CHAPTER 44

AIR-TO-AIR ENERGY RECOVERY

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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.

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
Comfort-to-comfort	Swimming pools Locker rooms Residential Smoking exhaust Operating rooms Nursing homes Animal ventilation Plant ventilation General exhaust

Process-to-Process

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 (humidity), as 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 in order to prevent condensation and possible corrosion.

Process-to-Comfort

In process-to-comfort applications, waste heat captured from a process exhaust heats the 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 desired in process-to-process applications, recovery for process-to-comfort applications must be modulated during warm weather to prevent overheating of 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 over a year than do process-to-process applications.

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

Comfort-to-Comfort

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.

In addition to commercial and industrial energy recovery equipment, small-scale packaged ventilators with built-in heat recovery components known as heat recovery ventilators (HRVs) or energy recovery ventilators (ERVs) are available for residential and small-scale commercial applications.

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

ECONOMIC CONSIDERATIONS

Air-to-air energy recovery systems are used in new or 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 on an annual basis and with an acceptable payback period.

The annualized system owning, operating, and maintenance costs are discussed in the section on Uniform Annualized Costs Method in Chapter 35 of the 1999 *ASHRAE Handbook—Applications*. Although the capital cost and interest term in this method implies a simple value, it is in fact a complex function of the future value of money as well as all the design variables in the energy/heat exchanger. These variables include the mass of each material used, the cost of forming these materials into an energy/heat exchanger with a high effectiveness, the cost of auxiliary equipment and controls, and the cost of installation.

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

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must be imposed. Thus, unlike other HVAC designs, heat recovery systems should have a design temperature not at the ambient winter design temperature but rather at the temperature for maximum heat recovery rate.

The exchanger overall effectiveness ε should be high (see Table 2 for typical values); however, a high value of 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 a large fraction of the time. Thus, the overall LCC minimization problem for optimal design may involve 10 or more independent design variables as well as a number of specified constraints and operating conditions [see, for example, Besant and Johnson (1995)].

In addition, comfort-to-comfort energy recovery systems often operate with much smaller temperature differences than do most auxiliary air-heating and -cooling heat exchangers. These small temperature differences imply the need for more accurate energy transfer models if the maximum cost benefit or lowest LCC is to be realized.

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

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

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

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 the last dollar earned (i.e., the 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; values less than 3 years are often realized.

Other economic factors include the following.

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

Life-Cycle Cost. Air-to-air energy recovery cost benefits are best evaluated considering all capital, installation, operating, and energy-saving costs over the duration of the equipment life under its normal operating conditions in terms of a single cost relationship—the life-cycle cost. As a rule, neither the most efficient nor the least expensive energy recovery device will be most economical. The optimization of the life-cycle cost for maximum net savings may involve a large number of design variables, necessitating careful cost estimates and the use of an accurate model of the recovery system with all its design variables.

Energy Costs. The absolute cost of energy and the 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.

Other Conservation Options. Energy recovery should be evaluated against other cost-saving opportunities, including reducing or eliminating the primary source of waste energy through process modification.

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. Although using equipment with higher effectiveness results in more recovered energy, equipment cost and space requirements also increase with effectiveness.

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 the supply is coincident with the 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 washdown 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.

TECHNICAL CONSIDERATIONS

Ideal Air-to-Air Energy Exchange

An ideal air-to-air energy exchanger performs the following functions:

- Allows temperature-driven heat transfer between the participating airstreams
- Allows partial-pressure-driven moisture transfer between the two streams
- Totally blocks cross-stream transfer of air, other gases (in particular, pollutants), biological contaminants, and particulates

Heat transfer is widely recognized as an important vehicle for energy recovery 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 environment; in view of the uncomfortable climate, the indoor air is conditioned. Many local ordinances require a specified number of outdoor air changes per hour. If the energy exchanger is a heat exchanger but not a moisture exchanger, it facilitates the cooling of outdoor ventilation air as it passes through the exchanger en route to the indoor space. Heat flows from the incoming outdoor air to the outgoing (and cooler) exhaust air drawn from the indoor conditioned space. This heat transfer process does very little to mitigate the high humidity that is carried into the indoor space by the outdoor ventilation air. A substantial amount of power will be required to dehumidify that air to reduce its moisture content to a level acceptable for comfort.

On the other hand, if the energy exchanger can transfer both heat and moisture, the highly humid outdoor air will transfer moisture to the less humid indoor air as the two streams pass through the exchanger. The lowered humidity of the entering ventilation air will allow a substantial savings of energy.

Forms of Energy Transfer in an Air-to-Air Energy Exchanger

Two generic types of air-to-air energy exchangers may be considered for service as heat and moisture exchangers. One of these is the **rotating energy wheel regenerator**, frequently called a **heat wheel** or an **energy wheel**. The other is the **permeable-walled flat-plate recuperator**. Although the end results are similar, the transport mechanisms in these two categories of devices are very different.

In a rotating energy wheel regenerator, a desiccant film coating the surfaces of the wheel absorbs moisture as the wheel passes through the more humid airstream. Once absorbed, the moisture rides the moving wheel until it reaches the less humid airstream, where it is desorbed from the film into the airstream.

In a moisture-transferring fixed-wall recuperator, the walls are made of a material that is permeable to water vapor. The moisture passes through the walls when there is a difference in the magnitude of the vapor pressures between the two airstreams.

Rotating energy wheel regenerators and permeable-walled flat-plate recuperators are prone to cross-stream leakage of air and other gases. Advances in microporous film technology have demonstrated that membranes can be synthesized which will nearly totally block the transfer of air and other gases, while providing relatively free passage for the water vapor. In light of this, the long-range commercial potential for the permeable-walled recuperator appears favorable.

Rate of Energy Transfer

The rate of energy transfer depends on the intrinsic characteristics of the energy exchanger and on the operating conditions. Intrinsic properties include the geometry of the exchanger (parallel flow/counterflow/cross-flow, number of passes, fins), the thermal conductivity of the walls separating the streams, and the permeability of the walls to the passage of various gases. As in a conventional heat exchanger, heat 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 heat 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 encompassing release of latent energy followed by its subsequent transfer in the form of sensible heat is commonly called **latent heat transfer**.

Latent energy transfer between the airstreams occurs only when moisture is transferred from one stream to another without condensation, thereby maintaining the latent heat of condensation intact. Once the 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 energy wheel regenerators and permeable-walled flat-plate recuperators are used because of their moisture recovery function. The passage of air or other gases (e.g., pollutants) across the exchanger is a negative consequence. Cross-stream mass transfer may occur through leakage even when such transfer is unintended. The undetected presence of unintended leakage may alter the performance of the exchanger from its design value.

Conduction heat transfer differs in principle from the mass transfer. Certain phenomena relating to heat/mass exchanger performance must be recognized:

- The effectiveness for moisture transfer is likely not equal to the effectiveness for heat transfer.
- The total energy effectiveness is likely not equal to either the sensible effectiveness or the latent effectiveness.
- The total energy transfer may not equal the sum of the sensible and latent heat transfers.

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. The following example illustrates the potential for misinterpretation of total energy transfer:

Example 1. Assume that the inlet supply air has a high temperature with low humidity ratio, the inlet exhaust air has a low temperature with high humidity ratio, and the supply and exhaust mass airflows are equal. Assume also that the energy exchanger has been tested to ASHRAE *Standard* 84, which rated the sensible heat transfer effectiveness at 50% and the latent (water-vapor) transfer effectiveness at 50%. Plotted on a psychrometric chart, the water vapor transfer is equal to the heat transfer but in the opposite direction. Mass airflow and enthalpy transfer balances are equal.

Simple examination of the plots shows that total energy transfer is zero because the change in enthalpy due to heat transfer is equal in magnitude but opposite in direction to that due to water vapor transfer. The net total energy effectiveness is shown to be zero. The direction and rate of heat and moisture energy transfer must be considered as separate, independent entities.

Cross-stream mass transfer 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 total air pressure differences and is minimized by the bulk-flow resistance characteristics of the exchanger wall barrier. Moisture mass transfer, on the other hand, is driven by a combination of total pressure differences and vapor partial pressure differences. Cross-stream moisture transfer is maximized by the bulk-flow resistance and permeability of the exchanger wall barrier. High bulk-flow resistance retards viscous, bulk airflow and minimizes the effect of total pressure differences on air mass transfer. Total-pressure-driven mass transfer and partial-pressure-driven mass transfer may be mutually aiding or mutually opposing.

Heat, moisture, and air transfer rates are partially independent of and separate from one another. Heat is always driven cross-stream in a direction from higher to lower temperature. Air is predominantly driven cross-stream in a direction from higher to lower total 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 total and vapor pressure differences cross-stream will influence the net intensity and direction of moisture exchange.

Standard laboratory rating tests and predictive computer models give exchanger performance values individually 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 transfer and mass water vapor transfer must be separately determined by running rating tests for a given exchanger in a laboratory that is staffed and instrumented to accurately measure all performance parameters required in ASHRAE *Standard* 84 and ARI *Standard* 1060. It may be very difficult to adhere to any standard when field tests are made.

Performance Rating

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 the performance; and (3) specifies the types of test equipment for performing such tests. *Standard* 84 test methods minimize air leakage by specifying that all tests be conducted at zero pressure differential between supply outlet and exhaust inlet. Fan placement or effect on performance is not included in the standard. *Standard* 84 does not specify performance criteria for product certification or identify laboratories capable of performing these tests.

ARI *Standard* 1060, Rating Air-to-Air Energy Recovery Ventilation Equipment, is an industry-established standard for rating the performance of air-to-air heat/energy exchangers for use in energy recovery ventilation equipment. It establishes definitions, requirements for marking and nameplate data, and conformance conditions intended for the industry, including manufacturers, engineers, installers, contractors, and users. Air-to-air heat/energy exchangers for use in energy recovery ventilation equipment must be tested in accordance with ASHRAE *Standard* 84, except where modified by ARI *Standard* 1060. Standard temperature and humidity conditions at which equipment tests are to be conducted are specified. Published ratings must be reported for mass flow rate; pressure drop; and sensible, latent, and total effectiveness at the standard conditions specified. ARI *Certification Program* 1060 was established to verify ratings published by manufacturers.

In the field, air-to-air heat/energy exchanger performance may depart significantly from that measured under the idealized conditions in a test lab. System design and configuration will influence the amount and direction of (1) air leakage between airstreams or between the exchanger and its surroundings; (2) water vapor transfer between airstreams as a result of carryover, crossover, or leakage; (3) pressurization caused by fan placement; and (4) heat and vapor exchange to the surroundings.

Balanced mass airflows as required in the ASHRAE and ARI standard test methods are rarely achieved in field operation in air-handling systems. Fans are 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. In packaged air handlers, it is impractical to expect unit fans to operate year-round at zero air leakage, zero pressure differential, and balanced mass airflows, as required by these standard test methods. The application designer should evaluate the probable actual (versus ideal laboratory standard) performance of the exchanger relative to placement of exhaust and supply fans in the equipment for all expected weather conditions.

The effectiveness of air-to-air heat exchangers is commonly measured in terms of

- Sensible energy (heat) transfer: dry-bulb temperature
- Latent energy (water vapor or moisture) transfer: humidity ratio
- Total energy (heat and moisture) transfer: enthalpy

ASHRAE *Standard* 84 defines effectiveness as

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

Referring to Figure 1,

$$\varepsilon = \frac{w_s(x_2 - x_1)}{w_{min}(x_3 - x_1)} = \frac{w_e(x_3 - x_4)}{w_{min}(x_3 - x_1)} \tag{2}$$

where

ε = moisture (or water vapor mass), sensible, or total effectiveness
x = humidity ratio *W*, dry-bulb temperature *t*, or enthalpy *h* at locations indicated in Figure 1
w_s = supply air mass flow
w_e = exhaust air mass flow
w_{min} = the smaller of w_s and w_e

The leaving supply air condition is

$$x_2 = x_1 + \varepsilon\left(\frac{w_{min}}{w_s}\right)(x_1 - x_3) \tag{3}$$

and the leaving exhaust air condition is:

$$x_4 = x_3 - \varepsilon\left(\frac{w_{min}}{w_e}\right)(x_1 - x_3) \tag{4}$$

Equations (2), (3), and (4) assume steady-state conditions; no heat transfer between the heat exchanger and its surroundings; and no gains from cross-leakage, fans, or frost control devices. Furthermore, condensation or frosting does not occur or is negligible. Those assumptions are generally true for larger commercial applications but not for heat recovery ventilators (HRVs). CAN/CSA-C439, Standard Methods of Test for Rating the Performance of Heat-Recovery Ventilators, is used to rate small (under 400 cfm) packaged ventilators with heat recovery. The rating terms used in CAN/CSA-C439 for HRVs are **energy recovery efficiency** (i.e., the actual energy transfer efficiency) and **apparent sensible effectiveness** (i.e., a measure of the temperature rise of the supply airstream, including that resulting from conversion of fan motor energy to heat, air leakage, and heat transfer across cases).

A number of variables can affect these performance factors, whether the device is designed to transfer total energy or just sensible heat. These variables include (1) water vapor partial pressure differences, (2) heat transfer area, (3) air velocity through the heat exchangers, (4) airflow arrangement or geometric configuration, (5) supply and exhaust air mass flow rates, and (6) method of frost

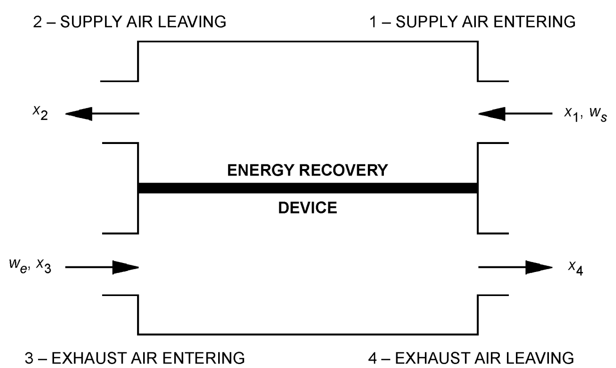

Fig. 1 Airstream Numbering Convention

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).

Sensible Versus Total Recovery

Sensible heat recovery devices do not transfer moisture. As water vapor changes state from vapor to liquid with no change in temperature, heat that is released into the same airstream is called the **latent heat of condensation**. Thus, air temperature is maintained in the airstream as a result of condensation. Because the airstream temperature is higher than it would have been without condensation, heat transfer is increased. Thus, latent heat is transferred only when the warmer airstream is cooled below its dew point and condensation occurs. In this case, latent heat is transferred, but moisture is not. Sensible heat exchangers should be used where latent heat transfer without moisture transfer is desired. Examples include indirect evaporative coolers, precool reheaters, and air dryers.

Total energy recovery devices (e.g., desiccant-coated rotary and permeable membrane energy exchangers) transfer both moisture and heat between the airstreams. The sensible (i.e., temperature) and latent (i.e., moisture) transfer effectivenesses for a given exchanger are different. Total heat exchange is desirable (1) in hot, humid climates where moisture transfer from supply air to the exhaust airstream reduces air-conditioning loads and (2) in cold, dry climates where moisture transfer from exhaust air to the supply airstream reduces humidification requirements. Barringer and McGugan (1989b) address sensible versus latent heat recovery for residential comfort-to-comfort applications.

Fouling

Fouling refers to an accumulation of dust or condensates on heat exchanger surfaces. By increasing the resistance to airflow and generally decreasing heat transfer coefficients, fouling reduces heat exchanger performance. The increased resistance increases fan power requirements and may reduce airflow.

Pressure drop across the heat exchanger core can be used as an indication of fouling and, with experience, may be used to establish cleaning schedules. Heat exchanger surfaces must be kept clean if system performance is to be maximized.

Corrosion

Process exhaust frequently contains substances requiring corrosion-resistant construction materials. If it is not known which materials are most corrosion-resistant for an application, the user and/or designer should examine on-site ductwork, review literature, and contact equipment suppliers prior to selecting materials. A corrosion study of heat exchanger construction materials in the proposed operating environment may be warranted if the 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 due to perforation or mechanical failure.

Cross-Leakage

Cross-leakage, cross-contamination, or mixing between supply and exhaust airstreams may occur in air-to-air heat exchangers. It 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).

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 is cooled from its inlet condition to its outlet condition. First there is a dry region with no condensate. Once the warm airstream is cooled below its dew point, there is a condensing region, which results in wetting of the heat exchange surfaces. If the heat exchange surfaces fall below freezing, the condensation will freeze. Finally, if the warm airstream temperature is reduced below 32°F, sublimation causes frost to form. The locations of these regions and rates of condensation and frosting depend on the duration of frosting conditions; the airflow rates; the inlet air temperature and humidity; the heat exchanger core temperature; the heat exchanger effectiveness; the geometry, configuration, and orientation; and the heat transfer coefficients.

Sensible heat exchangers, which are ideally suited to applications in which heat transfer is desired but humidity transfer is not (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 increase pressure drops significantly in heat exchangers with narrow airflow passage spacings. 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.

For frosting or icing to occur, an airstream must be cooled below 32°F and below its dew point. Total heat exchangers transfer moisture from the airstream with the higher moisture content (usually the warmer airstream) to the airstream with the lower moisture content. 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, permanently damaging the heat exchanger.

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; reducing the 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 543-RP (Phillips et al. 1989a, 1989b). ASHRAE 544-RP (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. (1989c) 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.

Pressure Drop

The pressure drop for each airstream through a heat exchanger depends on many factors, including exchanger design, mass flow rate, temperature, moisture, and inlet and outlet air connections. The exchanger pressure drop must be overcome by fans or blowers.

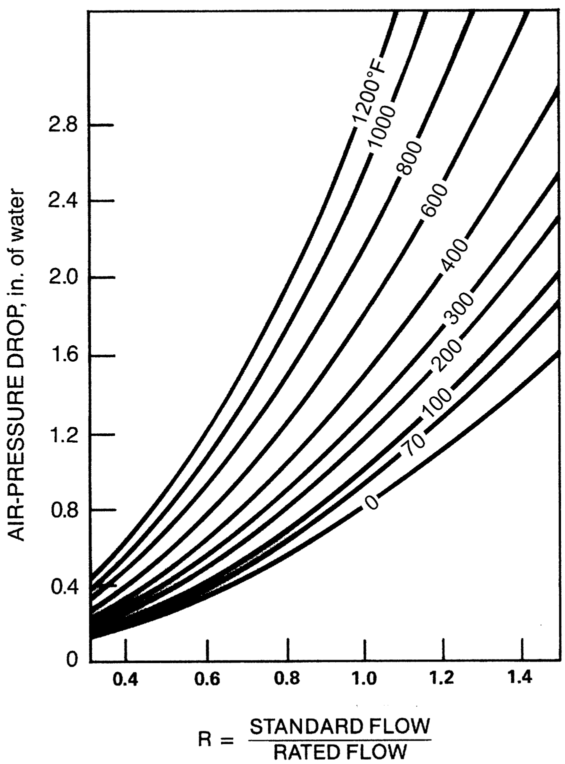

Fig. 2 Pressure Drop Versus Flow at Various Temperatures for Typical Plate Exchanger

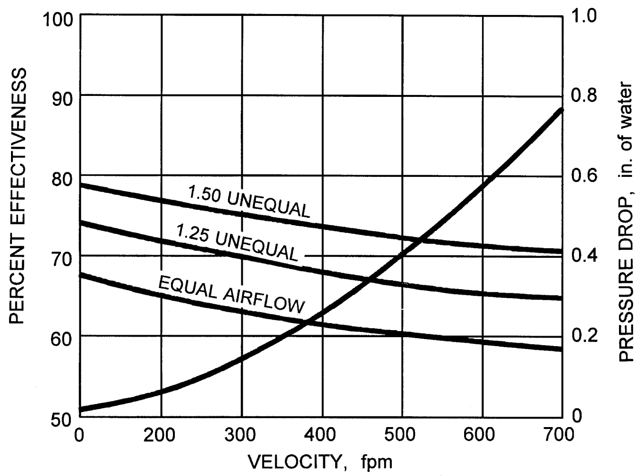

Fig. 3 Typical Flat-Plate Performance

When all other parameters are constant for a given exchanger, pressure drop increases

- With gas temperature (Figure 2)
- As plate contamination or fouling increases (e.g., due to condensation, frosting, or dust accumulation)
- If pressure differentials between airstreams deform airflow passages (plate-type exchangers)
- As barometric pressure increases
- With airflow velocity (Figure 3)

Face Velocity

Design face velocities for heat exchangers are based primarily on allowable pressure drop rather than on recovery performance. Recovery performance (effectiveness) decreases with increasing velocity, but the decrease in effectiveness is not as rapid as the increase in pressure drop (Figure 3). Low face velocities give lower pressure drop, higher effectiveness, and lower operating costs, but require larger units with higher capital costs and greater installation space.

Airflow Arrangements

Heat exchanger effectiveness depends to a great extent on the airflow direction and pattern of the supply and exhaust airstreams. Parallel-flow exchangers (Figure 4A), in which both airstreams move along heat exchange surfaces in the same direction, have a theoretical maximum effectiveness of 50%. Counterflow exchangers (Figure 4B), in which airstreams move along heat exchange surfaces in opposite directions, can have effectivenesses approaching 100%, but units designed for typical applications have a much lower effectiveness.

In practice, construction limitations favor designs in which the two airstreams interface in transverse (or cross-flow) configuration over much of the heat exchange surface (Figure 4C and Figure 4D).

The normal range of effectiveness for cross-flow heat exchangers is 50 to 70% (Figure 4C) and 60 to 85% (Figure 4D).

Maintenance

The method used to clean a heat exchanger depends on the transfer medium or mechanism used in the heat exchanger and on the nature of the material to be removed. Grease buildup 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 at the design stage 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 the air-conditioning equipment. Snow or frost can block the air supply filter and cause severe problems. Supply air filters should be removed in very cold weather to avoid frosting blockage of the inlet. Steps to ensure a continuous flow of supply air should be incorporated into the design.

Controls

Heat exchanger controls may function to control frost formation or to 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 prevent overheating ventilation supply air during cool to moderate weather or to prevent overhumidification. Modulation may be achieved by tilting heat pipes, changing rotational speeds of (or stopping) heat wheels, or bypassing part of one airstream around the

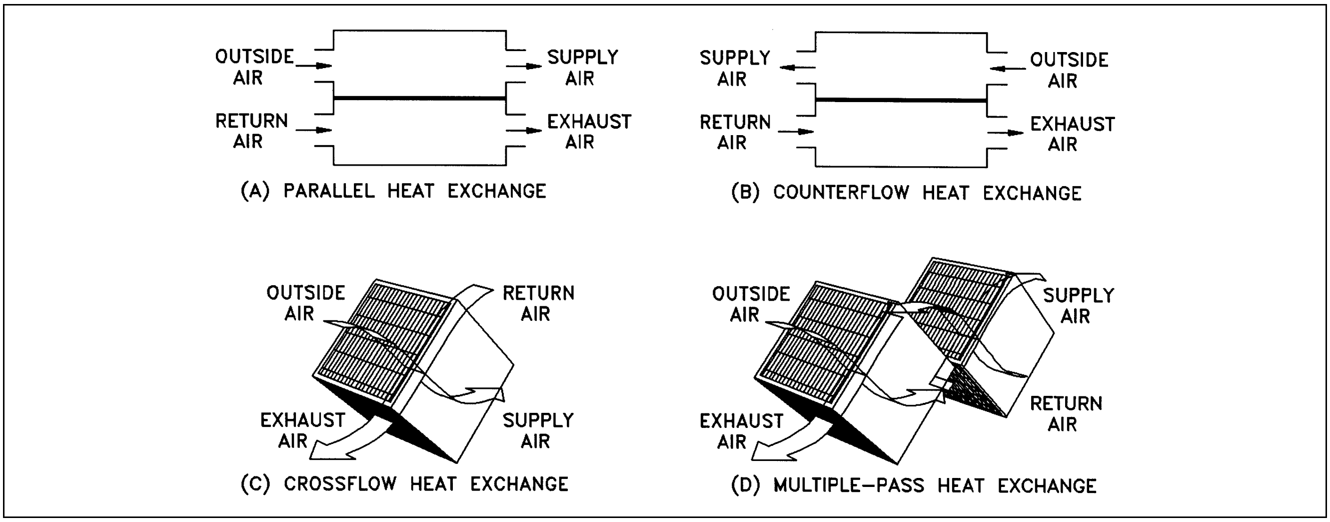

Fig. 4 Heat Exchanger Airflow Configurations

heat exchanger using face and bypass dampers (i.e., changing the supply-to-exhaust mass airflow ratio).

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 enthalpy of the airstream remains nearly constant, moisture content increases, and dry-bulb temperature decreases. The evaporatively cooled exhaust air can then be used to cool the supply air through an air-to-air heat exchanger. The heat exchanger may be applied either for year-round energy recovery or exclusively for its evaporative cooling benefits.

Indirect evaporative cooling has been applied with heat pipe heat exchangers, two-phase thermosiphon loops, and flat-plate heat exchangers for summer cooling (Scofield and Taylor 1986; Mathur 1990a, 1990b, 1992, 1993). 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 hence more heat transfer. Energy recovery is further enhanced by improved heat transfer coefficients due to 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 energy efficiency ratio (EER) of indirect evaporative cooling systems tends to be high, typically ranging from 30 to 70, depending on available wet-bulb depression.

Because less 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, so that smaller mechanical refrigeration systems can be selected. In some cases, the mechanical system may be eliminated. Chapter 19 of this volume and Chapter 50 of the 1999 *ASHRAE Handbook—Applications* have further information on evaporative cooling.

Precooling Air Reheater

In some applications, such as ventilation in hot, humid climates, supply air is cooled below the desired delivery temperature to condense moisture and reduce the humidity. Using this overcooled supply air to precool outdoor air reduces the air-conditioning load, which allows refrigeration equipment to be downsized and eliminates the need to reheat the supply air with purchased energy.

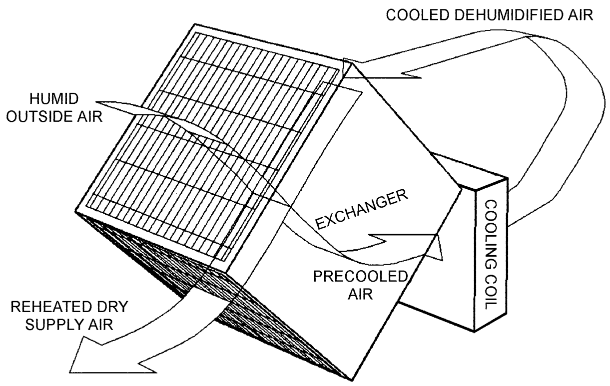

Fig. 5 Precooling Air Reheater

In this three-step process, illustrated in Figure 5, outdoor air passes through an air-to-air heat exchanger, where it is precooled by supply air leaving the cooling coil. It is then further cooled and dehumidified in the cooling coil. After leaving the cooling coil, it passes through the other side of the air-to-air heat exchanger, where it is reheated by the incoming supply air.

Fixed-plate, heat pipe, and rotary heat exchangers can be used to reheat precooled supply air. At part-load operating conditions, the amount of heat transferred from precooling to reheating may require modulation. This can be done using the heat rate control schemes noted in Table 2.

ENERGY RECOVERY CALCULATIONS

The rate of energy transfer to or from an airstream depends on the rate and direction of the heat transfer and on the rate and direction of the 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 are dependent on exchanger construction characteristics. Equation (5) is used to determine the rate of energy transfer when sensible (temperature) and latent (moisture) energy transfer occurs, while Equation (6) is used for sensible-only energy transfer.

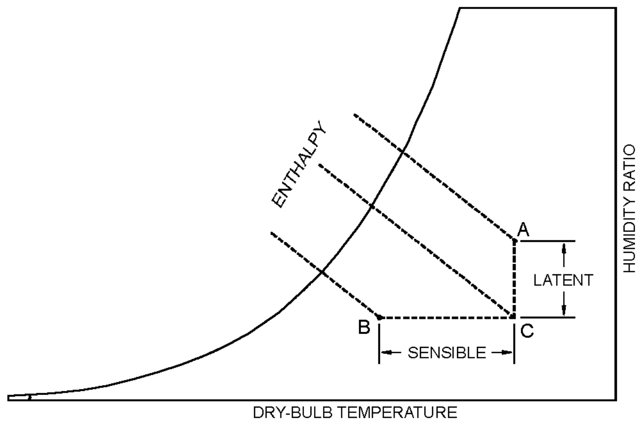

Fig. 6 Maximum Sensible and Latent Heat from Process A-B

$$q_{total} = 60Q\rho(h_{in} - h_{out}) \tag{5}$$

$$q_{sensible} = 60Q\rho c_p(t_{in} - t_{out}) \tag{6}$$

where

q_{total} = $q_{sensible} + q_{latent}$ = total energy transfer, Btu/h
$q_{sensible}$ = 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 energy recovered in air-to-air energy recovery applications.

Step 1. Calculate theoretical maximum moisture and energy transfer rates $w_{m,max}$ and q_{max}.

The airstream with the lower mass flow w_{min} limits heat and moisture transfer. Some designers specify and prefer working in scfm. In order 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 as scfm. If necessary, convert flow rates to mass flow rates (e.g., scfm or lb/min) to determine which airstream has the minimum mass.

If only sensible energy transfer occurs, the theoretical maximum rate of heat transfer q_{max}, using the airstream numbering convention from Figure 1, is $60\rho c_p Q_{min}(t_3 - t_1)$. If latent energy transfer occurs, the theoretical maximum energy transfer q_{max} is $60\rho Q_{min}(h_3 - h_1)$. The maximum moisture transfer rate $w_{m,max}$ is also implied by Equation (2) and is $w_{min}(W_3 - W_1)$, where W_3 and W_1 are the humidity ratios at state 3 and state 1.

The split between latent and sensible energy (enthalpy) potential flux can be determined by plotting the airstream conditions on a psychrometric chart as shown in Figure 6. 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 2. Establish the moisture, sensible, and total effectivenesses ε_m, ε_s, and ε_t.

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

Each effectiveness should be verified by the manufacturer for the air inlet conditions. If the exchanger selected does not perform at the specified effectiveness, its impact on the project should be considered. The manufacturer should answer the following questions as well:

- Does the published sensible effectiveness result from tests with condensation in the exhaust airstream?
- Are the published effectivenesses for sensible and total energy transfer different or are they assumed to be equal?
- Are published airflow rates based on standard or actual temperature and barometric pressure at the fan?
- Has the exchanger performance been verified by an independent laboratory to meet ASHRAE *Standard* 84 criteria at the specified airflows and inlet conditions?

The pressure drop for each airstream should be determined from the manufacturer's data for the design conditions to calculate fan requirements.

Step 3. Calculate actual moisture and energy (sensible and total) transfer.

The amount of energy transferred is the product of the effectiveness ε for the airstream with the lesser mass flow rate and the theoretical maximum heat transfer determined in Step 1 using Equation (2):

$$w_m = \varepsilon_m w_{m,\,max} \tag{7a}$$

$$q_{actual} = \varepsilon q_{max} \tag{7b}$$

where ε and q may be for sensible or total energy transfer.

Step 4. Calculate leaving air conditions for each airstream.

If an enthalpy or moisture-permeable heat exchanger is used, moisture (and its inherent latent energy) is transferred between airstreams. If a sensible-only heat exchanger is used, and 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 assumption of no flows other than at states 1, 2, 3, and 4 in Equation (2) is not valid. In spite of this, the same definitions for 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.) Equation (5) must be used to calculate the leaving air condition for airstreams in which inherent latent energy transfer occurs. Equation (6) may be used for an airstream if only sensible energy transfer is involved.

Step 5. Check the energy transfer balance between airstreams.

Total energy transferred from one airstream should equal total heat transferred to the other airstream. Calculate and compare the energy transferred to or from each airstream. Differences between these energy flows are usually due to measurement errors.

Step 6. Plot entering and leaving conditions on psychrometric chart.

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

EXAMPLES

Example 2. Sensible Heat Recovery in Winter

In this example, 10,000 scfm of exhaust air at 75°F and 10% rh preheats an equal mass of outdoor air at 0°F and 60% rh (ρ = 0.087 lb/ft³) using an air-to-air heat exchanger with moderate effectiveness (60%). Airflows are specified in scfm, so an air density of 0.075 lb/ft³ for both airstreams is appropriate. Determine the leaving supply air conditions, calculate energy recovered, and check heat exchange balance.

1. Calculate the theoretical maximum heat transfer.

   An examination of the two inlet conditions plotted on a psychrometric chart (Figure 7) indicates that, because the exhaust air has low relative humidity, latent energy transfer will not occur. Using Equation (6),

$$q_{max} = 60 \times 0.075 \times 0.24 \times 10{,}000(75 - 0) = 810{,}000 \text{ Btu/h}$$

2. Establish the sensible effectiveness.

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

3. Calculate actual heat transfer at given conditions.

$$q_{actual} = 0.6 \times 810{,}000 = 486{,}000 \text{ Btu/h recovered}$$

4. Calculate leaving air conditions.

   Because no moisture or latent energy transfer will occur, Equation (6) is used.

   a. Leaving supply air temperature t_2 is

$$t_2 = 0 + \left(\frac{486{,}000}{60 \times 0.24 \times 0.075 \times 10{,}000}\right) = 45°\text{F}$$

   b. Leaving exhaust air temperature t_4 is

$$t_4 = 75 + \left(\frac{-486{,}000}{60 \times 0.24 \times 0.075 \times 10{,}000}\right) = 30°\text{F}$$

5. Check performance.

$$q_s = 60 \times 0.24 \times 0.075 \times 10{,}000(45 - 0) = 486{,}000 \text{ Btu/h saved}$$

$$q_e = 60 \times 0.24 \times 0.075 \times 10{,}000(75 - 30) = 486{,}000 \text{ Btu/h recovered}$$

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

Example 3. Sensible Heat Recovery in Winter with Water Vapor Condensation

In this application, 10,000 cfm of exhaust air at 75°F and 28% rh (ρ = 0.075 lb/ft³) is used to preheat 9000 cfm of outdoor air at 14°F and 50% rh (ρ = 0.084 lb/ft³) using a heat exchanger with an effectiveness of 70%. Determine the leaving supply air conditions, calculate energy recovered, and check energy exchange balance.

The supply airstream has a lower airflow rate than the exhaust airstream, and so it may appear that the supply airstream will limit heat transfer. However, determination of mass flow rates for the given entry conditions shows that the mass flow rate of the supply airstream (756 lb/min) is slightly greater than that of the exhaust airstream (750 lb/min), 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. 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 (6) is used:

$$q_{max} = 60 \times 0.24 \times 0.084 \times 9000(75 - 14) = 664{,}000 \text{ Btu/h}$$

2. Select sensible effectiveness.

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

3. Calculate actual heat transfer at design conditions using Equation (7b):

$$q_{actual} = 0.7 \times 664{,}000 = 465{,}000 \text{ Btu/h recovered}$$

4. Calculate leaving air conditions.

   a. Leaving supply air temperature is calculated using Equation (6).

$$t_2 = 14 + \left(\frac{465{,}000}{60 \times 0.24 \times 0.084 \times 9{,}000}\right) = 56.7°\text{F}$$

   b. Condensation will occur on the exhaust side, so the leaving exhaust air enthalpy is determined using Equation (5). The entering exhaust air enthalpy, determined for the wet-bulb temperature (55.4°F) using a psychrometric chart, is 23.5 Btu/lb.

$$h_4 = 23.5 + \left(\frac{-465{,}000}{60 \times 0.075 \times 10{,}000}\right) = 13.2 \text{ Btu/lb}$$

   The wet-bulb temperature corresponding to an enthalpy of 13.2 Btu/lb is 35°F.

5. Check performance.

$$q_s = 60 \times 0.24 \times 0.084 \times 9000(56.7 - 14) = 465{,}000 \text{ Btu/h saved}$$

$$q_e = 60 \times 0.075 \times 10{,}000(23.5 - 13.2) = 465{,}000 \text{ Btu/h recovered}$$

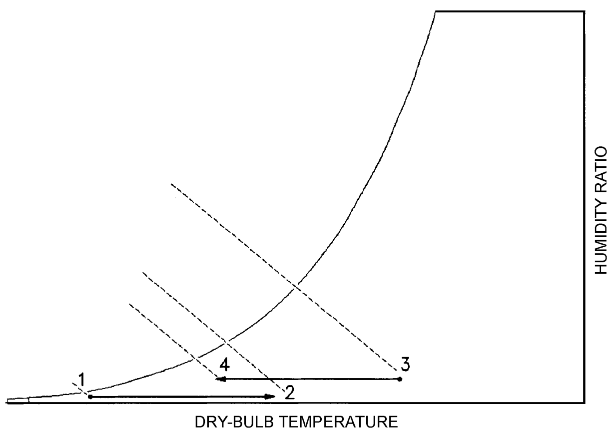

Fig. 7 Sensible Heat Recovery in Winter (Example 2)

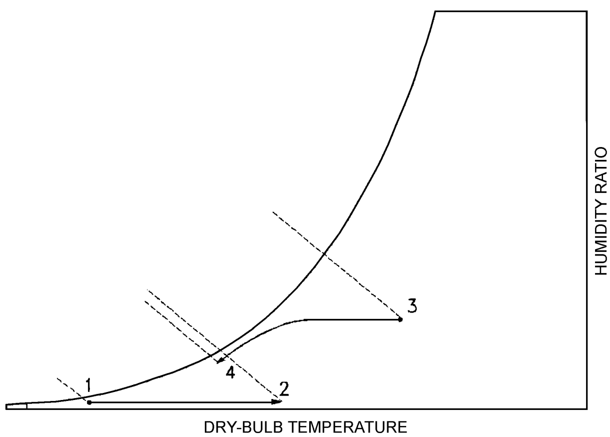

Fig. 8 Sensible Heat Recovery in Winter with Condensate (Example 3)

6. Plot conditions on psychrometric chart (Figure 8). Note that moisture will condense in the exhaust side of the heat exchanger.

Example 4. Total Heat Recovery in Summer

In this application, 10,000 cfm of exhaust air at 75°F and 63°F wet bulb (ρ = 0.075 lb/ft^3) is used to precool 8000 cfm of supply outdoor air at 95°F and 80°F wet bulb (ρ = 0.071 lb/ft^3) using a hygroscopic total energy exchanger. Determine the leaving supply air conditions, calculate energy recovered, and check energy exchange balance.

1. Calculate the theoretical maximum heat transfer.
2. The supply airstream is the lesser or limiting airstream for energy and moisture transfer. The hygroscopic energy exchanger will transfer moisture so the theoretical maximum heat transfer is calculated using Equation (5). Determine entering airstream enthalpies from psychrometric chart.

$$\text{Supply inlet } (95°\text{F db}, 80°\text{F wb}) \qquad h = 43.7 \text{ Btu/lb}$$
$$\text{Exhaust inlet } (75°\text{F db}, 63°\text{F wb}) \qquad h = 28.6 \text{ Btu/lb}$$

The intersection of sensible and latent heat transfer, shown as point A in Figure 9 (69.5°F wb, h = 33.8 Btu/lb).

$$q_{max} = 60 \times 0.071 \times 8000(43.7 - 28.6) = 515{,}000 \text{ Btu/h}$$
$$q_{latent} = 60 \times 0.071 \times 8000(43.7 - 33.8) = 337{,}000 \text{ Btu/h}$$
$$q_{sensible} = 60 \times 0.071 \times 8000(33.8 - 28.6) = 177{,}000 \text{ Btu/h}$$

3. Determine supply sensible and total effectiveness.

From manufacturer's selection data for the design conditions, the following effectiveness ratios are provided:

$$\varepsilon_{sensible} = 70\% \qquad \varepsilon_{total} = 56.7\%$$

4. Calculate energy transfer at design conditions.

$$q = 0.567 \times 515{,}000 = 292{,}000 \text{ Btu/h total recovered}$$
$$q_{sen} = -0.7 \times 177{,}000 = \underline{124{,}000 \text{ Btu/h sensible recovered}}$$
$$q_{lat} = 168{,}000 \text{ Btu/h latent recovered}$$

5. Calculate leaving air conditions.

Equation (6) is used to determine dry-bulb leaving conditions, and Equation (5) is used to determine leaving wet-bulb conditions.

a. Supply air conditions

$$t_2 = 95 + \left(\frac{-124{,}000}{60 \times 0.24 \times 0.071 \times 8000}\right) = 79.8°\text{F}$$
$$h_2 = 43.7 + \left(\frac{-292{,}000}{60 \times 0.071 \times 8000}\right) = 35.1 \text{ Btu/lb}$$

From the psychrometric chart, supply wet bulb = 71.1°F.

b. Exhaust air conditions

$$t_4 = 75 + \left(\frac{124{,}000}{60 \times 0.075 \times 0.24 \times 10{,}000}\right) = 86.5°\text{F}$$
$$h_4 = 28.6 + \left(\frac{292{,}000}{60 \times 0.075 \times 10{,}000}\right) = 35.1 \text{ Btu/lb}$$

From the psychrometric chart, exhaust wet bulb = 71.0°F.

6. Check total performance.

$$60 \times 0.071 \times 8000(43.7 - 35.1) = 292{,}000 \text{ Btu/h saved}$$
$$60 \times 0.075 \times 10{,}000(35.1 - 28.6) = 292{,}000 \text{ Btu/h recovered}$$

7. Plot conditions on psychrometric chart (Figure 9).

Example 5. Indirect Evaporative Cooling Recovery

In this application, 30,000 cfm of room air at 85°F and 63°F wb (ρ = 0.075 lb/ft^3) is used to precool 30,000 cfm of supply outdoor air at 101°F and 68°F wb (ρ = 0.070 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. Determine the leaving supply air conditions, calculate energy recovered, and check energy exchange balance.

First, determine the exhaust air condition entering the exchanger (i.e., after it is adiabatically cooled). Air at 85°F db, 63°F wb cools to 65°F db, 63°F wb as shown by the process line from point A to point 3 in Figure 10. In this problem the volumetric flows are equal, but the mass flows are not.

1. 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 (6).

$$q_{max} = 60 \times 0.24 \times 0.070 \times 30{,}000(101 - 65) = 1{,}089{,}000 \text{ Btu/h}$$

2. Establish the sensible effectiveness.

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

3. Calculate actual energy transfer at the design conditions.

$$q = 0.78 \times 1{,}089{,}000 = 849{,}000 \text{ Btu/h recovered}$$

4. Calculate leaving air conditions.

Because no moisture or latent heat is transferred, Equation (6) is used.

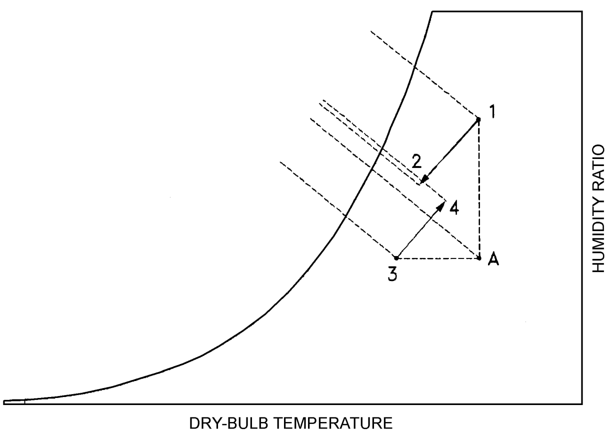

Fig. 9 Total Heat Recovery in Summer (Example 4)

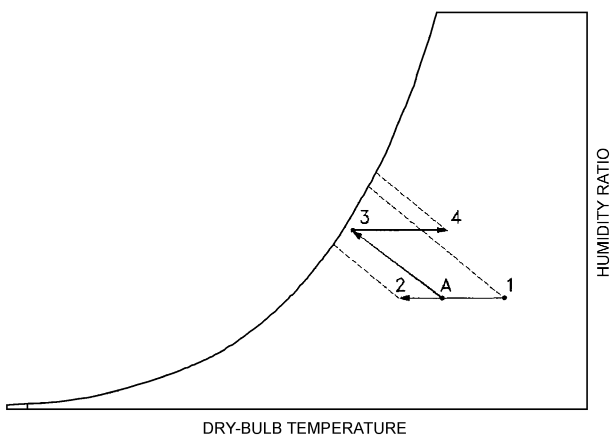

Fig. 10 Indirect Evaporative Cooling Recovery (Example 5)

a. Leaving supply air temperature is

$$t_2 = 101 + \left(\frac{-849{,}000}{60 \times 0.070 \times 0.24 \times 30{,}000}\right) = 72.9°\text{F}$$

b. Leaving exhaust air temperature is

$$t_4 = 65 + \left(\frac{849{,}000}{60 \times 0.24 \times 0.75 \times 30{,}000}\right) = 91.2°\text{F}$$

5. Check performance.

$$60 \times 0.070 \times 0.24 \times 30{,}000(101 - 72.9) = 849{,}000 \text{ Btu/h saved}$$
$$60 \times 0.075 \times 0.24 \times 30{,}000(91.2 - 65.0) = 849{,}000 \text{ Btu/h recovered}$$

6. Plot conditions on psychrometric chart (Figure 10), and confirm that no latent exchange occurred.

Example 6. Precooling Air Reheater Dehumidifier

In this application, 3300 cfm of outdoor supply air at 95°F and 80°F wb (ρ = 0.071 lb/ft^3) is used to reheat 3300 cfm of the same air leaving a cooling coil (exhaust) at 52.2°F and 52.1°F wb using a sensible heat exchanger as a precooling air reheater. The reheated air is to be between 75 and 78°F. In this application, the warm airstream is outdoor air and the cold airstream is the same air after it leaves the cooling coil. Determine the leaving precooled and reheated air conditions, calculate energy recovered, and check energy exchange balance.

1. Calculate the theoretical maximum energy transfer.

The air being reheated will have less mass than the outdoor air entering the precooler because moisture will condense from it as it passes through the precooler and cooling coil. Reheat is sensible heat only, so Equation (6) is used to determine the theoretical maximum energy transfer.

$$q_{max} = 60 \times 0.24 \times 0.071 \times 3300(95.0 - 52.2) = 144{,}000 \text{ Btu/h}$$

2. Establish the sensible effectiveness.

From the manufacturer's heat exchanger selection data, effectiveness is determined to be 58.4% at the designated operating conditions.

3. Calculate actual heat transfer at given conditions.

$$q = 0.584 \times 144{,}000 = 84{,}000 \text{ Btu/h}$$

4. Calculate leaving air conditions.

Because condensation is expected to occur as the outdoor airstream passes through the precooling side of the heat exchanger, Equation (5) is used to determine its leaving condition, which is the inlet condition for the cooling coil. The sensible heat transfer Equation (6) is used to determine the condition 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 80°F wb, is 43.7 Btu/lb.

$$h_2 = 43.7 + \left(\frac{-84{,}000}{60 \times 0.071 \times 3300}\right) = 37.7 \text{ Btu/lb}$$

The wet-bulb temperature for saturated air with this enthalpy is 73.6°F. This is point 2 on the psychrometric chart (Figure 11), which is near saturation. Note that this precooled air will be further dehumidified by a cooling coil from point 2 to point 3 in Figure 11.

b. Reheater leaving air conditions

$$t_4 = 52.2 + \left(\frac{84{,}000}{60 \times 0.24 \times 0.071 \times 3300}\right) = 77.1°\text{F}$$

Entering enthalpy, determined from the psychrometric chart for 52.2°F db and 52.1°F wb, is 21.4 Btu/lb.

$$h_4 = 21.4 + \left(\frac{84{,}000}{60 \times 0.071 \times 3300}\right) = 27.4 \text{ Btu/lb}$$

The wet-bulb temperature for air with this temperature and enthalpy is 61.4°F.

5. Check performance.

$$60 \times 0.071 \times 3300(43.7 - 37.7) = 84{,}000 \text{ Btu/h precooling}$$
$$60 \times 0.071 \times 0.24 \times 3300(77.1 - 52.2) = 84{,}000 \text{ Btu/h reheat}$$

6. Plot conditions on psychrometric chart (Figure 11).

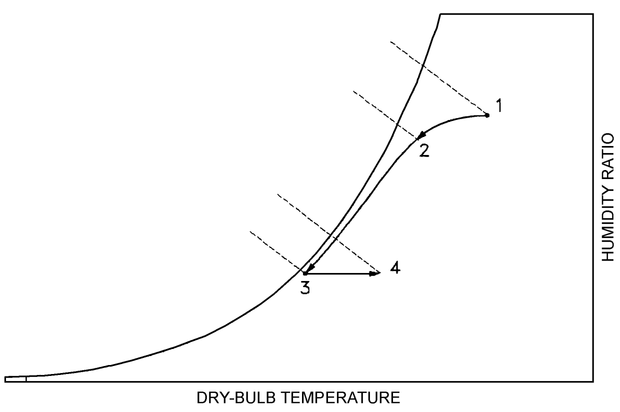

Fig. 11 Precooling Air Reheater Dehumidifier (Example 6)

EQUIPMENT

Table 2 provides comparative data for common types of air-to-air energy recovery devices. The following sections describe the construction, operation, and unique features of the various devices.

FIXED-PLATE EXCHANGERS

Fixed surface plate exchangers have no moving parts. Alternate layers of plates, separated and sealed (referred to as the heat exchanger core), form the exhaust and supply airstream passages (Figure 12). Plate spacings range from 0.1 to 0.5 in., depending on the design and the application. Heat is transferred directly from the warm airstreams through the separating plates into the cool airstreams. Design and construction restrictions inevitably result in cross-flow heat transfer, but additional effective heat transfer surface arranged properly into counterflow patterns can increase heat transfer effectiveness.

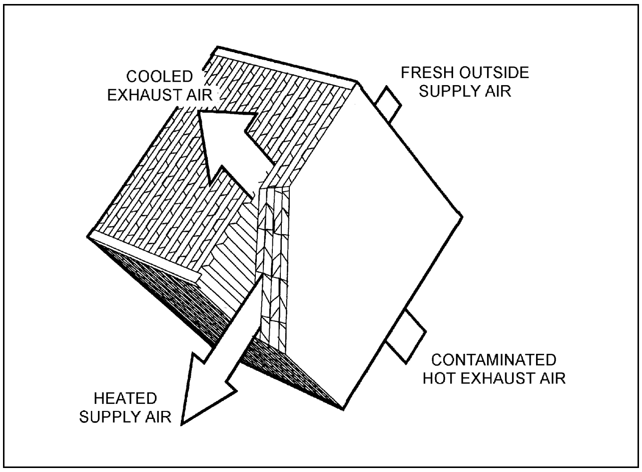

Fig. 12 Fixed-Plate Cross-Flow Heat Exchanger

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

	Fixed Plate	**Rotary Wheel**	**Heat Pipe**	**Runaround Coil Loop**	**Thermosiphon**	**Twin Towers**
Airflow arrangements	Counterflow Cross-flow Parallel flow	Counterflow Parallel flow	Counterflow Parallel flow	Counterflow Parallel flow	Counterflow Parallel flow	
Equipment size range, cfm	50 and up	50 to 70,000	100 and up	100 and up	100 and up	
Type of heat transfer (typical effectiveness)	Sensible (50 to 80%) Total (55 to 85%) Treated paper and poly membrane	Sensible (50 to 80%) Total (55 to 85%)	Sensible (45 to 65%)	Sensible (55 to 65%)	Sensible (40 to 60%)	Sensible (40 to 60%)
Face velocity, fpm (most common design velocity)	100 to 1000 (200 to 1000)	500 to 1000	400 to 800 (450 to 550)	300 to 600	400 to 800 (450 to 550)	300 to 450
Pressure drop, in. of water (most likely pressure)	0.02 to 1.8 (0.1 to 1.5)	(0.25 to 1.0)	(0.4 to 2.0)	(0.4 to 2.0)	(0.4 to 2.0)	0.7 to 1.2
Temperature range	−70 to 1500°F	−70 to 200°F	−40 to 95°F	−50 to 900°F	−40 to 104°F	−40 to 115°F
Typical mode of purchase	Exchanger only Exchanger in case Exchanger and blowers Complete system	Exchanger only Exchanger in case Exchanger and blowers Complete system	Exchanger only Exchanger in case Exchanger and blowers Complete system	Coil only Complete system	Exchanger only Exchanger in case	Complete system
Unique advantages	No moving parts Low pressure drop Easily cleaned	Latent (moisture mass) transfer Compact large sizes Low pressure drop	No moving parts except tilt Fan location not critical Allowable pressure differential up to 60 in. of water	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 Multiple units in a single system Efficient microbiological cleaning of both supply and exhaust airstreams
Limitations	Latent available in hygroscopic units only	Cold climates may increase service Cross-air contamination possible	Effectiveness limited by pressure drop and cost Few suppliers	High effectiveness requires accurate simulation model	Effectiveness may be limited by pressure drop and cost Few suppliers	Few suppliers
Cross-leakage	0 to 5%	1 to 10%	0%	0%	0%	0.025%
Heat rate control (HRC) schemes	Bypass dampers and ducting	Wheel speed control over full range	Tilt angle down to 10% of maximum heat rate	Bypass valve or pump speed control over full range	Control valve over full range	Control valve or pump speed control over full range

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 not uncommon.

Design Considerations

Plate exchangers are available in many configurations, materials, sizes, and flow patterns. Many are modular, and modules can be arranged to handle almost any airflow, effectiveness, and pressure drop requirement. Plates are formed with integral separators (e.g., ribs, dimples, ovals) 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 clean the heat transfer surfaces depends on the configuration.

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 construction material for plates because of its nonflammability and durability. Polymer plate exchangers have properties that may improve heat transfer by breaking down the boundary layer and are popular because of their corrosion resistance and cost-effectiveness. Steel alloys are used for temperatures exceeding 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 new microporous polymeric membranes, are used to transfer moisture, thus providing total (enthalpy) energy exchange.

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

Performance

Fixed-plate heat exchangers can economically achieve high sensible heat recovery and high 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 (i.e., pumping liquid, condensing and vaporizing gases, or transporting a heat transfer medium) inherent in other exchanger types. Simplicity and lack of moving parts add to the reliability, longevity, low auxiliary energy consumption, and safety performance of these exchangers.

Differential Pressure/Cross-Leakage

One of the advantages of the plate exchanger is that it is a static device built so that little or no leakage occurs between airstreams.

As velocity increases, the pressure difference between the two airstreams increases exponentially. High differential pressures may deform the separating plates and, if excessive, can permanently damage the exchanger. This may reduce effectiveness, alter design fan airflow, and cause excessive air leakage. This is not normally a problem because the differential pressures in most applications are less than 4 in. of water. In applications requiring high air velocities, high static pressures, or both, select exchangers that are designed for these conditions.

Condensing Within Exhaust Airstreams

Most plate exchangers are equipped with condensate drains, which remove the condensate and also wastewater when a water-wash system is used. 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.

Table 3 illustrates the effect of moisture content on the frost threshold for counterflow heat exchangers. Frosting can be controlled by preheating the incoming supply air, bypassing a portion of the incoming air, recirculating supply air through the exhaust side of the exchanger, or temporarily interrupting the supply air while maintaining exhaust. Bypassing reduces the ratio K in Table 3 and is generally more cost-effective than preheating.

For cross-flow plate heat exchangers, freezing and frost growth first occur at higher temperatures than those shown in Table 3. However, frost on cross-flow heat exchangers is less likely to block the exhaust airflow completely as with other types of exchangers. Generally, frost should be avoided unless a defrost cycle is included.

Microporous Permeable Membrane Fixed-Plate Exchangers

Fixed-plate heat exchangers can be fabricated of a permeable medium that transfers both moisture and heat from one airstream to the other. Media have been developed that minimize cross-leakage while maximizing moisture and energy transfer. Cross-stream mass transfer occurs without air transfer when the walls of an exchanger are made of such media. Suitable microporous permeable materials include cellulose, polymers, and other synthetic membranes.

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 one-half the exchanger in a counterflow pattern (Figure 13). Heat transfer media may be selected to recover sensible heat only or total heat (sensible heat 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 (1) condenses moisture from the airstream with the higher humidity ratio (either because the medium temperature is below its dew point or by means of absorption for liquid desiccants and adsorption for solid desiccants), with a simultaneous release of heat; and (2) releases the moisture through evaporation (and heat pickup) into the airstream with the lower humidity ratio. Thus, the moist air is dried while the drier air is humidified. In total heat transfer, both sensible and latent heat transfer occur simultaneously. Because rotary exchangers have a counterflow configuration and normally use small-diameter flow passages, they are quite compact and can achieve high transfer effectiveness.

Table 3 Frost Threshold Temperature for Various Counterflow Heat Exchanger Exhaust Air Conditions

Entering Exhaust Air		Frost Threshold Temperature t_1, °F			
		Ratio of Supply to Exhaust Airflow K			
t_3, °F	rh, %	0.5	0.7	1.0	2.0
60	30		15	23	32
60	40		15	23	32
60	50	–4		18	32
60	60	–9		13	32
70	30	–13		17	28
70	40	–21	–3	10	21
70	50	–27	–9		15
70	60	–32	–13	–1	10
75	30	–25	–4	10	23
75	40	–33	–12		15
75	50	–40	–20	–6	
75	60	–47	–26	–12	
80	30	–35	–11		19
80	40	–44	–20	–5	10
80	50	–53	–30	–14	
80	60	–62	–39	–23	–8
90	30	–58	–30	–11	
90	40			–24	–8
90	50				–20

Construction

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 man-made materials and provide either random flow 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, commonly used for comfort ventilation systems, 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 given values of airflow and pressure drop.

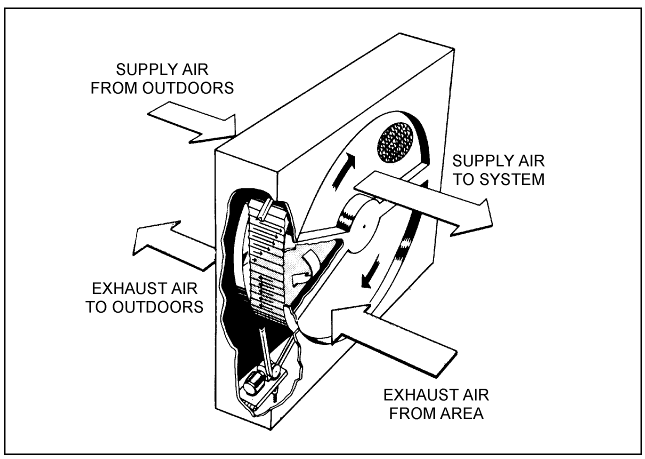

Fig. 13 Rotary Air-to-Air Energy Exchanger

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, 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 over 1000 ft^2/ft^3, depending on the type of medium and physical configuration. Media may also be classified according to 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 are fabricated 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-Contamination

Cross-contamination, or mixing, of air between supply and exhaust airstreams occurs in all rotary energy exchangers by two mechanisms—carryover and leakage. **Carryover** occurs as air is entrained within the volume of 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 clean rooms require stringent control of carryover. Carryover can be reduced to less than 0.1% of the exhaust airflow with a purge section (ASHRAE 1974).

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, purge, and carryover airflows.

Controls

Two control methods are commonly used to regulate wheel energy recovery. In the first, **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.

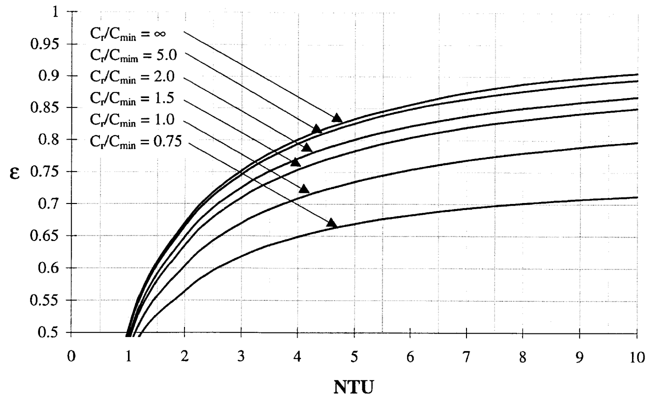

Fig. 14 Effectiveness of Counterflow Regenerator
(Shah 1981)

Figure 14 shows the effectiveness ε of a regenerative counterflow wheel versus number of transfer units (NTU). For sensible heat transfer only, with airflow balanced, convection-conduction ratio less than 4, and no leakage or cross-flow,

$$\text{NTU} = (UA)_{avg}/C_{min} \tag{8}$$

where

$(UA)_{avg}$ = product of modified overall heat transfer coefficient and heat exchange area, Btu/h·°F
C_{min} = minimum heat capacity rate of hot and cold airstreams, Btu/h·°F
C_r = heat capacity rate for air mass within rotary wheel, Btu/h·°F

Figure 14 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.

Mathematical models to describe the sensible and total energy effectiveness of regenerator wheels with hygroscopic coatings are under development. Until these models become accepted, however, desiccant wheels should be tested under conditions defined by ASHRAE *Standard* 84.

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 the outside air temperature is above the exhaust air temperature, the equipment operates at full capacity to cool the incoming air. During very cold weather, it may be necessary to heat the supply air, stop the wheel, or, in the case of small systems, use a defrost cycle for frost control.

Maintenance

Rotary enthalpy wheels require little maintenance. The following maintenance procedures ensure best performance:

- Clean the medium when lint, dust, or other foreign materials build up, following the manufacturer's instructions for that medium. Media treated with a liquid desiccant for total heat recovery must not be wetted.
- 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 do induction motors. Brushes should be replaced, and the commutator should be periodically turned and undercut.

- Inspect wheels regularly for proper belt or chain tension.
- Refer to the manufacturer's recommendations for spare and replacement parts.

COIL ENERGY RECOVERY (RUNAROUND) LOOPS

A typical coil energy recovery loop (Figure 15) 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 via counterflow piping through which an intermediate heat transfer fluid (typically water or an antifreeze solution) is pumped.

This system operates for sensible heat recovery only. In comfort-to-comfort applications, energy transfer is seasonally reversible—the supply air is preheated when the outdoor air is cooler than the exhaust air and precooled when the outdoor air is warmer.

Freeze Protection

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 the solution entering the exhaust coil at 30°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.

System Characteristics

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 the 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 values and low costs are to be realized (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). Manufacturer's 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.

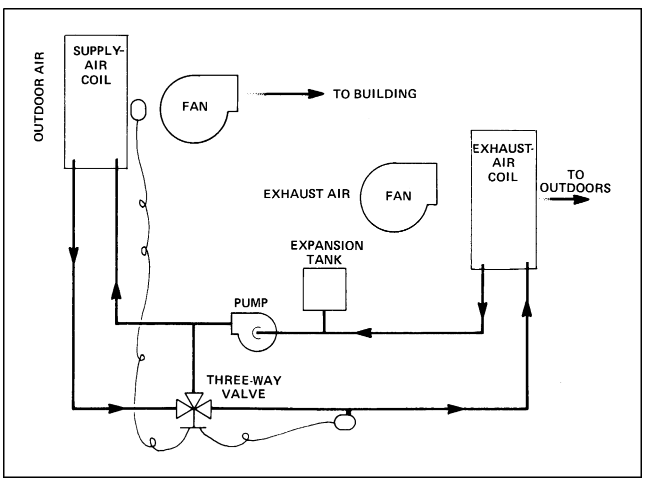

Fig. 15 Coil Energy Recovery Loop

Effectiveness

The coil energy recovery loop cannot transfer moisture from one airstream to another; 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%. Highest effectiveness does not necessarily give the greatest net cost saving.

The following example illustrates the capacity of a typical system:

Example 7. A waste heat recovery system heats 10,000 cfm of air from a 0°F design outdoor temperature to an exhaust dry-bulb temperature of 75°F and wet-bulb temperature of 60°F. (From Example 2, q_{max} = 810,000 Btu/h.) Air flows through identical eight-row coils at a 400 fpm face velocity. A 30% ethylene glycol solution flows through the coils at 26 gpm.

Figure 16 shows the effect of the 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 18.5°F. This constant output occurs because the valve has to control the temperature of the 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. Equation (2) may be used to calculate the sensible heat effectiveness.

When the three-way control valve operates at outside air temperatures of 18.5°F or lower, 414,000 Btu/h is recovered. At 18.5°F, the sensible heat effectiveness is 67.2%. At the 0°F design temperature, sensible effectiveness is 51% (ε = 414,000/810,000), and the leaving air dry-bulb temperature of the supply coil is 38.3°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 (as in Example 7), the sensible heat effectiveness decreases.

Construction Materials

Coil energy recovery loops incorporate coils constructed to suit the environment and operating conditions to which they are exposed. 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

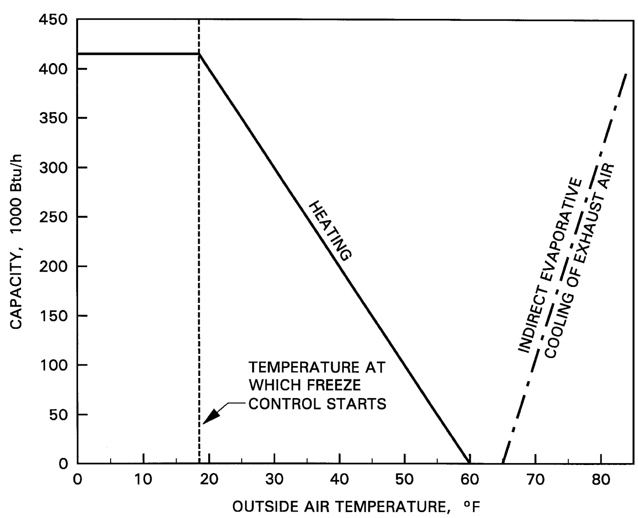

Fig. 16 Energy Recovery Capacity Versus Outside Air Temperature for Typical Loop

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

Cross-Contamination

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

Maintenance

Coil energy recovery loops require little maintenance. The only moving parts are the circulation pump and the 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.

Thermal Transfer Fluids

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 commonly used 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

A heat pipe heat exchanger is a passive energy recovery device. It has the outward appearance of an ordinary plate-finned water or steam coil, except that the tubes are not interconnected and the pipe heat exchanger is divided into evaporator and condenser sections by a partition plate (Figure 17). Hot air passes through the evaporator side of the exchanger, and cold air passes through the condenser side. Heat pipe heat exchangers are sensible heat transfer devices, but condensation on the fins does allow latent heat transfer, resulting in improved recovery performance.

Heat pipe tubes (Figure 18) are fabricated with an integral capillary wick structure, evacuated, filled with a suitable working fluid, and permanently sealed. The working fluid is normally a Class I refrigerant, but other fluorocarbons, water, and other compounds are used for applications with special temperature requirements.

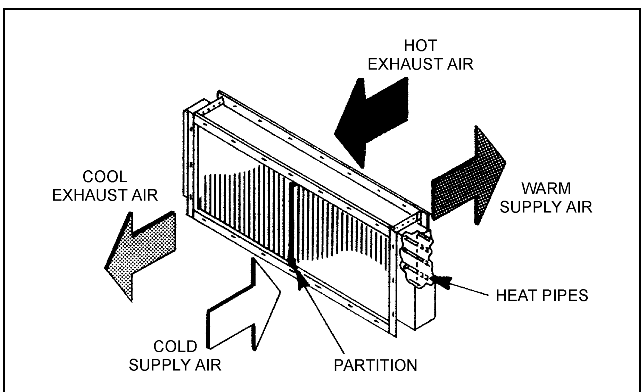

Fig. 17 Heat Pipe Heat Exchanger

Fin designs include continuous corrugated plate fin, continuous plain fin, and spiral fins. Modifying fin design and tube spacing changes pressure drop at a given face velocity.

Principle of Operation

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. 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, heat transfer along a heat pipe is up to 1000 times faster 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 such factors as wick design, tube diameter, working fluid, and tube (heat pipe) orientation relative to horizontal.

Construction Materials

HVAC systems use copper or aluminum heat pipe tubes with aluminum fins. 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 designed for finned tube heat exchangers have permitted 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.

Operating Temperature Range

Selection of 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 cause the formation of noncondensable gases that deteriorate performance.

Cross-Contamination

Heat pipe heat exchangers typically have zero cross-contamination for pressure differentials between airstreams of up to 50 in. of water. Constructing a vented double-wall partition between the air-

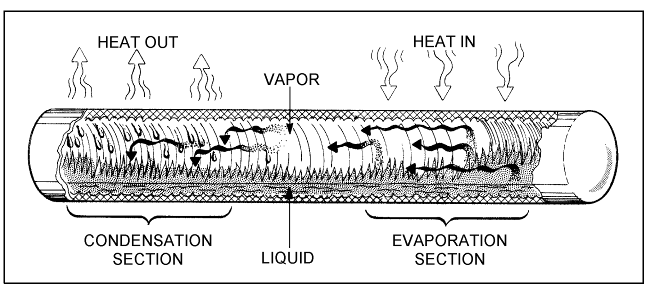

Fig. 18 Heat Pipe

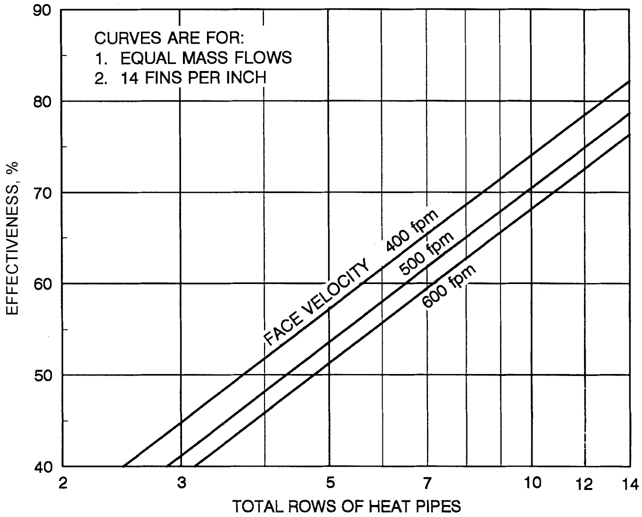

Fig. 19 Heat Pipe Exchanger Effectiveness

streams can provide additional protection against cross-contamination. If an exhaust duct is attached to the partition space, any leakage is withdrawn and exhausted from the space between the two ducts.

Performance

Heat pipe heat transfer capacity depends on design and orientation. Figure 19 presents a typical effectiveness curve for various face velocities and rows of tubes. As the number of rows increases, effectiveness increases at a decreasing rate. For example, doubling the number of rows of tubes in a 60% effective heat exchanger increases the effectiveness to 75%. The effectiveness of a counterflow heat pipe heat exchanger depends on the total number of rows such that two units in series yield the same effectiveness as a single unit of the same total number of rows. Series units are often used to facilitate handling, cleaning, and maintenance.

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 limit 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 the 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/in. is common. Wider fin spacings (8 to 10 fins/in.) are usually used for industrial applications. Heat pipe heat exchangers of the plate fin type can easily be constructed with different fin spacings 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 due to dirt buildup on the exhaust side surface.

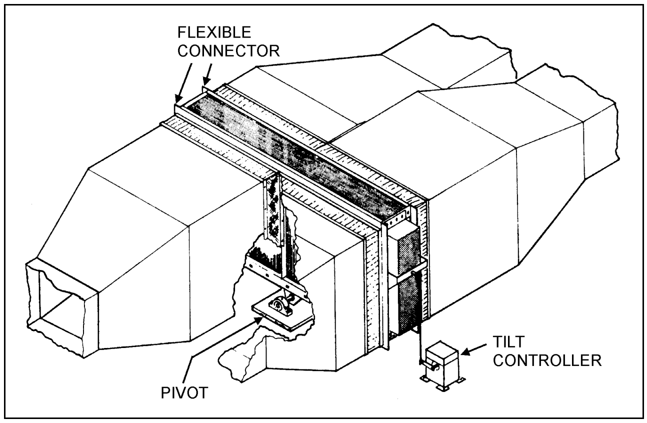

Fig. 20 Heat Pipe Heat Exchanger with Tilt Control

Controls

Changing the slope, or 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) the 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.

In practice, tilt control is effected by pivoting the exchanger about the center of its base and attaching a temperature-controlled actuator to one end of the exchanger (Figure 20). Pleated flexible connectors attached to the ductwork allow freedom for the small tilting movement (6° maximum).

The following are three functions for which tilt control may be desired:

- To change the operation from supply air heating to supply air cooling (i.e., to reverse the direction of heat flow) when seasonal changeover occurs.
- To modulate effectiveness to maintain desired supply air temperature. This kind of regulation is often required for large buildings to avoid overheating the air supplied to the interior zone.
- To decrease effectiveness to prevent frost formation at low outdoor air temperatures. With reduced effectiveness, the 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 be used for individual functions.

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 21). 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) are usually bactericidal and viricidal. 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.

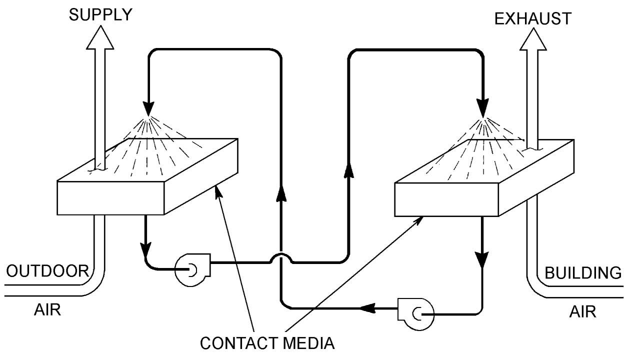

Fig. 21 Twin Tower Enthalpy Recovery Loop

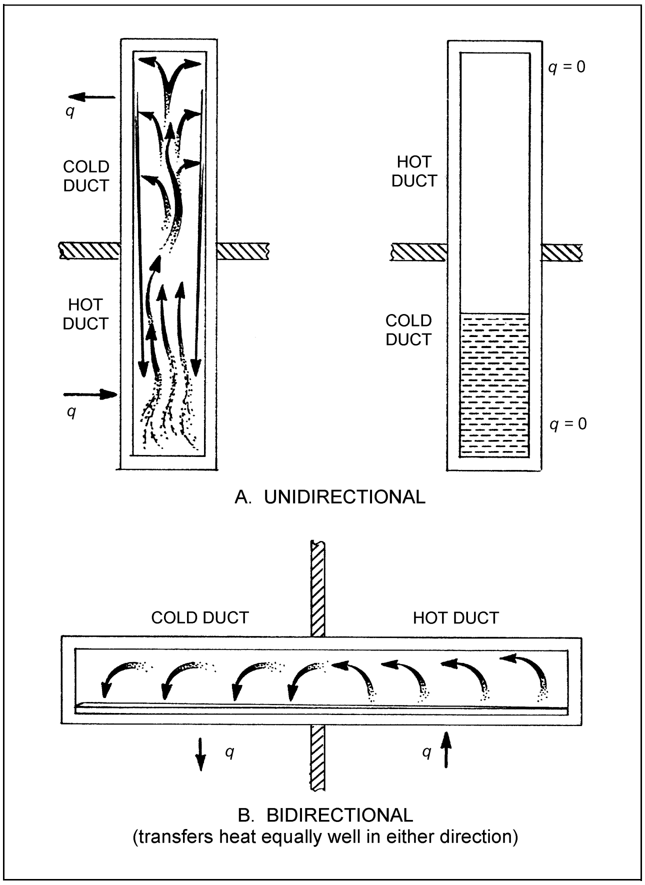

Fig. 22 Sealed Tube Thermosiphons

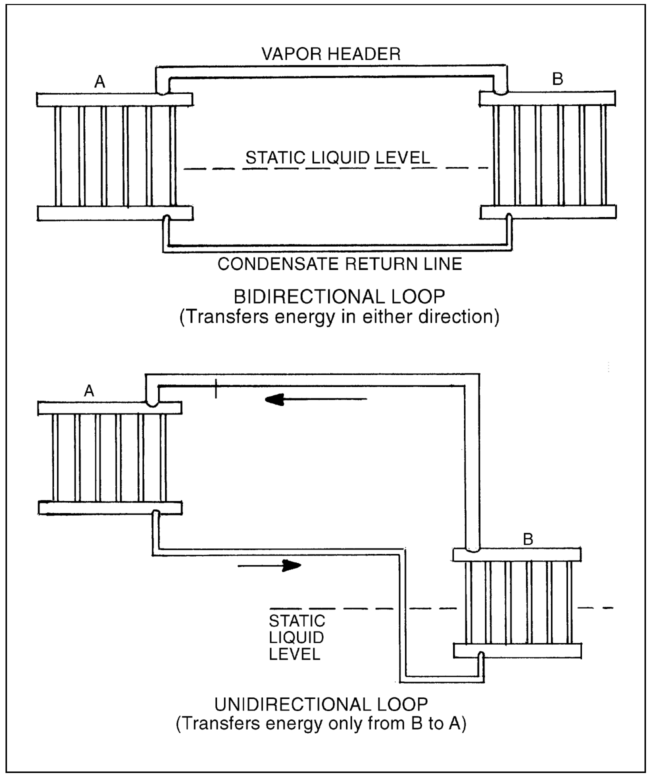

Fig. 23 Coil-Type Thermosiphon Loops

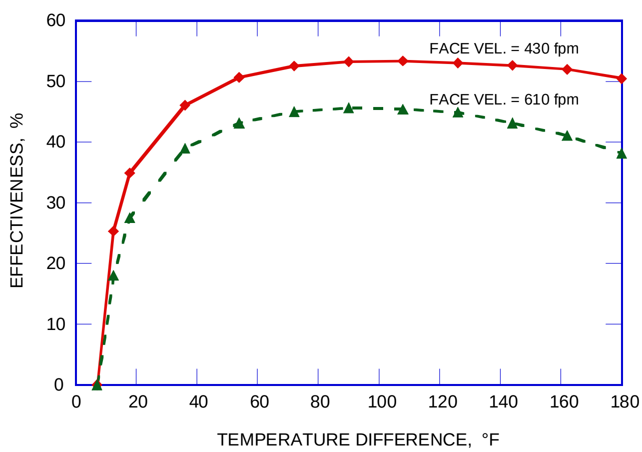

Fig. 24 Typical Performance of Two-Phase Thermosiphon Loop

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

Two-phase thermosiphon heat exchangers are sealed systems that consist of an evaporator, a condenser, interconnecting piping, and an intermediate working fluid that is present in both liquid and vapor phases. Two types of thermosiphon are used—sealed tube (Figure 22) and coil type (Figure 23). 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).

In thermosiphon systems, a temperature difference and gravity force a 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 the 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. As a result, thermosiphon heat exchangers may require a significant temperature difference to initiate boiling (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 1990c). Figure 24 shows the performance of a thermosiphon loop (Mathur and McDonald 1986).

Principle of Operation

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 two regions in a thermosiphon where liquid and vapor interfaces are present, the resulting vapor pressure difference causes vapor to flow from the warmer to the colder region. The flow is sustained by condensation in the cooler region and by evaporation in the warmer region. The condenser and evaporator must be oriented so that the condensate can return to the evaporator by gravity (Figure 22 and Figure 23).

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