Heat/Energy Recovery Technologies in Buildings

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Contents

Abstract

Heat/energy recovery technologies are exclusively applied in buildings so that more energy can be saved. Based on previous literatures, this chapter presents an analysis over the heat/energy recovery technologies in buildings. Firstly, the significance of heat/energy recovery technologies for building energy consumption was given briefly and some terms were introduced. Secondly, the components of a general heat/energy recovery system including heat exchanger, fan, and

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duct were explained. Particularly, as the core of heat/energy recovery system, different heat exchangers, such as fixed-plate, heat pipe, thermosyphon, loop and rotary wheel heat/energy exchangers were described in details. Then the performance indexes of heat/energy recovery system will be introduced and some impact factors on the performance will be discussed. Also, this chapter presents an analysis over experimental methods and rigs of these indexes. Together with that, the models in the literature for heat and mass transfer in the heat/energy recovery system to predict the performance were mentioned in details. Lastly, some typical application of heat/energy recovery in integrated energy-efficient system in buildings including heat/energy recovery ventilation, run-around heat/energy recovery system, heat pump with heat/energy recovery, and other potential application with heat/energy recovery in buildings were described to demonstrate the practical application of heat/energy recovery in buildings.

Introduction

Building energy consumption has accounted for nearly 40% of the total energy consumption due to higher requirement of indoor environment of residents derived from economic growth [1]. Building energy conservation is receiving more and more concern. Because lots of energy is lost in heating, air conditioning, and ventilating for buildings, heat recovery is important measure for building energy efficiency and can typically recover about 60–95% of the heat in exhaust air [2] or bring up to 20% energy reduction in building energy consumption in cold climate [3]. Therefore, heat/energy recovery in building applications was widely used in some countries such as Germany and Sweden [3, 4]. Heat/energy recovery has even become a requirement in national energy code or standard in some countries [5–7].

In some term, heat/energy recovery in building is limited as air-to-air heat or energy recovery, which refers to the process of recovering energy (heat/mass) from air with high temperature (humidity) to air at low temperature (humidity) to lower the heating or cooling energy for the supply air process [8]. As stated by Shurcliff [9], heat/energy recovery is defined as any device that removes in terms of extracts, recovers or salvages heat or mass from one air stream, and transfers it to another air stream to make use of the energy that would be lost to heat or cool the incoming air. However, some latent waste heat is not included in the definition, taking waste water or waste smoke exhausted from kitchen as an example. Obviously, these types of energy can also be transferred to supplied air to decrease air handling energy. Therefore, in this chapter, the concept of heat/energy recovery is extended to all heat or energy extract from waste flow.

Heat/energy recovery can be categorized as sensible heat recovery and latent heat or enthalpy recovery [10, 11]. Sensible heat recovery refers to the heat transfer caused by temperature difference while latent heat recovery comes from mass transfer of moisture in the two air streams as shown in Fig. 1. Analyzing from

Fig. 1 Schematic shown of heat recovery: (*1*) Sensible heat recovery; (*2*) latent heat recovery
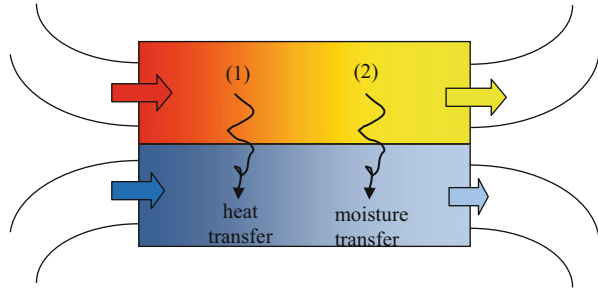
specific perspectives, as moisture transfer is always concomitant with heat transfer, latent heat recovery is also called as enthalpy recovery or total heat recovery.

Mardiana et al. [10] classified the heat/energy recovery technologies into three broad categories, including process to process, process to comfort, and comfort to comfort according to the state of the heat/energy recovered stream. In process-to-process system, heat is captured from exhaust stream and then transferred to the process supply air stream. Equipment is available to handle high temperatures level, such as exhaust smoke (high as 870 °C). These devices generally recover sensible heat while do not transfer latent heat. In dryers and ovens, some process-to-process heat recovery systems are often applied. However, in process-to-comfort system, heat is captured from a process exhaust to heat indoor air during winter. There are various applications, such as foundries, strip-coating plants, can plants, plating operations, pulp and paper plants, and other processing areas with heated process exhaust and large makeup air volume requirements [12]. The equipment is often applicable for high temperature stream and only sensible heat is recovered. Because energy is saved only in winter. When the weather is warm enough, the recovery system is not necessary and will be turned off. Process-to-comfort applications save less energy than process-to-process applications. Heat recovery device in comfort-to-comfort extract heating or cooling energy to the building supply air from the exhaust air of the building. This heat recovery system is often used in swimming pools, locker rooms, residential, operating rooms, nursing homes, animal ventilation, plant ventilation, and smoking exhaust [12].

In heat recovery system, heat transfer occurs between air to air, or air–fluid/ material–air or air–fluid–fluid–air as schematically, which is as shown in Fig. 2. In air to air recovery system, the supply air is directly neighbor to the exhaust through the heat exchanger. Heat is then transferred through the wall of the heat/energy exchanger between the two air streams. While, in air–fluid/material–air system, heat is extracted from the exhaust air to the immediate fluid or other material. Afterward, it will be transported to the supply air stream through the fluid or solid material movement. In air–fluid–fluid–air system, energy from the exhaust air stream is transferred to the working fluid though a heat exchanger and then the fluid with the energy is transported to another heat exchanger. Afterward, the energy will be transferred to another working fluid, which is transported to the supply air stream and transfer the energy to it. These systems depend on the application conditions. Because air to air heat recovery system is dominant in building application, it will be focused on in the following sections.

Fig. 2 Categories for heat transfer mediums:(*1*) air to air; (*2*) air–fluid/material–air; (*3*) air–fluid–fluid–air
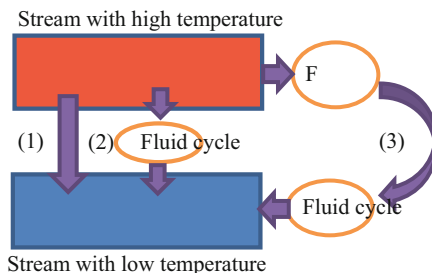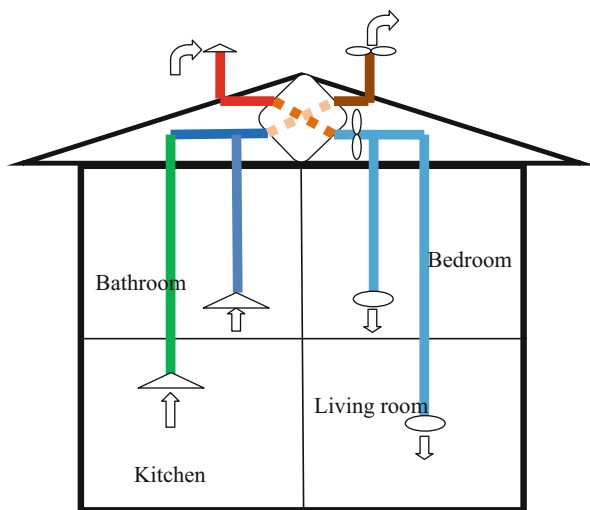

Fig. 3 Typical heat recovery system in a building [10]

Components of Heat/Energy Recovery System in Buildings and Their Physical Parameters

In common, a typical air to air heat recovery system in building consists of a heat exchanger core, where heat or energy occurs; ducts for incoming fresh air and outgoing stale air and two blower fans, which is as illustrated in Fig. 3. Those serve as the basic components for a heat recovery system. Some changes are necessary to be made to improve the performance of the system. For example, photovoltaic (PV) or hybrid photovoltaic and thermal (PV/T) panels can be added to the building roof to enhance the thermal efficiency [13].

Heat Exchanger

Heat exchanger is the core component in a heat recovery system. The heat transfer efficiency depends on the types and size of heat exchanger. There are various
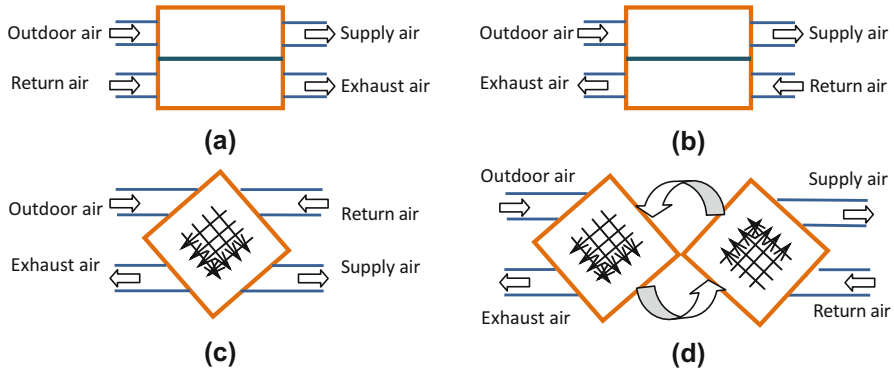
Fig. 4 Types of fixed-plate heat exchanger [12]: (a) Parallel-flow heat exchanger; (b) Counterflow heat exchanger; (c) Cross-flow heat exchanger; (d) Multi-pass heat exchanger

common types of heat exchanger for buildings, which include fixed-plate, heat pipe, rotary wheel, and run around [10].

Fixed-Plate Heat Exchanger

Fixed-plate heat exchanger serves as the most common heat exchanger for heat recovery system. In a heat exchanger, some thin plat plates with smooth or corrugate surfaces are stacked together with some internal flow channels. Similar to the water-water heat exchanger, the flow in the plat fixed-plate heat exchangers for air–air heat recovery system are classified as parallel flow, counter-flow, and cross-flow, which areas shown in Fig. 4. When the contact surface consists of metal material, such as copper and aluminum, only the sensible is commonly recovered. Moisture and heat are exchanged simultaneously to recover latent and sensible energy when porous materials are applied as heat transfer surfaces. Recently, more materials for fixed-plate heat exchanger besides metal materials are introduced as the market growth. Nasif et al. [14] studied the performance of a Z type flow heat exchanger by utilizing 98 μm thick 60gsm Kraft paper as the heat and moisture exchanger surface of applied in HVAC system as shown in Fig. 5. The efficiency of the heat exchanger is about 75% for sensible and 60% for latent energy. Lu et al. [15] employed plastic film to enhance the efficiency of heat transfer because the film vibrated when airflow passed through the channels. Results illustrated that the effectiveness of the heat exchanger varied from 0.65 to 0.85 with different airflow rate while pressure drop was lower than 20.0 Pa. Sensible polymer plate was also used in air-to-air heat recovery unit due to its ability to handle liquid and gases, the resistance to fouling and corrosion and the possible use in both humidification and dehumidification applications [16]. The efficiency was found to vary from 78% to 94% due to different air flow rates. Chen et al. [17] proposed polypropylene-based polymer hollow fiber heat exchangers (PHFHEs hereafter). The results revealed that the PHFHEs could provide a conductance per unit volume of 4×10^6 W/m^3 K, which is 2–8 times higher than the conventional metal heat exchangers. Though the exchangers were shell-and-
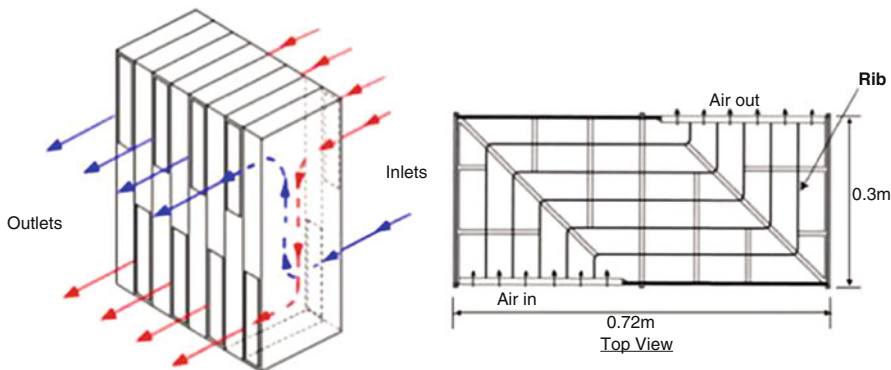
Fig. 5 Z type fixed-plate heat exchanger [14]

tube types, they may be promising to substitute metallic heat recovery system due to their low cost, corrosive resistance, and light weight features for building application. In addition to the material reformation for energy exchange performance improvement of heat exchanger for building application, some other methods for heat transfer enhancement were also developed for plate heat exchangers. For example, corrugated-plate heat exchangers are adopted in heat recovery in building because of their high thermal effectiveness and the small approach temperature operation [18]. Chevron-plate is the most commonly used plate pattern in a plate heat exchanger (PHE), which is as shown in Fig. 6. Theories and experiments proved that the swirling of the flow elevates heat transfer in the channels [19].

Heat Pipe Heat Exchanger

Heat pipe heat exchanger (HPHE hereafter) is a kind of efficient heat exchanger, which is widely used in air conditioning systems. The working principle of heat pipe heat exchanger is illustrated in Fig. 7. In one side of a heat pipe heat exchanger, liquid is heated by hot fluid stream outside the heat pipe and then is vaporized into vapor gas, in which the latent heat of vaporization is utilized to transfer heat. The vapor flows to another side and then is condensed to release the heat to the cool fluid. Then the condensed fluid returns back to the evaporator section through the capillary action of the wick or gravitational force in the heat pipe [20]. The HPHE can be used both in HVAC system and in natural ventilation [21] for heat recovery from exhaust air. The advantages of HPHE for heat recovery in building applications are listed as follows [22]: (1) the relatively small thermal resistance of the heat pipes, thus small overall temperature drop may be achieved with physical separation of the evaporator and condenser; (2) the capability of heat pipes to operate as nearly isothermal devices, where a wide range of energy inputs can be accommodated with little change in the source temperature; (3) the evaporator and condenser sections of the heat pipe function separately, and thus a wide variety of shapes and sizes of the heat pipes can be designed and produced and is easy to arrange in HVAC systems; and (4) the thermal response time for heat transfer is significantly less than other heat

Fig. 6 Geometry of a chevron-plate heat exchanger flow channel: (**a**) outward appearance, and (**b**) inter-plate corrugations [19]
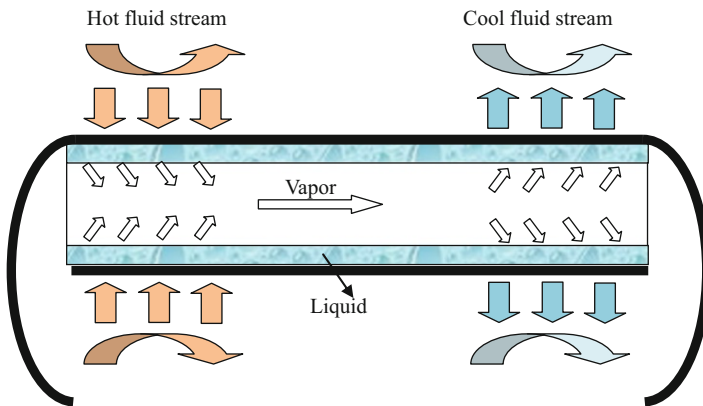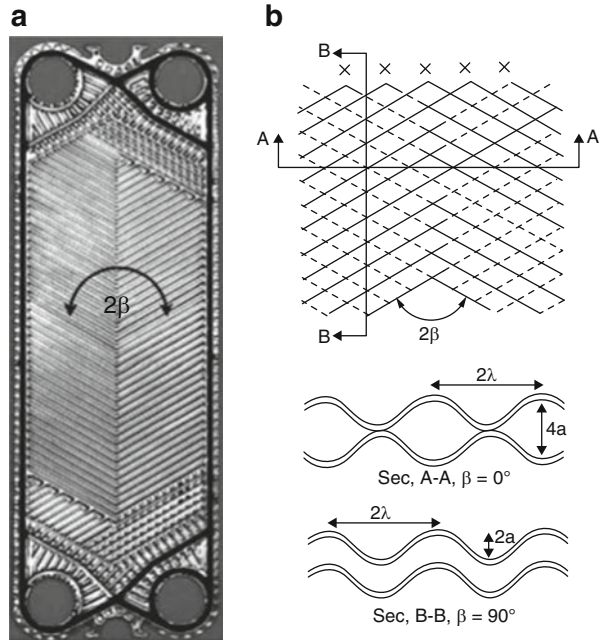
Fig. 7 Working principle of heat pipe

transfer devices. The most serious drawback of the HPHE for building application is low effectiveness, which is measured between 0.35 and 0.6 [23]. Apart from the conventional form of heat pipe, which is as shown in Fig. 7, other forms of heat pipes, such as vapor-dynamic thermosyphons, loop heat pipes, aluminum heat pipe panels, pulsating and "spaghetti" heat pipes, and sorption heat pipes, have been developed in recent years [20]. The conventional heat pipes dominate the application for heat pipe heat recovery system in buildings, where there are constraints in the air

Fig. 8 Thermosyphon and loop HPHEs [24]

Fig. 9 Connection of heat pipes in HPHE [24]

duct system, the thermosyphon heat pipes (THPs) or loop heat pipes (LHPs) are suitable because they are two-phase separated for easy arrangement in the duct of HVAC systems as shown in Fig. 8 [24]. Due to performance limitation of single heat pipe, a HPHE can consist of an array of HPs in one or more vertical rows. And these pipes are connected by the different forms as shown in Fig. 9.
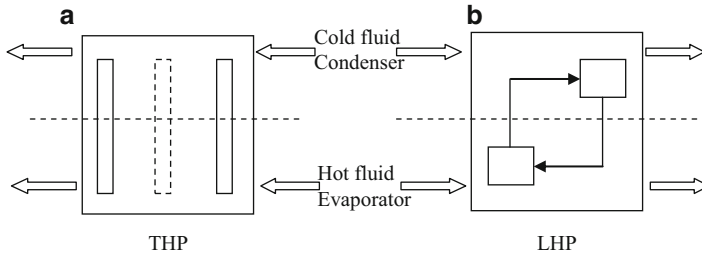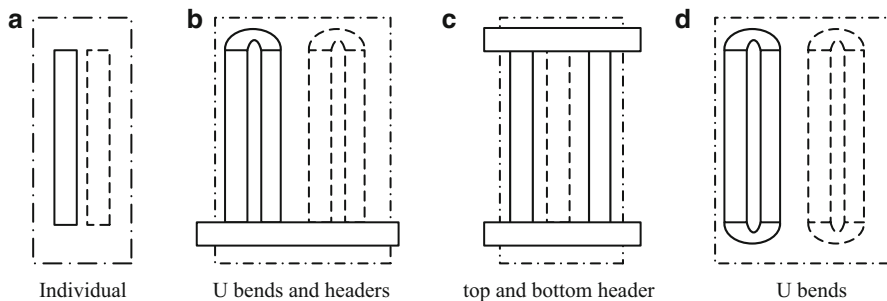
Rotary Wheel Heat Exchanger

Rotary wheel heat exchanger is a total energy exchanger consisting of a rotor with permeable storage media fitted in a case as shown in Fig. 10. The storage media will transfer energy from hot fluid to cold fluid when the wheel rotates periodically from one fluid to another with a relative low rotor speed range of 3–15 rpm. Media surface areas expose to airflow vary from 300 to over 4000 m^2/m^3, depending on the type of medium and physical configuration. The performance of energy exchanger will be influenced greatly. The storage media may also be classified by their ability to recover sensible heat only or total heat. Media for sensible heat recovery is made of aluminum, copper, stainless steel, and Monel (a kind of a nickel–copper alloy). Media for total heat recovery mainly comprises desiccant materials [12]. Commonly used desiccant materials include activated carbon, activated alumina, molecular sieve, silica gel, lithium chloride, calcium chloride, and so forth. The key features of these materials include large saturated adsorption amount, easy reactivation, and the adsorption
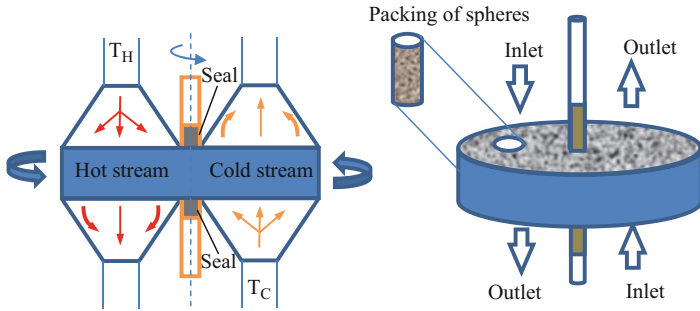
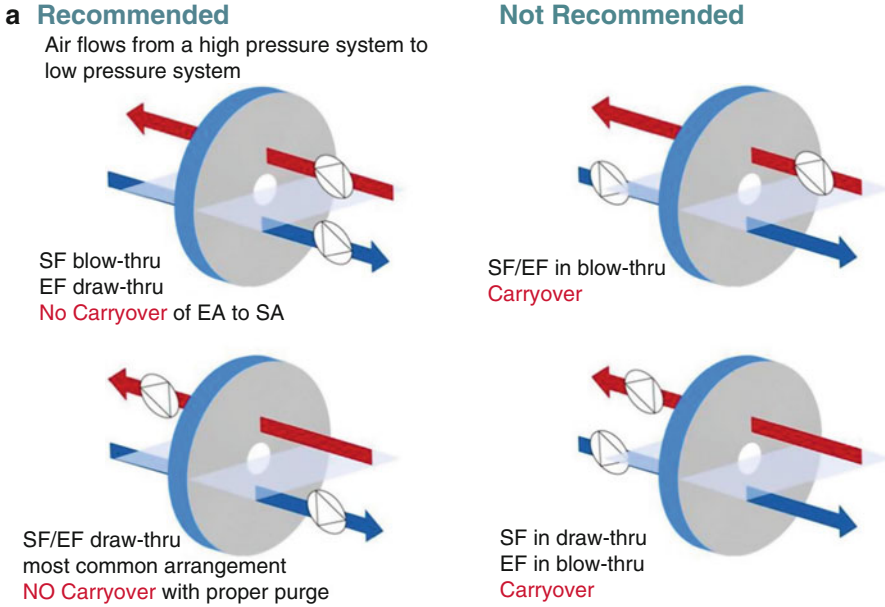**Fig. 10** Schematic shown of rotary wheel heat exchanger [27]

performance close to Type 1 M material [25]. Recent years, some other alternative desiccant materials were investigated including modification of conventional desiccant, natural rock-based desiccant, bio-desiccant, and composite desiccant [26].

The advantages of rotary wheels lie in the capability of recovering both sensible and latent heat and trouble-free operation. The temperature efficiency (see section "Performance evaluation of heat/energy recovery system and impact factors") of rotary wheel can get beyond 80% [10]. And the largest problem is intermixing of the two air stream due to two mechanisms: carryover and seal leakage [12, 28]. Some methods can be used to reduce cross leakage. For example, cross leakage can be reduced by properly arranging the blowers so that the leakage of outdoor air to the exhaust air stream will be promoted and the pressure difference between the two stream smaller will be clearly illustrated, which is as shown in Fig. 11a. A purge section also can be installed on the heat exchanger to reduce carryover leakage as shown in Fig. 11b [12]. Though recirculating some air is not a concern in many applications, critical applications such as hospital operating rooms, laboratories, and cleanrooms require stringent control of carryover. Carryover can be reduced to less than 0.1% of the exhaust airflow with a purge section yet cannot be completely eliminated. The theoretical carryover of a wheel without a purge section is directly proportional to the speed of the wheel and the void volume of the medium (75–95% void, depending on type and configuration) [12].

Fans and Duct

Fan Power

Apart from the heat exchanger, efficiency and size of fan as well as ducting characteristics also affects the heat or energy exchange performance of the heat/energy recovery systems. In the same power class in the heat/energy recovery system, global efficiency of fans varies a lot, which is as shown in Fig. 12. Min and Su [30] found that greater fan power leads to a larger total heat transfer rate, with the maximum total heat transfer rate occurring at a smaller channel height. Therefore, specific fan power should be addressed when operating energy consumption of heat recovery system is involved. Only low specific fan power makes the heat recovery system advantageous [31]. The fan power of

a **Recommended**   **Not Recommended**

Air flows from a high pressure system to
low pressure system

SF blow-thru
EF draw-thru
No Carryover of EA to SA

SF/EF draw-thru
most common arrangement
NO Carryover with proper purge

SF/EF in blow-thru
Carryover

SF in draw-thru
EF in blow-thru
Carryover

Fan arrangements-preventing cross flow leakage

b

High pressure area
OA

RA   purge air

Low pressure area
EA
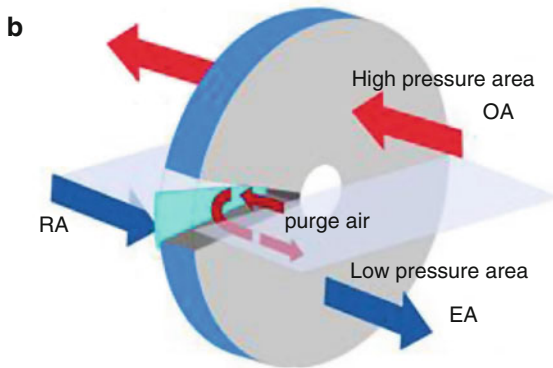
Purge sector-preventing carryover leakage

Fig. 11 Measures to prevent air leakage for rotary wheel heat exchanger [29]. *SF*: supply fan; *EF*: exhaust fan; *OA*: outdoor air; *RA*: return air; *EA*: exhaust air

heat recovery system was influenced by three factors [32]: (1) the overall efficiency of the fan; (2) the resistances or pressure drop in the system; (3)the air flow velocity through the heat exchanger and ductwork. That means that not only fan but also ductwork contributes to the specific power of fan. Shurcliff [9] suggested that ducts should be short and free of sharp bends as feasible and of ample diameter to decrease the pneumatic resistance together with fans or blowers of small, low power and quiet type. Because the

Fig 12 Global fan efficiency of heat recovery unit [34]
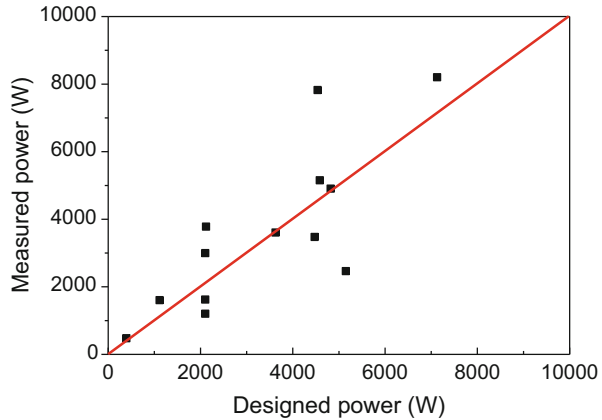
air flow in the ducts is commonly turbulent, the resistance can be estimated by some correlations [33]. Usually, the duct length ranges from 3 to 18 m and duct diameters of 0.1–0.2 m are satisfactory, producing low pressure drops at common flow rate [9].

Heat Loss Along the Ducts

Heat loss along the ducts also affects efficiency. Resistance depending on duct results in flow distribution, which is substantial for heat loss along the duct. The pressure drop and heat transfer coefficient are dominated by duct cross sections [35, 36].

Efficiency in the Core Heat Exchanger

Fan together with ductwork can affect the flow rate so that the efficiency in the core heat exchanger is also affected. Figure 13 illustrates that the efficiency of the two plate-fin cores in a heat recovery system varies with the flow rate, according to experimental and modeling results. Based on this, it can be concluded that when flow rate increases, the efficiency will decrease sharply. Therefore, caution is needed when selecting fan and designing ducts for a heat recovery system.

Performance Evaluation of Heat/Energy Recovery System and Impact Factors

## Performance Evaluation of Heat/Energy Recovery System

Performance Indices

Effectiveness

In sensible heat recovery system, only heat transfer occurs. The temperature of the supply air would be changed. In common, the sensible heat recovery system is evaluated through sensible efficiency or effectiveness. Sensible efficiency or effectiveness is defined as energy ratio, which is expressed by Eq. 1.

Fig. 13 Efficiency of the two plate-fin heat exchanger under various air flow rates [35]
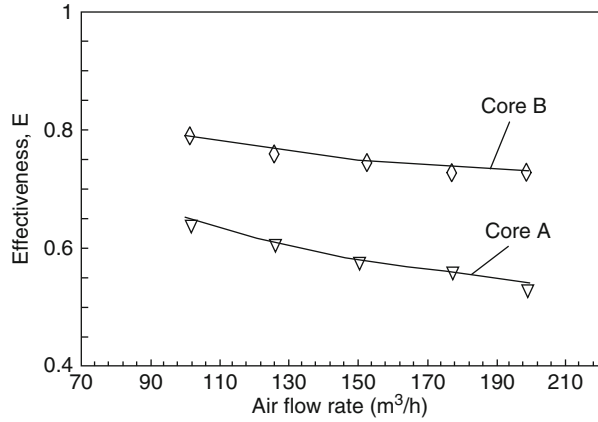
$$\varepsilon_S = \frac{c_{p,s}\,\dot{m}_s(t_2 - t_1)}{(c_p\dot{m})_{\min}(t_3 - t_1)} = \frac{c_{p,e}\,\dot{m}_e(t_3 - t_4)}{(c_p\dot{m})_{\min}(t_3 - t_1)} \tag{1}$$

In this Eq. 1, t_1 and t_2 are the temperature of supply fluid stream before and after the heat recovery system, while t_3 and t_4 are the temperature of exhaust fluid stream before and after the heat recovery system, which is as shown in Fig. 14. "c_p" is the specific heat capacity of supply fluid or exhaust fluid while $\dot{m}$ refers to the mass flow rate of supply fluid or exhaust fluid. The subscripts "s" and "e" express supply and exhaust fluid stream, respectively.

Sensible efficiency is also expressed by the temperature ratio or temperature efficiency when mass flow rates of supply and exhaust air are equal as Eq. 2.

$$\varepsilon_S = \varepsilon_T = \frac{t_2 - t_1}{t_3 - t_1} = \frac{t_3 - t_4}{t_3 - t_1} \tag{2}$$

Similarly, total efficiency or effectiveness and latent efficiency are defined as latent heat ratio expressed by Eqs. 3 and 4.

$$\varepsilon_L = \frac{h_{fg}\,\dot{m}_s(d_2 - d_1)}{h_{fg}(\dot{m})_{\min}(d_3 - d_1)} = \frac{h_{fg}\,\dot{m}_e(d_3 - d_4)}{h_{fg}(\dot{m})_{\min}(d_3 - d_1)} \tag{3}$$

$$\varepsilon_t = \frac{\dot{m}_s(h_2 - h_1)}{(\dot{m})_{\min}(h_3 - h_1)} = \frac{\dot{m}_e(h_3 - h_4)}{(\dot{m})_{\min}(h_3 - d_1)} \tag{4}$$

where, h_{fg} is the evaporation enthalpy of water; d is the absolute humidity; h is the enthalpy of water vapor.

Performance of sensible heat recovery system is also evaluated through the number of transfer units (*NTU*), which is used in heat exchanger. *NTU* is defined by Eq. 5.
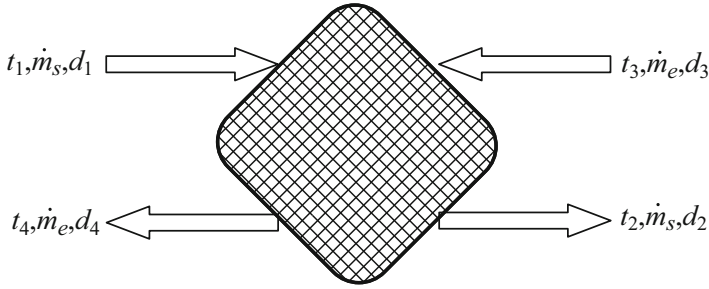
Fig. 14 Airstream numbering convention of a heat/energy recovery system [33]

$$NTU = \frac{K \cdot A}{(c_p \dot{m})_{\min}} \tag{5}$$

where, K refers to the overall heat transfer coefficient; A is the heat transfer area.

NTU represents the heat transfer capacity, which can determine temperature or enthalpy difference of the supply or exhaust air through heat exchanger. Efficiency can be determined by ε-NTU method according to some correlations between NTU and recovery efficiency for different flow types as listed in Table 1 [36, 37]. The theoretic studies found that there is a positive correlation between effectiveness and NTU not only in terms of sensible heat recovery but also in terms of total energy recovery [38, 39], which is as shown in Fig.15.

Rate of Energy Transfer

Efficiency provides a measure for capacity of heat recovery while energy transfer rate measures the quantity of heat recovery. For example, when the air flow is small, the efficiency will be high while the heat transfer rate will be low. The rate of energy transfer depends on the operating conditions and the intrinsic characteristics of the energy exchanger, such as the geometry of the exchanger (number of passes, fins), flow types (parallel flow/counter flow/cross flow), and material physical property (thermal conductivity, thickness, permeability) [12].

Experimental Determination of Efficiency

The efficiency of sensible, latent and total heat recovery can be determined in laboratory tests or field measurements based upon the definitions or Eqs. 1, 2, 3, and 4. Field measurements depend on the specific recovery system. The test conditions are uncontrolled and variable for the practical conditions. Therefore, the results from different field measurements actually are not comparable. In the laboratory, some typical experimental rigs are required, according to ASHRAE Standard 84 and AHRI Standard 1061. ASHRAE Standard 84 provides some guidelines to test the performance of the heat exchanger of heat recovery system and (1) establishes a uniform method of testing to obtain performance data; (2) specifies the data required, calculations to be used, and

Table 1 Correlations between NTU and efficiency of sensible heat recovery heat exchanger [37]

Flow types	Correlations
Cross-flow heat recovery	$\varepsilon = 1 - e^{NTU^{0.22}\left(e^{-NTU^{0.78}}-1\right)}$
Parallel heat recovery	$\varepsilon = \frac{1-e^{-NTU(1+C)}}{1+C}$ a
Count-flow heat recovery	$\varepsilon = \frac{1-e^{-NTU(1-C)}}{1-Ce^{-NTU(1-C)}}$
Balanced flow	$\varepsilon = \frac{NTU}{1+NTU}$

[a]C- minimum-to-maximum capacity ratio.

Fig. 15 Variation of effectiveness with NTU [38]
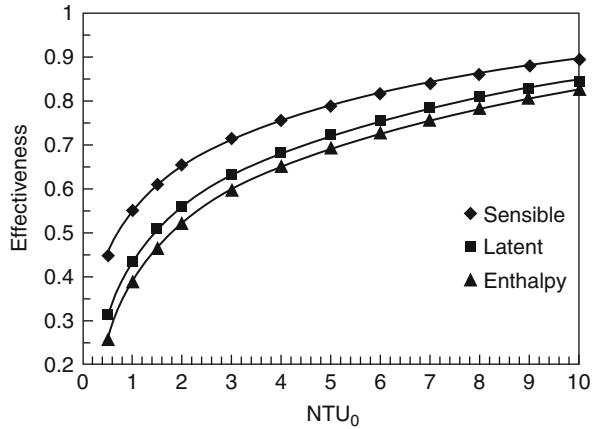
reports procedures to test each of seven independent performance factors (including ε_s, ε_L, ε_t, air pressure drops, air transfer ratio (EATR), which characterizes the fraction of exhaust air transferred to the supply air; and outdoor air correction factor(OACF), which is the ratio of supply inlet to outlet air flow) and their uncertainty limits; and (3) specifies the types of test equipment. Yau [40] established experimental rigs to obtain the performance factors of heat pipe exchangers, which is as shown in Fig. 16. This process must satisfy the requirement of ASHRAE Standard 84. The rigs imitate the working process of a heat recovery system with heat pipe heat exchangers. The dry and wet bulb temperatures of supply and return air after and before heat pipe heat exchanger, DBT_1, WBT_1, DBT_2, WBT_2, DBT_4, WBT_4, DBT_5, and WBT_5 were measured. As the air flow in supply and return air is balanced, simplified Eqs. 1, 3 and 4 were adopted to calculate the sensible, latent and total heat recovery efficiency.

AHRI Standard 1061 requires balanced airflow rates (no leakage between supply and return air). Besides, it also requires satisfying the following standard operating condition: Winter: Outdoor air with dry bulb temperature 1.7 °C and wet bulb temperature 0.6 °C, Indoor (room) air with dry bulb temperature 21.1 °C and wet bulb temperature 14.4 °C; Summer: Outdoor air with dry bulb temperature 35 °C and wet bulb temperature 25.6 °C, Indoor (room) air with dry bulb temperature 23.9 °C and wet bulb temperature 17.2 °C. As the test conditions are specific, the results from the laboratory measurements are comparable for different heat recovery equipment.
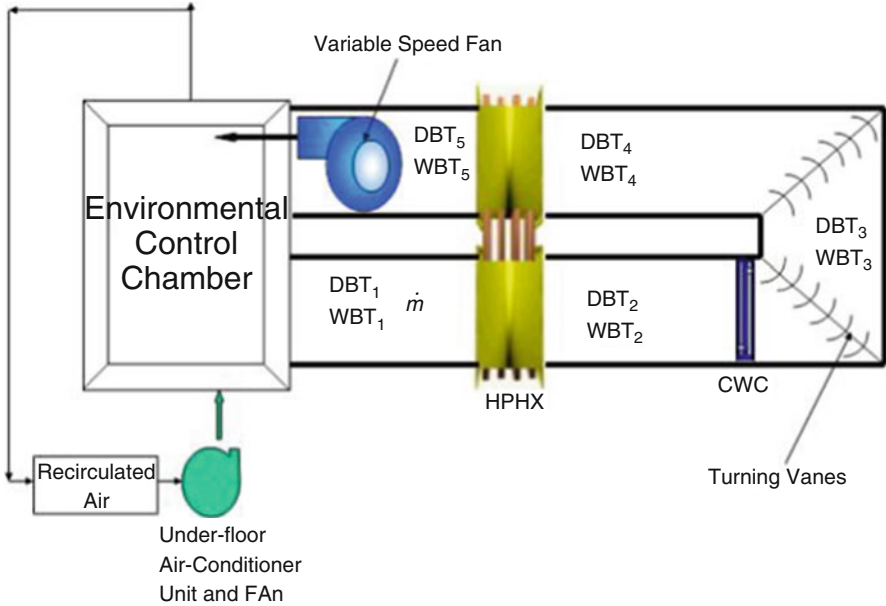
Fig. 16 Schematic diagram for the experimental setup for testing the heat recovery efficiency of heat pipe heat exchanger [40]

Performance Modeling of Heat/Energy Recovery System

Detailed modeling can provide more physical insights into mechanism on heat and mass transfer of heat recovery, which are both of great importance for the evaluation of the performance of heat/energy recovery and can be used to optimize the system. Physical model for the core heat or energy exchangers serves as the key for modeling heat recovery process. As the main difference between fixed-plate heat exchanger and membrane-based energy exchanger is whether there is moisture transfer through the plate, fixed-plate heat exchanger can be regarded as the special membrane-based energy exchanger without consideration of moisture transfer [35, 38]. The model for heat and mass transfer in the core membrane-based energy exchanger, as schematically shown in Fig. 17, was set up under the followings assumptions [38]: (1) heat conduction and vapor diffusion of the two air streams along flow directions (axial) are negligible compared to energy transport and vapor convection by bulk flow; (2) adsorption of water vapor and membrane material occurs in equilibrium adsorption-state; (3) both the heat conductivity and the water diffusivity in the membrane are constants; (4) the heat of sorption is assumed constant and equal to the heat of vaporization. Based on the assumptions listed above, the general normalized governing equations with boundary conditions for steady heat and mass transfer can be described as Eqs. 6, 7, 8, 9, 10, 11, 12, 13, 14, 15, 16, and 17.
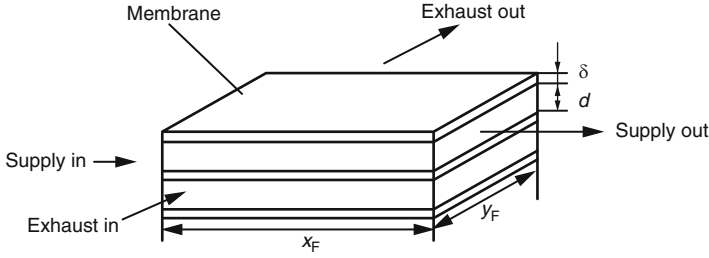
Fig. 17 Schematic of a cross-flow enthalpy exchanger with membrane cores [38]

Supply air stream:

$$\frac{\partial T_s}{\partial x^*} = 2NTU_s(T_m - T_s) \tag{6}$$

$$\frac{\partial \omega_s}{\partial x^*} = 2NTU_{Ls}(\omega_{ms} - \omega_s) \tag{7}$$

Exhaust:

$$\frac{\partial T_s}{\partial y^*} = 2NTU_e(T_m - T_e) \tag{8}$$

$$\frac{\partial \omega_e}{\partial x^*} = 2NTU_{Le}(\omega_{me} - \omega_e) \tag{9}$$

Membrane:

$$\frac{\partial T_m}{\partial z^*} = a_1 \frac{\partial^2 T_m}{\partial x^{*2}} + a_2 \frac{\partial^2 T_m}{\partial y^{*2}} + a_3 \frac{\partial^2 T_m}{\partial z^{*2}} \tag{10}$$

At the inlet and outlet of supply air stream:

$$T_s \mid_{x^*=0} = T_{si} \tag{11}$$

$$\omega_s \mid_{x^*=0} = \omega_{si} \tag{12}$$

At the inlet and outlet of exhaust air stream:

$$T_e \mid_{y^*=0} = T_{ei} \tag{13}$$

$$\omega_e \mid_{y^*=0} = \omega_{ei} \tag{14}$$

At the sides of the membrane:

$$\frac{\partial T_m}{\partial x^*} \mid_{x^*=0 \text{ or } 1} = 0 \tag{15}$$

Fig. 18 Comparison between the modeled and experimentally obtained efficiencies [38]

$$\frac{\partial T_m}{\partial y^*}\Big|_{y^*=0 \text{ or } 1} = 0 \tag{16}$$

At the interface between the membrane and supply air:

$$\frac{\partial T_m}{\partial z^*}\Big|_{z^*=0} = Nu_s(T_s - T_m) + \frac{D_{wm}}{\lambda_m}L_w(\theta_{ms} - \theta_{me}) \tag{17}$$

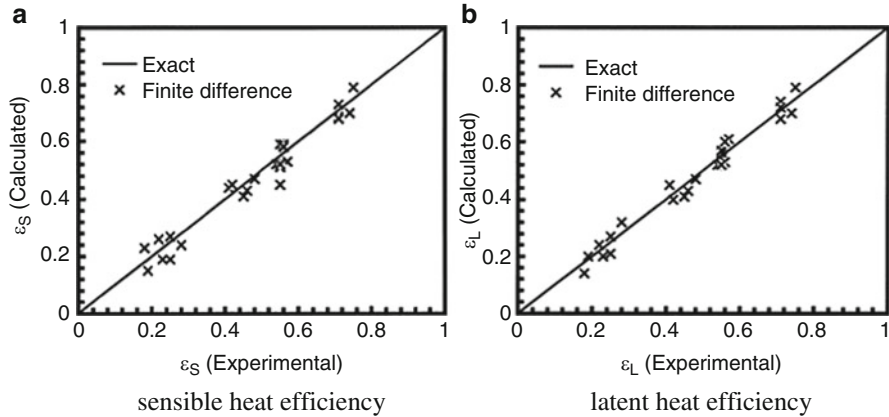where, T and ω refer to the temperature and moisture content; subscripts s, e, and m denote supply air, exhaust air, and membrane; x^*, y^*, and z^* are the normalized coordinates along supply air flow, return air flow, and membrane thickness direction respectively and are defined as $x^* = \frac{x}{x_F}$, $y^* = \frac{y}{y_F}$, $z^* = \frac{z}{\delta}$;

NTU and NTU_L are defined by $NTU_i = \frac{n_i h_i x_F y_F}{\dot{m}_i c_{pi}}$, $NTU_{Li} = \frac{k x_F y_F}{\dot{m}_i}$, where n refers to the number of channels, h is convective heat transfer coefficient, $\dot{m}$ means the mass flow rate of air streams, k is convective mass transfer coefficient, c_p is the specific heat capacity and $i = s$ or e.

The temperature and humidity distribution along supply or return air stream can be obtained by solving Eqs. 6, 7, 8, 9, 10, 11, 12, 13, 14, 15, 16, and 17 through numerical methods. Thus, the efficiency can be calculated based on Eqs. 1, 2, 3, and 4 easily. Niu and Zhang [38] employed the model to calculate the sensible and total heat efficiency for a cross-flow membrane-based enthalpy exchanger in the literature [41]. It has been found that the modeled efficiencies agreed well with the experimental results, which are as shown in Fig. 18. The model can also be used to investigate more details, which are often difficult to obtain through experimental methods. For example, Niu and Zhang [9] drew the exquisite distribution of temperature and humidity within the supply and exhaust air channels, as shown in Fig. 19. It can be seen from Fig. 19 that the great
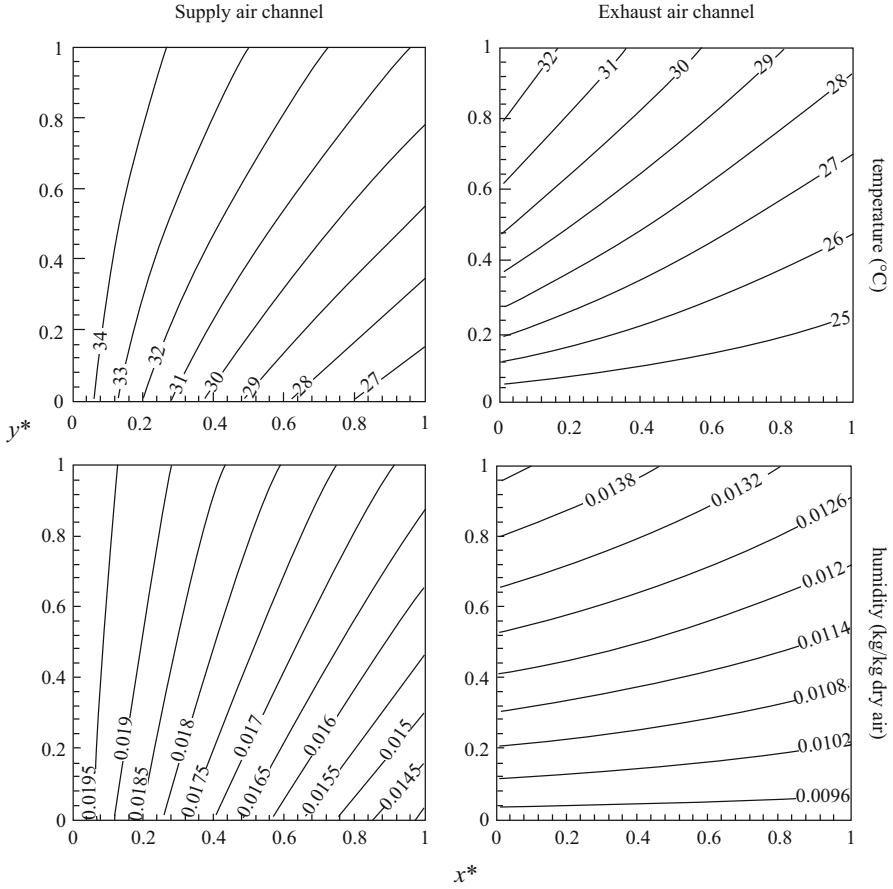
Fig. 19 Modeled temperature and humidity distribution within the supply and exhaust air channel [38]

temperature and humidity difference between the supply and exhaust air at the left bottom are found, so some measures may be made to improve the local heat and mass transfer to lower the temperature and humidity difference to elevate the global efficiency. Therefore, the model is more suitable to optimize the heat recovery design than experiments.

For the rotary wheel heat recovery system, Ge et al. [42] reviewed some proposed physical models including the general mathematical model, gas-side resistance (GSR) model, gas and solid-side resistance (GSSR) model, and empirical models. Similar to the models for membrane-based heat recovery system, the models for the rotary wheel heat recovery systems are also used into the assessment of the performance and the optimization of the design.

Fig. 20 The influence of temperature of inlet air stream on sensible and total heat efficiency for HPHE heat recovery system [40]

Impact Factors of Heat Recovery Efficiency

Supply Air Temperature

The effect of supply air temperature may differ a lot because of the different types of heat exchangers. For heat pipe sensible heat recovery system, supply air temperature may affect flow loss performance and pressure loss a little so that efficiency would be also affected slightly, while for other types of sensible heat recovery system [21], there is little influence of supply temperature on efficiency [16, 40, 43].
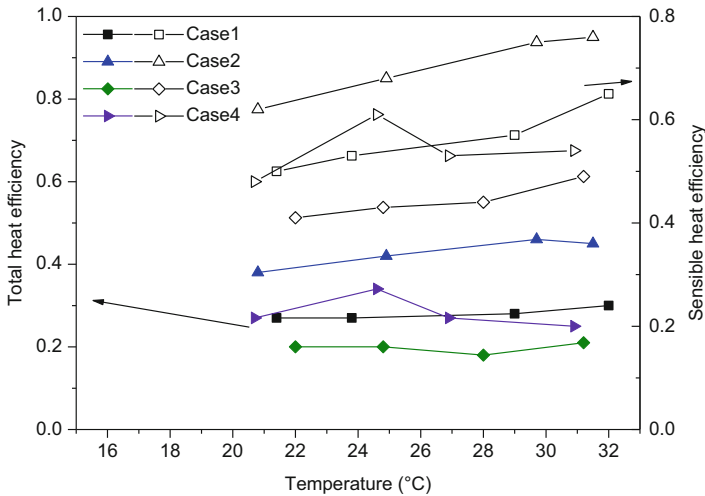
In theory, the performance of heat recovery system with fixed-plate type heat exchanger is not affected by the temperature of inlet fluid [43]. For HPHE heat recovery system, Yau [40] tested the efficiency of heat recovery system for different temperature and humidity of inlet air stream in various designed experimental rigs. And the measured results on the effect of inlet air temperature on sensible and total heat efficiency are plotted in Fig. 20. From that, it can be concluded that for both sensible and total effectiveness, the temperature of the inlet air appears to have minor influence.

For heat recovery system with membrane-based heat exchanger, the temperature of supply air may affect the heat recovery efficiency. Niu and Zhang [38] investigated the effect of the temperature of supply air based on a validated heat and mass transfer model for membrane-based heat exchanger. The variations of sensible, latent, and total heat effectiveness against different inlet air temperatures are shown in Fig. 21. Similar to the fixed-plate heat exchanger, the temperature of supply air has little impact on the sensible heat efficiency. Meanwhile, the great impact of the supply air temperature on the latent and total heat efficiency appears

Fig. 21 The influence of temperature of inlet air stream on sensible and total heat efficiency for membrane-based heat exchanger heat recovery system [38]
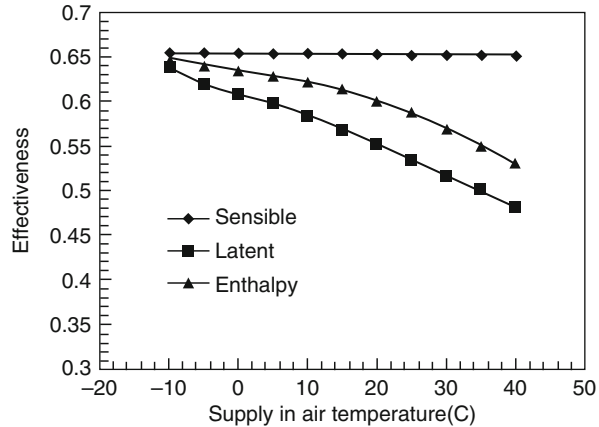
especially when the temperature is above than 15 °C with the unchanged relative humidity. The efficiency decreases with the increase of supply air temperature. The authors explained it by increasing moisture mass diffusion resistance defined as Eq. 18. According to Eq. 18, when temperature increases, the moisture mass diffusion resistance will increase so that the efficiency decreases.

$$\gamma_m = \frac{\delta}{D_m} \frac{10^6 (1 - C + C/\phi)^2 \phi^2}{e^{(5294/T)} w_{\max} C} \tag{18}$$

Air Flow

The air flow rate has significant impact over the heat efficiency for sensible heat recovery system because it determines the air velocity through the heat exchangers and ductworks. When air flow rate increases, the air velocity increases for the same heat recovery system so that air flows quickly through the heat exchanger surfaces. Thus, the heat efficiency decreases because heat transfer is insufficient. The results were experimentally validated for heat pipe [44], which is as shown in Fig. 22a. Besides, similar phenomenon was found for membrane heat exchanger [14]. In addition, pressure loss increases with the increase of air velocity, which is as shown in Fig. 22b, thus the energy consumption of fans also increases. Consequently, when air flow rate increases, the heat recovery efficiency will decrease. However, the transferring heat quantity in a period of time may increase because more air flows through the system in the time due to larger air flow rate.

Relative Humidity of Supply Air

Obviously, for the heat recovery system with fixed-plate heat exchangers, the relative humidity of supply air has little effect on the efficiency except when condensation of water vapor occurs on the surfaces, which is rare for this type of heat recovery system. For the total heat recovery system, taking membrane-based or

Fig. 22 Influence of air velocity on heat recovery system [44]

Heat efficiency vs. air velocity

Pressure loss vs. air velocity
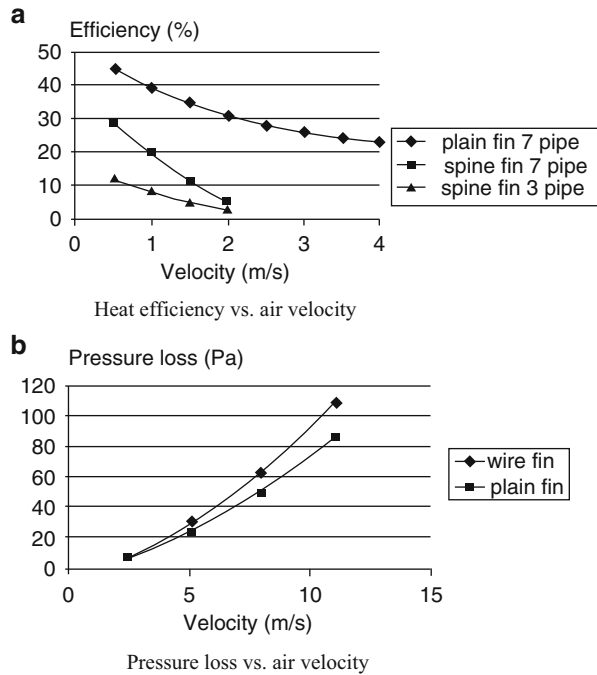
rotation wheel heat recovery systems as examples, moisture is transferred through the heat exchanger so that the relative humidity may have effect on the total or latent heat efficiency. The resistance of moisture transfer will be affected by the relative humidity, which depends on the water vapor sorption constant, C. According to the theoretical study [43, 45], it is illustrated that the moisture resistance of the membrane increases for $C < 1$, leading to the increase of latent heat recovery efficiency. Besides, it decreases when $C > 1$ while remains constant when $C = 1$ (corresponds to the Henry sorption) when the humidity increases, as indicated in Fig. 23 [45]. Therefore, the relative humidity may affect the latent and total heat efficiency. However, the CFD simulation by Yaici et al. [46] promoted that the relative humidity had minor effects on heat or energy recovery efficiency. The bias between the findings may be explained by different water vapor sorption constants. The further explanation needs more systematic experiments. On the other hand, although the membrane thermal resistance is also affected by the humidity due to the coupling effects of the heat and mass transfer, the influence of the humidity on the sensible heat efficiency is still quite limited because membrane thermal resistance accounts for only a small fraction of the total thermal resistance (10% or less) [45].

Inclination of Heat/Energy Exchanger
As the inclination of heat/energy exchanger may affect heat and mass transfer through the exchanger, it may affect the heat/energy efficiency. There are two possible effects of inclination: increasing tilt should reduce external condensate
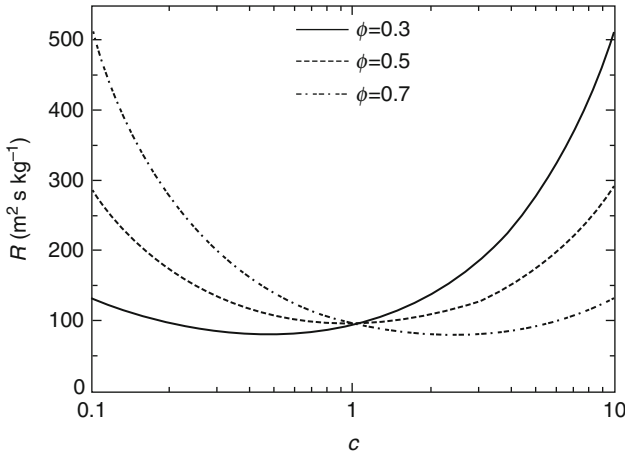
Fig. 23 The membrane moisture transfer resistance with different water vapor sorption constant when humidity difference is different [45]

film thickness and hence to result in the decrease of thermal resistance (i.e., the increase of effectiveness) as the heat transfer coefficient for the wet surface is negatively correlated with the thickness of the condensation water film; but increasing tilt should have an adverse effect on internal condensate return back to the evaporator to make heat exchange with low efficiency. Yau [40] measured the effect of inclination of the HPHE on the efficiency. According to the result, it has been found that there was also no clear evidence from these tests that inclining the HPHE results in consistently enhanced performance.

Heat/Energy Recovery in Integrated Energy-Efficient System for Building Application

Heat/Energy Recovery Ventilation (HRV or ERV)

Heat/energy recovery ventilation (HRV/ERV) is the most common heat recovery system for building application whose principle is to recover heat from the exhaust air and then get it transferred to the supply air through a heat exchanger. In some contexts, heat recovery system for building application refers to only HRV/ERV [12] as shown in Fig. 3.

HRV/ERV is often used in active ventilation systems or mechanical ventilation systems, in which fan or blower is employed. As ventilation is necessary for buildings to maintain good indoor air quality for persons' health, comfort, and effective working or learning performance, energy consumption due to heating or cooling outdoor air and fans power is inevitable especially in hot summer or cold winter. Due to the evaluation standards of indoor air quality, the ventilation loads cover 20–50% of the heating demand for new and retrofitted buildings [34, 47, 48].
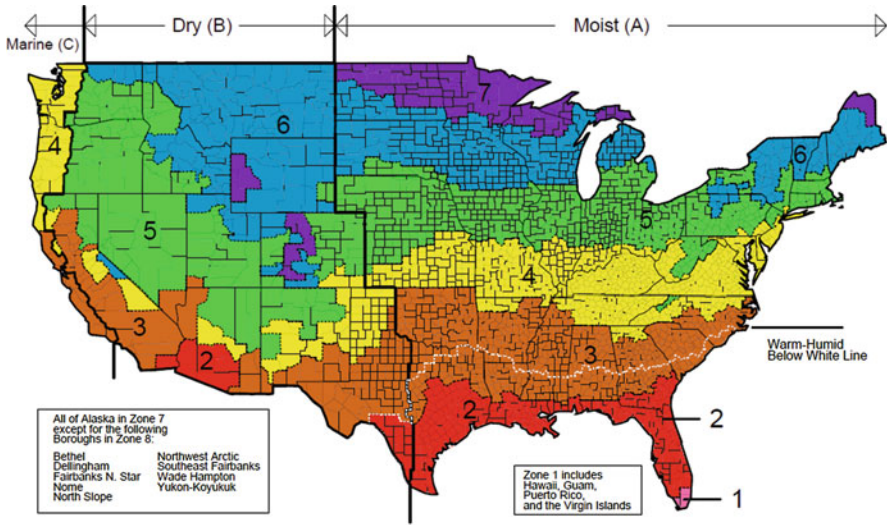
Fig. 24 Map of DOE's proposed climate zones [52, 53]

Meanwhile, the temperature or enthalpy of exhaust air balancing supply air is often lower than outdoor air in summer or higher than outdoor air in winter. Thus, heat or energy exchange between exhaust and supply is so called heat or energy recovery. For active ventilation system, in which the electrical energy consumption powers the fans, the energy saving potential could be in some cases very low or not assured, which depends on the benefit of HRV/ERV systems against the electricity consumption [32]. As a matter of fact, in some HRV/ERV systems, it is revealed from measurements that energy savings in several cases were small or even negative [3, 4, 34, 48–50]. Energy savings of HRV are affected by local climate, indoor temperature set-points, building thermal properties, ventilation air flow rate, and heat recovery system characteristics [32].

Local Climate

Zhong and Kang [51] explored the applicability of air-to-air heat recovery system in China. Through their studies, it has been found that the sensible energy saving in severe cold climate could be six times more than in hot summer and warm winter climate zone. Therefore, local climate has a significant influence on energy saving of HRV/ERV. In some location, HRV/ERV is not suitable for energy saving. ASHRAE standard [7] points out where the HRV is not suitable in not only USA but also other international zones. In USA, zones 1 and 2 as shown in Fig. 24 are not suitable for applying HRV/ERV to recover heating energy. And in zones 3C, 4C, 5B, 5C, 6B, 7, and 8, HRV is not suitable for recovering cooling energy.

Kang et al. [49] analyzed the outdoor temperature for application of HRV/ERV based on a simple mathematical model. There are two critical temperatures being

proposed, namely the first and second critical temperatures, t_{C1} and t_{C2} as derived as Eqs. 19 and 20.

$$t_{C2} = t_i - \frac{\dot{Q}_g}{\left[\rho c_p V_e + KF\right]} \tag{19}$$

$$t_{C2} = t_i - \frac{\dot{Q}_g}{\left[\rho c_p V_e (1 - \eta_s) + KF\right]} \tag{20}$$

In this equation, t_i is the set indoor temperature; $\dot{Q}_g$ is the sum of the internal heat gain; ρ, c_p are the density and specific heat capacity, respectively; V_e is the fresh air flow rate; η_s is the sensible heat recovery efficiency; K and F are the heat transfer coefficient and area, respectively.

Because of t_{C1} and t_{C2}, the winter condition of fresh air falls into three different regions: (1) **Recovery region:** outdoor air temperature is lower than the second critical temperature, i.e., $t_e < t_{C2}$. This is very common for commercial buildings in winter. And when the outdoor air temperature falls into this region, HRV can also operate as it is planned for air conditioning systems. (2) **Transition region:** outdoor air temperature is greater than the second critical temperature but less than the first critical temperature, i.e., $t_{C2} < t_e < t_{C1}$. Besides, the fresh air rate is equal to the minimum fresh air requirement while the energy consumption of the air conditioning systems with heat recovery will be in the same order or higher than that of the system without heat recovery. And the cooling or heating energy for fresh air can be uncounted. In this region, the HRV should be used in variable fresh air rate for maximum energy or heat recovery. (3) **Impermissible recovery region:** outdoor air temperature is greater than the first critical temperature in winter ($t_e > t_{C1}$). In this region, outside air is cooler than room air while room still needs to be cooled. Therefore, the heat recovery is impermissible.

Similarly, the first and second critical humidity ratio, denoted as w_{C1} and w_{C2}, respectively, are also defined as Eqs. 21 and 22. The winter outdoor humidity is divided into three different regions for latent heat recovery in HRV.

$$w_{C1} = w_i - \frac{W_i}{V_e} \tag{21}$$

$$w_{C2} = w_i - \frac{W_i}{V_e (1 - \eta_L)} \tag{22}$$

Based on the analysis, the following conclusions were drawn for HRV/ERV application in China: (a) Latent heat recovery is not suitable for ventilation energy savings in the four climate zones (including severe cold zone, cold zone, hot summer and cold winter zone, and hot summer and warm winter zone, which areas shown in Fig. 25) in winter for supermarkets of China, as the internal moisture emissions stay too high. (b) If a variable rotational speed fan is employed to introduce fresh air into the buildings, for all the cities except Guangzhou (in the hot summer and warm

Fig. 25 Climate map of China and the locations of the four selected cities (*I*: severe cold zone; *II*: cold zone; *III*: mild; *IV*: hot summer and cold winter zone; *V*: hot summer and warm winter zone)
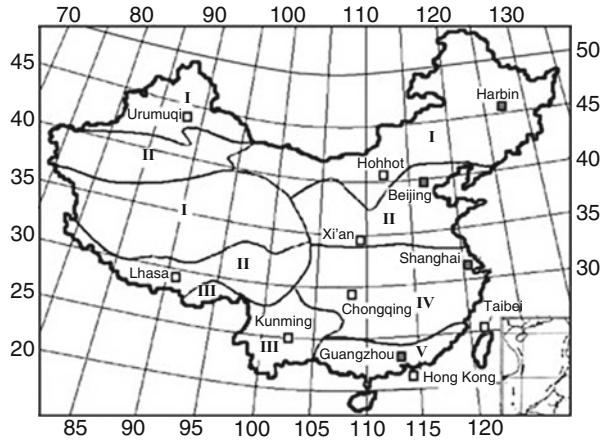
winter zone of China as shown in Fig. 25), heat recovery does always function as what it is planned in winter. If not, there is the risk of increasing energy consumption in all the cities except Harbin (in severe cold zone as shown in Fig. 25). Certainly, investment specific cost is an important factor especially for the zone where the outdoor climate falls into region II. Zhang and Kang [51] analyzed the applicability of HRV/ERV in China indifferent operations, with consideration of the investment specific cost. The above findings are mainly for the application in winter. Further studies for the summer are also needed to give some suggestions on operation optimization.

Indoor Temperature Set-Points

From Eqs. 19 and 20, it can be found indoor temperature set-points have obvious impact on feasibility of HRV/ERV. Zhou et al. [50] employed Energy Plus program to explore the energy performance of ERV along with the availability of HRV/ERV both for Beijing and Shanghai weather conditions with different indoor temperature set-points. They proposed a new concept, λ, which is defined as the ratio of total heat recovery to energy supply by conditioning equipment including HVAC devices and ERV, to evaluate the application of HRV/ERV at different indoor temperature set-points. It is illustrated from the simulated results that the seasonal average λ correlates negatively with indoor temperature set-points in winter, which is as depicted in Fig. 26. When the indoor temperature is set lower, more heat/energy recovery of ERV is available. As the cooling energy is so small in the two cities in winter that it can be ignored, the ERV is always suitable for the two cities for different indoor temperature set-point. On the contrary, in the middle seasons (spring and autumn) and summer, both heating and cooling heat recovery may occur. The seasonal average λ is positively linear with indoor temperature set-points. Besides, it has also been found that the total energy supply decreases with the increasing temperature set-point. For Beijing summer, when the set-point ranges from 24 °C to 25 °C and from 25 °C to 26 °C, respectively, the heating recovery rises equally by 39.2% while the cooling recovery decreases respectively by 30.3% and 39.1%. In

Fig. 26 λ average of
Shanghai and Beijing with
different indoor temperature
set-point in the (**a**) winter and
(**b**) the middle and hot seasons
[50]
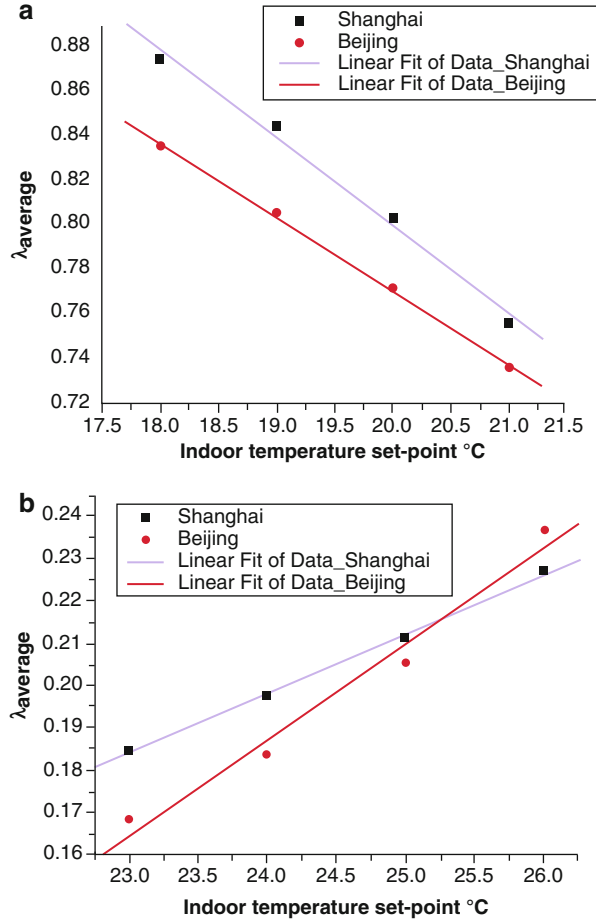
addition, when the indoor temperature set-point is higher than 24 °C, the heating
recovery will be stronger than the cooling recovery for the Beijing climate. There-
fore, in that case, the utilization of ERV is uneconomical because the heating
recovery in summer is not expected. Moreover, extra energy supply from air-
conditioning equipment is required to offset the unwanted load resulted from
ventilation in fact. For the Shanghai climate in summer, the utilization of ERV is
beneficial all the time. However, when the set-point is higher than 25 °C, this benefit
may be weakened.

Apart from the above two factors, building thermal properties may have great
impact on heat recovery. Juod is [54] defined a balance outdoor temperature for
winter as the outdoor temperature for which heat gains are equal to heat losses and
above which the HRV becomes useless. When the building is well insulated or
airtight with high occupant densities like supermarkets, the balance outdoor temper-
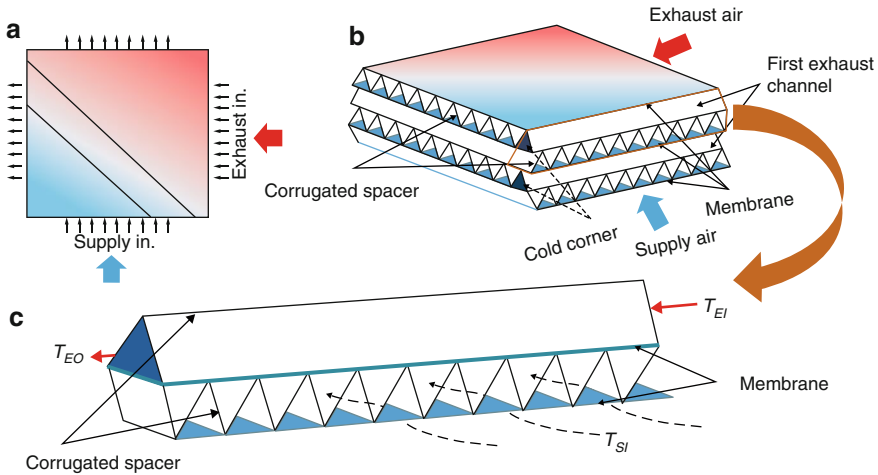ature is low. Thus, HRV may not be beneficial for long time in winter.

Fig. 27 The schematic of frosting risk with positions for membrane-based heat exchanger in HRV/ERV. (**a**) Qualitative membrane temperature distribution, (**b**) the schematic view of cross-flow exchanger structure with corrugated aluminum spacer and channels with highest risk of frosting, (**c**) a detailed view of most likely frosting spot and the research domain for developing frosting limits model [55]

Freezing

For HRV or ERV, freezing or condensation might be a problem in cold winter in some regions where warm exhaust air inside HRV/ERV units is often cooled below its dew point. As shown in Fig. 27, temperature and humidity distribute unevenly in the exhaust air channel. Therefore, the frosting position may be uneven. When frosting occurs, the pressure drop will increase and the heat exchange will become weak. Figure 28 measures the pressure drop in supply air and exhaust air channel for HRV and ERV respectively. It is clear that pressure drop increases in the exhaust side where the risk of frosting is higher, while no change in the supply side was captured [55]. Therefore, pressure drop measurement with time can be used to detect frosting. Besides, freezing inside cores eventually impede the exhaust airflow, resulting in lower efficiency of heat recovery and air change for indoor environment [56–59]. The study revealed that the onset temperature of freezing at supply airflow ranged from −3 °C to −7 °C for cross-flow heat exchanger and −8 °C to −12 °C for cross-flow energy exchanger [58, 60]. Therefore, some defrost control measures are required for both HRVs and ERVs in the region with cold climate to prevent core from freezing and blocking during cold days [58].

Compared with active HRV or ERV, passive HRV or ERV does not need fan power to promote air flow and hence to save some delivery energy. There are several types of ERV in passive natural ventilation which have already been attempted by the scientists. In common, heat pipe heat exchange was used in natural ventilation because of its low press loss [44, 61]. CFD technique was employed by Gan et al. [62] to study natural ventilation with heat recovery for a building with solar chimney

Fig. 28 The pressure drop in supply and exhaust air channel measured by Liu et al. [55]

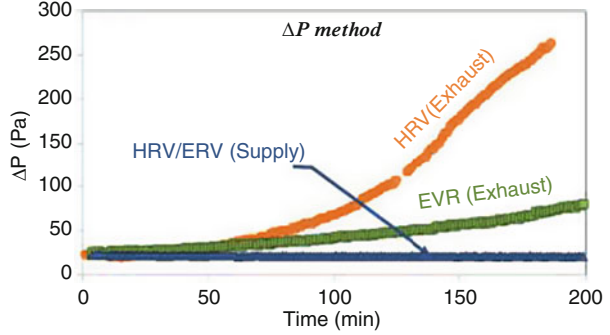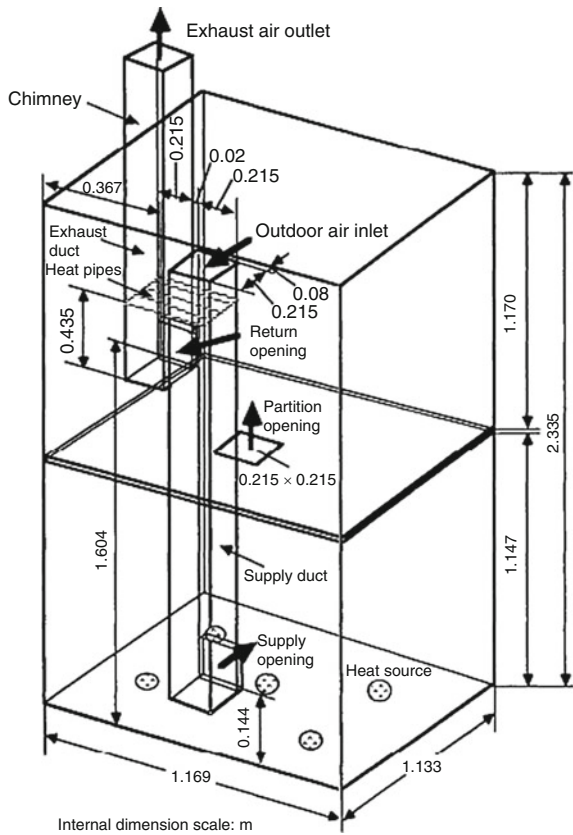**Fig. 29** Schematically shown of a two zone natural ventilation chamber with a solar chimney and heat pipe heat recovery [63]

and heat pipe heat exchange was applied for heat recovery. And some measurements were also made in a test chamber as shown in Fig. 29. It was shown that installing heat pipes in the chimney for heat recovery not only increased the flow resistance but also decreased the thermal buoyancy effect.

Other researchers [64] once promoted a conceptual design of applying a rotary wheel energy exchanger for energy recovery in natural ventilation. In such design, a rotary wheel is attached to the bottom of a wind tower, which is as shown in Fig. 30. Connor et al. [64] studied whether the air flow through a wind tower is significantly affected by the inclusion of a rotary thermal wheel by CFD method at the base of the wind tower. It is illustrated from the results that the wind tower with rotary thermal wheel can meet the guideline ventilation rates above an inlet air velocity of 3 m/s for a standard occupancy density of 1.8 m^2 per person. And no significant effect of the rotary thermal wheel on the air flow through the wind tower and into the model room was observed from the simulated results. Meanwhile, a recovery of 3 °C from the exhaust stream to the inlet stream could be generated so that up to 20% in heating costs can be avoided [64]. Therefore, this system is seemingly a promising technology for some buildings to recover heating energy without deliver energy.

Run-Around Heat Recovery System in Buildings or HVAC Systems

Run-around liquid is the core of run-around heat recovery systems. It transfers sensible and latent energy from exhaust or over heat region to supply or heat demand region with a significant distance as shown in Fig. 31. The system is categorized as comfort to comfort heat recovery system. The coupling liquid may be aqueous ethylene glycol solution or other solution such as LiBr [65] due to their antifreeze property. The liquid pump is employed for the liquid circulation between the heat and cool sides to fulfill heat or energy transportation. Obviously, it is not necessary that the supply air flow and exhaust air flow are adjacent like air to air HRV. Thus if the cross-contamination is of concern, the run-around heat recovery system is more suitable compared with twin tower energy or heat recovery system. There are at least
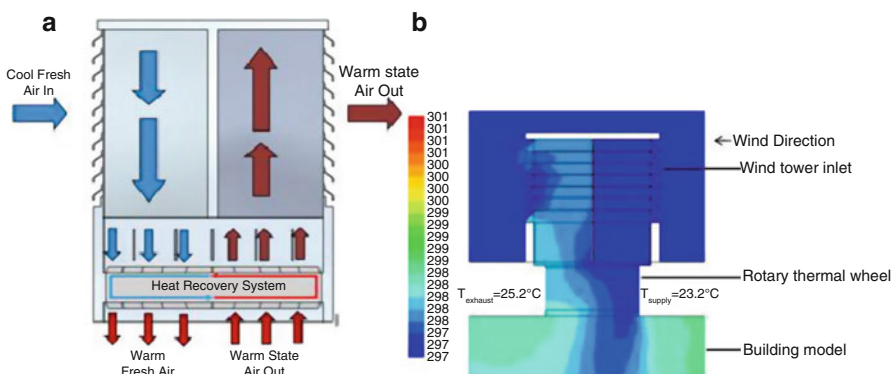
Fig. 30 Ventilation through the proposed wind tower system with integrated heat recovery technology [64]: (**a**) Schematically shown of wind tower with rotary wheel heat recovery system; (**b**) Temperature contour across the wind tower and rotary thermal wheel at an inlet velocity of 10 m/s with inlet temperature of 23.2 °C

Fig. 31 Schematic diagram of a run-around heat recovery system [65]
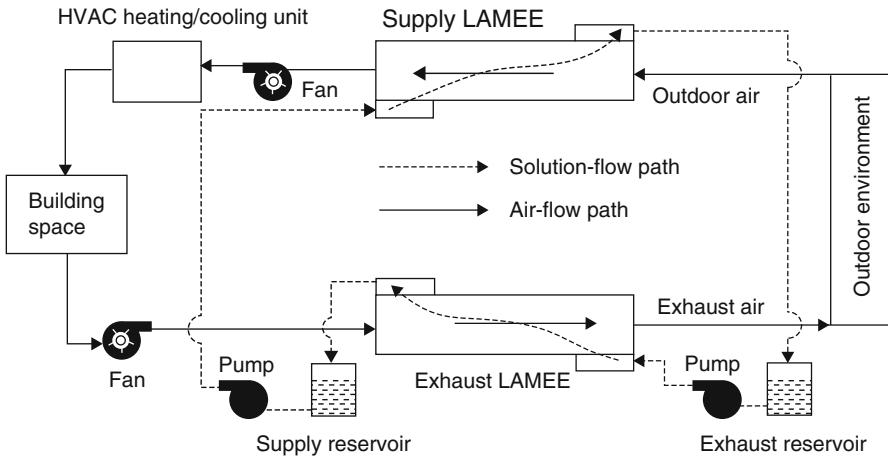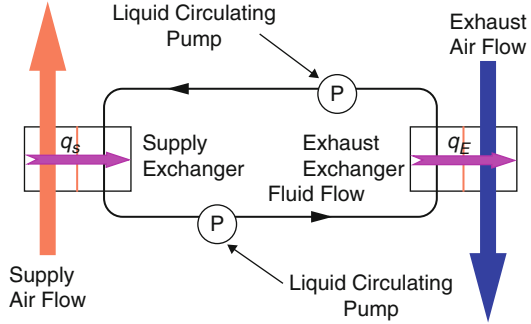
Fig. 32 Schematic of a run-around membrane energy exchanger (RAMEE) system in a HVAC line [68]

two heat exchangers in the system that are utilized to exchange heat or energy with supply and exhaust air flow respectively, which is another difference from air to air HRV and more complicated than HRV.

Apart from air to air HRV, pressure loss is an extra issue. This is because it would affect the size and energy consumption of the circulation pump. In Hviid and Svendsen's experiment, when the temperature efficiency is 75.6%, the pressure loss is 0.74 Pa [66]. Davidsson et al.'s experimental results also indicated that the pressure loss can be limited in a low range with high efficiency. Therefore, the system may be possible in passive ventilation systems [67].

To improve the design of run-around heat recovery system, new run-around membrane energy recovery system was proposed by Mahmud et al. [68], which is as illustrated in Fig. 32. In the system, membrane-based heat exchanger instead of the common metal plate heat exchanger was employed. The magnesium-chloride ($MgCl_2$) solution with a concentration of about 32% on mass basis was used as transport liquid. When the system runs, water will exchange between the air and
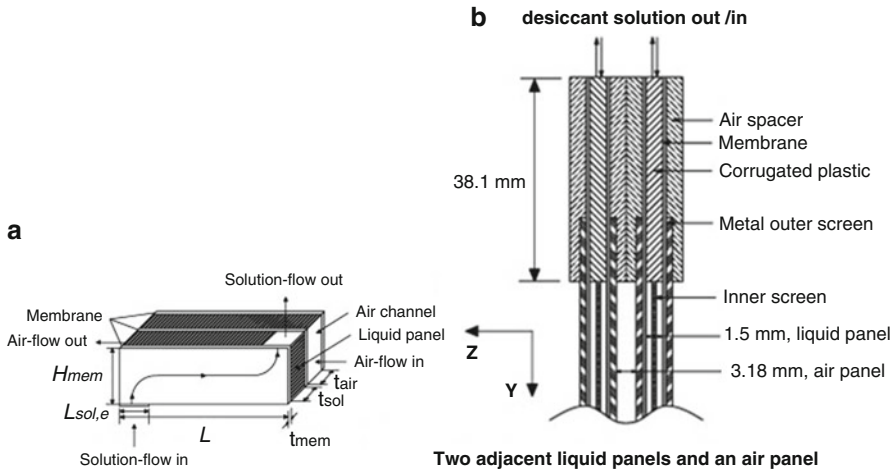
Fig. 33 The draw of counter-cross-flow liquid-to-air membrane energy exchanger: (**a**) the 3-D view; (**b**) cross-sectional front view of two adjacent liquid panels [68]

Table 2 AHRI summer and winter standard-test conditions [69]

	Summer	Winter
Outdoor air	35 °C dry bulb, 26 °C wet bulb, 49% RH	1.7 °C dry bulb, 0.6 °C wet bulb, 82% RH
Indoor air	24 °C dry bulb, 17 °C wet bulb, 49.6% RH	21 °C dry bulb, 14 °C wet bulb, 46% RH

the solution through the membrane so that the volume of desiccant solution changes. Hence, two desiccant reservoirs are placed in the system just upstream of each pump so that the volume change can be balanced. Two counter-cross-flow liquid-to-air membrane energy exchangers (LAMEE) was applied for heat and water transfer between the air and the solution. In the system, the liquid-to-air membrane was well designed with the cross-sectional front view of two adjacent liquid panel, which are as shown in Fig. 33. A plastic inner screen (0.77 mm thickness) is sandwiched between two semipermeable membranes to construct a liquid panel. Two corrugated plastic plates of 75 mm × 38.1 mm × 1.5 mm are placed at two diagonal opposite corners of a liquid panel to provide openings to the liquid headers. Desiccant solution can flow in or out through the small channels inside these corrugated plates. Each membrane is fixed with metal outer screen (12.0 mm squares and shown in Fig. 33b). Air flow channels are sandwiched by two liquid panels. Thus the assembly of 10 liquid panels with 9 air channels is bolted between 2 aluminum plates.

The system was measured through a test facility and the supply and exhaust air conditions are based on the AHRI summer and winter standard-test conditions, which are as listed in Table 2. Quasi-steady-state system effectiveness was obtained as shown in Fig. 34. From that, it can be seen how the sensible, latent and energy
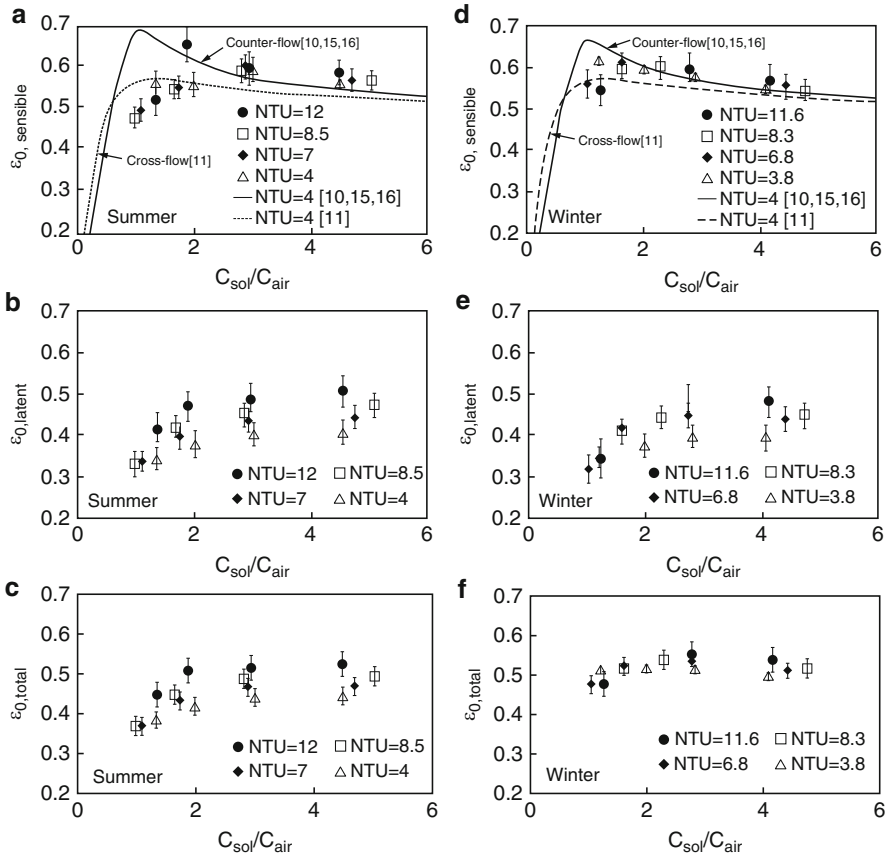
Fig. 34 Quasi-steady-state system effectiveness (sensible, latent and total) for (**a**–**c**) summer and (**d**–**f**) winter test conditions [68]

efficiency of the system varies with the ration of heat capacity rate of air and desiccant (C_{sol}/C_{air}) and *NTU* under summer and winter conditions. The system efficiency is slightly higher in the winter season than what it is in the summer season mostly because there is greater temperature difference between the supply and exhaust air streams in winter than in summer [68]. Finally, a maximum total effectiveness near to 55% at the winter operating conditions was observed when *NTU* ≈ 12 and $C_{sol}/C_{air} \geq 2$.

Heat Recovery in Heat Pump System

In some conditions, one part of rooms need be heated while another part of rooms need to be cooled in a building. When these rooms were equipped with heat pump in a cooling or heating system, heat or energy can be extracted from the rooms with

Fig. 35 Schematic draw of heat pump for cooling and heating rooms simultaneously
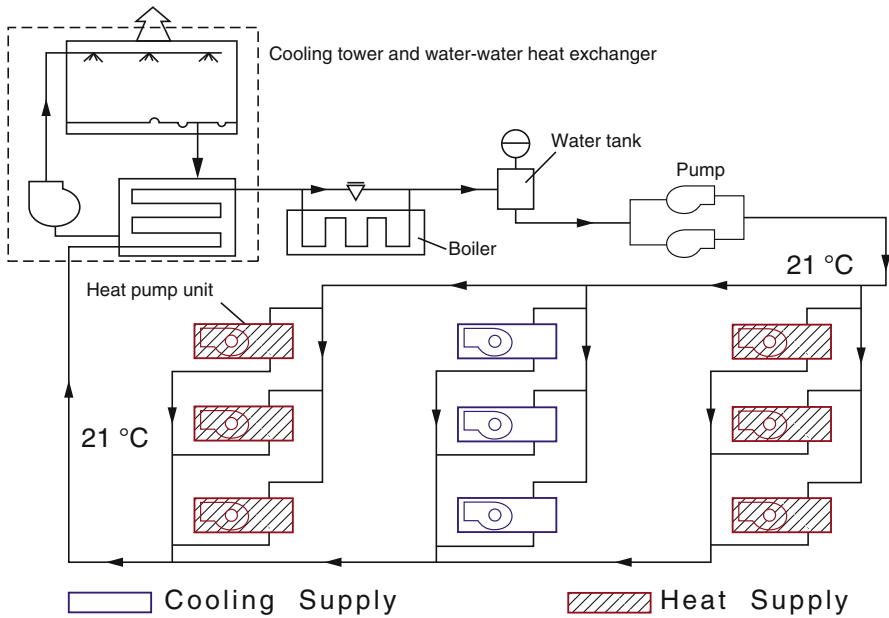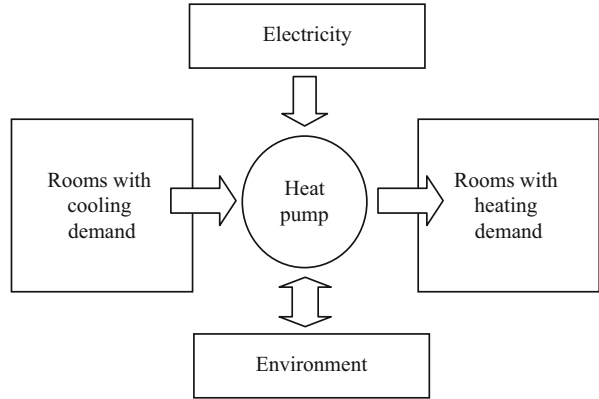
Fig. 36 The structure of WLHPS [70]

cooling demand to supply to satisfy the heating demand of other rooms, which is as shown in Fig. 35. That is deemed as a special form of heat or energy recovery.

Water-loop heat pump system (WLHPS) is such system which serves different rooms with different energy demands. The system was first put forward in 1960s and has been applied as a kind of energy saving for centralized air conditioning systems. The working principle of this system is shown in Fig. 36. It is composed with some heat pumps together with closed water-piping loop. The water is deemed as heat source when the heat pumps work for heating the rooms or as heat sink when the heat

Fig. 37 Suitable applied
areas of WLHPS when
electric boiler used in China
(North area of the dash line)
[70]

pumps serve for cooling the rooms. Therefore, the heat extracted from the rooms
with cooling requirement can be recovered into the loop-water and then be released
into the rooms with heating requirement. For example, inner zones in a building are
needed to be cooled even in winter while outer zones needed to be heated. The heat
discharged from the inner zones can be transferred to outer zones by the circulation
water through WLHPS and heat recovery can be realized in this way.

Lian et al. [70] proposed a mathematic model to analyze characteristics of energy
saving of the WLHPS in China. According to the results from the research, it was
indicated that the WLHPS cooperating with electric boiler has obvious impact over
the energy saving for the most districts in China particularly in the areas along the
Coast of Yangtze River and its northern parts as shown in Fig. 37. This system will
not save energy if it is used in south China. That is because in those areas the cooling
load is much larger than heating load so that no extra heat or energy can be recovered.
In common, the moderate inner load and the moderate ratio of inner zone area to outer
zone area are most suitable to the application of the WLHPS, particularly under the
condition that the WLHPS can be applied in the buildings where the ratio of cooling
load to heating load is not larger than 42 [70]. In addition, their calculation found that
the maximum rate of energy saving can reach to 19.29%, and the most suitable ratio of
cooling load to heating load for the WLHPS is about 4.8.

Buonomano et al. [71] also developed a purposely designed performance simu-
lation model for WLHPS systems. Using the model, the authors conducted assess-
ment over the hourly, daily, and seasonal HVAC system performance in Europe
climate zones. Based on the analysis of some cases, comparison of energy saving
between WLHP and traditional HVAC systems (including the system with water to
water electric chiller and traditional natural gas boiler; and the system air to water
electric chiller and traditional natural gas boiler) was made. According to the results,
it was shown that building heating and cooling simultaneity would influence relative
primary energy saving of the WLHP system versus traditional systems [71].
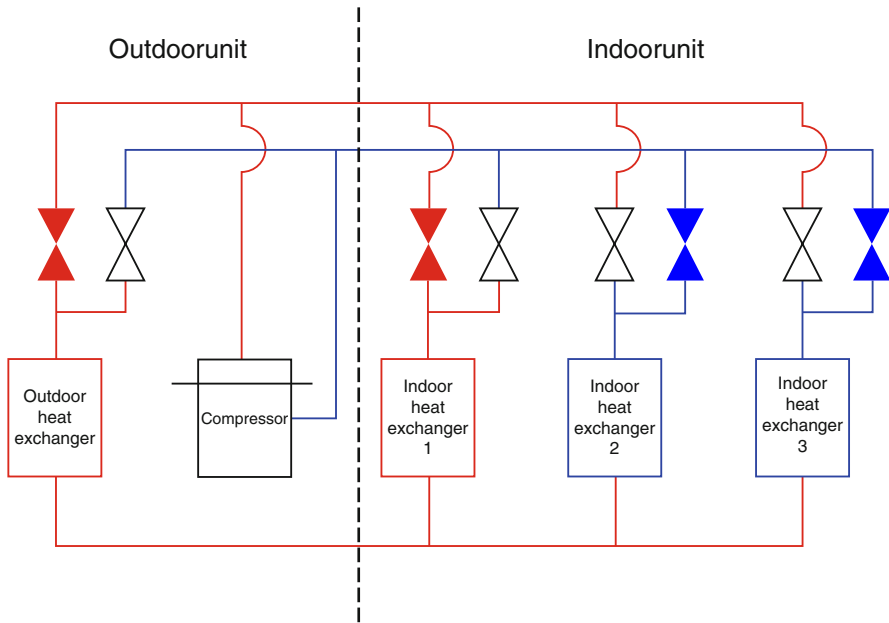
Fig. 38 Schematic diagram of HR-VRF system [72]

Concurrently, relative primary energy saving also depends on the increase of the electricity to gas consumption ratio obtained by shifting from traditional to WLHP systems. It was found that the highest primary energy savings were detected in climatic areas around 3000 Kd in Europe. And for the eventual operating economic cost, the saving by WLHP versus the considered traditional systems is largely decided by national electricity and natural gas costs. The best results are found in Denmark and northern France (between 35% and 40%).

Heat recovery cooperating with variable refrigerant flow (VRF) system called as HR-VRF system is another system suitable for the modern buildings with big inner zones. The schematic diagram of the structure and the recycle of the refrigerant of HR-VRF system are shown in Fig. 38. Indoor unit 1 is running for heating outer zones in the building with its heat exchanger as a condenser of the heat pump while indoor units 2 and 3 are running for cooling inner zones with their heat exchangers as evaporators. When the heating capacity of indoor unit 1 and the cooling capacity of indoor units 2 and 3 are in the balance, the outdoor heat exchanger closes to release no heat to the outdoor environment. Thus the heat can be recovered from indoor units 2 and 3 and afterward be released into indoor unit 1.Or, the outdoor exchanger supplies heating capacity or cooling capacity to balance the heat among the indoor units. There is still some heat recovery from indoor units 2 and 3into indoor unit 1. Li and Wu [72] developed a new energy simulation module to evaluate the energy features of the system. The simulated results illustrated that 6–17% energy can be saved through the HR-VRF system. As VRF system is a very popular air

Fig. 39 A typical residential shower installation with a single-pass counter-flow heat exchanger [73]

conditioning system, lots of theoretical and experimental studies have been conducted. The details for heat recovery performance in the system may refer to these papers.

Other Possible Waste Heat Recovery System in Buildings

In some cases, heat or energy will be consumed together with some waste medium in buildings. Hot water for personal hygiene such as bathing or showering may consume 20% of the building energy [73]. Large amount of water with higher temperature than indoor air is drained and some energy will be wasted during bathing or showering. Therefore, the heat recovery equipment was installed in the drainage pipe to recapture the heat. Wong et al. [73] installed a single-pass counter-flow shell-to-tube heat exchanger beneath the shower drain in their experimental high-rise residential buildings of Hong Kong, which is as shown in Fig. 39. Thus the heat was extracted from waste water to transfer into the cold supply water. Then the heating water was reheated by instantaneous water heater. The thermal energy exchange is evaluated via the effectiveness-number of transfer units (ε-NTU) approach. The investigation indicated that the system can recover 4–15% shower water heat through a 1.5 m long single-pass counter-flow heat exchanger for a drainage pipe of diameter 50 mm [73]. Through simulation, potential energy saving is expected to positively correlate with heat transfer area and drainage pipe diameter [73]. Certainly, space limitations for the installations, designs of heat exchanger with justified payback period will affect the application of the recovery system.

In some regions, fireplace is used to heat indoor space by burning fuels or firewood to produce and transfer heat. In the heating process, 10–20% heat is exhausted with waste smoke out of chimney. Mahmoud et al. [74] proposed a waste heat recovery system, which can be applied to chimney for the extraction of the heat from exhaust smoke to heat water to produce hot water in Lebanon. During experiments, the water temperature increased from 10 °C to 78 °C in 1 h. Through analysis, it was found that the average total heat transfer rate gained by water was 7.54 kW, including 2.33 kW from convection with the pipes and 5.21 kW from convection and radiation at the bottom surface of the water tank [74]. Although the system is only a prototype, it looks promising for the rural region in some developing countries. By the same token, the system may also be applied in some kitchen exhaust smoke duct or some localized heating design such as Kang in China [75] or Ondol in Korea [76].

Summary

After the introduction of the concept of heat recovery and the components of a general heat recovery system, a literature review on some heat recovery technologies in buildings was made. It has been found that some kinds of heat recovery technologies for different situation were proposed and many theoretic and experimental works have been conducted to verify, evaluate, and improve the heat or energy recovery efficiency. It has been proven from the research that due amount of energy can be saved through these technologies in the indoor environment. In the past, the heat or energy recovery technologies emphasized on the efficiency of heat and/or mass transfer in the heat or energy exchanger. However, the concern on the coupling with other technologies and integrated heat or energy recovery efficiency is becoming a trend because the energy saving expectation of the heat or energy recovery in building depends on both the heat or energy exchanger and many other factors such as outdoor climate, HVAC system types, operation schedule, and indoor environment setting.

Nowadays, heat or energy recovery system is applied mostly in ventilation system, including natural and mechanical ventilation. Especially, in passive house, heat or energy recovery system in natural ventilation almost becomes a required item, though this application might be limited by some problems such as economic cost and outdoor pollution. Besides, heat pump extends the application of heat recovery. In some technologies such as WLHPS and VRF, heat or energy recovery is implied when heating and cooling requirement coexist in a building. However, a handful amount of combination with other technologies such as evaporative cooling and desiccant dehumidification is made and few studies have been performed to test the efficiency and improve them [10]. In addition, some new technologies such as heat recovery technologies integrated with photovoltaic panel, solar energy collector, and other renewable technologies sound operable. However, they were not mentioned in this chapter because few research can be found.

References

1. Laustsen J (2008) Energy efficiency requirements in building codes, energy efficiency policies for new buildings. OECD/IEA, Paris
2. Cuce PM, Riffat S (2015) A comprehensive review of heat recovery systems for building applications. Renew Sustain Energy Rev 47:665–682
3. Fehrm M, Reiners W, Ungemach M (2002) Exhaust air heat recovery in buildings. Int J Refrig 25(4):439–449
4. Lazzarin RM, Gasparella A (1998) Technical and economical analysis of heat recovery in building ventilation systems. Appl Therm Eng 18(1):47–67
5. Natural Resources Canada (2015) National Energy Code of Canada for Buildings. NECB-2015, NRC, Ottawa, Canada
6. Ministry of Housing and Urban-Rural Development of China (2014) Green Building Evaluation Labelling Standard. GB/T 50378–2014, China Architecture and Building Press, Beijing, China
7. American Society of Heaing, Refrigeration and Air-Conditioning (ASHRAE) (2010) Energy Standard for buildings except low-rise residential buildings. ANSI/ASHRAE/IESNA Standard 90 1-2007, Atlanta
8. Riffat S, Gan G (1998) Determination of effectiveness of heat-pipe heat recovery for naturally-ventilated buildings. Appl Therm Eng 18(3):121–130
9. Shurcliff WA (1998) Air-to-air heat-exchangers for houses. Annu Rev Energy 13(1):1–22
10. Mardiana-Idayu A, Riffat S (2012) Review on heat recovery technologies for building applications. Renew Sustain Energy Rev 16(2):1241–1255
11. Besant RW, Simonson CJ (2000) Air-to-air energy recovery. ASHRAE J 42(5):31
12. ASHRAE (2012) Chapter 26: air-to-air energy recovery equipment. In: ASHRAE handbook. HVAC System and Equipment, Atlanta
13. Mei L, Infield D, Eicker U et al (2006) Cooling potential of ventilated PV façade and solar air heaters combined with a desiccant cooling machine. Renew Energy 31(8):1265–1278
14. Nasif M, Al-Waked R, Morrison G et al (2010) Membrane heat exchanger in HVAC energy recovery systems, systems energy analysis. Energ Buildings 42(10):1833–1840
15. Lu Y, Wang Y, Zhu L et al (2010) Enhanced performance of heat recovery ventilator by airflow-induced film vibration (HRV performance enhanced by FIV). Int J Therm Sci 49(10):2037–2041
16. Fernández-Seara J, Diz R, Uhía FJ et al (2011) Experimental analysis of an air-to-air heat recovery unit for balanced ventilation systems in residential buildings. Energ Conver Manage 52(1):635–640
17. Chen X, Su Y, Aydin D et al (2016) Experimental investigations of polymer hollow fibre heat exchangers for building heat recovery application. Energ Buildings 125:99–108
18. Rajendran S, Kalaikadal DS, Manglik RM (2013) Characterization and prediction of swirl-induced enhanced heat transfer in sinusoidal-corrugated plates. ASHRAE Trans 119(2):1–8
19. Metwally H, Manglik RM (2004) Enhanced heat transfer due to curvature-induced lateral vortices in laminar flows in sinusoidal corrugated-plate channels. Int J Heat Mass Transf 47(10):2283–2292
20. Vasiliev LL (2005) Heat pipes in modern heat exchangers. Appl Therm Eng 25(1):1–19
21. Shao L, Riffat S (1997) Flow loss caused by heat pipes in natural ventilation stacks. Appl Therm Eng 17(4):393–399
22. Yau Y (2008) The heat pipe heat exchanger: a review of its status and its potential for coolness recovery in tropical buildings. Build Serv Eng Res Technol 29(4):291–310
23. Zhang L, Lee W (2011) Evaluating the use heat pipe for dedicated ventilation of office buildings in Hong Kong. Energ Conver Manage 52(4):1983–1989
24. Ong K (2016) Review of heat pipe heat exchangers for enhanced dehumidification and cooling in air conditioning systems. Int J Low Carbon Technol 11(3):416–423. https://doi.org/10.1093/ijlct/CTU029
25. Waugaman D, Kini A, Kettleborough C (1993) A review of desiccant cooling systems. J Energy Resour Technol 115(1):1–8

26. La D, Dai Y, Li Y et al (2010) Technical development of rotary desiccant dehumidification and air conditioning: a review. Renew Sustain Energy Rev 14(1):130–147
27. Dallaire J, Gosselin L, Da Silva AK (2010) Conceptual optimization of a rotary heat exchanger with a porous core. Int J Therm Sci 49(2):454–462
28. Swanepoel D, Kröger D (1996) Rotary regenerator design theory and optimisation. R&D J 12:90–97
29. Shiminski J. http://www.dac-hvac.com/energy-recovery-wheels-understanding-cross-contamination-leakage, Posted on June 6, 2012
30. Min J, Su M (2010) Performance analysis of a membrane-based energy recovery ventilator: Effects of membrane spacing and thickness on the ventilator performance. Appl Therm Eng 30(8):991–997
31. Laverge J, Janssens A (2012) Heat recovery ventilation operation traded off against natural and simple exhaust ventilation in Europe by primary energy factor, carbon dioxide emission, household consumer price and exergy. Energ Buildings 50:315–323
32. El Fouih Y, Stabat P, Rivière P and et al (2012) Adequacy of air-to-air heat recovery ventilation system applied in low energy buildings. Energ Buildings 54: 29-39.
33. ASHRAE (2013) Chapter 21: Duct design. In: ASHRAE Handbook: fundamentals. ASHRAE, Atlanta
34. Roulet CA, Heidt F, Foradini F et al (2001) Real heat recovery with air handling units. Energ Buildings 33(5):495–502
35. Zhang LZ (2009) Flow maldistribution and thermal performance deterioration in a cross-flow air to air heat exchanger with plate-fin cores. Int J Heat Mass Transf 52(19):4500–4509
36. Mardiana A, Riffat S (2013) Review on physical and performance parameters of heat recovery systems for building applications. Renew Sustain Energy Rev 28:174–190
37. Kays W, London A (1984) Compact heat exchangers, 3rd edn. McGraw-Hill, New York
38. Niu J, Zhang L (2001) Membrane-based enthalpy exchanger: material considerations and clarification of moisture resistance. J Membr Sci 189(2):179–191
39. Nellis GF, Pfotenhauer JM (2005) Effectiveness-NTU relationship for a counterflow heat exchanger subjected to an external heat transfer. J Heat Transfer 127(9):1071–1073
40. Yau YH (2007) Experimental thermal performance study of an inclined heat pipe heat exchanger operating in high humid tropical HVAC systems. Int J Refrig 30(7):1143–1152
41. Zhang Land Jiang Y (1999) Heat and mass transfer in a membrane-based energy recovery ventilator. J Membr Sci 163(1):29–38
42. Ge T, Li Y, Wang R et al (2008) A review of the mathematical models for predicting rotary desiccant wheel. Renew Sust Energy Rev 12(6):1485–1528
43. Min J, Su M, Wang L (2012) Experimental and theoretical investigations of membrane-based energy recovery ventilator performance. Int J Air-Cond Refrig 20(1):115–119
44. Shao L, Riffat S, Gan G (1998) Heat recovery with low pressure loss for natural veltilation. Energ Buildings 28(2):179–184
45. Min J and Wang L (2011) Membrane sorption property effects on transmembrane permeation. Chin Sci Bull 56(22): 2394–2399.
46. Yaïci W, Ghorab M, Entchev E (2013) Numerical analysis of heat and energy recovery ventilators performance based on CFD for detailed design. Appl Therm Eng 51(1):770–780
47. Orme M (2001) Estimates of the energy impact of ventilation and associated financial expenditures. Energ Buildings 33(3):199–205
48. Simonson C (2005) Energy consumption and ventilation performance of a naturally ventilated ecological house in a cold climate. Energ Buildings 37(1):23–35
49. Kang Y, Wang Y, Zhong K et al (2010) Temperature ranges of the application of air-to-air heat recovery ventilator in supermarkets in winter, China. Energ Buildings 42(12):2289–2295
50. Zhou Y, Wu J, Wang R (2007) Performance of energy recovery ventilator with various weathers and temperature set-points. Energ Buildings 39(12):1202–1210
51. Zhong K, Kang Y (2009) Applicability of air-to-air heat recovery ventilators in China. Appl Therm Eng 29(5):830–840

52. Briggs RS, Lucas R, Taylor Z (2003) Climate classification for building energy codes and standards: part 2-zone definitions, maps, and comparisons.ASHRAE. Transactions 109(1):122–130

53. Available from: http://energy-models.com/map-doe's-proposed-climate-zones

54. Juodis E (2006) Extracted ventilation air heat recovery efficiency as a function of a building's thermal properties. Energ Buildings 38(6):568–573

55. Liu P, Rafati Nasr M, Ge G et al (2016) A theoretical model to predict frosting limits in cross-flow air-to-air flat plate heat/energy exchangers. Energ Buildings 110:404–414

56. Fisk W, Archer K, Chant R et al (1984) Performance of residential air-to-air heat exchangers during operation with freezing and periodic defrosts. Lawrence Berkeley Lab., Berkeley

57. Kragh J, Rose J, Svendsen S(2005). Mechanical ventilation with heat recovery in cold climates. In: Proceedings of the 7th symposium on building physics in the Nordic Countries, pp 1–8

58. Zhang J, Fung AS (2015) Experimental study and analysis of an energy recovery ventilator and the impacts of defrost cycle. Energ Buildings 87:265–271

59. Rafati Nasr M, Fauchoux M, Besant RW et al (2014) A review of frosting in air-to-air energy exchangers. Renew Sustain Energy Rev 30:538–554

60. Fisk W, Chant R, Archer K et al (1985) Onset of freezing in residential air-to-air heat exchangers. ASHRAE Trans 91(1):145–158

61. Hughes BR, Chaudhry HN, Calautit JK (2014) Passive energy recovery from natural ventilation air streams. Appl Energy 113:127–140

62. Gan G, Riffat S (1998) A numerical study of solar chimney for natural ventilation of buildings with heat recovery. Appl Therm Eng 18(12):1171–1187

63. Gan G, Riffat S (1998) Measurement and simulation of air flow in a two-zone chamber with heat-pipe heat recovery. Int J Ambient Energy 19(2):93–103

64. O'Connor D, Calautit JK, Hughes BR (2014) A study of passive ventilation integrated with heat recovery. Energ Buildings 82:799–811

65. Vali A, Simonson J, Besant RW et al (2009) Numerical model and effectiveness correlations for a run-around heat recovery system with combined counter and cross flow exchangers. Int J Heat Mass Transf 52(25):5827–5840

66. Hviid CA, Svendsen S (2011) Analytical and experimental analysis of a low-pressure heat exchanger suitable for passive ventilation. Energ Buildings 43(2):275–284

67. Davidsson H, Bernardo R, Hellström B (2013) Theoretical and experimental investigation of a heat exchanger suitable for a hybrid ventilation system. Buildings 3(1):18–38

68. Mahmud K, Mahmood GI, Simonson CJ et al (2010) Performance testing of a counter-cross-flow run-around membrane energy exchanger (RAMEE) system for HVAC applications. Energ Buildings 42(7):1139–1147

69. Air-Conditioning, Heating, and Refrigeration Institute (AHRI) (2005) Performance rating for air-to-air exchangers for energy recovery ventilation equipment. AHRI STANDARD 1060-2005, Arlington

70. Lian Z, Park SR, Qi H (2005) Analysis on energy consumption of water-loop heat pump system in China. Appl Therm Eng 25(1):73–85

71. Buonomano A, Calise F, Palombo A (2012) Buildings dynamic simulation: water loop heat pump systems analysis for European climates. Appl Energy 91(1):222–234

72. Li YM, Wu JY (2010) Energy simulation and analysis of the heat recovery variable refrigerant flow system in winter. Energ Buildings 42(7):1093–1099

73. Wong LT, Mui KW, Guan Y (2010) Shower water heat recovery in high-rise residential buildings of Hong Kong. Appl Energy 87(2):703–709

74. Khaled M, Ramadan M, El Rabet MG et al (2013) Heating water using the recovered chimney waste heat-prototype and experimental analysis. In: Proceedings of the 25th international conference on microelectronics (ICM). IEEE, pp 1–4

75. Zhuang Z, Li Y, Chen B et al (2009) Chinese kang as a domestic heating system in rural northern China – a review. Energ Buildings 41(1):111–119

76. Son K, Kim S, Kim JT (2013) Thin flooring panel development for energy efficiency of an ondol heating system. Indoor Built Environ 22(1):131–138