Renewable and Sustainable Energy Reviews 162 (2022) 112417

Contents lists available at ScienceDirect

Renewable and Sustainable Energy Reviews

journal homepage: www.elsevier.com/locate/rser

A review of heat recovery technologies and their frost control for residential building ventilation in cold climate regions

H.Y. Bai [a], P. Liu [b,*], M. Justo Alonso [a,b], H.M. Mathisen [a]

[a] Department of Energy and Process Engineering, Norwegian University of Science and Technology, 7491, Trondheim, Norway
[b] Department of Architecture, Materials and Structures, SINTEF Community, 7465, Trondheim, Norway

ARTICLE INFO

Keywords:
Heat recovery
Cold climate
Indoor air quality
Frosting control
Energy-efficient ventilation

ABSTRACT

This study reviewed heat recovery technologies and their application in residential buildings in cold climates. The increasing efforts on reducing building energy use over the recent years have resulted in improved insulated and airtight buildings. Reduced infiltration can result in poor indoor air quality owing to insufficient ventilation rates.

In cold climates, an energy-efficient mechanical ventilation system with heat recovery provides controlled airflow rates and thereby ensures healthy indoor air quality while using a limited amount of energy. This is of particular importance during the heating season when the windows and doors are closed to sustain indoor thermal comfort. This review examines the various heat recovery technologies employed in residential buildings, including state-of-the-art approaches adapted to cold climates. These technologies were compared in terms of various criteria.

Frosting has been observed frequently in heat exchangers used in cold regions, which reduces their performance. Various frosting control strategies were discussed in this review including frosting prevention and defrosting. Currently, available frost control strategies degrade indoor air quality or result in increased energy consumption. Finally, The effects of applying heat recovery for mechanical ventilation on indoor air quality were addressed. Based on the current review, recommendations for future research are proposed.

1. Introduction

Heating, ventilation, and air-conditioning (HVAC) consumption has been increasing significantly as a result of global economic growth, the expansion of building development, and the improvements in living standards [1]. For example, 40%, 39%, and 40% of the primary energy consumption and 38%, 36%, and 30–40% of CO_2 emissions are attributed to the building sector in the US [2], Europe [3], and China [4,5], respectively.

A residential building is defined as a building with primary habitational functions. This includes several types of buildings, such as single or multiple family houses, dormitories, apartments, and hotels [6]. Although well insulated and airtight buildings have been built to reduce the energy consumption in residential buildings, this often leads to the problem of poor indoor air quality (IAQ) owing to the lack of infiltration [7]. Air purifiers or air cleaners have gained prominence as a way to improve the IAQ particularly in the heavily polluted regions. However, they may be inefficient and cause issues such as noise and ozone gases [8]. Although natural ventilation can provide a relatively good IAQ [7] given good outdoor air quality, it may yield high energy use for space heating is in cold regions in winter and shoulder seasons [9]. Consequently, mechanical ventilation has emerged as the primary method for ensuring IAQ and thermal comfort in most newly built commercial and residential buildings located in cold climate regions [10]. With mechanical ventilation, the heat loss from residential building ventilation systems can reach 35–40 kWh/m^2 [11]; nevertheless, 60–95% of this ventilation heat loss can be recovered using a heat recovery system [12].

Several literature reviews have focused on heat recovery in hot and humid climates [13–17]. Additionally, a number of reviews on heat recovery for building applications have been conducted [12,18,19]; However, to the authors' knowledge, substantial review work focused on the advances and the applications of heat recovery in cold climate regions are not found. One review focused on only the frosting problem and common defrosting measures in air-to-air heat exchangers in cold climate regions [20]. Whereas another review [21] focused on the heat recovery technology in cold climates for Zero Energy Buildings; however, the problem of frosting was not addressed. This study conducted a

* Corresponding author.
E-mail address: peng.liu@sintef.no (P. Liu).

https://doi.org/10.1016/j.rser.2022.112417
Received 3 December 2021; Received in revised form 25 March 2022; Accepted 29 March 2022
Available online 16 April 2022

Nomenclature

A	rotary wheel facial area (m^2)
F	rotary wheel air flow rate (m^3/h)
h	convective heat transfer coefficient (W/m^2K)
R	heat resistance (K/W)
T	temperature ($^\circ C$)

Greeks

δ	thickness (m)
ε	heat exchanger effectiveness
η	heat recovery efficiency
λ	thermal conductivity (W/mK)
τ	time (sec)
Φ	recovered heat (kW)

Subscripts

ave	average
c	cold
cov	convection
D	dew point
e	exhaust air
exh	extracted heat
f	frost layer
h	hot
i	inlet
mix	mixed
o	outlet
P	plate
r	ratio
rec	recovered
s	supply
w	warm

Abbreviations

ACH	air change per hour
AHU	air handling unit
ASHP	air source heat pump
COP	coefficient of performance
DCHE	desiccant coated heat exchanger
EAIW	exhaust air insulation wall
EMF	effective mass-flow fraction
FBRs	fixed-bed regenerators
FMHPA	flat micro heat pipe array
HDD	heating degree day
HRV	heat recovery ventilation system
HVAC	heating, ventilation and air-conditioning
IECC	international energy conservation code
IAQ	indoor air quality
PCMs	phase change materials
NTU	number of heat transfer unit
PLHP	pump-driven loop heat pipe system
RH	relative humidity (%)
SARS-CoV-2	severe acute respiratory syndrome corona virus 2
rpm	revolutions per minute
TVE	thermal variation easiness
US	united states
VAV	variable air volume system
VOCs	volatile organic compounds

comprehensive review of state-of-the-art heat recovery technologies and ongoing research activities within the field of residential buildings, specifically focusing on their applications in cold climate regions.

The review paper is structured as follows. Section 1 provides the background of this review by introducing the classification of cold climate regions and the design parameters of ventilation with heat recovery in cold climates. Section 2 explains the main types of heat recovery technologies for residential building applications in cold climate regions. A comprehensive comparison between different technologies is presented in terms of various criteria. Section 3 presents a detailed discussion on the frosting issue in heat recovery applications. Various frosting control strategies for different applications are compared and summarised. Section 4 explains the effects of applying heat recovery for mechanical ventilation on IAQ. Section 5 summarises the review by listing the primary conclusions and Section 6 proposes recommendations on future research. This review aims to provide researchers, project developers and ventilation designers with broad information and insights on heat recovery technologies and frosting control strategies in cold climate regions in terms of heat recovery effectiveness, energy-saving potential, frosting control, and potential impacts on IAQ.

1.1. Cold climate regions

For habitable areas, the heating degree days (HDDs) method has been adapted for climate classification [22]. Building America [23] determines building practices in the United States based on climate zones classified using HDDs, average temperatures, and precipitation. A Cold climate is defined as a region with 5400–9000 HDDs (18.3 $^\circ C$ as a base temperature); a Very-cold climate is defined as a region with 9000–12,600 HDDs; and a Subarctic climate is defined as a region with 12,600 HDDs or more. International Energy Conservation Code (IECC) map was also developed to provide a simplified and consistent approach to defining climate for implementation of various codes. Cold climate, Very-cold climate and Subarctic climate defined in Building America correspond to IECC climate zones 5, 6, 7, and 8 respectively.

Annual HDDs at the base temperature of 18.3 $^\circ C$ is illustrated in Fig. 1 [24]. In general, cold climate covers most of the areas north of 40° North latitude, with the coldest month average temperature below 0 °C. A significant temperature difference between indoor and outdoor environment results in enormous heat loss potential. Moreover, the outdoor air with sub-zero temperature also introduces the challenge of frosting for heat recovery systems during the heating season.

1.2. Design parameters of ventilation with heat recovery in cold climate regions

Design parameters are critical for calculating the potential of heat recovery in cold climate regions. They include the air flow rate and target IAQ during the heating season. In certain countries, a minimum air change per hour (ACH) is prescribed for the whole building. ACH is a measure of air volume being exchanged within 1 h [25].

In addition, a minimum amount of sensible or latent heat recovered is also required or recommended in certain countries as a guideline. It is expressed as the minimum heat recovery efficiency, which is defined by Ref. [26]:

$$\eta = \frac{\Phi_{rec}}{\Phi_{exh}} \tag{1}$$

where Φ_{rec} is the recovered sensible or latent heat (kW) and Φ_{exh} is the extracted sensible or latent heat (kW).

The guidelines, standards, and regulations of design parameters for residential buildings in various countries in cold climate are listed in Table 1.

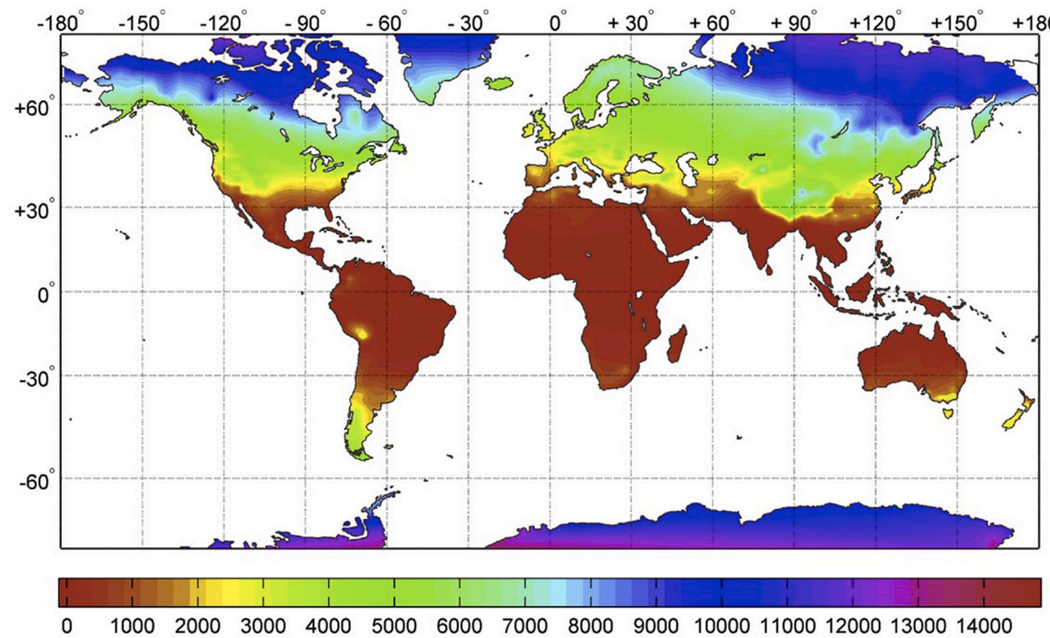


Fig. 1. Distribution of annual heating degree days (HDDs) at the base temperature of 18.3 °C [24].

2. Main types of heat recovery technologies for residential building applications in cold climate regions

The minimum average temperature of outdoor air can go below −30 °C in winter [50]. Supplying the unconditioned outdoor air directly into occupied spaces can result in thermal discomfort if no further heating is applied [51] and warming it to customary supply temperatures can result in substantial heating demands.

Heat recovery technologies have thus been commonly used in recent years to reduce the heating demands of residential buildings. Heat or thermal energy can be divided into sensible heat (energy) and latent heat (energy). During the heating season, the sensible energy in exhaust air can be transferred to the supply air, increasing its temperature while reducing the sensible heating demand for the residential building. Heat recovery devices used for recovering the sensible heat alone are the dominant type, and most countries that experience cold climates have enforced building regulations regarding the minimum sensible heat recovery efficiency, as presented in Table 1. Moisture in the exhaust air can also be transferred to the dry supply air provided total energy recovery technologies are applied. Although mechanical ventilation with heat recovery can be used as a stand-alone ventilator for a single room, in the case of other traditional centralised ventilation systems, it should be combined with the existing air-handling unit (AHU) [52]. Furthermore, indoor thermal comfort and moisture levels have garnered increasing attention over recent years; however, specific requirements about the latent heat recovery effectiveness have not been widely established in existing building regulations or standards.

A complete heat recovery ventilation system (HRV) typically comprises heat recovery units, fans, air ducts, exhaust air ducts, and air diffusers [12]. A schematic view of a typical HRV system in a residential building is presented in Fig. 2. In the following subsections, the main types of heat recovery technologies are presented and discussed.

2.1. Rotary wheels

Rotary wheels have been broadly used for heat recovery for building applications. Based on the characteristics of the wheel coating surface, they can be mainly classified into heat wheels and enthalpy wheels (or desiccant wheels). Heat wheels are designed primarily to recover sensible energy via transferring the sensible heat from the warm air to the cold air. In the enthalpy wheels additionally to sensible heat transfer, the moisture in the airflow with higher moisture levels can be transferred to the airflow with lower moisture levels.

The basic construction of rotary wheels is a rotating metallic wheel driven by a motor, where two airstreams are separated with brush lists. The rotating wheel acts as a thermal storage mass where the heat from one airstream is stored within the wheel until it is released to the other airstream later [12]. The schematic of a heat wheel is shown in Fig. 3.

Regarding enthalpy wheels, the metallic matrix is coated with hygroscopic materials such as silica gel and lithium chloride (LiCl) impregnated paper [52]. When the air with a higher moisture level passes through the wheel, the moisture is adsorbed by solid channel walls. Subsequently, the other air stream with lower moisture level passes through the wheel to regenerate the adsorbed moisture from the adsorbents [52].

Rotary wheels have been extensively used in buildings owing to their high effectiveness and high tolerance to frost [12]. The total effectiveness of rotary wheels can be over 80% with a counter-flow arrangement and compact design [18,55]. Compared with heat wheels, enthalpy wheels exhibit a higher total effectiveness because they recover both sensible and latent heat. However, their effectiveness is more dependent on operating conditions [56].

Jedlikowski et al. [57] explored the heat and mass transfer inside enthalpy wheels under high rotary speeds and sub-zero outdoor air temperature. They concluded that increasing the rotary speed facilitates more efficient heat and mass transfer, and less frost. However, the high rotary speeds may increase the cross-contamination. Tu et al. [58] indicated that the maximum heat and mass transfer efficiency of enthalpy wheels is affected by wheel thickness, specific surface area, air velocity, and heat and mass transfer coefficients.

Rotary wheels can be coupled with heat pumps for building applications. Ge et al. [59] developed an air source heat pump system (ASHP) combined with an enthalpy wheel, referred to as the 'hybrid ASHP system', for residential buildings in China with hot summer and cold winter climates. Under the heating mode operation, 14% of the energy can be saved using this hybrid ASHP system, as compared with the conventional ASHP system. Wallin and Claesson [60] retrofitted an AHU including a heat wheel, with an exhaust air heat pump in Sweden, as shown in Fig. 4. The heat pump extracted heat from the exhaust air and raised the temperature to a certain level, enabling the use of the

Table 1
Guidelines, standards, and regulations of design parameters for residential buildings in different counties in cold climate regions.

Required design parameters		Denmark [27,28]	Sweden [29]	Norway [30–32]	Finland [33–36]	Russia [37,38]	USA [25,39–42]	Canada [43–46]	China [47–49]
Air flow rate	ACH in the whole building	0.3 for residential buildings *	0.35 *	0.6 *	0.35 for new buildings *	N/A	0.35 for low-rise residential buildings	0.3 for low-rise residential buildings *	0.45–0.7 depending on the average living area per person *
	Enclosed Kitchen (exhaust)	20 l/s *	10 l/s *	10 l/s *	8 l/s *	16.7 l/s *	Demand-Controlled local ventilation: 50 l/s Continues local ventilation: 5 ach based on kitchen volume	5 l/s *	3 ACH *
	Bathroom (exhaust)	15 l/s for bathrooms, and toilets with bath/shower and sculleries *	10 l/s *	15 l/s *	10 l/s *	6.9 l/s *	Demand-Controlled local ventilation: 25 l/s Continues local ventilation: 10 l/s	10 l/s *	1 ACH *
	Bedroom and living room (supply)	0.3 l/s per m^2 of heated floor area *	0.35 l/s per m^2 of floor area *	26 m^3 fresh air per hour per planned bed *	6 l/s per person *	0.8 l/s per m^2 or 8.3 l/s per person *	Based on floor area and number of bedrooms	10 l/s *	N/A
	Toilet (exhaust)	10 l/s for toilets without bath/shower and sculleries *	10 l/s *	10 l/s *	10 l/s *	6.9 l/s *	10 l/s	10 l/s *	3 ACH *
Air tightness (tested at 50 Pa)		1.0 l/(s $\cdot m^2$) for new buildings No requirement for the refurbishment or renovation of existing buildings *	No particular requirement	0.6 l/(s $\cdot m^2$)	No particular requirement, but 1.0 l/(s $\cdot m^2$) is recommended	2-4 ACH for buildings with natural ventilation 2 ACH for buildings with mechanical ventilation	5 l/(s $\cdot m^2$) (at pressure difference of 0.3 Inches of Water Gauge) *	1.5 ACH	N/A
Indoor air condition	Temperature	No more than 100 h above 26 °C No more than 25 h above 27 °C	Minimum 18 °C for average dwellings *	19–26 °C easy work 16–26 °C medium work 10–26 °C heavy work	20–25 °C during the heating season *	Living room: 20–22 °C (winter) Kitchen: 19–21 °C (winter) Bathroom: 24–26 °C (winter)	20–24 °C during the winter	22 °C in all living spaces in winter	18–24 °C during the winter *
	Humidity	N/A	Maximum 3 g/ m^3 absolute humidity between indoor and outdoor environments *	Maximum 40% relative humidity during the winter	In accordance with space function to avoid moisture damage *	Living room: 30–45% (winter), no more than 60%	Maximum 65% relative humidity	35% in winter	30–60% relative humidity *
	Air velocity	Maximum 0.15 m/s	Maximum 0.15 m/s during the heating season *	N/A	Maximum 0.2 m/s in winter *	Maximum 0.15 m/s in winter	Maximum 0.15 m/s	N/A	Maximum 0.2 m/s *
	CO_2	1000 ppm	1000 ppm *	1800 mg/m^3 (1000 ppm)	700 ppm (best) 900 ppm (good) 1200 ppm (satisfactory)	400 cm^3/m^3 or less (high standard) 400–600 cm^3/m^3 (average)	1030 ppm	3500 ppm (acceptable long-term exposure)	$\leq$ 0.1% of 2000 ppm (day)
	HCHO (formaldehyde)	0.1 mg/m^3	0.08 ppm (0.1 mg/m^3) *	0.1 mg/m^3	0.03 mg/m^3 (best) 0.05 mg/m^3 (good) 0.1 mg/m^3 (satisfactory)	N/A	0.1 mg/m^3 (0.5 h)	100 ppb (acceptable short-term exposure) 40 ppb (acceptable long-term exposure)	0.1 mg/m^3 (1 h)
		N/A	N/A	Not set		N/A			0.6 mg/m^3 (8 h)

(*continued on next page*)

Table 1 (*continued*)

Required design parameters		Denmark [27,28]	Sweden [29]	Norway [30–32]	Finland [33–36]	Russia [37,38]	USA [25,39–42]	Canada [43–46]	China [47–49]
Total volatile organic compounds					200 μ g/m^3 (best) 300 μ g/m^3 (good) 600 μ g/m^3 (satisfactory)		Must be determined for each individual	200 μ g/m^3 (comfort level) 500 μ g/m^3 (building standard)	
Radon		100–300 Bq/m^3 (existing buildings) 100 Bq/m^3 (new buildings)	200 Bq/m^3 *	100 Bq/m^3 (recommended) 200 Bq/m^3 (maximum)	200 Bq/m^3	Depending on the type of ventilation	4 pCi/L (148 Bq/m^3)	200 Bq/m^3	400 Bq/m^3
Heat recovery efficiency	Minimum sensible heat recovery efficiency	80% *	70% if heated floor area is between 60 and 100 m^2 *	80% *	50% *	N/A	N/A	N/A	60% for public buildings only *
	Minimum latent heat recovery efficiency	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A
	Minimum total heat recovery efficiency	N/A	N/A	N/A	30% *	N/A	50% when the supply air ≥ 5000 cfm (8495 m^3/h), and the minimum outdoor air ≥ 70% of the supply air	55% *	60% for public buildings only *

* Building regulations or building codes.

recovered heat to reheat the return water from the ventilation heating coil. Their results showed that the heat recovery efficiency could be improved by 24%.

In cold and dry regions, the application of an enthalpy wheel may be beneficial for indoor humidity. Yang et al. [61] investigated the performance of silica gel desiccant wheel aiming to increase the indoor humidity level during the winter in China. They noted that the imposed system could satisfy the requirement of winter humidification in certain specific regions of China with ambient temperature ranging from −10 to 10 °*C*. Tu et al. [62,63] evaluated the humidification method for residential buildings in cold and dry climates of China using three-stage desiccant wheels. They concluded that the key to enhancing humidification performance involves increasing the desiccant wheel facial area ratio between dehumidification air and humidification air (A_r) as well as the ratio between dehumidification air and humidification air (F_r), while the rotation speed is less important.

The application of rotary wheels may be limited by the moving parts that require maintenance [64] and the cross-contamination risk. There are two types of cross-contamination: carryover and pressure leakages. The carryover leakage is caused when a small fraction of exhaust or supply airflow is trapped in void spaces or porous coating material and is subsequently carried over to the other airflow with the rotation of the wheel [65]. The pressure leakage occurs between gaps because of the pressure difference, which can be further divided into radial and angular leakages depending on the direction [66]. According to the experimental analysis by Han [66], the carryover leakage is proportional to the rotary speed, whereas the pressure leakage increases with airflow rate, and the heat recovery efficiency can be significantly enhanced by reducing the leakage. The impact of the pressure leakage can be prevented through a proper fan arrangement [67]. Further, carryover leakage can be reduced via a purge section, which purges the rotary wheel with outdoor air prior to its rotation into the supply air side [67]. Ruan et al. [68] developed a one-dimensional transient model to analyse the performance of an enthalpy wheel with a purge section. The optimum wheel depth and rotation speed were determined to achieve the maximum performance with the purge section. However, the long-time reliability of rotary wheels is questionable as the ability of the material to adsorb moisture from exhaust air deteriorates with time [52].

2.2. *Flat-plate heat exchangers*

Flat-plate heat exchangers have no moving parts, which guarantee long-time high reliability. The static flat-plate heat exchanger eases the sealing and reduces cross-contaminations [52]. Flat-plate sensible-only heat exchangers have been constructed using many thin plates stacked together with small spacings between them as air channels. Plates can be either smooth or corrugated, such as plate-fin channels [18,52]. Owing to the use of metals such as aluminium and steel alloys as plates in flat-plate sensible-only heat exchangers, only sensible heat can be recovered. The structure of a flat-plate heat exchanger with a cross-flow configuration is shown in Fig. 5 [69].

If water vapour-permeable plates replace conventional plates, the flat-plate heat exchanger can recover both sensible and latent heat from exhaust air. This type of heat exchanger is referred to as a flat-plate enthalpy exchanger. Previously, paper was used to transfer heat and moisture owing to its low cost; however, its application in cold climates is limited because the paper may be damaged by the condensation or frost in the exchanger [70]. Mohammad [70] tested a heat exchanger using 45 and 60 gsm Kraft paper, and their moisture efficiency are approximately 30% and 40%, respectively, at air velocity of 1.5 m/s. More studies have further confirmed the moisture efficiency of using paper in the flat-plate exchangers being lower than 40% [52,71]. Certain membranes have latent efficiency above 65%; however, the higher price than aluminium is a drawback [72].

In cold regions, air-to-air flat-plate enthalpy exchangers are used to recover heat and moisture from the warm and humid extract air. Yaïci

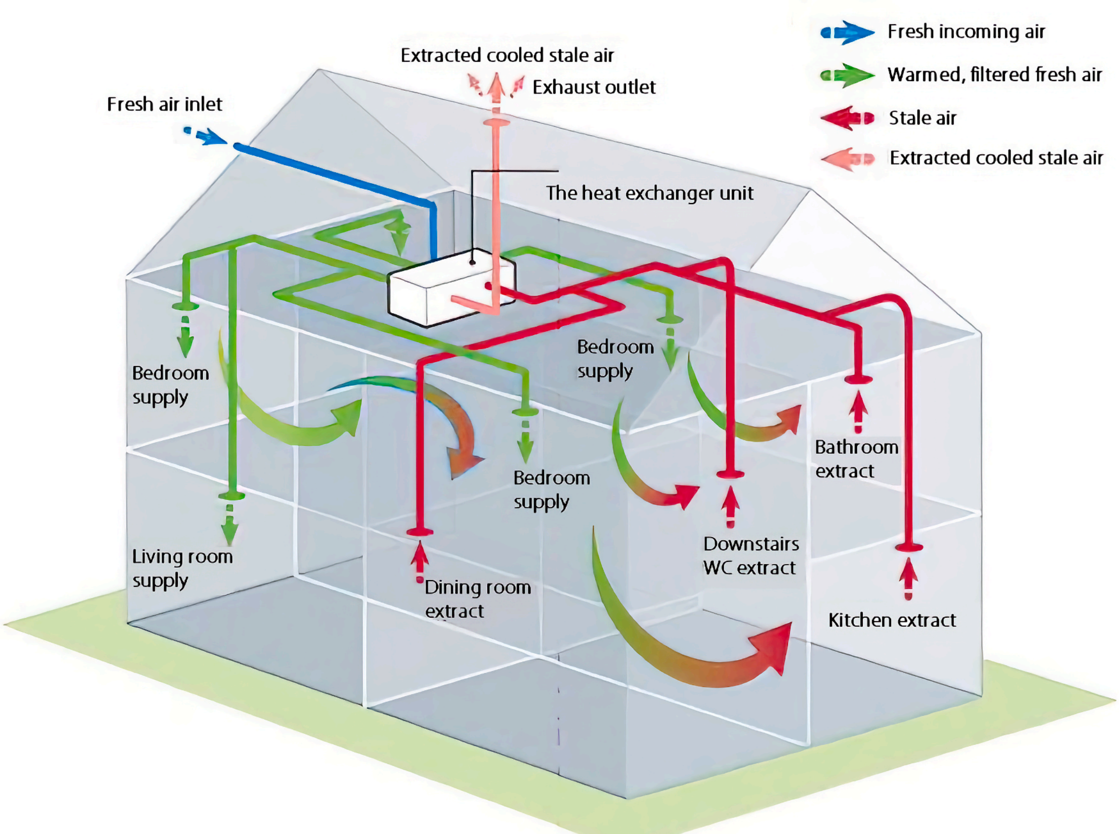


Fig. 2. Schematic of a typical heat recovery ventilation system (HRV) in a residential building [53].

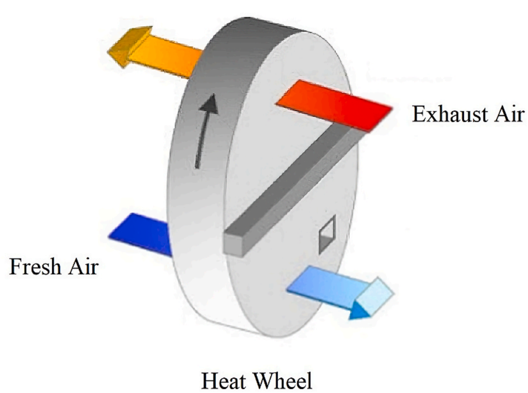


Fig. 3. Schematic of a heat wheel [54].

et al. [73] numerically analysed membrane-based flat-plate energy recovery ventilators (ERVs); their results showed that the effectiveness of ERVs decreases with an increase in the supply-exhaust air velocity and that the latent effectiveness was noticeably enhanced with an increase in the diffusivity of the water in the membranes. Zhang [74] studied cross-corrugated channels and compared them with the normal flat-plate channels in enthalpy exchangers. They found that the mixing effect caused by cross-corrugated shaped channels intensified the convective mass transfer coefficient on the membrane surfaces. Wang et al. [75] examined the energy saving potentials for a variable air volume (VAV) system implemented with membrane-based flat-plate ERVs in various climates. A maximum HVAC energy saving ranging from 39% to 49% was predicted under winter conditions.

Although cross-flow configuration is the most popular type for the simplicity of construction, its sensible effectiveness is normally lower than 75% unless two heat exchangers are used in series [76]. Higher effectiveness can be achieved using a counter-flow arrangement; however, due to the difficulty of sealing and connecting with the ducts, flow contamination may occur [21]. Nasif et al. [77] designed and constructed a Z-flow configuration enthalpy exchanger, which provided a counter-flow pattern over most of heat and mass transfer areas made of 98 μm thick Kraft paper. Consequently, heat and mass transfer can be very close to pure counter-flow pattern. Sensible and latent effectiveness of such a system were predicted to be 75% and 60% respectively. Zhang [78] developed a quasi-counter-flow membrane-based enthalpy exchanger, and concluded that sensible and latent effectiveness of the proposed exchanger is between pure counter-flow and pure cross-flow enthalpy exchangers. Compared with a pure cross-flow enthalpy exchanger, a quasi-counter-flow enthalpy exchanger may improve the sensible and latent effectiveness by 5%. The performance improved with the increasing counter-flow areas. Liu et al. [79,80] constructed a similar quasi-counter-flow membrane enthalpy exchanger with corrugated aluminium mesh spacers, focusing on its application in cold climates. Authors found that the proposed membrane enthalpy exchanger produced relatively high sensible and latent effectiveness in cold climates due to its counter-flow nature, and they were very independent of the outdoor air temperature when there is no condensation or frosting occurring. Fig. 6 shows a comparison between this novel combined counter- and cross-flow configuration for flat-plate heat exchangers from different literature.

Flat-plate heat exchangers introduced above are air-to-air heat exchangers, where heat exchange occurs between cold and hot air directly. The concept of run-around heat exchangers was first introduced by Kays and London [81], where it was referred to as a liquid-to-air exchangers system. The system comprised two normal liquid-to-air heat exchangers, which were coupled together via a circulation of a heat transfer medium, for example, water or liquid metal. Liquid heat transfer medium gained

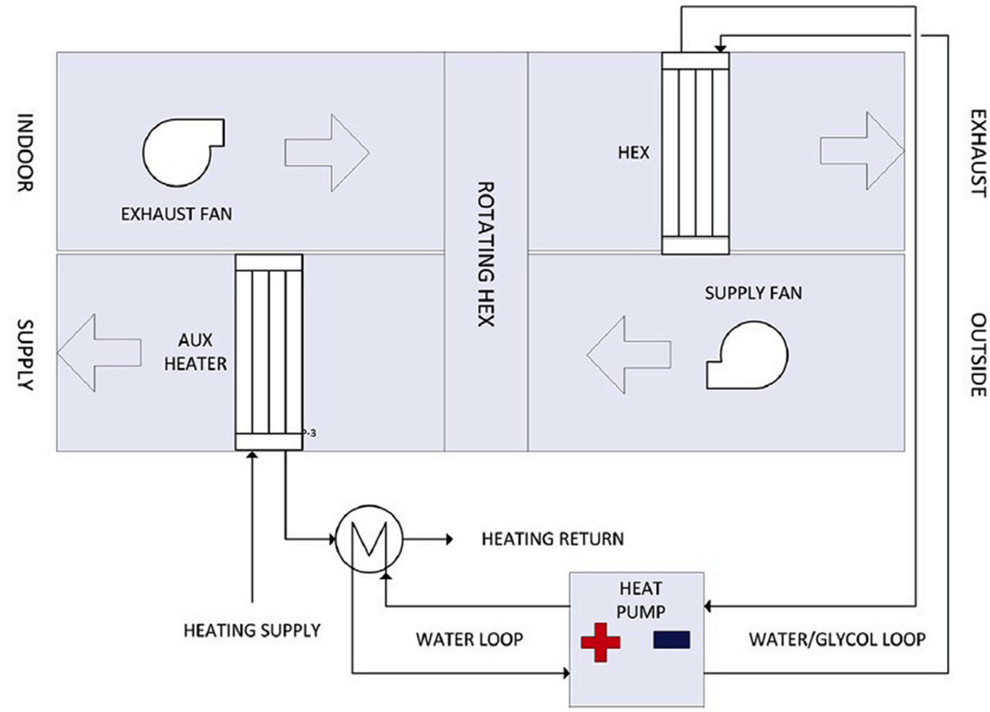


Fig. 4. Air-handling unit (AHU) with combined heat wheel and exhaust air heat recovery [60].

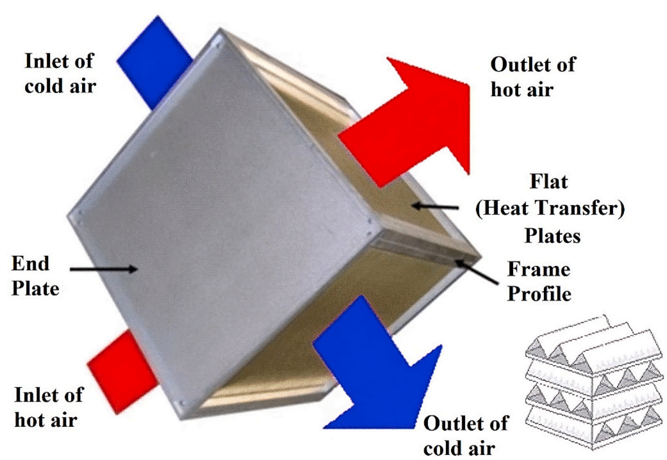


Fig. 5. Structure of a cross-flow flat-plate heat exchanger [69].

the heat from hot airstream and released it to cold airstream with the circulation. The authors reported several advantages of such a system, such as a more compact arrangement for a simpler gas-ducting situation. However, this system required a larger heat transfer area and an additional complicated liquid coupling circuit; moreover, it lacked a universally satisfactory heat transfer medium. Thus, only sensible heat transfer was considered in this early application. A similar system referred to as 'coil energy recovery loops' was introduced in Refs. [76, 82], where sensible heat could be transferred from warm to cold airstream through an intermediate heat transfer fluid or freeze-protected solution linking two heat exchangers. It was stated that this system was suitable for applications where supply air duct is located remotely from exhaust air ducts, which would afford high flexibility in building renovations. A basic arrangement of the run-around heat recovery system is depicted in Fig. 7.

Davidsson et al. [83] designed a system for natural/hybrid ventilation system intended for cold climate using shell and tube style liquid-to-air heat exchangers. Copper solar collector absorbers were used as pipes with water flowing inside. The explored ventilation system achieved a sensible effectiveness of 80% with a pressure loss of 1 Pa and provided high heat recovery rates for buildings in cold climates provided defrosting strategies were used to prevent water from freezing.

Aqueous solution of organic solvents such as ethylene glycol and diethylene glycol, and inorganic solvents such as lithium chloride, calcium chloride, and lithium bromide can be used to achieve latent heat recovery [84]. Liquid desiccant and air can be in direct contact [76], or through the use of membrane-based liquid-to-air heat exchangers. A two-dimensional steady state numerical model was developed to study heat and mass transfer in a membrane-based run-around heat exchanger for building heat recovery application using aqueous ethylene glycol [85] and aqueous lithium bromide [86] as the coupling fluid, respectively. The effectiveness of each cross-flow flat-plate heat exchanger and the run-around system were found to be related to the number of heat transfer unit (NTU), and thermal capacity ratio of heat exchangers. In addition, the overall effectiveness of the system was also found to be dependent on coupling fluid and airstreams flow rate, geometric parameters of heat exchangers, and inlet operating conditions.

Wallin et al. [87] built a run-around coil heat recovery system with two cross-flow heat exchangers connected by mixed water and ethylene glycol for cold climate owing to its anti-freeze nature. Various factors affecting the system performance were measured and identified, and it was found that brine flow rate, ethylene glycol concentration, and system pressure influenced the heat recovery rate.

2.3. Heat pipes

Heat pipe uses phase change materials to transfer heat from warm to cold airstreams. The warm airstream flows over the evaporator end of the heat pipe to vaporize the working fluid, which is then driven to the condenser end under the effect of the difference in vapour pressure. Subsequently, the vaporized working fluid releases the latent heat at the condenser end on encountering the cold airstream, and the condensed fluid is thereafter transferred back to the evaporator side via the wick effect. Thus, the working fluid inside heat pipes evaporates and

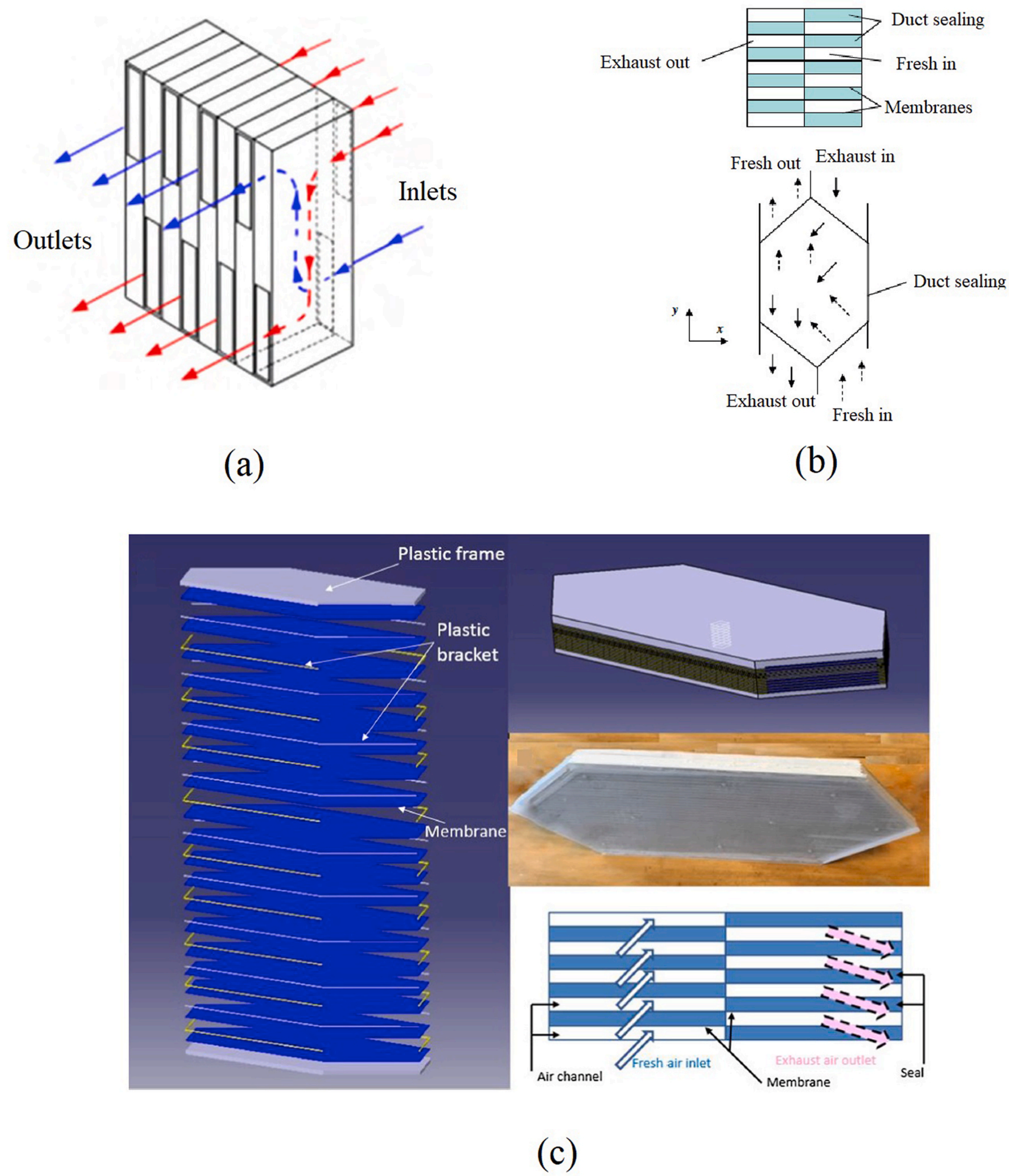


Fig. 6. Different structures of combined counter- and cross-flow flat-plate heat exchangers (a) Z-shape enthalpy exchanger [77] (b) quasi-counter-flow membrane-based enthalpy exchanger [78] (c) quasi-counter-flow membrane enthalpy exchanger with corrugated aluminium mesh spacers [79,80].

condenses in a closed cycle as long as there exists a temperature difference between the two airstreams to drive this process [76].

Heat pipes avoid the contamination between supply and exhaust airstreams. They are relatively reliable owing to the absence of moving parts [21] and suitable for natural ventilation systems for low pressure drops [88]. No additional pump or other sources of power are required to drive the operation. Further, the effective heat transfer coefficient was stated to be several orders of magnitudes higher than that of good solid conductors [89]. Heat pipes can be embedded in a metal cooling plate, or be assembled with a fin stack for fluid heat transfer [90]. For heat recovery application, copper or aluminium heat pipes were assembled with copper or aluminium fins to avoid problems with various thermal expansions of metal materials [76]. In addition, heat transfer performance is affected by various parameters such as wick design, tube diameter, type of working fluid, and heat pipe orientation relative to horizontal. The heat transfer effectiveness of heat exchanger with heat pipes installed is related to air face velocity and rows of heat pipes. Thus, the selection of working fluid is of critical importance for the long-term operation.

Calautit et al. [91] integrated heat pipes into a multi-directional wind tower, whose application was conventionally limited to the summer season for areas in cold climates. Their design is shown in Fig. 8, wherein, as evident, warm and contaminated indoor air passes through a series of heat pipes that absorb the heat and transfer it to a perpendicular heat sink installed at the top of the wind tower. Thereafter, cold and fresh outdoor air was sucked in the wind tower under the pressure difference, and it was heated in a heat sink. Therefore, the energy required for heating the fresh air to the required temperature in winter was considerably reduced as the fresh air temperature was raised without using additional power. Further, CFD simulations and wind tunnel tests

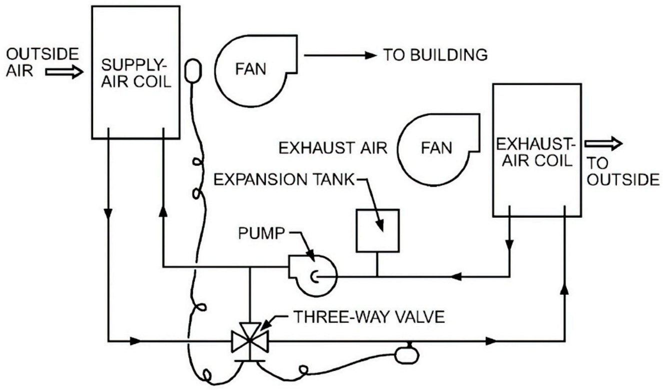


Fig. 7. A basic arrangement of run-around heat recovery system [76,82].

showed that by using heat pipes in the wind tower, the proposed system was capable of raising the supply air temperature by 4.5 K during winter.

Diao et al. [92] conducted an experimental test on a small flat heat pipe heat recovery device, which used a flat micro heat pipe array (FMHPA) with welded, serrated, and staggered fins on the surface as the heat exchanger to recover heat for residential buildings. The effects of the outdoor and indoor temperature difference, air flow rate, and the number of FMHPA rows on the heat transfer rate were studied. The authors found that the heat transfer efficiency and system COP increased gradually with the temperature difference between outdoor and indoor air, under winter conditions. Further, the maximum heat transfer efficiencies when using two, three, and four rows of FMHPAs were found to be 57.9%, 70.6%, and 76.2%, respectively. Zhou et al. [93] proposed a pump-driven loop heat pipe (PLHP) system to recover heat from the exhaust air of residential buildings. The core of this heat exchanger comprised bundles of cooper pipes and sheets of aluminium wave fins, serving as a tube-fin style heat exchanger. Further, a self-priming magnetic pump was used to drive the system for various reasons: the conventional heat pipe needed to be started under a certain temperature difference, limitations during use in both summer and winter seasons as the evaporator must be installed below the condenser, and a weak driving force [94]. The proposed system showed significant potential for recovering heat from exhaust air; the heat transfer capacity, temperature effectiveness, and COP were 6.63 kW, 33.8%, and 14.2 under the winter conditions, respectively.

2.4. Other types of heat recovery technologies for residential buildings in cold climates

Apart from the major heat recovery technologies introduced in previous sections, other types of heat recovery technologies for residential buildings in cold climates have been reported in published literature.

Zhang et al. [95] developed a heat recovery envelope, which was referred to an exhaust air insulation wall (EAIW), composed of air permeable porous materials. Compared with conventional exhaust air recovery design, which normally requires an air-to-air heat exchanger, EAIW provided an alternative for direct and local use of exhaust air in each room. By allowing exhaust air to exfiltrate through the porous layer the velocity vector of exhaust air was parallel to the temperature gradient in the porous layer, implying the best convective heat transfer. Consequently, the conduction heat transfer towards indoor space was notably decreased because of the enhanced convective heat transfer between the building envelope and exhaust air. Their design focused initially on heat recovery during the hot summer season, however, authors also reported that by reversing the airflow within porous layer from exfiltration to infiltration using a reversible fan, EAIW could also be used in the cold winter season. In such a case, the infiltration flow in the porous layer could trap and use the outward conductive heat loss. Thus, the fresh infiltration air was preheated.

Twin-tower enthalpy recovery loop is a method for recovering both sensible and latent heat from exhaust air, which includes supply and exhaust towers where airstreams and connecting liquid desiccants are in direct contact [76]. Connecting fluid is usually a salt solution that is an effective anti-freeze medium. Thus, outdoor air temperature as low as $-40\,^{\circ}C$ can be tolerated without freezing or frosting problems. However, their applications in residential buildings are lacking in previous literature.

Fixed-bed regenerators (FBRs) were used in commercial and large residential buildings to recover sensible heat from exhaust air and latent heat when manufactured with desiccant treating [76]. These devices, as shown in Fig. 9, are normally staggered; this implies that, when half of the ventilation system is exhausted, the other half introduces outdoor air into the space. Following the recovery period (period 1), the system alters the direction to recharge the device subjected to the outdoor airstream. The other half of the system transfers the energy to outdoor air until the subsequent recovery period. Owing to this repeated exhaust regeneration, FBRs can operate in extreme cold weather conditions with outdoor air temperatures below $-40\,^{\circ}C$, without requiring an additional defrosting strategy. However, the outlet air temperature and humidity from FBRs are not in a steady state, which constitutes a challenge for testing facilities.

VENTIREG, which was proposed and studied by Aristov et al. [97], refers to a design comprised an adsorbent bed (Ad bed) and a heating

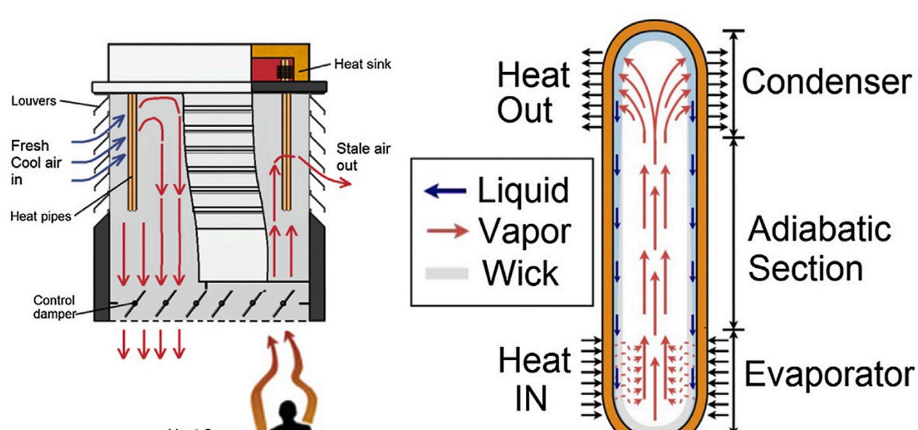


Fig. 8. Multi-directional wind tower design with integrated heat pipes [91].

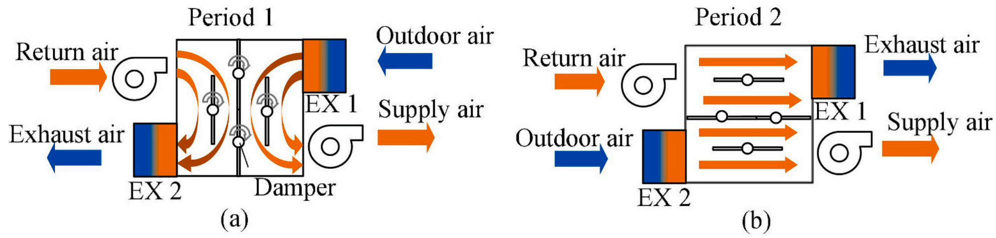


Fig. 9. Schematic of operation of an FBR with two stationary exchangers [96].

storing bed (HS bed), as shown in Fig. 10 (a). A complete recovery cycle consisted of outflow and inflow modes. In outflow mode, warm and humid indoor air was extracted through dry Ad bed, dried and warm air entered cold HS bed and heated it up, and then dry and cold air was exhausted. In inflow mode, the airflow direction switched, cold and dry outdoor air was extracted through a warm HS bed. Thus, the air was heated up. Subsequently, the air entered the humid Ad bed to regain the moisture being stored in outflow mode, and warm and humid air was supplied to occupied space. Their design was tested under Western Siberian condition, and results showed 90% of total energy, and 70%–90% of moisture being recovered from indoor extract air. Further, the formation of the frost at unit exit was also prevented as indoor air was dried in the Ad bed prior to being cooled in the HS bed. However, Shkatulov et al. [98] pointed out that VENTIREG system has two main drawbacks: heat released during the moisture adsorption in Ad bed increased the temperature of Ad bed, which reduced the ability of moisture adsorbing, implying the need for a larger adsorbent load. In addition, a considerable amount of electricity was required for extracting air through Ad and HS beds, which reduced the system's energy efficiency. They designed a new VentireC system to overcome these problems, which substituted the fixed Ad bed with thermally coupled desiccant-coated heat exchangers (DCHEs), as shown in Fig. 10 (b). Heat released during moisture adsorption in DCHE 1 was transferred to DCHE 2 through heat transfer fluid. Apart from improved dehumidification and humidification ability of DCHE, its hydraulic resistance was proved to be approximately ten times smaller than that of fixed Ad bed with same adsorbent mass, which resulted in notable saving in electricity consumption.

Comparison between various heat recovery technologies for residential building applications introduced in this section is summarised in Table 2 below.

3. Frosting problems and frosting control strategies for application of heat recovery in cold climate regions

In cold climate regions, the frosting problem is very common in heat and energy exchangers, which decreases their performance significantly. The frost inside air channels can partially or fully block channels, increasing the pressure drop through exchangers, or decreasing air flow rates [115]. Increased energy consumption by fans is required to overcome the pressure drop across the exchanger [115]. In addition, the formation of frost deteriorates the exchanger effectiveness as it results in a decrease in the heat transfer rate between two airflows [116]. Moreover, draught negatively affecting thermal comfort occurs in ventilated spaces owing to the low supply air temperature [117]. Consequently, the exchanger can be physically damaged in the long term, which would increase the maintenance cost. In this section, frosting problems in different types of heat recovery exchangers and various defrosting strategies for different applications in the literature are reviewed.

3.1. *Frosting in exchangers*

In the following section, first, the mechanisms of frost formation in sensible heat only exchanger and total heat exchanger are illustrated and compared. Different parameters affecting the frost formation, and its impact on heat and mass transfer inside exchangers are also reviewed.

3.1.1. *Mechanism of frosting formation in exchangers*

Frosting occurs in a heat exchanger if the temperature of a heat transfer surface is lower than 0 °*C*, and also lower than the dew point temperature of the airflow in contact with the surface [115]. The formation of the frost layer is a transient process. According to Iragorry, the process can be divided into four steps [118]: Dropwise Condensation (Nucleation), Tip-Growth Period (Frost Formation), Densification, and Bulk-Growth (Frost Layer). These steps are classified by different frost structures caused by the variations of frost layer temperature in contact with the air during the transient process. Chen et al. [119] investigated a typical condensation-freezing frost formation, and the growth of frost on initially clean cold surfaces was divided into two continuous stages.

The first early stage referred to the crystal growth period, which is relatively short (approximately 15–100 s) and is characterised by small water droplet (average diameters of 50 mm–90 mm) condensation and freezing covering approximately 60% of surface area. The latter stage is a fully developed frost growth period. During this period, the fully developed frost layer is characterized by a porous media made of a solid ice matrix and pores filled with moist air. The properties of the frost layer are explained in detail in Section 3.1.2.

In cold climate regions, frosting typically occurs in the exhaust air

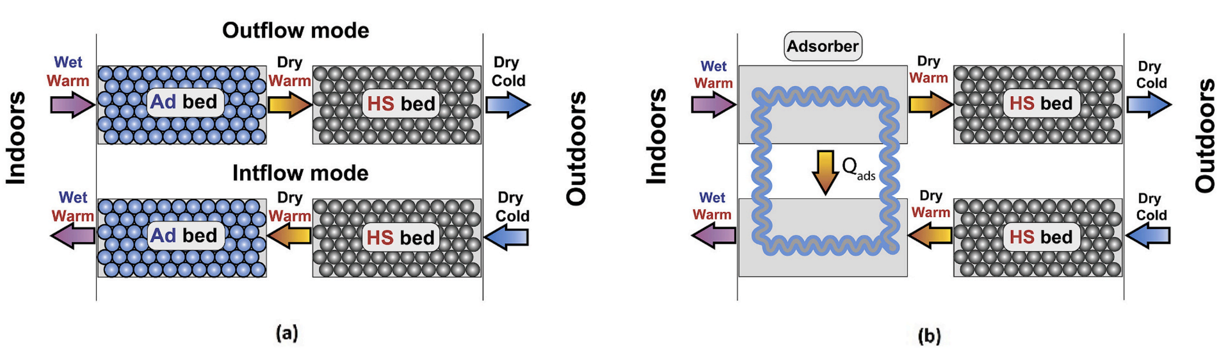


Fig. 10. (a) schematic of VENTIREG (b) schematic of VentireC [98].

Table 2
Comparison between various heat recovery technologies for residential building applications.

Reviewed aspects	Different types of heat recovery technologies used for residential building applications									
	Rotary wheel		Flat-plate heat exchanger			Heat pipe	Other types			
	Heat wheel	Enthalpy wheel	Flat-plate sensible-only heat exchanger	Flat-plate enthalpy exchanger	Run-around heat recovery system		EAIW	Fixed-bed regenerator	VENTIREG	VentireC
References	[21,52,54,76, 99–101]	[21,52,59–61,64, 65,76,102–107]	[18,21,52,76, 82]	[21,52,73, 76–80,108–110]	[76,81–83,87,111]	[21,76,89,91–93, 112,113]	[95,114]	[76,96]	[97,98]	[98]
Typical flow arrangement	Parallel flow Counter flow	Parallel flow Counter flow	Counter flow Cross flow	Counter flow Cross flow Combined counter-cross-flow	Counter flow Cross flow Combined counter-cross-flow	Counter flow Parallel flow	Counter flow	Parallel flow	Exhaust and supply air are in reversed direction	Exhaust and supply air are in reversed direction
Moving parts	Yes	Yes	No	No	Coupling working fluid	Multi-phase fluid inside heat pipe	No	Damper	No	Coupling working fluid
Face velocity (m/s)	2–5	2.5–5	1–5	1–3	1.5–3	2–4	N/A	1–2.5	N/A	N/A
Pressure drop (pa)	100–300	100–300	100–1000	100–500	150–500	150–500	N/A	50–300	N/A	N/A
Operating temperature range ($^{\circ}C$)	−55-800	−55-800	−60-800	−40-60	−45-500	−40-93	N/A	−55-60	−28-20	As low as −30
Types of heat being recovered	Mainly sensible	Sensible and latent	Sensible only	Sensible and latent	Sensible, or sensible and latent	Sensible only	Sensible only	Sensible, or sensible and latent with desiccant coating	Sensible and latent	Sensible and latent
Typical sensible effectiveness (%)	65–80	65–80	60-80 (cross flow) 70-90 (counter flow)	60-80 (cross flow) 70-90 (counter flow) 80-85 (combined counter-cross-flow)	65-70 (sensible-only run-around system) 60-80 (membrane-based run-around system)	40–60	N/A	80–90	86–96	N/A
Typical latent effectiveness (%)	–	65–80	–	Minimum 40 for paper Minimum 65 for membranes 80-85 (combined counter-cross-flow)	50–65	–	–	60–80	70–90	Improved compared with VENTIREG
Typical total effectiveness (%)	–	50–80	–	35–70	–	–	–	50–80	Up to 95%	N/A
Advantages	Low pressure drop Compact design	Low pressure drop Compact design Moisture recovery	No moving part Low pressure drop No cross contamination	No moving part Low pressure drop No cross contamination Moisture recovery	Exhaust air ducts can be separated from supply air ducts No cross contamination	No moving part Low pressure drop No cross contamination	Direct and local use of exhaust air in each room No heat exchanger needed	Few moving part Moisture recovery possible with desiccant coating Adaption to extreme cold	No moving part Air temperature and humidity independent control Low cost Exceptional adaption to extreme cold climate	Air temperature and humidity independent control Exceptional adaption to extreme cold climate Improved moisture regeneration efficiency

(*continued on next page*)

Table 2 (*continued*)

Reviewed aspects	Different types of heat recovery technologies used for residential building applications									
	Rotary wheel		Flat-plate heat exchanger			Heat pipe	Other types			
	Heat wheel	Enthalpy wheel	Flat-plate sensible-only heat exchanger	Flat-plate enthalpy exchanger	Run-around heat recovery system		EAIW	Fixed-bed regenerator	VENTIREG	VentireC
								climate without defrost strategies		Less hydraulic resistance and electricity consumption
Disadvantages	Moving part Air short circuiting Cross contamination Sensible heat recovery only	Moving part Air short circuiting Cross contamination	Large size Sensible heat recovery only	Large size Long-term maintenance	Additional coupling fluid design Long-term maintenance	Exhaust and supply air ducts must be closely installed Working fluid must match local climate	Lack of studies for cold climate applications	Air outlet conditions are not steady state	adsorption heat reduces moisture regeneration efficiency Significant consumption of electricity	Additional coupling fluid design
Integration into building systems	Mechanical ventilation Passive ventilation Integrated with PV/T	Mechanical ventilation Integrated with Air Source Heat Pump (ASHP)	Mechanical ventilation	Mechanical ventilation Integrated with a roof structure	Mechanical ventilation Passive ventilation Integrated with heat pump	Mechanical ventilation Wind tower Natural ventilation Domestic hot water	Natural ventilation	Mechanical ventilation	Mechanical ventilation	Mechanical ventilation
Applications in cold climates	Need defrost strategy	Better frost limit Need defrost strategy	Need defrost strategy	Better frost limit Need defrost strategy	Good adaption into cold climates as anti-freeze natural of salt solutions	Need more studies	Need more studies	Adaption to extreme cold climate without defrost strategies	Exceptional adaption to extreme cold climate as independent humidity and temperature control	Exceptional adaption to extreme cold climate as independent humidity and temperature control

Table 3
Comparison between various frosting detection methods.

Method	Visual observation	Laser beam	Energy effectiveness	Pressure drop	Temperature difference
References	[20,129,132,134]	[129]	[20,129,132,134]	[20,129,132–134]	[132]
Principle	Visual inspection	Channel blockage by frost growth	Frost layer change overall heat transfer coefficient	Channel blockage by frost growth	Airflow maldistribution due to frost growth
Testing parameters	Photos	Laser signal	Ait temperature and humidity	Static pressure	Air temperature
Devices	Endoscope or Borescope Optical fibre	Laser light Optical fibre	Thermocouple Humidity sensor	Pressure transducer Pitot tube	Thermocouple
Types of heat exchangers to be detected	Rotary wheels and flat-plate heat exchangers	Rotary wheels and flat-plate heat exchangers	Rotary wheels and flat-plate heat exchangers	Rotary wheels and flat-plate heat exchangers	Flat-plate heat exchangers
Applications	Laboratory	Laboratory	Laboratory, practical application	Laboratory, practical application	Laboratory, practical application
Expected trend with frost growth	–	Decrease	Increase initially, then decrease	Increase	Increase
Advantages	Easy operating Efficient Accurate	N/A	Quantitative Inexpensive	Efficient Quantitative Inexpensive Good indicator of the amount of frost	Efficient Quantitative Less affected by operating parameters and heat exchanger type
Disadvantages	Limitation due to narrow channels Damage to electronics Expensive Not quantitative	Expensive Not quantitative	Inefficient Inaccurate	Expensive Short lifespan Inaccurate	Does not indicate the amount of frost

side of heat exchanger as the extract air from indoor spaces has a higher moisture content. The temperature of the plate surface is given in Eq. (2) [115].

$$T_p = \frac{T_s h_s + T_e h_e}{h_s + h_e} \tag{2}$$

where T_p is the temperature of the surface of the plate ($^\circ C$); T_s and T_e are temperatures of the supply airflow and exhaust airflow, respectively ($^\circ C$); and h_s and h_e are the convective heat transfer coefficients on the supply and exhaust air sides, respectively (W/m^2K). If h_s and h_e are not equal, T_p approaches the airflow with a higher convective heat transfer coefficient. Further, T_p increases if the water vapour condenses, as the convective heat transfer coefficient on the warm air side is enhanced as a result of the condensation heat transfer to the surface and the formation of a rough film of water on the plate surface. At the inlet of the cold airflow, the convective heat transfer coefficient is considerably increased owing to the entrance effect. Therefore, T_p decreases and is closer to the temperature of the cold airflow. These two effects exert opposite influences on T_p, as qualitatively presented in Fig. 11 [115].

The psychometric processes in supply and exhaust airflows in a sensible-only heat exchanger in cold climates is shown in Fig. 12 (a). The exhaust air is cooled along the air channels without any change to its humidity ratio. For an enthalpy type heat exchanger, both the temperature and humidity ratio of exhaust air are reduced during the heat exchange process, as depicted in Fig. 12 (b) [20]. Further, the dew point temperature of the exhaust air is reduced because of the additional mass transfer, implying that frosting occurs at much lower outdoor air temperatures in a total heat exchanger than in a sensible-only heat exchanger. Thus, by using an enthalpy type heat exchanger, such as an enthalpy wheel or membrane-based flat-plate enthalpy exchanger, the system adaption in cold climate regions can be significantly improved.

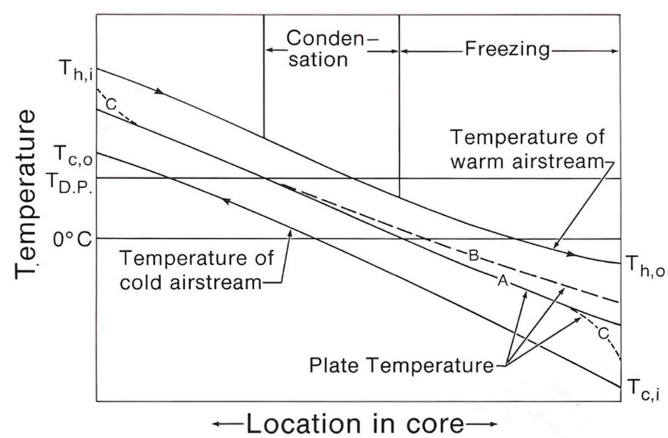


Fig. 11. Illustration of airflow and plate surface temperatures versus location in a counter-flow plate heat exchanger. Line A represents T_p when $h_s = h_e$; Line B represents the impact of condensation; Line C represents the impact of entrance effects. $T_{h,i}$, $T_{h,o}$, $T_{c,i}$, $T_{c,o}$, and $T_{D,P}$ represent warm airflow temperatures at inlet and outlet, cold airflow temperature at inlet and outlet, and dew point temperature of warm air, respectively [115].

3.1.2. *Impacts of frosting on heat exchangers*

The effects of frosting on heat exchangers are closely related to the frosting properties, such as frost growth rate, frost structure, frost density, thermal conductivity, and surface roughness [20]. As introduced in Section 3.1.1, a typical condensation-freezing frost formation comprises an early crystal growth period, followed by a fully developed period. The simulation of the first stage was considered to be unnecessary for the case of frosting over many hours [119]; thus, most studies have focused on the simulation of the second stage. O'Neal and Tree [120] presented a review of the available correlations for fully developed frost thickness, thermal conductivity, and heat transfer coefficient on the frost surface, considering simple geometries. Yang and Lee [121] proposed a numerical model that considered the air flow for predicting the frost properties and the heat and mass transfer within the frost layer on a cold plate. The proposed model was found to predict the heat transfer coefficient more accurately than the previous model, which used heat transfer coefficients obtained from non-frosting conditions. The authors then extended their study [122] and developed dimensionless correlations for predicting the frosting properties on a cold plate; air temperature, absolute humidity, velocity, cold plate temperature, and frosting time were considered. The correlations were expressed as functions of the Reynolds number, dimensionless time, absolute humidity, and dimensionless temperature, and were subsequently verified by experimental tests to be able to predict the average frost properties when the cooling plate temperature ranged from −35 °*C* to −15 °*C*. Lee et al. [123] developed a

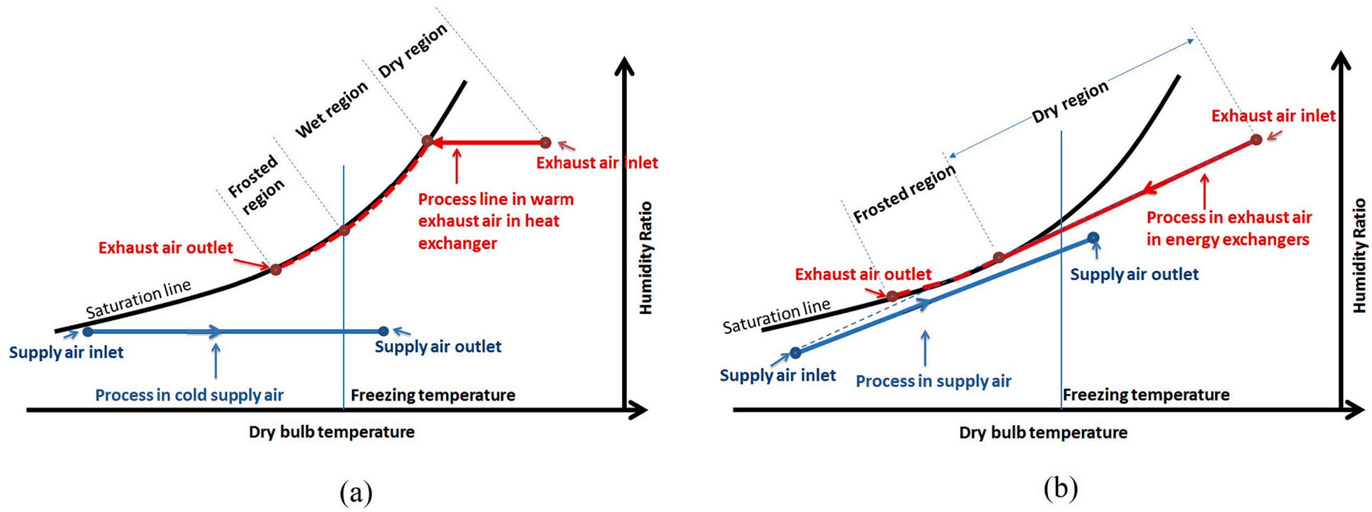


Fig. 12. Psychometric processes of supply air and exhaust air in a sensible-only heat exchanger (a) and total heat exchanger (b) [20].

mathematical model to predict frost formation, considering the airflow and the frost layer. Compared with previous models, their model estimated the frost thickness, density, and surface temperature more accurately. According to their results, the heat transfer rate decreased rapidly at the beginning of fully developed stages because of the increase in the frost thickness; however, its decrease rate gradually reduced over time. Furthermore, the density of the frost layer exhibited the highest value at the inlet, owing to the leading edge effect.

Kondepudi and O'Neal [124] presented a review of frosting on extended surfaces, and found that the fin efficiency was reduced by 20% owing to a 1–3 mm frost layer. The overall heat transfer coefficient increased at the initial stage of frosting, referred to as initial surge, which was attributed to increased surface area and surface roughness. However, the initial surge was soon offset by the increase in thermal resistance of the frost layer. The frost-to-air heat transfer coefficient was predicted higher under the same condition on a frosting layer than it on a non-frosting layer because of the surface roughness [120]. However, the surface roughness effect was effective only in the turbulent flow region, while in most exchangers, the airflows are laminar owing to narrow spacing and low air velocity. Yang et al. [125] developed a numerical model to investigate the frosting behaviour of a fin-and-tube heat exchanger. Although no surface roughness effect consideration was used in their model, the initial surge of heat transfer coefficient was still noticed. Huang et al. [126] built a numerical model to examine the effect of frost thickness on the heat transfer of fin-and-tube heat exchangers. The authors highlighted two main influences of the frost layer on the heat transfer: the growth of frost layer thickness narrows the flow channels, which increases the air flow velocity and air side convective heat transfer coefficient. Further, the thermal resistance of the frost layer is also increased. The total thermal resistance from air to fin was affected by these two parts. The initial surge of the heat transfer rate was attributed to the high thermal conductivity of the frost layer at the early stage, while surface roughness had a minor impact.

Gu and Li [127] explained the basic heat and mass transfer equations during desublimation frost formation. The authors provided differential heat and mass governing equations for the frost-air interface and pointed out that the total heat conducted to the cold plate comprised the heat convection from moist air to the frost layer and the desublimation heat released at the frost-air interface during phase change. Chen et al. developed numerical models to simulate the frost characteristics and heat transfer on flat-plate heat exchangers [128] and plate-fin heat exchangers [119]. The frost layer was simulated as a porous medium with coupled heat and mass diffusion, composed of a solid ice matrix and pores filled with moist air. Their results demonstrated that the blockage of air flow resulting from the increase in the pressure drop caused by frost formation across the heat exchanger by up to a factor of 8. Nevertheless, the heat transfer rate decreased by only 20%, implying that frost growth has a more dramatic influence on the pressure drop than the heat transfer rate. A numerical model of the frost growth for a rotary enthalpy wheel was developed [129] based on the previous model [119]; the authors concluded that the frost thickness increased the most in the middle of the wheel over time, and 50% of the blockage was caused in the first 20 min. Kragh et al. [117] conducted an experimental study on the impacts of frosting on mechanical ventilation systems for single-family houses in Denmark. One of the most direct influences of frost formation was the increased pressure drop across the exhaust airflow, which decreased the warm air flow rate. In addition, the sensible effectiveness of the counter-flow heat exchanger was also reduced significantly, from 80% to as low as 30%.

Fig. 13 shows the heat transfer mechanism in a plate-type heat exchanger with a frost layer, and its impact on heat exchangers, as summarised from the above discussion. h_w and h_c are the convective heat transfer coefficients (W/m^{2}K) for warm and cold air, respectively; λ_w and λ_c are the thermal conductivities (W/mK) for warm and cold air, respectively; δ_f and δ_p are the thicknesses (m) of the frost layer and plate, respectively; and $R_{w,conv}$, R_f, R_p, and $R_{c,conv}$, are the heat resistances (K/W) at the warm air side, frost layer, plate, and cold air side, respectively. Although several studies have attempted to experimentally or numerically investigate frost formation, the mechanism of the frost growth in heat exchangers remains unclear. Further, numerical modelling of the frost growth is yet limited to specific surface geometries and operating conditions; this can attribute to many reasons: coupled heat and mass transfer and complicated quasi-steady boundary conditions [127], difficulties in the development of physical models for the desublimation within frost layers [130], changes in frost properties along cold surfaces over time [20], and continuous variations in the frost-air interface temperature and alternating melting and sublimation within the frost layer as the result of the changes in the frost-air interface temperature [20].

3.2. Frosting detection methods

Frost formation is a complicated transient process and detecting the initiation of frost formation is of significant research interest. A reliable frosting detection method can aid in controlling the defrost process in terms of energy efficiency [131]. The easiest and most accurate method to detect frosting is through visual observations when the heat exchanger allows [20]. However, applying this method in practice

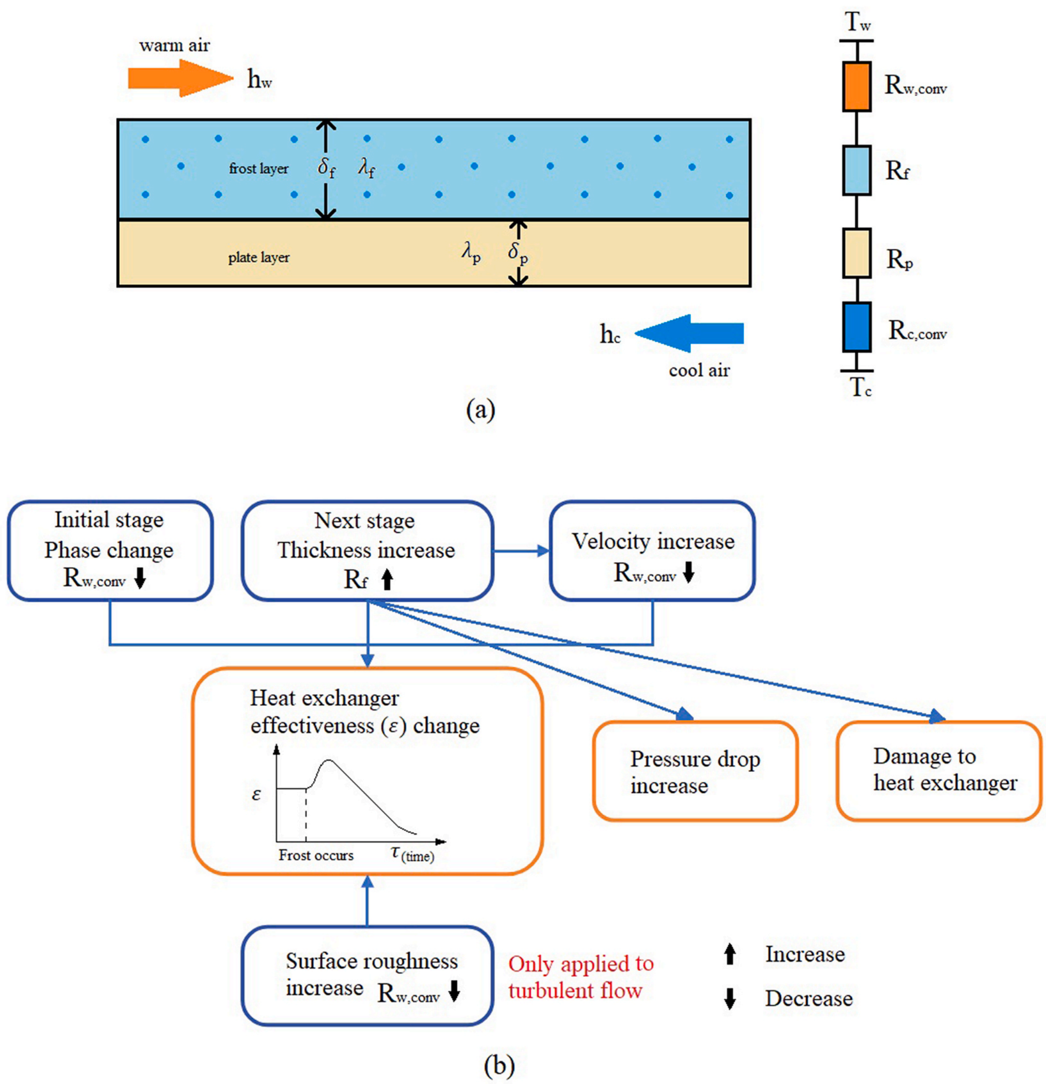


Fig. 13. Heat transfer mechanism explained in a plate-type heat exchanger with frosting (a) and impacts of frosting on heat exchanger summarised from the literature discussed in Section 3.1.2 (b).

remains challenging, as the narrow size of the air channels in heat or energy exchangers makes it challenging to employ visual equipment, such as cameras. Moreover, the low temperatures inside the air channels and frost layer may damage such electronics [132]. The labour intensive feature of the visual observation also limits its use in practice.

Another method to indirectly detect frosting is through identifying the changes of heat and/or moisture effectiveness. Once a frost layer begins forming, it changes heat and mass transfer resistance and thus causes changes in heat and moisture effectiveness. Pressure drop across the exchanger has been considered a reliable approach in determining the presence of frost, and thus is the most used one [20]. The pressure drop control method is based on the fact that the static pressure drop across exchanger increases as partial blockage of air channels by frost layer [132].

Mahmood and Simonson [133] experimentally examined the frosting conditions in an enthalpy wheel under extremely cold conditions. The frost initiation and frost growth in the wheel pores were determined by an instantaneous increase in the pressure drop across the air channel. It was thus stated that the pressure drop was a reliable indicator to determine frost formation. Shang et al. [129] presented an experimental analysis for the frost growth in a desiccant-coated wheel, where three different methods were used to investigate frost growth. They used 1) a borescope at the exhaust outlets to observe and photograph frost growth inside the wheel channels over time, 2) laser beam scanning to indicate the existence of frost blockages, and 3) static pressure ports on pitot tubes to measure the static pressure drops across the wheel. Thereafter, based on their results from the pressure drop tests, cyclic fluctuations and a gradual increase in the average static pressure drop over 2 h were observed. Fluctuations in the laser transmission and a gradual decrease in its average signal also revealed the same conditions caused by frost accumulation.

Nasr et al. [132] applied various methods for detecting frosting in membrane-based heat exchangers, including visual inspection, change in effectiveness, and change in static pressure drop. In addition, a new method based on only the temperature difference measurements was also addressed. Temperatures of airflows were measured by 12 thermocouples placed at the outlet to capture a non-uniform temperature profile. Locations of thermocouples, pitot tubes and endoscopes in the exchanger are shown in Fig. 14.

In frost-free conditions, the bulk mean temperature at the outlet of the air channels can be measured by averaging twelve thermocouples, noted as T_{ave}, as the airflow was uniform at the outlet. Nevertheless, the airflow was no longer uniform due to the growth of frost. Thus, the air bulk mean temperature was measured further downstream, after the header and a mixer to obtain a more accurate result, noted as T_{mix}. Consequently, the difference between T_{ave} and T_{mix} was used as an in-

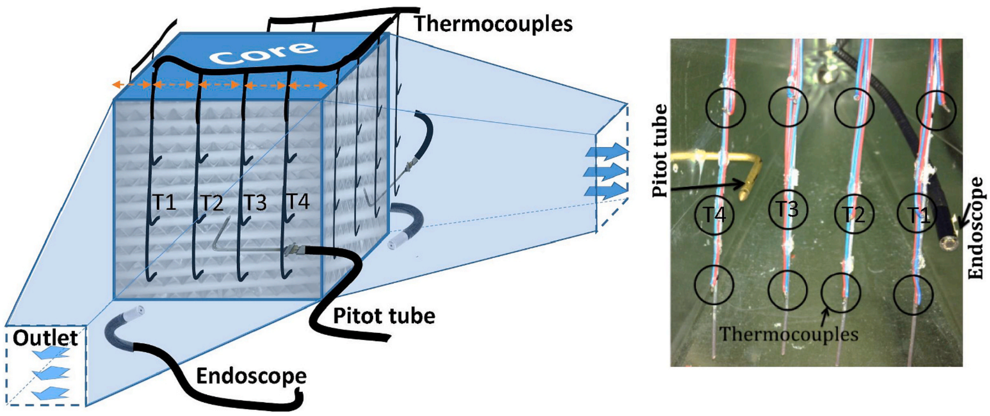


Fig. 14. Locations of thermocouples, pitot tubes and endoscopes in the exchanger [132].

dicator to determine the growth of frost. Among these four methods, the pressure drop and temperature difference methods, which detects frosting with quantitative results, were recommended as they were more responsive and accurate.

3.3. Frosting limit

The condition under which frost starts forming in exchangers is termed as the frosting limit [132]. The frosting limit has gained considerable attention recently, as it constitutes a criterion for selecting or designing heat and mass exchangers and determining the frost control preheating set point temperature [80].

Kragh et al. [117] examined the impacts of frosting on mechanical ventilation systems for single-family houses in Denmark. A typical heat exchanger composed of plastic was used, and its frosting limit was determined to be $-5\ ^{\circ}C$ for the outdoor air temperature when the indoor air relative humidity was 42%. Ruth et al. [135] investigated the frosting limit of an aluminium heat wheel, and they observed frosting when the outdoor air temperature varied from -26 to $-16\ ^{\circ}C$, accompanied by an indoor air relative humidity of 25–30%.

The above articles focused on frosting limits for sensible-only heat exchangers. However, the application of water vapour permeable membranes in flat-plate enthalpy exchangers and enthalpy wheels can reduce the dew point temperature of the extract air [132]. Thus, such systems can tolerate much lower outdoor air temperatures. Holmberg [136] numerically predicted the frost start temperature for an enthalpy wheel to be approximately 5–10 $^{\circ}C$ lower than for a heat wheel. The frosting limit in enthalpy wheels was related to operating conditions, heat exchanger design and coating material [137], which made a general value of frosting limit difficult to obtain. Mahmood and Simonson [133] tested an enthalpy wheel under extreme cold operating conditions and reported that when indoor air relative humidity was 30%, frost was observed at an outdoor temperature of $-29\ ^{\circ}C$. When indoor air relative humidity was increased to 40%, frost was observed when the outdoor temperature was $-20\ ^{\circ}C$. Fisk et al. [115] compared frosting limits for a cross-flow plate heat exchanger, a counter-flow plate heat exchanger, and a cross-flow plate enthalpy exchanger constructed from treated paper. It was summarised that the frosting limit was related to both operating conditions and heat exchanger design. Frost was initiated at a significantly lower temperature in enthalpy exchangers than in heat exchangers, particularly when indoor air relative humidity was high.

Most of the research in frosting limit mentioned above was obtained from experimental measurements. Liu et al. [138] developed mathematical models to determine frosting limits for a counter-flow flat-plate enthalpy exchanger and a cross-flow flat-plate enthalpy exchanger [139]. The frosting limit was defined as a specific combination of supply air temperature and exhaust air relative humidity. The critical temperature, which was defined as the lowest outdoor air temperature to maintain membrane surface temperature above freezing point at most likely frosting position, was predicted to be 5–10 $^{\circ}C$ lower in enthalpy exchanger than that in sensible heat exchanger, which is consistent with previous research [136]. Further, an analytical frosting limit model was also developed for a quasi-counter-flow flat-plate enthalpy exchanger [80]. The authors concluded that airflow rates, exhaust air temperature, and channel spacing had a minor impact on frosting limit compared with the influence of membrane diffusivity. In addition, a similar analytical model was developed by Navid et al. [140] for predicting frosting limits for water vapour permeable membranes. They confirmed that increasing membrane mass transfer rate considerably delayed frost formation and enhanced the frosting limit.

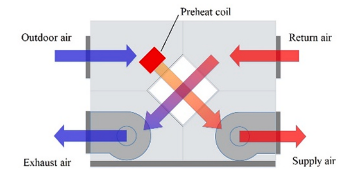
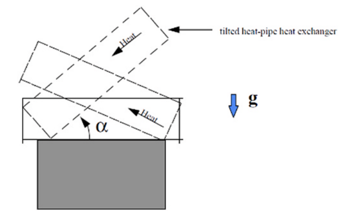
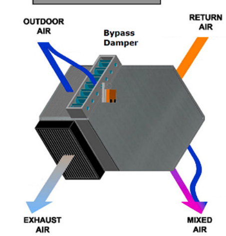
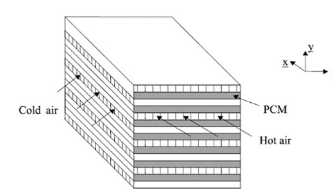
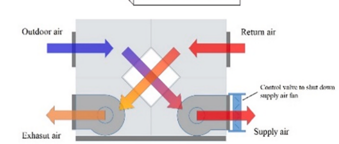
3.4. Frosting control strategies for various heat recovery technologies

The frosting control can be divided into two strategies. The first strategy is to prevent the occurrence of frosting in the first place, which is referred to as frosting prevention. The second strategy is to routinely remove frost from heat or energy exchanger surfaces after the frost formation, which is referred to as defrosting [141]. In this section, various frosting control strategies for residential building applications are discussed. All frosting control strategies explained in the following subsections are summarised and illustrated in Table 4.

3.4.1. Frosting prevention strategies

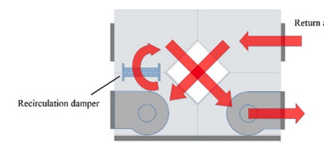
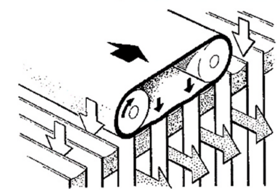
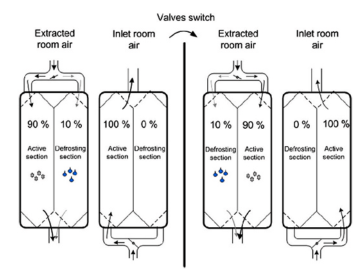
Frosting prevention can be achieved by preheating the outdoor air, heating the extract air, bypassing outdoor air or reducing the exchanger effectiveness. Preheating outdoor air can be applied to different types of heat exchangers. When the outdoor air temperature is below the frosting limit, the heating coil integrated into the supply side of the heat recovery unit is activated to preheat the outdoor air. Liu et al. [142] studied the energy-saving potential for a heat recovery system with preheating. According to their result, the amount of energy used for preheating was lower than that used for air post-conditioning after the heat recovery unit, and the difference was even more dramatic in a membrane-based enthalpy exchanger. Although this strategy was stated to have a higher upfront cost than other frosting control strategies, it still resulted in considerable operating cost savings in cold climates where frosting control is needed for a long period of time [76]. However, the amount of energy recovered from extract air was significantly reduced when applying this strategy because the temperature difference between outdoor and extract air inlets was narrowed when preheating the supply air [141]. The preheating strategy was conventionally controlled by a pre-set constant target temperature. Ko et al. [143] proposed a novel control strategy, which was a real-time based frost threshold temperature prediction model that considered exchanger effectiveness, extract air temperature, and relative humidity. Compared with constant target outdoor temperature ($-5\ ^{\circ}C$) control, their control resulted in 1–21%

Table 4
Summary of various frosting control strategies for residential building heat recovery applications.

Reviewed aspects				Applications	Control methods	Advantages	Disadvantages	Schematic diagrams	References
Frosting control strategies	Frosting Prevention	Preheating supply air		Any type of heat recovery techniques	Pre-set constant temperature Real-time based pre-set temperature	Simple operation	Relatively high upfront cost Reduced amount of recovered energy		[76, 141–144]
		Reducing exchanger effectiveness	Rotary speed adjustment	Rotary wheels	Pressure drop	Simple operation	N/A	–	[76,136, 137]
			Tilt control	Heat pipes	N/A	N/A	Extra control devices		[76,145, 146]
			Bypassing supply air	Flat-plate exchangers, rotary wheels, heat pipes	Pre-set temperature Exhaust side pressure drop	Simple operation High reliability No building depressurisation	Low overall efficiency Poor indoor thermal comfort		[76,117, 147–149]
			Pump speed adjustment	Run-around heat recovery system	N/A	N/A	N/A	–	[76]
		Improving frosting limits	Membranes with higher mass diffusivity	Flat-plate enthalpy exchangers	N/A	Lower critical outdoor temperature	High cost	–	[80]
			Higher coating hydrophilicity	Enthalpy wheels	N/A	Lower critical outdoor temperature	High cost Weak adhesive force High pressure drop	–	[137,150, 151]
			Phase-Change Materials (PCMs)	Flat-plate heat exchangers	N/A	High performance	High cost Extra control devices		[152]
	Defrosting	Exhaust-only		Flat-plate exchangers Heat pipes	Outdoor air temperature Exhaust air temperature	Cost-effective Simple operation Suitable for spaces need source control Suitable for semi-cold climates	Interrupted ventilation Poor IAQ Negative building pressure Not suitable in extremely cold arctic climates		[76,117, 144,153, 154]
		Recirculating indoor air		Flat-plate exchangers	Outdoor air temperature	Cost-effective No problem of negative	Poor IAQ Reduced effectiveness		

(*continued on next page*)

Table 4 (*continued*)

Reviewed aspects		Applications	Control methods	Advantages	Disadvantages	Schematic diagrams	References
				building pressure Most efficient in extremely cold climates	Not efficient in milder climates		[76,141, 153,155, 156]
	Hallgren's method	Flat-plate exchangers	N/A	Simple design No special modification to exchanger body Continuous defrosting	Extra cost of control system		[157]
	Two exchangers coupled in serial connection	Flat-plate exchangers	N/A	Continuous defrosting	Extra cost of additional exchanger, valve, and control system	–	[117]
	Two exchangers coupled in parallel	Flat-plate exchangers	N/A	Simple construction Continuous defrosting without supplementary heating High efficiency Low pressure drop	Extra cost of additional exchanger, valve, and control system Bigger size than other exchangers with the similar capacity		[20,158]

reduction in preheating coil capacity and 7%–72% reduction in energy consumption. Compared with constant target temperature (−10 °*C*) control, developed novel control consumed more energy; however, conventional control failed to prevent frosting during 1%–62% of total number in winter hours, which resulted in increased energy consumption for subsequent defrosting.

Reducing heat exchanger effectiveness is another strategy for frosting prevention. For a rotary wheel, effectiveness can be changed by adjusting the rotary speed. Simonson and Besant [137] conducted a numerical study in heat and mass transfer in an enthalpy wheel under sorption, condensation and frosting conditions. Their results revealed that uncontrolled condensation was stopped by reducing the rotary speed to 5 rpm, and the frosting was prevented by further decreasing the rotary speed to 1 rpm. In the case of a heat pipe, the effectiveness can be reduced by tilt control to reduce the condensate flowing back to the evaporator side [76]. Jasti et al. [145] compared the effectiveness of a heat pipe with different tilt angles, and an optimum inclination angle for the best thermal performance was determined as 60 °. Guo et al. [146] experimentally tested the impacts of various factors on the performance of an air-to-air heat pipe exchanger. Their research concluded that a positive inclination angle caused a sharp decrease in effectiveness. For a run-around heat recovery system, its effectiveness can be reduced by slowing the pump speed down for frosting prevention [76].

Bypassing outdoor air is a frosting prevention strategy that can be used for flat-plate heat exchangers, rotary wheels, and heat pipes while maintaining 100% of supply airflow rates [76]. In this strategy, when the outdoor air temperature drops below the frosting limit temperature, bypass damper at upstream of the heat exchanger in the supply air side modulates to reduce the amount of outdoor air flowing through the heat exchanger, or part of outdoor air is bypassed into the supply outlet, while the exhaust airflow rate remains constant. However, reducing the supply side airflow rate decreases the amount of energy recovered. When the exhaust side airflow rate remains unchanged, the exhaust outlet temperature rises above the frost threshold. This strategy was controlled by either a pre-set temperature or exhaust side pressure drop. In a study conducted by Kragh et al. [117], once the exhaust outlet temperature was below 3 °*C*, the control system reduced the outdoor airflow rate flowing through the exchanger until the exhaust outlet temperature was above 5 °*C*. This strategy offers advantages such as simple structure and high reliability [20,76].

Pacak et al. [147] compared preheating and bypassing of the outdoor air strategies, concluding that heat exchangers equipped with bypass dampers had very low overall heat recovery efficiency, and it required significantly more power for air treatment than required for preheating strategy. Thus, the preheating strategy was stated as more energy-efficient than bypassing. Similar results were presented in Ref. [148], where a comparison between preheating and bypassing was conducted under three different cold climates. The results revealed that preheating method outperformed than bypassing method as the actual effectiveness of heat exchangers using preheating was equal to the effectiveness without frosting. In Saskatoon, Canada, the energy saving percentage using the preheating method was 44%, which was twice of the saving percentage using bypassing (22%).

Frosting can be prevented by improving the frosting limit itself. This strategy can be applied to flat-plate enthalpy exchangers by increasing membrane mass diffusivity [80] and to enthalpy wheels by improving the moisture adsorption capacity of the coating materials [137,150, 151]. Qarnia et al. [152] used walls filled with phase-change materials (PCMs) sandwiched between cold and warm airflows in a cross-flow flat-plate heat exchanger. In this design, the heat stored in the PCMs prevented the surface temperature of the walls on the exhaust air side from dropping below the frosting temperature. Moreover, an electrical heating element was also used on the supply air side to further prevent frosting. According to their numerical simulation, when the electric power exceeded 300 W/m^2 for the heat transfer area, frosting could be avoided by using 3-mm-thick PCM layers. Moreover, when no electrical load was applied, frosting could still be avoided by increasing the PCM layer thickness to 6 mm.

3.4.2. Defrosting strategies

Exhaust-only defrosting strategy is among the most cost-effective and simple strategies to defrost a heat exchanger [144]. This technique periodically shuts down the supply air fan while using the warm exhaust airflow to heat the exchanger to defrost the frost layer. This strategy is commonly used for flat-plate heat exchangers and heat pipes, which is suitable for indoor environment that constantly requires source control. However, this strategy is not suitable for indoor environment that requires continuous outdoor air supply as the ventilation is interrupted and high IAQ requirements are not satisfied during the defrosting. Furthermore, negative indoor pressure may introduce outdoor pollutants due to the resulting uncontrolled infiltration [76,144].

A set point temperature was used to control the operation of the exhaust-only defrosting method. The supply air damper was closed entirely at the pre-set temperature, and an exhaust-only system was created until the temperature increased sufficiently to open it again. Robbin et al. [153] monitored the frosting conditions of eight different ERVs installed in Fairbanks, Alaska, and various defrosting cycles were applied for each ventilator. For instance, the ventilator named Panasonic FV-04VE1 normally operated for 1 h, operated in the exhaust-only mode for 30 min at 0 °*C*, and operated in the exhaust-only mode below −6.67 °*C* with a low fan speed. For the Lifebreath 150 ERVD ventilator, the supply air damper was closed once the temperature was less than −10 °*C*. The total heat recovery efficiency reached 36% and 57% for these two ventilators, respectively. Nyman and Simonson [154] presented a method that used the temperature of the exhaust air after the energy exchanger as an indicator to determine whether a defrosting measure was needed. Three defrosting pre-set temperatures were selected: 30 °*C*, indicating no defrosting control was applied; 5 °*C*, as a typical value in Finland; and 10 °*C*, as a more conservative value. The authors suggested 5 °*C* as the pre-set temperature considering its good defrosting effect and lowest energy waste, based on a life cycle analysis for 50 years. Kragh et al. [117] compared several frosting control strategies and stated that the exhaust-only strategy was suitable for cold climates such as Northern Europe but not serviceable in extremely cold arctic climates, as defrosting an exchanger filled with frost required several hours.

Another cost-effective and simple defrosting strategy is recirculating indoor air. When applying this strategy, the supply and exhaust air dampers were closed; thus, the indoor air was recirculated through the heat exchanger to defrost the frost layer. Because the ventilation was temporarily stopped and there was no outdoor air entering the occupied space during the defrosting cycle, this strategy was not suitable for applications that require high IAQ [76]. However, compared with the exhaust-only strategy, this strategy does not have the problem of negative building pressure [153]. This strategy was considered by Phillips et al. [141] to be the most efficient strategy in severely cold climates, but not the most efficient in milder climates. The recirculation strategy can be controlled by a pre-set temperature and a pre-set operation schedule [153]. In Phillips et al. [141], a heat recovery ventilator was operated in a pre-set recirculation mode for a fixed 2 min period at fixed time intervals (20 min). Further, a function was provided to calculate the heat recovery rate by applying this strategy.

Hallgren [157] proposed a novel flat-plate heat exchanger that continuously defrosted a small number of channels while all remaining channels were in normal operation. In this design, a small number of exchanger channels were inactive due to a plate continuously moving across the supply inlet side to block a small part of the opening, leaving time for the heat exchanger to defrost the inactive channels. The developed device was claimed to maintain a very thin frost layer such that at all time it had no significant impact on the heat exchanger and pressure drop. Kragh et al. [117] developed a design with two exchangers coupled in serial connection. In such a design, when frosting was forming in one exchanger, the flow direction was switched, and the frozen exchanger started defrosting while the other one operated normally. The disadvantage of such a system was the extra cost of an additional exchanger, valves, and a control system, which made this design not cost-effective. An improved design [158] proposed alternating the flow in two parallel coupled heat exchangers. The designed heat exchanger consists of two identical sections that can be cyclically defrosted by switching airflows between sections. Thus, one section was active and the other was passive (defrosting). After an adjusted time interval, the airflows switched. The authors stated that the developed heat exchanger had a simple construction and could provide continuous defrosting without supplementary heating. Further, the pressure drop was low, and the efficiency was high, which was tested to be 88%. However, Nasr et al. [20] pointed out that this system had a disadvantage, which was its size was bigger than other exchangers with a similar capacity.

In summary, Section 2 introduced and compared different heat recovery technologies used for residential buildings located in cold climate regions, and Section 3 introduced frosting control strategies. These are summarised in Fig. 15. Further research on the frosting control strategies that can provide indoor environments with continuous ventilation and simultaneously frosting risk-free operation in heat exchangers should be conducted in the future.

4. Effects of heat recovery on indoor relative humidity, temperature, virus, and gaseous pollutants transfer

4.1. Indoor relative humidity and temperature

Indoor RH was believed to have a significant impact on occupants' health and building structure [159]. In dry and cold areas, mechanical ventilation and indoor heating devices may significantly reduce indoor RH, which can result in various negative results such as poor thermal comfort, sensory irritation symptoms in the eyes, bad sleep quality, virus survival, static electricity, and voice disruption [160]. Furthermore, in dry and cold climates where the indoor RH tends to be low, the addition of humidifiers for better indoor humidity levels may increase the space latent heating load [160]. Moreover, improperly cleaned or maintained mist humidifiers can release aerosols containing dissolved minerals and opportunistic pathogens into the air, which become a potential pollution source and a health threat [161].

A possible alternative to humidifiers is mechanical ventilation with moisture recovery, which provides the space with outdoor air and also serves to maintain more acceptable indoor humidity with a low latent heat load. Barringer and McGugan [160] simulated the sensible and latent heat load in three reference houses in Winnipeg, MB with rotary non-desiccant heat exchanger and rotary desiccant exchanger based on their enthalpy balance model. Indoor RH setpoint was set to 35% for the simulation. The results showed that the rotary non-desiccant exchanger recovered a small amount of moisture and produced savings slightly greater than those for sensible-only plate exchangers. In contrast, the rotary desiccant exchanger was maintained an indoor RH level of 35% without any additional humidifier input for all cases except one house with a 40 L/s ventilation rate. For each house modelled, the energy savings from applying rotary desiccant exchanger ranged from 3300 to 12,000 kWh per year for ventilation rates ranging 20–70 L/s, and the annual saving was approximately $165 to $600. Consequently, the authors proposed a selection guide for selecting a proper heat recovery technology for various locations to maintain desired indoor RH levels based on average winter temperature, outdoor humidity ratio, and ventilation rate. Tafelmeier et al. [162] analysed energy saving potentials of sensible and total heat recovery systems when maintaining indoor RH at 50%, considering the impacts of different control strategies on both energy and cost savings. The authors proposed a novel humidity-based control strategy, and it was found that the use of total heat recovery was by far more beneficial than sensible heat recovery if

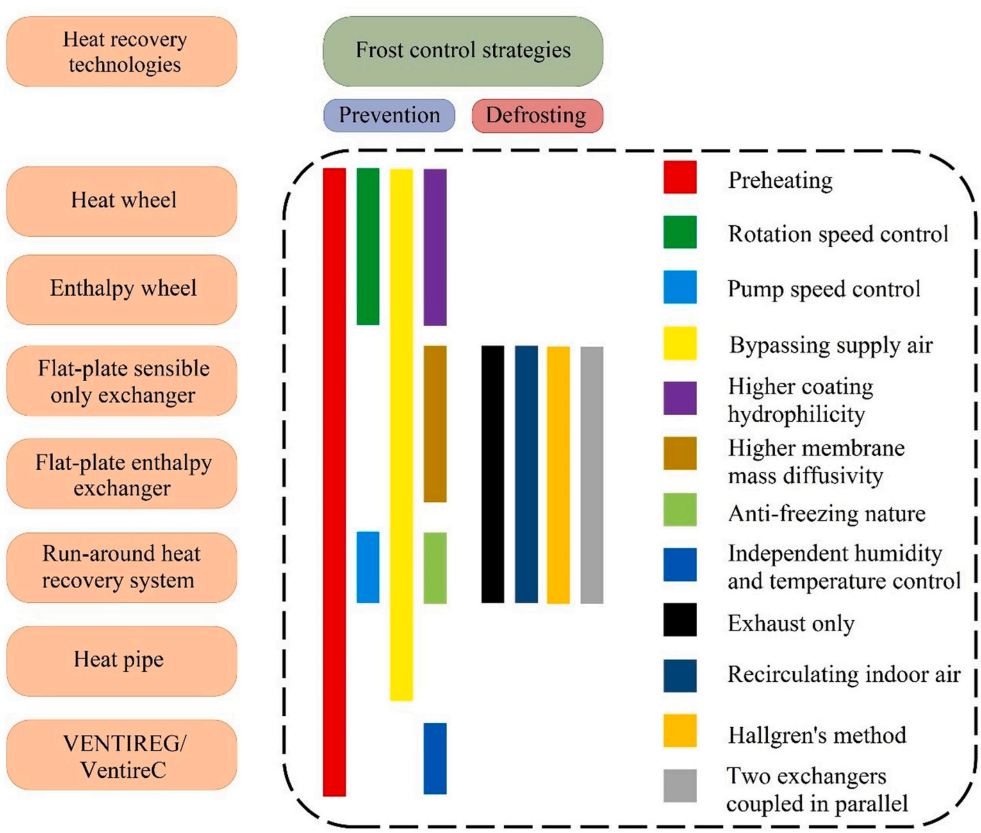


Fig. 15. Summary of heat recovery technologies, and available frosting control strategies.

there is no control for preventing excessive humidity. However, energy saving potentials between sensible and total heat recovery became similar if a humidity control was applied.

Indoor air temperature was found to have a strong impact on the perception of the IAQ, which decreases with an increase in the indoor air temperature at a constant pollution level [163]. However, low indoor temperatures can lead to indoor thermal discomfort [164]. Mechanical or natural ventilation can introduce outdoor air; however, without heat recovery or heating devices, the outdoor air temperature can be too low for direct supply. Furthermore, the use of additional heating devices reduces the overall energy efficiency. Prasauskas et al. [165] compared the IAQ in three Finnish and Lithuanian multifamily buildings before and after retrofits. The existing residential buildings had natural ventilation, whereas HRV was applied in the retrofitted buildings. Their results indicated dramatic improvements in the thermal condition after the retrofits. For example, in the case studies in Lithuania, the lowest measured indoor average temperature was raised by 3 $^{\circ}C$, i.e., from 15.1 to 18.2 $^{\circ}C$, which satisfied the lowest recommended indoor air temperature of 18 $^{\circ}C$ in Lithuania during the winter season, without using an additional heating device.

4.2. Virus transmission while using HRV

Although it is essential to maintain the IAQ via HRV owing to its air exchange nature and economic benefits, the IAQ can also deteriorate owing to improper installations of heat recovery units. Since 2019, an unprecedented viral disease Covid-19 has brought the globe to a halt, impacting most of mankind's activities [166]. Covid-19 belongs to the group of coronavirus, also known as Severe Acute Respiratory Syndrome Corona Virus 2 (SARS-CoV-2) [167]. HRV plays an important role in reducing long-range airborne transmission in buildings because the exposure from droplet aerosols can be reduced with a sufficient ventilation rate [168,169]. The possibility of viral particles being transmitted via heat recovery devices had been considered at the beginning of the pandemic. REHVA COVID-19 Guidance on how to operate HVAC [168] points out that the plate heat exchangers and enthalpy plate exchangers with permeable membranes, where air separation is guaranteed, are not sensitive for virus transmission. Rotary wheels have been shown to have almost no particle-bound pollutants, including bacteria, viruses, and fungi transmitted to the air when constructed, installed, and maintained properly. Pressure adjustment and bypassing have been suggested for heat recovery units when critical leakage has been detected to avoid cross contamination [168]. ASHRAE concluded that well-designed and well-maintained HRV systems should remain operating in residential, commercial buildings and medical facilities because maintaining at least normal ventilations rates, with proper temperature and humidity control of indoor spaces are of vital importance for maintaining health and combatting infectious bioaerosols [170]. However, it is also recommended by ASHRAE that HRV systems should be inspected for proper operation when an infectious outbreak has occurred or is expected in buildings to prevent the contribution of return air with infectious bioaerosols to the supply air.

4.3. Gaseous pollutants transmission while using HRV

Although virus particle transmission via heat recovery units is not an issue if they are properly installed and maintained, the transmission of gaseous pollutants in rotary wheels as a result of exhaust and supply air coming into contact with the same surfaces should not be ignored [168, 171]. Through adsorption and desorption processes, a range of gaseous pollutants can be transferred inside rotary wheels, especially when hygroscopic materials are used [172]. Indoor VOCs have been extensively studied previously for their ubiquity and indisputable negative influences on human health [173]. Although ventilation is recommended for reducing the indoor VOC concentration, the mass transfer of VOCs indoors is a rather complicated process, and the influence of ventilation on the resulting VOC concentration or VOC emission rate has not been established clearly [174]. Roulet et al. [106] investigated the transfer of VOCs from the exhaust air to the supply air using different methods. Their results demonstrated that leakages are typically negligible using the tracer gas method, and that adsorption–desorption is the main contributor to VOC transfer. Zhang et al. [107] numerically studied the transfer mechanism of VOCs in different adsorbents used for enthalpy wheels and concluded that, apart from pure diffusion, the convective mass transfer also plays an important role in VOC transfer. Pei et al. [175] developed a highly selective desiccant that could adsorb more water vapour and fewer VOCs for rotary wheels. This novel material was prepared by amine-functionalisation and subsequent self-assembly with malic acid on silica gel, and the selectivity of moisture/toluene and moisture/acetaldehyde were improved by 3.6 and 2.3 times, respectively, as compared with those for raw silica gel. For ventilation systems with heat recovery technologies using rotary wheels, a particular concern during winter operation is formaldehyde, which is a highly water-soluble compound and can be easily adsorbed onto the wheel structure from the exhaust air side and then desorb when the wheel encounters the supply air. Approximately 28–29% of formaldehyde is transferred from the exhaust air to the supply air, and a higher ventilation rate is required to adequately dilute the indoor contaminant concentrations [104]. Andersson et al. [105] experimentally tested the transfer of formaldehyde in six air-to-air enthalpy wheels in Northern Sweden. They concluded that rotary wheels could be applied for buildings where the air pollutants level is relatively low, provided the exchanger is appropriately installed and maintained. In order to prevent the adverse effects of possible microbial growth, proper draining of condensed water and defrosting of accumulated ice during winter operation are recommended [176].

5. Recommendations for future research

Based on the review of heat recovery technologies for residential buildings in cold climate regions, the following suggestions are proposed for future research:

- Development of novel heat recovery technologies that can be better tolerated in extremely cold climate regions without requiring extra energy use.
- Research on new membrane materials that have better moisture transfer ability and thus enhanced frosting limits, and simultaneously high selectivity to pollutants.
- Measurement of the long-term performance of MEE considering the effects of ageing and fouling on membranes.
- Numerical model of the frost growth process on heat exchanger surfaces that can reliably predict frost properties and their influence on heat exchanger performance under more general situations.
- New strategies for frost control that are energy-efficient and sustain continuous ventilation for heat recovery.
- Methods and criteria for choosing appropriate heat recovery technologies and frosting control strategies for different applications in various regions considering energy and indoor environment.
- Further research on heat recovery technologies integrated with renewable energy such as photovoltaic [177].

6. Conclusions

Mechanical ventilation provides airtight residential buildings with adequate air exchange rates, thereby ensuring a good IAQ. Through integration with heat recovery technologies, a significant amount of the energy used for air handling can be saved when employing mechanical ventilation. Literature related to heat recovery technologies for mechanical ventilation in residential buildings, with a specific focus on the applications in cold climates, were reviewed in this paper. According to these studies, numerous types of heat recovery technologies can be incorporated with mechanical ventilation, such as rotary wheels, flat-plate heat exchangers, run-around heat exchangers, and heat pipes, in order to recover either the heat and/or moisture from the extract air in the heating season. Novel designs such as run-around heat recovery, FBRs, VENTIREG, and VentireC have been introduced over recent years; these approaches exhibit better adaption in cold climates. Furthermore, frosting in the exhaust air channels is one of the most common problems associated with heat exchangers in cold climates; this is a rather complicated phase change and dynamic process that needs further research.

Based on this review, main conclusions were drawn as below:

- Heat loss from a residential building ventilation system can be up to 35–40 kWh/m^2, and 60–95% of ventilation heat loss can be recovered by heat recovery system. Minimum sensible heat recovery efficiency is required in the range of 70%–80% in most countries, and minimum total heat recovery efficiency is only required in a few countries, such as Finland, USA, and Canada.
- Heat wheels have been broadly used mainly due to their high heat recovery efficiency and high tolerance to frost in cold climates. Cross-contamination is a critical issue of using heat wheel, which consists of carryover and pressure leakages. The cross-contamination can be reduced by using a purge section and proper fan arrangement.
- Flat-plate exchangers are more favourable for their high long-term reliability. Carryover problems are avoided by easy duct sealing. Further, water vapour-permeable membranes are used by replacing metal plates to recover both sensible and latent heat, while frosting limits are enhanced through moisture recovery when operating in winter.
- Quasi-counter-flow configuration in flat-plate exchangers has been developed recently, which provides easier duct sealing than pure counter-flow configuration. The effectiveness can be improved by approximately 5% compared with the cross-flow exchanger.
- Novel heat recovery technologies for residential buildings have been introduced in recent years such as fixed-bed regenerator with two stationary exchangers, VENTIREG and VentireC. They have exceptional adaption in cold climates with less or even no frost control measures. The operating temperatures can be as low as -40, -28 , and -30 $^\circ C$ without additional defrosting strategies for fixed-bed regenerator, VENTIREG, and VentireC, respectively.
- The mechanism of frosting growth in heat exchangers remains unclear. Most of the numerical models predicting frosting properties and their effects on heat exchanger performance are based on rather simplified assumptions, and they are limited to specific surface geometries and operating conditions. In principle, heat transfer decreases with the growth of frost. The frosting increases pressure drop through the exchanger, which results in more energy use for fans and degrades IAQ due to the resulting low ventilation rates. The accumulated frost inside the channels may damage the exchanger construction.
- Frosting limit is mainly associated with outdoor air temperature and indoor air humidity levels. The frosting limit is crucial for selecting heat exchangers and determining preheating triggering conditions in frosting control. The frosting limit can be improved notably by enhancing moisture recovery in enthalpy exchangers. Further, the critical outdoor temperature is predicted to be 5–10 °C lower in enthalpy exchangers than that in the sensible-only heat exchangers.
- Using visual observations to detect frost inside heat exchangers is accurate. However, this method is labour intensive and not always possible due to the limited space and view in AHU. Changes in pressure drop and effectiveness are more applicable in practice to identify frost onset because of their ease of use and automatic and continuous monitoring traits.
- Frosting control strategies contain frosting prevention and defrosting. Both require additional energy consumption or sacrificing IAQ. The criteria for selecting proper frosting control strategies for different applications in various climates need to be established.
- Sufficient mechanical ventilation with heat recovery has been proven to be efficient at improving IAQ with low energy use. The transfer of particle-bound pollutants, including airborne bacteria, viruses and fungi, is not an issue when HRVs are properly installed and maintained. However, gaseous pollutants such as formaldehyde can be transferred from exhaust air to supply air in rotary wheels through the adsorption–desorption process especially when hygroscopic rotor materials are used.

Declaration of competing interest

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

Acknowledgements

This study is a part of the Defreeze MEE Now project 296489, which is financially supported by the Research Council of Norway and Flexit AS, and it is a close collaboration between Flexit AS, Norwegian University of Science and Technology (NTNU) and SINTEF. The authors acknowledge Ms. Anneli Halfvardsson and Dr. Fredrik Karlsson from Flexit AS for their broad experience and support.

References

[1] Guo S, et al. Modelling building energy consumption in China under different future scenarios. Energy 2021;214:119063.
[2] Becerik-Gerber B, et al. Civil Engineering Grand Challenges: Opportunities for Data Sensing, Information Analysis, and Knowledge Discovery28; 2013.
[3] Ahmad AS, et al. A review on applications of ANN and SVM for building electrical energy consumption forecasting. Renew Sustain Energy Rev 2014;33:102–9.
[4] Lin B, Liu H. CO2 emissions of China's commercial and residential buildings: Evidence and reduction policy. Build Environ 2015;92:418–31.
[5] Yi Jiang DY, Guo Siyue. Shan hu, China building energy use 2018. Beijing: Building Energy Research Center of Tsinghua University; 2019.
[6] Types of residential buildings and their site selection. Available from: https://theconstructor.org/building/types-site-selection-residential-building/5995/.
[7] Alwetaishi M, Gadi M. New and innovative wind catcher designs to improve indoor air quality in buildings. Energy Built Environ. 2020.
[8] Peng Z, Deng W, Tenorio R. An integrated low-energy ventilation system to improve indoor environment performance of school buildings in the cold climate zone of China. Build Environ 2020;182:107153.
[9] Izadyar N, et al. Impacts of façade openings' geometry on natural ventilation and occupants' perception: a review. Build Environ 2020;170:106613.
[10] 2013 ASHRAE handbook : fundamentals. Atlanta, GA: ASHRAE; 2013.
[11] Dodoo A, Gustavsson L, Sathre R. Primary energy implications of ventilation heat recovery in residential buildings. Energy Build 2011;43(7):1566–72.
[12] Mardiana-Idayu A, Riffat SB. Review on heat recovery technologies for building applications. Renew Sustain Energy Rev 2012;16(2):1241–55.
[13] Bai H, et al. State-of-art in modelling methods of membrane-based liquid desiccant heat and mass exchanger: a comprehensive review. Int J Heat Mass Tran 2018;125:445–70.
[14] Zhang L-Z. Progress on heat and moisture recovery with membranes: from fundamentals to engineering applications. Energy Convers Manag 2012;63:173–95.
[15] Abdel-Salam MRH, et al. State-of-the-art in liquid-to-air membrane energy exchangers (LAMEEs): a comprehensive review. Renew Sustain Energy Rev 2014;39:700–28.
[16] Huang S-M, Zhang L-Z. Researches and trends in membrane-based liquid desiccant air dehumidification. Renew Sustain Energy Rev 2013;28:425–40.
[17] Yang B, et al. A review of membrane-based air dehumidification. Indoor Built Environ 2015;24(1):11–26.
[18] Zeng C, Liu S, Shukla A. A review on the air-to-air heat and mass exchanger technologies for building applications. Renew Sustain Energy Rev 2017;75:753–74.
[19] O'Connor D, Calautit JKS, Hughes BR. A review of heat recovery technology for passive ventilation applications. Renew Sustain Energy Rev 2016;54:1481–93.
[20] Rafati Nasr M, et al. A review of frosting in air-to-air energy exchangers. Renew Sustain Energy Rev 2014;30:538–54.
[21] Justo Alonso M, et al. Review of heat/energy recovery exchangers for use in ZEBs in cold climate countries. Build Environ 2015;84:228–37.

[22] Spinoni J, Vogt J, Barbosa P. European degree-day climatologies and trends for the period 1951–2011. Int J Climatol 2015;35(1):25–36.
[23] Guide to determining climate regions by county. U.S. Department of energy's building America. 2015.
[24] Mourshed M. Climatic parameters for building energy applications: a temporal-geospatial assessment of temperature indicators. Renew Energy 2016;94:55–71.
[25] ASHRAE, standard 62.2-2013: ventilation and acceptable indoor air quality in low-rise residential buildings. 2013 [Atlanta, GA].
[26] Juodis E. Extracted ventilation air heat recovery efficiency as a function of a building's thermal properties. Energy Build 2006;38(6):568–73.
[27] B.a.H. Ministry of Transport. Danish building regulations - codes (BR18). 2018.
[28] DS 474 Code for Thermal Indoor Climate. Danish standard foundation (DS). 1995.
[29] författningssamling, B., BFS. 3-BBR 2014;21:2014.
[30] Building technical regulations (TEK17) with guidance, d.f. byggkvalitet. 2017.
[31] Advice for a good indoor climate in the home. Available from: https://www.fhi.no/ml/miljo/inneklima/fremhevede-artikler-inneklima-og-helse/godt-inneklima-brosjyre/; 2016.
[32] Health NIoP. Recommended professional standards for indoor climate. 2015.
[33] Environment Mot. D2 National building code of Finland: Indoor climate and ventilation of buildings. 2012 [Helsinki].
[34] Environment Mot. National Building Code of Finland: Indoor climate and ventilation of new buildings. 2017 [Helsinki].
[35] Environment Mot. National Building Code of Finland: Energy performance of new buildings. 2017 [Helsinki].
[36] Säteri J. FINNISH CLASSIFICATION OF INDOOR CLIMATE 2000: REVISED TARGET VALUES. 2002.
[37] Gost 30494-2011: residential and public buildings. Indoor microclimate parameters, F.A.f.T.R.a. Metrology. 2011 [Moscow].
[38] Thermal performance of the buildings. Moscow: 2012; 2012.
[39] ASHRAE, standard 62.2-2016, ventilation and acceptable indoor air quality in residential buildings. 2016 [Atlanta, GA].
[40] ASHRAE, standard 55-2013 thermal environmental conditions for human occupancy. 2013 [Atlanta, GA].
[41] ASHRAE, standard 90.1-2019: energy standard for buildings except low-rise residential buildings. 2019 [Atlanta, GA].
[42] Concil IC. 2012 International Energy Conservation Code (IECC). 2012.
[43] Canada NRCo, editor. National building code of Canada; 2020 [Ottawa].
[44] Group, C. CAN/CSA-F326-M91 (R2019) Residential Mechanical Ventilation Systems. 2019.
[45] Group, C. CAN/CSA C439-18 Standard laboratory methods of test for rating the performance of heat/energy-recovery ventilators. 2018.
[46] Canada NR. 2012 R-2000 Standard. 2012.
[47] China, M.o.H.a.U.-R.C.o.t.P.s.R.o.. GB 50189-2015 Design standard for energy efficiency of public buildings. 2015 [Beijing].
[48] China, M.o.H.a.U.-R.C.o.t.P.s.R.o.. GB50736–G52016 Code for design of heating ventilation and air conditioning. 2016 [Beijing].
[49] China, M.o.H.a.U.-R.C.o.t.P.s.R.o.. GB T18883-2002 Indoor Air Quality Standard. 2002 [Beijing].
[50] Shi J, et al. Characteristics, trend, and precursors of extreme cold events in northwestern North America. Atmos Res 2020:105338.
[51] Cao X, et al. A novel packaged outdoor air dehumidifier with exhaust air heat pump – Experiment and simulation. Appl Therm Eng 2020;181:115986.
[52] Zhang LZ. Total heat recovery: heat and moisture recovery from ventilation air. , New York: Nova Science Publishers Inc; 2008.
[53] Awbi HB. Indoor air quality in UK homes and its impact on health. 2015.
[54] Khanmohammadi S, Shahsavar A. Energy analysis and multi-objective optimization of a novel exhaust air heat recovery system consisting of an air-based building integrated photovoltaic/thermal system and a thermal wheel. Energy Convers Manag 2018;172:595–610.
[55] Men Y, Liu X, Zhang T. Performance comparison of different total heat exchangers applied for waste heat recovery. Appl Therm Eng 2021;182:115715.
[56] Nóbrega CEL, Brum NCL. Modeling and simulation of heat and enthalpy recovery wheels. Energy 2009;34(12):2063–8.
[57] Jedlikowski A, Kanaś P, Anisimov S. Heat and mass transfer inside the rotary heat exchanger operating under high speed rotor conditions. Int J Heat Mass Tran 2020;152:119558.
[58] Tu R, Liu X-H, Jiang Y. Performance comparison between enthalpy recovery wheels and dehumidification wheels. Int J Refrig 2013;36(8):2308–22.
[59] Ge F, et al. Energy savings potential of a desiccant assisted hybrid air source heat pump system for residential building in hot summer and cold winter zone in China. Energy Build 2011;43(12):3521–7.
[60] Wallin J, Claesson J. Improving heat recovery using retrofitted heat pump in air handling unit with energy wheel. Appl Therm Eng 2014;62(2):823–9.
[61] Yang F, et al. Investigation on humidification performance of silica gel rotary wheel system in winter. Build Environ 2020;183:107064.
[62] Tu R, Li J, Hwang Y. Fresh air humidification in winter using desiccant wheels for cold and dry climate regions: optimization study of humidification processes. Int J Refrig 2020;118:121–30.
[63] Tu R, Li J, Hwang Y. Performance analysis of desiccant wheels assisted fresh air humidifiers in winter using natural gas boilers: applied in cold and dry climate regions. Int J Refrig 2020;119:24–36.
[64] Nasif Ms MG, Behnia M. Heat and mass transfer in air to air enthalpy heat exchangers, in 6th international conference on heat transfer, fluid mechanics and thermodynamics. 2005 [Matsushima, Japan].
[65] Shah RK, Skiepko T. Influence of leakage distribution on the thermal performance of a rotary regenerator. Appl Therm Eng 1999;19(7):685–705.
[66] Han H. An experimental study on air leakage and heat transfer characteristics of a rotary-type heat recovery ventilator. Int Air Condition Refrige 2005;13:83–8.
[67] Shiminski J. Energy Recovery Wheels | Understanding Cross Contamination/ Leakage. Available from. 2012. : https://www.dac-hvac.com/energy-recovery-wheels-understanding-cross-contamination-leakage/.
[68] Ruan W, Qu M, Horton WT. Modeling analysis of an enthalpy recovery wheel with purge air. Int J Heat Mass Tran 2012;55(17):4665–72.
[69] Ma J, Lu C, Liu H. Fault diagnosis for the heat exchanger of the aircraft environmental control system based on the strong tracking filter. PLoS One 2015:10. p. e0122829.
[70] Nasif M. Analysis and modelling of membrane heat exchanger in HVAC energy recovery systems. Mechanical and Manufacturing Engineering. UNSW, PhD 2008.
[71] Zhang L-Z, Liang C-H, Pei L-X. Heat and moisture transfer in application scale parallel-plates enthalpy exchangers with novel membrane materials. J Membr Sci 2008;325(2):672–82.
[72] Min J, Su M. Performance analysis of a membrane-based enthalpy exchanger: Effects of the membrane properties on the exchanger performance348; 2010. p. 376–82.
[73] Yaïci W, Ghorab M, Entchev E. Numerical analysis of heat and energy recovery ventilators performance based on CFD for detailed design. Appl Therm Eng 2013;51(1):770–80.
[74] Zhang L-Z. Convective mass transport in cross-corrugated membrane exchangers. J Membr Sci 2005;260(1):75–83.
[75] Wang L, Curcija D, Breshears J. The energy saving potentials of zone-level membrane-based enthalpy recovery ventilators for VAV systems in commercial buildings. Energy Build 2015;109:47–52.
[76] 2020 ASHRAE handbook—HVAC systems and equipment. Atlanta, GA: ASHRAE; 2020.
[77] Nasif M, et al. Membrane heat exchanger in HVAC energy recovery systems, systems energy analysis. Energy Build 2010;42(10):1833–40.
[78] Zhang L-Z. Heat and mass transfer in a quasi-counter flow membrane-based total heat exchanger. Int J Heat Mass Tran 2010;53(23):5478–86.
[79] Liu P, et al. Performance of a quasi-counter-flow air-to-air membrane energy exchanger in cold climates. Energy Build 2016;119:129–42.
[80] Liu P, et al. A frosting limit model of air-to-air quasi-counter-flow membrane energy exchanger for use in cold climates. Appl Therm Eng 2017;111:776–85.
[81] Kays WM, London AL. Compact heat exchangers. 1984.
[82] Howell RH. Principles of heating ventilating and air conditiong, 8th edition. Atlanta, GA: ASHRAE; 2017.
[83] Davidsson H, Bernardo LR, Hellström B. Theoretical and experimental investigation of a heat exchanger suitable for a hybrid ventilation system. MDPI. Buildings 2013;2013:18–38.
[84] Chen X, et al. Recent progress in liquid desiccant dehumidification and air-conditioning: a review. Energy Built Environ. 2020;1(1):106–30.
[85] Fan H, et al. Run-around heat recovery system using cross-flow flat-plate heat exchangers with aqueous ethylene glycol as the coupling fluid. Build Eng 2005;111:901–10.
[86] Fan H, et al. Performance of a Run-Around System for HVAC Heat and Moisture Transfer Applications Using Cross-Flow Plate Exchangers Coupled with Aqueous Lithium Bromide12; 2006. p. 313–36.
[87] Wallin J, Madani H, Claesson J. Ventilation heat recovery with run around coil-System analysis and a study on efficency improvment-Part I. In: Ashrae RAL Symposium on Sustainability and Green Buildings; 2009.
[88] Shao L, Riffat SB. Flow loss caused by heat pipes in natural ventilation stacks. Appl Therm Eng 1997;17(4):393–9.
[89] Raghavendra CR, Hasavimath K, Naik K. Study on effectiveness of heat pipe heat exchanger with copper tube cylinders. Mater Today Proc 2020.
[90] Lin S, Broadbent J, McGlen R. Numerical study of heat pipe application in heat recovery systems. Appl Therm Eng 2005;25(1):127–33.
[91] Calautit JK, O'Connor D, Hughes BR. A natural ventilation wind tower with heat pipe heat recovery for cold climates. Renew Energy 2016;87:1088–104.
[92] Diao YH, et al. Experimental study on the heat recovery characteristic of a heat exchanger based on a flat micro-heat pipe array for the ventilation of residential buildings. Energy Build 2017;152:448–57.
[93] Zhou F, Duan W, Ma G. Thermal performance of a pump-driven loop heat pipe as an air-to-air energy recovery device. Energy Build 2017;151:206–16.
[94] Zhang S, Ma G, Zhou F. Experimental study on a pump driven loop-heat pipe for data center cooling. J Energy Eng 2015;141(4):4014054.
[95] Zhang C, et al. Dynamic thermal performance and parametric analysis of a heat recovery building envelope based on air-permeable porous materials. Energy 2019;189:116361.
[96] Krishnan EN, et al. Development of a small-scale test facility for effectiveness evaluation of fixed-bed regenerators. Appl Therm Eng 2020;174:115263.
[97] Aristov YI, Mezentsev IV, Mukhin VA. A new approach to regenerating heat and moisture in ventilation systems. Energy Build 2008;40(3):204–8.
[98] Shkatulov A, et al. Novel adsorption method for moisture and heat recuperation in ventilation: composites "LiCl/matrix" tailored for cold climate. Energy 2020;201:117595.
[99] O'Connor D, Calautit J, Hughes BR. Effect of rotation speed of a rotary thermal wheel on ventilation supply rates of wind tower system. Energy Proc 2015;75:1705–10.
[100] O'Connor D, Calautit JK, Hughes BR. A study of passive ventilation integrated with heat recovery. Energy Build 2014;82:799–811.
[101] Calautit JK, et al. Development of a natural ventilation windcatcher with passive heat recovery wheel for mild-cold climates: CFD and experimental analysis. Renew Energy 2020;160:465–82.

[102] De Antonellis S, et al. Desiccant wheels for air humidification: an experimental and numerical analysis. Energy Convers Manag 2015;106:355–64.
[103] Roulet C-A, et al. Real heat recovery with air handling units. Energy Build 2001; 33:495–502.
[104] Hult EL, Willem H, Sherman MH. Formaldehyde transfer in residential energy recovery ventilators. Build Environ 2014;75:92–7.
[105] Andersson B, et al. Mass transfer of contaminants in rotary enthalpy exchangers. Indoor Air 1993;3(2):143–8.
[106] Roulet C-A, et al. Effect of chemical composition on VOC transfer through rotating heat exchangers. Energy Build 2002;34(8):799–807.
[107] Zhang G, et al. Numerical study on the contribution of convective mass transfer inside high-porosity adsorbents in the VOC adsorption process. Indoor Built Environ 2013;22:551–8.
[108] Fernández-Seara J, et al. Experimental analysis of an air-to-air heat recovery unit for balanced ventilation systems in residential buildings. Energy Convers Manag 2011;52(1):635–40.
[109] Vali A, et al. Numerical model and effectiveness correlations for a run-around heat recovery system with combined counter and cross flow exchangers. Int J Heat Mass Tran 2009;52(25):5827–40.
[110] Vali A, et al. Numerical modeling of fluid flow and coupled heat and mass transfer in a counter-cross-flow parallel-plate liquid-to-air membrane energy exchanger. Int J Heat Mass Tran 2015;89:1258–76.
[111] Dhital P, Besant RW, Schoenau GJ. Integrating run-around heat exchanger systems into the design of large office buildings. In: United States: American society of heating, refrigerating and air-conditioning engineers, inc. Atlanta: GA (United States); 1995.
[112] Xue L, et al. Operation characteristics of air–air heat pipe inserted plate heat exchanger for heat recovery. Energy Build 2019;185:66–75.
[113] Hughes BR, Chaudhry HN, Calautit JK. Passive energy recovery from natural ventilation air streams. Appl Energy 2014;113:127–40.
[114] Alongi A, Angelotti A, Mazzarella L. Experimental investigation of the steady state behaviour of Breathing Walls by means of a novel laboratory apparatus. Build Environ 2017;123:415–26.
[115] Fisk W, et al. Onset of freezing in residential air-to-air heat exchangers. 1984.
[116] Fisk W, et al. Performance of residential air-to-air heat exchangers during operation with freezing and periodic defrosts. 1984.
[117] Kragh J, Rose J, Svendsen S. Mechanical ventilation with heat recovery in cold climates. In: Proceedings of the Seventh Nordic Symposium on Building Physics in Nordic Countries; 2005.
[118] Iragorry J, Tao Y-X, Jia S. Review article: a critical Review of Properties and Models for frost formation analysis. HVAC R Res 2004;10(4):393–420.
[119] Chen H, Thomas L, Besant R. Modeling frost characteristics on heat exchanger fins: Part I. Numerical model. 2000:106 [p. PA/].
[120] O'Neal D, Tree DR. A review of frost formation in simple geometries. Build Eng 1985;91:267–81.
[121] Yang D-K, Lee K-S. Modeling of frosting behavior on a cold plate. Int J Refrig 2005;28(3):396–402.
[122] Yang D-K, Lee K-S. Dimensionless correlations of frost properties on a cold plate. Int J Refrig 2004;27(1):89–96.
[123] Lee K-S, Jhee S, Yang D-K. Prediction of the frost formation on a cold flat surface. Int J Heat Mass Tran 2003;46(20):3789–96.
[124] Kondepudi S, O'Neal D. The effects of frost growth on extended surface heat exchanger performance: a review. Build Eng 1987;93:258–74.
[125] Yang D-K, Lee K-S, Song S. Modeling for predicting frosting behavior of a fin–tube heat exchanger. Int J Heat Mass Tran 2006;49(7):1472–9.
[126] Huang J-M, et al. The effects of frost thickness on the heat transfer of finned tube heat exchanger subject to the combined influence of fan types. Appl Therm Eng 2008;28(7):728–37.
[127] Xianghong G, Xiangying L. Basic heat and mass transfer regularity on sublimation frost-formation phenomenon. J Luoyang Univ 1999;14(4):35–8.
[128] Chen, H., R.W. Besant, and Y.X. Tao. Frost characteristics and heat transfer on a flat plate under freezer operating conditions: Part 2, Numerical modeling and comparison with data. 1999. United States: American Society of Heating, Refrigerating and air-conditioning engineers, inc., Atlanta, GA (US).
[129] Shang W, Chen H, Besant RW. Frost growth in regenerative wheels. J Heat Tran 2005;127(9):1015–26.
[130] Xianghong G, Weixiu K. Study on the development of physical model of sublimation frost formation. J Dalian Univ 1999;20(6):34–8.
[131] Maldonado JM, et al. Frost detection method on evaporator in vapour compression systems. Int J Refrig 2020;110:75–82.
[132] Rafati Nasr M, et al. Experimental methods for detecting frosting in cross-flow air-to-air energy exchangers. Exp Therm Fluid Sci 2016;77:100–15.
[133] Mahmood GI, Simmonson CJ. Frosting condition for an energy wheel in laboratory simulated extreme cold weather. In: 7th International Cold Climate HVAC Conference; 2012. p. 223–31.
[134] Kim M-H, Lee K-S. Determination method of defrosting start-time based on temperature measurements. Appl Energy 2015;146:263–9.
[135] Ruth DW, Fisher DR, Gawley HN. Investigation of frosting in rotary air-to-air heat exchangers. Build Eng 1975;81:410–7.
[136] Holmberg RB. Prediction of condensation and frosting limits in rotary wheels for heat recovery in buildings. Build Eng 1989;95:64–9.
[137] Simonson CJ, Besant RW. Heat and moisture transfer in energy wheels during sorption, condensation, and frosting conditions. J Heat Tran 1998;120(3): 699–708.
[138] Liu P, et al. Frosting limits for counter-flow Membrane Energy Exchanger (MEE) in cold climates. Indoor Air Q Clim 2014:488–96.
[139] Liu P, et al. A theoretical model to predict frosting limits in cross-flow air-to-air flat plate heat/energy exchangers. Energy Build 2016;110:404–14.
[140] Navid P, Niroomand S, Simonson CJ. An analytical model for predicting frosting limit in membranes. Int J Refrig 2019;99:316–26.
[141] Phillips EG, et al. Comparison of freezing control strategies for residential air-to-air heat recovery ventilators. 1989.
[142] Liu P, et al. Energy transfer and energy saving potentials of air-to-air membrane energy exchanger for ventilation in cold climates. Energy Build 2017;135:95–108.
[143] Ko J, Park J, Jeong J-W. Energy saving potential of a model-predicted frost prevention method for energy recovery ventilators. Appl Therm Eng 2020: 116450.
[144] Afshin M, Tardif M, Rice T. Frost control strategies for air-to-air energy recovery, in ASHRAE Seminar 22, 2019 Annual Conference. 2019 [Kansas City].
[145] Raghuram J, et al. Thermal performance of a selected heat pipe at different tilt angles. IOP Conf Ser Mater Sci Eng 2017;225:12043.
[146] Guo P, Ciepliski DL, Besant RW. A Testing and HVAC design Methodology for air-to-air heat pipe heat exchangers. HVAC R Res 1998;4(1):3–26.
[147] Pacak A, et al. Analysis of freeze protection methods for recuperators used in energy recovery from exhaust air. E3S Web Conf 2017;22:129.
[148] Rafati Nasr M, et al. Evaluation of defrosting methods for air-to-air heat/energy exchangers on energy consumption of ventilation. Appl Energy 2015;151:32–40.
[149] Jedlikowski A, et al. Frost formation and freeze protection with bypass for counter-flow recuperators. Int J Heat Mass Tran 2017;108:585–613.
[150] Moallem E, et al. Experimental measurements of the surface coating and water retention effects on frosting performance of microchannel heat exchangers for heat pump systems. Exp Therm Fluid Sci 2012;39:176–88.
[151] Liu Z, et al. Experimental investigations of frost release by hydrophilic surfaces. Front Energy Power Eng China 2010;4(4):475–87.
[152] El Qarnia H, Lacroix M, Mercadier Y. Use of a phase change material to prevent frosting in a compact crossflow air exchanger. Energy Convers Manag 2001;42 (10):1277–96.
[153] Garber-Slaght R, Stevens V, Madden D. energy recovery ventilators in cold climates. Cold Clim Hous Res Center: Fairbanks, AK 2014.
[154] Nyman M, Simonson CJ. Life cycle assessment of residential ventilation units in a cold climate. Build Environ 2005;40(1):15–27.
[155] Phillips EG, et al. A Model to compare freezing control Strategies for residential air-to-air heat recovery ventilators. Build Eng 1989;95:475–83.
[156] Ninomura, P.T. and R. Bhargava, Heat recovery ventilators in multifamily residences in the arctic. Conference: annual meeting of the American society of heating, refrigeration and air-conditioning engineers, inc. (ASHRAE), san diego, CA (United States), 24-28 jun 1995. 1995: American society of heating, refrigerating and air-conditioning engineers, inc., Atlanta, GA (United States). Medium: X; Size: pp. 961–966.
[157] Hallgren K. In: Patent US, editor. Method and device for defrosting heat exchanger without impairment of its heat exchange efficiency; 1981 [United States].
[158] Kragh J, et al. New counter flow heat exchanger designed for ventilation systems in cold climates. Energy Build 2007;39(11):1151–8.
[159] Wolkoff P. Indoor air humidity, air quality, and health – An overview. Int J Hyg Environ Health 2018;221(3):376–90.
[160] Barringer CG, McGugan CA. Effect of residential air-to-air heat and moisture exchangers on indoor humidity. Build Eng 1989:95.
[161] Sahai D. Evidence Brief: humidifier use in health care. Toronto, ON: Queen's Printer for Ontario; 2017. O.A.f.H.P.a.P.P.H. Ontario.
[162] Stefanie T, Pernigotto G, Gasparella A. Annual performance of sensible and total heat recovery in ventilation systems: humidity control constraints for European climates. Buildings 2017;7:28.
[163] Fang L, Clausen G, Fanger P. Impact of temperature and humidity on the perception of indoor air quality. Indoor Air 2004;8:80–90.
[164] Section 2 of the Building Air Quality Guide: Factors Affecting Indoor Air Quality, U.S.E.P. Agency, Editor.
[165] Prasauskas T, et al. Effects of energy retrofits on indoor air quality in three northern European countries. Energy Proc 2016;96:253–9.
[166] Lipinski T, et al. Review of ventilation strategies to reduce the risk of disease transmission in high occupancy buildings. Int J Thermofluid 2020:100045. 7-8.
[167] Coronavirus disease (COVID-19) weekly epidemiological update and weekly operational update available from. https://www.who.int/emergencies/diseases/novel-coronavirus-2019/situation-reports/.
[168] REHVA COVID-19 guidance document V4.1. Federation of European heating. Brussels: Ventilation and Air Conditioning Associations; 2020.
[169] COVID-19: Improv Indoor Ventil 2022. Available from: https://www.canada.ca/en/public-health/services/diseases/2019-novel-coronavirus-infection/prevention-risks/covid-19-improving-indoor-ventilation.html#a2.
[170] ASHRAE TC 5.5 practical guidance for epidemic operation of energy recovery ventilation (ERV) systems. Available from: https://www.ashrae.org/file%20library/technical%20resources/covid-19/practical-guidance-for-epidemic-operation-of-ervs.pdf; 2020.
[171] Dodoo A. Primary energy and economic implications of ventilation heat recovery for a multi-family building in a Nordic climate. J Build Eng 2020;31:101391.
[172] Ruud S. Återföring av föroreningar i roterande värmeväxlare-en litteraturstudie/ kunskapsöversikt. 1993.
[173] Wolkoff P, Nielsen GD. Organic compounds in indoor air-their relevance for perceived indoor air quality? Atmos Environ 2001;35(26):4407–17.
[174] Caron F, et al. Behaviour of individual VOCs in indoor environments: how ventilation affects emission from materials. Atmos Environ 2020;243:117713.

[175] Pei L-X, Lv Z-M, Zhang L-Z. Selective adsorption of a novel high selective desiccant for prospective use in heat and moisture recovery for buildings. Build Environ 2012;49:124–8.
[176] Seppänen O. VENTILATION, ENERGY AND INDOOR AIR QUALITY. 2002.
[177] Shahsavar, A. and M. Arıcı, Effect of glass cover on the energy and exergy performance of a combined system including a building integrated photovoltaic/thermal system and a sensible rotary heat exchanger. Int J Energy Res.