CHAPTER 26

AIR-TO-AIR ENERGY RECOVERY EQUIPMENT

AIR-TO-AIR energy recovery is the process of recovering heat and/or moisture between two airstreams at different temperatures and humidities. This process is important in maintaining acceptable indoor air quality (IAQ) while keeping energy costs low and reducing overall energy consumption and carbon dioxide emission. This chapter describes various technologies for air-to-air energy recovery. Thermal and economic performance, maintenance, and related operational issues are presented, with emphasis on energy recovery for ventilation.

Air-to-air energy recovery should be considered for every building in which energy is used to condition the air. This is consistent with ASHRAE's strategic plan to support a sustainable built environment, and aligns with the United Nations International Panel for Climate Change's call for action to reduce the emissions of carbon dioxide related to energy use.

Energy can be recovered either in its sensible (temperature only) or latent (moisture) form, or a combination of both from multiple sources. Sensible energy can be extracted, for example, from outgoing airstreams in dryers, ovens, furnaces, combustion chambers, and gas turbine exhaust gases to heat supply air. Units used for this purpose are called **sensible heat exchange devices** or **heat recovery ventilators (HRVs)**. Devices that transfer both heat and moisture are known as **energy** or **enthalpy devices** or **energy recovery ventilators (ERVs)**. HRVs and ERVs are available for commercial and industrial applications as well as for residential and small-scale commercial uses.

Air conditioners use significant energy to dehumidify moist airstreams. Excessive moisture in the air of a building can result in mold, allergies, and bacterial growth. ERVs can enhance dehumidification with packaged unitary air conditioners. Introducing outdoor or ventilation air is the primary means of diluting air contaminants to achieve acceptable indoor air quality. ERVs can cost-effectively provide large amounts of outdoor air to meet a building's minimum ventilation requirements as prescribed in ASHRAE *Standards* 62.1 and 62.2.

Types of ERVs include fixed-plate heat exchangers, rotary wheels, heat pipes, runaround loops, thermosiphons, and twin-tower enthalpy recovery loops. Performance is typically characterized by effectiveness; pressure drop, pumping, or fan power of fluids; cross flow (i.e., amount of air leakage from one stream to the other); and frost control (used to prevent frosting on the heat exchanger). Recovery efficiency, the ratio of output of a device to its input, is also often considered. In energy recovery ventilators, effectiveness refers to the ratio of actual energy or moisture recovered to the maximum possible amount of energy and/or moisture that can be recovered.

Fluid stream pressure drops because of the friction between the fluid and solid surface, and because of the geometrical complexity of the flow passages. Pumping or fan power is the product of the fluid volume flow rate and pressure drop. Economic factors such as cost of energy recovered and capital and maintenance cost (including pumping power cost) play a vital role in determining the economic feasibility of recovery ventilators for a given application.

The preparation of this chapter is assigned to TC 5.5, Air-to-Air Energy Recovery.

Table 1 Typical Applications for Air-to-Air Energy Recovery

Method	Application
Comfort-to-comfort	Residences
	Offices
	Classrooms
	Retail
	Bars and restaurants
	Swimming pools
	Locker rooms
	Operating rooms
	Nursing homes
	Animal ventilation
	Plant ventilation
	Smoking exhaust
Process-to-process	Dryers
and	Ovens
Process-to-comfort	Flue stacks
	Burners
	Furnaces
	Incinerators
	Paint exhaust
	Welding exhaust

1. APPLICATIONS

Air-to-air energy recovery systems may be categorized according to their application as (1) process-to-process, (2) process-to-comfort, or (3) comfort-to-comfort. Some typical air-to-air energy recovery applications are listed in Table 1.

In **comfort-to-comfort** applications, the energy recovery device lowers the enthalpy of the building supply air during warm weather and raises it during cold weather by transferring energy between the ventilation air supply and exhaust airstreams.

Air-to-air energy recovery devices for comfort-to-comfort applications may be sensible heat exchange devices (i.e., transferring sensible energy only) or energy exchange devices (i.e., transferring both sensible energy and moisture). These devices are discussed further in the section on Additional Technical Considerations.

When outdoor air humidity is low and the building space has an appreciable latent load, an ERV can recover sensible energy while possibly slightly increasing the latent space load because of water vapor transfer within the ERV. It is therefore important to determine whether the given application calls for HRV or ERV.

HRVs are suitable when outdoor air humidity is low and latent space loads are high for most of the year, and also for use with swimming pools, chemical exhaust, paint booths, and indirect evaporative coolers.

ERVs are suitable for applications in schools, offices, residences and other applications that require year-round economical preheating and/or precooling of outdoor supply air.

In **process-to-process** applications, heat is captured from the process exhaust stream and transferred to the process supply airstream. Equipment is available to handle process exhaust temperatures as high as 1600°F.

Process-to-process recovery devices generally recover only sensible heat and do not transfer latent heat, because moisture transfer is usually detrimental to the process. In cases involving condensable gases, less recovery may be desired to prevent condensation and possible corrosion.

In **process-to-comfort** applications, waste heat captured from process exhaust heats building makeup air during winter. Typical applications include foundries, strip-coating plants, can plants, plating operations, pulp and paper plants, and other processing areas with heated process exhaust and large makeup air volume requirements.

Although full recovery is usually desired in process-to-process applications, recovery for process-to-comfort applications must be modulated during warm weather to prevent overheating the makeup air. During summer, no recovery is required. Because energy is saved only in the winter and recovery is modulated during moderate weather, process-to-comfort applications save less energy annually than do process-to-process applications.

Process-to-comfort recovery devices generally recover sensible heat only and do not transfer moisture between airstreams.

2. BASIC HEAT OR HEAT AND WATER VAPOR TRANSFER RELATIONS

The second law of thermodynamics states that heat energy always transfers from a region of high temperature to one of low temperature. This law can be extended to say that mass transfer always occurs from a region of high vapor pressure to one of low vapor pressure. The conceptual energy recovery exchanger in Figure 1 facilitates this transfer (without mixing supply and exhaust airstreams) across a separating wall (shown by a thick horizontal line in Figure 1) made of a material that conducts heat and is permeable to water vapor. Heat is transferred when there is a difference in temperature between the two airstreams. Moisture is transferred when there is a difference in vapor pressure between the two airstreams.

On a typical summer day, supply air at temperature, humidity, or enthalpy of x_1 and mass flow rate m_s enters the ERV, while exhaust air from the conditioned space enters at conditions x_3 and m_3. Because conditions at x_3 are lower than conditions at x_1, heat and mass transfer from the supply airstream to the exhaust airstream because of differences in temperature and vapor pressures across the separating wall. Consequently, the supply air exit properties decrease, while those of the exhaust air increase.

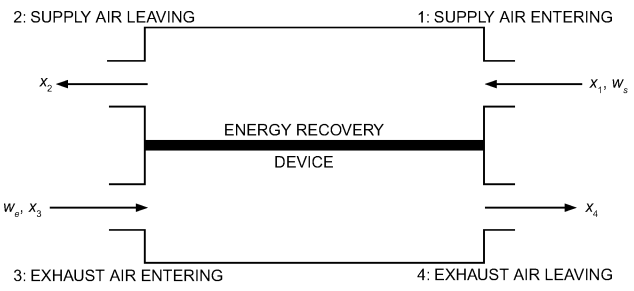

Fig. 1 Airstream Numbering Convention

Effectiveness

When mass flow rates, temperatures, and humidities of the inlets and outlets are known, the sensible, latent, or total effectiveness of the exchanger for those operating conditions can be determined.

Conversely, if the respective effectiveness of the exchanger at specific mass flow rates is known or rated, and the temperatures and humidities of the inlets are specified, it is possible to estimate conditions at the outlets, as long as confounding variables such as occurrence of condensation or internal leakages are insignificant.

The general definition of transfer effectiveness ε, on which ASHRAE *Standard* 84 bases its definitions of effectiveness, is

$$\varepsilon = \frac{\text{Actual transfer of moisture and/or energy}}{\text{Maximum possible transfer between airstreams}} \quad (1)$$

Referring to Figure 1, the gross sensible effectiveness ε_s or $\varepsilon_{sensible}$ of an energy recovery ventilator is given as

$$\varepsilon_{sensible} = \frac{\dot{m}_2(c_{p,1}T_1 - c_{p,2}T_2)}{\dot{m}_{min}(c_{p,1}T_1 - c_{p,3}T_3)} \quad (2a)$$

Referring to Figure 1, the gross latent effectiveness ε_L or ε_{latent} of an energy recovery ventilator is given as

$$\varepsilon_{latent} = \frac{\dot{m}_2(h_{fg,1}W_1 - h_{fg,2}W_2)}{\dot{m}_{min}(h_{fg,1}W_1 - h_{fg,3}W_3)} \quad (2b)$$

Referring to Figure 1, the gross total effectiveness ε_T or ε_{total} of an energy recovery ventilator is given as

$$\varepsilon_{total} = \frac{\dot{m}_2(h_1 - h_2)}{\dot{m}_{min}(h_1 - h_3)} \quad (2c)$$

where

$\dot{m}_n$ = mass flow rate at station n, lb_m/h
$\dot{m}_{min}$ = minimum of $\dot{m}_2$ and $\dot{m}_3$, lb_m/h
$c_{p,n}$ = specific heat of dry air at station n, $Btu/lb_m·°F$
h_{fg} = heat of vaporization of water, $Btu/lb_m·°F$
T_n = dry-bulb temperature at station n, °F
W_n = humidity at station n, $lb_{m\,w}/lb_{m\,da}$
h_n = enthalpy at station n, Btu/lb_m

Note that in Equations (2a), (2b), and (2c), the denominators express the theoretical maximum sensible, latent, or total energy transfer.

An additional figure of merit, which characterizes the reduction in heating or cooling load associated with the supply air, is the **enthalpy recovery ratio**, given by

$$\text{Enthalpy recovery ratio} = \frac{h_1 - h_2}{h_1 - h_3} \quad (3)$$

where h is enthalpy at station n, in Btu/lb.

Equation (3) bears a superficial resemblance to Equation (2c) which defines gross total effectiveness, but when the supply and exhaust mass flows through the exchanger are different, the two metrics may have different values. Using the enthalpy recovery ratio when designing HVAC systems for compliance with ASHRAE *Standard* 90.1 is detailed in the section on Performance Ratings.

The leaving supply air conditions and heat transfer rates easily can be estimated under ideal conditions: steady-state operation; no heat or moisture transfer between the heat exchanger and its surroundings; no cross-leakage; and no energy gains or losses from motors, fans, or frost control devices, with negligible condensation or frosting. Under such ideal conditions, the leaving supply temperature is

$$T_2 = \frac{\dot{m}_2 c_{p,1} T_1 - \varepsilon_{sensible} \dot{m}_{min}(c_{p,1} T_1 - c_{p,3} T_3)}{\dot{m}_2 c_{p,2}} \tag{4a}$$

and the exhaust air temperature is

$$T_4 = \frac{\dot{m}_2 c_{p,3} T_3 + \varepsilon_{sensible} \dot{m}_{min}(c_{p,1} T_1 - c_{p,3} T_3)}{\dot{m}_2 c_{p,2}} \tag{4b}$$

The leaving supply air humidity is

$$W_2 = \frac{\dot{m}_2 h_{fg,1} W_1 - \varepsilon_{latent} \dot{m}_{min}(h_{fg,1} W_1 - h_{fg,1} W_3)}{\dot{m}_2 h_{fg,2}} \tag{4c}$$

And the exhaust air humidity is

$$W_4 = \frac{\dot{m}_2 h_{fg,3} W_3 + \varepsilon_{latent} \dot{m}_{min}(h_{fg,1} W_1 - h_{fg,3} W_3)}{\dot{m}_2 h_{fg,2}} \tag{4d}$$

The leaving supply air enthalpy is

$$h_2 = h_1 - \varepsilon_{total} \frac{\dot{m}_{min}}{\dot{m}_s}(h_1 - h_3) \tag{4e}$$

And the exhaust air enthalpy is

$$h_4 = h_3 - \varepsilon_{total} \frac{\dot{m}_{min}}{\dot{m}_s}(h_1 - h_3) \tag{4f}$$

Use Equations (4e) and (4f) with caution. When the enthalpies of the entering airstreams are very close but the temperatures and humidities are not, it is possible for the calculated total effectiveness to be greater than or less than both of the sensible or latent effectivenesses, or even to be greater than 100% or less than 0%. For example, in some conditions, the supply airstream flowing through an ERV may gain heat energy ($+q_s$) from the adjoining stream but lose latent energy ($-q_L$) if it transfers the water vapor to the adjoining stream. Therefore, it is generally best to determine the expected enthalpy of leaving air by first calculating the expected temperature and humidity of the leaving air.

The sensible heat energy transfer q_s in the energy recovery exchanger can be estimated from

$$q_s = 60\dot{m}_2(c_{p,1} T_1 - c_{p,2} T_2) = 60 Q_2 \rho_2 (c_{p,1} T_1 - c_{p,2} T_2) \tag{5a}$$

where

q_s = sensible heat transfer, Btu/h
c_p = specific heat of air, Btu/lb$_m$·°F
Q_2 = volume flow rate of supply outlet air, cfm
ρ_2 = density of dry outlet supply air, lb/ft^3
t_1, t_2, t_3 = inlet and exit temperatures of supply inlet, supply outlet, and exhaust inlet airstreams, respectively, °F
$\dot{m}_2$ = mass flow rate of supply air inlet, lb/min

The latent heat transfer q_L in the energy recovery exchanger can be estimated from

$$q_L = 60\dot{m}_2(h_{fg,1} W_1 - h_{fg,2} W_2) = 60 Q_2 \rho_2 (h_{fg,1} W_1 - h_{fg,2} W_2) \tag{5b}$$

where

q_L = latent heat transfer, Btu/h
h_{fg} = enthalpy or heat of vaporization of water, Btu/lb
W_1, W_2, W_3 = inlet and exit humidity ratios of the supply inlet, supply outlet, and exhaust inlet airstreams, respectively, lb$_w$/lb$_{da}$

Similarly, humidity transfer q_w in the ERV can be estimated from

$$q_w = 60\dot{m}_2(W_1 - W_2) = 60 Q_2 \rho_2 (W_1 - W_2) \tag{5c}$$

where q_w is humidity transfer, in lb$_w$/h

In some cases, the total energy transfer q_T or q_{total} can be estimated from

$$q_{total} = 60\dot{m}_2(h_2 - h_1) = 60 Q_2 \rho_2 (h_1 - h_2) \tag{5d}$$

where q_h is total energy transfer, in Btu/h

However, in the somewhat unusual case when the supply airstream gains sensible energy but loses water content, q_{total} is best estimated from

$$q_{total} = q_s + q_L \tag{5e}$$

Heat or energy exchange effectiveness as defined in Equations (1), (2a), (2b), and (2c) is used to characterize each type of energy transfer in air-to-air exchangers. For a given set of inlet properties and flow rates, knowledge of each effectiveness allows the designer to calculate the sensible, latent, and total energy transfer rates using Equations (5a), (5b), and (5d), respectively. These effectiveness values can be determined either from measured test data or using correlations that have been verified in the peer-reviewed engineering literature. These correlations can also be used to predict energy transfer rates and outlet air properties for operating conditions different from those used for certification purposes. Predicting effectiveness for noncertified operating conditions using certified test data is the most common use of correlations for HVAC designs. Although correlations are not available for all types of air-to-air exchangers under all operating conditions, they are available for the most common types of air-to-air exchanger under operating conditions which have no condensation or frosting.

Rate of Energy Transfer

The rate of energy transfer depends on the operating conditions and the intrinsic characteristics of the energy exchanger, such as the geometry of the exchanger (parallel flow/counterflow/cross-flow, number of passes, fins), thermal conductivity of walls separating the streams, and permeability of walls to various gases. As in a conventional heat exchanger, energy transfer between the airstreams is driven by cross-stream dry-bulb temperature differences. Energy is also transported piggyback-style between the streams by cross-stream mass transfer, which may include air, gases, and water vapor. In another mode of energy transfer, water vapor condenses into liquid in one of the two airstreams of the exchanger. The condensation process liberates the latent heat of condensation, which is transferred to the other stream as sensible heat; this two-step process is also called **latent heat transfer**.

Latent energy transfer between airstreams occurs only when moisture is transferred from one airstream to another without condensation, thereby maintaining the latent heat of condensation. Once moisture has crossed from one airstream to the other, it may either remain in the vapor state or condense in the second stream, depending on the temperature of that stream.

Rotating-wheel and permeable-walled flat-plate energy recovery units are used because of their moisture exchange function. Some cross-stream mass transfer may also occur through leakage even when such transfer is unintended. This may alter exchanger performance from its design value, but for most HVAC applications with exhaust air from occupied spaces, small transfers to the supply air are not important. However, these transfers can and should be evaluated during design, and in many cases can be controlled.

Heat transfer differs in principle from mass transfer. Heat transfer only occurs when there is a temperature difference. In the case of air-to-air exchange between the supply and exhaust airflow, heat transfer by conduction and convection only occurs when there is a temperature difference between these airstreams. Remember the following facts about heat/mass exchanger performance:

- Effectiveness for moisture transfer may not equal the effectiveness for heat transfer.
- Total energy effectiveness may not equal either the sensible or latent effectiveness.

Net total energy transfer and effectiveness need careful examination when the direction of sensible (temperature-driven) transfer is opposite to that of latent (moisture or water vapor) transfer.

ERV performance is expressed by the magnitudes of pumping power and sensible, latent, or total energy recovered. The energy recovered is estimated from the exit temperatures or humidity ratios, which are directly related to the effectiveness. Effectiveness is a function of two parameters: the number of transfer units (NTU) and thermal flow capacity ratio C_r.

$$\text{NTU} = UA/60C_{min} \tag{6}$$

$$C_r = C_{min}/C_{max} \tag{7}$$

where

U = overall heat transfer coefficient, related to flow rates and dimensions of fluid flow path in heat exchanger, Btu/h·ft^2·°F
A = area of heat exchanger, ft^2
C_{min}, C_{max} = minimum and maximum capacity rates, expressed for sensible as $m_s c_{p,s}$ and $m_e c_{p,e}$

See Figure 8 for the variation of effectiveness with NTU for a rotary heat wheel. Variation of effectiveness is similar for plate exchangers.

Example 1. Inlet supply air enters an ERV with a flow rate of 9350 cfm at 95°F and 20% rh. Inlet exhaust air enters with a flow rate of 9050 cfm at 75°F and 50% rh. Assume that the energy exchanger was tested under ASHRAE *Standard* 84, which rated the sensible heat transfer effectiveness at 50% and the latent (water vapor) transfer effectiveness at 50%. With the simplifying assumptions that the specific heat of air is 0.24 Btu/lb·°F and the latent heat of vaporization is 1100 Btu/lb, determine the sensible, latent, and net energy transferred in the exchanger.

Solution:

From the psychrometric chart, the properties of air at 95°F and 20% rh are

$V_1 = 14.14$ ft^3/lb $\quad h_1 = 30.6$ Btu/lb $\quad w_1 = 0.0071$ lb/lb of dry air

and the properties of air at 75°F and 50% rh are

$V_3 = 13.68$ ft^3/lb $\quad h_3 = 28.15$ Btu/lb $\quad w_3 = 0.0093$ lb/lb of dry air

The mass flow rate at state 1 is obtained from

$$m_1 = \frac{Q_1}{V_1} = \frac{9350\ \text{ft}^3/\text{min}}{14.14\ \text{ft}^3/\text{lb}} = 660\ \text{lb/min}$$

Similarly, the mass flow rate at state 3 is obtained from

$$m_3 = \frac{Q_3}{V_3} = \frac{9050\ \text{ft}^3/\text{min}}{13.68\ \text{ft}^3/\text{lb}} = 660\ \text{lb/min}$$

Exit temperatures of the supply airstream can be obtained from Equation (4a):

$$t_2 = \frac{(660)(0.24)95 - 0.50(660)\{(0.24)95 - (0.24)75\}}{(660)0.24} = 85°\text{F}$$

The exit humidity of the supply airstream is found from Equation (4c):

$$\begin{aligned} w_2 &= \{(660)(1100)0.0071 - 0.50(660)[(1100)0.0071 \\ &\quad - (1100)0.0093]\}/[(660)1100] \\ &= 0.0080\ \text{lb}_w/\text{lb}_{da} \end{aligned}$$

The sensible heat transfer q_s in the exchanger is found from Equation (5a):

$$q_s = 60 \times 660(0.24 \times 95 - 0.24 \times 85) = 95{,}040\ \text{Btu/h}$$

The latent heat transfer q_L in the exchanger is found from Equation (5b):

$$q_L = 60 \times 660[(1100)(0.0071 - 0.0080)] = -39{,}204\ \text{Btu/h}$$

The net heat energy transfer q_s in the exchanger is found from Equation (5d):

$$q = q_s + q_L = 95{,}040 - 39{,}204 = 55{,}836\ \text{Btu/h}$$

If the incoming outdoor air conditions had been at 95°F and 14% rh, then the net energy gained by the exhaust airstream would have been zero.

Fan Power

It is important to estimate the fan power required to move the supply and exhaust airstream through the exchanger.

The fan power P_s required by the supply air is estimated from

$$P_s = \frac{Q_s\ \Delta p_s}{6356\eta_f} \tag{8}$$

The fan power P_e required by the exhaust air is estimated from

$$P_e = \frac{Q_e\ \Delta p_e}{6356\eta_f} \tag{9}$$

where

P_s = fan power for supply air, hp
P_e = fan power for exhaust air, hp
Q_s = supply volume flow rate, cfm
Q_e = exhaust volume flow rate, cfm
Δp_s = pressure drop of supply air caused by fluid friction, in. of water
Δp_e = pressure drop of exhaust air caused by fluid friction, in. of water
η_f = overall efficiency of fan and motor or product of fan and motor efficiencies

If pressure drops through the exchanger are not known at the operating conditions, they may be estimated from a standard pressure drop rating at a reference condition. Ideally, the density, viscosity, and Reynolds number of the air are known, which allows use of the following approach to estimate operating pressure drops.

The density and viscosity of air vary with temperature, and therefore pressure drop through the exchanger varies with temperature as well as flow rate. The variation of viscosity with temperature is given by the Sutherland law as

$$\frac{\mu}{\mu_o} = \left(\frac{T}{T_o}\right)^{3/2}\left(\frac{T_o + S}{T + S}\right) \tag{10}$$

where

μ = dynamic viscosity, lb/ft·min
μ_o = dynamic viscosity at the reference temperature, lb/ft·min
T = absolute temperature, °F
T_o = reference temperature, °F
S = constant = 198.7°F

The Reynolds number can be estimated from

$$Re_D = \frac{(\rho V)_{av} D_h}{\mu T} \quad (11)$$

where

ρ = air density, lb/ft³
V = average velocity in flow channels, fpm
D_h = hydraulic diameter of flow channels, ft
μ_T = dynamic viscosity at average temperature T, lb/ft·min

When flow through the exchanger is transitional or turbulent (i.e., when the Reynolds number Re_D for airflow through the exchanger is in the range $5 \times 10^3 \le Re_D \le 10^5$), then, treating air as an ideal gas, the pressure drop Δp at any temperature T is related to the pressure drop Δp_o at reference temperature T_o and is expressed as

$$\frac{\Delta p}{\Delta p_o} = \left(\frac{m}{m_o}\right)^{1.75}\left(\frac{T}{T_o}\right)^{1.375}\left(\frac{T_o + S}{T + S}\right)^{0.25} \quad (12)$$

where

Δp = pressure drop, in. of water
Δp_o = pressure drop at reference temperature, in. of water
m = mass flow rate at operating condition, lb/h
m_o = mass flow rate at rating condition, lb/h

For fully developed laminar flow through an energy exchanger, i.e. where $Re_D < 2000$, the corresponding dimensionless pressure drop relation similar to Equation (12) is given as

$$\frac{\Delta p}{\Delta p_o} = \left(\frac{m}{m_o}\right)\left(\frac{T}{T_o}\right)^{3/2}\left(\frac{1 + C/T_o}{1 + C/T}\right) \quad (13)$$

The equations above cannot be used in the case of (1) flow channels that are not reasonably smooth; (2) flow Reynolds numbers that are out of range; (3) significant exchanger fouling caused by condensation, frost, or dust; (4) excessive nonuniform property distributions inside the exchanger; or (5) significant pressure deformation of the flow channels (e.g., some plate cross-flow exchangers).

The total pumping power P of the ERV can be given as

$$P = P_s + P_e \quad (14)$$

3. TYPES OF AIR-TO-AIR HEAT EXCHANGERS

Ideal Air-to-Air Energy Exchange

An ideal air-to-air energy exchanger

- Allows temperature-driven heat transfer between participating airstreams
- Allows partial-pressure-driven moisture transfer between the two streams
- Minimizes cross-stream transfer of air, other gases (e.g., pollutants), biological contaminants, and particulates
- Optimizes energy recovery performance to minimize pressure drop while providing reasonable cost, dimensions, and weight

Heat transfer is an important energy recovery vehicle from airstreams that carry waste heat. The role of moisture transfer as an energy recovery process is less well known and merits explanation.

Consider an air-to-air energy exchanger operating in a hot, humid climate in a comfort air-conditioning application. If the energy exchanger exchanges heat but not moisture, it cools outdoor ventilation air as it passes through the exchanger to the indoor space. Heat flows from the incoming outdoor air to the outgoing (and cooler) exhaust air drawn from the indoor conditioned space. This does very little to mitigate the high humidity carried into the indoor space by the outdoor ventilation air and may even increase the relative humidity in the conditioned space, resulting in increased refrigeration and/or reheat to dehumidify the air and achieve acceptable comfort conditions. On the other hand, if the energy exchanger transfers both heat and moisture, the humid outdoor supply air transfers moisture to the less-humid inside exhaust air as the streams pass through the energy exchanger. The lower humidity of the entering ventilation air requires less energy input for comfort conditioning.

Fixed-Plate Heat Exchangers

Plate exchangers are available in many configurations, materials, sizes, and flow patterns. Many have modules that can be arranged to handle almost any airflow, effectiveness, and pressure drop requirement. Plates are formed with spacers or separators (e.g., ribs, dimples, ovals) constructed into the plates or with external separators (e.g., supports, braces, corrugations). Airstream separations are sealed by folding, multiple folding, gluing, cementing, welding, or any combination of these, depending on the application and manufacturer. Ease of access for examining and cleaning heat transfer surfaces depends on the configuration and installation.

Heat transfer resistance through the plates is small compared to the airstream boundary layer resistance on each side of the plates. Heat transfer efficiency is not substantially affected by the heat transfer coefficient of the plates. Aluminum is the most popular plate construction material because of its nonflammability and durability. Polymer plate exchangers may improve heat transfer by causing some turbulence in the channel flow, and are popular because of their corrosion resistance and cost-effectiveness. Steel alloys are used for temperatures over 400°F and for specialized applications where cost is not a key factor. Plate exchangers normally conduct sensible heat only; however, water-vapor-permeable materials, such as treated paper and microporous polymeric membranes, may be used to transfer moisture, thus providing total (enthalpy) energy exchange.

Most manufacturers offer modular plate exchangers. Modules range in capacity from 25 to 10,000 cfm and can be arranged into configurations exceeding 100,000 cfm. Multiple sizes and configurations allow selections to meet space and performance requirements.

Plate spacing ranges from 0.1 to 0.5 in., depending on the design and application. Heat is transferred directly from the warm airstreams through the separating plates into the cool airstreams. Usually design, construction, and cost restrictions result in the selection of cross-flow exchangers, but additional counterflow patterns can increase heat transfer effectiveness.

Normally, both latent heat of condensation (from moisture condensed as the temperature of the warm exhaust airstream drops below its dew point) and sensible heat are conducted through the separating plates into the cool supply airstream. Thus, energy is transferred but moisture is not. Recovering 80% or more of the available waste exhaust heat is possible.

Fixed-plate heat exchangers can achieve high sensible heat recovery and total energy effectiveness because they have only a primary heat transfer surface area separating the airstreams and are therefore not inhibited by the additional secondary resistance (e.g., pumping liquid, in runaround systems or transporting a heat transfer medium) inherent in some other exchanger types. In a cross-flow arrangement (Figure 2), they usually do not have sensible effectiveness greater than 75% unless two devices are used in series as shown in Figure 3.

One advantage of the plate exchanger is that it is a static device with little or no leakage between airstreams. As velocity increases, the pressure difference between the two airstreams increases. High differential pressures may deform the separating plates and, if excessive, can permanently damage the exchanger, significantly reducing

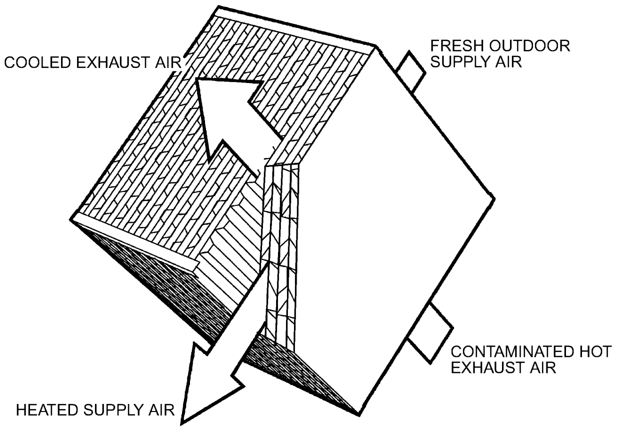

Fig. 2 Fixed-Plate Cross-Flow Heat Exchanger

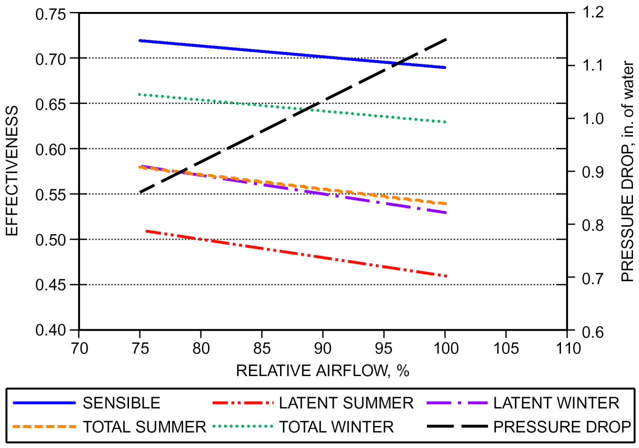

Fig. 3 Variation of Pressure Drop and Effectiveness with Airflow Rates for Membrane Plate Exchanger

the airflow rate on the low-pressure side as well as the effectiveness and possibly causing excessive air leakage. This is not normally a problem because differential pressures in most applications are less than 4 in. of water. In applications requiring high air velocities, high static pressures, or both, plate exchangers built to withstand these conditions are available and should be specified.

Most sensible plate exchangers have condensate drains, which remove condensate and also wastewater in water-wash systems. Heat recovered from a high-humidity exhaust is better returned to a building or process by a sensible heat exchanger rather than an enthalpy exchanger if humidity transfer is not desired.

Frosting can be controlled by preheating incoming supply air, bypassing part of the incoming air, recirculating supply air through the exhaust side of the exchanger, or temporarily interrupting supply air while maintaining exhaust. However, frost on cross-flow heat exchangers is less likely to block the exhaust airflow completely than with other types of exchangers. Generally, frost should be avoided unless a defrost cycle is included.

Fixed-plate heat exchangers can be made from permeable membranes designed to maximize moisture and energy transfer between airstreams while minimizing air transfer. Suitable permeable microporous membranes for this emerging technology include cellulose, polymers, and other synthetic materials such as hydrophilic electrolyte. Hydrophilic electrolytes are made from, for example, sulphonation chemistry techniques and contain charged ions that

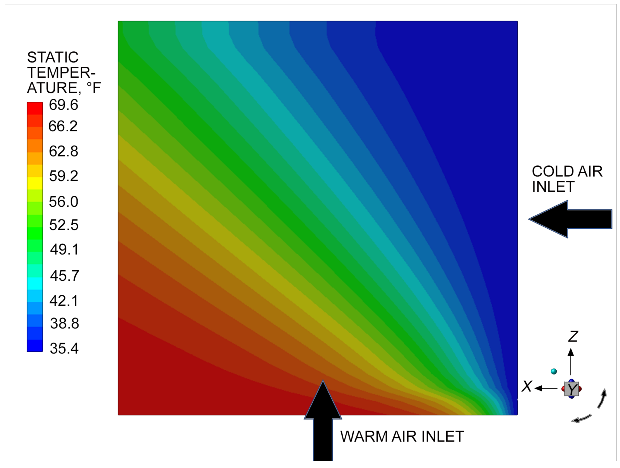

Fig. 4 Typical Temperature Stratification at Outlets of Cross-Flow Heat Exchanger

attract polar water molecules; adsorption and desorption of water occur in vapor state.

Airstreams exiting a cross-flow plate exchanger display temperature stratification when there is a temperature difference between the two airstreams (Figure 4). Enthalpy plate exchangers will also display humidity stratification. This should be considered if the temperature of one of these airstreams is to be measured (e.g., to control operation of downstream conditioning equipment or of a defrost mechanism).

Heat exchanger effectiveness depends heavily on the airflow direction and pattern of the supply and exhaust airstreams. Parallel-flow exchangers (Figure 5A), in which both airstreams move along heat exchange surfaces in the same direction, have a theoretical maximum effectiveness of 50%. Counterflow exchangers (Figure 5B), in which airstreams move in opposite directions, can have a theoretical effectiveness approaching 100%, but typical units have a lower effectiveness. Theoretical effectiveness for cross-flow heat exchangers is somewhat lower than for counterflow, and typical units have effectiveness of 50 to 70% (Figure 5C) and 60 to 85% for multiple-pass exchangers (Figure 5D).

In practice, construction limitations favor designs that use transverse flow (or cross-flow) over much of the heat exchange surface (Figures 5C and 5D).

Rotary Air-to-Air Energy Exchangers

A rotary air-to-air energy exchanger, or **rotary enthalpy wheel**, has a revolving cylinder filled with an air-permeable medium having a large internal surface area. Adjacent supply and exhaust airstreams each flow through half the exchanger in a counterflow pattern (Figure 6). Heat transfer media may be selected to recover sensible heat only or total (sensible plus latent) heat.

Sensible heat is transferred as the medium picks up and stores heat from the hot airstream and releases it to the cold one. Latent heat is transferred as the medium adsorbs water vapor from the higher-humidity airstream and desorbs moisture into the lower-humidity airstream, driven in each case by the vapor pressure difference between the airstream and energy exchange medium. Thus, the moist air is dried while the drier air is humidified. In total heat transfer, both sensible and latent heat transfer occur simultaneously. Sensible-only wheels (not coated with desiccant) can also transfer water via a mechanism of condensation and reevaporation driven by dew point and vapor pressure; the effectiveness varies strongly with conditions. Because rotary exchangers have a counterflow configuration and normally use small-diameter flow passages, they are quite compact and can achieve high transfer effectiveness.

Air contaminants, dew point, exhaust air temperature, and supply air properties influence the choice of materials for the casing, rotor structure, and medium of a rotary energy exchanger. Aluminum, steel, and polymers are the usual structural, casing, and rotor materials for normal comfort ventilating systems. Exchanger media are fabricated from metal, mineral, or synthetic materials and provide either random or directionally oriented flow through their structures.

Random-flow media are made by knitting wire into an open woven cloth or corrugated mesh, which is layered to the desired configuration. Aluminum mesh is packed in pie-shaped wheel segments. Stainless steel and monel mesh are used for high-temperature and corrosive applications. These media should only be used with clean, filtered airstreams because they plug easily. Random-flow media also require a significantly larger face area than directionally oriented media for a given airflow and pressure drop.

Directionally oriented media are available in various geometric configurations. The most common consist of small (0.06 to 0.08 in.) air passages parallel to the direction of airflow. Air passages are very similar in performance regardless of their shape (triangular, hexagonal, parallel plate, or other). Aluminum foil, paper, plastic, and synthetic materials are used for low and medium temperatures. Stainless steel and ceramics are used for high temperatures and corrosive atmospheres.

Media surface areas exposed to airflow vary from 100 to 1000 ft^2/ft^3, depending on the type of medium and physical configuration. Media may also be classified by their ability to recover sensible heat only or total heat. Media for sensible heat recovery are made of aluminum, copper, stainless steel, and monel. Media for total heat recovery can be from any of a number of materials and treated with a desiccant (typically zeolites, molecular sieves, silica gels, activated alumina, titanium silicate, synthetic polymers, lithium chloride, or aluminum oxide) to have specific moisture recovery characteristics.

Cross-Leakage. Cross-leakage (mixing between supply and exhaust airstreams) can occur in all rotary energy exchangers by two mechanisms: carryover and seal leakage. Cross leakage can be reduced by placing the blowers so that they promote leakage of outdoor air to the exhaust airstream. A purge section also can be installed on the heat exchanger to reduce cross leakage.

In many applications, recirculating some air is not a concern. However, critical applications such as hospital operating rooms, laboratories, and cleanrooms require stringent control of carryover. Carryover can be reduced to less than 0.1% of the exhaust airflow with a purge section but cannot be completely eliminated.

The theoretical carryover of a wheel without a purge section is directly proportional to the speed of the wheel and the void volume of the medium (75 to 95% void, depending on type and configuration). For example, a 10 ft diameter, 8 in. deep wheel with a 90% void volume operating at 14 rpm has a carryover volumetric flow of

$$\pi(10/2)^2(8/12)(0.9)(14) = 660 \text{ cfm}$$

If the wheel is handling a 20,000 cfm balanced flow, the percentage carryover is

$$\frac{660}{20,000} \times 100 = 3.3\%$$

The exhaust fan, which is usually located at the exit of the exchanger, should be sized to include leakage and purge airflows.

Control. Two control methods are commonly used to regulate wheel energy recovery. In **supply** or **exhaust air bypass** control, the amount of supply air allowed to pass through the wheel establishes the supply air temperature. An air bypass damper, controlled by a wheel supply air discharge temperature sensor, regulates the proportion of supply air permitted to bypass the exchanger.

The second method regulates the energy recovery rate by varying wheel rotational speed. The most frequently used **variable-speed drives** are (1) a silicon controlled rectifier (SCR) with variable-speed dc motor, (2) a constant-speed AC motor with hysteresis coupling, and (3) an AC frequency inverter with an AC induction motor. The dehumidification capacity and reheating effectiveness of a energy wheel can be varied by changing the wheel speed (Figure 7A) or by bypassing air around the wheel (Figure 7B) (Moffitt 2010). Figure 7A is a typical capacity curve for varying the wheel's rotation speed when the outdoor air is cooler and drier than the exhaust air; when outdoor air conditions are cooler and more humid, capacity may increase at low speeds (Simonson et al. 2000).

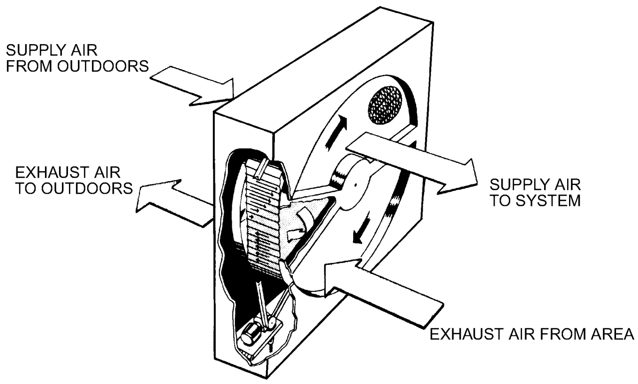

Fig. 6 Rotary Air-to-Air Energy Exchanger

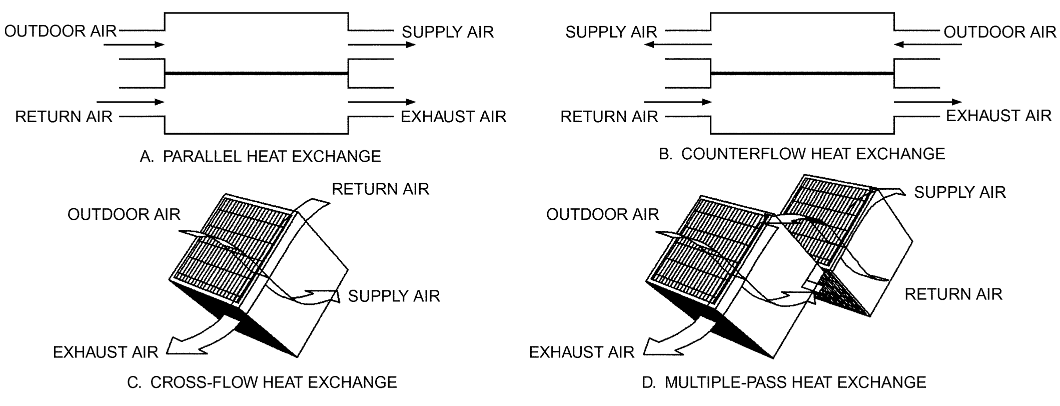

Fig. 5 Plate Heat or Heat and Mass Exchanger Airflow Configurations

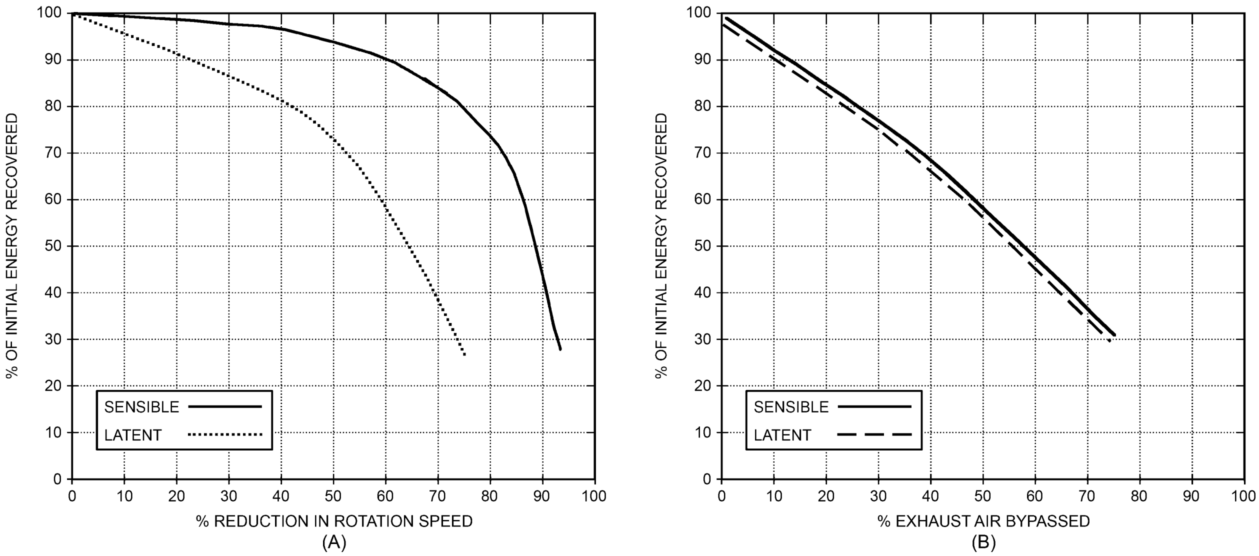

Fig. 7 Latent and Sensible Effectiveness Versus (A) Wheel Speed and (B) Bypassed Air
(Moffitt 2010)

Figure 8 shows the effectiveness ε, sensible heat transfer only, with balanced airflow, convection/conduction ratio less than 4, and no leakage or cross-flow, of a regenerative counterflow heat wheel versus number of transfer units (NTU). This simple example of a regenerative wheel also shows that regenerative counterflow rotary effectiveness increases with wheel speed (C_r is proportional to wheel speed), but there is no advantage in going beyond C_r/C_{min} = 5 because the carryover of contaminants increases with wheel speed. See Kays and Crawford (1993) or Shah (1981) for details.

Rotary energy or enthalpy wheels are more complex than heat wheels, but recent research has characterized their behavior using laboratory and field data (Johnson et al. 1998).

A dead band control, which stops or limits the exchanger, may be necessary when no recovery is desired (e.g., when outdoor air temperature is higher than the required supply air temperature but below the exhaust air temperature). When outdoor air temperature is above the exhaust air temperature, the equipment operates at full capacity to cool incoming air. During very cold weather, it may be necessary to heat the supply air, stop the wheel, or, in small systems, use a defrost cycle for frost control.

Rotary enthalpy wheels require little maintenance and tend to be self-cleaning because the airflow direction is reversed for each rotation of the wheel. The following maintenance procedures ensure best performance:

- Clean the medium when lint, dust, or other foreign materials build up, following the manufacturer's instructions.
- Maintain drive motor and train according to the manufacturer's recommendations. Speed-control motors that have commutators and brushes require more frequent inspection and maintenance than induction motors. Brushes should be replaced, and the commutator should be periodically turned and undercut.
- Inspect wheels regularly for proper belt or chain tension.
- Refer to the manufacturer's recommendations for spare and replacement parts.

Coil Energy Recovery (Runaround) Loops

A typical coil energy recovery loop (Figure 9) places extended-surface, finned-tube water coils in the supply and exhaust airstreams of a building or process. The coils are connected in a closed loop by

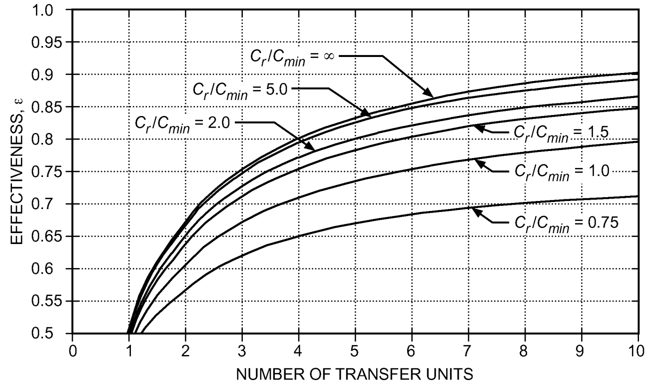

Fig. 8 Effectiveness of Counterflow Regenerator as a Function of NTU
(Shah 1981)

counterflow piping through which an intermediate heat transfer fluid (typically water or antifreeze solution) is pumped.

Moisture must not freeze in the exhaust coil air passage. A dual-purpose, three-way temperature control valve prevents the exhaust coil from freezing. The valve is controlled to maintain the temperature of solution entering the exhaust coil at 40°F or above. This condition is maintained by bypassing some of the warmer solution around the supply air coil. The valve can also ensure that a prescribed air temperature from the supply air coil is not exceeded.

Coil energy recovery loops are highly flexible and well suited to renovation and industrial applications. The loop accommodates remote supply and exhaust ducts and allows simultaneous transfer of energy between multiple sources and uses. An expansion tank must be included to allow fluid expansion and contraction. A closed expansion tank minimizes oxidation when ethylene glycol is used.

Standard finned-tube water coils may be used; however, these need to be selected using an accurate simulation model if high effectiveness and low costs are needed (Johnson et al. 1995). Integrating runaround loops in buildings with variable loads to achieve maximum benefits may require combining the runaround simulation with building energy simulation (Dhital et al. 1995). Manufacturers' design curves and performance data should be used when selecting coils, face velocities, and pressure drops, but only when the design data are for the same temperature and operating conditions as in the runaround loop.

When testing chilled-water coils for certification, AHRI *Standard* 410 requires testing in the range of 35 to 55°F. Care should be taken when using manufacturers' selection programs for sizing coils for runaround loops. It is common to use a cooling coil design for these systems, but average water temperatures can vary outside the range for which these coils are typically used. For example, in commercial comfort conditioning, average water temperatures in a runaround loop can range from 25°F (–20°F outdoors, 70°F return) to 80°F (95°F outdoors, 75°F return) in a single application, whereas cooling coils are normally applied using a steady chilled-water supply designed from a narrower range.

The coil energy recovery loop cannot transfer moisture between airstreams; however, indirect evaporative cooling can reduce the exhaust air temperature, which significantly reduces cooling loads. For the most cost-effective operation, with equal airflow rates and no condensation, typical effectiveness values range from 45 to 65%. The highest effectiveness does not necessarily give the greatest net life-cycle cost saving.

Typically, the sensible heat effectiveness of a coil energy recovery loop is independent of the outdoor air temperature. However, when the capacity is controlled, the sensible heat effectiveness decreases.

Coil energy recovery loops use coils constructed to suit their environment and operating conditions. For typical comfort-to-comfort applications, standard coil construction usually suffices. In process-to-process and process-to-comfort applications, the effect of high temperature, condensable gases, corrosives, and contaminants on the coil(s) must be considered. Above 400°F, special construction may be required to ensure a permanent fin-to-tube bond. The effects of condensable gases and other adverse factors may require special coil construction and/or coatings. Chapters 23 and 27 discuss the construction and selection of coils in more detail.

Complete separation of the airstreams eliminates cross-contamination between the supply and exhaust air.

Coil energy recovery loops require little maintenance. The only moving parts are the circulation pump and three-way control valve. However, to ensure optimum operation, the air should be filtered, the coil surface cleaned regularly, the pump and valve maintained, and the transfer fluid refilled or replaced periodically. Fluid manufacturers or their representatives should be contacted for specific recommendations.

The thermal transfer fluid selected for a closed-loop exchanger depends on the application and on the temperatures of the two airstreams. An inhibited ethylene glycol solution in water is common when freeze protection is required. These solutions break down to an acidic sludge at temperatures above 275°F. If freeze protection is needed and exhaust air temperatures exceed 275°F, a nonaqueous synthetic heat transfer fluid should be used. Heat transfer fluid manufacturers and chemical suppliers should recommend appropriate fluids.

Heat Pipe Heat Exchangers

Figure 10 shows a typical heat pipe assembly. Hot air flowing over the evaporator end of the heat pipe vaporizes the working fluid. A vapor pressure gradient drives the vapor to the condenser end of the heat pipe tube, where the vapor condenses, releasing the latent energy of vaporization (Figure 11). The condensed fluid is wicked or flows back to the evaporator, where it is revaporized, thus completing the cycle. Thus the heat pipe's working fluid operates in a closed-loop evaporation/condensation cycle that continues as long as there is a temperature difference to drive the process. Using this mechanism, the heat transfer rate along a heat pipe is up to 1000 times greater than through copper (Ruch 1976).

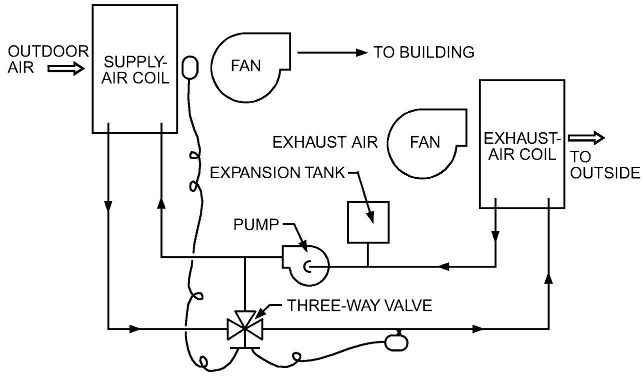

Fig. 9 Coil Energy Recovery Loop

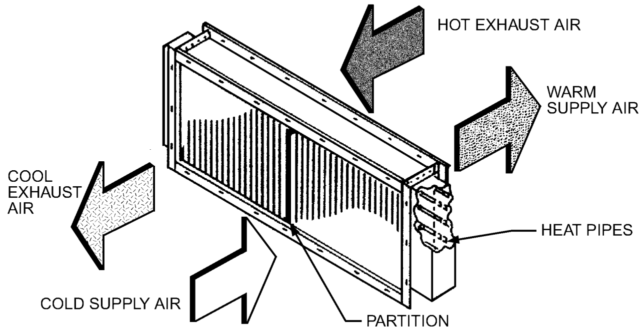

Fig. 10 Heat Pipe Assembly

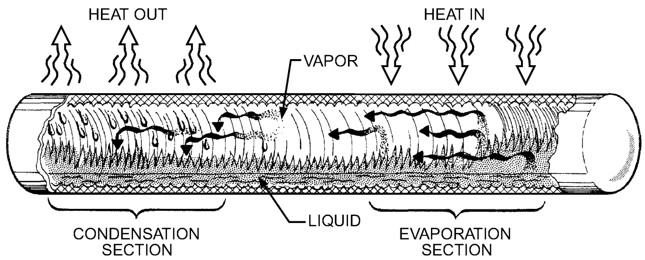

Fig. 11 Heat Pipe Operation

Energy transfer in heat pipes is often considered isothermal. However, there is a small temperature drop through the tube wall, wick, and fluid medium. Heat pipes have a finite heat transfer capacity that is affected by factors such as wick design, tube diameter, working fluid, and tube (heat pipe) orientation relative to horizontal. In current designs, a wick is replaced by circumferential grooves that facilitate capillary-action flow of condensed refrigerant back to the evaporator section.

HVAC systems use copper or aluminum heat pipe tubes with aluminum or copper fins. Fin designs include continuous corrugated plate, continuous plain, and spiral. Modifying fin design and tube spacing changes pressure drop at a given face velocity.

For process-to-comfort applications with large temperature changes, tubes and fins are usually constructed of the same material to avoid problems with different thermal expansions of materials. Heat pipe heat exchangers for exhaust temperatures below 425°F are most often constructed with aluminum or copper tubes and fins. Protective coatings allow inexpensive aluminum to replace exotic metals in corrosive atmospheres; these coatings have a minimal effect on thermal performance.

Heat pipe heat exchangers for use above 425°F are generally constructed with steel tubes and fins. The fins are often aluminized to prevent rusting. Composite systems for special applications may be created by assembling units with different materials and/or different working fluids.

Selecting the proper working fluid for a heat pipe is critical to long-term operation. The working fluid should have high latent heat of vaporization, a high surface tension, and a low liquid viscosity over the operating range; it must be thermally stable at operating temperatures. Decomposition of the thermal fluids can form noncondensable gases that deteriorate performance. For low-temperature applications, gases such as helium can be used as working fluid; for moderate temperatures, commercial refrigerants and liquids such as water can be used; and for high temperatures, liquid metals such as sodium or mercury can be used.

Heat pipe heat exchangers typically have no cross-contamination for pressure differentials between airstreams of up to 50 in. of water. A vented double-wall partition between the airstreams can provide additional protection against cross contamination. If an exhaust duct is attached to the partition space, any leakage is usually withdrawn and exhausted from the space between the two ducts.

Heat pipe heat transfer capacity depends on design and orientation. Figure 12 shows a typical effectiveness curves for various face velocities and rows of tubes. As the number of rows increases, effectiveness increases at a decreasing rate. For example, doubling the number of rows of tubes in a 60% effective heat exchanger increases the effectiveness to 75%. The effectiveness of a counterflow heat pipe heat exchanger depends on the total number of rows such that two units in series yield the same effectiveness as a single unit of the same total number of rows. Series units are often used to facilitate handling, cleaning, and maintenance. Effectiveness also depends on outdoor air temperature and the ratio of mass flow rates of the airstreams. Typically, heat capacity in the cooling season increases with a rise in outdoor air temperature. It has an opposite effect during the heating season. Effectiveness typically increases with the ratio of mass flow rates of the fluids (flow rate of the fluid with warmer entering temperature over that of cooler entering fluid temperature).

The heat transfer capacity of a heat pipe increases roughly with the square of the inside diameter of the pipe. For example, at a given tilt angle, a 1 in. inside diameter heat pipe will transfer roughly 2.5 times as much energy as a 5/8 in. inside diameter pipe. Consequently, heat pipes with large diameters are used for larger-airflow applications and where level installation is required to accommodate both summer and winter operation.

Heat transfer capacity is virtually independent of heat pipe length, except for very short heat pipes. For example, a 4 ft long heat pipe has approximately the same capacity as an 8 ft pipe. Because the 8 ft heat pipe has twice the external heat transfer surface area of the 4 ft pipe, it will reach its capacity limit sooner. Thus, in some applications, it is more difficult to meet capacity requirements as the heat pipes become longer. A system can be reconfigured to a taller face height and more numerous but shorter heat pipes to yield the same airflow face area while improving system performance. Also, under limiting conditions, adding rows increases performance more proportionally. In typical HVAC comfort applications, the heat pipe capacity is not reached, and longer heat pipes can be used to recover more energy. The limit for a level (two-season) 1/2 in.-diameter heat pipe, using R-410A and six rows deep, is 200 in. in most HVAC comfort applications.

The selection of fin design and spacing should be based on the dirtiness of the two airstreams and the resulting cleaning and maintenance required. For HVAC applications, 11 to 14 fins per in. (fpi) is common. Wider spacing (8 to 10 fpi) is usually used for industrial applications. Plate-fin heat pipe heat exchangers can easily be constructed with different fin spacing for the exhaust and supply airstreams, allowing wider fin spacing on the dirty exhaust side. This increases design flexibility where pressure drop constraints exist and also prevents deterioration of performance caused by dirt build-up on the exhaust side surface.

Changing the tilt of a heat pipe controls the amount of heat it transfers. Operating the heat pipe on a slope with the hot end below (or above) the horizontal improves (or retards) condensate flow back to the evaporator end of the heat pipe. This feature can be used to regulate the effectiveness of the heat pipe heat exchanger (Guo et al. 1998). Introducing the action of gravity makes the device a hybrid heat pipe/thermosiphon.

Tilt control is achieved by pivoting the exchanger about the center of its base and attaching a temperature-controlled actuator to one end of the exchanger (Figure 13). Pleated flexible connectors attached to the ductwork allow freedom for the small tilting movement of only a few degrees.

Tilt control may be desired

- To change from supply air heating to supply air cooling (i.e., to reverse the direction of heat flow) during seasonal changeover
- To modulate effectiveness to maintain desired supply air temperature (often required for large buildings to avoid overheating air supplied to the interior zone)
- To decrease effectiveness to prevent frost formation at low outdoor air temperatures (with reduced effectiveness, exhaust air leaves the

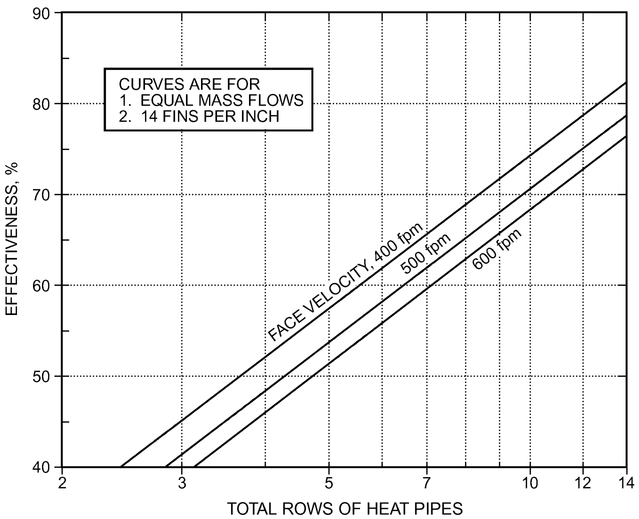

Fig. 12 Heat Pipe Exchanger Effectiveness

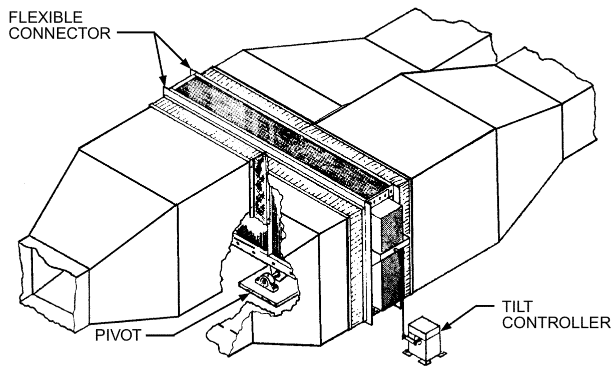

Fig. 13 Heat Pipe Heat Exchanger with Tilt Control

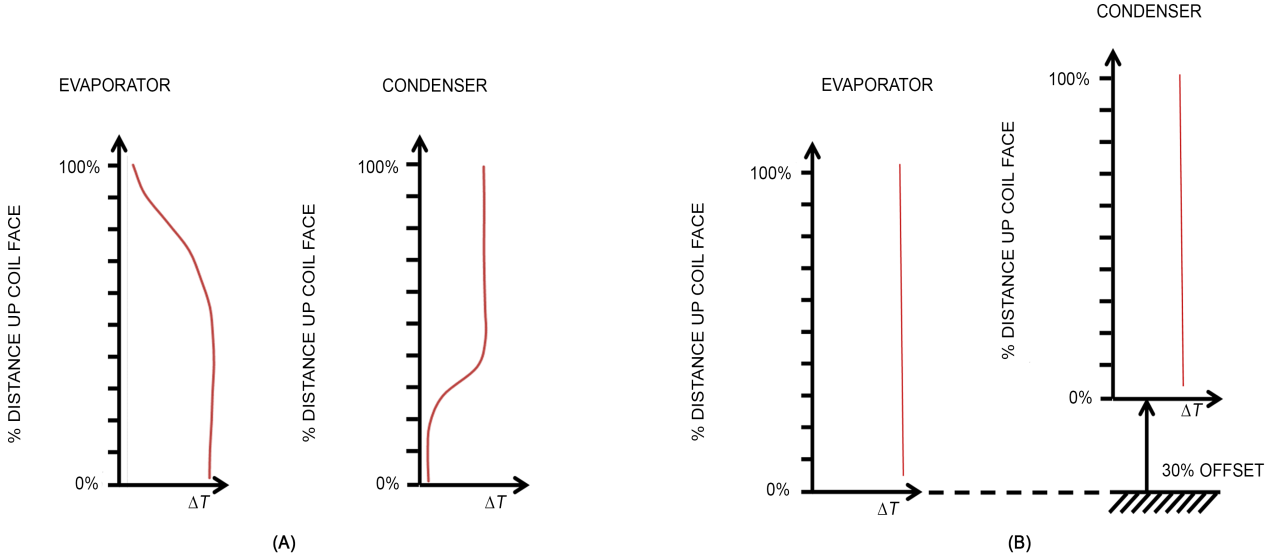

Fig. 16 Heat Recovered Across Evaporator and Condenser Coils for (A) Level Thermosiphon and (B) Thermosiphon with Elevated Condenser

unit at a warmer temperature and stays above frost-forming conditions)

Other devices, such as face-and-bypass dampers and preheaters, can also be used to control the rate of heat exchange.

Recent design improvements allow bidirectional heat transfer in heat pipes. Some heat pipe manufacturers have eliminated the need for tilting for capacity control or seasonal changeover. Once installed, the unit is removed only for routine maintenance. Capacity and frost control can also be achieved through bypassing airflow over the heat pipes, as for air coils. Bypassing airflow is the preferred method of control in applications where the average (annual) recovery efficiency ratio (RER) is important, because the reduced airflow through the heat pipe results in reduced air pressure losses during periods of modulation.

Example 2. Sensible Heat Energy Recovery in a Heat Pipe. Outdoor air at 50°F enters a six-row heat pipe with a flow rate of 660 lb/min and a face velocity of 500 fpm. Exhaust air enters the heat pipe with the same velocity and flow rate but at 75°F. The pressure drop across the heat pipe is 0.6 in. of water. The supply air density is 0.08 lb/ft^3. The efficiency of the electric motor and the connected fan are 90 and 75%, respectively. Assuming the performance characteristics of the heat pipe are as shown in Figure 18, determine the sensible effectiveness, exit temperature of supply air to the space, energy recovered, and power supplied to the fan motor.To simplify the calculations, assume the specific heat is a constant of 0.24 Btu/lb·°F.

Solution:

From Figure 12, at face velocity of 500 fpm and with six rows, the effectiveness is about 58%. Because the mass flow rate of the airstreams is the same and assuming their specific heat of 0.24 Btu/lb·°F is the same, then the exit temperature of the supply air to the space can be obtained from Equation (4a):

$$T_2 = \frac{(660)(0.24)50 - 0.58(660)(0.24)(50 - 75)}{(660)(0.24)} = 64.5\text{°F}$$

The sensible energy recovered can be obtained from Equation (5a), incorporating the simplification of specific heat to a single value, as

$$q_s = (60)(660)(0.24)(64.5 - 50) = 139{,}200 \text{ Btu/h}$$

The supply air fan power can be obtained from Equation (8) as

$$Q_s = \frac{660}{0.08} = 8250 \text{ ft}^3\text{/min}$$

$$P_s = [(8250)(0.6)]/[(6356)(0.9)(0.75)] = 1.15 \text{ hp}$$

Because there are two airstreams, neglecting the difference in the air densities of the airstreams, the total pumping power of the heat pipe is twice the above value (i.e., 2.3 hp).

Thermosiphon Heat Exchangers

Two-phase thermosiphon heat exchangers are sealed systems that consist of an evaporator, a condenser, interconnecting piping, and an intermediate working fluid that is present in both liquid and vapor phases. Two types of thermosiphon are used: a sealed tube (Figure 14) and a coil type (Figure 15). In the **sealed-tube thermosiphon**, the evaporator and the condenser are usually at opposite ends of a bundle of straight, individual thermosiphon tubes, and the exhaust and supply ducts are adjacent to each other (this arrangement is similar to that in a heat pipe system). In **coil-type thermosiphons**, evaporator and condenser coils are installed independently in the ducts and are interconnected by the working fluid piping (this configuration is similar to that of a coil energy recovery loop).

A thermosiphon is a sealed system containing a two-phase working fluid. Because part of the system contains vapor and part contains liquid, the pressure in a thermosiphon is governed by the liquid temperature at the liquid/vapor interface. If the surroundings cause a temperature difference between the regions where liquid and vapor interfaces are present, the resulting vapor pressure difference causes vapor to flow from the warmer to the colder region. The flow is sustained by condensation in the cooler region and by evaporation in the warmer region. The condenser and evaporator must be oriented so that the condensate can return to the evaporator by gravity (Figures 14 and 15). For coil-type thermosiphon loops (Figure 15), static liquid level or refrigerant charge has a significant effect on system performance. The static charge should be optimized: a low static charge leads to dryout in the evaporator and reduced system performance, and a high charge suppresses condensation in the condenser. When the coils are on the same vertical level, as in Figure 15B, a certain amount of suppressed condensation in the condenser is inevitable because a certain static liquid head is required in the bottom of the condenser to drive liquid through the loop into the evaporator (Wei Qu 2010). Figure 16A shows a typical temperature change profile

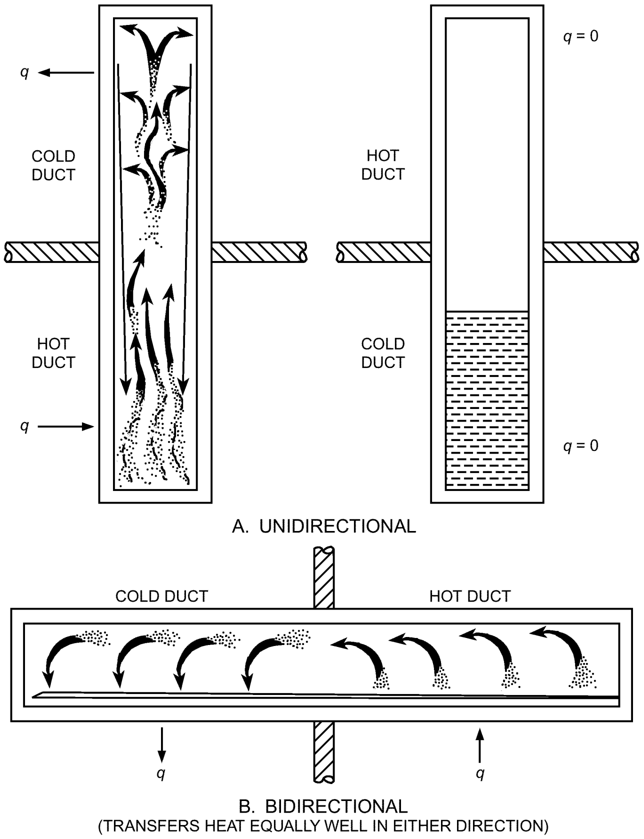

Fig. 14 Sealed-Tube Thermosiphons

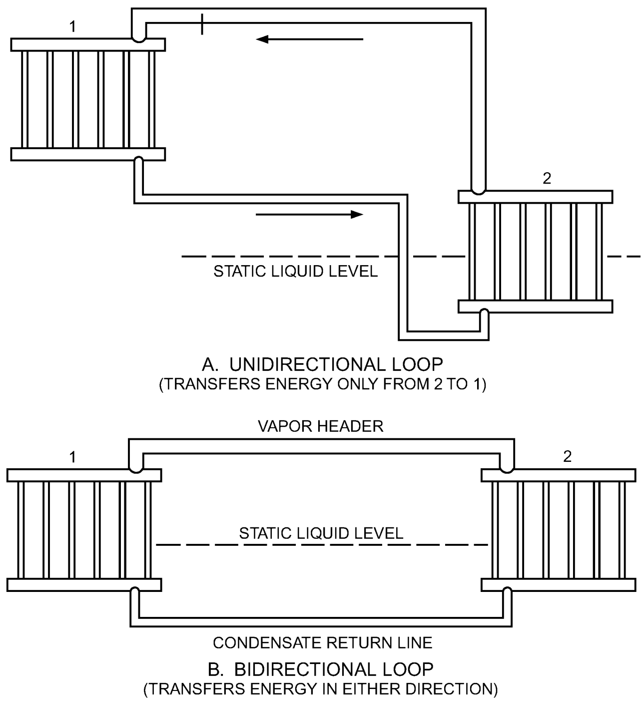

Fig. 15 Coil-Type Thermosiphon Loops

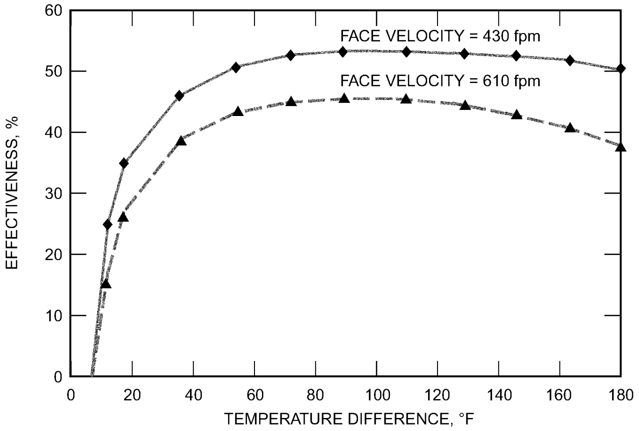

Fig. 17 Eight-Row, 12 fpi Unidirectional System; Static Charge of 80%
(Mathur and McDonald 1986)

along the vertical section of a condenser, where lower ΔT indicates loss of energy recovery potential, and the thermosiphon manufacturer should quote performance as an average across the whole airstream. Wei Qu (2010) found that the condenser need not be placed much higher than the evaporator to reach full potential; Figure 16B shows that raising the condenser 25 to 30% of the coil face height above the evaporator can result in negligible variation in the temperature profile.

In thermosiphon systems, a temperature difference and gravity force are required for the working fluid to circulate between the evaporator and condenser. As a result, thermosiphons may be designed to transfer heat equally in either direction (bidirectional), in one direction only (unidirectional), or in both directions unequally.

Although similar in form and operation to heat pipes, thermosiphon tubes are different in two ways: (1) they have no wicks and hence rely only on gravity to return condensate to the evaporator, whereas heat pipes use capillary forces; and (2) they depend, at least initially, on nucleate boiling, whereas heat pipes vaporize the fluid from a large, ever-present liquid/vapor interface. Thus, thermosiphon heat exchangers may require a significant temperature difference to initiate boiling (Mathur and McDonald 1987; McDonald and Shivprasad 1989). Thermosiphon tubes require no pump to circulate the working fluid. However, the geometric configuration must be such that liquid working fluid is always present in the evaporator section of the heat exchanger.

Thermosiphon loops differ from other coil energy recovery loop systems in that they require no pumps and hence no external power supply, and the coils must be appropriate for evaporation and condensation. Two-phase thermosiphon loops are used for solar water heating (Mathur 1990a) and for performance enhancement of existing (i.e., retrofit applications) air-conditioning systems (Mathur 1997a, 1997b, 1997c, 1997d). Two-phase thermosiphon loops can be used to downsize new air-conditioning systems and thus reduce the overall project costs. Figure 17 shows thermosiphon loop performance (Mathur and McDonald 1986) for an eight-row, 12 fpi system in unidirectional mode. Figure 18 shows the performance of the same system as a function of percent static charge for an overall temperature difference of 63°F.These systems have been designed with various working fluids like *n*-pentene and R-134a (Samba et al. 2013) with lower saturation pressures at atmospheric pressure, water (Mathur 1994) at negative operating pressures at typical ambient condition, and napthalene for high-temperature applications (Orr et al. 2016).

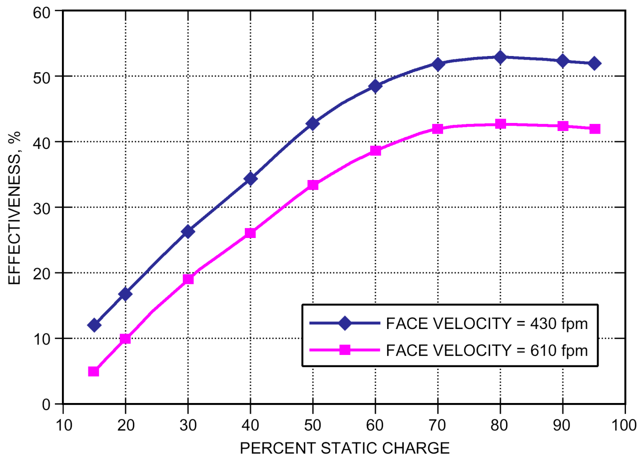

Fig. 18 Typical Performance of Two-Phase Thermosiphon Loop as Function of Percent Static Charge: Eight Rows, 12 fpi Unidirectional System; Overall Temperature Difference of 63°F
(Mathur and McDonald 1986)

These energy recovery systems are available to enhance system performance (similar to heat pipe heat exchangers) of existing systems (Cieslinski and Fiuk 2013; Mathur 1994; Yau 2007) or to downsize new equipment. Using two-phase thermosiphon loop systems or other energy recovery systems can improve the overall EERs of the systems and can earn significant LEED® points. These systems are also used for exhaust heat recovery for automotive application (Orr et al. 2016) and for cooling telecommunications systems (Samba et al. 2013).

Example 3. Sensible Heat Energy Recovery in Two-Phase Thermosiphon Loop. Outdoor air at 10°F enters an eight-row. two-phase thermosiphon loop with a face area of 2 ft^2 at a face velocity of 430 fpm. Exhaust air enters the two-phase thermosiphon loop with the same velocity with same face area at 75°F. Pressure drop across the heat pipe is 0.72 in. of water. The supply air density is 0.0843 lb/ft^3. Efficiencies of the electric motor and connected fan are 90 and 75%, respectively. Assuming the performance characteristics of the two-phase thermosiphon as shown in Figure 26, determine the sensible effectiveness $\varepsilon_{sensible}$, exit temperature of supply air to the space T_2, sensible energy recovered q_s, and power supplied to the fan motor p_s.

Solution:

From the data given, the overall temperature difference between supply and exhaust airstreams is 65°F (i.e., 75 – 10 = 65). From Figure 17, at face velocity of 430 fpm and with eight rows, the effectiveness is about 52.5%. Because the mass flow rate of the airstreams is the same, and, for simplification, assuming specific heat is constant at 0.24 Btu/lb·°F1, then the exit temperature of the supply air to the space can be obtained as follows:

$$t_2 = 10 + 0.525(75 - 10) = 44.1°\text{F}$$

The sensible energy recovered can be obtained from a variation of Equation (5a) as

$$q_s = 60(430)(2)(0.0843)(0.24)(44.1 - 10) = 35{,}599 \text{ Btu/h}$$

The supply air fan power can be obtained using Equation (8) as

$$Q_s = (430)(2) = 860 \text{ ft}^3\text{/min}$$

$$P_s = [(860)(0.72)]/[(6356)(0.9)(0.75)] = 0.15 \text{ hp}$$

Because there are two airstreams, the total pumping power of the two-phase thermosiphon loop (neglecting the differences in air density between the airstreams) is approximately twice the value of P_s (i.e., 0.30 hp).

Liquid-Desiccant Cooling Systems

Liquid-desiccant cooling systems (LDCSs) have great potential in reducing energy costs associated with meeting latent loads in high-occupancy commercial building applications. An LDCS can produce very dry air as long as the desiccant solution is supplied to its dehumidifier at low temperatures. However, an LDCS requires additional components: two or three plate-to-plate heat exchangers, a separate cooling tower, pump, fan, and a packed-bed humidifier and regenerator.

When exposed to moist air at the proper temperature and concentration, some chemical solutions (e.g., lithium chloride, lithium bromide, calcium chloride, triethylene glycol) can either absorb water vapor from, or release water vapor to, the airstream. These chemicals, known as liquid desiccants, are used to dehumidify air with significantly less energy consumption than by a mechanical refrigeration system. Liquid desiccants also have the advantage of removing bacteria and dust from the air compared to solid desiccant wheels or mechanical vapor compression systems. It requires a lower-temperature (less than 180°F) heat source for regeneration, which can be provided by solar energy or waste heat from local equipment. Moisture transfer between the desiccant solution and process air is caused by the difference in vapor pressures at the desiccant surface and that of the air passing over it. The desiccant collects moisture when its vapor pressure is lower than that of the water vapor in the air, and releases it when its vapor pressure is higher. Absorption of water vapor releases heat energy, including the heat of vaporization and, in some cases, additional heat. After absorbing water vapor from the processes air, the solution temperature increases slightly, and its concentration drops as absorbed water vapor becomes part of the solvent of the solution. For cyclic operation, the desiccant solution is regenerated to restore its original concentration; this involves heating the solution with a low-grade heat source and exposing the heated solution to an airstream, as shown in Figure 19 (Williams 2007). Often, outdoor air is used as a scavenging airstream in regeneration. Heat added to the desiccant solution for regeneration is of low grade and can be obtained from a source at no more than 160°F. Such low-temperature energy sources are often available as waste heat in many applications.

As seen in Figure 19, the hot, humid process air to be dehumidified enters the packed-bed dehumidifier or absorber at state 1 and leaves at state 2. The concentrated desiccant solution enters the absorber at state 5 and leaves at state 6 as dilute solution after absorbing water vapor from the process airstream. The dilute solution now passes through a liquid-to-liquid plate heat exchanger to recover some of the heat energy from the concentrated solution exiting the regenerator. The dilute solution is heated further to the required temperature in the desiccant solution heating coil and is sprayed over the incoming outdoor scavenging air passing through the regenerator. For air-to-air energy recovery purposes, the exhaust air from the space can be used for greater effectiveness in place of outdoor scavenging air. Because of the high inlet temperature of the desiccant solution, its vapor pressure is relatively higher than that of the incoming air, causing moisture transfer from the solution to the air. The inlet temperature of the desiccant solution to the regenerator for the given state of the air at state 8 is such that the desiccant concentration is restored to its value at state 6 as it exits the regenerator. The concentrated desiccant solution leaving the regenerator transfers heat energy to the incoming dilute solution exiting the packed-bed dehumidifier, in the liquid-to-liquid plate heat exchanger. Further temperature reduction of the desiccant solution, if required before it enters the absorber, can be accomplished by using cooling water from a cooling tower in the desiccant cooling coil. During extreme high-latent-load conditions, chilled water, rather than water from a cooling tower, may be necessary to cool the desiccant solution before it enters the absorber. The temperature of the solution at the inlet of the absorber strongly influences the air-dehumidifying capacity. The higher the concentration and lower the temperature of the solution, the higher the moisture absorbed and lower the humidity of the exiting air from the absorber.

Types of Desiccant Solutions. Desiccant solutions used by different investigators, as well as their key features, are shown in Table 2.

Lithium chloride (LiCl) is a good candidate material because it has good desiccant characteristics and does not vaporize in air at ambient temperature, but it is corrosive. Calcium chloride ($CaCl_2$) is an equally good candidate for desiccant choice and is less expensive than LiCl. A 50%/50% combination of LiCl and $CaCl_2$ is also being investigated. Triethylene glycol (TEG) has a low vapor pressure that enables effective dehumidification in the absorber, but it also makes it vulnerable to evaporation. Special coatings may be required to avoid corrosion, and efficient absorber design may be needed to prevent liquid or desiccant carryover.

Fluid Flow Configuration. Four different types of fluid flow configurations are used by different investigators in the packed beds for absorbers and regenerators.

- Counter flow
- Parallel flow
- Cross flow
- Falling film

Table 2 Types of Liquid-Desiccant Solutions

Desiccant Type	Typical Concentration Range, %	Operating Temperatures, °F
LiCl	34 to 43	77 to 81
TEG	92 to 98	68 to 114
LiBr	42.6 to 57	68 to 93
$CaCl_2$	40 to 43	81
KCOOH	72.8 to 74	71 to 77

Source: Jain and Bansal (2007).

Most applications are counter flow, because it provides the maximum enthalpy and humidity effectiveness for the packed beds.

Types of Packing. Packing arrangements are basically either structured or random types. In some cases, vertical surfaces or tubes are used, over which the desiccant solution falls as a thin film. The purpose of packing is to provide the maximum surface area for the desiccant solution to come in contact with the process air. The greater the area, the greater the dehumidification. The main parameters affecting bed arrangement are the enhancement of heat and mass transfer, air pressure drop, and pumping power to circulate the fluids through the bed. The main advantage of structured packing is the low pressure drop, whereas that of random packing is high heat and mass transfer but with high pressure drop. Packing compactness is typically expressed in terms of packing density a_t, in units of m^2/m^3. Typically, packing densities exceeding 200 m^2/m^3 have been achieved in the field. A higher packing density allows for more dehumidification but also produces greater pressure drop for the fluid streams. In random packing arrangements, packing is made up of Berl saddles, Raschig rings, and saddles of various trademarked designs, whereas in structured packing, the packing consists of cellulose rigid media pads, wood grids, expanded metal lash packing, or double spiral rings.

Twin-Tower Enthalpy Recovery Loops

One method of ventilation air enthalpy recovery is liquid desiccant twin-tower systems. In this air-to-liquid, liquid-to-air enthalpy recovery system, sorbent liquid circulates continuously between supply and exhaust airstreams, alternately contacting both airstreams directly in contactor towers (Figure 20). This liquid transports water vapor and heat. The sorbent solution is usually a halogen salt solution such as lithium chloride, calcium chloride, and water.

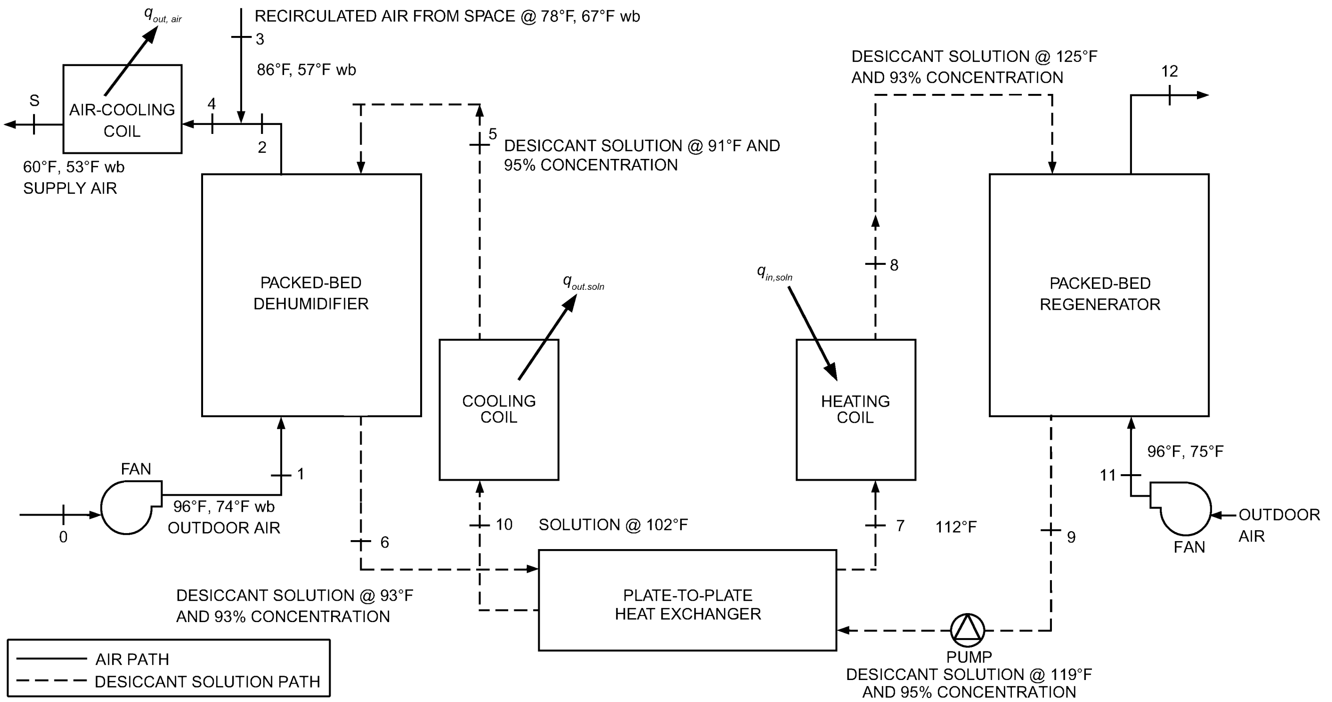

Fig. 19 Typical Dehumidification of Outdoor Air by Liquid-Desiccant System
(Williams 2007)

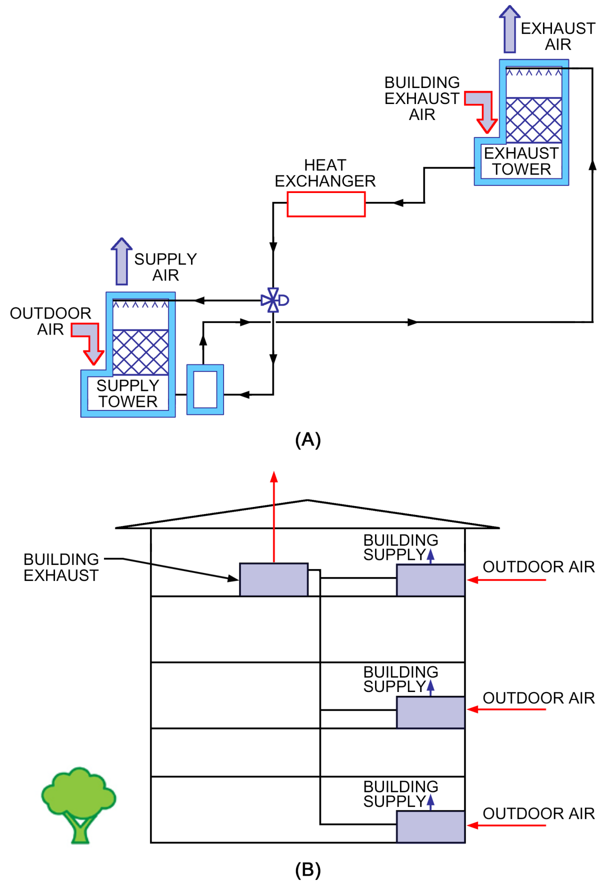

Fig. 20 Twin-Tower System Performing Indirect Evaporative Air Cooling: (A) Schematic and (B) Installation in Building

Pumps circulate the solution between supply and exhaust contactor towers.

Both vertical and horizontal airflow contactor towers are available, and both types can be combined in a common system. Contactor towers of both configurations are commonly available with airflow capacities of up to 84,000 scfm.

In the vertical configuration, the supply or exhaust air passes vertically through the contact surface counterflow to the sorbent liquid to achieve high contact efficiency. In the horizontal configuration, air passes horizontally through the contact surface cross flow to the sorbent liquid, yielding a slightly lower contact efficiency.

The contactor surface is usually made of nonmetallic materials. Air leaving the contactor tower passes through demister pads, which remove any entrained sorbent solution. The halogen salt solution does not offgas or vaporize.

Design Considerations. Twin-tower enthalpy recovery systems operate primarily in the comfort conditioning range; they are not suitable for high-temperature applications. During summer, these systems operate efficiently with building supply outdoor air temperatures as high as 115°F. Winter supply air temperatures as low as –40°F can be tolerated without freezing or frosting problems because the sorbent solution is an effective antifreeze at all useful concentrations. During weather conditions that are not advantageous for energy recovery, the system automatically shuts down the pump(s), thereby saving operating power. When outdoor conditions warrant energy recovery, the system restarts automatically. Because contactor supply and exhaust air towers are independent units connected only by piping, supply and exhaust air fans can be located wherever desired. Contactor towers can operate with air inlet static pressure from –6 to 6 in. of water, but require less than 1.8 in. of water air-side pressure drop. The exhaust contactor tower may operate at a higher internal static pressure than the supply contactor tower without danger of exhaust-to-supply cross contamination.

Particulate cross contamination does not occur because wetted particulates remain in the sorbent solution until they are filtered. Limited gaseous cross contamination may occur, depending on the solubility of the gas in the sorbent solution. Sorbent solutions, especially halide brines, are bactericidal. Lithium or calcium chloride as used in twin-tower systems is viricidal against all viruses.

If the building or process exhaust contains large amounts of lint, animal hair, or other solids, filters should be placed upstream of the exhaust air contactor tower; if it contains large amounts of gaseous contaminants, such as chemical fumes and hydrocarbons, investigate the possibility of cross contamination as well as the possible effects of contaminants on the sorbent solution. When using twin-tower systems in controlled-humidity applications in colder climates, saturation effects (which can cause condensation, frosting, and icing in other types of equipment) may overdilute the twin towers' sorbent solution. Heating the sorbent liquid supplied to the supply air contactor tower can prevent dilution, as shown in Figure 20A. Heating raises the discharge temperature and humidity of air leaving the supply contactor tower, thus balancing the humidity of the system and preventing overdilution.

A thermostat sensing the leaving air temperature from the supply air contactor tower is commonly used to control the solution heater to deliver constant-temperature air regardless of outdoor temperature. Automatic addition of makeup water to maintain a fixed concentration of the sorbent solution enables the twin-tower system to deliver supply air at fixed humidity during cold weather. Thus, the system provides a constant supply air temperature and humidity without preheat or reheat coils or humidifiers.

Any number of supply air towers can be combined with any number of exhaust air towers, as shown in Figure 20B. If a sufficient elevation difference exists between supply and exhaust towers, gravity may be used to return the sorbent solution from the upper tower(s) in lieu of pumped flow. Twin-tower energy recovery systems can be retrofitted to existing buildings. The towers are connected to each other with piping. The solution piping can be easily installed inside or outside the building. Twin-tower enthalpy recovery systems operate with only routine maintenance. Complete instructions on procedures as well as spare-parts lists are included in operating manuals relevant to each installation. Periodically, the circulation pumps, spray nozzles, liquid transfer controls, and mist eliminators need checking and adjusting or maintenance. Halide brine solutions are normally used as energy transfer media in twin-tower systems. Manufacturers provide technical support, including solution sampling services to monitor and report changes in concentration and pH, thus ensuring continued maximum performance.

Fixed-Bed Regenerators

Fixed-bed regenerators (FBRs) are available in many configurations, sizes, and airflow patterns. In general, FBRs contain one or two cores for the purpose of energy transfer, and some means of reversing airflow through its core(s) to alternately store and release energy.

In North America, double-core FBRs are more common; single-core FBRs are available but have not been widely adopted. Double-core FBRs are packaged with two cores and an airflow control module to alternate supply and exhaust air through each core. These units are used in large residential, commercial, and industrial applications with airflow capacity in the range of 50 to 100,000 cfm. Cost and physical size are the main barriers between double-core FBRs and the residential home market.

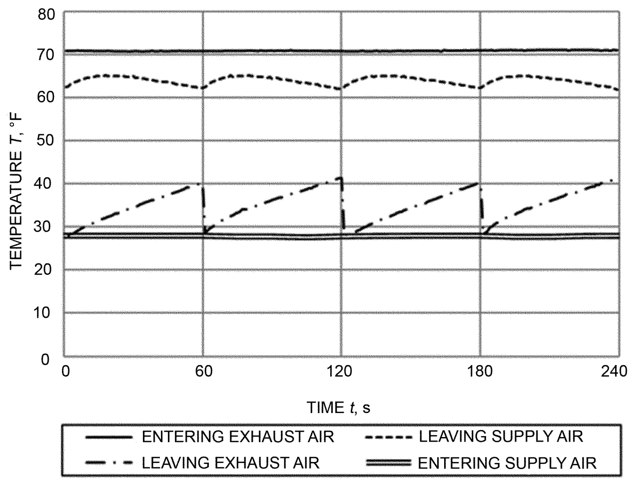

Fig. 22 Typical Temperature Profile of Fixed-Bed Regenerator Operating at 60 s Recovery Periods

Unlike other energy recovery exchangers, in FBRs the outlet air temperatures and humidities are never steady state; instead, they rise and fall cyclically with each reversal of airflow. This poses challenges to test facilities, which generally seek to take measurement only when the device under test is operating in steady state with very small variations in outlet conditions. The latest edition of ASHRAE *Standard* 84 (in progress) introduces techniques for data reduction and humidity averaging to deal with this challenge. As more manufacturers enter the market, certified performance data may become available.

Arrangement. Cores are typically split into smaller cells to facilitate transportation, handling, and cleaning. Plates are formed with spacers (e.g., ribs or dimples) constructed into the plates or by alternating corrugation angle between adjacent plates (direction of the angle spaces plates apart). Corrugation patterns and plate thickness varies depending on manufacturer. Plate spacing ranges from 0.1 to 0.5 in.. Aluminum cores are the most popular construction, but some manufacturers also offer polypropylene cores for corrosive or lightweight applications.

Airflow control modules consist of a single deflector plate or a multiblade damper system. Early designs included pneumatic actuators controlled by solenoids and flow valves. Electric actuators were introduced to improve synchronization and to reduce switchover time of multiblade dampers to 0.5 to 1.5 s. Rapid switchover is of key importance to minimize exhaust air transfer between recovery periods.

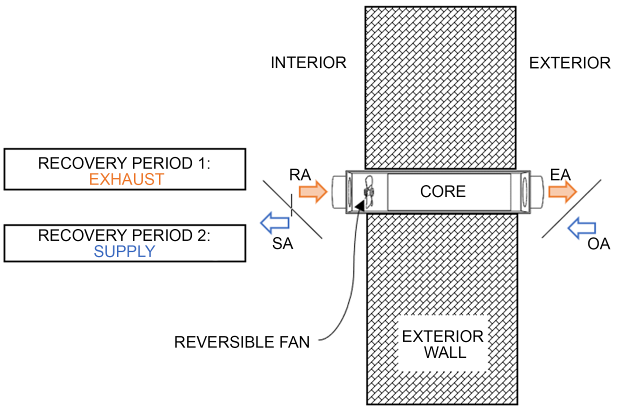

Fig. 23 Single-Core Fixed-Bed Regenerator

In Europe, various types of single-core devices are used for residential, apartment, condo, or renovation projects with limited ceiling space. Often, the only option is a room-based decentralized ductless ventilation system, which consists of several single-core FBRs installed in the exterior walls of a space. The individual devices are synchronized via interconnecting wiring or wireless communication to alternately store and release energy from a single honeycomb-shaped, ceramic core.

Operation. Typically, the devices are staggered so half of the ventilation system is exhausting while the other half is bringing tempered outside air into the space. After a certain period of time, called the recovery period, the system switches direction to recharge the devices that have been in the outside airstream. The other half of the system now transfers energy to the outside air stream until the next recovery period.

Many single-core FBRs are equipped with a relatively small fan, capable of changing its direction at the end of each recovery period to supply air to, or exhaust air from, the space. Depending on the manufacturer, the device is designed with two opposite fans in adjacent channels, airflow deflectors, and/or backdraft flaps to alternate exhaust and supply air through the core. Performance curves of the fan must be considered for high-rise buildings and locations with high winds. Stack effect and wind direction can be detrimental to supply airflow rate or effectiveness of single-core FBRs.

Recovery periods in the range of 50 to 70 s are common to both single-core and double-core FBRs. Manufacturers report sensible effectiveness typically in the range of 80 to 90%. Latent recovery is also possible. With sensible-only FBRs, latent recovery can occur in

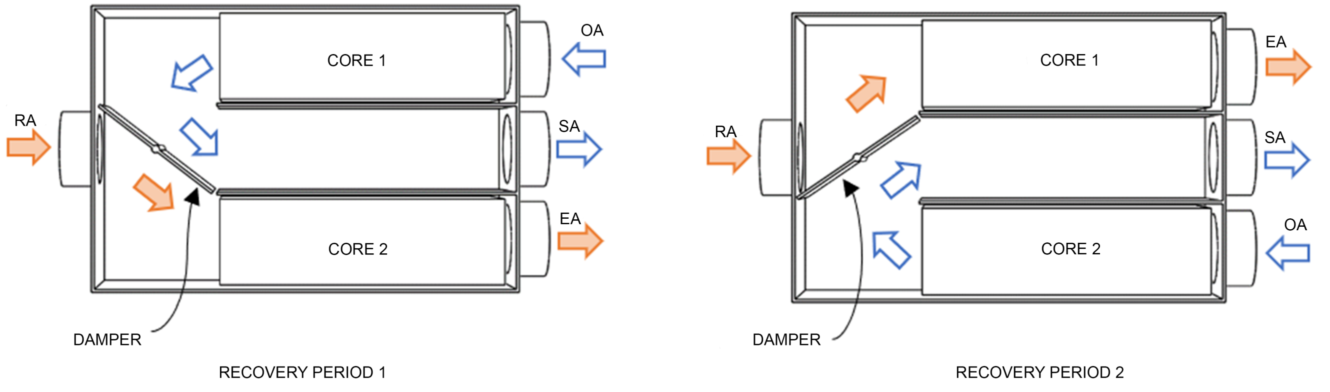

Fig. 21 Double-Core Fixed-Bed Regenerators

condensing conditions (i.e., heating mode). FBRs also can be manufactured with desiccant treatment providing latent recovery in all conditions. Manufacturers report latent effectiveness in desiccant-treated FBRs in the 60 to 80% range, and in non-desiccant-treated FBRs, as high as 70% (depending on conditions). Longer recovery periods up to 3 h are used for free cooling mode. Some manufacturers vary the recovery period between 20 to 120 s to control discharge air temperature and/or humidity. Shorter recovery periods improve effectiveness, but the drawback is a higher exhaust air transfer ratio.

Exhaust air transfer for outdoor double-core FBRs is reported by manufacturers to be in the range of 3 to 5%. Indoor applications with long ducts connecting the unit to the ambient suffer from higher exhaust transfer rates due to the volume of exhaust air drawn back from these ducts. Generally, duct length is kept under 25 ft to limit increase in pressure drop and exhaust air transfer.

An advantage of FBRs is the repeated exhaust regeneration, which allows this technology to operate in extremely cold climates below –40°F without the need of auxiliary defrost strategies. Typically, discharge air from a double-core FBR is condition by a heater, or an interlocked air handling unit before entering the space.

4. PERFORMANCE RATINGS

Performance ratings should be based on tests performed in qualified laboratories that are staffed and instrumented to meet requirements of ASHRAE *Standard* 84 (in progress) and AHRI *Standard* 1060.It may be very difficult to adhere to any standard when field tests are made.

Performance Ratings for Air-to-Air Heat or Heat and Mass Exchangers

ASHRAE *Standard* 84 (1) establishes a uniform method of testing for obtaining performance data; (2) specifies the data required, calculations to be used, and reporting procedures for testing each of seven independent performance factors and their uncertainty limits; and (3) guides selection of test equipment. The independent performance factors are sensible, latent, and total effectivenesses e_s, e_l, and e_t; the enthalpy recovery ratio; supply and exhaust air pressure drops ΔP_s and ΔP_e; exhaust air transfer ratio (EATR), which characterizes the fraction of exhaust air transferred to the supply air; and outdoor air correction factor (OACF), which is the ratio of supply inlet to outlet airflow. An additional useful parameter, which can be evaluated only when fan efficiency is known or assumed, is the recovery efficiency ratio.

AHRI *Standard* 1060 is an industry-established standard for rating air-to-air heat/energy exchanger performance for use in energy recovery ventilation equipment. This standard, based on ASHRAE *Standard* 84, establishes definitions, requirements for marking and nameplate data, and conformance conditions intended for the industry, including manufacturers, engineers, installers, contractors, and users. Standard temperature and humidity conditions at which equipment tests are to be conducted are specified for summer and winter conditions. Published ratings must be reported for the performance factors specified in ASHRAE *Standard* 84. The AHRI certification program using *Standard* 1060 is used to verify ratings published by manufacturers. At this time the scope of the certification program includes plate, rotary, and heat pipe exchangers. Coil energy recovery systems, twin-tower enthalpy recovery loops, and fixed-bed regenerators are not in the scope of the certification program.

Before 2020, AHRI *Standard* 1060 required the publication of ratings based on performance of the rated exchanger at two balanced airflow rates selected by the manufacturer, at standard winter and summer conditions:

Winter: Outdoor air at t_1 = 35°F and t_{w1} = 33°F
Indoor (room) air at t_3 = 67°F and t_{w3} = 58°F and $p_2 - p_3 = 0$

Summer: Outdoor air at t_1 = 95°F and t_{w1} = 78°F
Indoor (room) air at t_3 = 75°F and t_{w3} = 63°F and $p_2 - p_3 = 0$

These ratings applied to operation with the static pressure at supply air outlet equal to the static pressure at the exhaust air inlet. Ratings for the leakage metrics EATR and OACF also were to be provided at these static pressure conditions, as well as at two additional static pressure conditions selected by the manufacturer.

As of 2020, the AHRI certification program verifies ratings generated by a manufacturer's prediction software across a broad psychrometric range, at any combination of airflows and static pressures supported by the software. The program also verifies that all of the ratings required for a complete characterization of exchanger performance are provided.

Some metrics can be provided optionally. The new **energy recovery ratio** (Equation [3]) is used to characterize the performance of the exchanger in relation to its use in a system. This metric is important because, as of 2019, ASHRAE *Standard* 90.1 requires that energy recovery systems have an enthalpy recovery ratio of at least 50% (i.e., a change in enthalpy of outdoor air supply equal to 50% of the difference between outdoor air and entering exhaust air enthalpies at design conditions).

Standard 90.1 previously used 50% effectiveness at design conditions, but this metric measures performance of the exchanger and is less relevant when defining the performance of an enthalpy recovery system.

It is more difficult to measure performance of a heat or heat/mass exchanger when condensation occurs. ASHRAE *Standard* 84 provides theoretical approaches, but in practice it is challenging to measure the rate at which condensate is generated, which is required to confirm the test meets steady-state requirements. CSA *Standard* C439-2018 provides a method to measure some energy and ventilation parameters of a complete ventilating unit equipped with heat or heat and mass recovery and active frost prevention or recovery means. However, this is intended for use with packaged ventilating units, and is not intended to characterize the conditions under which frost occurs in the exchanger itself.

Performance Ratings for Residential Ventilators with Air-to-Air Heat or Heat and Mass Exchangers

Residential ventilation products are certified by the Home Ventilating Institute (HVI 2015). Heat recovery ventilators and energy recovery ventilators (H/ERVs) are certified for their ventilation performance at high speed and for their energy recovery performance at manufacturer's selected airflows. The *HVI Certified Products Directory* (www.hvi.org/proddirectory) is useful for comparing product performance on a level playing field. H/ERVs must be tested at prescribed conditions in accordance with CSA *Standard* C439 at an HVI-designated laboratory to be eligible for HVI certification.

HVI issues certified ratings for H/ERVs for airflow (net supply, gross exhaust, gross supply) and for energy recovery performance parameters include the following:

- Sensible recovery efficiency (SRE)
- Adjusted sensible recovery efficiency (ASRE)
- Total sensible recovery efficiency (TRE) in cooling modes
- Adjusted total sensible recovery efficiency (ATRE) in cooling modes
- Latent recovery moisture transfer
- Very-low-temperature ventilation reductions

Note that ratings for ASRE are to be used in energy-modeling software where fan power is a separate modeling input, to avoid

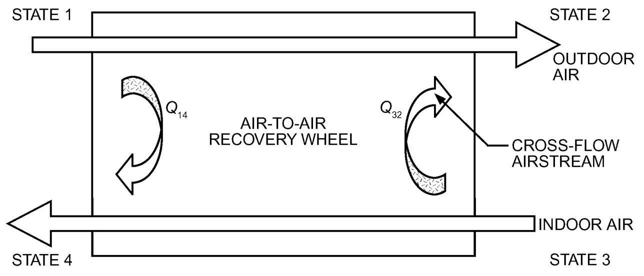

Fig. 24 Air Leakage in Energy Recovery Units

counting fan power twice. Residential H/ERVs are certified as a packaged unit, whereas larger commercial H/ERVs typically carry certified ratings for energy recovery performance of just the core. Products must be listed in the online directory to be considered HVI certified (i.e., not listed means not certified).

5. ADDITIONAL TECHNICAL CONSIDERATIONS

The rated effectiveness of energy recovery units is obtained under balanced flow conditions (i.e., the flow rates of supply and exhaust airstreams are the same). However, these ideal conditions do not always exist, because of design for positive building pressure, air leakage, fouling, condensation, or frosting, and the following other factors.

Air Leakage

Air leakage refers to any air that enters or leaves the supply or exhaust airstreams. Zero air leakage in either airstream would require m_1 to equal m_2 and m_3 to equal m_4. External air leakage occurs when the ambient air surrounding the energy recovery system leaks into (or exits) either or both airstreams. Internal air leakage occurs when holes or passages are open to the other airstream. Internal air leakage occurs when heat or energy exchanger design allows (1) tangential air movement in the wheel's rotational direction and (2) air movement through holes in the barrier between airstreams. Under some pressure differentials, air leaks in and out of each airstream in nearly equal amounts, giving the illusion that there is no air leakage. Heat and water vapor transfer could appear to be greater than it is in reality.

Leakage varies with heat exchanger type and design, airstream static pressure differences, and the physical condition of the heat exchanger (see Table 3). Air leakage is seldom zero because external and internal air pressures are usually different, causing air to leak from areas of higher pressure to those of lower pressure. Cross-leakage, cross-contamination, or mixing between supply and exhaust airstreams may occur in air-to-air heat exchangers and may be a significant problem if exhaust gases are toxic or odorous.

Air leakage between incoming fresh air and outgoing exhaust air can be classified into two mechanisms: cross-flow and carryover.

Cross-flow leakage is caused primarily by difference in static pressures between states 2 and 3 and/or between states 1 and 4, as shown in Figure 24. This is a major cause of cross-flow leakage, and underscores the importance of specifying precise locations for fans that circulate the airstreams. Cross-flow can also be caused by factors such as provisions for surging, geometrical irregularities, and local velocity distribution of the airstreams.

Carryover occurs in rotary recovery units because of wheel rotation out of one airstream into the other. Exhaust air trapped in cavities in the heat transfer medium is carried into the fresh airstream by the wheel's rotation.

The test estimate of the air leakage is characterized by two dimensionless parameters: the exhaust air transfer ratio (EATR) and outdoor air correction factor (OACF). During these tests, $m_2 = m_3$, while an inert tracer gas is introduced at station 3 (Shang et al. 2001a):

$$\text{EATR} = \frac{c_2 - c_1}{c_3 - c_1} \tag{15}$$

where c_1, c_2, and c_3 are the concentrations of inert gas at states 1, 2, and 3, respectively.

EATR is the ratio of the concentration increase in supply air relative to the maximum concentration difference between stations 3 and 1. It is representative of the air mass leakage from exhaust air into supply air, when the air-to-air heat exchanger is operating at mass flow rates and static pressures equal to those at which the device was tested.

$$\text{OACF} = \frac{m_1}{m_2} \tag{16}$$

where m_1 and m_2 are the mass flow rates of incoming fresh airstream at states 1 and 2, respectively.

OACF gives the ratio of outdoor airflow required at the inlet (station 1) relative to the supply flow at 2 to compensate for air that leaks into or out of the exchanger, to meet the required net supply airflow to the building space, when the air-to-air heat exchanger is operating at mass flow rates and static pressures equal to those at which the device was tested.

Ideal operating conditions exist when there is no air leakage between the streams, EATR is close to zero, and OACF approaches 1. Deviations from ideal conditions indicate air leakage between the airstreams, which complicates the determination of accurate values for pressure drop and effectiveness. Methods to estimate actual flow rates at states 1, 2, 3, and 4 when air leakage exists and the values of the parameters EATR and OACF are known are discussed in Friedlander (2003) and Moffitt (2003). EATR and OACF must be determined when evaluating any HRV or ERV in which leakage between airstreams occurs, when evaluating the actual flow rate of outdoor air supplied for a given ventilation requirement, and when estimating the capacity of ventilator fans, as shown by the following.

Air Capacity of Ventilator Fans

For a given ventilation requirement Q_v, the density-corrected volume flow rate capacity Q_{fan} of the supply fan is greater than Q_2 if air leakage exists (see Figure 24).

In the following calculations, be sure to obtain the value for EATR that applies to the device when it operates at the airflow volumes and static pressure that will exist in the specific application. The leakage flow rate from the exhaust to the supply air $Q_{3\text{-}2}$ can be estimated by the equation.

$$Q_3 - Q_2 = Q_2 \frac{\text{EATR}}{100} \tag{17}$$

If the ventilation requirement is Q_v, then the actual volume flow rate of supply air to the space is calculated as

$$Q_2 = Q_v + Q_{3-2} = Q_v + Q_2 \frac{\text{EATR}}{100} \tag{18}$$

This can be simplified as

$$Q_2 = \frac{Q_v}{1 - (\text{EATR}/100)} \tag{19}$$

which gives the quantity of air entering the space. Assuming steady-state conditions, and balanced flow through the energy recovery ventilator, the density corrected quantity of air leaving the building space (Q_3) should be same as air supplied to the space Q_2.

$$Q_3 = Q_2 = \frac{Q_v}{1-(\mathrm{EATR}/100)} \tag{20}$$

Combining Equations (16) and (20) gives the required supply air inlet volume q_1 as

$$Q_1 = Q_2(\mathrm{OACF}) = \frac{Q_v(OACF)}{1-(\mathrm{EATR}/100)} \tag{21}$$

Caution should be used with Equations (20) and (21) because in many practical applications, Q_2 is not equal to Q_3.

Equation (21) gives the volume flow rate capacity of the intake, which equals the exhaust outlet for balanced airflow. (Fan capacity depends on location. This statement only applies to the supply fan at station 1 [blow-through]; its capacity would change at station 2 [draw-through].)

Pressure Drop

Pressure drop for each airstream through an energy recovery unit depends on many factors, including exchanger design, mass flow rate, temperature, moisture, and inlet and outlet air connections. The pressure drop must be overcome by fans or blowers. Because the power required for circulating airstreams through the recovery unit is directly proportional to the pressure drop, the pressure drop through the energy recovery unit at its operating conditions should be known. The pressure drop may be used with the fan efficiency to characterize the energy used by the exchanger and in turn the efficiency (not effectiveness) of an application.

Maintenance

The method used to clean a heat exchanger depends on the transfer medium or mechanism used in the energy recovery unit and on the nature of the material to be removed. Grease build-up from kitchen exhaust, for example, is often removed with an automatic water-wash system. Other kinds of dirt may be removed by vacuuming, blowing compressed air through the passages, steam cleaning, manual spray cleaning, soaking the units in soapy water or solvents, or using soot blowers. The cleaning method should be determined during design so that a compatible heat exchanger can be selected.

Cleaning frequency depends on the quality of the exhaust airstream. Residential and commercial HVAC systems generally require only infrequent cleaning; industrial systems, usually more. Consult equipment suppliers about the specific cleaning and maintenance requirements of systems being considered.

Filtration

Filters are recommended and should be placed in both the supply and exhaust airstreams to reduce fouling and thus the frequency of cleaning. Exhaust filters are especially important if the contaminants are sticky or greasy or if particulates can plug airflow passages in the exchanger. Supply filters eliminate insects, leaves, and other foreign materials, thus protecting both the heat exchanger and air-conditioning equipment. Snow or frost can block the air supply filter and cause severe problems. Specify steps to ensure a continuous flow of supply air.

Controls

Heat exchanger controls may control frost formation or regulate the amount of energy transferred between airstreams at specified operating conditions. For example, ventilation systems designed to maintain specific indoor conditions at extreme outdoor design conditions may require energy recovery modulation to provide an economizer function, to prevent overheating ventilation supply air during cool to moderate weather or to prevent overhumidification. Modulation methods include tilting heat pipes, changing rotational speeds of (or stopping) heat wheels, or bypassing part of one airstream around the heat exchanger using dampers (i.e., changing the supply-to-exhaust mass airflow ratio).

Fouling

Fouling, an accumulation of dust or condensates on heat exchanger surfaces, reduces heat exchanger performance by increasing resistance to airflow, interfering with mass transfer, and generally decreasing heat transfer coefficients. Increased resistance to airflow increases fan power requirements and may reduce airflow.

Increased pressure drop across the heat exchanger core can indicate fouling and, with experience, may be used to establish cleaning schedules. Reduced mass transfer performance (latent effectiveness) indicates fouling of permeable membranes or desiccant sorption sites. Heat exchanger surfaces must be kept clean to maximize system performance.

Corrosion

Process exhaust frequently contains corrosive substances. If it is not known which construction materials are most corrosion resistant for an application, the user and/or designer should examine on-site ductwork, review literature, and contact equipment suppliers before selecting materials. A corrosion study of heat exchanger construction materials in the proposed operating environment may be warranted if installation costs are high and the environment is corrosive. Experimental procedures for such studies are described in an ASHRAE symposium (ASHRAE 1982). Often contaminants not directly related to the process are present in the exhaust airstream (e.g., welding fumes or paint carryover from adjacent processes).

Moderate corrosion generally occurs over time, roughening metal surfaces and increasing their heat transfer coefficients. Severe corrosion reduces overall heat transfer and can cause cross-leakage between airstreams because of perforation or mechanical failure.

Condensation and Freeze-Up

Condensation, ice formation, and/or frosting may occur on heat exchange surfaces. Ignoring entrance and exit effects, four distinct air/moisture regimes may occur as the warm airstream cools from its inlet condition to its outlet condition. First, there is a dry region with no condensate. Once the warm airstream cools below its dew point, there is a condensing region, which wets the heat exchange surfaces. If the heat exchange surfaces fall below freezing, the condensation freezes. Finally, if the warm airstream temperature falls below its dew point and is also below freezing temperature, sublimation causes frost to form. The locations of these regions and rates of condensation and frosting depend on the duration of frosting conditions; airflow rates; inlet air temperature and humidity; heat exchanger core temperature; heat exchanger effectiveness; geometry, configuration, and orientation; and heat transfer coefficients.

Sensible heat exchangers, which are ideally suited to applications in which heat transfer is desired but humidity transfer is not (e.g., swimming pools, kitchens, drying ovens), can benefit from the latent heat released by the exhaust gas when condensation occurs. One pound of moisture condensed transfers about 1050 Btu to the incoming air at room temperature. Equipment design should provide for continuous draining of the condensate from the unit. Transfer of latent heat occurs at the point of condensation, and little available energy remains in the condensate.

Condensation increases the heat transfer rate and thus the sensible effectiveness; it can also significantly increase pressure drops in heat exchangers with narrow airflow passage spacing. Frosting fouls the heat exchanger surfaces, which initially improves energy transfer but subsequently restricts the exhaust airflow, which in turn reduces the energy transfer rate. In extreme cases, the exhaust airflow (and supply, in the case of heat wheels) can become blocked. Defrosting a fully blocked heat exchanger requires that the unit be shut down for an extended period. As water cools and forms ice, it expands, which may seriously damage the heat exchanger core unless the water is entirely removed.

For frosting or icing to occur, an airstream must be cooled to a dew point below 32°F. Enthalpy exchangers transfer moisture from the airstream with higher moisture content (usually the warmer airstream) to the less humid one. As a result, frosting or icing occurs at lower supply air temperatures in enthalpy exchangers than in sensible heat exchangers. Cross-flow heat exchangers are more frost tolerant than other plate exchangers because frost blockage only occurs over a fraction of the exchanger exhaust airflow passages.

For these reasons, some form of freeze control must be incorporated into heat exchangers that are expected to operate under freezing conditions. Frosting and icing can be prevented by preheating supply air or reducing heat exchanger effectiveness (e.g., reducing heat wheel speed, tilting heat pipes, bypassing some supply air around the heat exchanger). Alternatively, the heat exchanger may be periodically defrosted. Manufacturers of commercially available packaged heat or energy ventilators routinely incorporate frosting control strategies or mechanisms appropriate to their equipment and its targeted application.

ASHRAE research project RP-543 discusses performance of several freeze control strategies (Phillips et al. 1989a, 1989b). ASHRAE research project RP-544 (Barringer and McGugan 1989a, 1989b) covers performance of enthalpy heat exchangers. Many effective defrost strategies have been developed for residential air-to-air heat exchangers, and may also be applied to commercial installations. Phillips et al. (1992) describe frost control strategies and their impact on energy performance in various climates.

If cyclic frost methods are used, the defrost period should be long enough to completely remove the frost, ice, and water during each defrost period.

Total heat exchangers transfer moisture from the more humid airstream to the less humid one. In winter, this results in an exhaust stream that is partially dehumidified as it is cooled. As a result, frosting or icing generally occurs at lower temperatures than in sensible-only heat exchangers. On the other hand, sensible heat exchangers can benefit from the latent heat released in a high-humid-exhaust airstream when condensation occurs.

See the Bibliography for sources of more information on frost growth and control.

Frost Control Strategies for Air-to-Air Energy Recovery Systems

In all cases, the potential for frost is in the warm-return-air side of the heat exchanger, which cools during a winter condition. For energy recovery devices, the exact outdoor air temperatures at which frosting will start depends on the temperature and humidity of the exhaust air and the type and performance of the energy recovery device. This temperature is known as the frost threshold temperature. The frost threshold varies by specific technology, and significantly by whether water vapor is transferred between airstreams:

- **Flat Plate Heat Exchangers.** In flat plate heat exchangers (Figures 2 and 4), warm return air cools as it passes through the heat exchanger. Moisture from the warm air condenses and creates water droplets on the media; as cool outdoor air enters the heat exchanger, the water droplets start to freeze, creating ice on the cold corner of the core. Figure 1 shows a typical temperature distribution of warm and humid; and cold and dry air streams. The circled area shows the potential problem area for frost formation. In general, frost buildup becomes a problem when outdoor air temperatures reach about 23°F for sensible devices, or approximately –5°F for total energy devices. Figure 3 shows frost threshold for energy (Afshin 2019) and heat recovery ventilators.
- **Energy Recovery Wheels.** In energy recovery wheels (Figure 6), warm return air cools as it passes through the rotating wheel. Moisture from the warm air condenses and creates water droplets on the surface of the wheel. The water molecules are then absorbed by the desiccant on the enthalpy wheel. As the wheel rotates through the cold supply air, the water can start to freeze and form ice before the water droplets can be cleared from the wheel surface. Ice typically forms on the internal surface of the wheel, near the outdoor air intake which can cause damage and reduce airflow and performance.
- **Heat Pipe, Thermosiphon Loop, and Runaround Glycol Heat Exchangers.** In these technologies, the return air is cooled by a fluid passing through a coil. (Figures 10 and 15) For frost to form, the coil fin surface must be freezing and below the return air dew point. For frost to build up, the return air must be cooled below freezing. Using the sensible effectiveness, the user can calculate the outdoor air temperature where frost would be expected. For example, with a sensible effectiveness of 50%, an outdoor air temperature of –6°F would cool 70°F return air to 32°F.

There are a variety of different approaches that can be used to determine the frost threshold temperature, including measuring temperature, humidity, or a variation in pressure drop. A frost control strategy is then determined to either eliminate the frost once it builds up on the exchanger (before it becomes too severe), or to prevent frost formation from ever starting at all. The choice between frost control and frost prevention will depend on the robustness of the air-to-air energy recovery device and the application.

An effective frost control strategy is one that meets the requirements of the application while being cost-effective and device appropriate. Figure 25 shows the possible control strategies (Afshin 2019) for controlling frost for plate heat exchangers used as ERVs.

In all cases of frost prevention, the amount of energy recovery is reduced and there will need to be additional heat added into the system. Energy efficiencies for each solution vary somewhat, and the best choice depends on application and the amount of upfront cost that can be tolerated.

Frost Prevention.

- *Preheat coil to heat up exhaust air*. Preheat frost control is a preventative strategy that can be used with any air-to-air energy recovery device. The objective is to prevent frost from occurring within the heat exchanger, while maintaining 100% or continuous ventilation. Energy recovery is reduced, because the difference in temperature between the preheated outside air and the return air has reduced. Heating coils (electric, steam, or hot water) are duct-mounted or integrated into the unit in the outdoor airstream so that the entering outdoor air temperature is preconditioned to a temperature above the frost threshold for the technology being used. While preheat typically has higher upfront costs than other strategies, it can result in significant operating savings in climates where frost control is required for a long period of time (more than a few hours). The preheater should be sized for the coldest outdoor air design temperature and to the appropriate outdoor air temperature where the exhaust air frost threshold is reached based on the design conditions and airflows for the type of heat exchanger selected. Figure 3 shows frost threshold characteristics for energy and heat recovery ventilators.

- *Face and bypass damper* (Figure 25D). Face and bypass frost control is a preventative strategy that can be used for flat plates, heat pipes, and rotary wheel exchangers, with the objective of preventing frost formation while maintaining 100% ventilation. As the outdoor air becomes colder, face and bypass dampers upstream of the heat exchanger modulate to reduce the amount of outdoor air flowing through the heat exchanger. This reduces the amount of energy recovered and keeps the exhaust temperature above the frost threshold. The supply and exhaust fans, as well as the outdoor air and exhaust air dampers, continue to operate during the frost control cycle. With this strategy there is no interrupted ventilation which makes it ideal for harsher environments or source control applications like laboratories. Furthermore, as both fans run continuously there is no depressurization of the building, eliminating the potential for combustion appliance backdraft into the occupied space. The lower leaving supply air temperature would potentially require post-conditioning, or some type of terminal reheat within the space to ensure that occupants remain comfortable under extreme conditions. A further advantage would be that the face-and-bypass can also be used for free cooling (economizer) and/or supply air temperature control in different modes of operation if required.
- *Performance modulation*. The heat exchanger can be controlled to ensure that frost does not form in the return air, by reducing the effectiveness and, as a result, suppressing the frost threshold. For example, wheel rotational speed can be reduced, runaround pumps can be slowed, heat pipes can be equipped with control valves or tilted. Since the heat exchanger is recovering less energy, more heating is required in the process downstream.

Defrosting.

1. *Exhaust defrosting* (Figure 25B). Exhaust-only defrost is one of the most cost-effective and simple strategies to implement. It periodically defrosts ice forming on the heat exchanger by shutting down the supply fan to remove the source of cold air, while using the warm exhaust air to heat up the exchanger. When the unit goes into a defrost cycle, the exhaust fan continues to operate, the supply fan is deenergized and the outdoor air damper closes. This method is most commonly used with flat plate or heat pipe heat exchangers and is ideal for source control applications where continuous exhaust is required. One drawback of this method is that ventilation is interrupted when the supply fan shuts down during the defrost cycle, which may not be acceptable as the equipment may not meet indoor air quality (IAQ) requirements as defined by ASHRAE Standard 62.1. This also creates negative indoor pressure, resulting in infiltration.
2. *Recirculation defrost* (Figure 25B). Recirculation defrost is also cost-effective and simple, commonly used in light commercial stand-alone air-to-air ERVs that are not used as a primary ventilation system. Ice formation on the heat exchanger is periodically defrosted by shutting down the exhaust fan, by closing the outdoor and exhaust air dampers, and by opening a recirculation air damper to remove the source of cold air. When the unit goes into a defrost cycle, the supply fan remains on to recirculate building exhaust air back into the occupied space. The exhaust air goes through the heat exchanger and provides defrosting in the absence of cold outdoor air. A drawback of this method is that ventilation is interrupted when the exhaust fan shuts down during the defrost cycle. This may not be acceptable in all applications and may not meet indoor air quality (IAQ) requirements as set out in ASHRAE Standard 62.1. It may however be acceptable in more moderate climates where freezing conditions occur only during unoccupied hours for a few hours per year.

Indirect Evaporative Air Cooling

In indirect evaporative air cooling, exhaust air passing through a water spray absorbs water vapor until it becomes nearly saturated. As the water evaporates, it absorbs sensible energy from the air, lowering its temperature. The evaporatively cooled exhaust air can then be used to cool supply air through an air-to-air heat exchanger, which

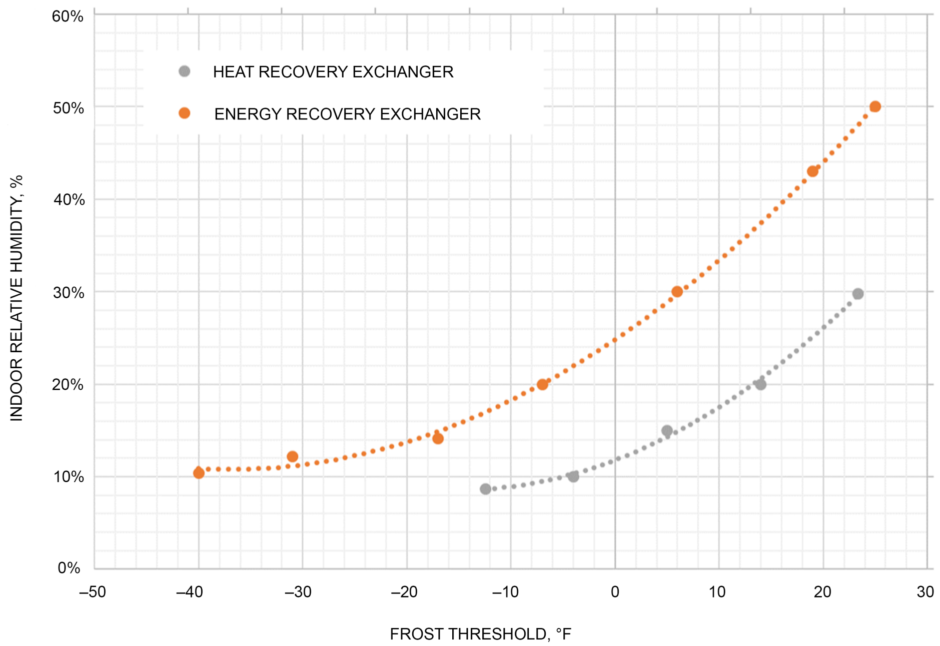

Fig. 26 Examples of Frost Threshold Temperatures for Energy and Heat Recovery Exchangers
(Afshin 2019)

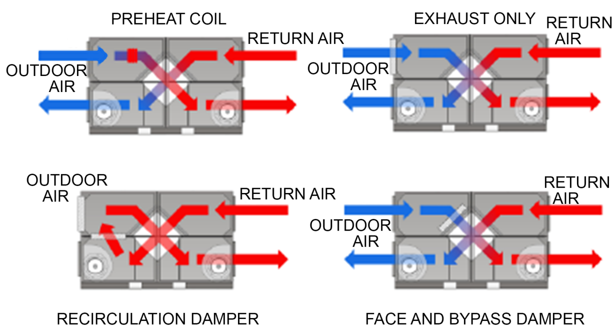

Fig. 25 Common Frost Control Strategies for Fixed-Plate Exchangers

may be used either for year-round energy recovery or exclusively for its evaporative cooling benefits. This process follows a constant wet-bulb line on a psychrometric chart: the airstream enthalpy remains nearly constant, moisture content increases, and dry-bulb temperature decreases.

Indirect evaporative cooling has been used with heat pipe heat exchangers, runaround coil loop exchangers, and flat-plate heat exchangers for summer cooling (Dhital et al. 1995; Johnson et al. 1995; Mathur 1990a, 1990b, 1992, 1993; Scofield and Taylor 1986). Exhaust air or a scavenging airstream is cooled by passing it through a water spray, wet filter, or other wetted media, resulting in a greater overall temperature difference between the supply and exhaust or scavenging airstreams and thus more heat transfer. Energy recovery is further enhanced by improved heat transfer coefficients because of wetted exhaust-side heat transfer surfaces. No moisture is added to the supply airstream, and there are no auxiliary energy inputs other than fan and water pumping power. The coefficient of performance (COP) tends to be high, typically from 9 to 20, depending on available dry-bulb temperature depression. The dry-bulb temperature decrease in the exhaust airstream caused by evaporative cooling tends to be 85 to 95% of the maximum available difference between the exhaust air inlet dry-bulb and wet-bulb temperatures. Therefore, exhaust air evaporative cooling is usually most cost effective in hot, dry climates where the evaporator can be used frequently to obtain large exhaust air dry-bulb temperature depressions.

Without a bypass scheme for either the evaporator or air-to-air heat exchanger, the net annual energy costs include the extra annual fan power for these devices as well as the benefit of evaporative cooling.

Because less mechanical cooling is required with evaporative cooling, energy consumption and peak demand load are both reduced, yielding lower energy bills. Overall mechanical refrigeration system requirements are reduced, allowing use of smaller mechanical refrigeration systems. In some cases, the mechanical system may be eliminated. Chapter 41 of this volume and Chapter 53 of the 2019 *ASHRAE Handbook—HVAC Applications* have further information on evaporative cooling.

Example 4. Indirect Evaporative Cooling Recovery. Room air at 86°F and 63°F wb (ρ = 0.075 lb/ft^3) with a flow rate of 32,000 cfm is used to precool 32,000 cfm of supply outdoor air at 102°F and 68°F wb (ρ = 0.071 lb/ft^3) using an aluminum fixed-plate heat exchanger and indirect evaporative cooling. The evaporative cooler increases the exhaust air to 90% rh before it enters the heat exchanger. The sensible effectiveness is given by the manufacturer as 78%. Assuming EATR = 0 and OACF ≈ 1, determine the leaving supply air conditions and energy recovered, and check the energy exchange balance.

Solution:

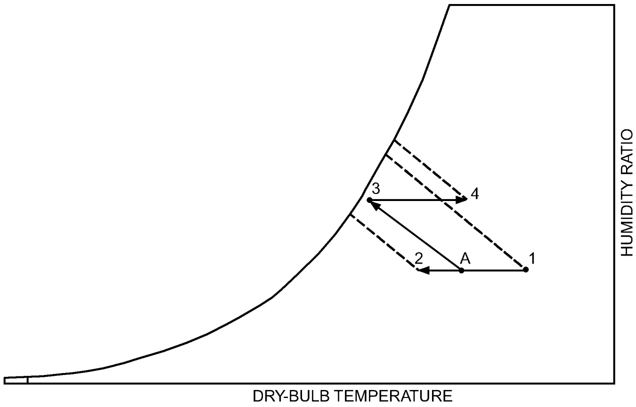

Fig. 27 Indirect Evaporative Cooling Recovery (Example 4)

First, determine the exhaust air condition entering the exchanger (i.e., after it is adiabatically cooled). Air at 86°F db, 63°F wb cools to 64°F db, 63°F wb as shown by the process line from point A to point 3 in Figure 27. In this problem the volumetric flows are equal, but the mass flows are not.

1. Because data on pressure drop are missing, skip this step.

2. Calculate the theoretical maximum heat transfer.

Based on a preliminary assessment, the supply air is not expected to cool below its wet-bulb temperature of 68°F. Thus, use the denominator of sensible heat effectiveness Equation (2a) and, for simplification, assume the specific heat of 0.24 Btu/lb·°F.

$$q_{max}\text{ (sensible)} = (60)(0.071\text{ lb/ft}^3)(32{,}000\text{ ft}^3\text{/min})(0.24\text{ Btu/lb}\cdot°\text{F}) \times (102 - 64) = 1{,}240{,}000\text{ Btu/h}$$

3. Establish the sensible effectiveness.

From manufacturer's exchanger selection data for indirect evaporative coolers, an effectiveness of 78% is found to be appropriate.

4. Calculate actual energy transfer at the design conditions.

$$q_{actual} = (0.78)(1{,}240{,}000\text{ Btu/h}) = 967{,}000\text{ Btu/h recovered}$$

5. Calculate leaving air conditions.

a. Leaving supply air temperature is

$$t_2 = 102°\text{F} + \frac{-967{,}000\text{ Btu/h}}{(60)(0.071\text{ lb/ft}^3)(32{,}000\text{ ft}^3\text{/min})(0.24\text{ Btu/lb}\cdot°\text{F})} = 72.4°\text{F}$$

b. Leaving exhaust air temperature is

$$t_4 = 64°\text{F} + \frac{967{,}000\text{ Btu/h}}{(60)(0.075\text{ lb/ft}^3)(32{,}000\text{ ft}^3\text{/min})(0.24\text{ Btu/lb}\cdot°\text{F})} = 92.0°\text{F}$$

6. Check performance.

$$q_s = (0.071\text{ lb/ft}^3)(60)(32{,}000\text{ ft}^3\text{/min})(0.24\text{ Btu/lb}\cdot°\text{F})(102 - 72.4°\text{F}) = 968{,}000\text{ Btu/h recovered}$$

$$q_e = (0.075\text{ lb/ft}^3)(60)(32{,}000\text{ ft}^3\text{/min})(0.24\text{ Btu/lb}\cdot°\text{F})(92 - 64°\text{F}) = 968{,}000\text{ Btu/h recovered}$$

7. Plot conditions on psychrometric chart (Figure 27), and confirm that no latent exchange occurred.

Because EATR = 0 and OACF is 1, Steps 8 to 10 are not presented here.

Use of Economizer

ASHRAE *Standard* 90.1 specifies the use of economizer whenever the conditions of the outdoor air, especially enthalpy, are lower than that of the return air. Typically, the use of economizer depends on factors such as the characteristics of the building envelope, number of occupants, internal loads, and the location of the building. An economizer may be required for buildings in U.S. climate zones other than 1a,1b, 2a, and 2b, provided the cooling capacity is higher

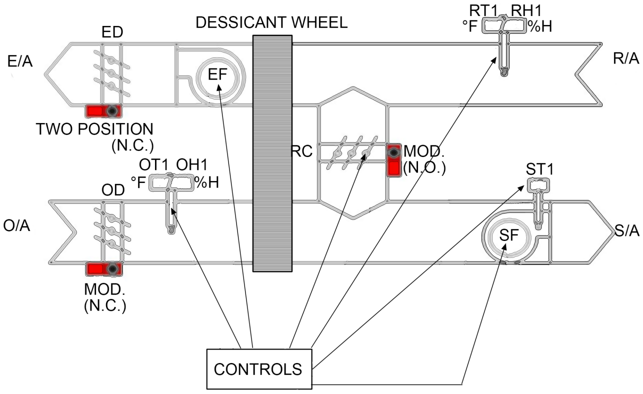

Fig. 28 Economizer with Dessicant Wheel
(Pieper 2015)

than 11 tons in the moist region and 5 tons for the dry regions. Exceptions might exist for other regions for just a few hours of the year, which may not justify the use of economizer for economic considerations. A schematic of the economizer along with energy recovery wheel as shown in Figure 28.

If the outdoor temperature appears better than that of the return air (Pieper 2015), there would be a call for the economizer to engage, but if the enthalpy of the outdoor air could be worse than that of the return air, the economizer should not be in operation. Enthalpy comparison prevents excess moisture being brought into the space. It is also important to note that not all outdoor air passes through the energy recovery device, in order to prevent excessive fan power associated with recovery device and bypass dampers. Dampers shown in Figure 28 have a wide range of flow rates such that the system may provide 100% outdoor air or less flow rate in combination with the return air through return air damper. The controls may be programmed to supply the required supply air conditions such that the mass flow rate and the enthalpy of the supply air satisfy the following equation.

$$m_s(h_r - h_s) = q_t \tag{22}$$

where

m_s = mass flow rate of the supply air, lb/h
h_r = enthalpy of the room air, Btu/lb$_{DA}$
h_s = enthalpy of the supply air, Btu/lb$_{DA}$
q_t = total building load at the time, Btu/h

Building loads vary each hour, as does the sensible heat ratio (SHR), however, for the peak load conditions, the supply air conditions are determined based on the supply air temperature (T_s) as shown in Figure 29. The letters X and Y in Figure 29 indicate the states of supply and room air, respectively, and that the SHR line AO on the protractor is parallel to the line XY.

In Figure 29, it is assumed that SHR = 0.7, however it changes by the hour. The challenge lies in the ability of the control system (shown in Figure 28) to modulate the flow rates of the outdoor air and the return air.

6. COMPARISON OF AIR-TO-AIR HEAT OR HEAT AND MASS EXCHANGER CHARACTERISTICS

It is difficult to compare different types of air-to-air energy recovery systems based on overall performance. They can be compared based on certified ratings such as sensible, latent, and total effectiveness or on air leakage parameters. Comparing them on payback period or maximum energy cost savings requires accurate values of their capital cost, life, and maintenance cost, which vary

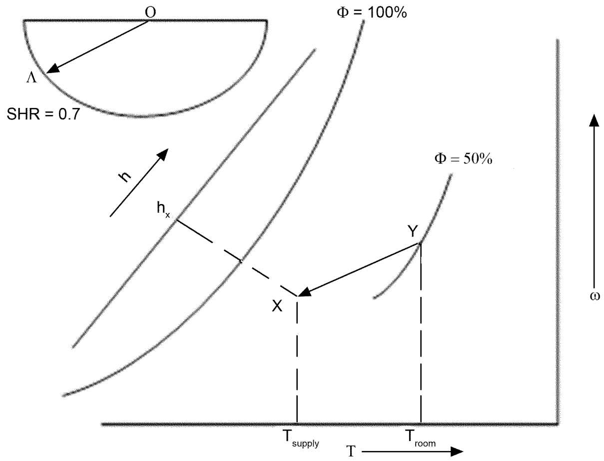

Fig. 29 Supply Conditions for a Given Load *qt* and Given SHR

from product to product for the same type of recovery system. Without such data, and considering the data available in the open literature such as that presented by Besant and Simonson (2003), use Table 3's comparative data for common types of air-to-air energy recovery devices.

7. USE OF AIR-TO-AIR HEAT OR HEAT AND MASS EXCHANGERS IN SYSTEMS

Characterizing System Efficiency of Heat or Energy Recovery Ventilators

A measure of energy recovery ventilator performance is the relative magnitude of actual energy recovered and power supplied to fans to circulate the airstreams. The cost of power supplied to the fans depends on the pressure drop of airstreams, volume flow rate, and combined efficiency of the fan motor systems. The quality of power supplied to the fans is high, and its cost per unit energy is much higher than the quality and cost of energy recovered in the ventilator. The magnitude and costs of these two forms of energy vary over the year. Besant and Simonson (2003) suggest that a parameter such as **recovery efficiency ratio (RER)** may be introduced to characterize the efficiency of the recovery ventilators:

$$\text{RER} = \frac{\int (\text{rate of energy recovered})\,dt}{\int (\text{rate of power supplied to fan motors})\,dt} \tag{23}$$

RER is similar to the energy efficiency ratio (EER) for chillers or unitary air-conditioning equipment. Besant and Simonson (2003) also suggest that the entire system performance, including the recovery ventilator, can be represented by the ratio of COP and RER.

RER is typically expressed in units of Btu/h; energy recovered can be sensible, latent, or total, in Btu/h, and energy expended is that spent for circulating exhaust and supply air through the energy recovery unit, expressed in watts. In AHRI *Guideline* V, RER$_{Total}$, RER$_{sensible}$, and RER$_{latent}$ are defined as

$$\text{RER}_{Total} = \frac{\varepsilon_{net\ total}\ m_{min}(h_1 - h_3)}{P_{blower} + P_{comp}} \tag{24a}$$

$$\text{RER}_{sensible} = \frac{\varepsilon_{net\ sensible}\ m_{min} c_p(t_1 - t_3)}{P_{blower} + P_{comp}} \tag{24b}$$

Table 3 Comparison of Air-to-Air Energy Recovery Devices

	Fixed Plate	Membrane Plate	Energy Wheel	Heat Wheel	Heat Pipe	Runaround Coil Loop	Thermosiphon	Liquid Desiccant	Fixed-Bed Regenerator
Airflow arrangements	Counterflow Cross flow[a]	Counterflow Cross flow[a]	Counterflow Parallel flow	Counterflow	Counterflow Parallel flow	—	Counterflow Parallel flow	—	Counterflow
Equipment size range, cfm	50 and up	50 and up	50 to 74,000 and up	50 to 74,000 and up	100 and up	100 and up	100 and up	—	50 and up
Typical sensible effectiveness ($m_s = m_e$), %[c]	50 to 75	55 to 75	65 to 80	65 to 80	40 to 60[b]	45 to 65[b]	40 to 60	40 to 60[b]	80 to 90[c]
Typical latent effectiveness,* %[c]	0	25 to 60	50 to 80	0	0	0	0	50 to 75[b,d]	60 to 80 with dessicant coating[c]
Total effectiveness,* %[c]	20 to 50	35 to 70	55 to 80	25 to 60	15 to 35	—	—	40 to 75[d]	50 to 80[c]
Face velocity, fpm	200 to 1000	200 to 600	500 to 1000	400 to 1000	400 to 800	300 to 600	400 to 800	300 to 450	200 to 500
Pressure drop, in. of water	0.4 to 4	0.4 to 2	0.4 to 1.2	0.4 to 1.2	0.6 to 2	0.6 to 2	0.6 to 2	0.7 to 1.2	0.2 to 1.2
EATR, %	0 to 2	0 to 5	0.5 to 10	0.5 to 10	0 to 1	0	0	0	3 to 5[c]
OACF	0.97 to 1.06	0.97 to 1.06	0.99 to 1.1	1 to 1.2	0.99 to 1.01	1.0	1.0	1.0	0.90 to 1[c]
Temperature range, °F	–75 to 1470	–40 to 140	–65 to 1470	–65 to 1470	–40 to 200	–50 to 930	–40 to 104	–40 to 115	–65 to 140
Typical mode of purchase	Exchanger only Exchanger in case Exchanger and blowers Complete system	Exchanger only Exchanger in case Exchanger and blowers Complete system	Exchanger only Exchanger in case Exchanger and blowers Complete system	Exchanger only Exchanger in case Exchanger and blowers Complete system	Exchanger only Exchanger in case Exchanger and blowers Complete system	Coil only Complete system	Exchanger only Exchanger in case	Complete system	Exchanger and damper Complete system
Advantages	No moving parts Low pressure drop Easily cleaned	No moving parts Low pressure drop Low air leakage Moisture/mass transfer	Moisture/mass transfer Compact large sizes Low pressure drop Available on all ventilati on system platforms	Compact large sizes Low pressure drop Easily cleaned	No moving parts except tilt Fan location not critical Allowable pressure differential up to 2 psi	Exhaust airstream can be separated from supply air Fan location not critical	No moving parts Exhaust airstream can be separated from supply air Fan location not critical	Latent transfer from remote airstreams Efficient microbiological cleaning of both supply and exhaust airstreams	Few moving parts Defrost strategy not required Low maintenance Easily cleaned Moisture/mass transfer if dessicant-coated
Limitations	Large size at higher flow rates	Few suppliers Long-term maintenance and performance unknown	Supply air may require some further cooling or heating Some EATR without purge	Some EATR with purge	Effectiveness limited by pressure drop and cost Few suppliers	Predicting performan ce requires accurate simulation model	Effectiveness may be limited by pressure drop and cost Few suppliers	Few suppliers Maintenance and performance unknown	Indoor units may require additional airflow selector/damper to control EATR Some EATR
Heat rate control (HRC) methods	Bypass dampers and ducting	Bypass dampers and ducting	Bypass dampers and wheel speed control	Bypass dampers and wheel speed control	Tilt angle down to 10% of maximum heat rate	Bypass valve or pump speed control	Control valve over full range	Control valve or pump speed control over full range	Bypass damper and ducting Recovery period timing

[a]Rated effectiveness values are for balanced flow conditions for cross flow. Effectiveness values increase slightly if flow rates of either or both airstreams are higher than flow rates at which testing is done.

[b]Data not based on third-party certified data.

[c]Data based on typical range of third-party certified data.

[d]Face velocity of 250 to 500 fpm.

EATR = exhaust air transfer ratio

OACF = outdoor air correction factor

$$\mathrm{RER}_{latent} = \frac{\varepsilon_{net\ latent}\ \dot{m}_{min} h_{fg}(\omega_1 - \omega_3)}{P_{blower} + P_{comp}} \tag{24c}$$

where

$\dot{m}_{min}$ = minimum mass flow rate of supply and exhaust airstreams

h_1 = enthalpy at state 1

t_1 = temperature at state 1

ω_1 = humidity ratio at state 1

P_{comp} = direct power input of recovery component

P_{blower} = blower power required for energy recovery component

$$P_{blower} = \left[\frac{Q_{blower\ supply}\Delta P_{supply}}{C\eta_{fan/motor\ supply}}\right] + \left[\frac{Q_{blower\ exhaust}\Delta P_{exhaust}}{C\eta_{fan/motor\ exhaust}}\right] \tag{25}$$

where

C = unit conversion constant, 8.52 ft^3·in. of water/min·W

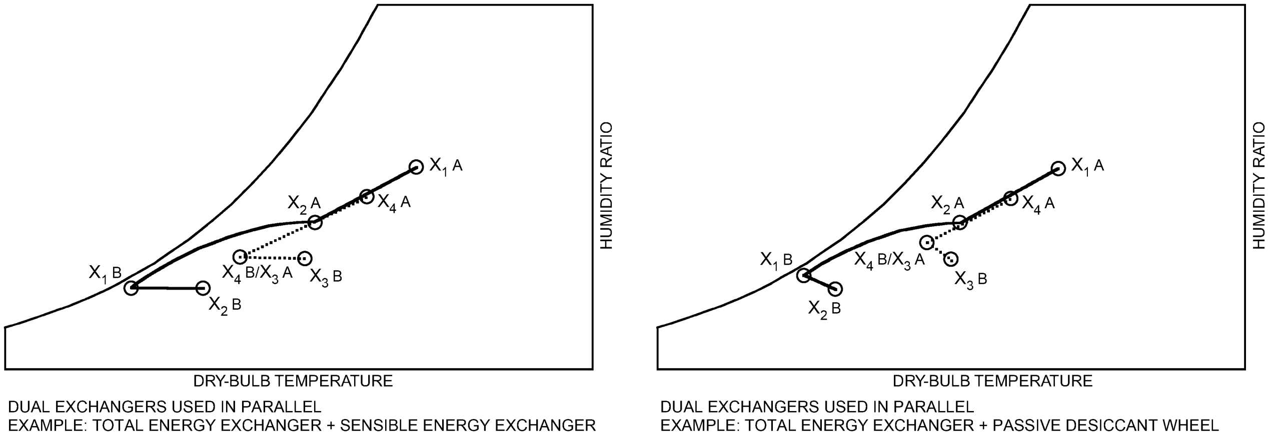

Fig. 31 Psychrometric Processes of Exchangers in Parallel Mode
(Moffitt 2011)

ΔP = pressure loss of component for supply or exhaust airstreams, in. of water
$Q_{blower\ supply}$=supply fan airflow, cfm
$Q_{blower\ exhaust}$=exhaust fan airflow, cfm
$\eta_{fan/motor}$=combined fan and motor efficiency

Note that other alternatives (e.g., comparing operating points on a fan curve) that accurately characterize the additional fan power required by the component are acceptable means of obtaining blower power.

For a flow rate of 1000 cfm and typical values of airstream pressure drop and motor and fan efficiencies, $RER_{sensible}$ can range from 25 to 30 Btu/Wh. Considering the conversion factor of 1 W = 3.413 Btu/h, the calculated value of RER suggest that the energy recovered is about 7.3 to 8.8 times that used to operate the energy recovery unit. The quality of thermal energy is about one-third that of electric power, so the actual energy recovered is about three times the cost of energy spent for energy recovery.

The **combined efficiency (CEF)** is the ratio of the net heating/cooling delivered to the total electric power consumed. CEF can also be related to the sensible, latent, or total recovered energy and is related to the RER; energy efficiency ratio (EER), which is typically provided by manufacturers of unitary equipment; and load ratio *Y*. It represents the percentage of the system load (heating, cooling, humidification and/or dehumidification) met by the energy recovery component. The higher load ratio allows use of smaller equipment, thereby affecting capital costs. $CEF_{cooling}$ can be estimated as

$$CEF_{cooling} = \frac{1}{Y_c/RER + (1 - Y_c)/EER} \tag{26}$$

where

$$Y_c = \frac{\text{Net cooling capacity of energy recovery unit}}{\text{System net cooling capacity}} \tag{27}$$

$$EER = \frac{\text{Net cooling capacity}}{\text{Total electric consumption}} \tag{28}$$

For a typical value of EER = 10 for a unitary system, and a load ratio of about 0.3, the $CEF_{cooling}$ can range from 14 to 18 Btu/Wh. Use of an energy recovery unit and its effects on system sizing and annual net savings for various climatic regions are discussed in AHRI *Guideline* V.

Selection of Heat or Energy Recovery Ventilators

Heat and energy recovery ventilators are available as heat exchangers only or as a complete system, including the heat exchangers and fan/motor systems, as indicated in Table 3. Energy recovery is also available integrated into unitary air-conditioning equipment or in both standard and custom air-handling systems. Selection of such units is primarily dictated by the quantity of ventilation air. Several manufacturers have developed software or tables to help select these units. The user may have to determine the required fan size (see Example 9), if only the heat exchanger is to be purchased.

However, the true overall system performance is the life-cycle cost, which takes into account the capital and maintenance costs. Because of lack of sufficient data on these factors, they are not presented in Table 3.

Systems with Multiple Energy Recovery Exchangers

Multiple exchangers are often used in air-handling systems that have a cooling coil, to enhance dehumidification capability. The first air-to-air exchanger in the outdoor airstream is used to recover exhaust energy and reduce the required coil capacity. The second exchanger's purpose is to increase the coil's latent removal. This second exchanger is either a sensible heat exchanger or a passive desiccant wheel. A sensible exchanger (e.g., coil loop, plate exchanger, heat pipe, wheel) reheats the air as it leaves the cooling coil. This simultaneously precools the return air. The cooled return air that is exhausted then precools the outdoor air by passing through the first exchanger. A passive dehumidification wheel works similarly: it removes water vapor from air leaving the coil and transfers the vapor, with some heat, to the return air. The second exchanger can be in parallel with the first exchanger, as shown in Figure 30 (Moffitt 2011). The corresponding processes are shown in Figure 31 (Moffitt 2011); use of a passive desiccant wheel is shown on the right side of the figure.

The second exchanger can also be placed in series with the cooling coil, as shown in Figure 32 (Moffitt 2011), where it has a similar effect. The first exchanger reduces the cooling capacity by recovery exhaust energy. The second exchanger improves latent cooling and reduces the sensible cooling delivered by the system. The corresponding processes are shown in Figure 33 (Moffitt 2011); use of a passive desiccant wheel is shown on the right side of the figure.

Using Air-to-Air Heat Exchangers to Modify the Latent Capacity Ratio of Cooling Coils

Air-to-air energy recovery can be used in a series application where heat is transferred from one location in an airstream (where the heat is detrimental) to another location in the same airstream (where the heat is beneficial). The most common example of this application is in series energy recovery around a cooling coil for

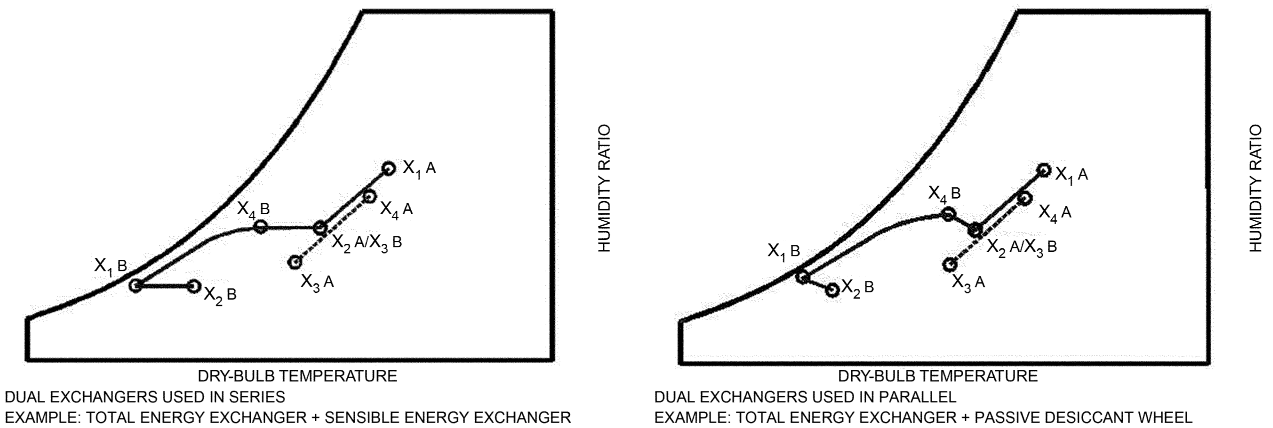

Fig. 33 Psychrometric Processes of Exchangers in Series Mode
(Moffitt 2011)

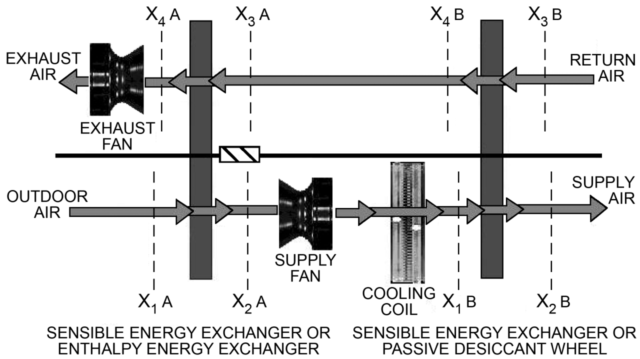

Fig. 30 Multiple Energy Recovery Exchangers in Parallel Mode

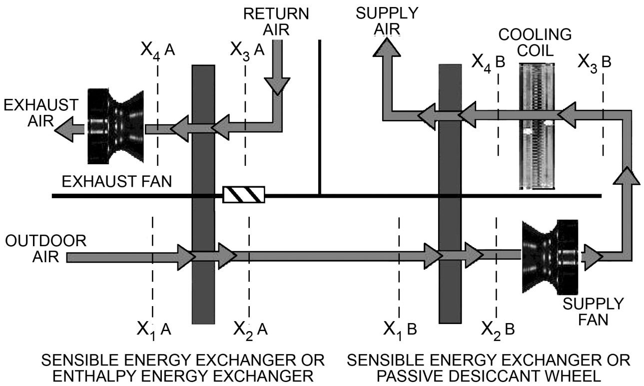

Fig. 32 Multiple Energy Recovery Exchangers in Parallel Mode
(Moffitt 2011)

summer dehumidification. During dehumidification the warm, humid outside air is cooled by a cooling coil so the dry bulb temperature reaches the desired dew point. This air is then reheated to reduce its relative humidity (for indoor air quality, IAQ), and to make it more comfortable before entering the building space. The in-series energy recovery device is placed in front of and after the cooling coil, where it provides precooling (reducing load on the cooling coil) and reheat (reducing load on the reheating equipment). Figure 34 provides an example and shows the driving force (left) and the enthalpy savings (right). In some cases, the outdoor air reaches saturation during precool, and humidity is removed before the air enters the cooling coil. The total energy consumption (kWh) and peak demand (kW) load are both reduced, yielding lower energy bills and healthier IAQ (Mathur 1997a, 1997b, 1997c, 1997d).

Since there must be energy balance between the precool side and reheat side, the changes in enthalpy will always be equal and opposite. Where there is no condensation during precool, the ΔT across the precool side will be equal and opposite to the reheat side. When condensation occurs during precool, the ΔT across the precool side will be smaller than the reheat side.

Even on direct-expansion (DX) systems where hot gas reheat is a resource created by the process, in-series energy recovery can be used to reduce compressor capacity and downsize equipment to reduce first costs, particularly where condensation takes place on more of the DX coil, which increases the cycle's coefficient of performance (COP).

Because both precooling and reheating are beneficial to the system, the RER in dehumidification applications is double that of an identical exchanger used in energy recovery. Also, the driving force is the dry-bulb temperature difference between the outdoor air and the cooling coil leaving temperature. For example, on a 90°F day and with 55°F air leaving the cooling coil, the driving force across the energy recovery device is 45°F. In contrast, a traditional air-to-air energy recovery device with outdoor air at 90°F and return air at 75°F is exposed to a driving temperature difference of only 15°F, a third of the in-series energy recovery device. Furthermore, when the energy recovery device is in economizer mode (outdoor air temperatures of about 55 to 75°F) the in-series recovery device is still recovering energy.

The in-series energy recovery device should be sized for the highest dry-bulb temperature design condition, since this is when the device is exposed to the highest driving force and creates most reheat. This ensures the system will never provide too much reheat during the year. This approach is the simplest and is used when the primary goal is to downsize cooling equipment. The system designer must recognize that when the outdoor temperature is lower, that the driving force is also lower, and the device provides less reheat. If the design reheat is required during the whole cooling season, then supplemental reheat is still required after the in-series energy recovery device, albeit less than in a system without energy recovery.

An alternative design strategy is to optimize energy recovery. In this strategy, the system designer selects a device with a higher effectiveness than in the previous strategy, and provides a means

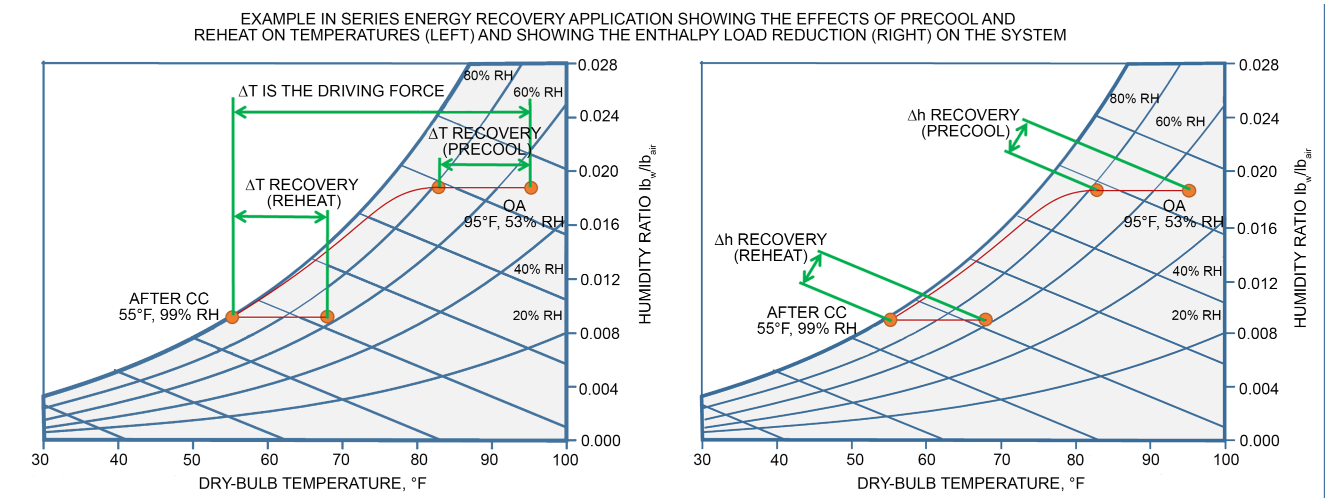

Fig. 34 Psychrometric Chart: In-Series Energy Recovery

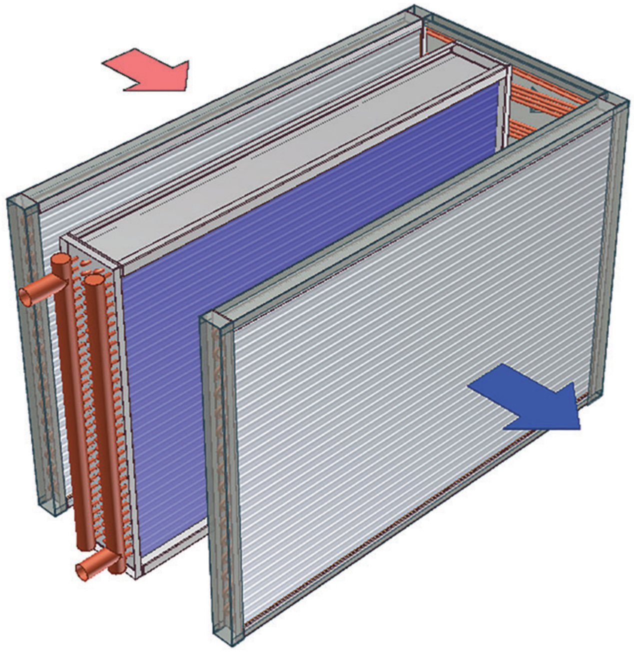

Fig. 35 Wraparound (In-Series) Heat Pipe

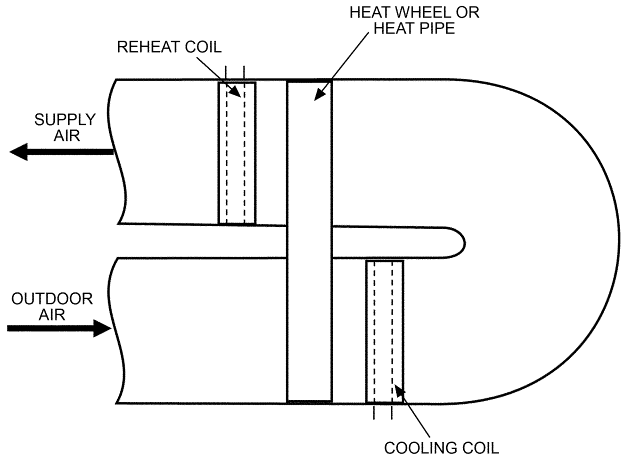

Fig. 36 Flat Heat Recovery Device in Wraparound (In-Series) Configuration

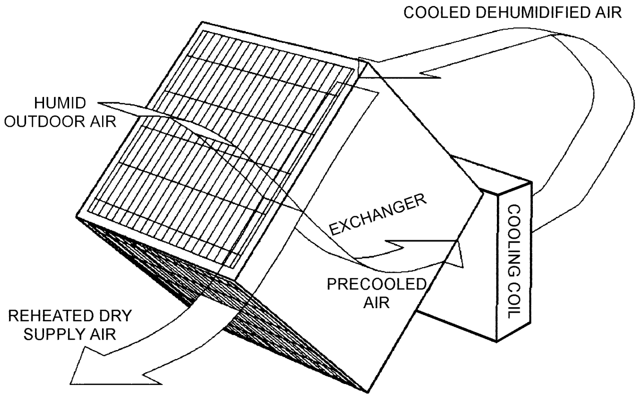

Fig. 37 Flat Plate Heat Exchanger in Wraparound (In-Series) Configuration

of modulating its performance. By modulating the performance during the warmest outdoor air conditions, excessive reheat is avoided. During cooler conditions the device operates at full performance to recover more energy. The device is usually sized with a mind to pressure drop, recognizing that the airstream passes through the device twice, and initial purchase cost and is optimized against annual energy savings.

Examples of devices are wraparound heat pipes and thermosiphons (Figures 35 and 38), flat heat exchangers like wheels and heat pipes (Figure 36), and plate heat exchangers (Figure 37). Footprint and condensation management should be considered when designing the layout. Where a controllable device is required, options include: wraparound heat pipes and thermosiphons that use valves to control refrigerant flow; bypass around the device (but not through the cooling coil); or, for wheels, modulating wheel speed.

Example 5. Precooling Air Reheater Dehumidifier. In this application, 3400 cfm of outdoor supply air at 95 and 81°Fwb (ρ = 0.069 lb/ft^3) is used to reheat 3400 cfm of the same air leaving a cooling coil (exhaust) at 52.1 and 51.8°F wb using a sensible heat exchanger as a precooling

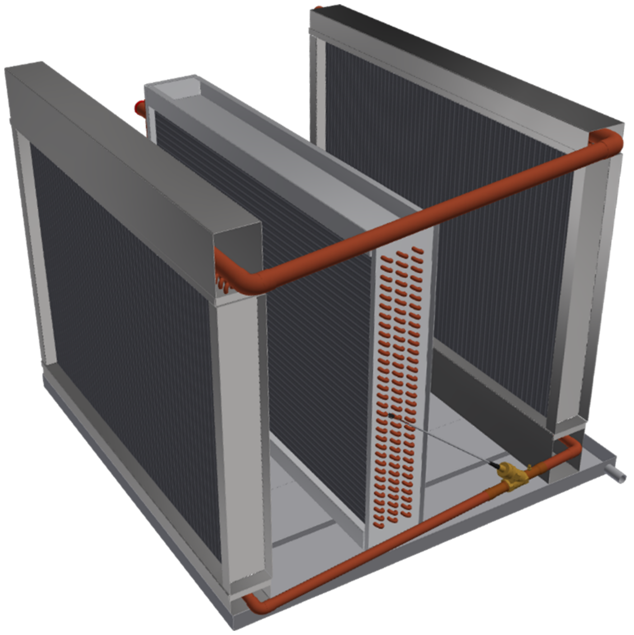

Fig. 38 Wraparound (In-Series) Thermosiphon with a Valve for Modulation

air reheater. The reheated air is to be between 75 and 79°F. In this application, the warm airstream is outdoor air and the cold airstream is the same air after it leaves the cooling coil. The unit's manufacturer lists its effectiveness as 58.4%. For EATR = 0 and OACF ≈1, determine the leaving precooled and reheated air conditions and energy recovered, and check the energy exchange balance.

Solution:

Step 1. Because data on pressure drop are missing, skip this step.

Step 2. Calculate the theoretical maximum energy transfer.

The air being reheated will have less mass than the outdoor air entering the precooler because moisture will condense from it as it passes through the precooler and cooling coil. Reheat is sensible heat only, so the denominator of Equation (2a) is used to determine the theoretical maximum energy transfer. To simplify the calculation, it is assumed that specific heat of dry air is constant at 0.24 Btu/lb$_m$·°F.

$$qs = 60(0.69)(3400)(0.24)(95 - 52.1) = 145,000 \text{ Btu/h saved}$$

Step 3. Establish the sensible effectiveness. The manufacturer gives the effectiveness as 58.4% at the designated operating conditions.

Step 4. Calculate actual energy transfer at design conditions.

$$q_{actual} = 0.584(145,000) = 84,700 \text{ Btu/h}$$

Step 5. Calculate leaving air conditions. Because condensation occurs as the outdoor airstream passes through the precooling side of the heat exchanger, use Equation (4e) to determine its leaving enthalpy, which is the inlet condition for the cooling coil. Sensible heat transfer Equation (5a) is used to determine the temperature of air leaving the preheat side of the heat exchanger.

a. Precooler leaving air conditions

Entering enthalpy, determined from the psychrometric chart for 95°F db and 81°F wb, is 44.6 Btu/lb.

$$h_2 = 44.6 + \frac{-84,700}{60(0.069)(3400)} = 38.6 \text{ Btu/lb}$$

The wet-bulb temperature for saturated air with this enthalpy is 75°F. This is point 2 on the psychrometric chart (Figure 39), which is near saturation. Note that this precooled air is further dehumidified by a cooling coil.

b. Reheater leaving air conditions

$$t_4 = 52.1 + \frac{84,700}{60(0.069)(3400)(0.24)} = 77.2°\text{F}$$

Entering enthalpy, determined from the psychrometric chart for 52.1°F db and 51.8°F wb, is 21.3 Btu/lb.

$$h_4 = 21.3 + \frac{84,700}{60(0.069)(3400)} = 27.3 \text{ Btu/lb}$$

The wet-bulb temperature for air with this temperature and enthalpy is 61.3°F.

Step 6. Check performance.

$$q_s = 60(0.069)(3400)(44.6 - 38.6) = 84,500 \text{ Btu/h precooling}$$

$$q_e = 60(0.069)(3400)(0.24)(77.2 - 52.1) = 84,800 \text{ Btu/h reheat}$$

Step 7. Plot conditions on psychrometric chart (Figure 39). Because EATR = 0 and OACF ≈ 1, Steps 8 to 10 are not presented here.

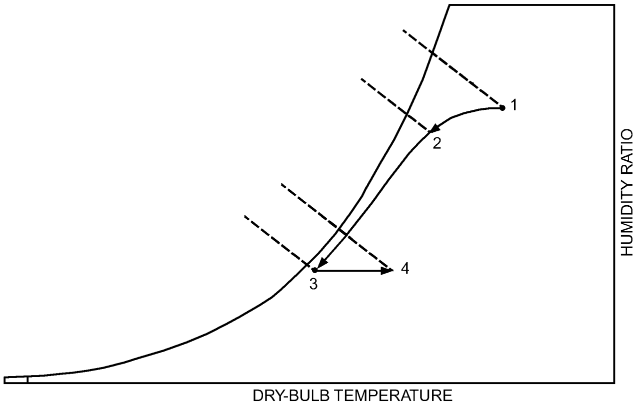

Fig. 39 Precooling Air Reheater Dehumidifier (Example 5)

Dessicant and Heat Wheel Systems

The Center for Climate and Energy Solutions' 2015 international conference on climate change emphasized reducing greenhouse gas emissions and requiring more efficient equipment, including HVAC systems (which consume significant electrical power) (C2ES 2015). Sensible and desiccant wheels can be used in combination with a heat pump direct evaporative cooler (DEC) to achieve this goal, as shown in the following cases.

The heat dissipated in the condenser of a heat pump can be used to regenerate the desiccant wheel (Sheng et al. 2015) and has a potential of significant energy cost savings. At design conditions for Atlanta, Georgia, for a 100% replacement air system as shown in Figure 40 , this approach can reduce cooling capacity requirements by 52% and net electrical shaft power requirements by more than 24% compared to the conventional vapor-compression system (Dhamshala 2016).

Figure 41 shows an advanced desiccant system for 100% outdoor supply air (OSA) mode with heat and desiccant wheels and direct evaporative cooler (DEC) in combination with a heat pump. This system could reduce cooling capacity requirements by 59% and net electrical shaft power requirements by close to 30% at the same design conditions as for Figure 40 (Dhamshala 2016).

A simplified system suitable for surgery rooms (able to provide optimum temperature and humidity control) consists of a preheater, desiccant wheel with type 3 desiccant, and a cooling coil, as shown

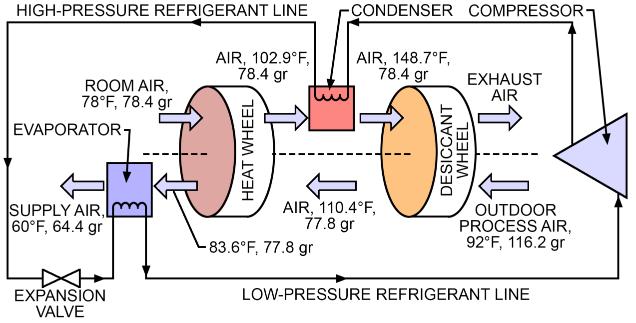

Fig. 40 Heat Pump Augmented by Heat and Dessicant Wheels

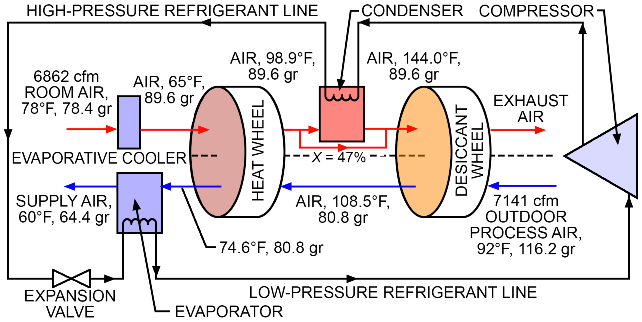

Fig. 41 Heat Pump Augmented by Evaporative Cooler and Heat and Dessicant Wheels

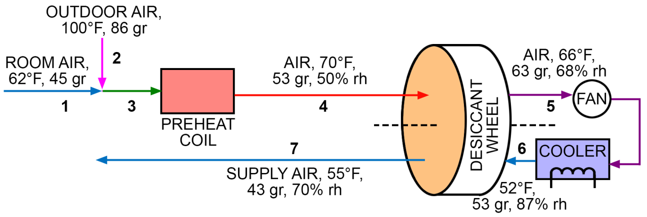

Fig. 42 System for Surgery Room
(based on concepts presented in Murphy 2006)

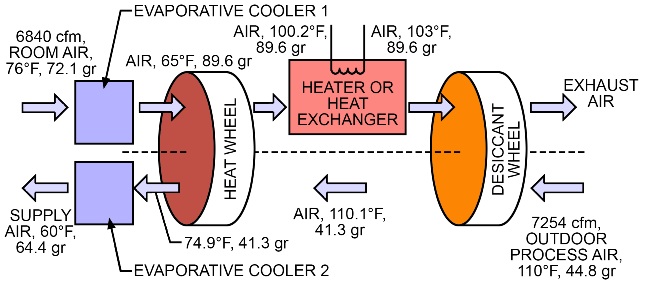

Fig. 43 Dessicant System of Evaporative Coolers and Heat and Dessicant Wheels

in Figure 42 . The type 3 desiccant in the desiccant wheel has a large affinity to adsorb water vapor when the relative humidity of the air is high (as it is in states 6 and 7), whereas it gives off moisture at states 4 and 5 (which have lower humidity). The simplicity of this system not only reduces maintenance costs but can also save significant energy costs.

A desiccant system consisting of two evaporative coolers and heat and desiccant wheels for a 100% replacement air application, as shown in Figure 43 , could provide all the space summer cooling needs without the use of an air conditioner. Such a system can reduce net electrical power requirements by more than 90% at design conditions for the city of Phoenix, Arizona (Dhamshala 2016). The only electrical power required is for the fan power and rotation of the wheels. The magnitude of estimated electrical power savings should be of similar range for the entire dry (B) region of the ASHRAE climatic zones, with small differences.

Note that the minuscule heat energy required (6.15 kW) in the heater can easily be obtained from an air-to-air heat exchanger using outdoor air (which, at 103°F, exceeds the minimum temperature) before it enters the desiccant wheel. The cost of water supplied to the evaporative coolers is estimated to be negligible.

Accurate estimates of energy cost savings over a year can be obtained through detailed computer simulations of systems for a specific building with given building loads using the hourly weather data. Highly specialized desiccant systems using a DOAS can be designed to serve the buildings with specific loads or needs (Moffitt 2015). Sultan et al. 2015 provide an overview of various widely used solid-desiccant air systems.

In the case that outdoor air requirements are less than 100%, use of direct evaporative coolers adds more humidity into the supply air. An alternative to replace the direct evaporative cooler is the indirect evaporative cooler (M-cycle), as shown in Figure 44 for the cooling season, which has a potential to save 32% of cooling capacity, while the system shown in Figure 45 (suitable for winter heating) has a potential to save 22% of heating costs.

These systems are particularly suitable for sustainable systems, where the emphasis is placed on the use of renewable energy resources such as solar, wind, and bioenergy. A PV/T panel system in combination with water-to-water heat pump, absorption chiller, and thermal storage can provide the energy requirements for cooling and heating seasons.

However, these systems are also suitable if renewable energy resources are not available at the site. The heat energy required by the heater placed between the heat and desiccant wheels to regenerate the air can be obtained from the condenser of the air-conditioner, similar to the system shown in Figure 14.

Sustainability. Recently, due to climate change, greater emphasis has been on reducing the greenhouse gas emissions (UN Climate Change Report; UNFCCC 2018), and on the rapid adoption of renewable energy resources. In light of the urgent need, the systems shown in Figures 44 and 45 pave the way to accommodate the use of renewable energy resources to meet the future goals of the HVAC systems to be compatible with smart grid applications. The desiccant and heat wheels shown in Figures 44 and 45 and not only perform the energy recovery from the outgoing exhaust air but also meet the peak loads of the building through proper selection of flow rates, mixing of outdoor air with exhaust air, and regenerating temperature. Several other options of these variations may be possible to suit the level of regenerating temperatures.

Ge et al. (2015), Tu et al. (2015), and Zeng et al. (2014) find that two-stage desiccant systems can be designed to use low-grade heat energy resources, such as waste energy at the site or solar collectors, which can yield greater energy cost savings.

To best use existing energy resources available at a specific temperature or grade, perform a series of second-law analyses on a variety of desiccant systems (including a two-stage unit) to find the maximum energy cost savings and to identify the components of the system that have largest exergy destruction (Enteria et al. 2013; Liu et al. 2016; Tu et al. 2015a, 2015b).

An emerging technology combining liquid-desiccant solutions (e.g., lithium chloride) with an evaporative cooler has the potential to save significant energy costs (Gao et al. 2015). A simplified schematic suitable for use in various climates is shown in Figure 46.

Note that the minuscule heat energy required (6.15 kW) in the heater can easily be obtained from an air-to-air heat exchanger using outdoor air (which, at 103°F, exceeds the minimum

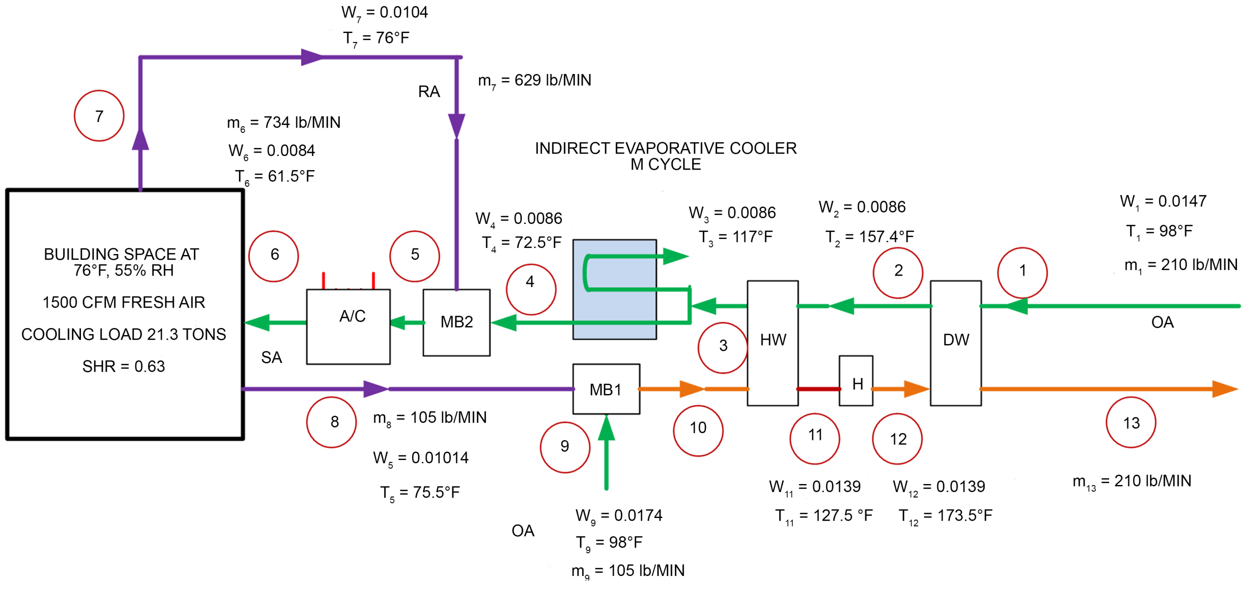

Fig. 44 Dessicant and Heat Wheels with Indirect Evaporative Cooler (M-Cycle)
Dhamshala et al. (2019)

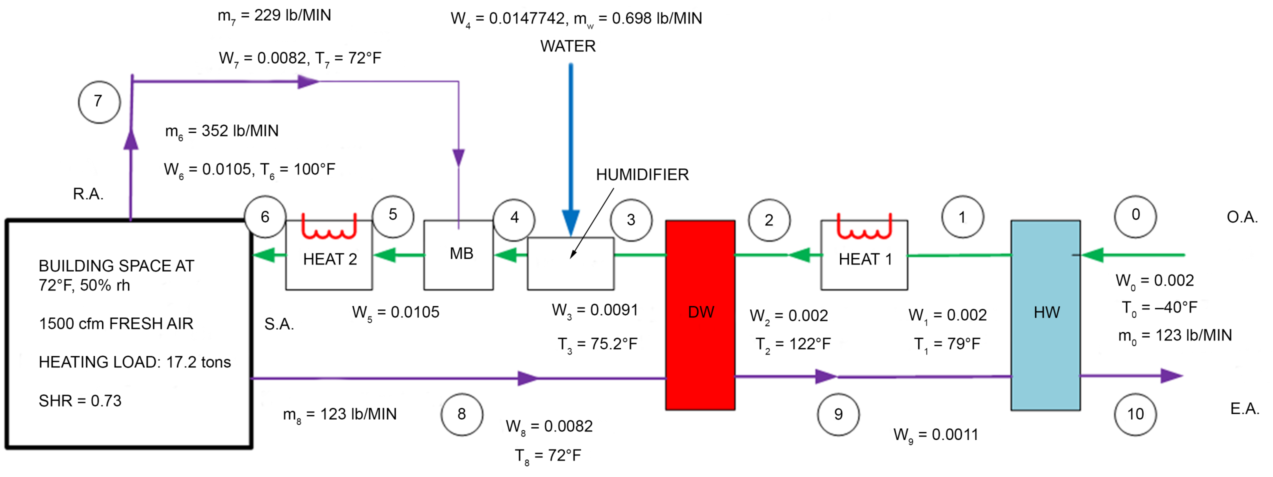

Fig. 45 Dessicant and Heat Wheels with Humidifier for Winter Heating
Dhamshala et al. (2019)

temperature) before it enters the desiccant wheel. The cost of water supplied to the evaporative coolers is estimated to be negligible.

Heat energy is required to regenerate the liquid desiccant solution; it can use waste heat energy, if available at the site, or use solar collectors. A combination of heat and desiccant wheels with this liquid-desiccant system might provide an economical system that is environmentally friendly.

8. ECONOMIC CONSIDERATIONS

Air-to-air energy recovery systems are used in both new and retrofit applications. These systems should be designed for the maximum cost benefit or least life-cycle cost (LCC) expressed either over the service life or annually and with an acceptable payback period.

The annualized system owning, operating, and maintenance costs are discussed in Chapter 38 of the 2019 *ASHRAE Handbook—HVAC Applications*. Although the capital cost and interest term in this method imply a simple value, it is in fact a complex function of the future value of money as well as all the design variables in the energy/heat exchanger. These variables include the mass of each material used, cost of forming these materials into a highly effective energy/heat exchanger, cost of auxiliary equipment and controls, and cost of installation.

The **operating energy cost** for energy recovery systems involves functions integrated over time, including variables such as flow rate, pressure drop, fan efficiency, energy cost, and energy recovery rate. The calculations are complex because air-heating and/or cooling loads are, for a range of supply temperatures, time dependent in most buildings. Time-of-use schedules for buildings can impose different ventilation rates for each hour of the day. Electrical utility charges often vary with time of day, amount of energy used, and peak power load. For building ventilation air-heating applications, the peak heat recovery rate usually occurs at the outdoor supply temperature at which frosting control throttling must be imposed. In addition to designing for the winter design temperature, heat recovery systems

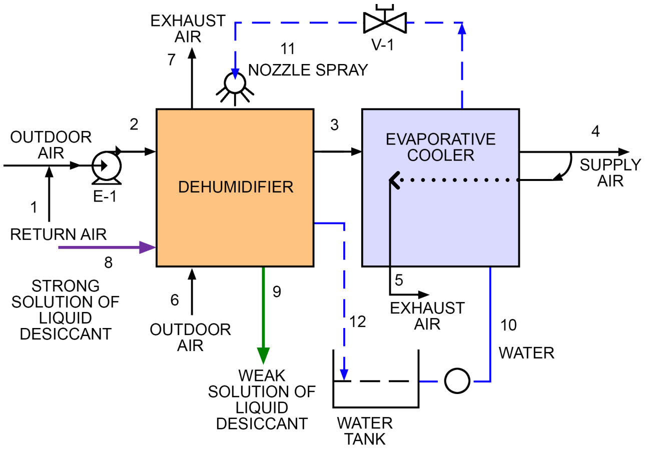

Fig. 46 Schematic of Liquid Dessicant System and Evaporative Cooler
NREL

should be optimized for peak heat recovery rate, taking frost control into account.

Overall exchanger effectiveness ε should be high (see Table 3 for typical values); however, a high ε implies a high capital cost, even when the exchanger is designed to minimize the amount of materials used. Energy costs for fans and pumps are usually very important and accumulate operating cost even when the energy recovery system must be throttled back. For building ventilation, throttling may be required much of the time. Thus, the overall LCC minimization problem for optimal design may involve 10 or more independent design variables as well as several specified constraints and operating conditions (see, e.g., Besant and Johnson [1995]).

In addition, comfort-to-comfort energy recovery systems often operate with much smaller temperature differences than most auxiliary air-heating and cooling heat exchangers. These small temperature differences need more accurate energy transfer models to reach the maximum cost benefit or lowest LCC. Most importantly, recovered energy at design may be used to reduce the required capacity of heating and cooling equipment, which can be significant in both system performance/efficiency and economics.

The **payback period (PP)** is best computed after the annualized costs have been evaluated. It is usually defined as

$$\begin{aligned} PP &= \frac{\text{Capital cost and interest}}{\text{Annual operating enery cost saved}} \\ &= \frac{C_{s,init} - \text{ITC}}{C_e(1 - T_{inc})}\text{CRF}(i'', n) \end{aligned} \tag{29}$$

where

$C_{s,\,init}$ = initial system cost
ITC = investment tax credit for energy-efficient improvements
C_e = cost of energy to operate the system for one period
T_{inc} = net income tax rate where rates are based on last dollar earned (i.e., marginal rates) = (local + state + federal rate) – (federal rate) (local + state rate)
CRF = capital recovery factor
i'' = effective discount rate adjusted for energy inflation
n = total number of periods under analysis

The inverse of this term is usually called the **return on investment (ROI).** Well-designed energy recovery systems normally have a PP of less than 5 years, and often less than 3 years. Paybacks of less than 1 year are not uncommon in comfort-to-comfort applications in hot, humid climates, primarily because of the reduced size of cooling equipment required.

Other economic factors include the following.

System Installed Cost. Initial installed HVAC system cost is often lower for air-to-air energy recovery devices because mechanical refrigeration and fuel-fired heating equipment can be reduced in size. Thus, a more efficient HVAC system may also have a lower installed total HVAC cost. The installed cost of heat recovery systems becomes lower per unit of flow as the amount of outdoor air used for ventilation increases.

Life-Cycle Cost. Air-to-air energy recovery cost benefits are best evaluated considering all capital, installation, operating, and energy-saving costs over the equipment life under normal operating conditions in terms of the life-cycle cost. As a rule, neither the most efficient nor the least expensive energy recovery device will be most economical. Optimizing the life-cycle cost for maximum net savings may involve many design variables, requiring careful cost estimates and use of an accurate recovery system model with all its design variables (see, e.g., Besant and Simonson [2000]).

Energy Costs. The absolute cost of energy and relative costs of various energy forms are major economic factors. High energy costs favor high levels of energy recovery. In regions where electrical costs are high relative to fuel prices, heat recovery devices with low pressure drops are preferable.

Amount of Recoverable Energy. Economies of scale favor large installations. Equipment is commercially available for air-to-air energy recovery applications using 50 cfm and above.

Grade of Exhaust Energy. High-grade (i.e., high-temperature) exhaust energy is generally more economical to recover than low-grade energy. Energy recovery is most economical for large temperature differences between the waste energy source and destination.

Coincidence and Duration of Waste Heat Supply and Demand. Energy recovery is most economical when supply coincides with demand and both are relatively constant throughout the year. Thermal storage may be used to store energy if supply and demand are not coincident, but this adds cost and complexity to the system.

Proximity of Supply to Demand. Applications with a large central energy source and a nearby waste energy use are more favorable than applications with several scattered waste energy sources and uses.

Operating Environment. High operating temperatures or the presence of corrosives, condensable gases, and particulates in either airstream results in higher equipment and maintenance costs. Increased equipment costs result from the use of corrosion- or temperature-resistant materials, and maintenance costs are incurred by an increase in the frequency of equipment repair and wash down and additional air filtration requirements.

Effect on Pollution Control Systems. Removing process heat may reduce the cost of pollution control systems by (1) allowing less expensive filter bags to be used, (2) improving the efficiency of electronic precipitators, or (3) condensing out contaminant vapors, thus reducing the load on downstream pollution control systems. In some applications, recovered condensable gases may be returned to the process for reuse.

Effect on Heating and Cooling Equipment. Heat recovery equipment may reduce the size requirements for primary utility equipment such as boilers, chillers, and burners, as well as the size of piping and electrical services to them. Larger fans and fan motors (and hence fan energy) are generally required to overcome increased static pressure loss caused by the energy recovery devices. Auxiliary heaters may be required for frost control.

Effect on Humidifying or Dehumidifying Equipment. Selecting total energy recovery equipment results in the transfer of moisture from the airstream with the greater humidity ratio to the airstream with the lesser humidity ratio. This is desirable in many situations because humidification costs are reduced in cold weather and dehumidification loads are reduced in warm weather.

9. ENERGY AND/OR MASS RECOVERY CALCULATION PROCEDURE

The rate of energy transfer to or from an airstream depends on the rate and direction of heat transfer and water vapor (moisture) transfer. Under customary design conditions, heat and water vapor transfer will be in the same direction, but the rate of heat transfer often will not be the same as the rate of energy transfer by the cross-stream flow of water vapor. This is because the driving potentials for heat and mass transfer are different, as are the respective wall resistances for the two types of transport. Both transfer rates depend on exchanger construction characteristics.

The following general procedure may be used to determine the performance and energy recovered in air-to-air energy recovery applications at each operating condition.

Step 1. Determine supply and exhaust air pressure drops Δp_s and Δp_e across exchanger.

Request air pressure drops Δp_s and Δp_e across the heat or energy exchanger from the manufacturer, who may have certified AHRI *Standard* 1060 test condition data obtained using ASHRAE *Standard* 84 as a test procedure and analysis guide. These data may be extrapolated to non-AHRI conditions by the manufacturer using correlations such as Equations (12) or (13), if their restrictions are satisfied. For other flow conditions, somewhat different correlations may be more accurate to determine the pressure drop.

Step 2. Calculate theoretical maximum moisture and energy transfer rates m_{max} and q_{max}.

The airstream with the lower mass flow m_{min} limits heat and moisture transfer. Some designers specify and prefer working with airflows stated at standard temperature and pressure conditions. To correctly calculate moisture or energy transfer rates, the designer must determine mass flow rates. For this reason, the designer must know whether airflow rates are quoted for the entry conditions specified or at standard temperature and pressure conditions. If necessary, convert flow rates to mass flow rates (e.g., scfm to lb/min) and then determine which airstream has the minimum mass.

The theoretical maximum sensible heat, latent heat and total energy rates are given by the denominators of Equations (2a), (2b), and (2c), respectively.

The split between latent and sensible energy can be determined by plotting airstream conditions on a psychrometric chart as shown in Figure 47. Maximum sensible heat transfer is represented by a horizontal line drawn between the two dry-bulb temperatures, and maximum latent energy transfer is represented by the vertical line.

Step 3. Establish moisture, sensible, and total effectiveness ε_s, ε_L, and ε_t.

Each of these ratios is obtained from manufacturers' product data using input conditions and airflows for both airstreams. The effectiveness for airflows depends on (1) exchanger construction, including configuration, heat transfer material, moisture transfer properties, transfer surface area, airflow path, distance between heat transfer surfaces, and overall size; and (2) inlet conditions for both airstreams, including pressures, velocities, temperatures, and humidities. In applications with unequal airflow rates, the enthalpy change will be higher for the airstream with the lesser mass flow.

Step 4. Calculate actual moisture (latent) and energy (sensible, latent or total) transfer rates.

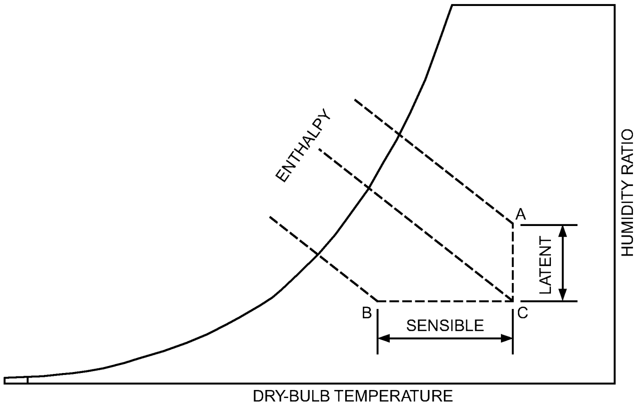

Fig. 47 Maximum Sensible and Latent Heat from Process A-B

The actual moisture, sensible heat, latent heat, and total energy rates are given by Equations (5a), (5b), (5c), and (5d or 5e), respectively.

Step 5. Calculate leaving air properties for each airstream using Equations (4a), (4b), (4c), (4d), (4e), and (4f).

With an enthalpy or moisture-permeable heat exchanger, moisture (and its inherent latent energy) is transferred between airstreams. With a sensible-only heat exchanger, if the warmer airstream is cooled below its dew point, the resulting condensed moisture transfers additional energy. When condensation occurs, latent heat is released, maintaining that airstream at a higher temperature than if condensation had not occurred. This higher air temperature (potential flux) increases the heat transfer to the other airstream. The sensible and total effectiveness are widely used because the energy flow in the condensate is relatively small in most applications. (Freezing and frosting are unsteady conditions that should be avoided unless a defrost cycle is included.) Equations (4c) and (4d) must be used to calculate the leaving air humidity conditions, and Equations (4e) and (4f) to calculate the enthalpy values for airstreams in which inherent latent energy transfer occurs. Equations (4a) and (4b) may be used for airstreams if only sensible energy transfer is involved.

Step 6. Check energy transfer balance between airstreams.

Equation (5a) can be used to estimate the sensible energy transfer rate into the supply airstream, and can be adapted to estimate the energy transfer rate out of the exhaust airstream by substituting T_3 for T_1, and substituting T_4 for T_2. Equation (5d) or (5e) can estimate the total energy for the two airstreams, using similar substitutions. Total energy transferred from one airstream should equal total heat transferred to the other. Calculate and compare the energy transferred to or from each airstream. Differences between these energy flows are usually because of measurement errors.

Step 7. Plot entering and leaving conditions on psychrometric chart.

Examine the plotted information for each airstream to verify that performance is reasonable and accurate.

(*Steps 8 to 10 apply only when EATR ≠ 0 and OACF ≠ 1.*)

Step 8. Obtain data on exhaust air transfer ratio (EATR > 0 and typically 0.05 > EATR > 0 for regenerative wheels).

Request the EATR data from the manufacturer, who may have certified AHRI *Standard* 1060 test condition data obtained using ASHRAE *Standard* 84 as a test procedure and analysis guide. These data may be extrapolated to non-AHRI test conditions using correlations relating EATR to air pressure differences between the supply and exhaust and, for rotary regenerative wheels, carryover caused by wheel rotation. Shang et al. (2001a) show that, for regenerative wheels, a correlation may be developed between EATR and carryover ratio, R_c, and OACF, but for other air-to-air exchangers EATR will be very small or negligible.

Step 9. Obtain data on outdoor air correction factor (OACF ≈ 1 and typically 0.9 < OACF < 1.1 for regenerative wheels).

Request the OACF data from the manufacturer, who may have certified AHRI *Standard* 1060 test condition data obtained using ASHRAE *Standard* 84 as a test procedure and analysis guide. These data may be extrapolated to non-AHRI test conditions using correlations relating OACF to pressure differences (Shang et al. 2001b), for regenerative wheels; for other exchangers, OACF will be very nearly 1.0.

Step 10. Correct the supply air ventilation rate, moisture transfer rate, and energy transfer rates for EATR ≠ 0 and OACF ≠ 1.0.

Values of EATR significantly larger than zero and OACF significantly different than 1.0 imply the air-to-air exchanger is transferring air between the exhaust and supply airstreams. This transfer may be important, especially for some devices such as regenerative wheels. Shang et al. (2001b) show a method to correct the energy rates when EATR ≠ 0 and OACF ≠ 1. The procedure to correct the supply air ventilation rate is shown in Example 9.

Example 6. Sensible Heat Recovery in Winter. Exhaust air at 75°F and 10% rh with a flow rate of 800 lb/min preheats an equal mass flow rate of outdoor air at 0°F and 60% rh ($\rho = 0.087$ lb/ft^3) using an air-to-air heat exchanger with a measured effectiveness of 60%. Airflows are specified in scfm, so an air density of 0.075 lb/ft^3 for both airstreams is appropriate. Assuming EATR = 0 and OACF ≈ 1, determine the leaving supply air temperatures and energy recovered, and check the heat exchange balance. To simplify the calculation, assume that specific heat of dry air is constant at 0.24 Btu/lb$_m$·°F

Solution:

Note: the numbers correspond to the steps in the calculation procedure.

1. Because data on pressure drop are missing, skip this step.

2. Calculate the theoretical maximum heat transfer.

The two inlet conditions plotted on a psychrometric chart (Figure 48) indicate that, because the exhaust air has low relative humidity, latent energy transfer does not occur. Using the denominator of Equation (2a), the theoretical maximum sensible heat transfer rate q_s is

$$q_{max} = (60)(800\ \text{lb/min})(0.24\ \text{Btu/lb}\cdot°\text{F})(75 - 0) = 864{,}000\ \text{Btu/h}$$

3. Establish the sensible effectiveness.

From manufacturer's literature and certified performance test data, effectiveness is determined to be 60% at the design conditions.

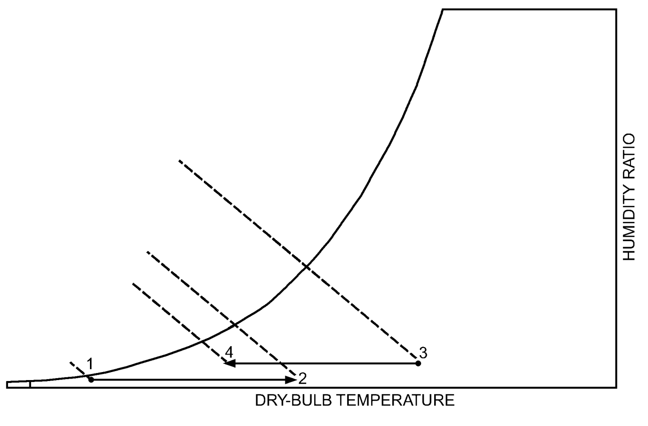

Fig. 48 Sensible Heat Recovery in Winter (Example 6)

4. Calculate actual heat transfer at given conditions.

Using Equation (5a),

$$q_s = (0.6)(864{,}000\ \text{Btu/h}) = 518{,}400\ \text{Btu/h}$$

5. Calculate leaving air conditions.

Because no moisture or latent energy transfer will occur,

a. Leaving supply air temperature t_2 is given as

$$t_2 = 0°\text{F} + \frac{518{,}400\ \text{Btu/h}}{(60)(800\ \text{lb/min})(0.24\ \text{Btu/lb}\cdot°\text{F})} = 45°\text{F}$$

b. Leaving exhaust air temperature t_4 is given as

$$t_4 = 75°\text{F} - \frac{518{,}400\ \text{Btu/h}}{(60)(800\ \text{lb/min})(0.24\ \text{Btu/lb}\cdot°\text{F})} = 30°\text{F}$$

6. Using Equation (5a), check performance.

$$q_s = (60)(800\ \text{lb/min})(0.24\ \text{Btu/lb}\cdot°\text{F})(45 - 0) = 518{,}400\ \text{Btu/h saved}$$

7. Plot conditions on psychrometric chart to confirm that no moisture exchange occurred (Figure 48).

Because EATR = 0 and OACF ≈ 1, Steps 8 to 10 of the calculation procedure are not presented here.

Example 7. Sensible Heat Recovery in Winter with Water Vapor Condensation. Exhaust air at 75°F and 28% rh ($\rho = 0.075$ lb/ft^3) and flow rate of 10,600 cfm is used to preheat 9500 cfm of outdoor air at 14°F and 50% rh ($\rho = 0.084$ lb/ft^3) using a heat exchanger with a sensible effectiveness of 70%. Assuming EATR = 0 and OACF ≈ 1, determine the leaving supply air conditions and energy recovered, and check the energy exchange balance.

Solution:

The supply airstream has a lower airflow rate than the exhaust airstream, so it may appear that the supply airstream limits heat transfer. However, determination of mass flow rates for the given entry conditions shows that the mass flow rate of the supply airstream (47,900 lb/h) is slightly greater than that of the exhaust airstream (47,600 lb/h), so exhaust is the limiting airstream. Nevertheless, because the mass difference is negligible, it is convenient to use supply air volume as the limiting airstream. Also, to simplify the calculation, assume that specific heat of dry air is constant at 0.24 Btu/lb$_m$·°F.

1. Because data on pressure drop are missing, skip this step.

2. Calculate the theoretical maximum sensible heat transfer.

The limiting airstream, the supply airstream, will be preheated in the heat exchanger, so it is not subject to condensation. Therefore, the denominator of Equation (2a) is used:

$$q_{max} = (60)(9500\ \text{ft}^3/\text{min})(0.084\ \text{lb/ft}^3)(0.24\ \text{Btu/lb}\cdot°\text{F})(75 - 14)$$
$$= 700{,}000\ \text{Btu/h}$$

3. Select sensible effectiveness.

From manufacturer's literature and performance test data, the sensible effectiveness is determined to be 70% at the design conditions.

4. Calculate actual heat transfer at design conditions using Equation (5a):

$$q_s = (0.7)(700{,}000\ \text{Btu/h}) = 490{,}000\ \text{Btu/h}$$

5. Calculate leaving air conditions.

a. Leaving supply air temperature is calculated by

$$t_2 = 14°\text{F} + \frac{490{,}400\ \text{Btu/h}}{(60)(0.084\ \text{lb/ft}^3)(9500\ \text{ft}^3/\text{min})(0.24\ \text{Btu/lb}\cdot°\text{F})} = 56.6°\text{F}$$

b. Because the dew point of exhaust air at inlet is 38.4°F, condensation occurs on the exhaust side, so the leaving exhaust air temperature cannot be determined using Equation (4b). The entering exhaust air enthalpy and humidity ratio are determined for the dry-bulb temperature of 75°F and 28% rh using a psychrometric chart and found to be $h_3 = 23.7$ Btu/lb and $w_3 = 0.0052$ lb/lb. However, the leaving exhaust air enthalpy can be determined by

$$h_4 = 23.7\ \text{Btu/lb} - \frac{490{,}000\ \text{Btu/h}}{(60)(0.075\ \text{lb/ft}^3)(10{,}600\ \text{ft}^3/\text{min})} = 13.43\ \text{Btu/lb}$$

Because the air will be saturated at the outlet of exhaust air, the dry-bulb or wet-bulb temperature and humidity ratio corresponding to an enthalpy of 13.43 Btu/lb is found to be t_4 = 35.7°F and w_4 = 0.0044 lb/lb. The rate of moisture condensed m_w is

$$m_w = m_e (w_3 - w_4) = (60)(800 \text{ lb/min})(0.0052 - 0.0044) = 38.4 \text{ lb/h}$$

6. Check performance.

$$q_s = (60)(0.084 \text{ lb/ft}^3)(9500 \text{ ft}^3\text{/min})(0.24 \text{ Btu/lb}\cdot°\text{F})(56.6 - 14) = 489{,}500 \text{ Btu/h}$$

Neglect the enthalpy of the condensed water by adding the energy lost through condensation of vapor to the sensible heat lost of the exhausting air.

$$q_e = (60)(0.075 \text{ lb/ft}^3)(10{,}600 \text{ ft}^3\text{/min})(0.24 \text{ Btu/lb}\cdot°\text{F})(75 - 34.5) + (38.4 \text{ lb/h})(1100 \text{ Btu/lb})$$
$$= 505{,}900 \text{ Btu/h saved, which is very close to } 489{,}500 \text{ Btu/h}$$

7. Plot conditions on psychrometric chart (Figure 49). Note that moisture condenses in the exhaust side of the heat exchanger.

Because EATR = 0 and OACF ≈ 1, Steps 8 to 10 of the calculation procedure are not presented here.

Example 8. Total Heat Recovery in Summer. Exhaust air at 75°F and 63°F wb (ρ = 0.073 lb/ft^3) with a flow rate of 10,600 cfm is used to precool 8500 cfm of supply outdoor air at 95°F and 81°F wb (ρ = 0.069 lb/ft^3) using a hygroscopic total energy exchanger. The sensible and total effectiveness for this heat exchanger are 70 and 56.7%, respectively. Assuming EATR = 0 and OACF ≈ 1, determine the leaving supply air conditions and energy recovered, and check the energy exchange balance. To simplify the calculation, assume that specific heat of dry air is constant at 0.24 Btu/lb$_m$·°F.

Solution:

1. Because data on pressure drop are missing, skip this step.
2. Calculate the theoretical maximum heat transfer.

The supply airstream is a lesser or limiting airstream for energy and moisture transfer. Determine entering airstream enthalpies and humidity ratio from psychrometric chart.

Supply inlet (95°F db, 81°F wb) h_1 = 44.6 Btu/lb w_1 = 0.0198 lb/lb

Exhaust inlet (75°F db, 63°F wb) h_3 = 28.5 Btu/lb w_3 = 0.0096 lb/lb

The theoretical maximum sensible and total heat transfer rates can be obtained as follows:

$$q_{max}\text{(sensible)} = (60)(0.069 \text{ lb/ft}^3)(8500 \text{ ft}^3\text{/min})(0.24 \text{ Btu/lb}\cdot°\text{F}) \times (95 - 75) = 169{,}000 \text{ Btu/h}$$

$$q_{max}\text{(total energy)} = (60)(0.069 \text{ lb/ft}^3)(8500 \text{ ft}^3\text{/min})(44.6 - 28.5) = 566{,}000 \text{ Btu/h}$$

3. Determine supply sensible and total effectiveness.

The manufacturer's selection data for the design conditions provide the following effectiveness ratios:

$$\varepsilon_s = 70\% \qquad \varepsilon_t = 56.7\%$$

4. Calculate energy transfer at design conditions.

$$q_t = (0.567)(566{,}000 \text{ Btu/h}) = 321{,}000 \text{ Btu/h total recovered}$$
$$q_s = -(0.7)(169{,}000 \text{ Btu/h}) = \underline{-118{,}000 \text{ Btu/h sensible recovered}}$$
$$q_{lat} = 203{,}000 \text{ Btu/h latent recovered}$$

5. Calculate leaving air conditions.

a. Supply air conditions

$$t_2 = 95°\text{F} + \frac{-118{,}000 \text{ Btu/h}}{(60)(0.069 \text{ lb/ft}^3)(8500 \text{ ft}^3\text{/min})(0.24 \text{ Btu/lb}\cdot°\text{F})} = 81°\text{F}$$

$$h_2 = 44.6 \text{ Btu/lb} + \frac{-321{,}000 \text{ Btu/h}}{(60)(0.069 \text{ lb/ft}^3)(8500 \text{ ft}^3\text{/min})} = 35.5 \text{ Btu/lb}$$

From the psychrometric chart, the supply air humidity ratio and wet-bulb temperature are found to be w_2 = 0.0145 and t_{w2} = 71.6°F.

b. Exhaust air conditions

$$t_4 = 75°\text{F} + \frac{118{,}000 \text{ Btu/h}}{(60)(0.073 \text{ lb/ft}^3)(10{,}000 \text{ ft}^3\text{/min})(0.24 \text{ Btu/lb}\cdot°\text{F})} = 86.2°\text{F}$$

$$h_4 = 28.5 \text{ Btu/lb} + \frac{321{,}000 \text{ Btu/h}}{(60)(0.073 \text{ lb/ft}^3)(10{,}600 \text{ ft}^3\text{/min})} = 35.4 \text{ Btu/lb}$$

From the psychrometric chart, the exhaust humidity ratio and wet-bulb temperature are found to be w_4 = 0.0134 and t_{w4} = 71.6°F.

6. Check total performance (Equation 5d).

$$q_t = (60)(0.069 \text{ lb/ft}^3)(8500 \text{ cfm})(44.6 - 35.5) = 320{,}000 \text{ Btu/h saved}$$

$$q_t = (60)(0.073 \text{ lb/ft}^3)(10{,}600 \text{ cfm})[(0.24 \text{ Btu/lb}\cdot°\text{F})(86.2 - 75) + (0.0134 - 0.0096)(1100 \text{ Btu/lb})]$$
$$= 319{,}000 \text{ Btu/h, which is close to } 320{,}000 \text{ Btu/h}$$

7. Plot conditions on psychrometric chart (Figure 50).

Because EATR = 0 and OACF ≈ 1, Steps 8 to 10 are not presented here.

Example 9. Total Energy Recovery with EATR ≠ 0 and OACF ≠ 1.0. An ERV manufacturer claims a product has performance characteristics as shown here at 850 ft^3/min:

Pressure drop Δp = 0.9 in. of water
$\varepsilon_{sensible}$ = 0.73
ε_{latent} = 0.68
ε_{total} = 0.715

A building has a ventilation requirement of 850 cfm and exhaust air at 75°F and 63°F wb (ρ = 0.075 lb/ft^3) is used to precool supply outdoor air at 95°F and 81°F wb (ρ = 0.072 lb/ft^3).

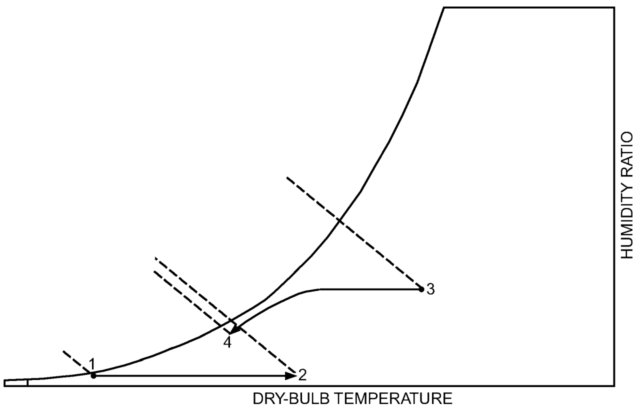

Fig. 49 Sensible Heat Recovery in Winter with Condensate (Example 6)

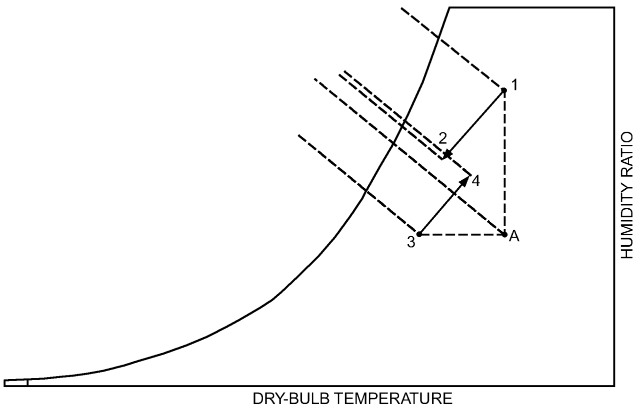

Fig. 50 Total Heat Recovery in Summer (Example 7)

(a) Assuming EATR = 0 and OACF ≈ 1, determine the leaving supply air conditions and energy recovered, and check the energy exchange balance.

(b) Assuming EATR = 5% and OACF = 1.05, determine the actual airflow rates.

In both cases, to simplify the calculation, assume that specific heat of dry air is constant at 0.24 Btu/lb$_m$·°F.

Solution:

1. From the manufacturer's claims, at a flow rate of 850 cfm, the pressure drop Δp = 0.9 in. of water. Assuming the effective efficiency of the fan motor combination is about 0.6, the power P_s required to circulate the supply air can be obtained from Equation (8) as

$$P_s = [(850\ \text{ft}^3/\text{min})(0.9)]/[(6356)(0.6)] = 0.2\ \text{hp}$$

Assuming the balanced flow the power required to circulate the exhaust air would be same, therefore the total power P required to circulate the airstreams would be twice this amount.

$$P = 0.4\ \text{hp}$$

2. Calculate the theoretical maximum heat transfer.

Determine entering airstream enthalpies and humidity ratio from the psychrometric chart.

Supply inlet (95°F db, 81°F wb) h_1 = 44.6 Btu/lb w_1 = 0.0198 lb/lb

Exhaust inlet (75°F db, 63°F wb) h_3 = 28.5 Btu/lb w_3 = 0.0096 lb/lb

The theoretical maximum heat transfer rates can be obtained as follows:

$$q_{max}(\text{sensible}) = (60)(0.072\ \text{lb/ft}^3)(850\ \text{ft}^3/\text{min})(0.24\ \text{Btu/lb}\cdot°\text{F}) \times (95 - 75) = 17{,}600\ \text{Btu/h}$$

$$q_{max}(\text{latent}) = (60)(0.072\ \text{lb/ft}^3)(850\ \text{ft}^3/\text{min})(11{,}000\ \text{Btu/lb}\cdot°\text{F}) \times (0.0194 - 0.0093) = 40{,}800\ \text{Btu/h}$$

$$q_{max}(\text{total}) = (60)(0.072\ \text{lb/ft}^3)(850\ \text{ft}^3/\text{min}) \times (36.6 - 20.6) = 58{,}750\ \text{Btu/h}$$

Note that sum of sensible and latent energy should equal the total energy.

3. Determine supply sensible and total effectiveness.

From the manufacturer's claims, at a flow rate of 850 cfm, ε_s = 0.73, ε_L = 0.68, and
ε_t = 0.715.

4. Calculate energy transfer at design conditions.

$$q_s = (0.73)(17{,}600\ \text{Btu/h}) = 12{,}850\ \text{Btu/h sensible recovered}$$

$$q_L = (0.68)(40{,}800\ \text{Btu/h}) = 27{,}750\ \text{Btu/h latent recovered}$$

$$q_t = (0.715)(58{,}750\ \text{Btu/h}) = 42{,}000\ \text{Btu/h total recovered}$$

5. Calculate leaving air conditions.

a. Supply air conditions

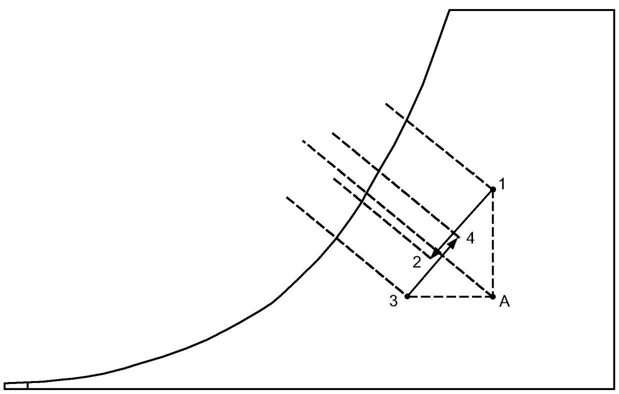

Fig. 51 Total Energy Recovery with EATR ≠ 0 and OACF ≠ 1 (Example 8)

$$t_2 = 95°\text{F} + \frac{-12{,}850\ \text{Btu/h}}{(60)(0.072\ \text{lb/ft}^3)(850\ \text{ft}^3/\text{min})(0.24\ \text{Btu/lb}\cdot°\text{F})} = 80°\text{F}$$

$$h_2 = 44.6\ \text{Btu/lb} + \frac{-42{,}000\ \text{Btu/h}}{(60)(0.072\ \text{lb/ft}^3)(850\ \text{ft}^3/\text{min})} = 33.16\ \text{Btu/lb}$$

From the psychrometric chart, the supply air humidity ratio and wet-bulb temperature are w_2 = 0.0129 and t_{w2} = 69.2°F.

b. Exhaust air conditions

$$t_4 = 75°\text{F} + \frac{12{,}850\ \text{Btu/h}}{(0.075\ \text{lb/ft}^3)(60)(850\ \text{ft}^3/\text{min})(0.24\ \text{Btu/lb}\cdot°\text{F})} = 89°\text{F}$$

$$h_4 = 28.5\ \text{Btu/lb} + \frac{42{,}000\ \text{Btu/h}}{(0.075\ \text{lb/ft}^3)(60)(850\ \text{ft}^3/\text{min})} = 39.5\ \text{Btu/lb}$$

From the psychrometric chart, the exhaust humidity ratio and wet-bulb temperature are found to be w_4 = 0.0164, t_{w4} = 76.1°F.

6. Check total performance.

$$q_t = (0.072\ \text{lb/ft}^3)(60)(850\ \text{ft}^3/\text{min})(36.6 - 25.1) = 42{,}230\ \text{Btu/h saved}$$

$$q_t = (0.075\ \text{lb/ft}^3)(60)(850\ \text{ft}^3/\text{min})[(0.24\ \text{Btu/lb}\cdot°\text{F})(89 - 75) + (0.0164 - 0.0096)(1100\ \text{Btu/lb})]$$
$$= 41{,}460\ \text{Btu/h, which is close to } 42{,}230\ \text{Btu/h}$$

7. Plot conditions on psychrometric chart (Figure 51).

8. Obtain data on EATR. (Given: EATR = 5% or 0.05.)

9. Obtain data on OACF. (Given: OACF = 1.05.)

10. Correct the supply air ventilation rate, the moisture transfer rate, and energy transfer rates EATR ≠ 0 and OACF ≠ 1.0.

The net ventilation rate is 850 cfm and the EATR = 0.05; therefore, the actual flow rate Q_2 to the space can be obtained from Equation (19) as

$$Q_3 = Q_2 = \frac{Q_v}{1 - (\text{EATR}/100)} = \frac{850\ \text{ft}^3/\text{min}}{1 - 5/100} = 895\ \text{cfm}$$

Because OACF = 1.05, the actual flow rate Q_1 of fresh air from outdoor can be calculated from Equation (21) as

$$Q_1 = Q_2(\text{OACF})\frac{Q_v(\text{OACF})}{1 - (\text{EATR}/100)} = \frac{(850\ \text{ft}^3/\text{min})(1.05)}{1 - 5/100} = 939\ \text{cfm}$$

To balance the flow rates into the ERV, the actual air flow rates at states 3 and 4 are as shown in Figure 52.

Supply and exhaust fan capacity should match the flows required at their locations. Example 9's results are for balanced flow. Assuming flow rates in the ERV are same as the outdoor air ventilation requirements, then the effectiveness would be same as that for no air leakage. If air leaks at the inlet and outlet of the energy recovery ventilator, then the exit conditions of air temperature and humidity at states 3 and 4 can be calculated as those of the airstream mixture. For instance, the properties at state 2 would be those of an airstream mixture at state 2 for no air leakage and air

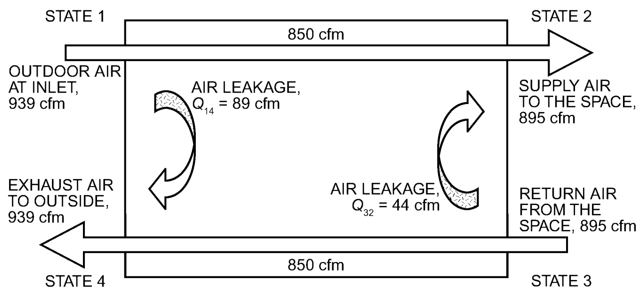

Fig. 52 Actual Airflow Rates at Various State Points (Example 8)

quantity (Q_{32}) at state 3. The error for these calculations should be less than 5%.

Shang et al. (2001a) show a method to accurately estimate the energy rates when EATR ≠ 0 and OACF ≠ 1.

10. SYMBOLS

A = area of recovery exchanger, ft^2
c_p = specific heat of moist air, Btu/lb ·°F
C = capital cost
C_e = cost of energy
C_p = specific heat of air Btu/lb·°F
C_r = ratio of C_{min}/C_{max}
$C_{s,int}$ = initial system capital cost
CRF = capital recovery factor
h = enthalpy, Btu/lb
h_{fg} = enthalpy of vaporization, Btu/lb
i = arbitrary state, or discount rate
ITC = income tax credit
m_s = mass flow rate of supply moist air from outdoors, lb/min
m_e = mass flow rate of exhaust moist air, lb/min
m_{min} = minimum mass flow rate of supply and exhaust airstreams, lb/min
n = number of years in economic consideration, years
NTU = number of transfer units = UA/C_{min}
p = pressure, in. of water
pp = payback period, years
P = pumping power, Btu/h
q = heat transfer rate, Btu/h
Q = volume flow rate, cfm
S = reference temperature, °F
t = moist air temperature at state i, °F
t_{w3} = wet-bulb temperature of moist air at state 3, °F
T = absolute temperature, °F, or tax
U = overall heat transfer coefficient, Btu/h·ft^2·°F
V = mean velocity, fpm
w = humidity ratio

Greek Letters

ε_s = sensible effectiveness of heat or energy wheel
ε_L = latent effectiveness of energy wheel
ε_t = total effectiveness of ERV
η = efficiency
ρ = density, lb/ft^3
σ = volume fraction
ϕ = relative humidity
ω = rotational speed of the wheel, rpm

Subscripts

a = air
e = exhaust side of heat/energy exchanger, exit or energy
f = fan or fan motor combination
if = threshold temperature of the outdoor air for freezing to occur
in = indoor conditions of building space
inc = increment
h = hydraulic
L = latent
max = maximum value
min = minimum value
n = station number indicating supply and exhaust inlets and outlets (see Figure 1)
o = reference state or outlet
p = constant pressure
s = supply side or suction side
t = total

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