

CHAPTER 26. AIR-TO-AIR ENERGY RECOVERY EQUIPMENT

AIR-TO-AIR energy recovery is the process of recovering heat or/and moisture between two airstreams at different temperatures and humidities. This process is important in maintaining acceptable indoor air quality (IAQ) while maintaining low energy costs and reducing overall energy consumption. 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.

Energy can be recovered either in its sensible (temperature only) or latent (moisture) form, or 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 much 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.

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](#).

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

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

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.

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 or/and precooling of outdoor supply air.

2. BASIC THERMODYNAMICS

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 ERV facilitates this transfer 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. 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_e . 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. Exit properties of these two streams can be estimated, knowing the flow rates and the effectiveness of the heat exchanger.

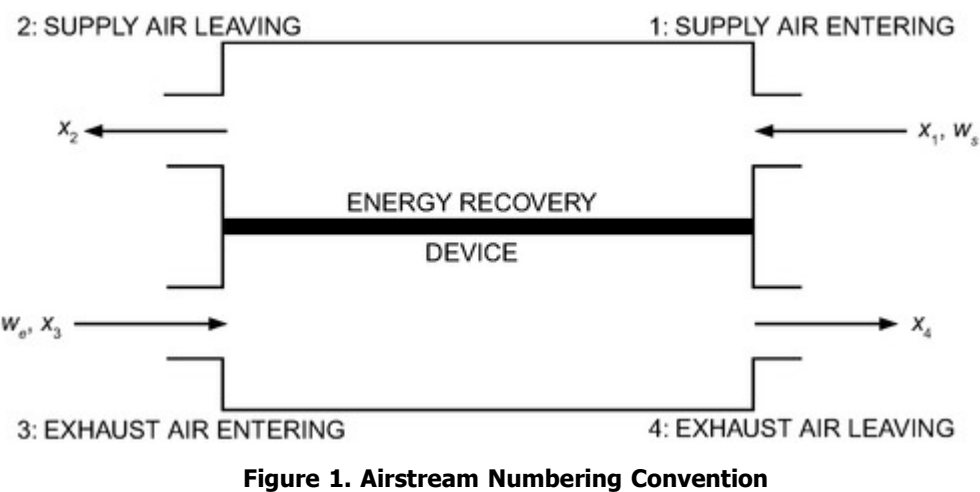


Figure 1. Airstream Numbering Convention

ASHRAE Standard 84 defines effectiveness as

$$\epsilon = \frac{\text{Actual transfer of moisture or energy}}{\text{Maximum possible transfer between airstreams}} \tag{1}$$

Thermodynamics of Heat Recovery Ventilators

From [Figure 1](#), the sensible effectiveness ϵ_s of a heat recovery ventilator is given as

$$\epsilon_s = \frac{q_s}{q_{s,max}} = \frac{m_s c_{ps} (t_2 - t_1)}{C_{min} (t_3 - t_1)} = \frac{m_e c_{pe} (t_3 - t_4)}{C_{min} (t_3 - t_1)} \tag{2a}$$

where q_s is the actual sensible heat transfer rate given by

$$q_s = \epsilon_s q_{s,max} \tag{2b}$$

where $q_{s,max}$ is the maximum sensible heat transfer rate given by

$$q_{s,max} = 60 C_{min} (t_3 - t_1) \tag{2c}$$

where

- q_s = sensible heat transfer rate, Btu/h
 - $q_{s,max}$ = maximum sensible heat transfer rate, Btu/h
 - ϵ_s = sensible effectiveness
 - t_1 = dry-bulb temperature at location 1 in [Figure 1](#), °F
 - m_s = supply dry air mass flow rate, lb/min
 - m_e = exhaust dry air mass flow rate, lb/min
 - C_{min} = smaller of $C_{ps} m_s$ and $C_{pe} m_e$
 - C_{ps} = supply moist air specific heat at constant pressure, Btu/lb · °F
 - C_{pe} = exhaust moist air specific heat at constant pressure, Btu/lb · °F
- Assuming no water vapor condensation in the HRV, the leaving supply air condition is

$$t_2 = t_1 - \epsilon_s \frac{C_{min}}{m_s c_{ps}} (t_1 - t_3) \tag{3a}$$

and the leaving exhaust air condition is

$$t_4 = t_3 + \epsilon_s \frac{C_{min}}{m_e c_{pe}} (t_1 - t_3) \tag{3b}$$

[Equations \(2\), \(3a\), and \(3b\)](#) assume steady-state operating conditions; 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. Furthermore, condensation or frosting does not occur or is negligible. These assumptions are generally nearly true for larger commercial HRV applications. Note that the HRV only allows transfer of sensible heat energy associated with heat transfer because of temperature difference between the airstreams or between an airstream and a solid surface. These equations apply even in winter, if there is no condensation in the HRV.

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

$$q_s = 60 m_s c_{ps} (t_2 - t_1) = 60 Q_s \rho_s c_{ps} (t_2 - t_1) \tag{3c}$$

$$q_s = 60 m_e c_{pe} (t_4 - t_3) = 60 Q_e \rho_e c_{pe} (t_4 - t_3) \tag{3d}$$

$$q_s = 60 \epsilon_s m_{min} c_p (t_1 - t_3) \tag{3e}$$

where

- Q_s = volume flow rate of supply air, cfm
 - Q_e = volume flow rate of exhaust air, cfm
 - ρ_s = density of dry supply air, lb/ft³
 - ρ_e = density of dry exhaust air, lb/ft³/kg/m³
 - t_1, t_2, t_3, t_4 = inlet and exit temperatures of supply and exhaust airstreams, respectively
 - m_{min} = smaller of m_s and m_e
- Because C_{ps} and C_{pe} are nearly equal, these terms may be omitted from [Equations \(1\) to \(3e\)](#).
- Sensible heat exchangers (HRVs) can be used in virtually all cases, especially for swimming pool, paint booth, and reheat applications. [Equations \(1\) to \(3e\)](#) apply for both HRVs and ERVs with appropriate selection of x_1, x_2, x_3, x_4 .

Thermodynamics of Energy Recovery Ventilators

The ERV allows the transfer of both sensible and latent heat, the latter because of the difference in water vapor pressures between the airstreams or between an airstream and a solid surface. ERVs are available as desiccant rotary wheels and also as membrane plate exchangers; although other gases may also pass through the membrane (Sparrow et al. 2001a) of membrane plate energy exchangers, it is assumed in the following equations that only the water vapor is allowed to pass through the membrane.

From [Figure 1](#), assuming no condensation in the ERV, the latent effectiveness ϵ_L of an energy recovery ventilator is given as

$$\epsilon_L = \frac{q_L}{q_{L,max}} = \frac{m_s h_{fg} (w_1 - w_2)}{m_{min} h_{fg} (w_1 - w_3)} = \frac{m_e h_{fg} (w_4 - w_3)}{m_{min} h_{fg} (w_1 - w_3)} \tag{4a}$$

where q_L is the actual latent heat transfer rate given by

$$q_L = \epsilon_L q_{L,max} \tag{4b}$$

where $q_{L,max}$ is the maximum heat transfer rate given by

$$q_{L,max} = 60 m_{min} h_{fg} (w_1 - w_3) \tag{4c}$$

where

- ϵ_L = latent effectiveness
 - h_{fg} = enthalpy of vaporization, Btu/lb
 - w = humidity ratios at locations indicated in [Figure 1](#)
 - m_s = supply dry air mass flow rate, lb/min
 - m_e = exhaust dry air mass flow rate, lb/min
 - m_{min} = smaller of m_s and m_e
- Because the enthalpy of vaporization from [Equation \(4a\)](#) can be dropped out from numerator and denominator, [Equation \(4a\)](#) can be rewritten as

$$\epsilon_m = \frac{m_w}{m_{w,max}} = \frac{m_s (w_1 - w_2)}{m_{min} (w_1 - w_3)} = \frac{m_e (w_4 - w_3)}{m_{min} (w_1 - w_3)} \tag{4d}$$

where ϵ_m is moisture effectiveness, numerically equal to latent effectiveness ϵ_L , and m_w is actual moisture transfer rate given by

$$m_w = \epsilon_m m_{w,max} \tag{4e}$$

where $m_{w,max}$ is the maximum moisture transfer rate given by

$$m_{s,max} = m_{w,min} (w_1 - w_3) \tag{4f}$$

Assuming no water vapor condensation in the ERV, the leaving humidity ratios can be given as follows. The supply air leaving humidity ratio is

$$w_2 = w_1 - \epsilon_L \frac{m_{w,min}}{m_s} (w_1 - w_3) \tag{5a}$$

and the leaving exhaust air humidity ratio is

$$w_4 = w_3 + \epsilon_L \frac{m_{w,min}}{m_e} (w_1 - w_3) \tag{5b}$$

The total effectiveness ϵ_t of an energy recovery ventilator is given as

$$\epsilon_t = \frac{q_t}{q_{t,max}} = \frac{m_s (h_2 - h_1)}{m_{min} (h_3 - h_1)} = \frac{m_e (h_3 - h_4)}{m_{min} (h_3 - h_1)} \tag{6a}$$

where q_t is the actual total energy transfer rate given by

$$q_t = \epsilon_t q_{t,max} \tag{6b}$$

where $q_{t,max}$ is the maximum total energy transfer rate given by

$$q_{t,max} = 60 m_{min} (h_1 - h_3) \tag{6c}$$

where

- ϵ_t = total effectiveness
 - h = enthalpy at locations indicated in [Figure 1](#), Btu/lb
 - m_s = supply dry air mass flow rate, lb/min
 - m_e = exhaust dry air mass flow rate, lb/min
 - m_{min} = smaller of m_s and m_e
- The leaving supply air condition is

$$h_2 = h_1 - \epsilon_t \frac{m_{min}}{m_s} (h_1 - h_3) \tag{7a}$$

and the leaving exhaust air condition is

$$h_4 = h_3 + \epsilon_t \frac{m_{min}}{m_e} (h_1 - h_3) \tag{7b}$$

Assuming the stream at state 1 is of higher humidity, the latent heat recovery q_L from the ERV can be estimated from

$$q_L = 60 m_s h_{fg} (w_1 - w_2) = 60 Q_s \rho_s h_{fg} (w_1 - w_2) \tag{8a}$$

$$q_L = 60 m_e h_{fg} (w_4 - w_3) = 60 Q_e \rho_e h_{fg} (w_4 - w_3) \tag{8b}$$

$$q_L = 60 \epsilon_L m_{min} h_{fg} (w_1 - w_3) \tag{8c}$$

where

- h_{fg} = enthalpy of vaporization or heat of vaporization of water vapor, Btu/lb
 - w_1, w_2, w_3, w_4 = inlet and exit humidity ratios of supply and exhaust airstreams, respectively
- The total energy transfer q_t between the streams is given by

$$q_t = q_s + q_L = 60 m_s (h_{1s} - h_{2s}) = 60 Q_s \rho_s (h_{1s} - h_{2s}) = 60 [m_s c_{ps} (t_1 - t_2) + m_s h_{fg} (w_1 - w_2)] \tag{9}$$

$$q_t = q_e + q_L = 60 m_e (h_{4e} - h_{3e}) = 60 Q_e \rho_e (h_{4e} - h_{3e}) = 60 [m_e c_{pe} (t_4 - t_3) + m_e h_{fg} (w_4 - w_3)] \tag{10a}$$

where

- h_{1s} = enthalpy of supply air at inlet, Btu/lb
- h_{1e} = enthalpy of exhaust air at inlet, Btu/lb
- h_{2s} = enthalpy of supply air at outlet, Btu/lb
- h_{2e} = enthalpy of exhaust air at outlet, Btu/lb

ERV's can be used where there is an opportunity to transfer heat and mass (water vapor) (e.g., humid areas, schools, offices with large occupancies). Latent energy transfer can be positive or negative depending on the direction of decreasing vapor pressure. Depending on conditions, the supply airstream flowing through an ERV may gain heat energy (+ q_s) from the adjoining stream, but lose latent energy (− q_s) if it transfers the water vapor to the adjoining stream. Heat and latent energy gain may be in the same or opposite direction. The total net energy gain is the difference between q_s and q_L , as shown in Example 1.

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%. Assuming the specific heat of air is 0.24 Btu/lb · °F and the latent heat of vaporization to be 1100 Btu/lb, determine the sensible, latent, and net energy gained by the exhaust air.

Solution:

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

$V_1 = 14.14 \text{ ft}^3/\text{lb}$ $h_1 = 30.6 \text{ Btu/lbw}_1 = 0.0071 \text{ lb/lb of dry air}$

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

$V_3 = 13.68 \text{ ft}^3/\text{lb}$ $h_3 = 28.15 \text{ Btu/lbw}_3 = 0.0093 \text{ lb/lb of dry air}$

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

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

These equal mass flow rates conform with ASHRAE *Standard* 84. Exit temperatures of the airstreams can be obtained from the [Equations \(3a\)](#) and [\(3b\)](#) as follows:

$$t_2 = 95^\circ\text{F} - 0.5 \frac{(660 \text{ lb/min})(0.24 \text{ Btu/lb}\cdot^\circ\text{F})}{(660 \text{ lb/min})(0.24 \text{ Btu/lb}\cdot^\circ\text{F})} (95^\circ\text{F} - 75^\circ\text{F}) = 85^\circ\text{F}$$
$$t_4 = 75^\circ\text{F} + 0.5 \frac{(660 \text{ lb/min})(0.24 \text{ Btu/lb}\cdot^\circ\text{F})}{(660 \text{ lb/min})(0.24 \text{ Btu/lb}\cdot^\circ\text{F})} (95^\circ\text{F} - 75^\circ\text{F}) = 85^\circ\text{F}$$

The exit humidity of the airstreams is found from [Equations \(5a\)](#) and [\(5b\)](#) as follows:

$$w_2 = 0.0071 - 0.5 \frac{(660 \text{ lb/min})}{(660 \text{ lb/min})} (0.0071 - 0.0093) = 0.0082 \text{ lb/lb of dry air}$$
$$w_4 = 0.0093 + 0.5 \frac{(660 \text{ lb/min})}{(660 \text{ lb/min})} (0.0071 - 0.0093) = 0.0082 \text{ lb/lb of dry air}$$

The sensible heat gained by the exhaust stream is found from [Equation \(3c\)](#) as

The latent heat gained by the exhaust stream is found from [Equation \(8a\)](#) as

$$q_L = (660 \text{ lb/min})(1100 \text{ Btu/lb})(0.0082 - 0.0093) = -799 \text{ Btu/min}$$

The net heat energy gained by the exhaust airstream is therefore

$$q = q_s + q_L = 1584 - 799 = 785 \text{ Btu/min}$$

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. The outlet exhaust airstream enthalpy at 85°F and 0.0082 lb/lb of dry air is given in the psychrometric chart as 29.4 Btu/lb. The net heat gained by the exhaust airstream [found from [Equation \(10\)](#)] is close to 945 Btu/min.

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

$$P_s = Q_s \Delta p_s / 6356 \eta_f \tag{11}$$

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

$$P_e = Q_e \Delta p_e / 6356 \eta_f \tag{12}$$

where

- P_s = fan power for supply air, hp
- P_e = fan power for exhaust air, hp
- 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

However, the density and viscosity of air vary with the temperature. The variation of viscosity with temperature is given by the Sutherland law as

$$\frac{\mu}{\mu_o} = \left(\frac{T}{T_o} \right)^{3/2} \left(\frac{T_o + S}{T + S} \right) \tag{13}$$

where

- T = absolute temperature, °F
- T_o = reference temperature, °F
- S = constant = 198.7°F

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

$$\frac{\Delta p}{\Delta p_o} = \left(\frac{m}{m_o} \right)^{1.75} \left(\frac{T}{T_o} \right)^{1.375} \left(\frac{T_o + S}{T + S} \right)^{0.25} \tag{14}$$

[Equation \(14\)](#) is only accurate when the Reynolds number Re_D for airflow through the exchanger is in the range

$$5 \times 10^3 \leq Re_D \leq 10^5$$

where

$$Re_D = \frac{(\rho V)_a D_h}{\mu_T} \tag{15}$$

where

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

For fully developed laminar flow through an energy exchanger (e.g., an energy wheel), the corresponding dimensionless pressure drop relation similar to [Equation \(14\)](#) is given as

$$\frac{\Delta p}{\Delta p_o} = \left(\frac{m}{m_o} \right) \left(\frac{\bar{T}}{\bar{T}_o} \right)^{3/2} \left(\frac{1 + C/\bar{T}_o}{1 + C/\bar{T}} \right) \tag{16}$$

where $Re_D < 2000$.
[Equations \(14\)](#) and [\(16\)](#) 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

$$P = P_s + P_e \tag{17}$$

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.

3. AIRFLOW ARRANGEMENTS

Heat exchanger effectiveness depends heavily on the airflow direction and pattern of the supply and exhaust airstreams. Parallel-flow exchangers ([Figure 2A](#)), in which both airstreams move along heat exchange surfaces in the same direction, have a theoretical maximum effectiveness of 50%. Counterflow exchangers ([Figure 2B](#)), 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 2C](#)) and 60 to 85% for multiple-pass exchangers ([Figure 2D](#)).

In practice, construction limitations favor designs that use transverse flow (or cross-flow) over much of the heat exchange surface ([Figures 2C](#) and [2D](#)).

Effectiveness

Heat or energy exchange effectiveness as defined in [Equation \(1\)](#) 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 \(3c\)](#), [\(8c\)](#), and [\(10a\)](#), 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.

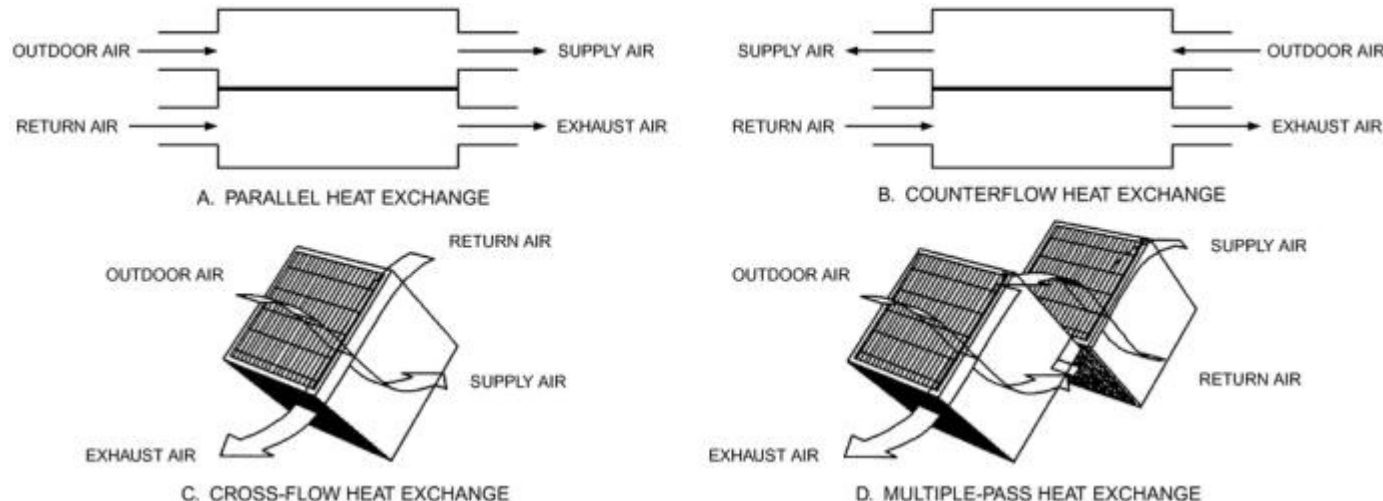


Figure 2. Heat Exchanger Airflow Configurations

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 .

$$NTU = UA/60C_{min} \tag{18}$$

$$C_r = C_{min}/C_{max} \tag{19}$$

where

- U = overall heat transfer coefficient, related to flow rates and dimensions of fluid flow path in heat exchanger, Btu/h · ft² · °F
- A = area of heat exchanger, ft²
- C_{max} = maximum of m_1c_{p1} and m_2c_{p2}

See [Figure 9](#) for the variation of effectiveness with NTU for a rotary heat wheel. Variation of effectiveness is similar for plate exchangers.

4. 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 actually is.

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 3](#). 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.

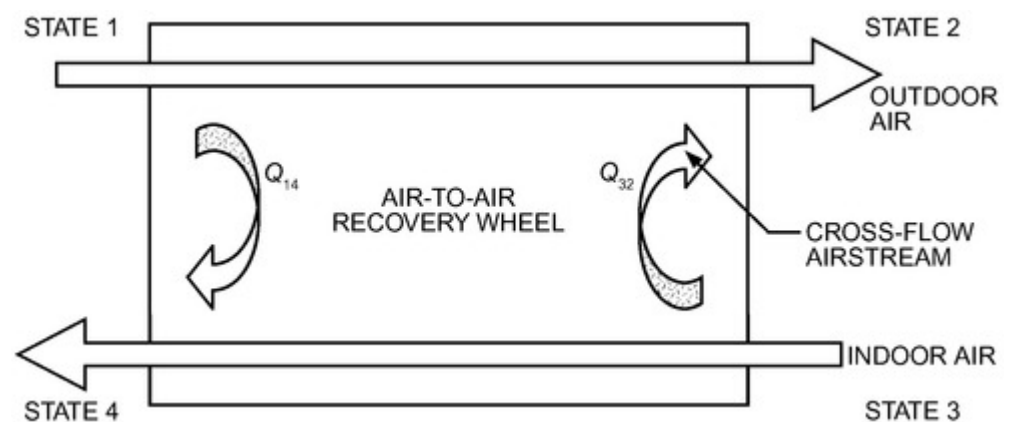


Figure 3. Air Leakage in Energy Recovery Units

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):

$$EATR = \frac{c_2 - c_1}{c_3 - c_1} \tag{20}$$

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.

$$OACF = \frac{m_1}{m_2} \tag{21}$$

where m_1 and m_2 are the mass flow rates of the incoming fresh airstream at state 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_1 of the supply fan is greater than Q_2 if air leakage exists (see [Figure 3](#)).

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 can be estimated by the equation

$$Q_{32} = Q_2 \left(\frac{\text{EATR}}{100} \right)$$

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

$$Q_2 = Q_v + Q_{32} = Q_v + Q_2 \left(\frac{\text{EATR}}{100} \right)$$

This may be simplified as

$$Q_2 = \frac{Q_v}{\left(1 - \frac{\text{EATR}}{100} \right)}$$

which gives the quantity of air entering the space. Assuming steady-state conditions, balanced flow through the energy recovery ventilator, the density corrected quantity of air leaving the building space (Q_1) should be same as air supplied to the space Q_2 .

$$Q_3 = Q_2 = \frac{Q_v}{\left(1 - \frac{\text{EATR}}{100} \right)}$$

Combining Equation (22) and Equation (21) gives

$$Q_1 = Q_2(\text{OACF}) = \frac{Q_v(\text{OACF})}{\left(1 - \frac{\text{EATR}}{100} \right)}$$

Equation (23) 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 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 snot 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.

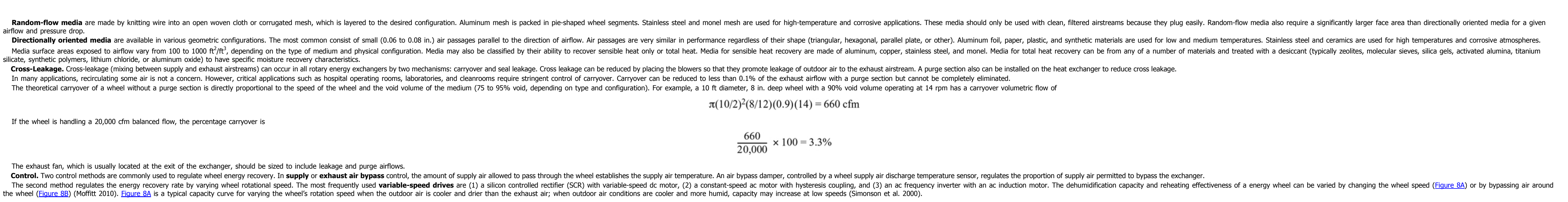
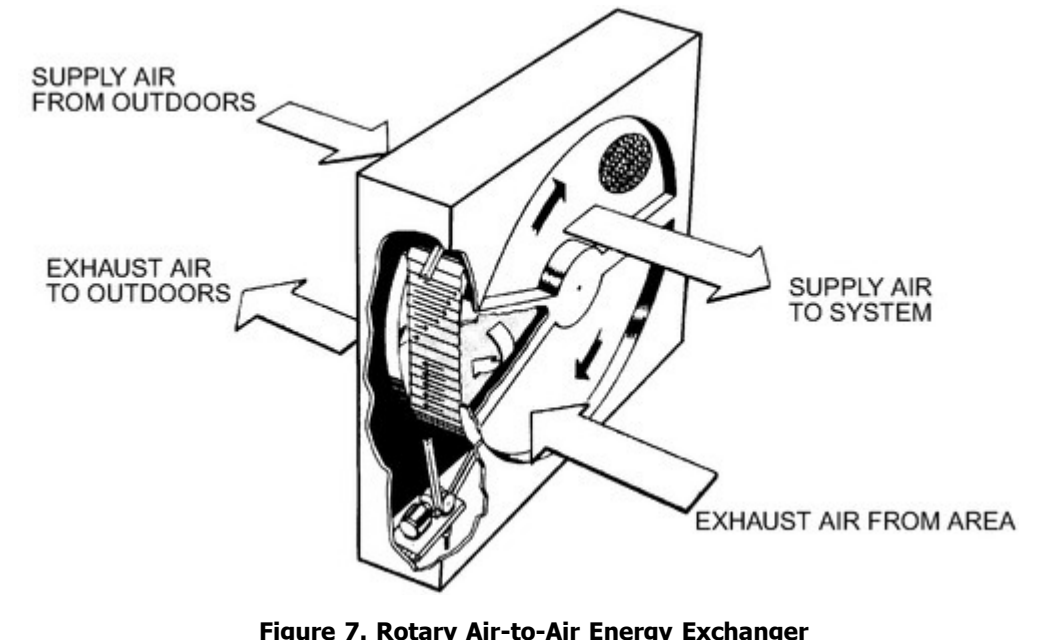
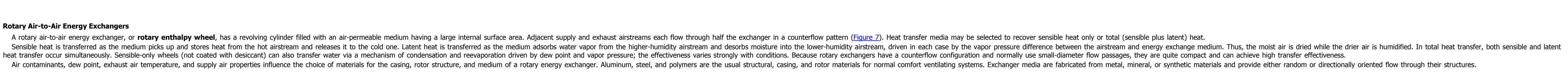
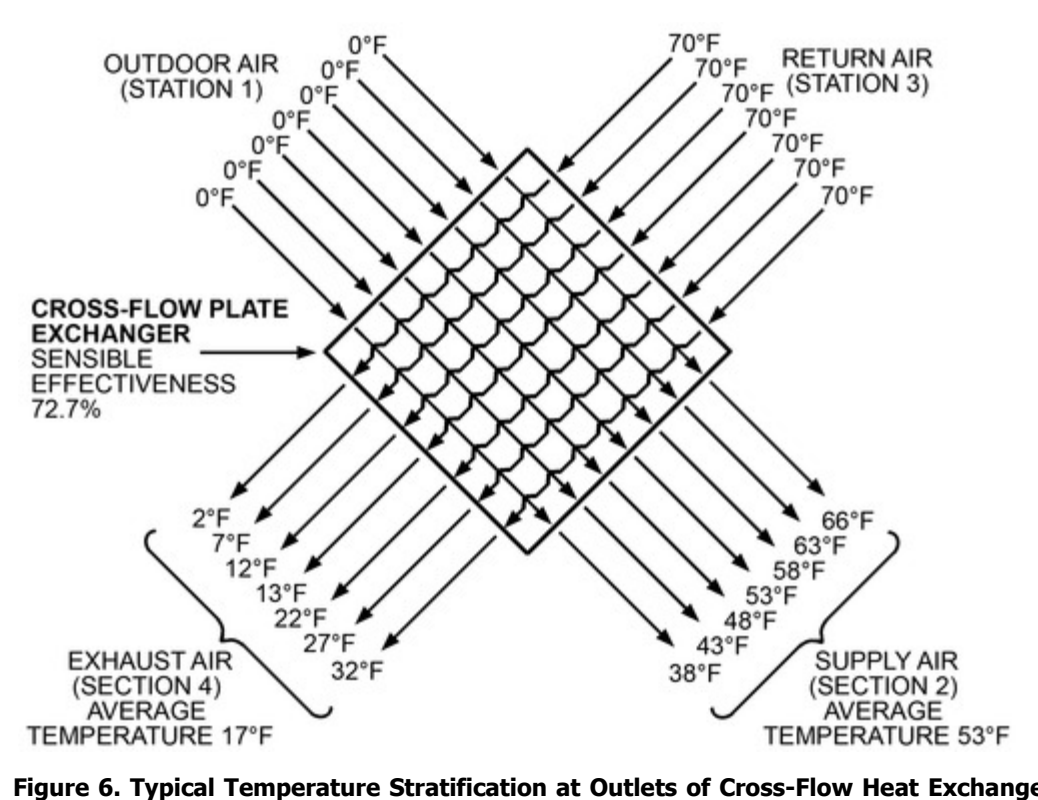
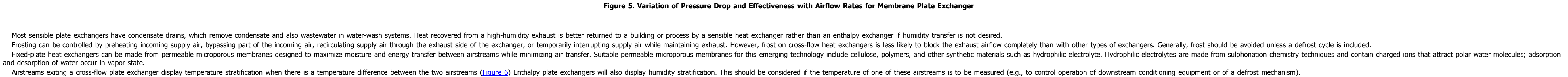
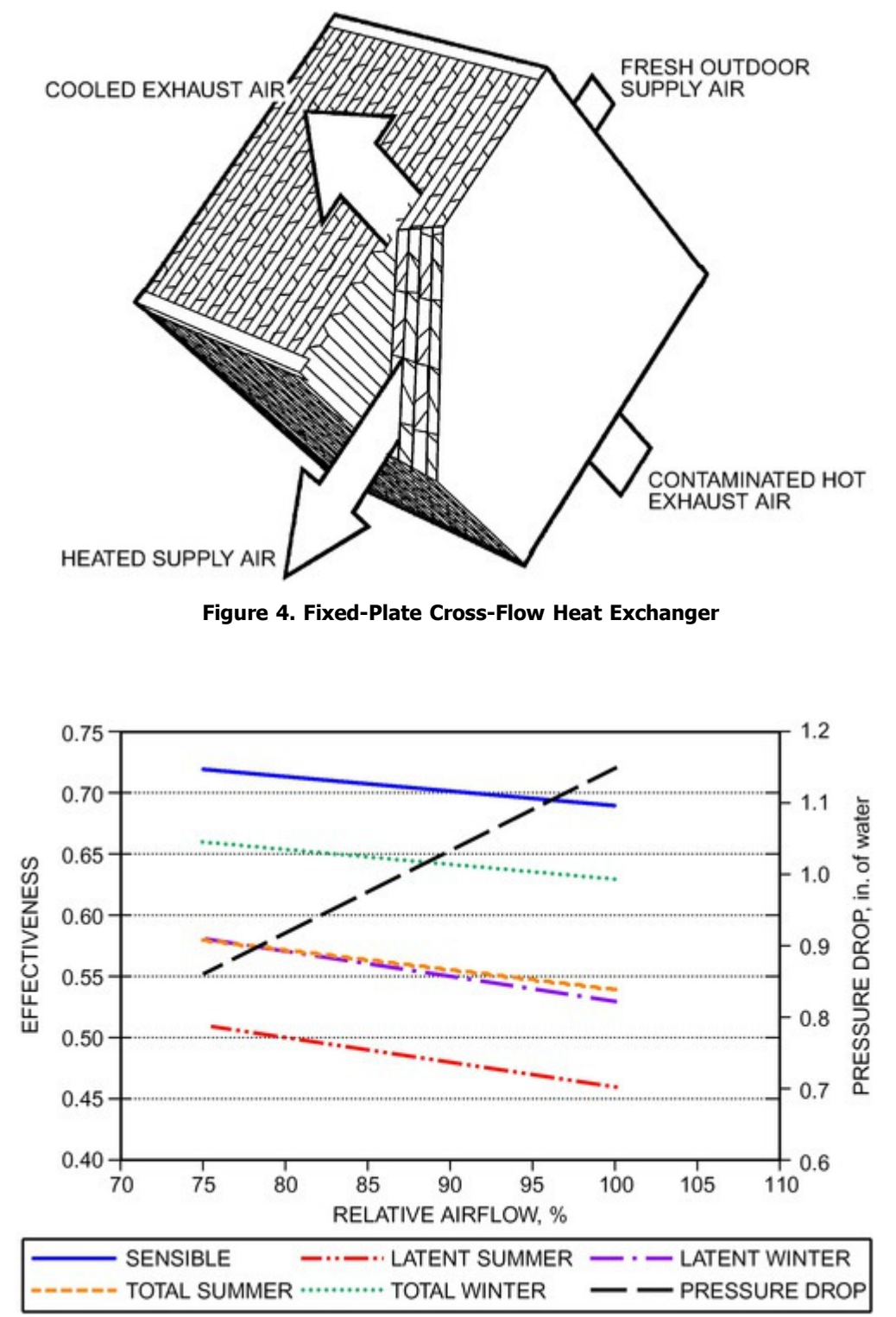
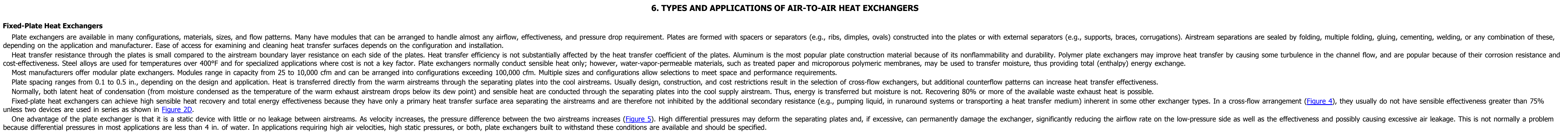
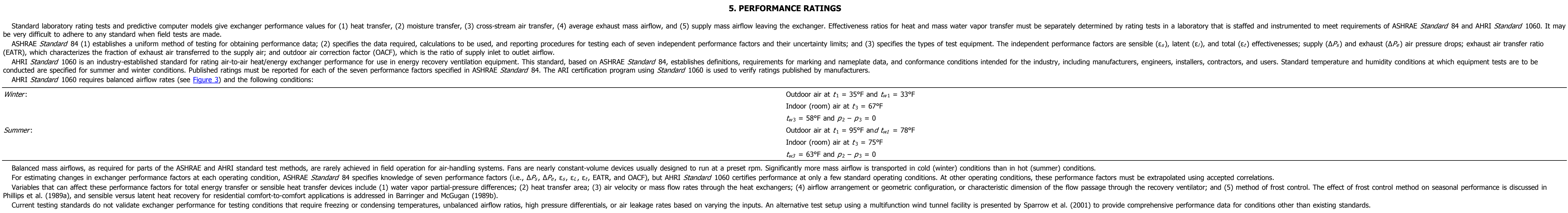


Figure 9 shows the effectiveness ϵ , sensible heat transfer only, with balanced airflow, convection-conduction ratio less than 4, and no leakage or cross-flow, of a regenerative counterflow heat wheel versus number of transfer units (NTU). This simple example of a regenerative wheel also shows that regenerative counterflow rotary effectiveness increases with wheel speed (C is proportional to wheel speed), but there is no advantage in going beyond $C/C_{min} = 5$ because the carryover of contaminants increases with wheel speed. See Kays and Crawford (1993) or Shah (1981) for details.

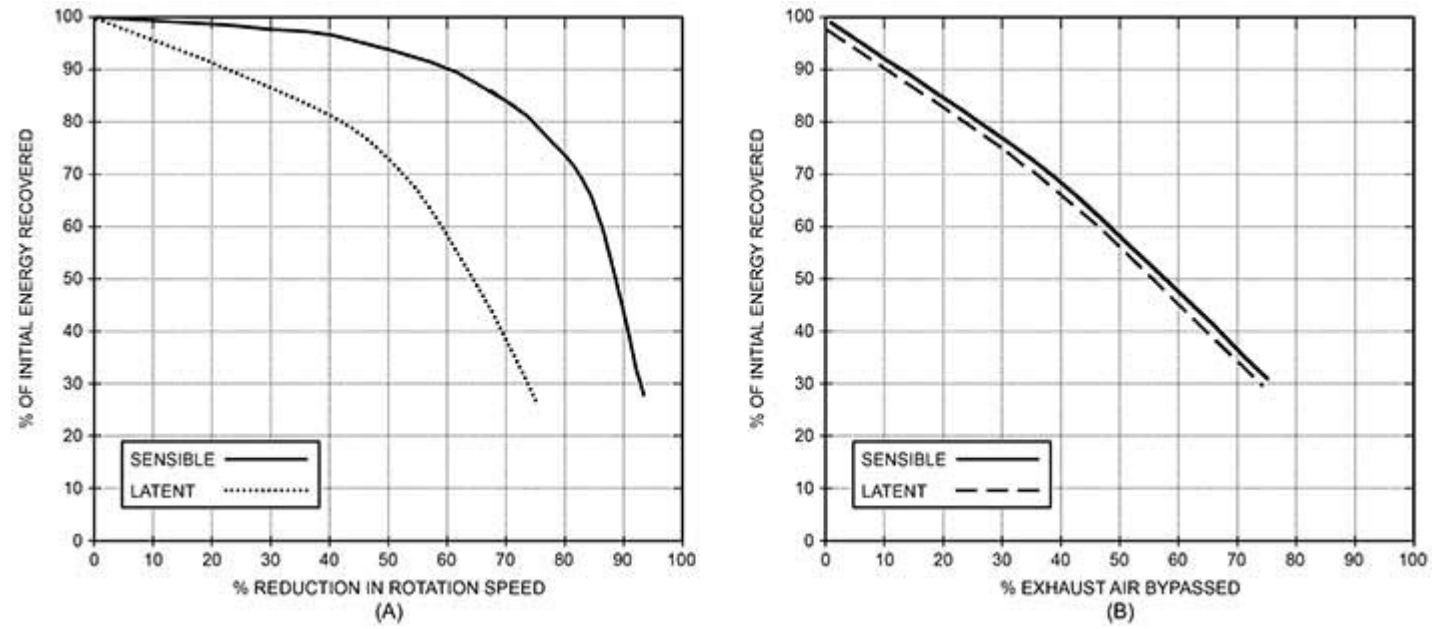


Figure 8. Latent and Sensible Effectiveness Versus (A) Wheel Speed and (B) Bypassed Air (Moffitt 2010)

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.

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 re-moves 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 10 (Moffitt 2011). The corresponding processes are shown in Figure 11 (Moffitt 2011); use of a passive desiccant wheel is shown on the right side of the figure.

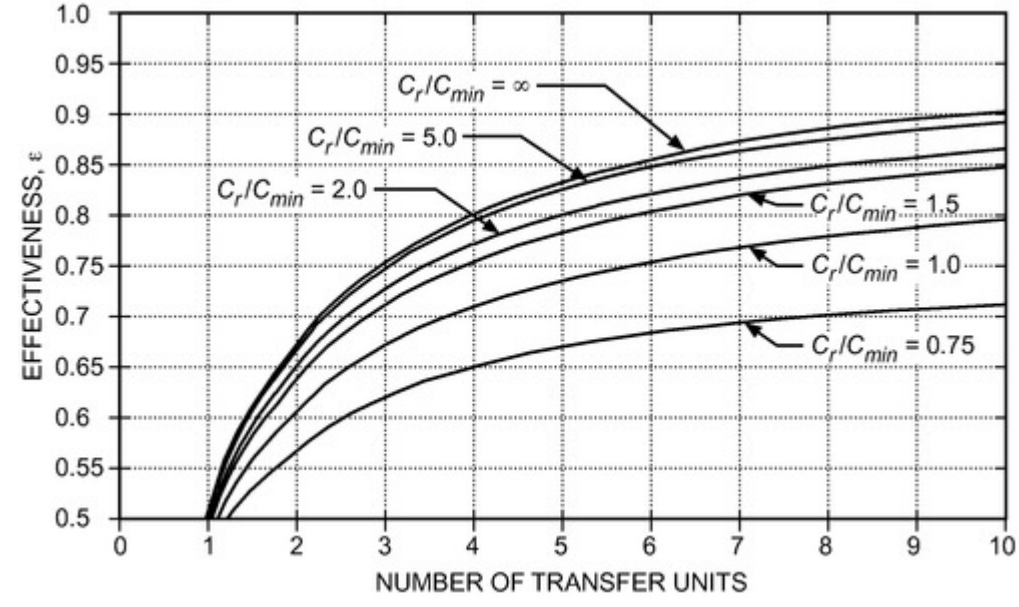


Figure 9. Effectiveness of Counterflow Regenerator (Shah 1981)

The second exchanger can also be placed in series with the cooling coil as shown in Figure 12 (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 13 (Moffitt 2011); use of a passive desiccant wheel is shown on the right side of the figure.

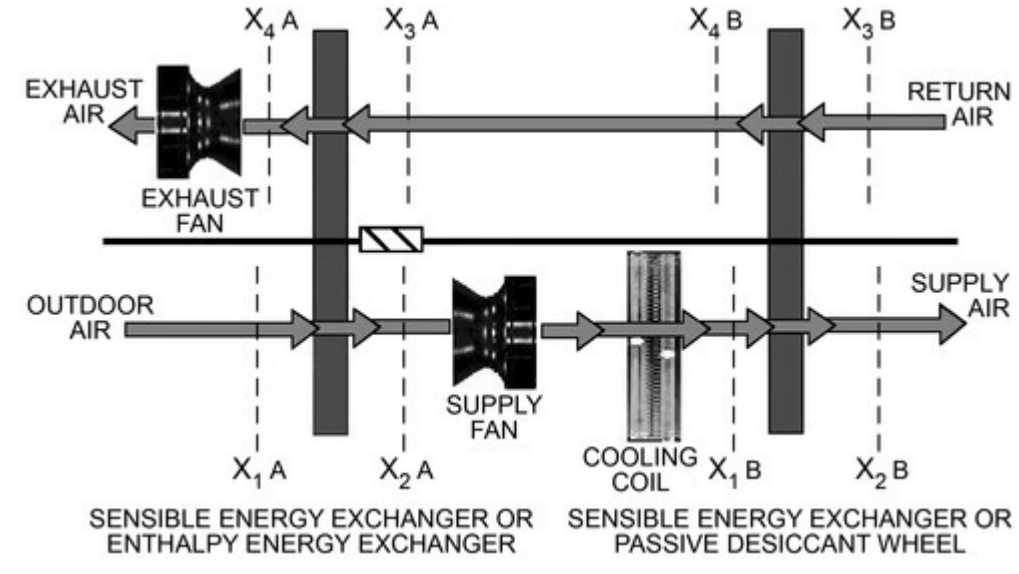


Figure 10. Multiple Energy Recovery Exchangers in Parallel Mode (Moffitt 2011)

Coil Energy Recovery (Runaround) Loops

A typical coil energy recovery loop (Figure 14) 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 counterflow piping through which an intermediate heat transfer fluid (typically water or antifreeze solution) is pumped.

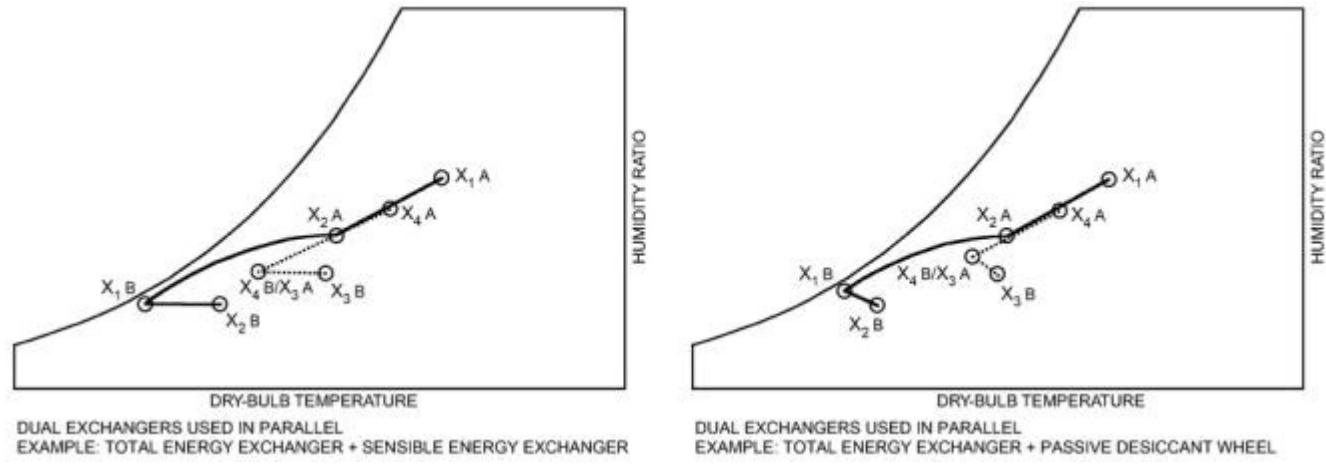


Figure 11. Psychrometric Processes of Exchangers in Parallel Mode (Moffitt 2011)

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.

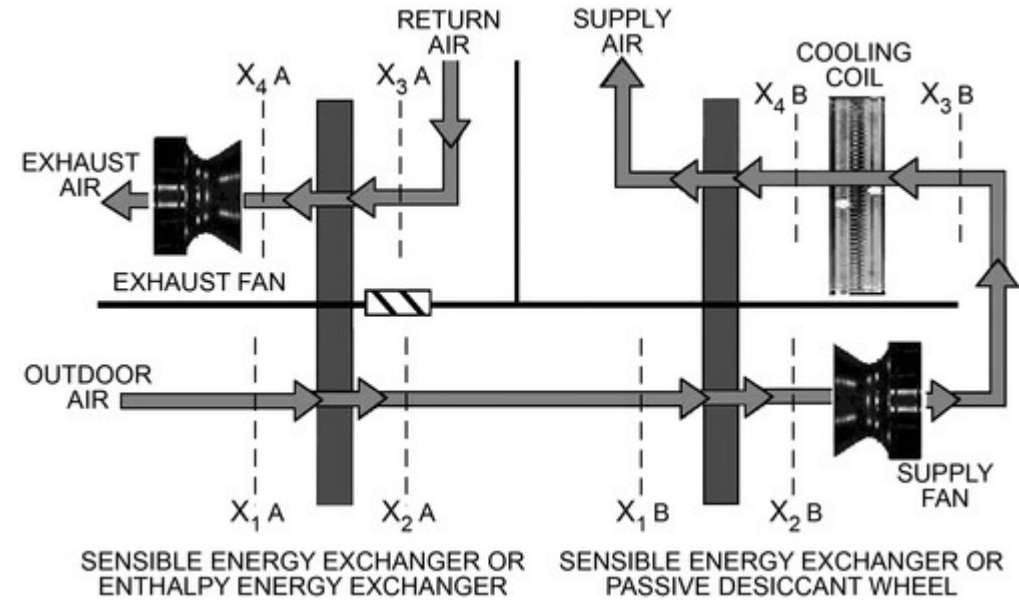


Figure 12. Multiple Energy Recovery Exchangers in Series Mode (Moffitt 2011)

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. 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.

Desiccant 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) (CZES 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.

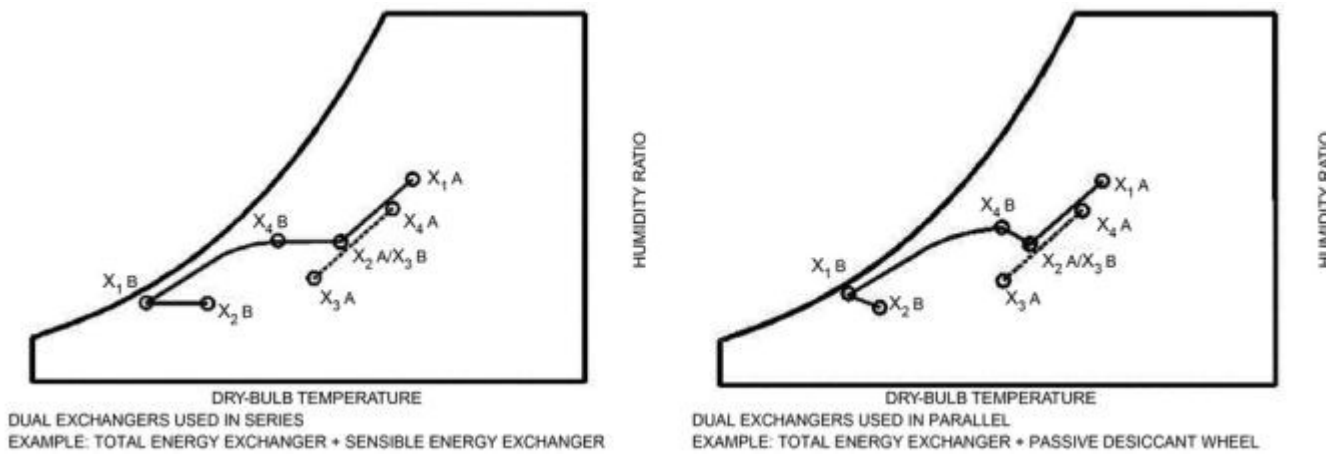


Figure 13. Psychrometric Processes of Exchangers in Series Mode (Moffitt 2011)

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 15, 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).

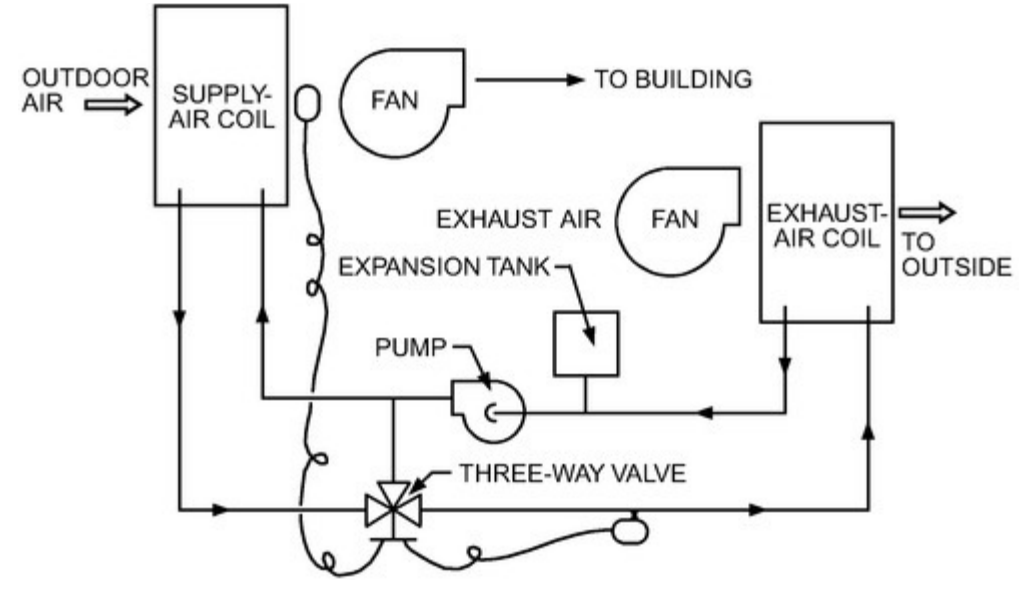


Figure 14. Coil Energy Recovery Loop

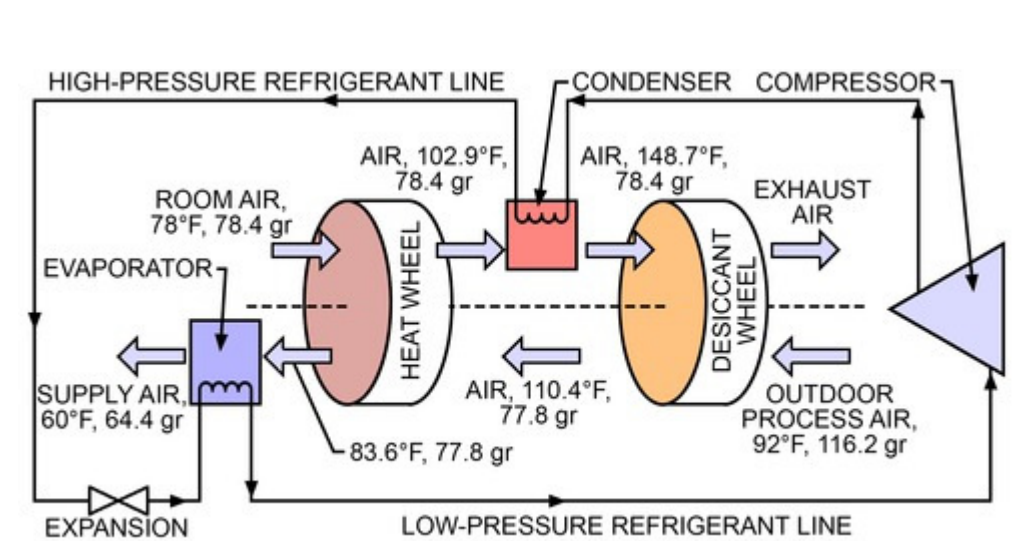


Figure 15. Heat Pump Augmented by Heat and Desiccant Wheels (Dhamshala 2016)

Figure 16 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 15 (Dhamshala 2016).

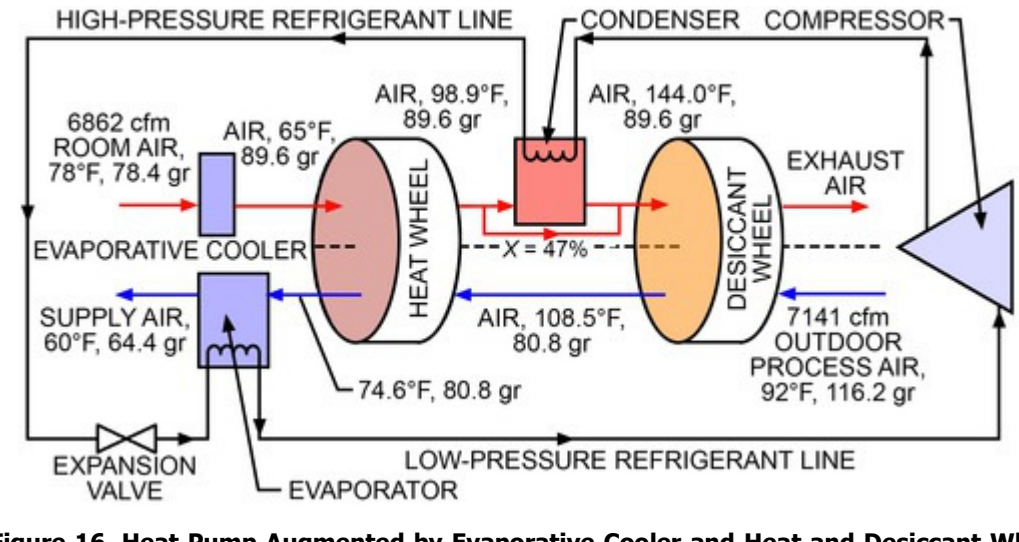


Figure 16. Heat Pump Augmented by Evaporative Cooler and Heat and Desiccant Wheels

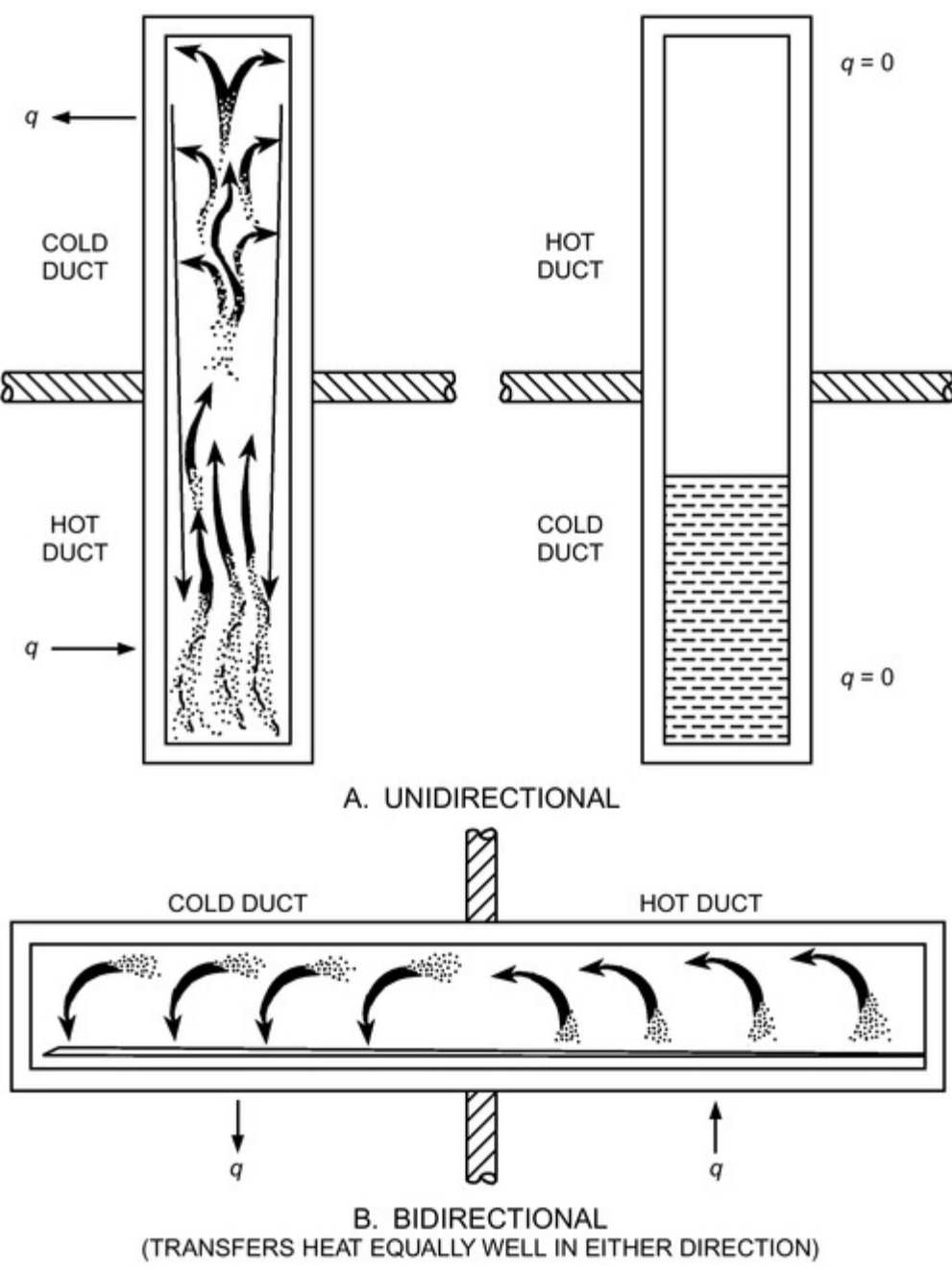


Figure 24. Sealed-Tube Thermosiphons

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 20 and 25). For coil-type thermosiphon loops (Figure 25), 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 boiling in the evaporator. Optimum charge levels have been found to be around 80% (Mathur 1990a, 1990b; Mathur and McDonald 1986).

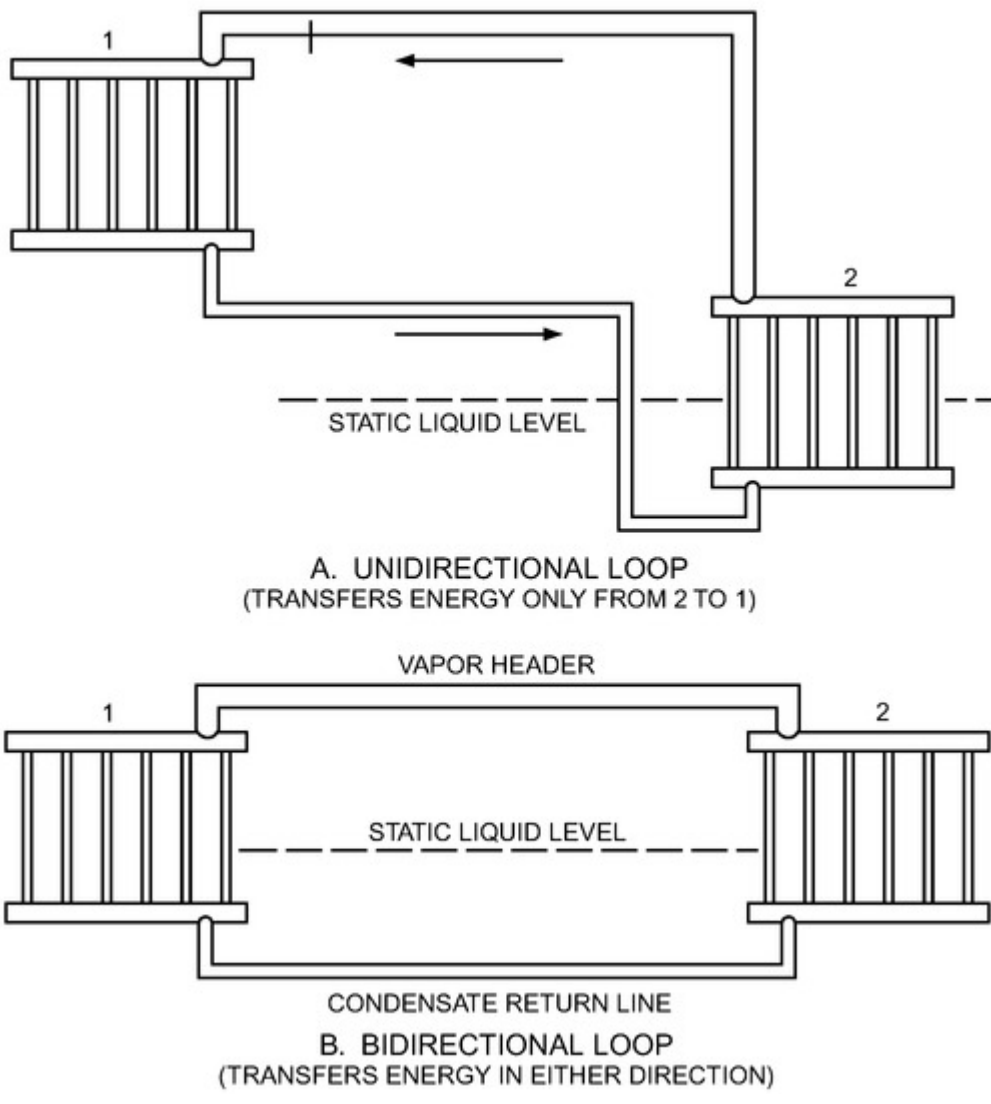


Figure 25. Coil-Type Thermosiphon Loops

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 unceally. 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 26 shows thermosiphon loop performance (Mathur and McDonald 1986) for an eight-row, 12 fpi system in unidirectional mode. Figure 27 shows the performance of the same system as a function of percent static charge for an overall temperature difference of 63°F.

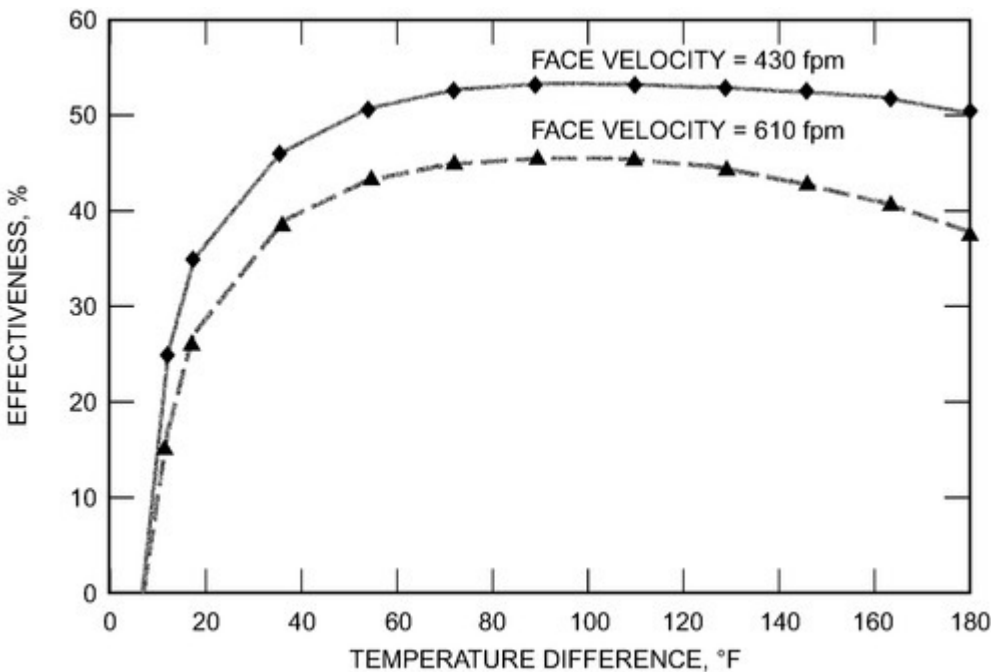


Figure 26. Eight-Row, 12 fpi Unidirectional System; Static Charge of 80% (Mathur and McDonald 1986)

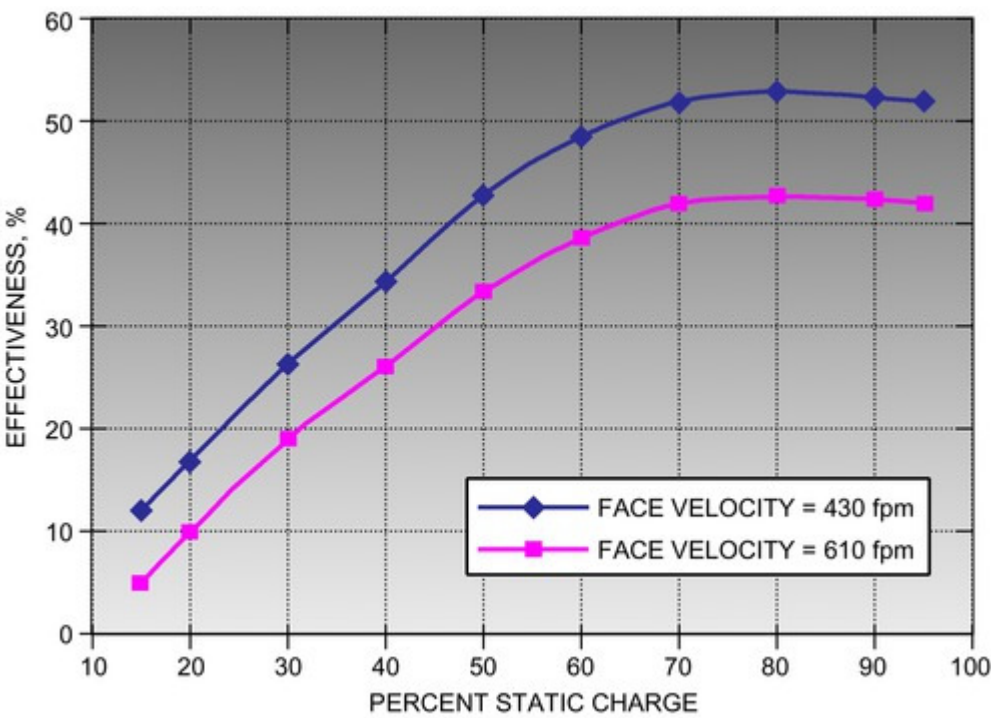


Figure 27. Typical Performance of Two-Phase Thermosiphon Loop as Function of Percent Static Charge: Eight Rows, 12 fpi Unidirectional System; Overall Temperature Difference of 63°F (Mathur and McDonald 1986)

These energy recovery systems are currently available to enhance the system performance (similar to heat pipe heat exchangers) of the current systems or to downsize the 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 Leadership in Energy and Environmental Design™ (LEED®) points. **Example 3. Sensible Heat Energy Recovery in a Two-Phase Thermosiphon Loop** Outdoor air at 10°F enters an eight-row, two-phase thermosiphon loop with a face area of 2 ft² 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/ft³. 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, exit temperature of supply air to the space, energy recovered, and power supplied to the fan motor.

Solution:

From the data given, the overall temperature difference between supply and exhaust airstreams is 65°F (i.e., 75 – 10 = 65). From Figure 26, 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 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 as follows:

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

$$q_s = 60(430 \text{ ft/min})(2 \text{ ft}^2)(0.0843 \text{ lb/ft}^3)(0.24 \text{ Btu/lb} \cdot ^{\circ}\text{F})(44.1 - 10) = 35,599 \text{ Btu/h}$$

The supply air fan power [Equation (11)] is

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

$$P_s = [(860 \text{ ft}^3/\text{min})(0.72 \text{ in. of water})]/[(6356)(0.9)(0.75)] = 0.15 \text{ hp}$$

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 (LDCS) 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 process 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 28 (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.

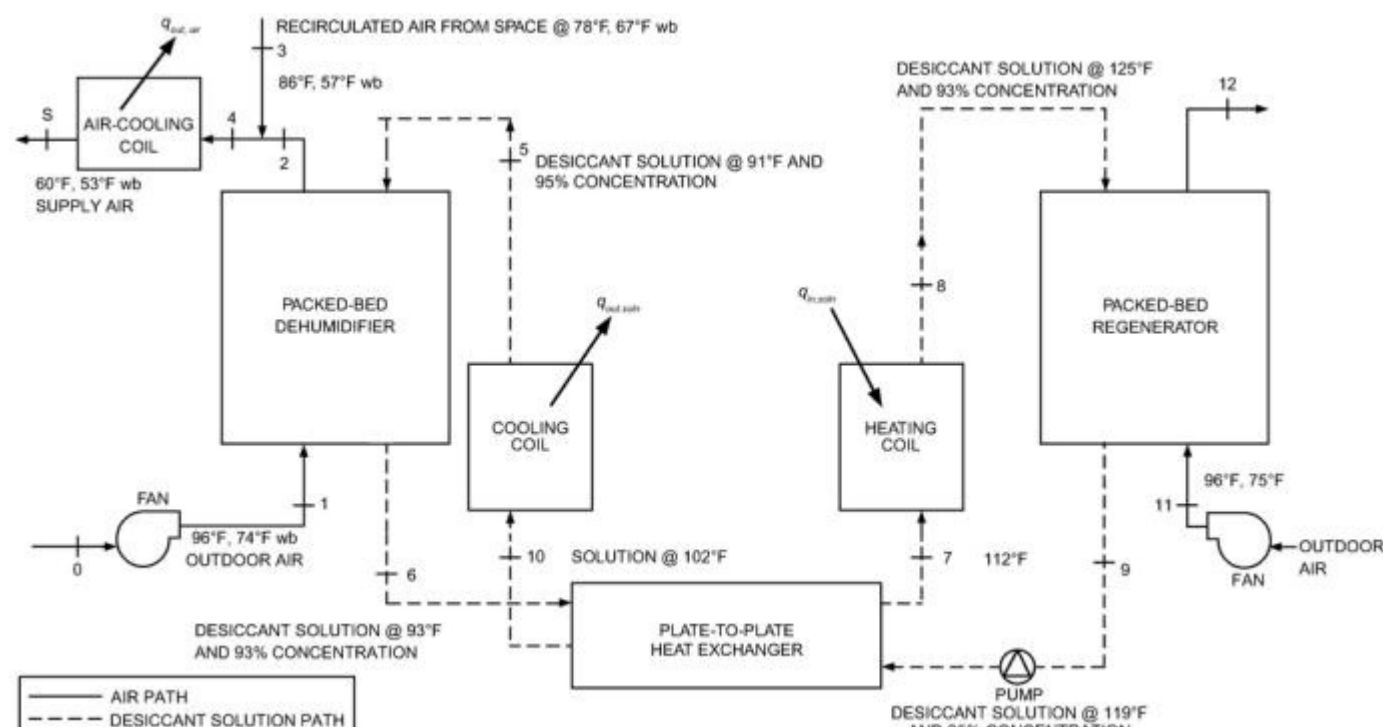


Figure 28. Typical Dehumidification of Outdoor Air by Liquid-Desiccant System (Williams 2007)

As seen in Figure 28, 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 temperature absorbed and lower the humidity of the exiting air from the absorber.

Table 2. Types of Liquid-Desiccant Solutions

Desiccant Type	No of Investigators	Typical Concentration Range, %	Operating Temperatures, °F
LiCl	7	34 to 43	77 to 81
TEG	5	92 to 98	68 to 114
LiBr	4	42.6 to 57	68 to 93
CaCl ₂	2	40 to 43	81
KCOOH	1	72.8 to 74	71 to 77

Source: Jain and Bansal (2007).

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 (CaCl₂) is an equally good candidate for desiccant choice and is less expensive than LiCl. A 50%/50% combination of LiCl and CaCl₂ 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

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_v , in units of m²/m³. Typically, packing densities exceeding 200 m²/m³ 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 lath 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 29). This liquid transports water vapor and heat. The sorbent solution is usually a halogen salt solution such as lithium chloride, calcium chloride, and water. 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.

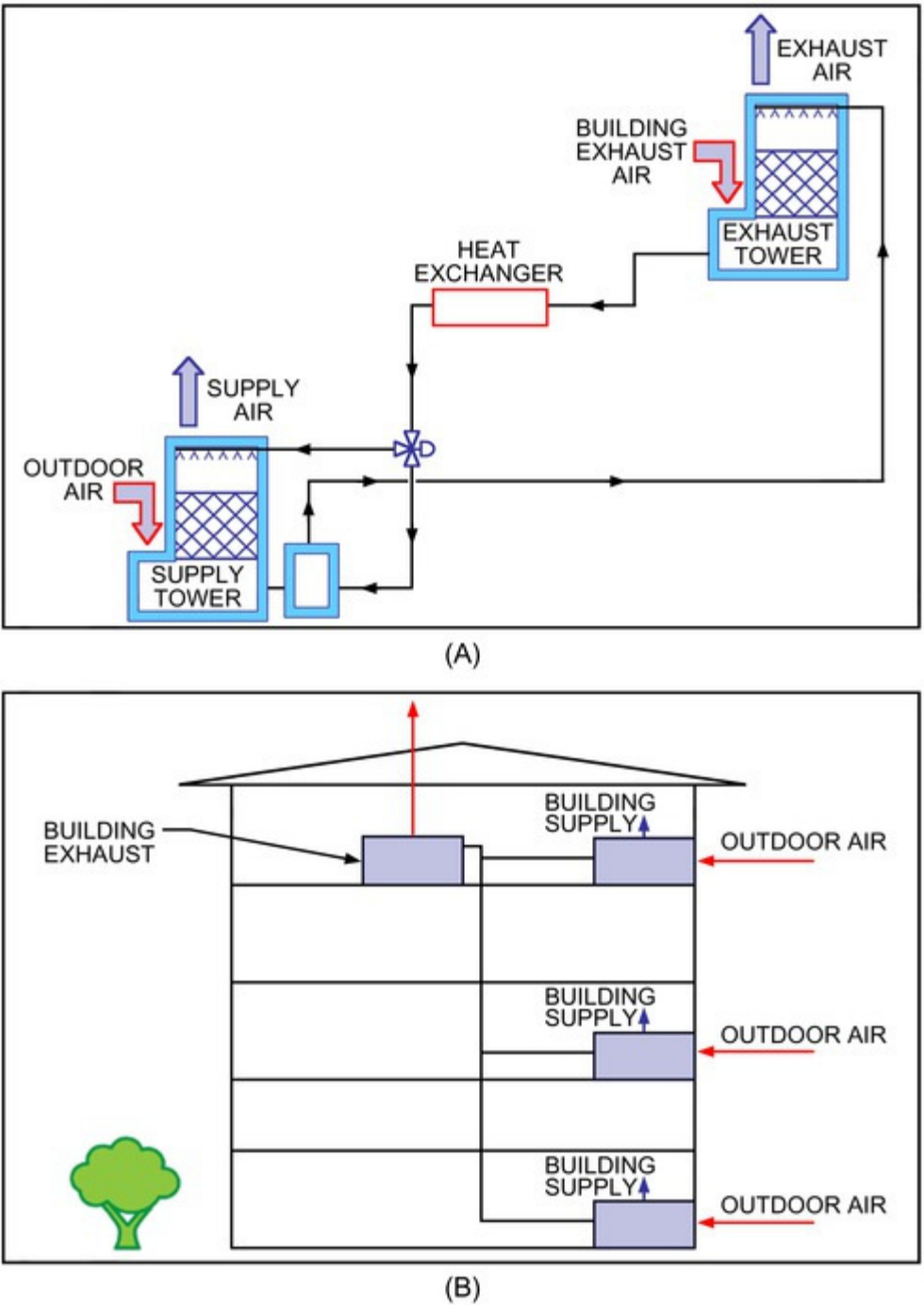


Figure 29. Twin-Tower System Performing Indirect Evaporative Air Cooling: (A) Schematic and (B) Installation in Building

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 denister 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 29A](#). 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 29B](#). 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.

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 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 65 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 52 of the 2015 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 ($p = 0.075 \text{ lb/ft}^3$) 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 ($p = 0.071 \text{ lb/ft}^3$) 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:

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 30](#). 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 sensible heat [Equation \(3c\)](#).

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

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.

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

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

- b. Leaving exhaust air temperature is

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

$$q_s = (0.071 \text{ lb/ft}^3)(60)(32,000 \text{ ft}^3/\text{min})(0.24 \text{ Btu/lb} \cdot ^\circ\text{F})(102 - 72.4) = 968,000 \text{ Btu/h recovered}$$
$$q_e = (0.075 \text{ lb/ft}^3)(60)(32,000 \text{ ft}^3/\text{min})(0.24 \text{ Btu/lb} \cdot ^\circ\text{F})(92 - 64) = 968,000 \text{ Btu/h recovered}$$

7. Plot conditions on psychrometric chart ([Figure 30](#)), and confirm that no latent exchange occurred.
- Because EATR = 0 and OACF = 1, Steps 8 to 10 are not presented here.

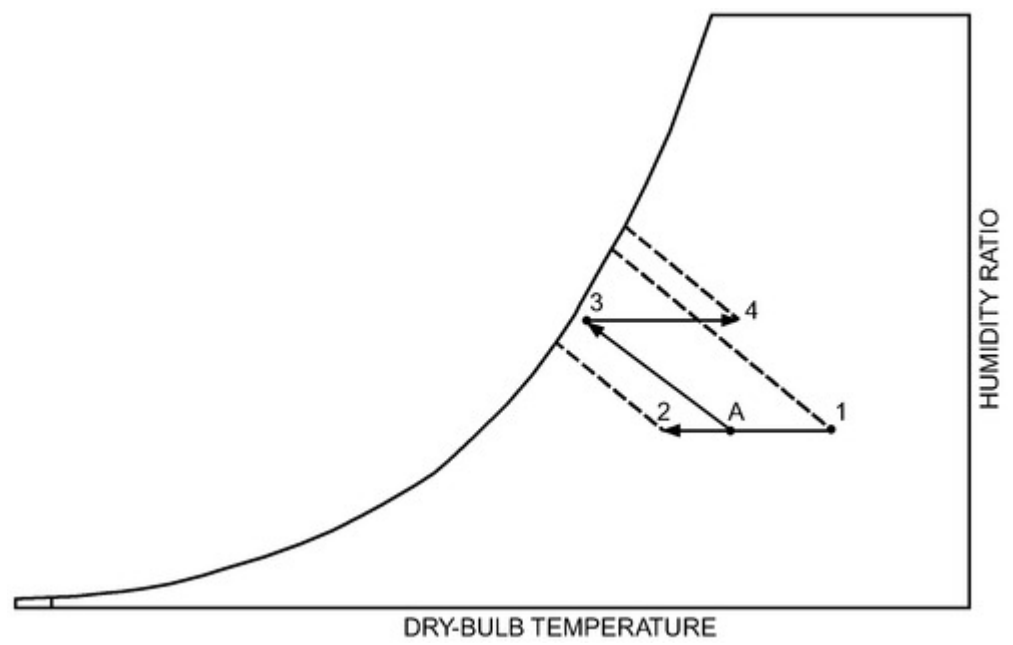


Figure 30. Indirect Evaporative Cooling Recovery (Example 5)

Precooling Air Reheaters (Series Application)

In some applications, such as ventilation in hot, humid climates, supply air is cooled below the desired delivery temperature to condense moisture and reduce humidity. Using this overcooled supply air to precool outdoor air reduces the air-conditioning load, which allows refrigeration equipment to be downsized and eliminates the need to reheat supply air with purchased energy. In this three-step process, shown in [Figure 31](#), outdoor air passes through an air-to-air heat exchanger, where it is precooled by supply air leaving the cooling coil. It is then further cooled and dehumidified in the cooling coil. It passes through the other side of the air-to-air heat exchanger, where the incoming supply air reheats it.

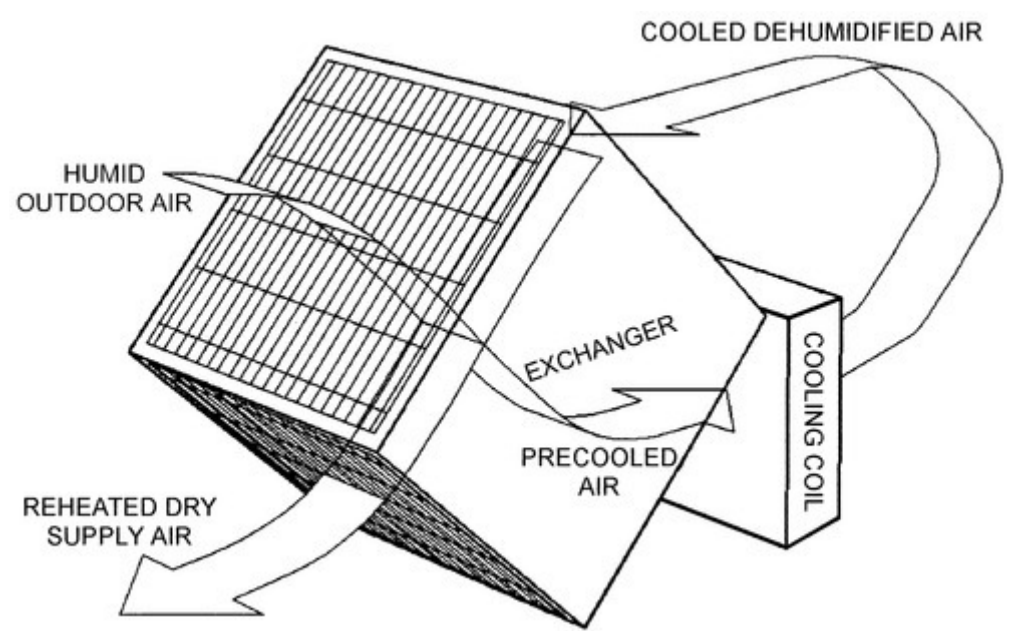


Figure 31. Precooling Air Reheater

Fixed-plate, heat pipe, and rotary heat wheel exchangers can be used to reheat precooled supply air. At part-load operation, the amount of heat transferred from precooling to reheating may require modulation. This can be done using the heat rate control schemes in [Table 3](#).

Example 5. Precooling Air Reheater Dehumidifier In this application, 3400 cfm of outdoor supply air at 95 and 81°F wb ($p = 0.069 \text{ lb/ft}^3$) 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 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:**
1. Because data on pressure drop are missing, skip this 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 [Equation \(2c\)](#) is used to determine the theoretical maximum energy transfer.

$$q_s = (60)(0.069 \text{ lb/ft}^3)(3400 \text{ ft}^3/\text{min})(0.24 \text{ Btu/lb} \cdot ^\circ\text{F})(95 - 52.1) = 145,000 \text{ Btu/h saved}$$

3. Establish the sensible effectiveness.
- The manufacturer give the effectiveness as 58.4% at the designated operating conditions.
4. Calculate actual energy transfer at design conditions.

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

5. Calculate leaving air conditions.
- Because condensation occurs as the outdoor airstream passes through the precooling side of the heat exchanger, use [Equation \(26\)](#) to determine its leaving enthalpy, which is the inlet condition for the cooling coil. Sensible heat transfer [Equation \(22\)](#) 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.

$$h_2 = 44.6 \text{ Btu/lb} + \frac{-84,700 \text{ Btu/h}}{(60)(0.069 \text{ lb/ft}^3)(3400 \text{ ft}^3/\text{min})} = 38.6 \text{ Btu/lb}$$

The wet-bulb temperature for saturated air with this enthalpy is 75°F. This is point 2 on the psychrometric chart ([Figure 32](#)), which is near saturation. Note that this precooled air is further dehumidified by a cooling coil from point 2 to point 3 in [Figure 32](#).

- b. Reheater leaving air conditions

$$t_4 = 52.1^\circ\text{F} + \frac{84,700 \text{ Btu/h}}{(60)(0.069 \text{ lb/ft}^3)(3400 \text{ ft}^3/\text{min})(0.24 \text{ Btu/lb} \cdot ^\circ\text{F})} = 77.2^\circ\text{F}$$

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

$$h_4 = 21.3 \text{ Btu/lb} + \frac{84,700 \text{ Btu/h}}{(60)(0.069 \text{ lb/ft}^3)(3400 \text{ ft}^3/\text{min})} = 27.3 \text{ Btu/lb}$$

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

6. Check performance.
7. Plot conditions on psychrometric chart ([Figure 32](#)).
- Because EATR = 0 and OACF = 1, Steps 8 to 10 are not presented here.

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
Airflow arrangements	Counterflow Cross flow ^a	Counterflow Cross flow ^a	Counterflow Parallel flow	Counterflow	Counterflow Parallel flow	—	Counterflow Parallel flow	—
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	—
Typical sensible effectiveness ($\eta_s = \eta_t$), % ^b	55 to 75	55 to 75	65 to 80	65 to 80	40 to 60 ^b	45 to 65 ^b	40 to 60 ^b	40 to 60 ^b
Typical latent effectiveness, % ^c	0	25 to 60	50 to 80	0	0	0	0	50 to 75 ^{d,e}
Total effectiveness, % ^f	20 to 50	35 to 70	55 to 80	25 to 60	15 to 35	—	—	40 to 75 ^f

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
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
EATR, %	0 to 2	0 to 5	0.5 to 10	0.5 to 10	0 to 1	0	0	0
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
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
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
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 ventilation 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
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 performance requires accurate simulation model	Effectiveness may be limited by pressure drop and cost Few suppliers	Few suppliers Maintenance and performance unknown
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
EATR = exhaust air transfer ratio OACF = outdoor air correction factor ¹ 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. ² Data not based on third-party certified data. ³ Data based on typical range of third-party certified data. ⁴ Face velocity of 250 to 500 fpm.								

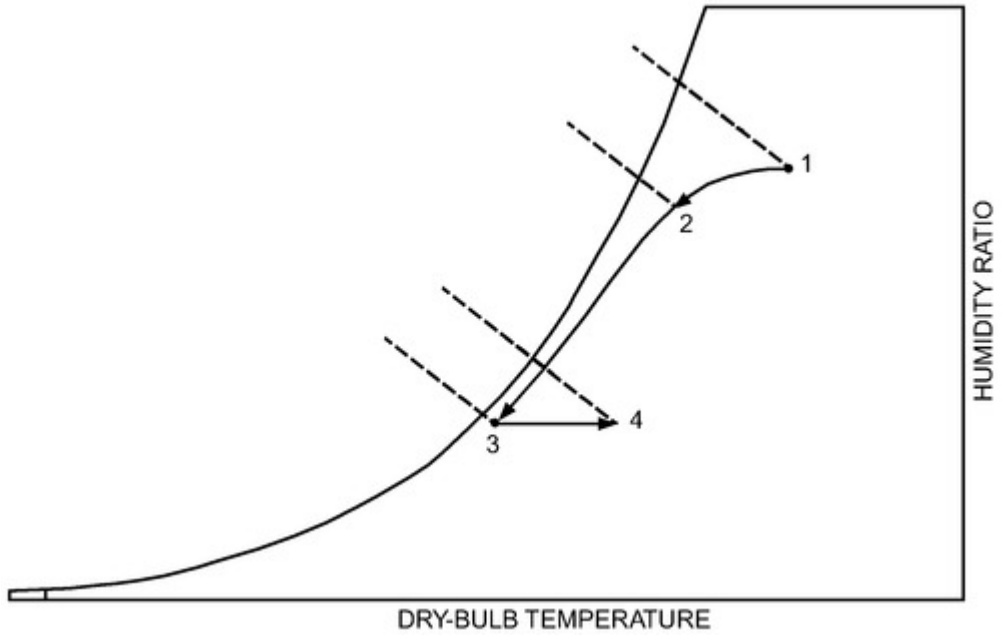


Figure 32. Precooling Air Reheater Dehumidifier (Example 5)

7. COMPARISON OF AIR-TO-AIR ENERGY RECOVERY SYSTEMS

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 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.

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:

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

(24)

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/Wh; 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

$$RER_{Total} = \frac{\epsilon_{net\ total} \dot{m}_{min}(h_1 - h_3)}{P_{blower} + P_{comp}}$$

(25a)

$$RER_{Sensible} = \frac{\epsilon_{net\ sensible} \dot{m}_{min} c_p (t_1 - t_3)}{P_{blower} + P_{comp}}$$

(25b)

$$RER_{Latent} = \frac{\epsilon_{net\ latent} \dot{m}_{min} h_{fg}(\omega_1 - \omega_3)}{P_{blower} + P_{comp}}$$

(25c)

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

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

(25d)

where

- C = unit conversion constant, 8.52 ft³-in. of water/min · W
- 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 γ . 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

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

(25e)

where

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

(25f)

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

(25g)

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](#).

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 37 of the 2015 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 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

$$PP = \frac{\text{Capital cost and interest}}{\text{Annual operating enery cost saved}}$$

(26)

$$= \frac{C_{s,init} - ITC}{C_e(1 - T_{inc})} CRF(i'', n)$$

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 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. [Equation \(26\)](#) is used to determine the rate of energy transfer when sensible (temperature) and latent (moisture) energy transfer occurs; [Equation \(27\)](#) is used for sensible-only energy transfer.

$$q_t = 60Q\rho(h_{in} - h_{out})$$

(27)

$$q_s = 60Q\rho c_p(t_{in} - t_{out})$$

(28)

where

- q_t = q_s + q_l = total energy transfer, Btu/h
- q_s = sensible heat transfer, Btu/h
- Q = airflow rate, cfm
- ρ = air density, lb/ft³
- c_p = specific heat of air = 0.24 Btu/lb · °F
- t_{in} = dry-bulb temperature of air entering exchanger, °F
- t_{out} = dry-bulb temperature of air leaving exchanger, °F
- h_{in} = enthalpy of air entering heat exchanger, Btu/lb
- h_{out} = enthalpy of air leaving heat exchanger, Btu/lb

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.

The required 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 94* as a test procedure and analysis guide. These data may be extrapolated to non-AHRI conditions by the manufacturer using correlations such as [Equations \(14\)](#) or [\(16\)](#), 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 $\dot{m}_{max}$ and $\dot{Q}_{max}$.

The airstream with the lower mass flow $\dot{m}_{min}$ limits heat and moisture transfer. Some designers specify and prefer working with airflow rates 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 moisture, sensible heat, latent heat and total energy rates are given by [Equations \(4f\)](#), [\(2c\)](#), [\(4c\)](#), and [\(6c\)](#), respectively.

The split between latent and sensible energy can be determined by plotting airstream conditions on a psychrometric chart as shown in [Figure 33](#). 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_t , and ϵ_t .

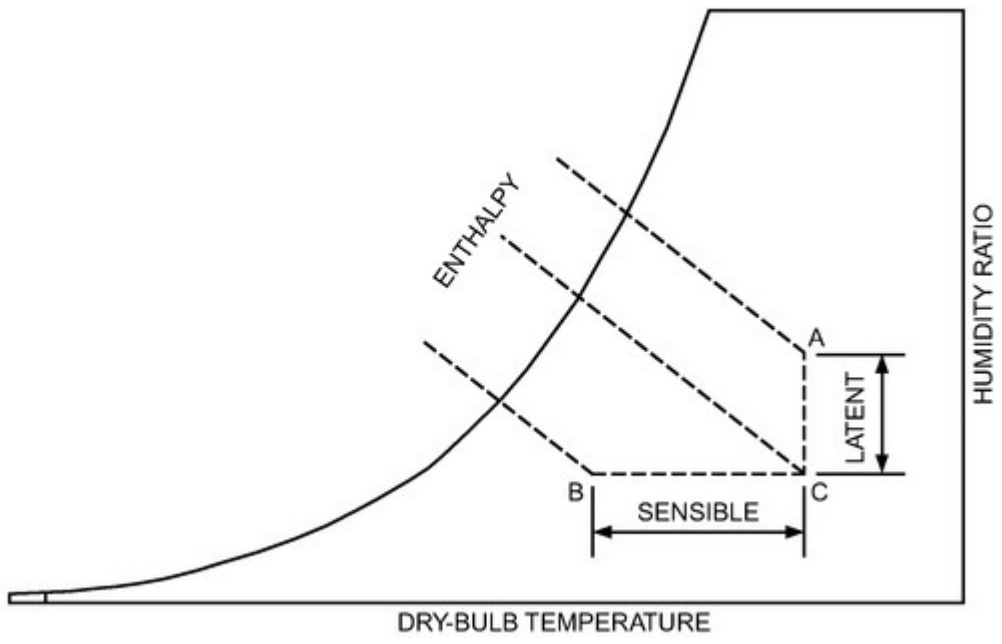


Figure 33. Maximum Sensible and Latent Heat from Process A-B

Each of these ratios is obtained from manufacturers' product data using input conditions and airflows for both airstreams. The effectiveness for airflow 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.

The actual moisture, sensible heat, latent heat, and total energy rates are given by [Equations \(4a\)](#), [\(2b\)](#), [\(4b\)](#), and [\(6b\)](#), respectively. Note that ϵ_m (mass effectiveness) = ϵ_t (latent effectiveness), as shown by [Equations \(4a\)](#) and [\(4d\)](#).

Step 5. Calculate leaving air properties for each airstream using [Equations \(3\)](#), [\(5\)](#), and [\(7\)](#).

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 \(5a\)](#) and [\(5b\)](#) must be used to calculate the leaving air humidity conditions, and [Equations \(7a\)](#) and [\(7b\)](#) to calculate the enthalpy values for airstreams in which inherent latent energy transfer occurs. [Equations \(3a\)](#) and [\(3b\)](#) may be used for airstreams if only sensible energy transfer is involved.

Step 5. Check energy transfer balance between airstreams.
[Equations \(3c\)](#) and [\(3d\)](#) can be used to estimate the sensible energy for the two airstreams, and [Equations \(8a\)](#) and [\(8b\)](#) can estimate the total energy for the two airstreams. 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 ($p = 0.087$ lb/ft³) 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/ft³ 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.

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

- Because data on pressure drop are missing, skip this step.
- Calculate the theoretical maximum heat transfer.
The two inlet conditions plotted on a psychrometric chart ([Figure 34](#)) indicate that, because the exhaust air has low relative humidity, latent energy transfer does not occur. Using [Equation \(2c\)](#), the theoretical maximum sensible heat transfer rate q_s is
$$q_{max} = (60)(800 \text{ lb/min})(0.24 \text{ Btu/lb} \cdot ^\circ\text{F})(75 - 0) = 864,000 \text{ Btu/h}$$
- Establish the sensible effectiveness.
From manufacturer's literature and certified performance test data, effectiveness is determined to be 60% at the design conditions.
- Calculate actual heat transfer at given conditions.
Using [Equation \(2b\)](#),
$$q_s = (0.6)(864,000 \text{ Btu/h}) = 518,400 \text{ Btu/h}$$
- Calculate leaving air conditions.
Because no moisture or latent energy transfer will occur,
 - Leaving supply air temperature t_2 is given as
$$t_2 = 0^\circ\text{F} + \frac{518,400 \text{ Btu/h}}{(60)(800 \text{ lb/min})(0.24 \text{ Btu/lb} \cdot ^\circ\text{F})} = 45^\circ\text{F}$$
 - Leaving exhaust air temperature t_4 is given as
$$t_4 = 75^\circ\text{F} - \frac{518,400 \text{ Btu/h}}{(60)(800 \text{ lb/min})(0.24 \text{ Btu/lb} \cdot ^\circ\text{F})} = 30^\circ\text{F}$$
- Using [Equations \(3c\)](#) and [\(3d\)](#), check performance.
$$q_s = (60)(800 \text{ lb/min})(0.24 \text{ Btu/lb} \cdot ^\circ\text{F})(45 - 0) = 518,400 \text{ Btu/h saved}$$
$$q_e = (60)(800 \text{ lb/min})(0.24 \text{ Btu/lb} \cdot ^\circ\text{F})(75 - 30) = 518,400 \text{ Btu/h saved}$$
- Plot conditions on psychrometric chart to confirm that no moisture exchange occurred ([Figure 34](#)).
Because EATR = 0 and OACF ≈ 1, Steps 8 to 10 of the calculation procedure are not presented here.

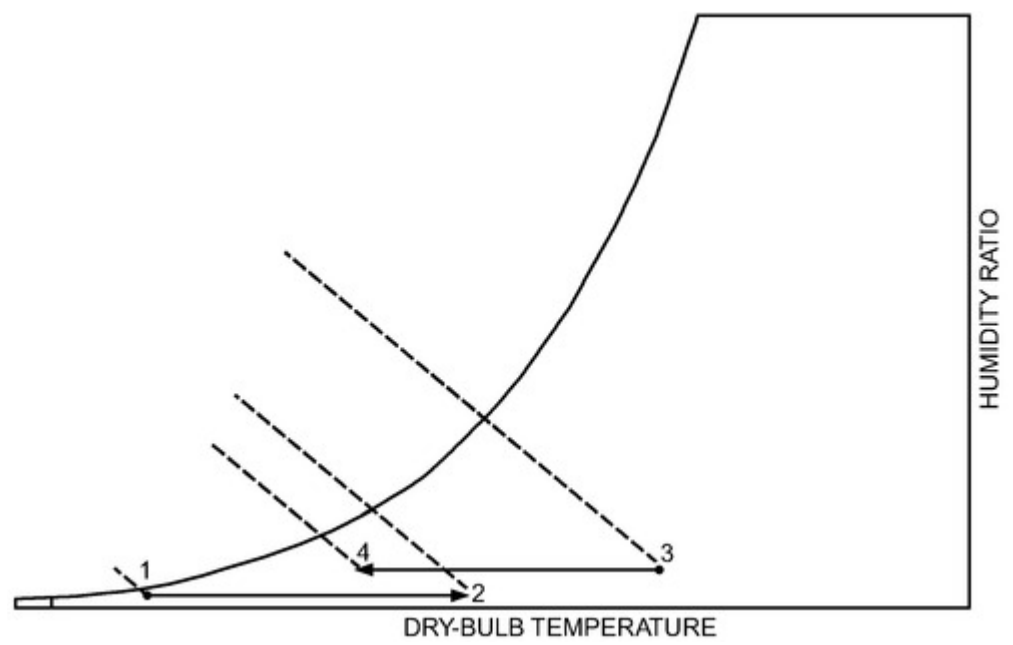


Figure 34. Sensible Heat Recovery in Winter (Example 6)

Example 7. Sensible Heat Recovery in Winter with Water Vapor Condensation Exhaust air at 75°F and 28% rh ($p = 0.075$ lb/ft³) and flow rate of 10,600 cfm is used to preheat 9500 cfm of outdoor air at 14°F and 50% rh ($p = 0.084$ lb/ft³) 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.

- Because data on pressure drop are missing, skip this step.
- Calculate the theoretical maximum sensible heat transfer.
The limiting airstream, will be preheated in the heat exchanger, so it is not subject to condensation. Therefore, [Equation \(2c\)](#) is used:
$$q_{max} = (60)(9500 \text{ ft}^3/\text{min})(0.084 \text{ lb/ft}^3)(0.24 \text{ Btu/lb} \cdot ^\circ\text{F})(75 - 14) = 700,000 \text{ Btu/h}$$
- Select sensible effectiveness.
From manufacturer's literature and performance test data, the sensible effectiveness is determined to be 70% at the design conditions.
- Calculate actual heat transfer at design conditions using [Equation \(2b\)](#):

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

$$t_2 = 14^\circ\text{F} + \frac{490,000 \text{ Btu/h}}{(60)(0.084 \text{ lb/ft}^3)(9500 \text{ ft}^3/\text{min})(0.24 \text{ Btu/lb} \cdot ^\circ\text{F})} = 56.6^\circ\text{F}$$

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

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 $t_4 = 35.7^\circ\text{F}$ and $w_4 = 0.0044$ lb/lb. The rate of moisture condensed m_w is

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

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

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.

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

- Plot conditions on psychrometric chart ([Figure 35](#)). 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.

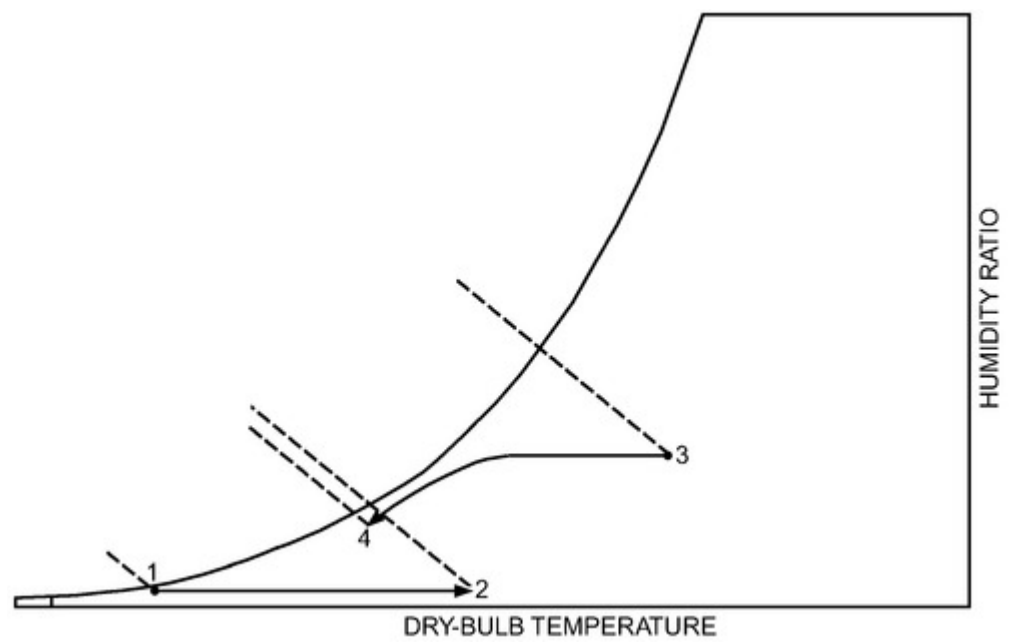


Figure 35. Sensible Heat Recovery in Winter with Condensate (Example 7)

Example 8. Total Heat Recovery in Summer Exhaust air at 75°F and 63°F wb ($p = 0.073$ lb/ft³) 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 ($p = 0.069$ lb/ft³) 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.

Solution:

- Because data on pressure drop are missing, skip this step.
- 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) $h_1 = 44.6 \text{ Btu/lb}$ $w_1 = 0.0198 \text{ lb/lb}$
Exhaust inlet (75°F db, 63°F wb) $h_3 = 28.5 \text{ Btu/lb}$ $w_3 = 0.0096 \text{ lb/lb}$

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

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

- Determine supply sensible and total effectiveness.
The manufacturer's selection data for the design conditions provide the following effectiveness ratios:
$$\epsilon_s = 70\% \quad \epsilon_t = 56.7\%$$

- Calculate energy transfer at design conditions.
$$q_t = (0.567)(566,000 \text{ Btu/h}) = 321,000 \text{ Btu/h total recovered}$$
$$q_s = -(0.7)(169,000 \text{ Btu/h}) = -118,000 \text{ Btu/h sensible recovered}$$
$$q_{lat} = 203,000 \text{ Btu/h latent recovered}$$

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

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

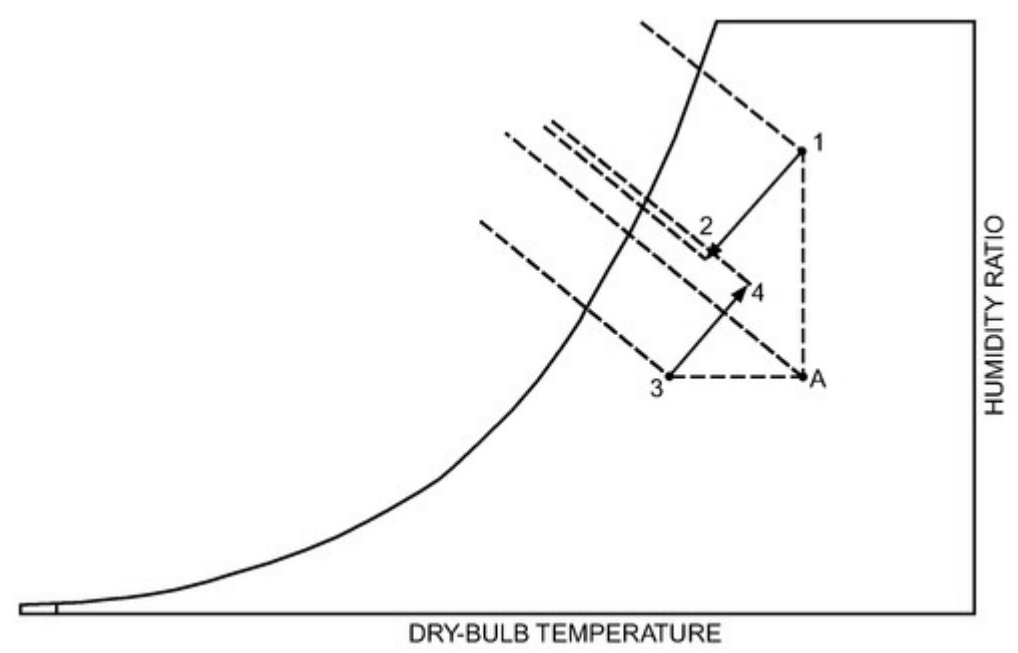
From the psychrometric chart, the supply air humidity ratio and wet-bulb temperature are found to be $w_2 = 0.0145$ and $t_{w2} = 71.6^\circ\text{F}$.

- Exhaust air conditions
$$t_4 = 75^\circ\text{F} + \frac{118,000 \text{ Btu/h}}{(60)(0.073 \text{ lb/ft}^3)(10,000 \text{ ft}^3/\text{min})(0.24 \text{ Btu/lb} \cdot ^\circ\text{F})} = 86.2^\circ\text{F}$$
$$h_4 = 28.5 \text{ Btu/lb} + \frac{321,000 \text{ Btu/h}}{(60)(0.073 \text{ lb/ft}^3)(10,600 \text{ ft}^3/\text{min})} = 35.4 \text{ Btu/lb}$$

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

- Check total performance ([Equations 9](#) and [10](#)).
$$q_t = (60)(0.069 \text{ lb/ft}^3)(8500 \text{ cfm})(44.6 - 35.5) = 320,000 \text{ Btu/h saved}$$
$$q_t = (60)(0.073 \text{ lb/ft}^3)(10,600 \text{ cfm})[(0.24 \text{ Btu/lb} \cdot ^\circ\text{F})(86.2 - 75) + (0.0134 - 0.0096)(1100 \text{ Btu/lb})] = 319,000 \text{ Btu/h, which is close to } 320,000 \text{ Btu/h}$$

- Plot conditions on psychrometric chart ([Figure 36](#)).
Because EATR = 0 and OACF ≈ 1, Steps 8 to 10 are not presented here.



- Assuming $EATR = 0$ and $OACF \approx 1$, determine the leaving supply air conditions and energy recovered, and check the energy exchange balance.
- If $EATR = 5\%$ and $OACF = 1.05$, determine the actual airflow rates.

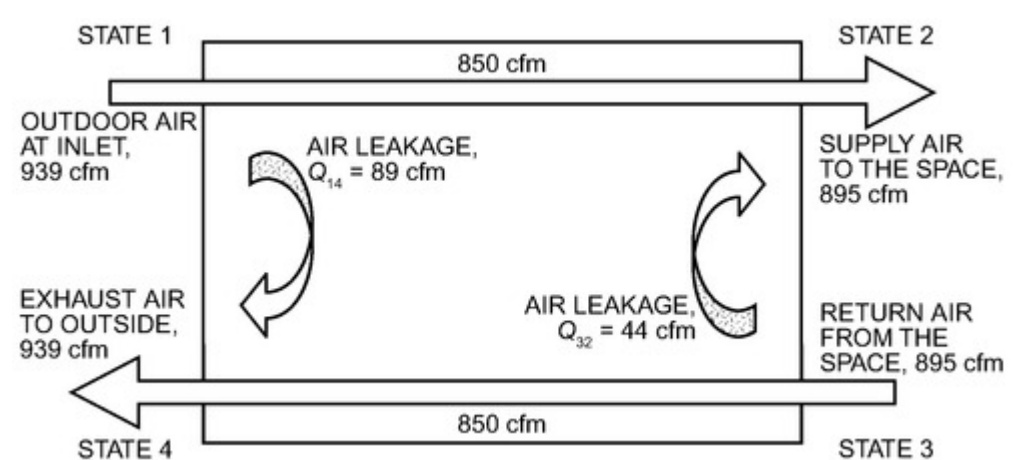
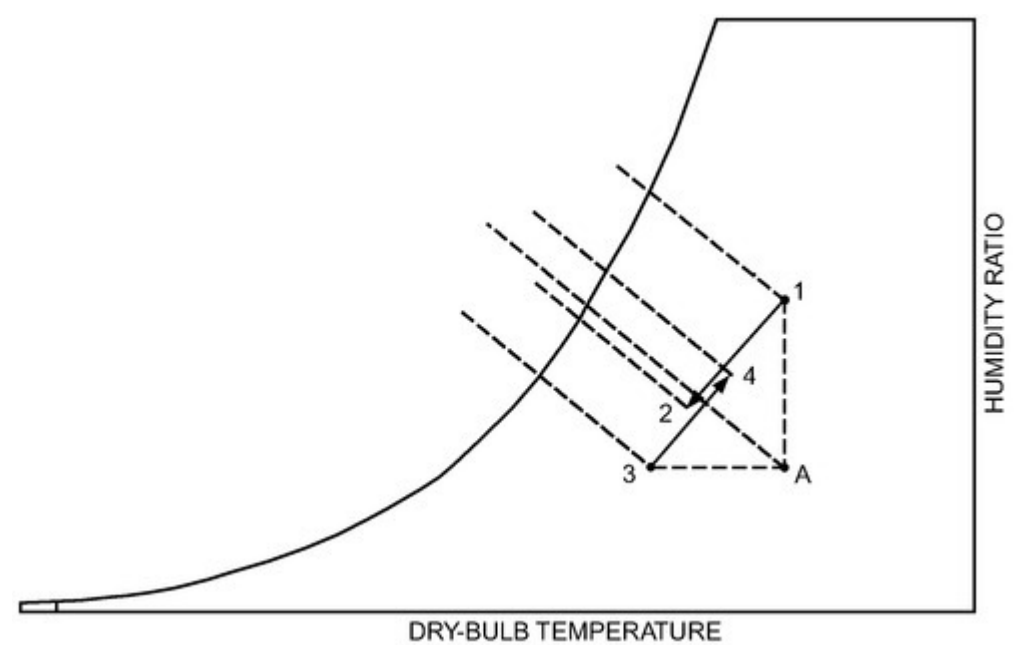
$$P_s = [(850 \text{ ft}^3/\text{min})(0.9)]/[(6356)(0.6)] = 0.2 \text{ hp}$$
 $P = 0.4 \text{ hp}$

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

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

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

$$q_t = (0.715)(58,750 \text{ Btu/h}) = 42,000 \text{ Btu/h total recovered}$$
$$t_4 = 75^\circ\text{F} + \frac{12,850 \text{ Btu/h}}{(0.075 \text{ lb/ft}^3)(60)(850 \text{ ft}^3/\text{min})(0.24 \text{ Btu/lb}\cdot^\circ\text{F})} = 89^\circ\text{F}$$

$$h_4 = 28.5 \text{ Btu/lb} + \frac{42,000 \text{ Btu/h}}{(0.075 \text{ lb/ft}^3)(60)(850 \text{ ft}^3/\text{min})} = 39.5 \text{ Btu/lb}$$
$$Q_3 = Q_2 = \frac{Q_v}{1 - \text{EATR}} = \frac{850 \text{ ft}^3/\text{min}}{1 - 5/100} = 895 \text{ cfm}$$
$$Q_1 = Q_2(\text{OACF}) \frac{Q_v(\text{OACF})}{1 - \text{EATR}} = \frac{(850 \text{ ft}^3/\text{min})(1.05)}{1 - 5/100} = 939 \text{ cfm}$$


A = area of recovery exchanger, ft²
 C_p = specific heat of moist air, Btu/lb · °F
 C = capital cost
 C_e = cost of energy
 C_r = ratio of $C_{\text{max}}/C_{\text{min}}$
 C_{int} = initial system capital cost
 CRF = capital recovery factor
 h = enthalpy, Btu/lb
 h_{fg} = enthalpy of vaporization, Btu/lb
 i = arbitrary rate, or discount rate
 ITC = income tax credit
 m_a = mass flow rate of supply moist air from outdoors, lb/min
 m_e = mass flow rate of exhaust moist air, lb/min
 m_{tot} = minimum mass flow rate of supply and exhaust airstreams, lb/min
 n = number of years in 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
 $t_{w,i}$ = wet-bulb temperature of moist air at state i , °F
 T = absolute temperature, °F, or $^{\circ}\text{C}$
 U = overall heat transfer coefficient, Btu · ft² · °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_E = total effectiveness of ERV
 η = efficiency
 ρ = density, lb/ft³
 ϕ = 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
 inc = indoor conditions of building space
 i = increment
 L = hydraulic
 L = latent
 max = maximum value
 min = minimum value
 o = reference state or outlet
 p = constant pressure

s = supply side or suction side
 t = total

REFERENCES

AHRI. 2014. Performance rating air-to-air heat exchangers for energy recovery ventilation heat equipment. *ANSI/AHRI Standard* 1060-2014. Air-Conditioning, Heating, and Refrigeration Institute, Arlington, VA.

AHRI. 2011. Calculating the efficiency of energy recovery ventilation and its effect on efficiency and sizing of building HVAC systems. *AHRI Guideline V-2011*. Air-Conditioning, Heating, and Refrigeration Institute, Arlington, VA.

ASHRAE. 1982. Symposium on energy recovery from air pollution control. *ASHRAE Transactions* 88(1):1197-1225.

ASHRAE. 2013. Ventilation for acceptable indoor air quality. *ANSI/ASHRAE Standard* 62.1-2013.

ASHRAE. 2013. Ventilation and acceptable indoor air quality in low-rise residential buildings. *ANSI/ASHRAE Standard* 62.2-2013.

ASHRAE. 2013. Method of testing air-to-air heat/energy exchangers. *ANSI/ASHRAE Standard* 89-2013.

Barringer, C.G., and C.A. McGugan. 1989a. Development of a dynamic model for simulating indoor air temperature and humidity. *ASHRAE Transactions* 95(2):449-460.

Barringer, C.G., and C.A. McGugan. 1989b. Effect of residential air-to-air heat and moisture exchangers on indoor humidity. *ASHRAE Transactions* 95(2):461-474.

Besant, R.W., and A.B. Johnson. 1995. Reducing energy costs using run-around systems. *ASHRAE Journal* 37(2):41-47.

Besant, R.W., and C.J. Simonsen. 2000. Air-to-air energy recovery. *ASHRAE Journal* 42(5):31-38.

Besant, R.W., and C. Simonsen. 2003. Air-to-air exchangers. *ASHRAE Journal* 45(4):42-50.

CZES. 2015. Center for Climate and Energy Solutions. www.czes.org.

Dhamshala, P. 2016. *Modern practices in design of air-conditioning and refrigerated systems*. The University of Tennessee, Chattanooga, Graphic Services, Chattanooga.

Dhital, P., R. Besant, and G.J. Schoenau. 1995. Integrating run-around heat exchanger systems into the design of large office buildings. *ASHRAE Transactions* 101(2):979-999.

Entesari, N., H. Yoshino, R. Takaki, A. Sakaki, and A. Mochida. 2013. First and second law analysis of the developed solar-desiccant air-conditioning system (SDACS) operation during the summer day. *Energy and Buildings* 60:239-251.

Friedlander, M. 2003. How certified ratings can improve system design. Seminar at ASHRAE Winter Annual Meeting, Chicago.

Gao, W.Z., Y.P. Cheng, A.G. Jiang, T. Liu, and K. Anderson. 2015. Experimental investigation on integrated liquid desiccant–Indirect evaporative air cooling system utilizing the Maisotsenko cycle. *Applied Thermal Engineering* 88:288-296.

Ge, T.S., Y.J. Dai, R.Z. Wang, and Y. Li. 2015. Performance of two-stage rotary desiccant cooling system with different regeneration temperatures. *Energy* 81:556-566.

Gio, P., D.L. Cipliski, and R.W. Besant. 1998. A testing and HVAC design methodology for air-to-air heat pipe heat exchangers. *International Journal of HVAC&R Research (now Science and Technology for the Built Environment)* 4(1):3-26.

Jain, S., and P.K. Bansal. 2007. Performance analysis of liquid desiccant dehumidification systems. *International Journal of Refrigeration* 30(5): 861-872.

Johnson, A.B., R.W. Besant, and G.J. Schoenau. 1995. Design of multi-coil run-around heat exchanger systems for ventilation air heating and cooling. *ASHRAE Transactions* 101(2):967-978.

Johnson, A.B., C.J. Simonsen, and R.W. Besant. 1998. Uncertainty analysis in the testing of air-to-air heat/energy exchangers installed in buildings. *ASHRAE Transactions* 104(18):1639-1650.

Kays, W.H., and M.E. Crawford. 1993. *Convective heat and mass transfer*. 3rd ed. McGraw-Hill, New York.

Liu, X., T. Zhang, Y. Zheng, and R. Tu. 2016. Performance investigation and exergy analysis of two-stage desiccant wheel systems. *Renewable Energy* 86:877-888.

Mathur, G.D. 1990a. Long-term performance prediction of refrigerant charged flat plate solar collector of a natural circulation closed loop. *ASME HTD* 157:19-27.

Mathur, G.D. 1990b. Indirect evaporative cooling using heat pipe heat exchangers. ASME Symposium, Thermal Hydraulics of Advanced Heat Exchangers, ASME Winter Annual Meeting, Dallas.

Mathur, G.D. 1992. Indirect evaporative cooling. *Heating/Piping/Air Conditioning* 64(4):60-67.

Mathur, G.D. 1993. Retrofitting heat recovery systems with evaporative coolers. *Heating/Piping/Air Conditioning* 65(9):47-51.

Mathur, G.D. 1997a. Performance enhancement of existing air conditioning systems. *Proceedings of Intersociety Energy Conversion Engineering Conference, Honolulu*, American Institute of Chemical Engineers, Paper #97367, pp. 1618-1623.

Mathur, G.D. 1997b. Using heat pipe heat exchangers for reducing high energy costs of treating ventilation air. *Proceedings of Intersociety Energy Conversion Engineering Conference*, American Institute of Chemical Engineers, vol. 2, pp. 1447-1452.

Mathur, G.D. 1997c. Predicting yearly energy savings using B1N weather data with heat pipe heat exchangers. *Proceedings of Intersociety Energy Conversion Engineering Conference*, American Institute of Chemical Engineers, vol. 2, pp. 1391-1396.

Mathur, G.D. 1997d. Performance enhancement of existing air conditioning systems. *Proceedings of Intersociety Energy Conversion Engineering Conference*, American Institute of Chemical Engineers, vol. 3, pp. 1618-1623.

Mathur, G.D. 2000. Controlling space humidity with heat pipe heat exchangers. *Proceedings of Intersociety Energy Conversion Engineering Conference*, American Institute of Chemical Engineers, vol. 2, pp. 835-842.

Mathur, G.D., and T.W. McDonald. 1986. Simulation program for a two-phase thermosiphon-loop heat exchanger. *ASHRAE Transactions* 92(2A): 473-485.

Mathur, G.D., and T.W. McDonald. 1987. Evaporator performance of finned air-to-air two-phase thermosiphon loop heat exchangers. *ASHRAE Transactions* 98(2):247-257.

McDonald, T.W., and D. Shivprasad. 1989. Incipient nucleate boiling and quench study. *Proceedings of CLIMA 2000* 1:347-352. Sarajevo, Yugoslavia.

Moffitt, R. 2010. Personal communication and reference, *Trane Application Engineering Manual* SYS-APM003-EN.

Moffitt, R. 2010. Using energy recovery to improve dehumidification performance and control. *Seminar* 34, ASHRAE Winter Meeting, Orlando.

Moffitt, R. 2011. *Energy wheel capacity control*. Trane.

Mostafa, A.A., and M.M. Nausa. 2007. Heat pipe heat exchanger for heat recovery in air conditioning. *Applied Thermal Engineering* 27(4):795-801.

Moffitt, R. 2015. Dedicated outside air system with dual energy recovery used with distributed sensible cooling equipment. *ASHRAE Transactions*, Paper AT-15-C047.

Murphy, J. 2006. Temperature and humidity control in surgery rooms. *ASHRAE Journal*, June.

Phillips, E.G., R.E. Chant, B.C. Bradley, and D.R. Fisher. 1989a. A model to compare freezing control strategies for residential air-to-air heat recovery ventilators. *ASHRAE Transactions* 95(2):475-483.

Phillips, E.G., R.E. Chant, D.R. Fisher, and B.C. Bradley. 1989b. Comparison of freezing control strategies for residential heat recovery ventilators. *ASHRAE Transactions* 95(2):484-490.

Phillips, E.G., D.R. Fisher, R.E. Chant, and B.C. Bradley. 1992. Freeze-control strategy and air-to-air energy recovery performance. *ASHRAE Journal* 34(12):44-49.

Ruch, M.A. 1976. Heat pipe exchangers as energy recovery devices. *ASHRAE Transactions* 82(1):1008-1014.

Scofield, M., and J.R. Taylor. 1986. A heat pipe economy cycle. *ASHRAE Journal* 28(10):35-40.

Shah, R.K. 1981. Thermal design theory for regenerators. In *Heat exchangers: Thermohydraulic fundamentals and design*. S. Kakac, A.E. Bergles, and F. Mayinger, eds. Hemisphere Publishing, New York.

Shang, W., M. Wawryn, and R.W. Besant. 2001a. Air crossover in rotary wheels used for air-to-air heat and moisture recovery. *ASHRAE Transactions* 107(2).

Shang, W., H. Chen, R.W. Evitts, and R.W. Besant. 2001b. Frost growth in regenerative heat exchangers: Part I—Problem formulation and method of solution; Part II—Simulation and discussion. *Proceedings of ASME International Mechanical Engineering Congress and Expo*, November, New York.

Sheng, Y., Y. Zhang, and G. Zhang. 2015. Simulation and energy savings analysis of high temperature heat pump coupling to desiccant wheel air conditioning system. *Energy* 83:583-596.

Simonsen, C.J., W. Shang, and R.W. Besant. 2000. Part-load performance of energy wheels: Part I—Speed control. *ASHRAE Transactions* 106(1): 286-300.

Sparrow, E.M., J.P. Abraham, and J.C.K. Tong. 2001. An experimental investigation on a mass exchanger for transferring water vapor and inhibiting the transfer of other gases. *International Journal of Heat and Mass Transfer* 44 (November):4313-4321.

Sultan, M., I.I. El-Sharkawy, T. Miyazaki, B.B. Saha, and S. Koyama. 2015. An overview of solid desiccant dehumidification and air conditioning systems. *Renewable and Sustainable Energy Reviews* 46:16-29.

Tu, R., X.-H. Liu, and Y. Jiang. 2015a. Lowering the regeneration temperature of a rotary wheel dehumidification system using exergy analysis. *Energy Conversion and Management* 89:162-174.

Tu, R., Y. Liu, and Y. Jiang. 2015b. Irreversible processes and performance improvement of desiccant wheel dehumidification and cooling systems using exergy. *Applied Energy* 145:331-344.

Williams, S. 2007. *Assessment of liquid desiccant cooling technology*. Master's thesis, The University of Tennessee, Chattanooga.

Yau, Y.H. 2007. Application of a heat pipe heat exchanger to dehumidification enhancement in a HVAC system for tropical climates—A baseline performance characteristics study. *International Journal of Thermal Sciences* 46(2):164-171.

Yau, Y.H. 2008. The use of a double pipe heat exchanger system for reducing energy consumption of treating ventilation air in an operating theatre—A full year energy consumption model simulation. *Energy and Buildings* 40(5):917-925.

Wu, X.P., P. Johnson, and A. Akbarzadah. 1997. Application of heat pipe heat exchangers to humidity control in air conditioning systems. *Applied Thermal Engineering* 17(6):561-568.

Zeng, D.Q., H. Li, Y.J. Dai, and A.X. Xie. 2014. Numerical analysis and optimization of solar hybrid one-rotor two-stage desiccant cooling and heating system. *Applied Thermal Engineering* 73:474-483.

BIBLIOGRAPHY

AHRI. 2014. Selecting, sizing, and specifying packaged air-to-air energy recovery ventilation equipment. *AHRI Guideline W*. Air-Conditioning, Heating, and Refrigeration Institute, Arlington, VA.

Andersson, B., K. Andersson, J. Sundell, and P.A. Zingmark. 1992. Mass transfer of contaminants in rotary enthalpy exchangers. *Indoor Air* 93(3): 143-148.

ASHRAE. 1974. Symposium on heat recovery. *ASHRAE Transactions* 80(1):302-332.

CSA. 1988. Standard methods of test for rating the performance of heat-recovery ventilators. CAN/CSA-C439-88. Canadian Standards Association, Rexdale, ON.

Cipliski, D.L., C.J. Simonsen, and R.W. Besant. 1998. Some recommendations for improvements to ASHRAE Standard 84-1991. *ASHRAE Transactions* 104(18):1651-1665.

Dhamshala, P., and S. Patel. 2016. *Use of heat and desiccant wheels in combination with a heat pump to reduce energy cost savings*. Master's thesis, The University of Tennessee, Chattanooga.

Dehl, F., T. Kuna, and N. Shrahama. 1993. A new development for total heat recovery wheels. *Energy Impact of Ventilation and Air Infiltration, 14th AIVC Conference, Copenhagen, Denmark*, pp. 261-268.

Inoue, T., and M. Monde. 2009. Operating limit of heat transport in two-phase thermosiphon with connecting pipe. *International Journal of Heat & Mass Transfer* 52:4519-4524.

Jouhara, H., and R. Meskimm. 2010. Experimental investigation of wraparound loop heat pipe heat exchanger used in energy efficient air handling units. *Energy* 35(12):4592-4599.

Jeffus, L. 2003. *Refrigeration and air conditioning: An introduction to HVAC/R*. Pearson Prentice Hall, Upper Saddle River, NJ.

Mathur, G.D. 1990. Indirect evaporative cooling using two-phase thermosiphon loop heat exchangers. *ASHRAE Transactions* 96(1):1241-1249.

Mathur, G.D., and T.W. McDonald. 1987. Evaporator performance of finned air-to-air two-phase thermosiphon loop heat exchangers. *ASHRAE Transactions* 98(2):247-257.

Mathur, G.D. 1996. Enhancing performance of an air conditioning unit system with a two-phase heat recovery loop retrofit. *Proceedings of Intersociety Energy Conversion Engineering Conference*, vol. 3, pp. 2027-2032.

McDonald, T.W., and D. Shivprasad. 1989. Incipient nucleate boiling and quench study. *Proceedings of CLIMA 2000* 1:347-352. Sarajevo, Yugoslavia.

Ninomura, P.T., and R. Bhargava. 1995. Heat recovery ventilators in multifamily residences in the Arctic. *ASHRAE Transactions* 101(2):961-966.

Nole-Baghdan, S.H., and G.R. Majidjan. 2000. Waste heat recovery using heat pipe heat exchanger (HPHE) for surgery rooms in hospitals. *Applied Thermal Engineering* 20(14):1271-1282.

Shang, W., and R.W. Besant. 2001. Energy wheel effectiveness evaluation: Part I—Outlet airflow property distributions adjacent to an energy wheel; Part II—Testing and monitoring energy wheels in HVAC applications. *ASHRAE Transactions* 107(2).

Sheng, Y., Y. Zhang, Y. Sun, L. Fang, J. Nie, and L. Ma. 2014. Experimental analysis and regression prediction of desiccant wheel behavior in high temperature heat pump and desiccant wheel air-conditioning system. *Energy and Buildings* 80:358-365.

Simonsen, C.J., and R.W. Besant. 1997. Heat and moisture transfer in desiccant coated rotary energy exchangers: Part I—Numerical model; Part II—Validation and sensitivity studies. *International Journal of HVAC&R Research (now Science and Technology for the Built Environment)* 3(4): 325-368.

Simonsen, C.J., and R.W. Besant. 1998. Heat and moisture transfer in energy wheels during sorption, condensation and frosting. *ASHRAE Journal of Heat Transfer* 120(3):699-708.

Simonsen, C.J., and R.W. Besant. 1998. Energy wheel effectiveness: Part I—Development of dimensionless groups; Part II—Correlations. *International Journal of Heat and Mass Transfer* 42(12):2161-2186.

Simonsen, C.J., D.L. Cipliski, and R.W. Besant. 1999. Determining performance of energy: Part I—Experimental and numerical methods; Part II—Experimental data and validation. *ASHRAE Transactions* 105(1): 177-205.

SMACNA. 1978. *Energy recovery equipment and systems*. Report.

Sparrow, E.M., J.P. Abraham, J.C. Tong, and G.L. Martin. 2001. Air-to-air energy exchanger test facility for mass and energy transfer performance. *ASHRAE Transactions* 107(2):450-456.

Stauder, F.A., and T.W. McDonald. 1986. Experimental study of a two-phase thermosiphon-loop heat exchanger. *ASHRAE Transactions* 92(2A):486-497.

Stauder, F.A., G.D. Mathur, and T.W. McDonald. 1985. Experimental and computer simulation study of an air-to-air two-phase thermosiphon-loop heat exchanger. *ASME 85-WA/HT-15*.

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