CHAPTER 25

AIR-TO-AIR ENERGY RECOVERY

AIR-TO-AIR energy recovery is the process of recovering energy or/and moisture from an airstream at a high temperature or humidity to an airstream at a low temperature or humidity. 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 outside 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 outside air to meet minimum ventilation requirements as prescribed in ASHRAE *Standards* 62.1 and 62.2.

Types of ERVs include compact air-to-air cross-flow heat exchangers, rotary wheels, heat pipes, runaround loops, thermosiphons, and twin-tower enthalpy recovery loops. Performance is typically measured by effectiveness, pressure drop or pumping power of fluids, cross-flow, (the amount of air leakage from one stream to the other), and frost control (used to prevent frosting on the heat exchanger). 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.

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 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.

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. Typical air-to-air energy recovery applications are listed in Table 1.

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. Process-to-process applications usually recover the maximum amount of energy. 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 heat 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.

Table 1 Applications for Air-to-Air Energy Recovery

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

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

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 outside 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 outside 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 outside supply air.

BASIC RELATIONS

The second law of thermodynamics states that heat energy always transfers from a region of high temperature to one of low temperature. This law can be extended to say that mass transfer always occurs from a region of high vapor pressure to one of low vapor pressure. The 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_3. Because conditions at x_3 are lower than conditions at x_1, heat and mass transfer from the supply airstream to the exhaust airstream because of differences in temperature and vapor pressures across the separating wall. Consequently, the supply air exit properties decrease, while those of the exhaust air increase. Exit properties of these two streams can be estimated, knowing the flow rates and the effectiveness of the heat exchanger.

ASHRAE *Standard* 84 defines effectiveness as

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

Heat Recovery Ventilators

From Figure 1, the sensible effectiveness ε_s of a heat recovery ventilator is given as

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

where q_s is the actual sensible heat transfer rate given by

$$q_s = \varepsilon_s\, q_{s,max} \qquad (2b)$$

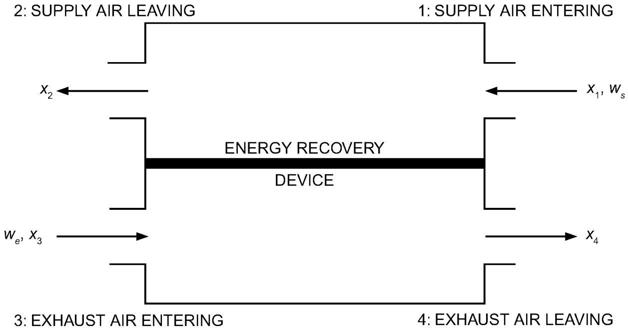

Fig. 1 Airstream Numbering Convention

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

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

where

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

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

and the leaving exhaust air condition is

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

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 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

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

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

$$q_s = 60 \varepsilon_s\, m_{min}\, c_p (t_1 - t_3) \qquad (3e)$$

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/ft^3
ρ_e = density of dry exhaust air, lb/ft^3
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 (4).

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, and x_4.

Energy Recovery Ventilators

The ERV allows the transfer of both sensible and latent heat, the latter due to 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

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

where q_L is the actual latent heat transfer rate given by

$$q_L = \varepsilon_L q_{L,max} \quad (4b)$$

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

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

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

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

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

$$m_w = \varepsilon_m m_{w,max} \quad (4e)$$

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

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

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

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

and the leaving exhaust air humidity ratio is

$$w_4 = w_3 + \varepsilon_L \frac{m_{w,min}}{m_s}(w_1 - w_3) \quad (5b)$$

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

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

where q_t is the actual total heat transfer rate given by

$$q_t = \varepsilon_t q_{t,max} \quad (6b)$$

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

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

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

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

and the leaving exhaust air condition is

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

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

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

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

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

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

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

$$\begin{aligned} q_t &= q_s + q_L = 60 m_e(h_{4e} - h_{3e}) = 60 Q_e \rho_e(h_{4e} - h_{3e}) \\ &= 60[m_e c_{pe}(t_4 - t_3) + m_e h_{fg}(w_4 - w_3)] \end{aligned} \quad (10a)$$

$$q_t = 60 \varepsilon_t m_{min}(h_{1s} - h_{3e}) \quad (10b)$$

where

h_{1s} = enthalpy of supply air at inlet, Btu/lb
h_{3e} = enthalpy of exhaust air at inlet, Btu/lb
h_{2s} = enthalpy of supply air at outlet, Btu/lb
h_{4e} = enthalpy of exhaust air at outlet, Btu/lb

ERVs 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. An airstream flowing through an ERV may gain heat energy ($+q_s$) from the adjoining stream, but will lose the latent energy ($-q_L$) if it transfers the water vapor to the adjoining stream, because of transfer of moisture. 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

$V_1 = 14.14$ ft³/lb $\quad h_1 = 30.6$ Btu/lb $\quad w_1 = 0.0071$ lb/lb of dry air

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

$V_3 = 13.68$ ft³/lb $\quad h_3 = 28.15$ Btu/lb $\quad w_3 = 0.0093$ lb/lb of dry air

The mass flow rate at state 1 is obtained from

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

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

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

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:

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

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

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

$$q_s = (660\ \text{lb/min})(0.24\ \text{Btu/lb}\cdot°\text{F})(85°\text{F} - 75°\text{F}) = 1584\ \text{Btu/min}$$

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

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

where

$$Re_D = \frac{(\rho V)_{av} D_h}{\mu_{\bar{T}}} \tag{15}$$

where

ρ = air density, lb/ft^3
V = average velocity in flow channels, fpm
D_h = hydraulic diameter of flow channels, ft
$\mu_{\bar{T}}$ = dynamic viscosity at average temperature $\bar{T}$, 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

$$\frac{\Delta p}{\Delta p_0} = \left(\frac{m}{m_0}\right)\left(\frac{\bar{T}}{\bar{T}_0}\right)^{3/2}\left(\frac{1 + C/\bar{T}_0}{1 + C/\bar{T}}\right) \tag{16}$$

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 due to 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

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 outside ventilation air as it passes through the exchanger to the indoor space. Heat flows from the incoming outside 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 outside 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 outside 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.

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 an effectiveness approaching 100%, but typical units have a lower effectiveness. Normal effectiveness for cross-flow heat exchangers is 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.

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/crossflow, 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 and permeable-walled flat-plate energy recovery units are used because of their moisture exchange function. Passage of air or other gases (e.g., pollutants) across the exchanger is a negative consequence. As well, some cross-stream mass transfer may 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.

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. The following facts about heat/mass exchanger performance must be recognized:

- The effectiveness for moisture transfer may not equal the effectiveness for heat transfer.
- The 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.

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

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

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 $m_s c_{ps}$ and $m_e c_{pe}$

Figure 7 depicts the variation of effectiveness with NTU for a rotary heat wheel.

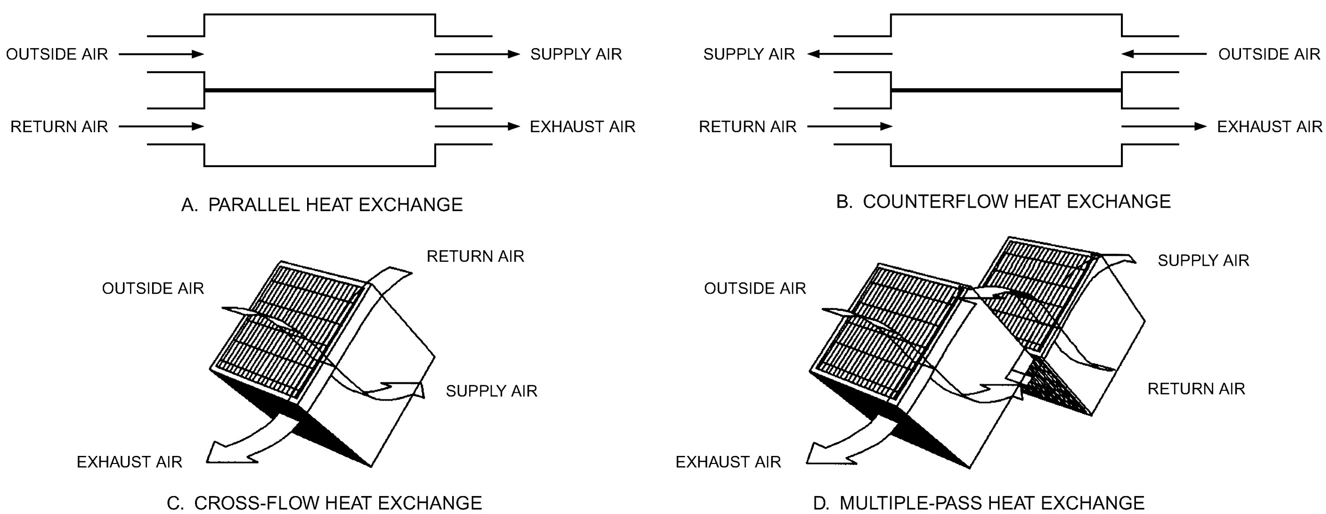

Fig. 2 Heat Exchanger Airflow Configurations

ADDITIONAL TECHNICAL CONSIDERATIONS

The rated effectiveness of energy recovery units is typically 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 due to design for positive building pressure, the presence of air leakage, fouling, condensation or frosting and several other factors as described below.

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. Cross-flow air leakage results from inadequate sealing construction between ambient and cross-stream seal interfaces. 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. Air leakage is seldom zero because external and internal air pressures are usually different, causing air to leak from higher-pressure regions to lower-pressure regions.

Cross-flow air leakage is usually caused by pressure differentials between airstreams. Carryover air leakage (specific to wheels) is caused by continuous rotation of trapped exhaust air in cavities in the heat transfer surface, which reverses airflow direction as the wheel rotates and spills this exhaust air into the supply airstream.

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. Cross-leakage varies with heat exchanger type and design, airstream static pressure differences, and the physical condition of the heat exchanger (see Table 2).

Air leakage between incoming fresh air and outgoing exhaust air is comprised of two paths called cross-flow and carryover leakage. **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. The quantitative estimate of the air leakage is expressed by two dimensionless parameters: the exhaust air transfer ratio (EATR) and outside air correction factor (OACF).

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

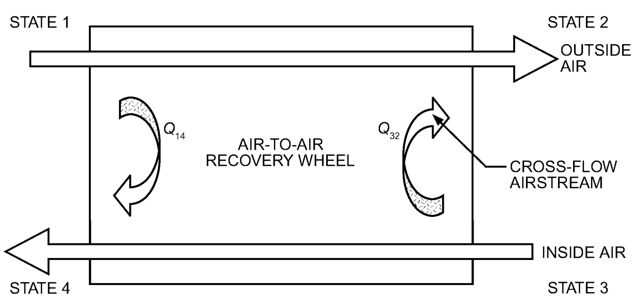

Fig. 3 Air Leakage in Energy Recovery Units

where c_1, c_2, and c_3 are the concentrations of inert gas at states 1, 2 and 3, respectively. Note that EATR represents an exhaust air leakage based on observed relative concentration of inert gas in supply airflow.

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

where m_1 and m_2 are the mass flow rates of the incoming fresh airstream at state 1 and 2, respectively. OACF helps estimate the extra quantity of outside air required at the inlet to compensate for the air that leaks into or out of the exchanger, and to meet the required net supply airflow to the building space. Ideal airflow 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 are useful in comparing HRVs and ERVs and in evaluating the actual flow rate of outside air supplied for a given ventilation requirement or in estimating the capacity of ventilator fans, as illustrated by the following.

Air Capacity of Ventilator Fans

For a given ventilation requirement Q_v, the volume flow rate capacity Q_1 of the supply fan is greater if air leakage exists (see Figure 3).

EATR is the percentage of supply air Q_2 that is made up of exhaust air Q_3 that has leaked Q_{32} through the device.

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

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, and negligible variation in air densities, the quantity of air leaving the building space (Q_3) should be same as air supplied to the space Q_2.

$$Q_3 = Q_2 = \frac{Q_v}{\left(1 - \frac{\text{EATR}}{100}\right)} \tag{22}$$

Assuming negligible change in air density and after substituting Equation (22) into Equation (21) gives

$$Q_1 = Q_2(\text{OACF}) = \frac{Q_v(\text{OACF})}{\left(1 - \frac{\text{EATR}}{100}\right)} \tag{23}$$

Equation (23) gives the volume flow rate capacity of the intake, which equals the exhaust 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).]

Cross-stream mass transfer of air and water vapor can be driven by two independent types of pressure differences: (1) cross-stream total pressure differences and (2) cross-stream partial pressure differences. Air mass movement is driven primarily by air pressure differences and is minimized by a high bulk-flow resistance of the exchanger wall barrier and adjacent air seals. Moisture mass transfer is driven by a combination of air pressure differences and vapor partial-pressure differences. Cross-stream moisture transfer is maximized by a low bulk-flow resistance, high moisture adsorption/desorption characteristics of desiccant coatings in total energy wheels, high moisture absorption/desorption characteristics of twin-tower coupling desiccant liquid, and high permeability of the exchanger wall barrier in permeable-walled flat-plate energy recovery units. High bulk-flow resistance retards viscous bulk airflow and minimizes the effect of air pressure differences on air mass transfer. Air-pressure- and partial-pressure-driven mass transfer may be additive or subtractive.

Heat, moisture, and air transfer rates are sometimes (but not generally) independent of and separate from one another. Heat is always driven cross-stream from higher to lower temperature. Air is predominantly driven cross-stream from higher to lower air pressure. Water vapor mass is driven cross-stream in an amount and direction influenced by several variables. Design and construction characteristics of the exchanger greatly influence whether the moisture mass is (1) transferred predominantly by riding on the cross-stream air mass or (2) separated from the air mass by a permeable desiccant, selective microporous membrane, or other moisture-separating device. The net effect of vapor pressure differences cross-stream influences the net intensity and direction of moisture exchange.

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 soot blowers. The cleaning method should be determined during design so that a compatible heat exchanger can be selected.

Cleaning frequency depends on the quality of the exhaust airstream. Residential and commercial HVAC systems generally require only infrequent cleaning; industrial systems, usually more. Equipment suppliers should be contacted regarding the specific cleaning and maintenance requirements of the systems being considered.

Filtration

Filters 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. If entrance and exit effects are neglected, 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 dewpoint, 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; the 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.

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. Total heat 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. In enthalpy heat exchangers, which use chemical absorbents, condensation may cause the absorbents to deliquesce, resulting in loss of absorbent.

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 the supply air, or reducing heat exchanger effectiveness (e.g., reducing heat wheel speed, tilting heat pipes, or bypassing part of the supply air around the heat exchanger). Alternatively, the heat exchanger may be periodically defrosted.

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

For sensible heat exchangers, system design should include drains to collect and dispose of condensation, which occurs in the warm airstream. In comfort-to-comfort applications, condensation may occur in the supply side in summer and in the exhaust side in winter.

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Frost Blockage and Control in Air-to-Air Exchangers

Air-to-air exchangers are widely used in HVAC building applications where outside air temperatures are often well below freezing in winter. As a result, condensation and freezing may occur in the exhaust airstream or both the exhaust and supply for regenerative heat or energy wheels. Condensation occurs in the exchanger when and where the local exchanger surface temperature is below the dew-point temperature but above the frost point. Freezing occurs when and where the surface temperatures are below the frost point. Low-density frost grows when surface temperatures are well below freezing.

Design should allow for continuous removal of any condensation. Ice and frost must be prevented or cyclically removed. Generally, frost growth is considered to be more serious because it can degrade exchanger performance, sometimes within only minutes. The consequences of frost growth are usually (1) increased pressure drop, (2) decreased mass flow rates of air, (3) decreased exchanger effectiveness, and (4) increased OACF.

Predicting frosting conditions for a given exchanger is usually a function of its operating conditions and performance factors. For example, for counterflow air-to-air exchangers, which essentially includes all types of exchanger except crossflow, an equation for the frost onset threshold of the supply air inlet temperature T_1 is given by:

$$T_{1f}(°\mathrm{F}) = \left(\frac{-6.0\dot{m}_e}{\varepsilon_s \dot{m}_e}\right)(1 + 3.5\varepsilon_L \phi_3)\left[1 + \frac{0.012\dot{m}_e}{\varepsilon_s \dot{m}_e}(T_3 - 20)\right] \quad (24)$$

with no frost control and $T_0 = t_1 < T_{1f}$, where T_{1f} is the threshold temperature when frost will grow without limit. The sensible effectiveness $\varepsilon_s > 0.5$ for energy wheels and $\varepsilon_l = 0.5$ for sensible heat exchangers even though there is no moisture transfer between the airstreams.

This equation suggests several methods for steady-state frost control when outside ambient air temperature T_o is less than T_{1f}: (1) preheat supply air so that $T_1 \geq T_{1f}$; (2) preheat exhaust air so that $T_{1f} \geq T_1 = T_0$; (3) bypass some supply air to increase exhaust to supply air mass flow rate m_e/m_s, which results in a new $T_{1f} \leq T_1 = T_0$; (4) increase exhaust air humidity for energy wheels so that $T_{1f} \leq T_1 = T_0$; (5) decrease the value of the sensible effectiveness (e.g., decreasing the wheel speed of energy wheels, tilt control on heat pipes, changing the liquid flow rate in the loop on runaround systems); or (6) any combination of these five methods.

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.

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. Similar frost control strategies can be used for counterflow exchangers, but Equation

(24) must be adapted: frost forms in the exhaust airflow passages whenever the outside air temperatures bring any part of the exhaust channel surfaces below the frost point temperature of about 23°F.

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

PERFORMANCE RATINGS

Standard laboratory rating tests and predictive computer models give exchanger performance values for (1) heat transfer, (2) moisture transfer, (3) cross-stream air transfer, (4) average exhaust mass airflow, and (5) supply mass airflow leaving the exchanger. Effectiveness ratios for heat and mass water vapor transfer must be separately determined by rating tests in a laboratory that is staffed and instrumented to meet requirements of ASHRAE *Standard* 84 and ARI *Standard* 1060. It may be very difficult to adhere to any standard when field tests are made.

ASHRAE *Standard* 84, Method of Testing Air-to-Air Heat Exchangers, (1) establishes a uniform method of testing for obtaining performance data; (2) specifies the data required, calculations to be used, and reporting procedures for testing each of seven independent performance factors and their uncertainty limits; and (3) specifies the types of test equipment. The independent performance factors specified by *Standard* 84 are sensible (ε_s), latent (ε_l), and total (ε_t) effectivenesses; supply (ΔP_s) and exhaust (ΔP_e) air pressure drops; exhaust air transfer ratio (EATR), which characterizes the fraction of exhaust air transferred to the supply air; and outside air correction factor (OACF), which is the ratio of supply inlet to outlet air flow.

ARI *Standard* 1060, Rating Air-to-Air Energy Recovery Ventilation Equipment, is an industry-established standard for rating air-to-air heat/energy exchanger performance for use in energy recovery ventilation equipment. This standard, based on ASHRAE *Standard* 84, establishes definitions, requirements for marking and nameplate data, and conformance conditions intended for the industry, including manufacturers, engineers, installers, contractors, and users. Standard temperature and humidity conditions at which equipment tests are to be conducted are specified for summer and winter conditions. Published ratings must be reported for each of the seven performance factors specified in ASHRAE *Standard* 84. The ARI certification program using *Standard* 1060 is used to verify ratings published by manufacturers.

ARI *Standard* 1060 requires balanced airflow rates (see Figure 3) and the following conditions:

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

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

Balanced mass airflows, as required for some ASHRAE and ARI standard test methods, are rarely achieved in field operation for air-handling systems. Fans are nearly constant-volume devices usually designed to run at a preset rpm. Significantly more mass airflow will be transported in cold (winter) conditions than in hot (summer) conditions.

For estimating changes in exchanger performance factors at each operating condition, ASHRAE *Standard* 84 specifies knowledge of seven performance factors (i.e., ΔP_s, ΔP_e, ε_s, ε_L, ε_t, EATR, and OACF), but ARI *Standard* 1060 certifies performance at only a few standard operating conditions. At other operating conditions, these performance factors must be extrapolated using accepted correlations.

Variables that can affect these performance factors for total energy transfer or sensible heat transfer devices include (1) water vapor partial-pressure differences; (2) heat transfer area; (3) air velocity or mass flow rates through the heat exchangers; (4) airflow arrangement or geometric configuration, or characteristic dimension of the flow passage through the recovery ventilator; and (6) method of frost control. The effect of frost control method on seasonal performance is discussed in Phillips et al. (1989a), and sensible versus latent heat recovery for residential comfort-to-comfort applications is addressed in Barringer and McGugan (1989b).

Current testing standards do not validate exchanger performance for testing conditions that require freezing or condensing temperatures, unbalanced airflow ratios, high pressure differentials, or air leakage rates based on varying the inputs. An alternative test setup using a multifunction wind tunnel facility is presented by Sparrow et al. (2001) to provide comprehensive performance data for conditions other than existing standards.

DESIGN CONSIDERATIONS OF VARIOUS ERV SYSTEMS

Fixed-Plate Heat Exchangers

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

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

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

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

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

Fixed-plate heat exchangers can achieve high sensible heat recovery and total energy effectiveness because they have only a primary heat transfer surface area separating the airstreams and are therefore not inhibited by the additional secondary resistance (e.g., pumping liquid, in runaround systems or transporting a heat transfer medium) inherent in some other exchanger types. In a cross-flow arrangement (Figure 4), they usually do not have sensible effectiveness greater than 75% unless two devices are used in series as shown in Figure 2D.

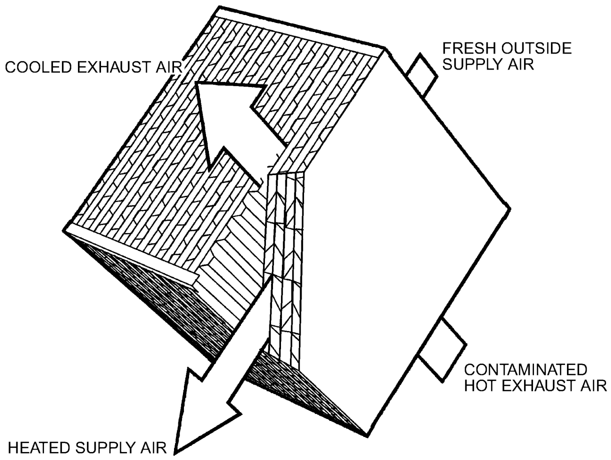

Fig. 4 Fixed-Plate Cross-Flow Heat Exchanger

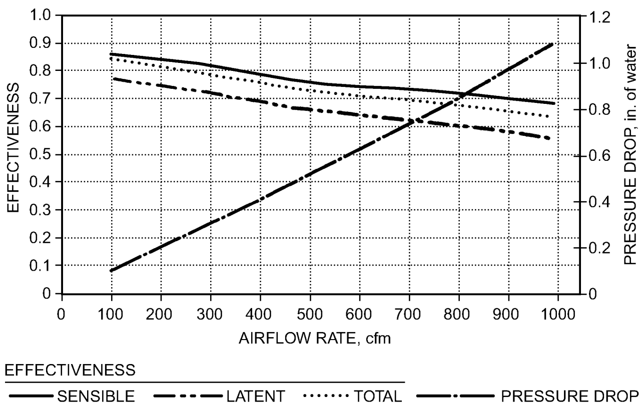

Fig. 5 Variation of Pressure Drop and Effectiveness with Air Flow Rates for a Membrane Plate Exchanger

One advantage of the plate exchanger is that it is a static device with little or no leakage between airstreams. As velocity increases, the pressure difference between the two airstreams increases (Figure 5). High differential pressures may deform the separating plates and, if excessive, can permanently damage the exchanger, significantly reducing the airflow rate on the low-pressure side as well as the effectiveness and possibly causing excessive air leakage. This is not normally a problem because differential pressures in most applications are less than 4 in. of water. In applications requiring high air velocities, high static pressures, or both, plates are not recommended.

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

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

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

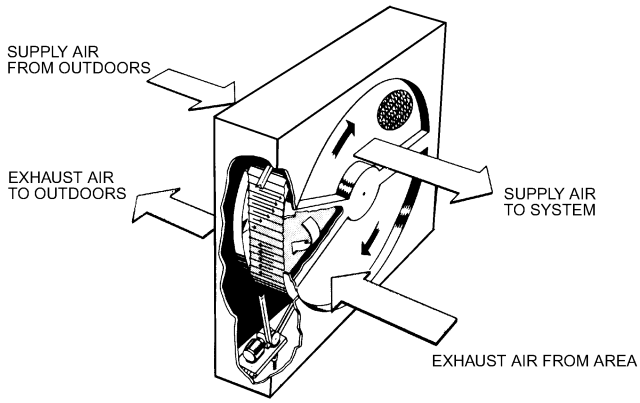

Fig. 6 Rotary Air-to-Air Energy Exchanger

sulphonation chemistry techniques and contain charged ions that attract polar water molecules; adsorption and desorption of water occur in vapor state.

Rotary Air-to-Air Energy Exchangers

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

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

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

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

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

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

Cross-Leakage. Cross-leakage, cross-contamination, or mixing between supply and exhaust airstreams occurs in all rotary energy exchangers by two mechanisms: carryover and seal leakage. **Carryover** occurs as air is entrained within the rotation medium and is carried into the other airstream. **Leakage** occurs because the differential static pressure across the two airstreams drives air from a higher to a lower static pressure region. Cross-contamination can be reduced by placing the blowers so that they promote leakage of outside air to the exhaust airstream. Carryover occurs each time a portion of the matrix passes the seals dividing the supply and exhaust airstreams. Because carryover from exhaust to supply may be undesirable, a **purge section** can be installed on the heat exchanger to reduce cross-contamination.

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

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

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

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

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

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

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

The second method regulates the energy recovery rate by varying wheel rotational speed. The most frequently used **variable-speed drives** are (1) a silicon controlled rectifier (SCR) with variable-speed dc motor, (2) a constant-speed ac motor with hysteresis coupling, and (3) an ac frequency inverter with an ac induction motor.

Figure 7 shows the effectiveness ε, sensible heat transfer only, with balanced airflow, convection-conduction ratio less than 4, and no leakage or cross-flow, of a regenerative counterflow heat wheel versus number of transfer units (NTU). This simple example of a regenerative wheel also shows that regenerative counterflow rotary effectiveness increases with wheel speed (C_r is proportional to wheel speed), but there is no advantage in going beyond C_r/C_{min} = 5 because the carryover of contaminants increases with wheel speed. See Shah (1981) or Kays and Crawford (1993) for details.

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

A dead band control, which stops or limits the exchanger, may be necessary when no recovery is desired (e.g., when outside air temperature is higher than the required supply air temperature but below the exhaust air temperature). When outside 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.

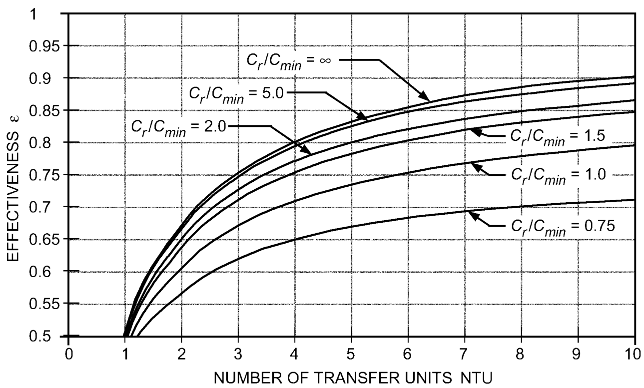

Fig. 7 Effectiveness of Counterflow Regenerator
(Shah 1981)

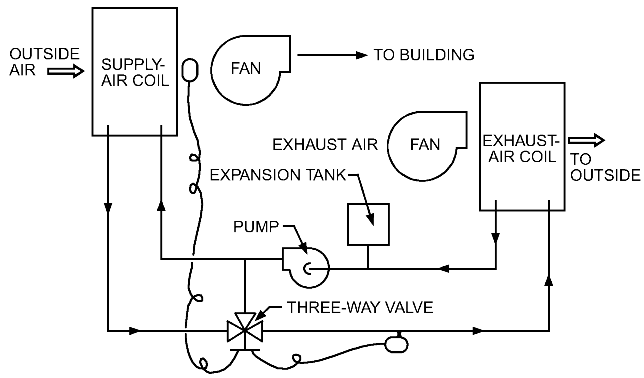

Fig. 8 Coil Energy Recovery Loop

Coil Energy Recovery (Runaround) Loops

A typical coil energy recovery loop (Figure 8) 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.

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

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

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

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.

The following example illustrates the capacity of a typical system:

Example 2. Runaround Loop Energy Recovery System. A waste heat recovery system heats 48,000 lb/h of air from a 0°F design outside temperature with an exhaust temperature of 75°F db and 60°F wb. Air flows through identical eight-row coils at 400 fpm. A 30% ethylene glycol solution flows through the coils at 3.4 cfm. Assuming the performance characteristic of the runaround loop as shown in Figure 9, determine the sensible effectiveness and exit temperature of the exhaust air (1) when the outside air temperature is at 0°F and (2) when it is at 18°F.

Solution:

Figure 9 shows the effect of outside air temperature on capacity, including the effects of the three-way temperature control valve. For this example, the capacity is constant for outside air temperatures below 18°F. This constant output of 413,000 Btu/h occurs because the valve has to control the temperature of fluid entering the exhaust coil to prevent frosting. As the exhaust coil is the source of heat and has a constant airflow rate, entering air temperature, liquid flow rate, entering fluid temperature (as set by the valve), and fixed coil parameters, energy recovered must be controlled to prevent frosting in the exhaust coil.

(1) Equation (2a) with the numerator equal to 413,000 Btu/h may be used to calculate the sensible heat effectiveness.

$$\varepsilon = \frac{413{,}000 \text{ Btu/h}}{(48{,}000 \text{ lb/h})(0.24 \text{ Btu/lb}\cdot\text{°F})(75-0)} = 48\%$$

From Equation (3b), the exhaust airstream exit temperature can be estimated as

$$t_4 = 75 + 0.48\frac{(48{,}000 \text{ lb/h})(0.24 \text{ Btu/lb}\cdot\text{°F})}{(48{,}000 \text{ lb/h})(0.24 \text{ Btu/lb}\cdot\text{°F})}(0-75) = 39\text{°F}$$

(2) When the three-way control valve operates at outside air temperatures of 18°F or lower, 413,000 Btu/h is recovered. Using the same equations, it can be shown that at 18°F, the sensible heat effectiveness is 64% and the exhaust air leaving dry-bulb temperature is 39°F. Above 60°F outside air temperature, the supply air is cooled with an evaporative cooler located upstream from the exhaust coil.

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

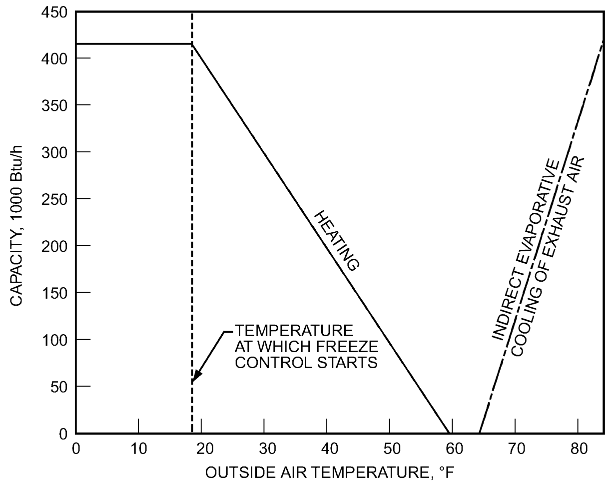

Fig. 9 Energy Recovery Capacity Versus Outside Air Temperature for Typical Loop

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. At temperatures 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 21 and 23 discuss the construction and selection of coils in more detail.

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

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

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

Heat Pipe Heat Exchangers

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

Energy transfer in heat pipes is often considered isothermal. However, there is a small temperature drop through the tube wall, wick, and fluid medium. Heat pipes have a finite heat transfer capacity that is affected by factors such as wick design, tube diameter, working fluid, and tube (heat pipe) orientation relative to horizontal.

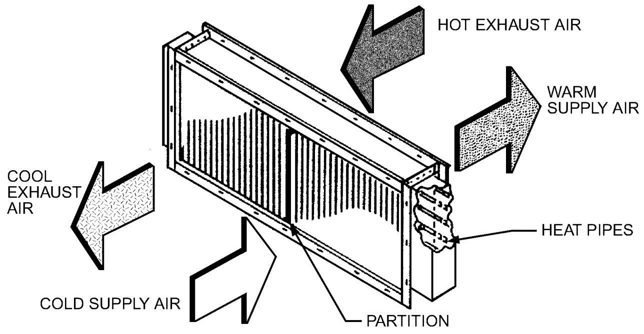

Fig. 10 Heat Pipe Assembly

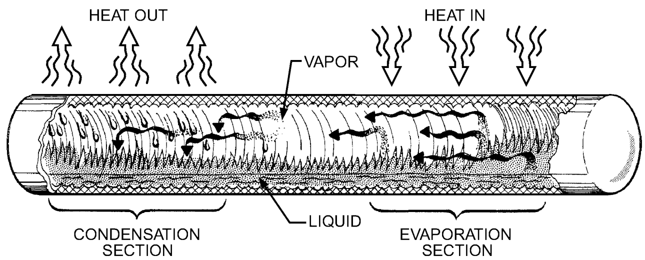

Fig. 11 Heat Pipe Operation

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

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

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

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

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

Heat pipe heat transfer capacity depends on design and orientation. Figure 12 shows a typical effectiveness curves for various face velocities and rows of tubes. As the number of rows increases, effectiveness increases at a decreasing rate. For example, doubling the number of rows of tubes in a 60% effective heat exchanger increases the effectiveness to 75%. The effectiveness of a counterflow heat pipe heat exchanger depends on the total number of rows such that

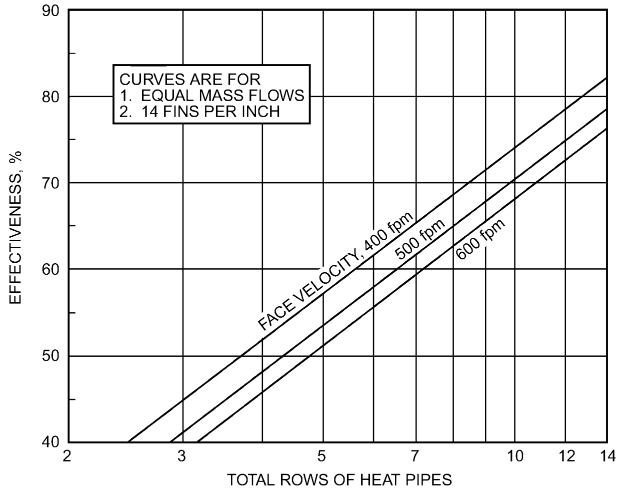

Fig. 12 Heat Pipe Exchanger Effectiveness

two units in series yield the same effectiveness as a single unit of the same total number of rows. Series units are often used to facilitate handling, cleaning, and maintenance. Effectiveness also depends on outside air temperature and the ratio of mass flow rates of the airstreams. Typically, heat capacity in the cooling season increases with a rise in outside air temperature. It has an opposite effect during the heating season. Effectiveness increases with the ratio of mass flow rates of the fluids (flow rate of the fluid with warmer entering temperature over that of cooler entering fluid temperature).

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

Heat transfer capacity is virtually independent of heat pipe length, except for very short heat pipes. For example, a 4 ft long heat pipe has approximately the same capacity as an 8 ft pipe. Because the 8 ft heat pipe has twice the external heat transfer surface area of the 4 ft pipe, it will reach its capacity limit sooner. Thus, in a given application, it is more difficult to meet capacity requirements as the heat pipes become longer. A system can be reconfigured to a taller face height and more numerous but shorter heat pipes to yield the same airflow face area while improving system performance.

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

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

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

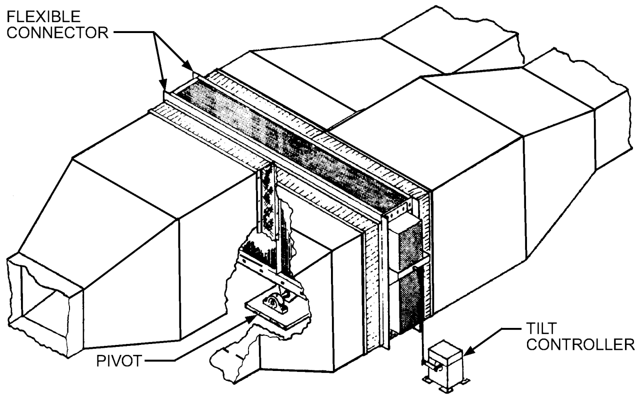

Fig. 13 Heat Pipe Heat Exchanger with Tilt Control

Tilt control may be desired

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

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

Bidirectional heat transfer in heat pipes is achieved through recent design improvements. Some heat pipe manufacturers have eliminated the need for tilting for capacity control or seasonal changeover. Once installed, the unit is removed only for routine maintenance. Capacity and frost control can also be achieved through bypassing airflow over the heat pipes, as for air coils.

Example 3. Sensible Heat Energy Recovery in a Heat Pipe

Outside air at 50°F enters a six-row heat pipe with a flow rate of 660 lb/min and a face velocity of 500 fpm. Exhaust air enters the heat pipe with the same velocity and flow rate but at 75°F. The pressure drop across the heat pipe is 0.6 in. of water. The supply air density is 0.08 lb/ft^3. The efficiency of the electric motor and the connected fan are 90 and 75%, respectively. Assuming the performance characteristics of the heat pipe are as shown in Figure 12, determine the sensible effectiveness, exit temperature of supply air to the space, energy recovered, and power supplied to the fan motor.

Solution:

From Figure 12, at face velocity of 500 fpm and with six rows, the effectiveness is about 58%. Because the mass flow rate of the airstreams is the same and assuming their specific heat of 0.24 Btu/lb·°F is the same, then the exit temperature of the supply air to the space can be obtained from Equation (3a):

$$t_2 = 50 - 0.58\frac{(660\ \text{lb/min})(0.24\ \text{Btu/lb}\cdot°\text{F})}{(660\ \text{lb/min})(0.24\ \text{Btu/lb}\cdot°\text{F})}(50 - 75) = 64.5°\text{F}$$

The sensible energy recovered can be obtained from Equation (3c) as

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

The supply air fan power can be obtained from Equation (11) as

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

$$P_s = [(8250\text{ft}^3/\text{min})(0.6)]/[(6356)(0.9)(0.75)] = 1.15\ \text{hp}$$

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

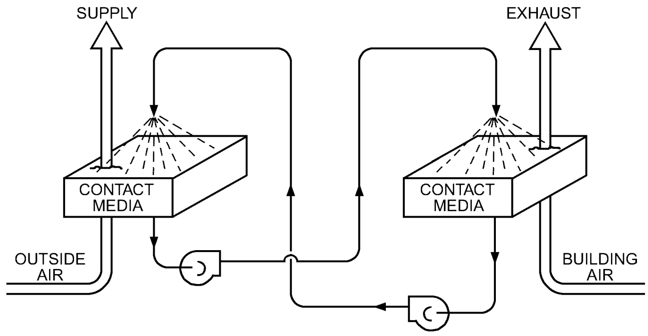

Fig. 14 Twin-Tower Enthalpy Recovery Loop

Twin-Tower Enthalpy Recovery Loops

In this air-to-liquid, liquid-to-air enthalpy recovery system, a sorbent liquid circulated between supply and exhaust contactor towers directly contacts both airstreams, transporting water vapor and energy between the airstreams (Figure 14). Supply air temperatures can be as high as 115°F or as low as –40°F. Any number of vertical and horizontal airflow contactor towers can be combined into a common system of any airflow capacity.

Leaving air passes through demister pads to remove entrained sorbent solution. Airstreams containing lint, animal hair, or other solids should be filtered upstream of the contactor towers. Wetted particles should be filtered from the sorbent solution, which minimizes particulate cross-contamination. Sorbent solutions (typically a halogen salt solution such as lithium chloride and water) usually contain bactericidal and viricidal additives. Testing has shown that contactor towers can effectively remove up to 94% of atmospheric bacteria, a desirable feature in health care applications. Limited gaseous cross-contamination may occur. If either airstream contains gaseous contaminants, their effects on the sorbent solution should be investigated.

In colder climates, moisture losses from the exhaust airstream may overdilute the sorbent solution. Heating the sorbent liquid entering the supply air contactor tower raises the discharge temperature and humidity of the leaving supply air, preventing overdilution. This, coupled with automatic makeup water addition, can maintain sorbent solution concentrations during cold weather, enabling the system to deliver air at a fixed humidity and temperature.

THERMOSIPHON HEAT EXCHANGERS

Two-phase thermosiphon heat exchangers are sealed systems that consist of an evaporator, a condenser, interconnecting piping, and an intermediate working fluid in both liquid and vapor phases. Two types of thermosiphon are used: a sealed tube (Figure 15) and a coil type (Figure 16). In the **sealed-tube thermosiphon**, the evaporator and the condenser are usually at opposite ends of a bundle of straight, individual thermosiphon tubes, and the exhaust and supply ducts are adjacent to each other (this arrangement is similar to that in a heat pipe system). In **coil-type thermosiphons**, evaporator and condenser coils are installed independently in the ducts and are interconnected by the working fluid piping (this configuration is similar to that of a coil energy recovery loop).

A thermosiphon is a sealed system containing a two-phase working fluid. Because part of the system contains vapor and part contains liquid, the pressure in a thermosiphon is governed by the liquid temperature at the liquid/vapor interface. If the surroundings cause a temperature difference between the regions where liquid and vapor interfaces are present, the resulting vapor pressure difference causes vapor to flow from the warmer to the colder region. The flow is sustained by condensation in the cooler region and by evaporation in the warmer region. The condenser and evaporator must be oriented so

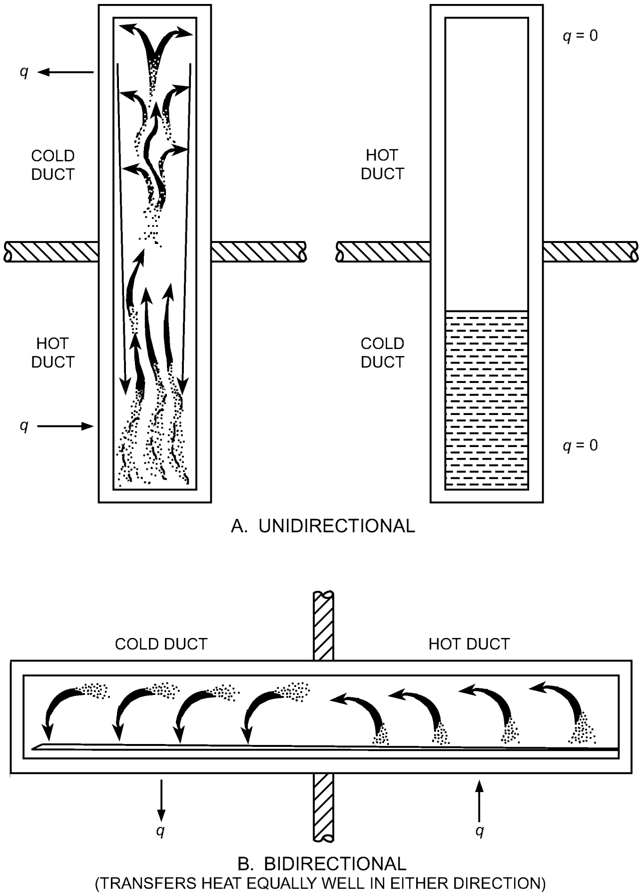

Fig. 15 Sealed-Tube Thermosiphons

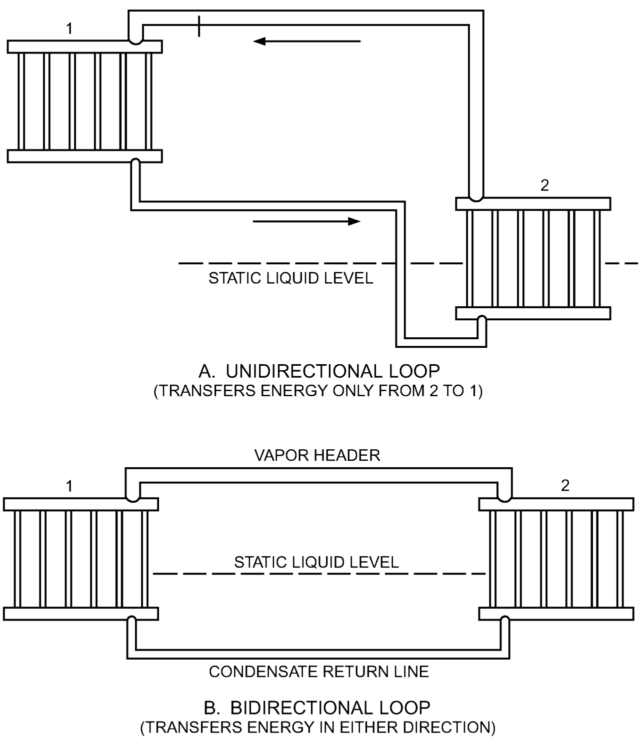

Fig. 16 Coil-Type Thermosiphon Loops

that the condensate can return to the evaporator by gravity (Figures 15 and 16).

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

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

Thermosiphon loops differ from other coil energy recovery loop systems in that they require no pumps and hence no external power supply, and the coils must be appropriate for evaporation and condensation. Two-phase thermosiphon loops are used for solar water heating (Mathur 1990a) and for performance enhancement of existing (i.e., retrofit applications) air-conditioning systems (Mathur 1997). Two-phase thermosiphon loops can be used to downsize new air-conditioning systems and thus reduce the overall project costs. Figure 17 shows thermosiphon loop performance (Mathur and McDonald 1986).

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. To compare them on payback period or maximum energy cost savings, accurate values of their capital cost, life, and maintenance cost, which vary from product to product for the same type of recovery system, must be known. Without such data, and considering the data available in the open literature such as that presented by Besant and Simonson (2003), use Table 2's comparative data for common types of air-to-air energy recovery devices.

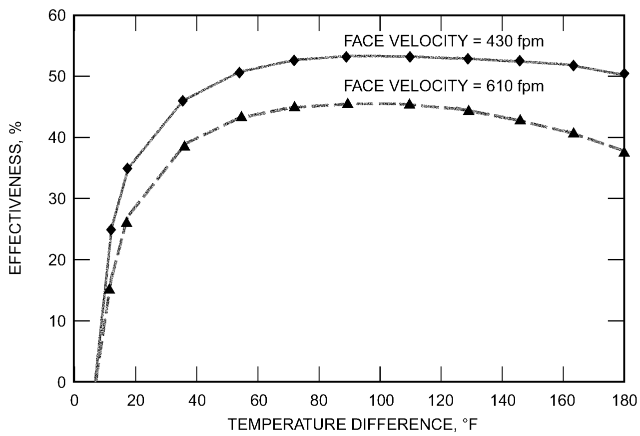

Fig. 17 Typical Performance of Two-Phase Thermosiphon Loop
(Mathur and McDonald 1986)

Table 2 Comparison of Air-to-Air Energy Recovery Devices

	Fixed Plate	Membrane Plate	Energy Wheel	Heat Wheel	Heat Pipe	Runaround Coil Loop	Thermosiphon	Twin Towers
Airflow arrangements	Counterflow Cross-flow	Counterflow Cross-flow	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 ($m_s = m_e$), %	50 to 80	50 to 75	50 to 85	50 to 85	45 to 65	55 to 65	40 to 60	40 to 60
Typical latent effectiveness,* %	—	50 to 72	50 to 85	0	—	—	—	—
Total effectiveness,* %	—	50 to 73	50 to 85	—	—	—	—	—
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 5	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	15 to 120	–65 to 1470	–65 to 1470	–40 to 105	–50 to 930	–40 to 105	–40 to 115
Typical mode of purchase	Exchanger only Exchanger in case Exchanger and blowers Complete system	Exchanger only Exchanger in case Exchanger and external 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 or 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 without 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

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

EATR = Exhaust Air Transfer Ratio
OACF = Outside Air Correction Factor

Long-Term Performance of Heat or Energy Recovery Ventilators

The type of seal used on a rotating wheel affects its performance as well as its capital cost. A measure of energy recovery ventilator performance is the relative magnitude of the actual energy recovered and the power supplied to the fans to circulate the airstreams. The cost of power supplied to the fans depends on the pressure drop of airstreams, volume flow rate, and the 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 ratio of energy recovered (RER) may be introduced to reflect the long-term performance of the recovery ventilators:

$$\mathrm{RER} = \frac{\int (\text{rate of energy recovered})\,dt}{\int (\text{rate of power supplied to the fan motors})\,dt} \tag{25}$$

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. However, the true overall system performance is the life-cycle cost or payback period, both of which take into account the capital and maintenance costs. Because of lack of sufficient data on these factors, they are not presented in Table 2.

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 2. Energy recovery is also available integrated into unitary air-conditioning equipment or in both standard and custom air-handling systems. Selection of such units primarily is dictated by the quantity of ventilation air. Several manufacturers have developed software or tables to help the user in selection of these units. The user may have to determine the required fan size (see Example 7), if only the heat exchanger is to be purchased.

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.

$$q_t = 60Q\rho(h_{in} - h_{out}) \qquad (26)$$

$$q_s = 60Q\rho c_p(t_{in} - t_{out}) \qquad (27)$$

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/ft^3
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.

Request air pressure drops Δp_s and Δp_e across the heat or energy exchanger from the manufacturer, who may have certified ARI *Standard* 1060 test condition data obtained using ASHRAE *Standard* 84 as a test procedure and analysis guide. These data may be extrapolated to non-ARI 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 m_{max} and q_{max}.

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

The theoretical maximum moisture, sensible heat, latent heat and total energy rates are given by Equations (4f), (2c), (4c), and (6c), respectively.

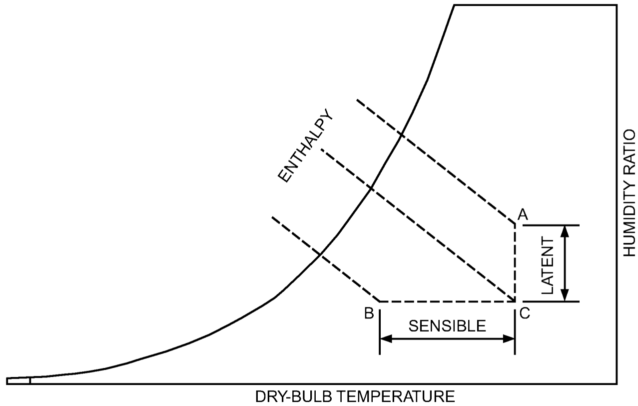

Fig. 18 Maximum Sensible and Latent Heat from Process A-B

The split between latent and sensible energy can be determined by plotting airstream conditions on a psychrometric chart as shown in Figure 18. 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 the moisture, sensible, and total effectiveness ε_s, ε_L, and ε_t.

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

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

The actual moisture, sensible heat, latent heat, and total energy rates are given by Equations (4e), (2b), (4b), and (6b), respectively. Note that ε_m (mass effectiveness) = ε_L (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 6. Check the 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.

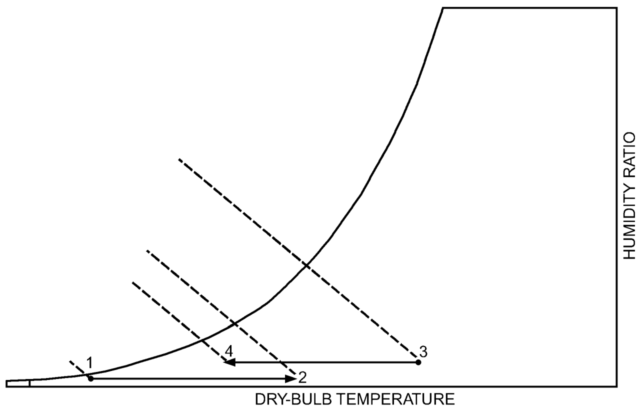

Fig. 19 Sensible Heat Recovery in Winter (Example 4)

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 ARI *Standard* 1060 test condition data obtained using ASHRAE *Standard* 84 as a test procedure and analysis guide. These data may be extrapolated to non-ARI test conditions using correlations relating EATR to air pressure differences between the supply and exhaust and, for rotary regenerative wheels, carryover due to 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 outside 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 ARI *Standard* 1060 test condition data obtained using ASHRAE *Standard* 84 as a test procedure and analysis guide. These data may be extrapolated to non-ARI 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 illustrated in Example 7.

Example 4. 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 (ρ = 0.087 lb/ft^3) using an air-to-air heat exchanger with a measured effectiveness of 60%. Airflows are specified in scfm, so an air density of 0.075 lb/ft^3 for both airstreams is appropriate. Assuming EATR = 0 and OACF ≈ 1, determine the leaving supply air temperatures and energy recovered, and check the heat exchange balance.

Solution:

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

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

2. Calculate the theoretical maximum heat transfer.

The two inlet conditions plotted on a psychrometric chart (Figure 19) 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

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

3. Establish the sensible effectiveness.

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

4. Calculate actual heat transfer at given conditions.

Using Equation (2b),

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

5. Calculate leaving air conditions.

Because no moisture or latent energy transfer will occur,

a. Leaving supply air temperature t_2 is given as

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

b. Leaving exhaust air temperature t_4 is given as

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

6. Using Equations (3c) and (3d), check performance.

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

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

7. Plot conditions on psychrometric chart to confirm that no moisture exchange occurred (Figure 19).

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

Example 5. Sensible Heat Recovery in Winter with Water Vapor Condensation

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

Solution:

The supply airstream has a lower airflow rate than the exhaust airstream, so it may appear that the supply airstream limits heat transfer. However, determination of mass flow rates for the given entry conditions shows that the mass flow rate of the supply airstream (47,900 lb/h) is slightly greater than that of the exhaust airstream (47,600 lb/h), so exhaust is the limiting airstream. Nevertheless, because the mass difference is negligible, it is convenient to use supply air volume as the limiting airstream.

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

2. Calculate the theoretical maximum sensible heat transfer.

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

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

3. Select sensible effectiveness.

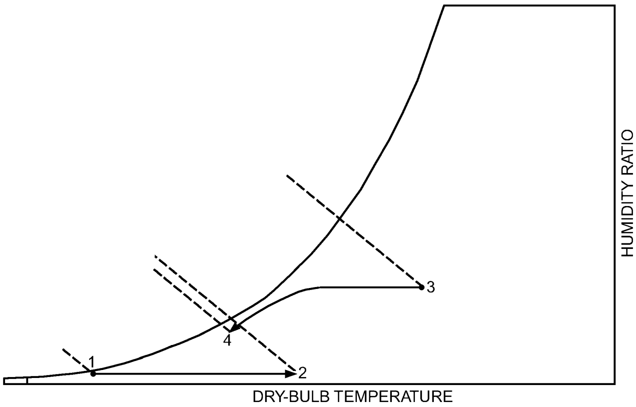

Fig. 20 Sensible Heat Recovery in Winter with Condensate (Example 5)

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

4. Calculate actual heat transfer at design conditions using Equation (2b):

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

5. Calculate leaving air conditions.

a. Leaving supply air temperature is calculated by

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

b. Because the dew point of exhaust air at inlet is 38.4°F, condensation occurs on the exhaust side, so the leaving exhaust air temperature cannot be determined using Equation (3b). The entering exhaust air enthalpy and humidity ratio are determined for the dry-bulb temperature of 75°F and 28% rh using a psychrometric chart and found to be h_3 = 23.7 Btu/lb and w_3 = 0.0052 lb/lb. However, the leaving exhaust air enthalpy can be determined by

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

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

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

6. Check performance.

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

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

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

7. Plot conditions on psychrometric chart (Figure 20). Note that moisture condenses in the exhaust side of the heat exchanger.

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

Example 6. Total Heat Recovery in Summer

Exhaust air at 75°F and 63°F wb (ρ = 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 (ρ = 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:

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

2. Calculate the theoretical maximum heat transfer.

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

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

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

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

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

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

3. Determine supply sensible and total effectiveness.

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

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

4. Calculate energy transfer at design conditions.

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

5. Calculate leaving air conditions.

a. Supply air conditions

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

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

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

b. Exhaust air conditions

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

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

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

6. Check total performance (Equations 9 and 10).

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

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

7. Plot conditions on psychrometric chart (Figure 21).

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

Example 7. Total energy recovery with EATR ≠ 0 and OACF ≠ 1.0

An ERV manufacturer claims a product has performance characteristics as shown in Figure 5. A building has a ventilation requirement of 850 cfm and exhaust air at 75°F and 63°F wb (ρ = 0.075 lb/ft³) is used to precool supply outdoor air at 95°F and 81°F wb (ρ = 0.072 lb/ft³).

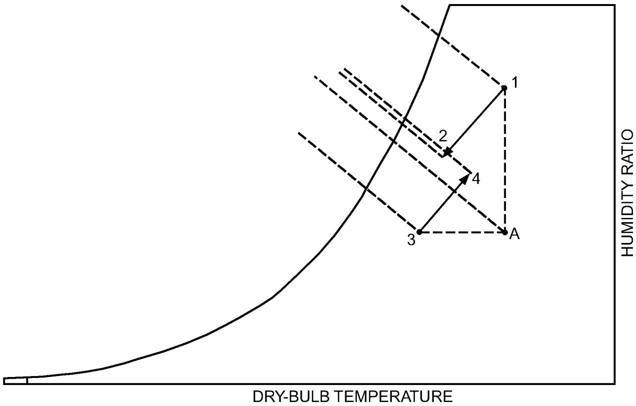

Fig. 21 Total Heat Recovery in Summer (Example 6)

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

(b) If EATR = 5% and OACF = 1.05, determine the actual air flow rates.

Solution:

1. From Figure 5, at a flow rate of 850 cfm, the pressure drop Δp = 0.9 in. of water. Assuming the effective efficiency of the fan motor combination is about 0.6, the power P_s required to circulate the supply air can be obtained from Equation (11) as

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

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

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

2. Calculate the theoretical maximum heat transfer.

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

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

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

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

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

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

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

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

3. Determine supply sensible and total effectiveness.

From Figure 5 at a flow rate of 850 cfm, ε_s = 0.73, ε_L = 0.68, and ε_t = 0.715.

4. Calculate energy transfer at design conditions.

q_s = (0.73)(17,600 Btu/h) = 12,850 Btu/h sensible recovered

q_L = (0.68)(40,800 Btu/h) = 27,750 Btu/h latent recovered

q_t = (0.715)(58,750 Btu/h) = 42,000 Btu/h total recovered

5. Calculate leaving air conditions.

a. Supply air conditions

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

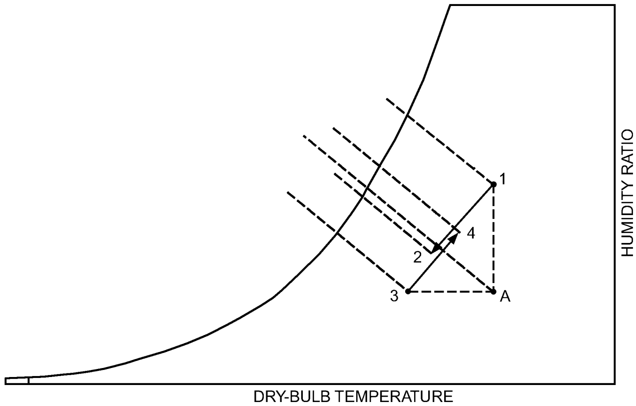

Fig. 22 Total Energy Recovery with EATR ≠ 0 and OACF ≠ 1 (Example 7)

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

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

b. Exhaust air conditions

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

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

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

6. Check total performance.

q_t = (0.072 lb/ft³)(60)(850 ft³/min)(36.6 – 25.1) = 42,230 Btu/h saved

q_t = (0.075 lb/ft³)(60)(850 ft³/min)[(0.24 Btu/lb·°F)(89 – 75) + (0.0164 – 0.0096)(1100 Btu/lb)] = 41,460 Btu/h, which is close to 42,230 Btu/h

7. Plot conditions on psychrometric chart (Figure 22).

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

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

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

The net ventilation rate is 850 cfm and the EATR = 0.05; therefore, the actual flow rate Q_2 to the space can be obtained from Equation (22) as

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

Because OACF = 1.05, the actual flow rate Q_1 of fresh air from outside can be calculated from Equation (23) as

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

To balance the flow rates into the ERV, the actual air flow rates at states 3 and 4 are as shown in Figure 23.

Supply and exhaust fan capacity should match the flows required at their locations. Example 7's results are for balanced flow. Assuming flow rates in the ERV are same as the outside air ventilation requirements, then the effectiveness would be same as that for no air leakage. If air leaks at the inlet and outlet of the energy recovery ventilator, then the exit conditions of air temperature and humidity at states 3 and 4 can be calculated as those of the airstream mixture. For instance, the properties at state 2 would be those of an airstream

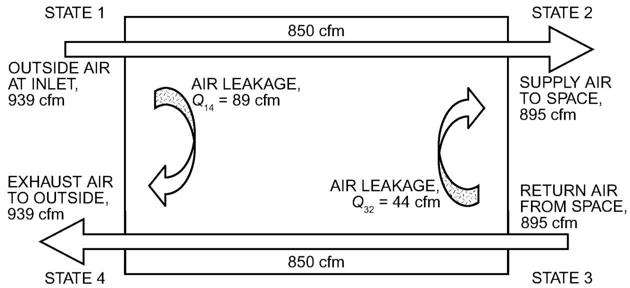

Fig. 23 Actual Airflow Rates at Various State Points (Example 7)

mixture at state 2 for no air leakage and air quantity (Q_{32}) at state 3. The error for these calculations should be less than 5%.

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

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. This process follows a constant wet-bulb line on a psychrometric chart. Thus, the airstream enthalpy remains nearly constant, moisture content increases, and dry-bulb temperature decreases. 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.

Indirect evaporative cooling has been used with heat pipe heat exchangers, two-phase thermosiphon loops, 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, a 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 COP tends to be high, typically from 9 to 20, depending on available dry-bulb temperature depression. The dry-bulb temperature decrease in the exhaust airstream caused by evaporative cooling tends to be 85 to 95% of the maximum available difference between the exhaust air inlet dry-bulb and wet-bulb temperatures. Therefore, exhaust air evaporative cooling is usually most cost-effective in hot, dry climates where the evaporator can be used frequently to obtain large exhaust air dry-bulb temperature depressions.

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

Because less mechanical cooling is required with evaporative cooling, energy consumption and peak demand load are both reduced, yielding lower energy bills. Overall mechanical refrigeration system requirements are reduced, allowing use of smaller mechanical refrigeration systems. In some cases, the mechanical system may be eliminated. Chapter 19 of this volume and Chapter 51 of the 2007 *ASHRAE Handbook—HVAC Applications* have further information on evaporative cooling.

Example 8. Indirect Evaporative Cooling Recovery

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

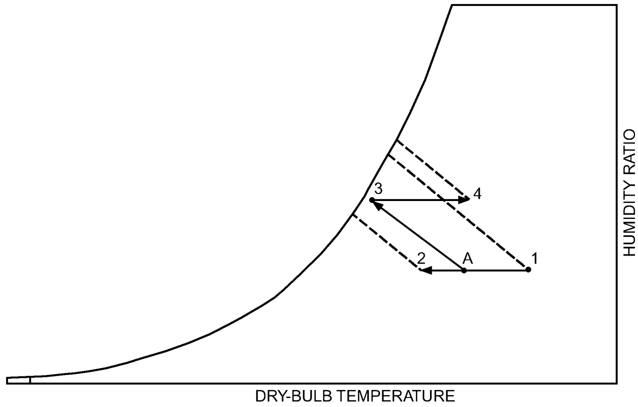

Fig. 24 Indirect Evaporative Cooling Recovery (Example 8)

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 24. In this problem the volumetric flows are equal, but the mass flows are not.

1. Because the 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).

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

3. Establish the sensible effectiveness.

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

4. Calculate actual energy transfer at the design conditions.

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

5. Calculate leaving air conditions.

a. Leaving supply air temperature is

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

b. Leaving exhaust air temperature is

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

6. Check performance.

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

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

7. Plot conditions on psychrometric chart (Figure 24), and confirm that no latent exchange occurred.

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

Precooling Air Reheater (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 outside 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, illustrated in Figure 25, outside 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. After leaving the cooling coil, it passes through the other side of the air-to-air heat exchanger, where the incoming supply air reheats it.

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 2.

Example 9. Precooling Air Reheater Dehumidifier

In this application, 3400 cfm of outdoor supply air at 95°F and 81°F wb (ρ = 0.069 lb/ft^3) is used to reheat 3400 cfm of the same air leaving a cooling coil (exhaust) at 52.1°F 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 the 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 pre-cooler 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.

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

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 outside 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 (27) 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.

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

The wet-bulb temperature for saturated air with this enthalpy is 75°F. This is point 2 on the psychrometric chart (Figure 26), which is near saturation. Note that this precooled air is further dehumidified by a cooling coil from point 2 to point 3 in Figure 26.

b. Reheater leaving air conditions

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

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.

$$q_s = (60)(0.069\ \text{lb/ft}^3)(3400\ \text{ft}^3/\text{min})(44.6 - 38.6)$$
$$= 84{,}500\ \text{Btu/h precooling}$$

$$q_s = (60)(0.069\ \text{lb/ft}^3)(3400\ \text{ft}^3/\text{min})(0.24\ \text{Btu/lb·°F})(77.2 - 52.1)$$
$$= 84{,}800\ \text{Btu/h reheat}$$

7. Plot conditions on psychrometric chart (Figure 26).

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

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 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, the cost of forming these materials into a highly effective energy/heat exchanger, the cost of auxiliary equipment and controls, and the cost of installation.

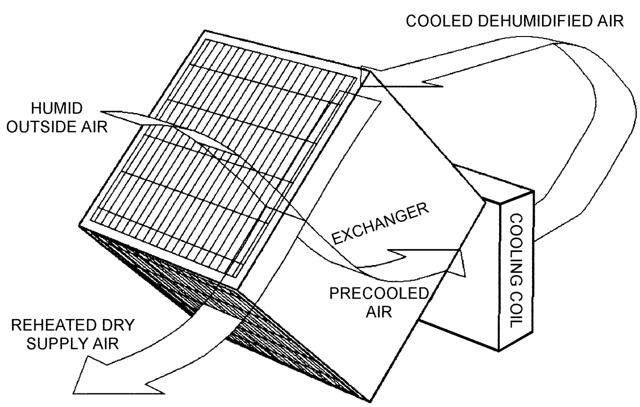

Fig. 25 Precooling Air Reheater

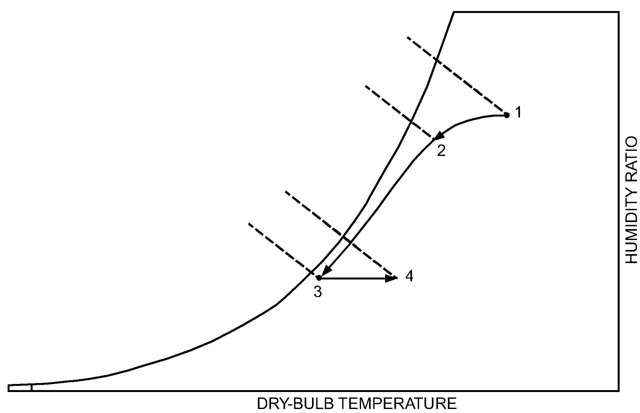

Fig. 26 Precooling Air Reheater Dehumidifier (Example 9)

Owning and operating costs are discussed in more detail in Chapter 36 of the 2007 *ASHRAE Handbook—HVAC Applications*.

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 outside 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 2 for typical values); however, a high ε implies a high capital cost, even when the exchanger is designed to minimize the amount of materials used. Energy costs for fans and pumps are usually very important and accumulate operating cost even when the energy recovery system must be throttled back. For building ventilation, throttling may be required much of the time. Thus, the overall LCC minimization problem for optimal design may involve 10 or more independent design variables as well as several specified constraints and operating conditions [see, e.g., Besant and Johnson (1995)].

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

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

$$\begin{aligned} PP &= \frac{\text{Capital cost and interest}}{\text{Annual operating energy cost saved}} \\ &= \frac{C_{s,init} - \text{ITC}}{C_e(1 - T_{inc})}\text{CRF}(i'', n) \end{aligned} \tag{28}$$

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 outside 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.

SYMBOLS

A = area of recovery exchanger, ft^2
c_p = specific heat of moist air, Btu/lb·°F
C = capital cost
C_e = cost of energy
C_r = ratio of C_{min}/C_{max}
$C_{s,int}$ = initial system capital cost
CRF = capital recovery factor
h = enthalpy, Btu/lb

h_{fg} = enthalpy of vaporization, Btu/lb
i = arbitrary state, or discount rate
ITC = income tax credit
m_s = mass flow rate of supply moist air from outside, lb/min
m_e = mass flow rate of exhaust moist air, 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_{w3} = wet-bulb temperature of moist air at state 3, °F
T = absolute temperature, °F, or tax
U = overall heat transfer coefficient, Btu/h·ft^2·°F
V = mean velocity, fpm
w = humidity ratio

Greek Letters

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

Subscripts

a = air
e = exhaust side of heat/energy exchanger, exit or energy
f = fan or fan motor combination
if = threshold temperature of the outside air for freezing to occur
in = indoor conditions of building space
inc = increment
h = 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

ARI. 2001. Rating air-to-air energy recovery ventilation equipment. ANSI/ARI *Standard* 1060-2001. Air-Conditioning and Refrigeration Institute, Arlington, VA.

ASHRAE. 2001. Ventilation for acceptable indoor air quality. ANSI/ASHRAE *Standard* 62.

ASHRAE. 1991. Method of testing air-to-air heat exchangers. ANSI/ASHRAE *Standard* 84-1991.

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

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.

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

Guo, P., D.L. Ciepliski, 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* 4(1):3-26.

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. Simonson, and R.W. Besant. 1998. Uncertainty analysis in the testing of air-to-air heat/energy exchangers installed in buildings. *ASHRAE Transactions* 104(1B):1639-1650.

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

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. 1997. 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. 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. 2003. (Personal communication and reference, *Trane Application Engineering Manual* SYS-APM003-EN.)

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: Thermal-hydraulic fundamentals and design*. S. Kakec, A.E. Bergles, and F. Maysinger, eds. Hemisphere Publishing, New York.

Shang, W., M. Wawryk, 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 regerative 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.

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.

BIBLIOGRAPHY

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):307-332.

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

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. Simonson. 2003. Air-to-air exchangers. *ASHRAE Journal* 45(4):42-50.

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

Ciepliski, D.L., C.J. Simonson, and R.W. Besant. 1998. Some recommendations for improvements to ASHRAE *Standard* 84-1991. *ASHRAE Transactions* 104(1B):1651-1665.

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

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

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

Phillips, E.G., R.E. Chant, B.C. Bradley, and D.R. Fisher. 1989. 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. 1989. Comparison of freezing control strategies for residential air-to-air 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.

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).

Simonson, 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* 3(4):325-368.

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

Simonson, 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.

Simonson, C.J., D.L. Cieplisky, 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., Mathur, G.D, 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.

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

Related Commercial Resources