Tables 1–4, MATLAB R2016a software was employed to solve the humidity balance differential equation. This enabled us to determine the hourly indoor humidity ratio, relative humidity, heat recovery efficiency, and heat recovery amount in air-conditioned spaces such as offices, classrooms, and commercial buildings in winter, where a total heat exchanger had been installed. Notably, any calculation results pertaining to relative air humidity exceeding 100% were omitted, as they fell outside the applicable range of Equations (3) and (4).

3.1. Calculation of the Indoor Relative Humidity

Figures 3 and 4, respectively, show how humidity and relative humidity in commercial rooms change over time. The calculation reveals that in the absence of total heat recovery, the indoor humidity ratio experiences a rapid surge within the first 2 h and subsequently maintains a gradual yet steady change. Specifically, under the calculation conditions of February 4th, the relative indoor humidity primarily fluctuates between 41% and 56%, whereas under other conditions, it surpasses 60%, a notably high humidity level. Conversely, when total heat recovery is implemented, the indoor humidity ratio undergoes a swift increase within a timeframe of 1.5 to 3.5 h, ultimately attaining saturation. This transformation underscores the profound influence of total heat recovery on indoor humidity, enabling the humidity ratio to reach a higher value within a shorter timeframe.

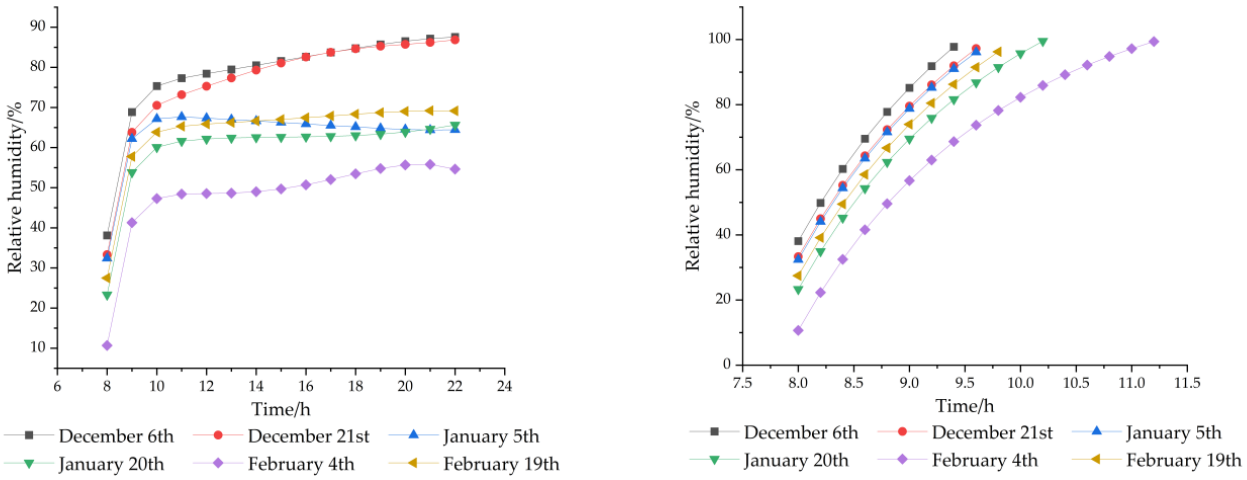


(**a**) Total heat exchanger not implemented. (**b**) Total heat exchanger implemented.

Figure 3. Humidity ratio curves in commercial buildings.

From Equation (7), it can be seen that when the outdoor air volume per capita satisfies $\frac{\mathrm{d}d_n(t)}{\mathrm{d}t} > 0$, namely $q < \frac{d_p(t)}{\rho(1-\eta_d)[d_n(t)-d_w(t)]}$, the indoor humidity ratio increases. Assuming a winter indoor temperature of 20 °C, an indoor air relative humidity upper limit of 60%, an outdoor air temperature of −1.5 °C, and a relative outdoor air humidity of 79%, the estimated average absolute humidity ratio exchange effectiveness of the total heat exchanger stands at 60%. Under these circumstances, the required outdoor air volume to maintain humidity balance is 44 $m^3/(h \cdot person)$. However, it is noteworthy that during the heating

period, outdoor humidity is typically higher than the designated outdoor air humidity. Therefore, the per capita outdoor air volume necessary to uphold the indoor humidity surpasses the recommended value stipulated in the specification. Consequently, under the specified calculation conditions, the implementation of total heat recovery in commercial spaces during winter is not advisable. In addition, latent heat recovery could potentially exacerbate the indoor thermal and humidity environment, thereby compromising overall energy-saving efficiency. The authors of [37,38] explicitly underscore that in environments characterized by high humidity, the bypass control mode of the total heat recovery equipment offers distinct advantages. In this mode, air bypasses the heat and moisture exchange core, thereby further minimizing ventilation energy consumption.



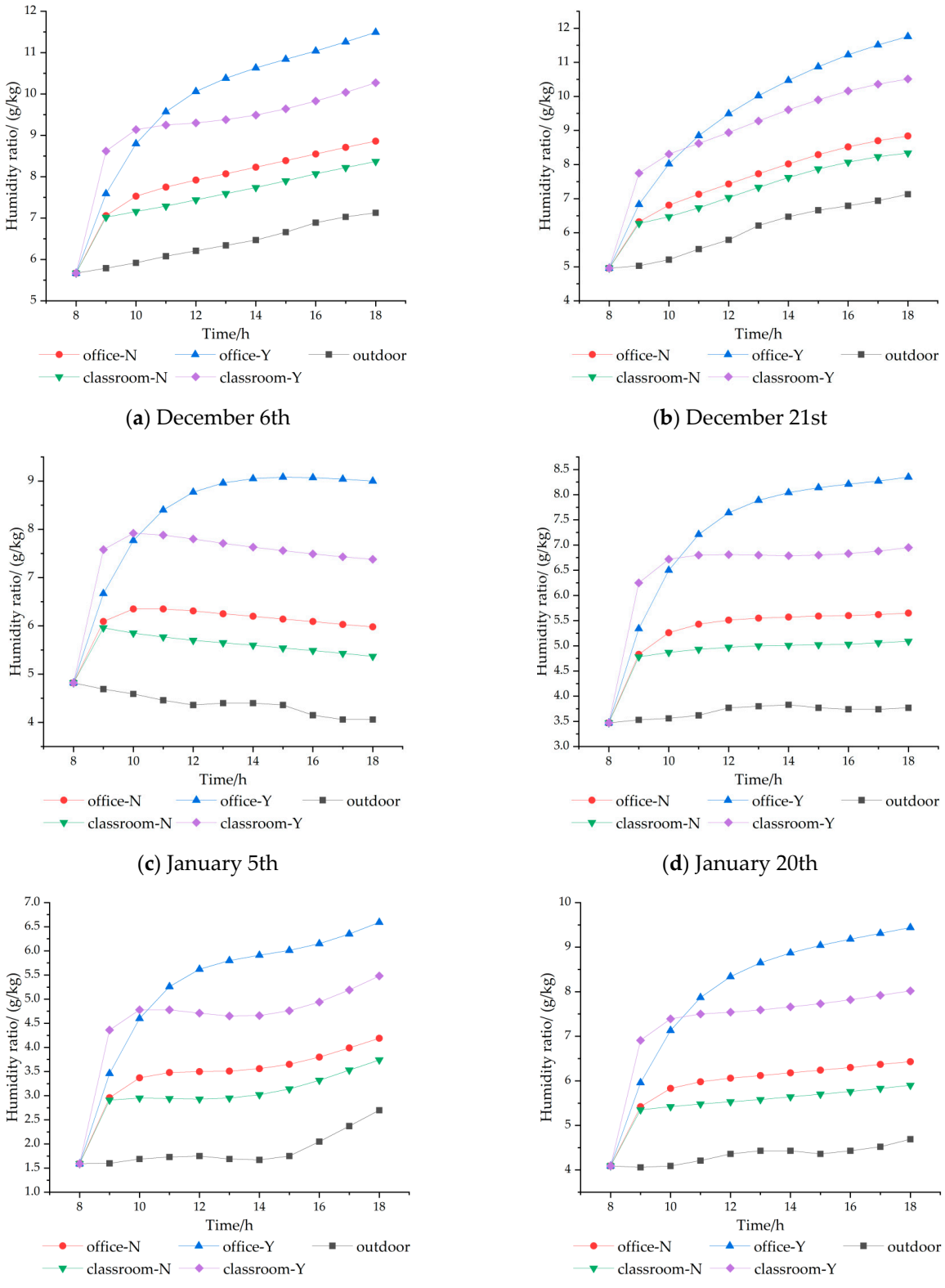
(**a**) Total heat exchanger not implemented. (**b**) Total heat exchanger implemented.

Figure 4. Relative humidity curves in commercial buildings.

Figure 5 shows how the humidity ratio of offices and classrooms changes over time. Specifically, the labels "office-N" and "classroom-N" signify scenarios where total heat recovery is not implemented, whereas "office-Y" and "classroom-Y" indicate instances where total heat recovery is in use. It is noteworthy that, within 3 h and 1 h, respectively, of people occupying these spaces, the humidity ratio in office rooms and other functional areas that utilized total heat recovery underwent a rapid increase. Subsequently, these ratios gradually maintain a trend similar to the outdoor humidity ratio.

When considering identical ventilation rates, the change rate of the indoor humidity ratio is influenced by various factors. These include the room's volume, the absolute humidity ratio exchange effectiveness of heat recovery, the absolute humidity ratio exchange effectiveness, the difference in indoor and outdoor humidity, and the room's humidity load. In particular, a smaller room volume and a smaller difference in indoor and outdoor humidity ratio, coupled with a higher humidity load and absolute humidity ratio exchange effectiveness, results in a more significant change in the indoor humidity ratio. The observed order of humidity ratio changes is commercial > classrooms > offices, which aligns with the calculation results presented in Figures 3 and 5.

Figure 6 shows how the relative humidity of offices and classrooms changes over time. Unlike the notable fluctuations observed in relative outdoor humidity during the evaluation period, the relative indoor humidity of these spaces exhibits a distinct pattern. Initially, within the first 1 to 3 h, a rapid increase in relative humidity was observed, followed by a gradual stabilization in the rate of change.



(**a**) December 6th (**b**) December 21st

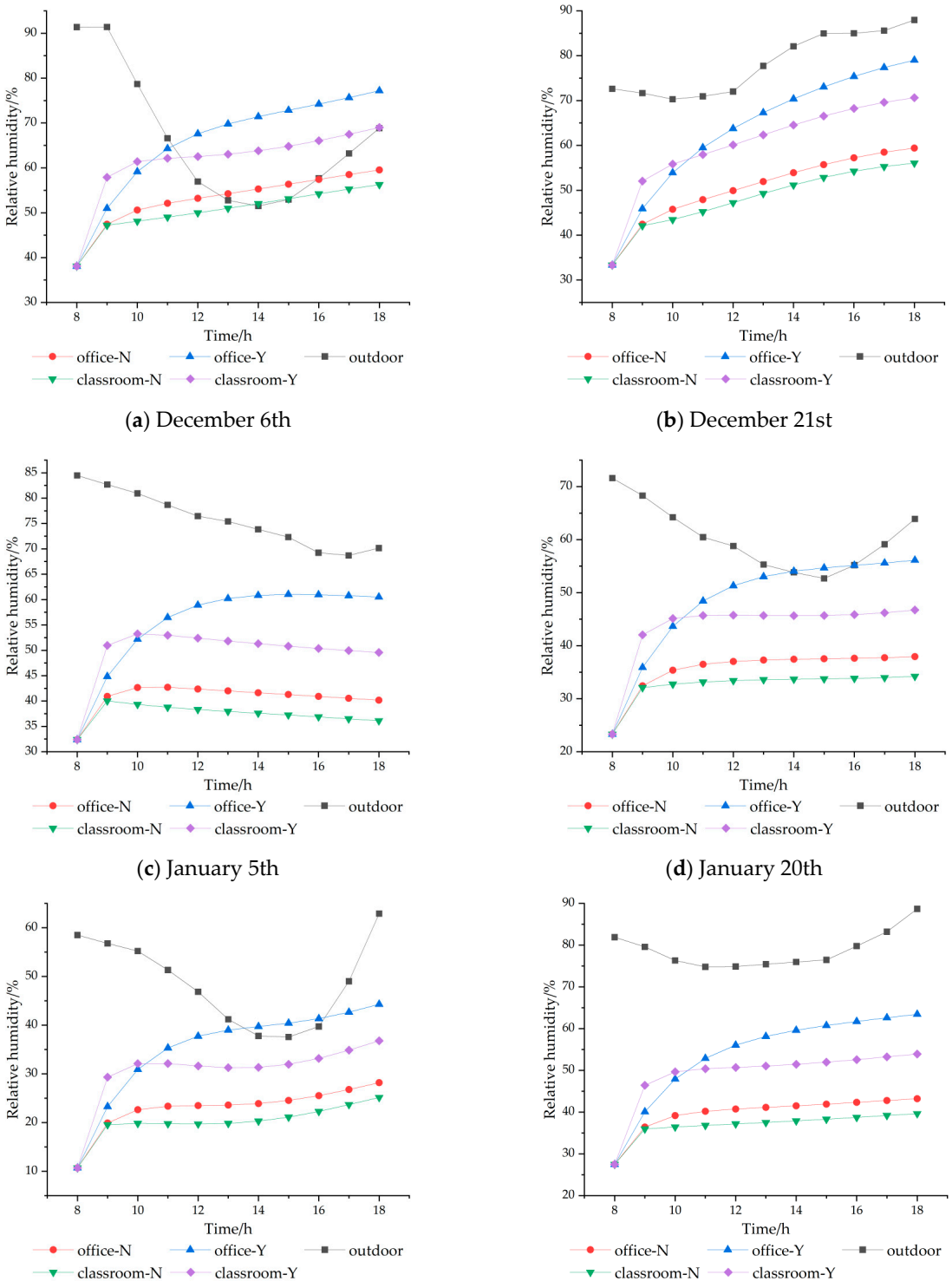
(**c**) January 5th (**d**) January 20th

(**e**) February 4th (**f**) February 19th

Figure 5. Humidity ratio curves of offices and classrooms.

Notably, February 4th stands out as the day with the lowest daily average temperature among the typical annual meteorological parameters. Consequently, both the outdoor humidity ratio and relative humidity were significantly lower compared to other conditions. In the absence of total heat recovery, the relative humidity levels in offices and

classrooms were particularly low, dipping below 30%, resulting in a noticeably dry indoor environment. However, the installation of a total heat exchanger significantly improved this situation, with relative indoor humidity stabilizing primarily between 30% and 44%. This underscores the beneficial impact of latent heat recovery on enhancing indoor thermal and humidity comfort.



(**a**) December 6th (**b**) December 21st

(**c**) January 5th (**d**) January 20th

(**e**) February 4th (**f**) February 19th

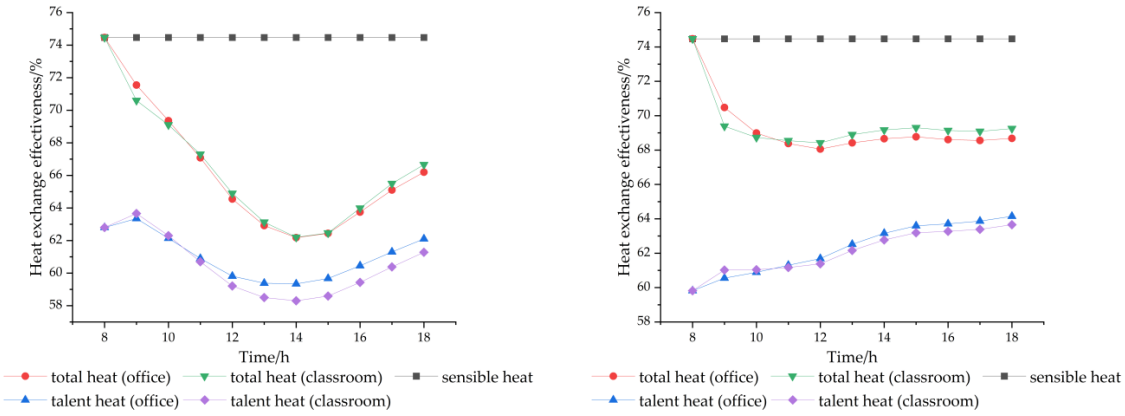
Figure 6. Relative humidity curves of offices and classrooms.

Under the calculation conditions, the relative humidity levels within offices and classrooms, without total heat recovery, primarily ranged between 32% and 63%. However, with the implementation of total heat recovery, the relative humidity on December 6th and December 21st exceeded 60%. For the remaining three conditions, the relative indoor humidity was maintained within a range of 42% to 63%.

It is worth noting that, given identical outdoor air parameters and heat recovery equipment configurations, a consistent trend was observed in the overall relative indoor humidity levels. Specifically, the relative humidity within commercial spaces tended to be the highest, followed by offices and then classrooms.

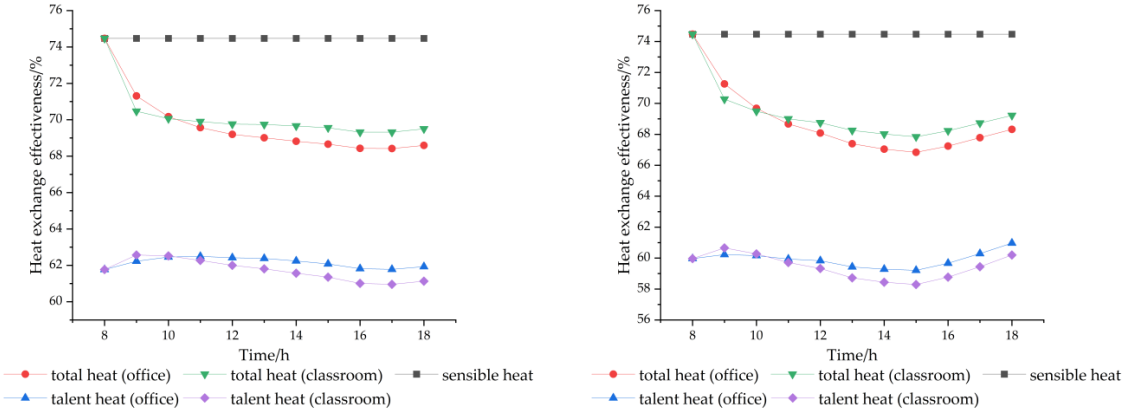
3.2. Calculation of Heat Recovery Efficiency

Figure 7 illustrates that the sensible exchange effectiveness of the total heat exchanger remains steady at the desired flow rate. This effectiveness is solely dependent on the number of heat transfer units and remains unaffected by variations in indoor and outdoor air parameters. Initially, when the total heat exchanger commences operation, there is no disparity in humidity ratio between indoor and outdoor air, resulting in the total exchange effectiveness being equivalent to the sensible exchange effectiveness. However, as the difference in humidity ratio between indoor and outdoor air gradually increases, the total exchange effectiveness decreases accordingly. Once a quasi-steady state is reached, with minimal changes in the humidity ratio difference, the absolute humidity ratio exchange effectiveness of heat recovery becomes primarily influenced by relative humidity. Additionally, the total exchange effectiveness of heat recovery is also influenced by the temperature difference between indoor and outdoor air.



(**a**) December 6th

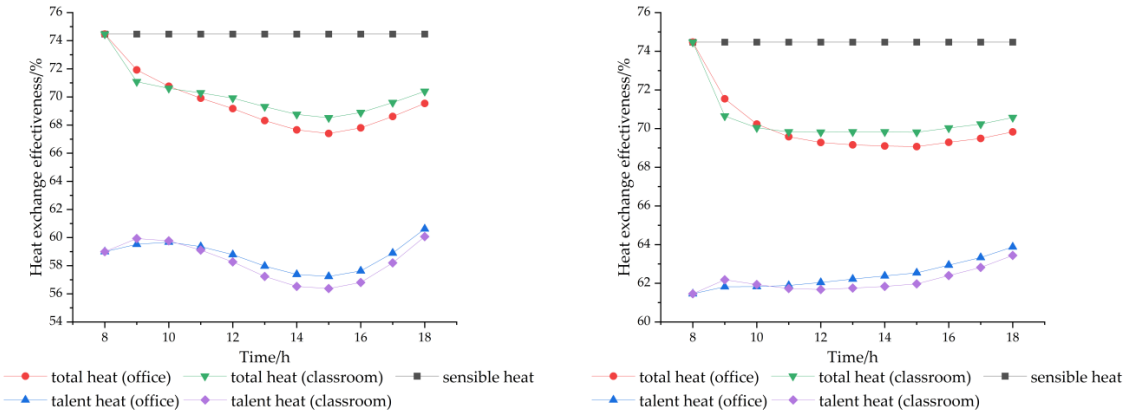
(**b**) December 21st



(**c**) January 5th

(**d**) January 20th

Figure 7. *Cont.*



(**e**) February 4th (**f**) February 19th

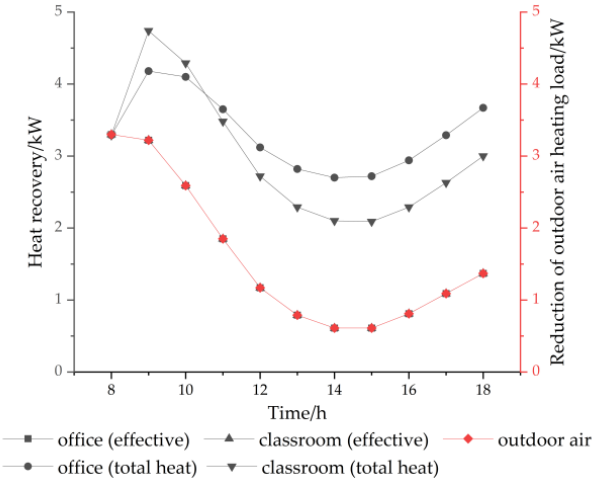
Figure 7. Heat recovery efficiencies of total heat exchanger in offices and classrooms.

During the calculation period, significant fluctuations were observed in the total exchange effectiveness of heat recovery. Taking the office room on December 6th as a representative example, the total exchange effectiveness of heat recovery exhibited significant variations, decreasing from 75% at 8:00 to a low of 62% around 14:00 and subsequently increasing to 66% by 18:00. Notably, under different typical daily calculation conditions, the exchange effectiveness differed significantly. Overall, the relative humidity of the air was higher in office spaces than in classrooms. Consequently, the absolute humidity ratio exchange effectiveness of heat recovery was also higher in offices compared to classrooms, while the trends for total exchange effectiveness were the opposite.

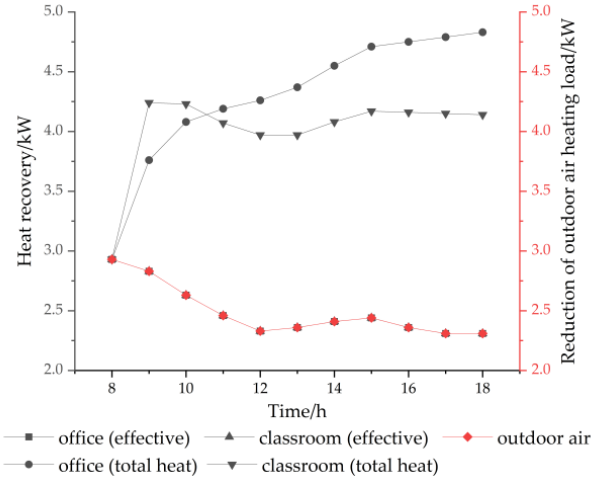
3.3. Heat Recovery Calculation

As shown in Figure 8, initially, the indoor-to-outdoor humidity ratios align, resulting in the total heat recovery being equivalent to the sensible heat recovery. Specifically, under the working conditions observed on February 4th, the effective heat recovery in both office and classroom settings is comparable or closely aligned with the total heat recovery. However, under other calculation conditions, the effective heat recovery matches the sensible heat recovery, representing the reduction in the outdoor air heating load. The sensible heat recovery is primarily influenced by the indoor–outdoor temperature difference, exhibiting an inverse relationship with outdoor temperature.

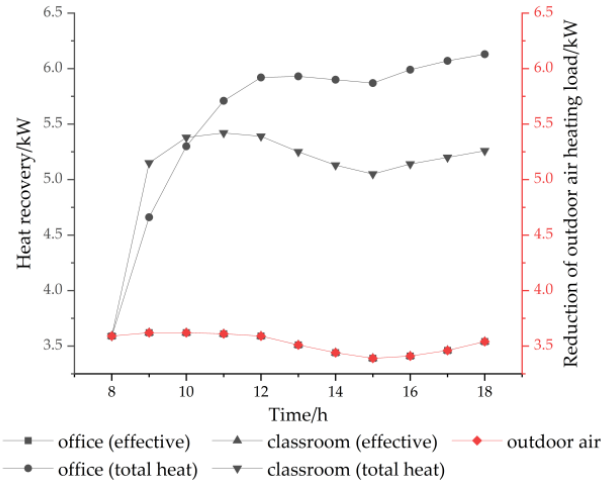
Typically, the energy-saving performance of a total heat exchanger is evaluated using the coefficient of performance COP_{rhs} of the heat recovery equipment. This coefficient compares the effective heat recovery of the device to the power consumption of the additional fan [39], i.e., $COP_{rhs} = \Delta Q/P$, where ΔQ represents the effective heat recovery, kW, and P represents the power consumption of the additional fan of the total heat exchanger, kW. According to established standards [35], the power consumption per unit air volume W_s should not exceed 0.45 w/(m^3/h) for total heat exchangers, 0.19 w/(m^3/h) for outdoor air systems, and 0.22 w/(m^3/h) for exhaust systems. Given this, when comparing with the combined operation scheme of the outdoor air system and the exhaust system, the power consumption P of the additional fan of the two-way ventilation system of total heat exchanger with rated air rate L = 1000 m^3/h is calculated as 0.49 kW, and the performance coefficient COP_{rhs} of total heat exchanger under various calculation conditions is shown in Figure 9.



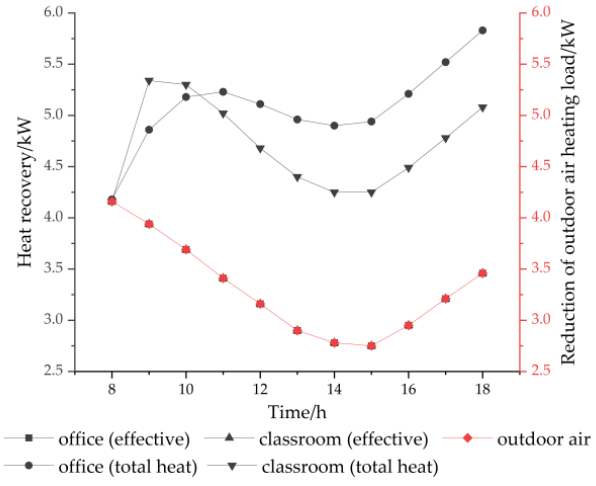
(**a**) December 6th



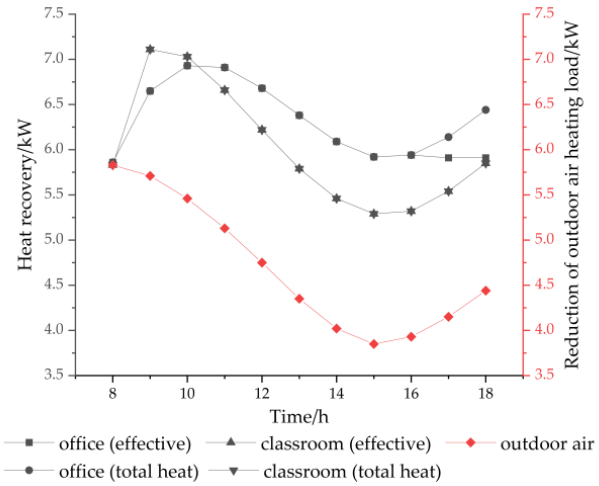
(**b**) December 21st



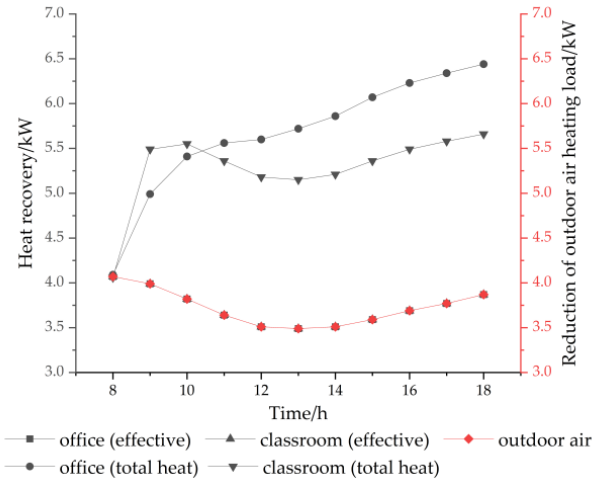
(**c**) January 5th



(**d**) January 20th



(**e**) February 4th



(**f**) February 19th

Figure 8. Office and classroom heat recovery and outdoor air load changes.

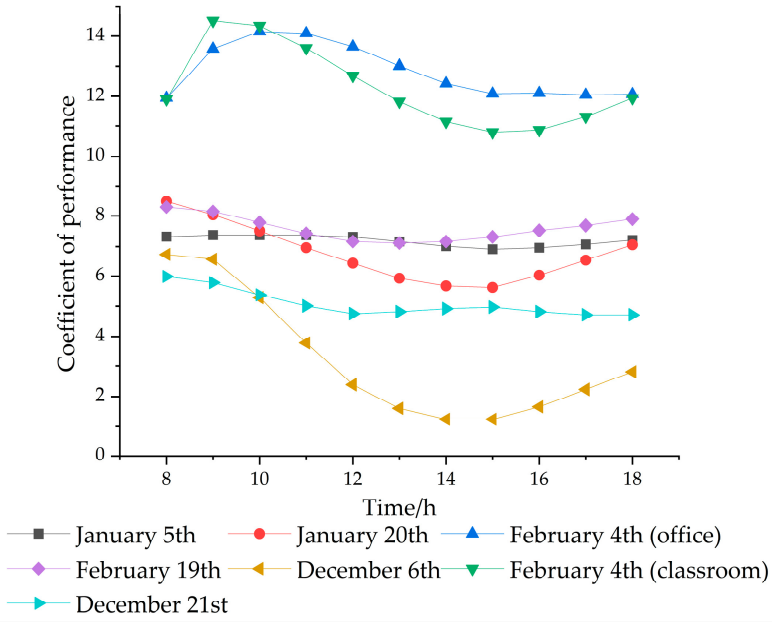


Figure 9. Coefficients of total heat exchanger performance.

Figure 9 demonstrates that the coefficients of the COP_{rhs} performance ranged from 1.2 to 2.8 during most periods on December 6th but exceeded 4.7 on the other days. This suggests that the heat recovery technology employed by the total heat exchanger was energy-efficient on the selected six typical days, excluding certain periods on December 6th. In contrast to utilizing the economic specific enthalpy difference as the activation and deactivation control signal for the total heat exchanger during summer, it is advisable to assess the energy-saving potential of the exchanger based on the temperature differentials between indoor and outdoor air in winter. This temperature difference can serve as a reliable activation and deactivation control signal for the equipment.

The total heat recovery and effective heat recovery in a period centered on the hour of i are approximately represented by Q_i and ΔQ_i, respectively, and measured as kW. The total heat recovery in the calculation period of air-conditioned offices and classrooms is:

$$Q = \sum_{i=9}^{17} Q_i + \frac{1}{2}(Q_8 + Q_{18}) \tag{16}$$

The effective heat recovery is:

$$\Delta Q = \sum_{i=9}^{17} \Delta Q_i + \frac{1}{2}(\Delta Q_8 + \Delta Q_{18}) \tag{17}$$

The effective heat recovery rate is defined as $\eta = \Delta Q/Q$, and its value is shown in Figure 10.

Figure 10 clearly illustrates that the effective heat recovery rate η of the total heat exchanger ranges from 46% to 70% under various calculation conditions, except on February 4th, where the effective heat recovery amount closely matches or equals the total heat recovery amount. Notably, the exchanger reached its lowest effective heat recovery rate on December 6th and its peak on February 4th. This discrepancy can be attributed to the relatively high outdoor air temperature and relative indoor–outdoor humidity on December 6th, leading to a higher absolute humidity ratio exchange effectiveness and a correspondingly lower sensible heat recovery amount, resulting in a relatively low effective heat recovery rate. Conversely, the conditions on February 4th were the opposite, explaining the higher effective heat recovery rate observed on that day.

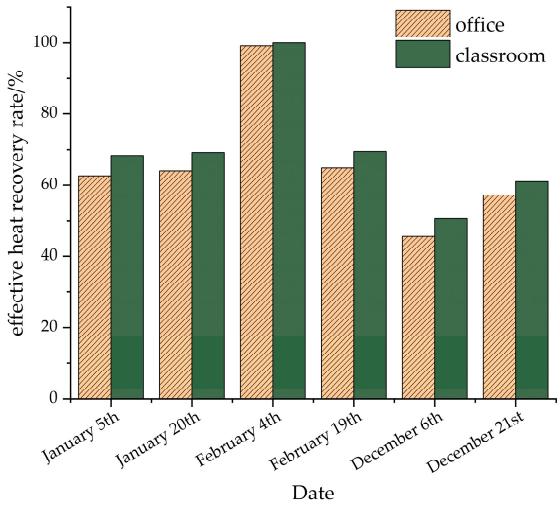


Figure 10. Effective heat recovery rate of total heat exchanger.

Previous research [31] has emphasized that in the interiors of buildings in winter, heat recovery does not always equate to effective heat recovery, as its effectiveness depends on the heating requirements of the air in a given room. Similarly, the same logic applies to the recovered latent heat. If the room environment does not necessitate humidification, the recovered latent heat should not be factored into the effective heat recovery calculation. Only when latent heat recovery genuinely reduces the load on the air conditioning system can it be deemed effective heat recovery. Therefore, when conducting energy-saving calculations, it is imperative to meticulously distinguish between heat recovery and effective heat recovery.

Zhang et al. [40,41] conducted a thorough analysis of the changes in room humidity when implementing a conventional outdoor air system without humidity control. Smith et al. [37] further delved into the characteristics of indoor humidity within environments utilizing rotary heat exchangers. During the calculation process, for non-hygroscopic rotary heat exchangers, it was postulated that all condensed moisture in the exhaust subsequently evaporated into the supply air. Meanwhile, for fully hygroscopic rotary heat exchangers, the assumption was made that the sensible exchange effectiveness aligned with the absolute humidity ratio exchange effectiveness. However, when compared to rotary heat exchangers, the plate heat recovery equipment exhibited significantly more complex characteristics in terms of moisture transfer and absolute humidity ratio exchange effectiveness. Despite Wang and Zhong's comprehensive elucidation of the moisture transfer mechanisms within the plate heat recovery equipment core and the intricacies of absolute humidity ratio exchange effectiveness calculations [26,27], the coupling effect between indoor humidity and absolute humidity ratio exchange effectiveness renders the calculation process extremely intricate. Therefore, previous studies [2,11–15,25,30], when estimating the humidity and heat recovery within air-conditioned spaces, often overlooked the dynamic fluctuations in indoor humidity and their consequential impact on absolute humidity ratio exchange effectiveness, ultimately leading to errors in calculations.

This enhanced approach holds immense significance for the study of indoor thermal and humidity environments, for accurate energy-saving calculations for outdoor air systems, and for the formulation of operational strategies for total heat recovery equipment. This work has achieved the following results:

(1) Based on the calculation of the moisture transfer mechanism and dynamic heat recovery efficiency of the plate heat recovery equipment, special attention was paid to the influence of the change in the indoor air state on the equipment, so as to more accurately evaluate the dynamic characteristics of the absolute humidity ratio exchange effectiveness.

(2) With the help of MATLAB software, the complex coupling effect between indoor air parameters and absolute humidity ratio exchange effectiveness was systematically considered, which provides an effective numerical analysis method for the optimization calculation of heat recovery efficiency.
(3) An accurate calculation method of the indoor air state when a total heat exchange device is installed was proposed, which provides a scientific basis for the determination of indoor thermal and humidity environmental parameters in air-conditioned rooms.
(4) The internal relationship between total heat recovery and effective heat recovery was explained and a corresponding mathematical model was established, which provides more accurate theoretical support and a practical reference for energy-saving calculations.

In this study, we conducted thorough research into the operational performance of plate heat recovery equipment using six representative typical days as case studies, marking a significant step towards accurate heat recovery calculations. Nevertheless, it is crucial to acknowledge that future research holds immense potential and vast opportunities. Specifically, we envision the integration of the computational methods proposed in this paper into HVAC system performance simulation tools, such as Energy Plus, to enable dynamic and precise calculations of heat recovery throughout the winter season. The implementation of such a methodology would significantly enhance computational efficiency, providing more accurate and efficient solutions for practical engineering applications.

However, it is worth noting that the hourly meteorological data from typical meteorological years utilized in this research are primarily intended for building energy consumption calculations and assessments. These data, being based on the statistical analyses of meteorological parameters over the years, lack the conditions for direct measurement and comparison with simulated calculation results. Future studies should focus on the comparative study of the theoretical calculation and actual measurement results under the actual meteorological parameters, so as to reveal the correlation and difference between the two more deeply. In addition, determining parameters such as occupant density, hourly occupancy rates, and per capita humidity gains in different buildings remains a crucial research direction. These parameters are fundamental to energy-saving calculations and directly influence the reliability of energy consumption calculations and assessments.

4. Conclusions

According to the humidity balance equation and outdoor air load analysis of air-conditioned spaces without humidity control, taking Ningbo City as an example, the relative indoor humidity and effective heat recovery of air-conditioned spaces in offices, classrooms, and commercial buildings in winter in terms of exhaust heat recovery amount were predicted by taking the typical daily meteorological parameters in typical years.

(1) The humidity ratio and relative humidity of the air in offices and classrooms generally reach a quasi-steady state equilibrium within 1~3 h of the operation of the total heat exchanger. Under the calculated six typical daily working conditions, the relative indoor humidity is generally maintained above 30% when the total heat exchanger is used. Under the calculated working conditions, the relative humidity of the rooms with a large per capita humidity gain, such as spaces in commercial buildings, can rapidly increase to more than 60% or can even reach saturation. On the other hand, latent heat recovery is not conducive to hot and humid indoor environments and energy saving, and it is not suitable to use total heat recovery.
(2) Relative indoor humidity depends on the absolute humidity ratio exchange effectiveness of heat recovery, the indoor and outdoor humidity difference, and the room humidity gain. According to the room humidity balance equation and the relative humidity limit, the required per capita outdoor air rate can be determined, which can be compared with the recommended value of the specification and used as the basis for judging the suitability of the total heat recovery technology.
(3) The effective heat recovery amount of a total heat exchanger cannot be calculated as the total heat recovery amount. Under the calculation conditions, the effective heat

recovery rate of the total heat exchanger is between 46% and 100%, which depends on the indoor and outdoor air temperature difference, the humidity difference, and the relative humidity value. For working conditions or areas with high outdoor temperatures and humidity, the effective heat recovery amount and total heat recovery amount of total heat exchangers differ greatly in winter, which should be taken into account in the suitability analysis of total heat exchangers.

(4) Compared to rooms with general outdoor air systems in winter, there is a coupling relationship between relative indoor air humidity and absolute humidity ratio exchange effectiveness when utilizing total heat exchangers. The calculation method provided in this paper can accurately calculate the indoor air state and effective heat recovery amount with total heat recovery and improve the accuracy of the suitability evaluation of total heat recovery.

(5) The calculation results are based on the calculation conditions. In view of the differences in the actual occupant density, hourly occupancy rate of individuals, per capita humidity gain, and other factors in different buildings, the above parameters should be determined by analyzing the indoor humidity and the heat recovery of specific buildings.

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Conflicts of Interest: The authors declare no conflicts of interest.

Nomenclature

Variables

η_h	Total exchange effectiveness, %
h_w	Enthalpy of outdoor air, kJ/kg
h_s	Enthalpy of supply air, kJ/kg
h_n	Enthalpy of exhaust air, kJ/kg
η_t	Sensible exchange effectiveness, %
η_d	Absolute humidity ratio exchange effectiveness, %
d_w	Humidity ratio of outdoor air, g/kg
d_s	Humidity ratio of supply air, g/kg
d_n	Humidity ratio of exhaust air, g/kg
t_w	Temperature of outdoor air, °C
t_n	Temperature of exhaust air, °C
NTU_D	Number of mass transfer units of total Heat exchanger
k_d	Total mass transfer coefficient, kg/(m^2·s)
F_1	Primary heat transfer area of the total Heat exchanger, m^2
g_f	Mass flow rate, kg/s
α_d	Convective mass transfer coefficient, kg/(m^2·s)
δ_p	Thickness of the wet exchange film, m
D_{wn}	Mass diffusion coefficient, kg/(m·s)
C	Calculated value corresponding to the adsorption type of the wet exchange film
φ_1	Relative humidity of outdoor air, %
φ_2	Relative humidity of indoor air, %
T	Average indoor and outdoor temperature, K
ω_{max}	Maximum water capacity of the film, kg/kg

d_{sa}	Humidity ratio of saturated air, g/kg
ρ	Air density, kg/m^3
L	Room ventilation rate, m^3/h
d (t)	Humidity gain, g/h
n_1	Number of people in the room
q	Per capita outdoor air volume, m^3/(h·person)
d_p (t)	Per capita humidity gain, g/(h·person)
V	Volume of the room, m^3
t	Time, h
n_2	Air change rate of the room, times per hour
q_0	Outdoor air heating load, kw
C_p	Specific heat at constant air pressure, J/(kg·K)
t_o	Dry-bulb temperature of the final state of outdoor air treatment, °C
h_o	Specific enthalpy of the final state of outdoor air treatment, kJ/kg
q_h	Recovered energy of the total heat exchanger, kw
h_s	Specific enthalpy of supply air of the total heat exchanger, kJ/kg
q_{sh}	Outdoor air heating load after heat and humidity exchange, kw
$h_{o'}$	Final specific enthalpy of outdoor air treatment, kJ/kg
Q_i	Total heat recovery centered on the hour of i
ΔQ_i	Effective heat recovery centered on the hour of i

References

1. *GB55015-2021*; Ministry of Housing and Urban-Rural Development of the People's Republic of China. General Code for Energy Efficiency and Renewable Energy Application in Buildings. China Architecture & Building Press: Beijing, China, 2021; pp. 2–73.
2. Wang, F.; Lee, M.; Chang, T.; Chen, Y.; Jung, R. Improving Indoor Air Quality and Thermal Comfort by Total Heat Exchanger for an Office Building in Hot and Humid Climate. *HVACR Res.* **2014**, *20*, 731–737. [CrossRef]
3. Jeon, B.-H.; Yang, S.-K.; Ahn, Y.-C.; Kang, Y.-H. Policies and Energy Efficiency of Heat Recovery Ventilators in South Korea. *Energies* **2023**, *16*, 7539. [CrossRef]
4. Men, Y.; Xiao, H.; Zhang, T. Experimental and Numerical Analysis on Heat and Moisture Recovery Performance of Enthalpy Wheel with Condensation. *Energy Convers. Manag.* **2021**, *246*, 114683. [CrossRef]
5. Fan, H.; Chen, L. Analysis of Heat and Mass Exchange Performance of Enthalpy Recovery Wheel. In Proceedings of the 11th International Symposium on Heating, Ventilation and Air Conditioning (ISHVAC 2019), Harbin, China, 12–15 July 2019; Springer: Singapore, 2020.
6. Liu, P.; Justo Alonso, M.; Mathisen, H. Global Sensitivity Analysis and Optimal Design of Heat Recovery Ventilation for Zero Emission Buildings. *Appl. Energy* **2023**, *329*, 120237. [CrossRef]
7. Di, Y.; Di, H.; Zhao, C.; Lu, Z.; Li, X.; Leng, J. An Experimental Study on Heat Recovery Performances of Three-Dimensional Heat Pipes in Air-Conditioning Systems. *Buildings* **2024**, *14*, 355. [CrossRef]
8. Zender-Świercz, E. A Review of Heat Recovery in Ventilation. *Energies* **2021**, *14*, 1759. [CrossRef]
9. Lu, Y.; Xu, W.; Sun, D.; Wang, S. Research on Calculation Method of Thermal Load of Outdoor Air for Passive Residential Building with Ultra-low Energy Consumption. *Build. Sci.* **2015**, *31*, 80–85.
10. Wei, D.; Tian, J.; Hao, C.; Tian, S.; Liu, S. Study on Fresh Air System for Nearly Zero Energy Residential Buildings in Severe Cold and Cold Regions. *Build. Sci.* **2019**, *35*, 102–108.
11. Cheng, S.; Wu, Z.; Qin, M. Study on Energy Saving Performance of Heat-Recovery-Ventilation System in Residential Buildings of Hot-Summer and Cold-Winter Areas in China in Summer. *Build. Sci.* **2018**, *34*, 25–31.
12. Duan, F. Performance of Exhaust Heat Recovery in Residential Building at Night. *J. Cent. South Univ. (Sci. Technol.)* **2015**, *46*, 1110–1117.
13. Zhou, Y.; Wu, J.; Wang, R. Performance Comparison of Energy Recovery Ventilator with Different Indoor and Outdoor Air Parameters. *Acta Energiae Solaris Sin.* **2008**, *29*, 198–203.
14. Wang, Q.; Liu, J.; Zhong, S.; Wang, X. Energy Saving Analysis of Exhaust Heat Recovery Technology in Beijing Library. *Cryog. Refrig.* **2022**, *51*, 55–59.
15. Yang, C.; Chen, D. Economic Analysis on Heat Recovery Systems from Exhauster Air. *J. Hunan Univ. (Nat. Sci.)* **2009**, *36*, 103–108.
16. Chen, J.; Hong, S.; Han, K. Experimental Study on the Performance of Graphene Plate-Fin Total Heat Exchanger. *Fluid Mach.* **2023**, *51*, 1–8.
17. Zhang, C. *The Experimental Study on the Performance of Plate-Fin Enthalpy Exchanger Cores*; Shanghai Jiaotong University: Shanghai, China, 2017.
18. Gao, A.; Yang, Y.; Jia, C.; Cui, J.; Wang, B.; Yan, Y. Polyethylene/Poly (N,N-Dimethyl Acrylamide) Hybrid Hydrogel Membrane for Total Heat Exchanger. *Chem. Eng. J.* **2023**, *470*, 143954. [CrossRef]

19. Cho, H.-J.; Cheon, S.-Y.; Jeong, J.-W. Energy Saving Potential of Latent Heat Exchanger-Integrated Dual Core Energy Recovery Ventilator. *Appl. Therm. Eng.* **2023**, *231*, 120989. [CrossRef]
20. Men, Y.; Liang, C.; Li, Z.; Tong, X. Configuration Optimization of a Membrane-Based Total Heat Exchanger with Cross-Corrugated Triangular Ducts Considering Thermal Economy and Entropy Generation. *Case Stud. Therm. Eng.* **2021**, *28*, 101446. [CrossRef]
21. Sabek, S.; Tiss, F.; Chouikh, R.; Guizani, A. Experimental Investigation and Numerical Validation of Total Heat Exchanger and Membrane Phenomena. *Energy Build.* **2016**, *133*, 131–140. [CrossRef]
22. Shen, Z.; Zhu, J.; Min, J. Airflow Maldistribution Effects on Membrane-Type Total Heat Exchanger Performance. *J. Enhanc. Heat Transf.* **2021**, *28*, 83–97. [CrossRef]
23. Shen, Z.; Min, J.; Duan, J. Numerical Study on Influence of Supply Inlet Air Parameter Distribution on a Membrane-type Total Heat Exchangers. *J. Tsinghua Univ. (Sci. Technol.)* **2020**, *60*, 958–966.
24. Kim, W.S. Design Optimization of Cross-Counter Flow Compact Heat Exchanger for Energy Recovery Ventilator. *Int. J. Air-Cond. Ref.* **2022**, *30*, 16. [CrossRef]
25. Feng, T.; Qian, Y.; Liu, F.; Yao, Y.; Xiao, L.; Wang, L. Research of Heat Recovery Energy for New Air Ventilator in Non-Standard Operating Conditions. *Build. Sci.* **2018**, *34*, 59–64.
26. Wang, Z. Performance Study of Membrane-Based Air to Air Enthalpy Exchanger. Master's Thesis, Tongji University, Shanghai, Chian, 2006.
27. Zhong, K.; Kang, Y.; Huang, P. Experimental Study on the Efficiency of Cross Flow Heat Recovery Ventilators. *HVAC* **2007**, *37*, 62–67.
28. Zhong, K.; Kang, Y. Effects of Outdoor Climate on the Efficiency of Air-To-Air Total Heat Exchanger. *HVAC* **2009**, *39*, 57–61.
29. Zhang, L.; Niu, J. Energy Requirements for Conditioning Fresh Air and The Long-Term Savings with a Membrane-Based Energy Recovery Ventilator in Hong Kong. *Energy* **2001**, *26*, 119–135. [CrossRef]
30. Wang, S.; Yuan, Y.; Sun, L.; Cao, X. Dynamic Analysis of Exhaust Air Total-Heat Recovery of Air Conditioning System in Summer Condition of Chengdu Using TRNSYS Simulation. *HVAC* **2016**, *46*, 128–132.
31. Zhang, Y.; Liu, Y.; Zhang, X. Analysis on Energy Saving Calculation of Exhaust Air Heat Recovery. *HVAC* **2015**, *45*, 12–15.
32. *JGJ/T346-2014*; China Academy of Building Research. Standard for Weather Data of Building Energy Efficiency. China Architecture & Building Press: Beijing, China, 2015; p. 9.
33. Shen, W.; Jiang, Z.; Tong, J. *Engineering Thermodynamics*, 3rd ed.; High Education Press: Beijing, China, 2001; p. 427.
34. *GB/T18883-2022*; National Health Commission of the People's Republic of China. Standard for Indoor Air Quality. Standards Press of China: Beijing, China, 2022; p. 2.
35. *DB33/1036-2021*; Zhejiang University Architectural Design and Research Institute Co., Ltd. Design Standard for Energy Efficiency of Public Buildings. China Building Materials Press: Beijing, China, 2021; pp. 29–32.
36. Lu, Y. *Practical Heating and Air Conditioning Design Manual*, 2nd ed.; China Architecture & Building Press: Beijing, China, 2008; p. 1548.
37. Smith, K.M.; Svendsen, S. The Effect of a Rotary Heat Exchanger in Room-Based Ventilation on Indoor Humidity in Existing Apartments in Temperate Climates. *Energy Build.* **2016**, *116*, 349–361. [CrossRef]
38. Cho, K.; Cho, D.; Kim, T. Effect of Bypass Control and Room Control Modes on Fan Energy Savings in a Heat Recovery Ventilation System. *Energies* **2020**, *13*, 1815. [CrossRef]
39. *DBJ33/T 1288-2022*; Zhejiang University Architectural Design and Research Institute Co., Ltd. Technical Specification for Energy-Saving Assessment of Civil Buildings. China Building Materials Press: Beijing, China, 2023; p. 217.
40. Zhao, L.; Wang, H.; Wang, A.; Ge, Z. Influence of Air Conditioning System without Humidification on Indoor Air Parameters in Winter of Kunming Region. *HAVC* **2017**, *47*, 30–34.
41. Zhao, L.; Wang, H.; Qin, J.; Ge, Z.; Han, J. Humidity Analysis of Air-Conditioned Rooms without Humidification in Winter. *J. Civ. Archit. Environ. Eng.* **2018**, *40*, 63–70.

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