Designing for air to air heat and moisture exchange in HVAC applications by C Besant, C Simonson and Wei Shang

Abstract

Air-to-air heat and moisture exchange between exhaust and supply airflows can substantially reduce HVAC costs. This paper outlines the design considerations that should be included when selecting a type of exchanger and shows how the performance of each recovery device can be determined. Energy wheels, which transfers both heat and water vapor, are given special consideration. The HVAC design for heat and moisture exchanger sizing is presented as a least life-cycle cost design problem.

Introduction

Conditioning outdoor supply air for building ventilation often is accompanied by high capital and operation costs for HVAC equipment and utilities. On the other hand, exhaust air, aside form some airborne contaminants such as odors, often has properties (e.g. temperature and moisture content) closer to the those required for the ventilation supply air than the ambient outdoor air. Exchangers, which can transfer heat and moisture between the exhaust and supply airstreams and exclude exhaust air contaminants, can substantially reduce HAVC costs and they are likely the least capital cost design alternative.

In the past air-to-air heat exchangers have been used between the exhaust and supply air especially for air preheating but moisture exchange was problematic before the advent of desiccant coated rotary wheels, also known as energy wheels or enthalpy wheels. Moisture exchange is most advantageous for humid outdoor air in summer but winter ambient air is often too low in moisture content for human comfort. Heat exchangers can only be used to transfer heat while energy wheels can be very effective in transferring both heat and moisture between the exhaust and supply air.

It is the purpose of this paper to explain the function and selection of each type of air-to-air exchanger and finally to explain how the least life-cycle cost design process for various air-to-air exchangers differs from most HVAC design problems.

Airflow Configuration for Heat and Moisture Exchange between Supply and Exhaust Airflows

Several types of air-to-air heat exchangers have been employed in HVAC applications. These include plate type, regenerative heat wheel, run-around and heat pipe heat exchangers (ASHRAE, 1996). These nearly passive devices have been most often configured into an HVAC system as in Figure 1. In warm and moderately warm climates heat exchangers that do not transfer moisture were not usually considered to be cost effective in a system as in Figure 1. However, when configured as in Figure 2a and b and when used with exhaust air evaporative coolers (Dhital et al.,1995) or as supply air pre-heaters and post-heaters with auxiliary air chillers as in Figure 2, they can be cost effective in warm climates (Hill and Jetter, 1994 and Paarporn, 1999). The system in Figure 2a allows for complete control of the supply air condition while Figure 2b is typical of some packaged units which rely on some auxiliary space heating not shown in this figure.

The energy wheel which transfers both heat and moisture between the supply and exhaust airstreams is a more recent device and it is favored for HVAC applications with large cooling loads requiring large power inputs for supply air moisture removal. Furthermore, because energy wheels are also effective heat exchangers, they are used for any application where moisture transfer is permitted. They are excluded from the supply air downstream of the auxiliary cooling coils in Figure 2a and b.

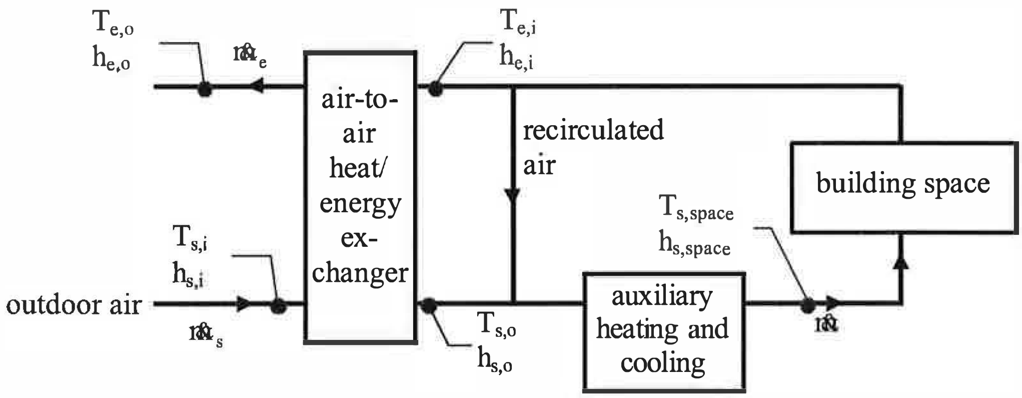
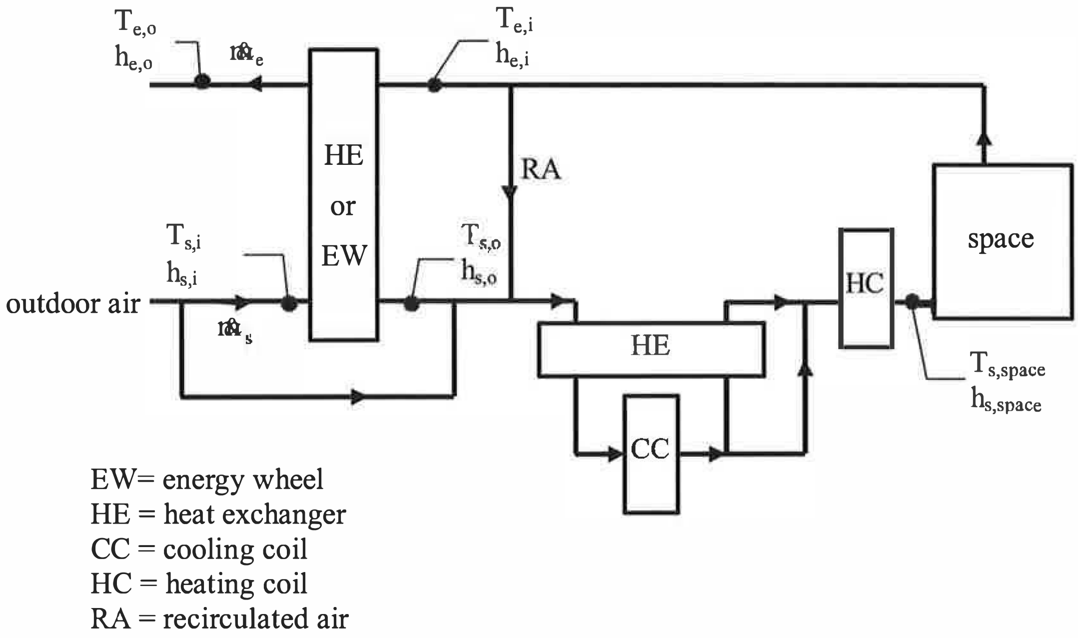
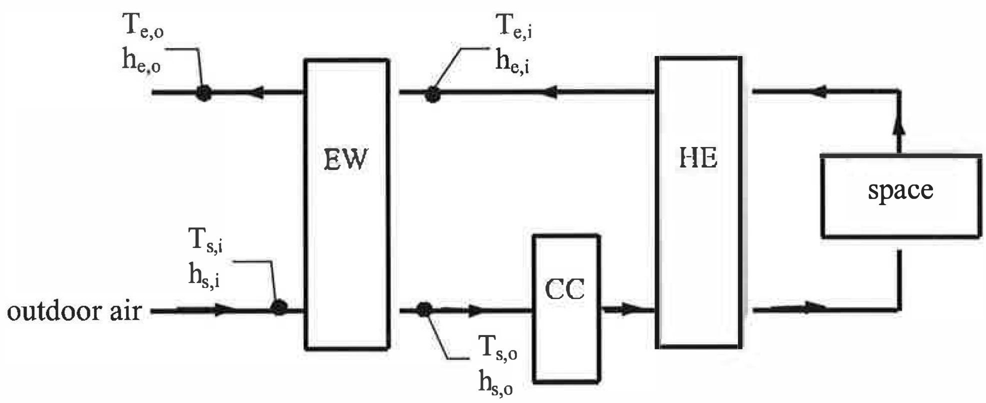


Figure 1. *Schematic of a HVAC system with air-to-air heat/energy recovery.*



***(a)** Site built system with some recirculated air*



***(b)** components in a packaged air conditioning and recovery unit*

Figure 2. *Schematics of HVAC systems with two air-to-air heat/energy recovery devices.*

Energy Changes in Air Flowing Through Air-to-Air Heat and Moisture Exchangers

Exhaust and supply air for buildings flow steadily through air-to-air exchangers such that one airstream loses energy while the other gains energy. There are three forms of energy content or storage in any airstream within an exchanger. Energy may be stored in any unit of mass of air due to its mechanical motion or velocity (e.g. kinetic energy), temperature and pressure (e.g. internal energy), gaseous/chemical composition (e.g. chemical phase energy) due to the mass fraction of water vapor. For air flows within exchangers, the changes in mechanical energy due to velocity changes are only important when considering pressure losses which accompany any airflow. For most operating conditions, the mechanical energy changes are usually much smaller than the energy changes measured by the air temperature or water vapor content change. When the air moisture concentration does not change, the internal energy changes are due to only temperature changes and are referred to as sensible energy changes. In the absence of leakage or cross contamination, changes in water vapor concentration in an air stream flowing through an exchanger are usually accompanied by a phase change between vapor and liquid (water) or solid (frost or ice). Thus, it is common to present this form of change in water vapor composition in air as a latent energy change and refer to the energy due to the concentration of water vapor in air as its latent energy.

The main reason for expressing any moisture concentration change in air as an energy change is because when this latent energy change is added to the sensible energy change for air flowing steadily within an exchanger, we obtain the enthalpy change which, excluding the mechanical energy change, gives the total energy change required to change the temperature and moisture content of air flowing through an exchanger.

The absolute values for sensible and latent energies are not important in air-to-air exchangers because only the energy changes that occur as the air flows through the exchanger on the supply and exhaust sides are of interest. Above freezing, the latent heat of evaporation or condensation (h_{fg}) is used and it is nearly constant for typical air property changes in air-to-air exchangers but is has a small temperature dependence. (Frosting conditions are always unsteady and they will be treated separately.) For a unit mass of dry air flowing, we can write the enthalpy, h, as the sum of the sensible and latent energy (ASHRAE Fundamentals, 1997)

$$h = C_p t + W\left(h_{fg_{ref}} + (C_{pv} \frac{dh_{fg}}{dt})\ t \right) \tag{1}$$

where,

C_p = specific heat of dry air, kJ/kg°F (Btu/lb°F),
C_{pv} = specific heat of water vapor, kJ/kg°C (Btu/lb°F),
t = temperature, °C (°F),
W = absolute humidity or moisture content (mass of water vapor per unit mass of dry air),
$h_{fg_{ref}}$ = reference value of h_{fg}, kJ/kg (Btu/lb).

The total energy change rate (sensible and latent) for either the supply or exhaust airflow in an air-to-air exchanger with moderate temperature differences becomes

$$\dot{m}\,\Delta h = \dot{m}\left[C_p\ \Delta t + \Delta W\left(h_{fg_{ref}} + (C_{pv} + \frac{dh_{fg}}{dt})\Delta t\right)\right], \tag{2}$$

where $\dot{m}$ is the mass flow rate of dry air.

The fundamental question that we might ask here is how do we design a device that will take full advantage of the energy in the exhaust air to condition the supply air?

Energy Balances and Performance Factors of Heat Exchangers

Conservation of energy within a heat exchanger dictates that, in the absence of heat or mass flow leaks to the surroundings, the heat rate of a heat exchanger is balanced by the rate of change of energy in the supply and exhaust air streams. Thus,

$$q = \dot{m}_s \Delta h_s = -\dot{m}_e \Delta h_e \tag{3}$$

where the subscript "s" refers to the supply and "e" refers to the exhaust airstreams. The fact that enthalpy is used in this equation and not just temperature for the sensible energy change includes the case of condensation within the heat exchanger for the case that energy in the condensate is negligible. When there is no condensation, there will be no change in humidity ratio and

$$\dot{m}\Delta h = \dot{m} C_p \Delta t \tag{4}$$

which is the sensible energy change rate.

Heat exchanger performance is given by the <u>sensible effectiveness,</u> ε_s, a dimensionless number between 0 and 1.0 that is defined by the equation

$$\varepsilon_s = \frac{q}{q_{max}} = \frac{\dot{m}_s (t_{s,o} - t_{s,i})}{\dot{m}_{min} (t_{e,i} - t_{s,i})} \tag{5}$$

where q_{max} is the maximum possible heat rate of a heat exchange of infinite core area operating with $\dot{m}_s, \dot{m}_e$, and a temperature difference between the exhaust inlet and supply inlet of $t_{e,i} - t_{s,i}$ and is defined by

$$q_{max} = \dot{m}_{min} C_p (t_{e,i} - t_{s,i}) \tag{6}$$

where $\dot{m}_{min}$ is the minimum mass flow rate of air for $\dot{m}_s$ and $\dot{m}_e$.

For sensible energy changes only for the air flow streams, ε_s is constant or very nearly constant – so the heat rate is readily calculated for any known operating condition with the equation

$$q = \varepsilon_s \dot{m}_{min} C_p (t_{e,i} - t_{s,i}) \tag{7}$$

For a given recuperator heat exchanger operating at steady state, ε_s depends only on the ratio of mass flow rates, $\dot{m}_s / \dot{m}_e$, and another dimensionless factor (number of transfer units) defined by

$$N_{tu} = \frac{UA}{\dot{m}_{min} C_p} \tag{8}$$

where U is the overall heat exchanger conductance and A is the surface area of the heat exchanger core.

For plate and heat pipe heat exchangers, the heat rate, q, for sensible energy changes can be written as

$$q = UA\Delta t_{lm}F \tag{9}$$

where Δt_{lm} is the log mean temperature difference for a counterflow heat exchanger and F is a temperature distribution correction factor for other flow directions through the heat exchanger (see most heat transfer texts, e.g. Incorpera and DeWitt, 1996). Ntu is seen to be closely related to the ratio of heat rate, q, divided by q_{max}. As a consequence, the sensible effectiveness, ε_s, increases indefinitely with an increase in N_{tu} (i.e., surface area A) but for large values of N_{tu}, the changes in ε_s are small for a given change in N_{tu}. Generally, it is impractical to use heat exchanger designs with N_{tu} greater than 5 while a value of less than 1.5 is too small because the sensible effectiveness will be small.

For a rotary heat wheel, ε_s is also dependent on another heat exchanger factor which accounts for the thermal capacity of the mass in the rotating core of the regenerative heat exchanger $(MC_p)_{core}$

$$N_\omega = \frac{(MC_p)_{core}\,\omega}{\dot{m}_{min}C_p} \tag{10}$$

where ω is the angular speed of the wheel. N_ω, like N_{tu}, should be kept in a certain narrow range for good design. Too high a value (e.g. greater than 5) causes excessive carry over of one airstream into the other and too low a value (e.g. less than 2) causes a low sensible effectiveness. Equations and graphs relating $\varepsilon_s(\frac{\dot{m}_s}{\dot{m}_e}, N_{tu})$ and $\varepsilon_s(\frac{\dot{m}_s}{\dot{m}_e}, N_{tu}, N_\omega)$ for plate and rotary heat exchangers are presented in Kays and London (1984) and Shah (1981) and Shah et al. (1988).

Run-around systems are somewhat similar to rotary heat wheels where the run-around liquid flow thermal capacitance rate between the exhaust reclaim and supply air heat exchangers $(\dot{m}Cp)_{liq}$ results in another non-dimensional factor

$$N_{liq} = \frac{(\dot{m}Cp)_{liq}}{\dot{m}_{min}Cp} \tag{11}$$

similar to (10). Theoretically, N_{liq} should be close to 1.0 for a good design, however, N_{liq} must be significantly greater than 1.0 because the liquid flow through each coil-tube heat exchanger should be turbulent. On the other hand, too high a value of N_{liq} limits the overall heat exchanger system effectiveness to less than 50%. Temperature dependent properties and transition fluid turbulence characteristics within the liquid flow do not permit simple correlations for run-around systems (Bennett et al., 1994).

Performance factors other than the sensible effectiveness should be considered in the design and selection of air-to-air heat exchangers (ASHRAE Std 84, 1991, ARI Std 1060, 1997). These are the frictional pressure losses due to airflow through the exchanger and air leakage and cross contamination. Air pressure losses are characterized by a direct measurement of the static pressure loss across the heat exchanger device. Air leakage can be characterized by a leakage flow measurement but cross contamination measurements require tracer gas testing.

Important Characteristics of Heat Exchangers

In this section, we explain some other characteristics of a few commonly used air-to-air heat exchangers in more detail. These are the plate type and heat pipe recuperators rotary heat wheels and run-around systems (ASHRAE, 1996).

Plate-type heat exchangers have been used in stoves and furnaces for more than 100 years where they heat room air while combustion air discharges out the chimney. In these devices, some of the sensible energy in the exhaust air after the combustion chamber is transferred to the supply room air in the heat exchangers by means of heat transfer across the heat exchanger core.

Theoretically, the sensible effectiveness for these plate devices of a given core area, A, is highest for the counterflow arrangement where the exhaust air flow direction is 180° different than the supply flow direction. Counterflow heat exchangers are often more expensive to manufacture than other types of plate heat exchangers because the inlet and outlet headers are geometrically complex. The most commonly used plate heat exchanger is the cross-flow heat exchanger where the supply and exhaust air flow directions are at 90° to each other and the inlet and outlet headers are simple.

Plate type heat exchangers are made from several types of materials including metal (e.g. aluminum sheet), plastic (e.g. polyethylene) and even paper. From the point of view of resistance to heat transfer within the heat exchangers, the material used in the core is of little importance. The most important heat transfer factors that can reduce the heat transfer resistance are the local airflow speeds and surface roughness effects within the core. Since airflow pressure drop can be a problem for plate heat exchangers, the heat exchange surface area per unit volume must not be too high and attention must be given to plate deformation under pressure differences. As a consequence, the materials used in plate heat exchanger cores are selected for other reasons such as: cost, ease of manufacturing, mechanical strength, fire resistance, cleanability, resistance to corrosion, and long-term durability.

Most air-to-air heat exchangers must be controlled for part-load conditions. Part-load operating conditions occur most of the time for many HVAC heat recovery applications because the supply air does not need to be heated to a temperature very close to room temperature. Indeed, many HVAC systems require the supply air temperatures for rooms to be several degrees cooler than room air because the internal energy dissipation from lights, people, computers, and other appliances adds to the sensible energy of the room air and this must be removed. Since air-to-air heat exchangers often have sensible effectivenesses of 70% or higher, part-load operating conditions may occur most of the time.

Plate-type heat exchangers have no internal means of controlling heat rates; therefore, they must use some form of face-and-bypass control. Face-and-bypass control is achieved by using external dampers to bypass part or all the supply airflow around the plate heat exchanger. For buildings that allow a large air infiltration rates the HVAC designer may depend on extra air infiltration to make up for a larger exhaust flow rate than a supply flow rate. For airtight buildings, bypass ducting should be used. This method permits heat recovery fractions from zero up to the sensible effectiveness of the plate exchanger when the supply and exhaust airflows are balanced.

Heat-pipe heat exchangers have been used in air-to-air energy recovery applications since the 1960's after they were first developed for the space program. Heat pipes use a sealed-in intermediate fluid (e.g. a refrigerant) which undergoes boiling on the end where heat is added and vapor condensation at the other end where heat is removed. The temperature difference required to sustain boiling and condensation within the same tube is usually very small (e.g. 1 or 2 °C (2 or 4 °F)) while the heat rates can be very large for a given tube size. Air-to-air heat-pipe heat exchangers use a large array of heat pipes usually with attached fins that span both the supply and exhaust airflow streams arranged in a side-by-side arrangement.

Heat-pipe heat exchanger airflow directions are counter flow for the supply and exhaust airflows through the array of tubes. They usually have high sensible effectivenesses and high heat exchange surface areas per unit volume. Normally they are constructed from aluminum or copper tubes with aluminum fins but for high temperature applications, steel may be used.

There are two commonly used methods to control the part-load performance of heat-pipes. These are face-and-bypass supply airflow control and heat-pipe tilt control. Heat-pipe tilt control is achieved by tilting the heat-pipe exchanger a few degrees from the horizontal so that the hot end of the heat pipes achieves a partial or complete dry out condition on the inside of each tube. When dry out occurs, the heat rate of the heat pipe heat exchanger may drop by up to a factor of ten compared to the case of a flooded hot end (Guo et al., 1998). Complete prevention of heat flow using tilt control cannot be achieved. If complete prevention of heat flow is required, the face-and-bypass method of control must be used.

Run-around heat exchangers have been used in air-to-air applications for several decades. Run-around systems require the pumped, freeze-protected liquid (i.e., aqueous glycol) to be circulated between separate exhaust and supply coil-tube heat exchangers. They offer the unique advantage that these coil-tube heat exchangers can be located a considerable distance from each other (i.e., up to 100m) and several heat exchangers may be used to reclaim exhaust air energy while only one supply air exchanger is used.

Run-around systems are not easy to design for a high effectiveness because the coupling liquid heat transfer properties vary with the operating temperature and two heat exchangers are necessarily coupled. Consequently, they often have an overall effectiveness less than 60% even when designed for the least life-cycle cost (Johnson et al., 1995).

Part-load operating conditions are met by bypassing some of the coupling fluid using a flow control valve so a complete bypass of all the coupling fluid results in no supply air heating or cooling.

Rotary heat wheels have been used for more than 50 years, first as heat recovery devices using hot chimney exhaust gases to preheat combustion air for boilers and gas turbines and later for HVAC applications.

The airflow directions are counterflow through the matrix core of the rotary heat wheel. Very high heat-exchange surface area per unit volume of core are often used because the air pressure drop due to air flow is often smaller than other types of heat exchangers. The rotation of heat wheels causes some air from the supply side to discharge into the exhaust side and vice versa. Consequently, the supply air maybe slightly contaminated by the exhaust air even when the rotational speed is small (e.g. 10 to 20 RPM) and a purge section is used to reduce the cross contamination to less than 1%. As a result, heat wheels are not used in applications where cross contamination must be zero.

The matrix cores of rotary heat wheels are usually metal although plastics and other materials have been used. In all cases, the matrix material is selected to achieve very large heat exchange surface areas (e.g. thin materials) and airflow passages with a small resistance to flow. The flow is usually laminar through a rotary heat wheel even though the duct flow is turbulent.

The part-load heat rate control for rotary heat wheels is achieved by controlling the wheel speed. At a wheel speed of zero, the heat rate will be zero and at full speed (e.g. 10 to 20 RPM) the maximum heat rate will be a maximum given by the sensible effectiveness (ε_s) times the maximum possible heat rate (q_{max}).

Coupled Heat and Moisture Transfer

Two methods of transferring both heat and moisture in the same air-to-air exchanger have been demonstrated and commercially developed. One method is to construct the exchanger using permeable or semi-permeable plates through which both heat and water vapor molecules are transferred by diffusion or capillary action. The other is to employ regenerative rotary wheels with desiccant coated surfaces that readily adsorb and desorb water vapor as well as cyclically transfer heat. These wheels have been well researched and are now widely available and used in a wide variety of HVAC heat and moisture recovery applications. Conversely, permeable and semi-permeable plate exchangers are under development and in need of independent research to clearly define their performance characteristics as well as their risks. With these facts in mind, only the regenerative rotary energy wheels (also called enthalpy wheels) will be considered.

Whereas, heat exchangers that transfer only heat between the supply and exhaust air-streams and cause a change in only the air temperature in each stream, energy wheels cause a change in temperature (t) and humidity ratio (W). For the heat exchangers, each airstream undergoes a change in its sensible energy but for the energy wheels, the total energy change in either air stream is characterized by its enthalpy (h) or the sum of the sensible and latent energy.

For heat exchangers, performance is characterized by a single factor - the sensible effectiveness (ε_s). For energy wheels, two performance factors are needed along with the operating condition of the exchanger. That is, we need to know the sensible energy and latent energy (or moisture) effectivenesses (e.g. ε_s and ε_l) at each operating condition. When the total energy change in the supply air is required, the change in enthalpy of this air must be determined. The total energy or enthalpy effectiveness (ε_t) defines this performance factor.

The definitions of latent and total effectivenesses are somewhat similar to equation (5) but the latent heat of evaporation is assumed to be a constant giving

$$\varepsilon_l = \frac{\dot{m}_s (W_{s,o} - W_{s,i})}{\dot{m}_{min} (W_{e,i} - W_{s,i})} \tag{12}$$

$$\varepsilon_t = \frac{\dot{m}_s (h_{s,o} - h_{s,i})}{\dot{m}_{min} (h_{e,i} - h_{s,i})} \tag{13}$$

and q_{max} is not interpreted as the maximum possible energy transfer rate for ε_s, ε_l or ε_t because under certain operating conditions these effectivenesses are not bound between 0 and 1.0.

The total energy or enthalpy effectiveness is also calculated from the equation (Simonson and Besant, 1997)

$$\varepsilon_t = \frac{\varepsilon_s + H^* \varepsilon_l}{1 + H^*} \tag{14}$$

where the operating condition factor (H^*) is calculated from the definition equation

$$H^* = \frac{1}{SHR} - 1 = K\frac{\Delta W}{\Delta t} = K\frac{W_{s,i} - W_{e,i}}{t_{s,i} - t_{e,i}} \tag{15}$$

where $K = 2500/{}^\circ C$ for SI units and $4500/{}^\circ F$ for inch-pound units and SHR is the well known "sensible heat ratio" as defined on the ASHRAE psychrometric chart. The total energy rate (q_t) is then written as

$$q_t = \dot{m}_{min} \varepsilon_t (h_{e,i} - h_{s,i}). \tag{16}$$

Whereas, the sensible energy effectiveness (ε_s) can be assumed to be a constant for a given heat exchanger over the full range of operating conditions, none of the performance factors $(\varepsilon_s, \varepsilon_l, \text{or } \varepsilon_t)$ can be assumed to be constant for a given energy wheel. They all depend on the operating condition, H*. That is, if one knows the effectivenesses at one operating condition, the effectivenesses at another operating condition may differ depending on the value of H*.

The HVAC designer must know how the energy wheel performance factors change with operating conditions when calculating the annual performance of an energy wheel. This is a more complex calculation than is necessary to calculate the total yearly energy savings for a heat exchanger where only sensible energy changes in the supply air are important. The designer for energy wheel applications needs to couple the variations in each expected operating condition (i.e., $h_{e,i}$ and $h_{s,i}$) with the corresponding change in total energy effectiveness, $\varepsilon_t(H^*)$, and then calculate the total energy change rate, q_t. The value of q_t for each operating condition can then be summed over the year.

Energy Wheels

Energy wheels for HVAC air-to-air energy recovery has its technological origin in desiccant wheel dryers. These dryers date back several decades. The technology for these two applications differ primarily in the operating conditions and wheel speed. Desiccant dryer wheels use high temperature air to regenerate the desiccant coating and wheel speeds of about 1 RPM. Energy wheels in HVAC air-to-air applications use room exhaust air and ambient outside air along with wheel speeds of about 20 RPM. The desiccant coating used for these two different applications is often the same.

More than one type of desiccant coatings may be used, such as molecular sieves and silica gels, but most energy wheel manufacturers use molecular sieve particles which are then bonded onto the metal or plastic film. Molecular sieves are characterized by their strong affinity for water vapor. At one temperature, a molecular sieve will absorb a quantity of water vapor that depends only on the ambient air relative humidity. A change in this relative humidity can change the moisture content significantly. A full range of relative humidities from 0 to 100% gives the range of moisture contents at that temperature. This relationship is called the moisture content isotherm. When the temperature is changed, a different isotherm will occur such that at higher temperatures the moisture content will decrease for the same ambient air relative humidity. Bonding agents are used to bind the molecular sieve particles to the exchanger surfaces.

The construction of the energy wheel is nearly identical to the heat wheel with a metal or plastic core and only differs by the desiccant coating on all the exchange internal surfaces of the wheel core. The diffusion of water vapor onto the desiccant coated surface is very much similar to the diffusion of heat into the metal matrix material of heat wheels and the resistance to these diffusion processes is similar and is dominated by the air resistance. The binding fluids and glues used in the construction of the energy wheel can decrease its performance. Therefore, provided the surfaces are coated with an adequate layer of desiccant coating material to adsorb and desorb water vapor, we may expect the latent (or moisture) effectiveness (ε_l) to have a value which is similar to the sensible effectiveness (ε_s) (e.g. more than 70%). However, the binding fluids and glues used in the construction of the energy wheel can decrease its performance.

In a series of research papers, numerical models have been developed for air-to-air energy wheels (Simonson and Besant, 1997 and Stiesch et at., 1995) and these models have been validated over a wide range of operating conditions (Simonson et al., 1999). Comprehensive dimensionless correlations have been developed for the sensible and latent energy effectivenesses with balanced supply and exhaust flow rates (Simonson and Besant, 1999) and unbalanced supply and exhaust airflow rates (Simonson et al., 2000). These correlations for sensible and latent effectiveness for unbalanced supply and exhaust airflow rates are as follows:

$$\varepsilon_s = \frac{NTU_o}{1+NTU_o}\left(1-\frac{1}{7.5Cr^*_o}\right) - \left[\frac{0.26\left(\frac{Cr^*_o}{Wm^2 Crm^*_o}\right)^{0.28}}{7.2(Cr^*_o)^{1.53}+\frac{210}{(NTU_o)^{2.9}}-5.2} + \frac{0.31\eta}{(NTU_o)^{0.68}}\right] H^* C^{*0.33}, \quad (17)$$

$$\varepsilon_l = \frac{NTU_o}{1+NTU_o}\left(1-\frac{1}{0.54(Cr^*_{mt,o})^{0.86}}\right)\left(1-\frac{1}{(NTU_o)^{0.51}(Cr^*_{mt,o})^{0.54}H^*}\right) \quad (18)$$

where,

$$NTU_o = \frac{1}{(\dot{m}Cp_a)_{min}}\left[\frac{1}{(hA_s)_s}+\frac{1}{(hA_s)_e}\right]^{-1}, \quad (19)$$

$$Cr^*_o = \frac{(MCp)_{m,dry}N}{(\dot{m}Cp_a)_{min}}, \quad (20)$$

$$Crm^*_o = \frac{M_{d,dry}N}{\dot{m}_{min}}, \quad (21)$$

$$Cr^*_{mt,o} = (Crm^*_o)^{0.58}\,Wm^{0.33}\left(\left.\frac{\partial u}{\partial \phi}\right|_{\phi_{ave}}\right)^{0.2}(Cr^*_o)^{1.13}\left(\frac{e^{\left(\frac{1482}{T_{ave}}\right)}}{47.9} - 1.26(\phi_{ave})^{0.5}\right)^{4.66}, \tag{22}$$

$$H^* = 2500\frac{\Delta W}{\Delta T} \quad \text{and} \tag{23}$$

$$\phi_{ave} = \frac{\dot{m}_s\phi_{s,i} + \dot{m}_e\phi_{e,i}}{\dot{m}_s + \dot{m}_e}. \tag{24}$$

where,

A_s	= heat and mass transfer surface area on the supply or exhaust side [m^2]
Cp	= specific heat capacity [J/(kg·K)]
Cp_a	= specific heat capacity of air [J/(kg·K)]
Cr^*	= heat capacity ratio
Cr^*_o	= overall matrix heat capacity ratio
Crm^*_o	= overall matrix moisture capacity ratio
$Cr^*_{mt,o}$	= overall matrix moisture capacity ratio
C^*	= ratio of the minimum to maximum mass flow rates of the air streams
H^*	= operating condition factor that represents the ratio of latent to sensible energy differences between the inlets of the energy wheel
h	= convective heat transfer coefficient [$W/(m^2 \cdot K)$]
M	= total mass of the energy wheel or heat exchanger [kg]
$(MCp)_{m,dry}$	= total heat capacity of the matrix of the dry energy wheel [J/K]
$M_{d,dry}$	= total mass of the dry desiccant [Kg]
$\dot{m}$	= mass flow rate of dry air [kg/s]
N	= rotational speed of the wheel [cycles/s]
NTU_o	= overall number of transfer units
u	= mass fraction of water in the desiccant [kg_w/kg_d]
Wm	= empirical coefficient used in the sorption isotherm describing the maximum moisture capacity of the desiccant [kg_w/kg_d]
ΔT	= temperature difference between supply and exhaust inlet conditions (C)
ΔW	= humidity ratio difference between supply and exhaust inlet conditions (kg_w/kg_a)

ϕ = relative humidity

ϕ_{ave} = average relative humidity

η = fraction of the phase change energy that is delivered directly to the air

The part-load control of energy wheels is inherently more complex than air-to-air heat exchangers because, as noted above, the effectivenesses are not constant and the designer has two air properties (e.g. temperature and humidity ratio) to consider at each operating condition. There can be operating conditions when it is important to decrease the supply inlet air temperature into a space but there is no need to decrease the humidity or vice versa. The control strategy to manage such conflicts may be dictated by mostly comfort conditions in the space but partly by operating cost factors.

It was noted previously that wheel speed control was the primary means of controlling the heat rate in rotary heat wheels. Energy wheel performance is much more complex at low wheel speeds making it more difficult to achieve the desired operating conditions (Simonson et al., 2000). As a consequence, face-and-bypass control should be used where a wide range of control is required for energy wheels.

Frost Control in Air-to-Air Exchangers

At air supply inlet temperatures below freezing, some parts or all the exchanger surfaces may drop below freezing. When this occurs in a heat recuperator, any water vapor condensation from the exhaust airflow may freeze and, when the surface is well below freezing, frost may start to grow in the exhaust airflow passages. For heat exchangers, the exact outdoor air temperatures when frosting will start to form depends on the temperature and humidity of the exhaust air and the type and performance of the heat exchanger. Typically, air supply temperatures less than about -10°C (14°F) often start to cause frosting problems for cross-flow plate heat exchangers, especially along one edge and corner of the cross-flow plates (Phillips et al., 1989). Most manufacturers will start to control for frost for temperatures less than –5°C (23°F). Rotary heat wheels are cyclically exposed to warm humid and cold supply air so frosting can occur anywhere on the wheel internal surfaces and it usually does not start to be a problem until slightly colder supply air inlet temperatures (i.e. -15 or -20°C (5 or -4°F)). When it does start to occur, blockage can be very rapid and the wheel may freeze to the frame. Heat-pipe and run-around exchangers are also counterflow devices, so they have a somewhat similar frost threshold condition as heat wheels but the consequences of frost growth are less severe because there is no rotating wheel. For energy wheels, research work on frost growth is only now being done. It appears that frosting problems may not occur within energy wheels until there is slightly colder supply air inlet temperatures than heat wheels (Simonson and Besant, 1998).

Regardless of the type of exchanger, there will be supply air inlet conditions when frost problems may occur. When frosting problems become significant (i.e., a large fraction of frost blockage and increase in air pressure drop) then a defrost cycle, supply air preheating, exhaust air preheating, or bypassing more of the supply air around the heat exchangers must be used. For heat wheels, wheel speed reduction may be used, but bypass appears to be best when very cold operating conditions are encountered. Otherwise the frost may grow to a point where the exhaust airflow rate will drop to a small fraction of the design flow rate for heat recuperators and for rotary wheels. Both supply and exhaust flows will diminish or the wheel may cease to rotate because it is frozen to the frame on the cold supply side. Thus it is more important to avoid freeze up problems for wheels than it is for other types of heat recuperators.

Preliminary Selection of the Type of Heat or Energy Recovery Device

The forgoing sections outlined some of the characteristics of several air-to-air heat exchangers and heat and moisture exchangers (i.e., energy wheels). Now we need to identify the considerations for selection of a type of exchanger and then consider how they should be sized.

The selection of a type of exchanger depends on: (1) their capital cost and (2) the climate, (3) HVAC application design, size and operating conditions, (4) auxiliary energy costs, (5) exchanger performance, and (6) some other usually small factors such as maintenance costs, space costs, and tax incentives and/or government regulations. A crucial question about climate is the cooling load compared to the heating load, especially the load for dehumidification of ambient supply air. Climates with relatively small cooling loads and large heating loads are favored for the selection of some type of heat exchanger and when this is not true the energy wheel is preferred.

When selecting a heat or energy recovery device, the indoor and outdoor design conditions and yearly weather data are important because the temperature and enthalpy differences between the outdoor and indoor air govern the energy recovery rate. Figure 3 presents the energy recovery rate per unit supply air flow rate for a sensible heat exchanger and an energy wheel as a function of the temperature difference between the outdoor (supply) and indoor (exhaust) temperatures, assuming all effectiveness values are 70%. Since the energy recovery of an energy wheel depends on the humidity level, one winter and two summer cases are included in Figure 3. During winter, the indoor conditions are 22°C (72°F) and 30% RH and the outdoor humidity is 80% RH. During summer, the indoor conditions are 24°C (75°F) and 50% RH and the outdoor humidity is 40% RH or 60% RH as marked in Figure 3. With the data in Figure 3 and the known supply airflow rate, the HVAC designer can estimate the expected energy recovery rates at various temperatures and relative humidities. Figure 3 shows that during the summer when the temperature difference between the indoors and outdoors is 10°C (18°F) and the outdoor humidity is 60% RH, 31 W of energy can be recovered per L/s of supply air flow. If the design requires 1,000 L/s [2,000 cfm] of outdoor ventilation air, 31 kW [105 kBtu/h] will be recovered at these conditions. In other words, the load on the cooling equipment will be reduced by 31 kW [105 kBtu/h or 8.7 tons]. When using the design conditions in conjunction with Figure 3, the designer can estimate how much the capacity of the heating and cooling equipment can be decreased when applying energy recovery devices.

Applying Figure 3 for the ventilation requirements of an entire country highlights the enormous energy and power generation savings possible when air-to-air heat/energy recovery is universally applied. During hot weather, a country or state may find that additional electrical generation is needed to meet the electrical demand. For a population of 1 million spending 90% of their time indoors and requiring 7.5 L/s (15 cfm) per person of outdoor ventilation air, 55 MW of energy can be conserved by universally applying sensible heat exchangers when the outdoor temperature is 10°C (18°F) warmer than the indoor temperature. If energy wheels are universally applied, 110 MW and 210 MW can be recovered at outdoor humidities of 40% RH and 60% RH respectively. This recovered energy can reduce the needed electrical production capacity in the country or state. Because of these large capacity reductions and with added pressure to curb green house gas emissions, it is not surprising that some countries have mandated energy recovery from ventilation air for certain installations.

Figure 3 shows significant energy transfer rates for both sensible and total energy recovery devices and highlights the higher energy transfer rates for energy wheels compared to sensible heat exchangers during the cooling season. For example, when ΔT = 10°C (18°F), energy wheels recover from 2 (40% RH) to 4 (60% RH) times as much energy as sensible heat exchangers. During the winter, an energy wheel recovers more energy than a sensible heat exchanger for ΔT > 15°C (27°F). When ΔT < 15°C (27°F) in large buildings, the sensible and total energy recovery devices will likely need to be controlled to prevent overheating. The energy recovery device, will need to be controlled to prevent overhumidifying as well.

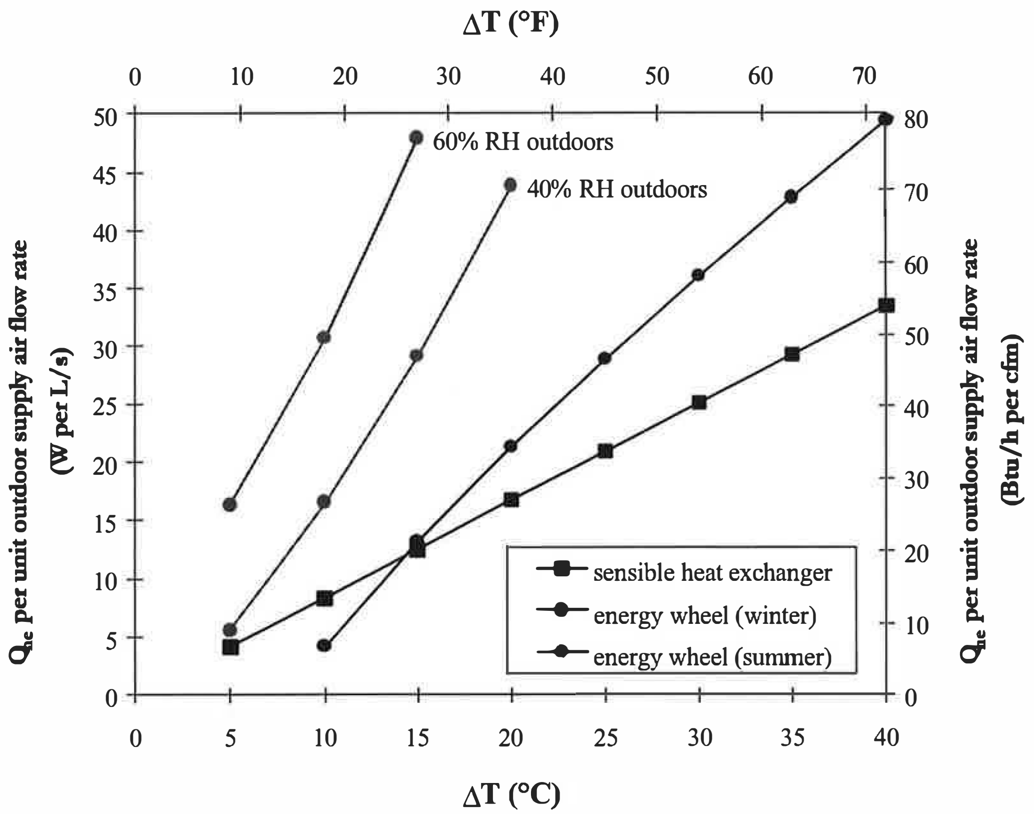


Figure 3. *Energy recovery rate per unit outdoor supply airflow rate for sensible and total energy exchangers under different operating conditions.*

The choice between a sensible and total heat exchanger for various climates can be clarified by plotting the design conditions on the psychrometric chart as in Figure 4. Figure 4 contains the 1% summer design WB/MDB and DB/MWB conditions for a few locations (ASHRAE, 1997) and shows that the region where a total energy exchanger is favored includes many of the shown summer design conditions. A sensible heat exchanger with evaporative cooler is also favored in some conditions, while a sensible heat exchange without evaporative cooler is favored in only dry conditions. (In fact an evaporative cooler can always increase the energy transfer of a sensible heat exchanger during the cooling season.) From the results in Figure 4, an energy wheel is favored generally if the humidity ratio is greater than 0.012 kg/kg (0.012 lb/lb). Recommended regions for the sensible and total energy exchangers are not shown for the heating season in Figure 4 because the energy implication of moisture transfer is only important during the heating season if the building is humidified. If the building is humidified, a total energy wheel is favored in nearly all locations. Even when a building has no humidification control, an energy wheel can provide slightly higher indoor humidity which can improve comfort and health (Green, 1985, and Toftum and Fenger, 1999).

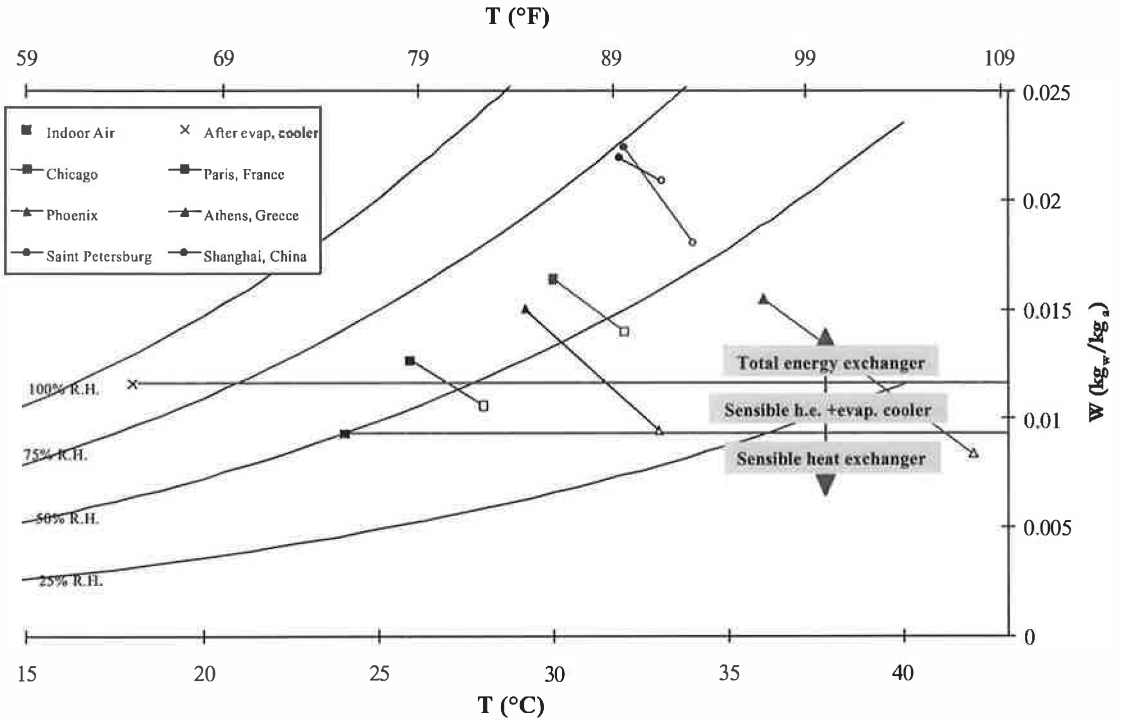


Figure 4. *Psychrometric chart showing the regions were a sensible heat exchanger (with and without an evaporative cooler) and an energy wheel will provide the greatest energy recovery during cooling. For comparison, the 1% summer design conditions (WB/MDB - closed symbols and DB/MWB - open symbols) are included for some cities.*

Although Figures 3 and 4 help to quantify the possible energy transfer rates (or capacity reductions) and the appropriate choice between a sensible and total energy recovery device at design conditions, off-design conditions should be considered as well when selecting an energy recovery device. Since the temperature and humidity difference across the supply and exhaust ducts is a time variable, the designer must use detailed hourly weather data and hours of system operation to obtain more detailed cost savings and to make a more accurate choice between sensible and total energy recovery devices. Building energy simulation with hourly weather data or a frequency distribution of temperatures and humidity throughout the year can provide valuable insight for the designer when resolving these issues.

Of course, Figures 3 and 4 are a simplification of all the important design factors in selecting a type of device because each HVAC design problem often has some unique constraints or requirements. For example, when the system includes an economizer, the heat or energy recovery device may not be used during moderate outdoor temperatures. Then the ability to control the heat and moisture transfer rates is essential.

Airflow rate requirements in an HVAC system are always important because larger systems require better designs and lower capital and operating costs per unit mass flow rate of ventilation air. Air-to-air exchangers should be considered for all large ventilation flow rates. This design consideration will become clear when total costs and savings are considered.

The HVAC design requirements for air supply temperature and humidity controls, air flow pressure loss limits, and cross contamination limits may eliminate certain air-to-air exchanger selections. Buildings with strict humidity as well as temperature controls are favored for energy wheels and those with only temperature controls are favored for heat exchangers. Strict airflow pressure loss requirements may eliminate some plate heat exchangers and favor wheels. Zero cross contamination requirements will eliminate wheels. Very cold air supply inlet temperatures will favor the most reliable frost control strategy for each type of device.

Auxiliary energy costs for heating and cooling depend first on utility charges for natural gas and electricity and then on the performance and cost factors for the heating and cooling plants. The final auxiliary energy costs are not just operating costs because they have significant capital cost implications for the size of the auxiliary heating and cooling plants used in buildings. These capital costs for the size of auxiliary heating and cooling plants can exceed the capital cost of air-to-air heat and energy recovery equipment. When this occurs, air-to-air heat and energy recovery systems should always be installed if the least capital and/or operating system cost is the design objective. Considering only the utility costs for natural gas and electricity, high electrical energy costs favor the selection of energy wheels when cooling loads are significant. High utility energy costs favor design optimization of the air-to-air heat or energy recovery systems.

Preliminary Cost Considerations

It was suggested earlier that air-to-air heat exchangers usually have payback durations of several years for new designs (e.g. 1 to 5 years) without considering the cost savings that might be realized by downsizing the capacity of boilers or chillers. These payback periods are usually calculated without considering the operational benefit of better indoor air quality due to higher fractions of outdoor air supply to recirculated air supply. Energy wheels, on the other hand, may have a much more rapid payback in new applications, especially in warm humid climates where they can offset a significant fraction of the chiller load and they operate for all weather conditions except perhaps when the economizer is handling all the ventilation air. When all the factors are considered, HVAC design with energy wheels will likely prove to be the lowest capital cost design for many applications - then operating cost savings from the energy wheel are an added benefit. The data in Figure 3 and Figure 4 further demonstrates this. Since the cost of air-to-air heat/energy recovery devices are about \$5 per L/s (\$2.50 per cfm) and the cost of heating and cooling equipment are typically \$100 to \$200 per kW (\$350 to \$700 per Btu/h) respectively, the results in Figure 3 show that, for many climates, the payback for heat/energy recovery devices will be immediate for new buildings. For example when using an energy wheel in Chicago (WB/MDB), the heating and cooling capacities can be reduced by 49 W and 19 W per L/s of ventilation air flow, which would reduce the capital investment for heating by \$5 per L/s and cooling by \$4 per L/s. These capital savings are similar to the price of the energy exchanger and therefore the capital cost of the system is expected to be about the same with and without the energy exchanger. In this case and many others, the energy savings that accrue during operation will result from essentially no investment. Retrofitting air-to-air heat exchangers or energy wheels into existing HVAC facilities is always more expensive. Each one usually has some unique constraints that must be accounted for in the retrofit costs.

Size Optimization and Least Life-Cycle Costs

In the past, HVAC designs have tended to be adequate or conservative in that they satisfy comfort conditions for building occupants for the 1% and 99% temperature and humidity design conditions for a given climate. Often heating and cooling plants are oversized to allow a rapid indoor temperature change but mostly to avoid the large cost of undersized equipment when loads are somewhat uncertain.

Air-to-air heat/energy recovery equipment sizing is different because very often this equipment is only considered for the reduction of heating and/or cooling loads not for replacing some capacity in the heating and cooling plants. When the design problem for air-to-air recovery does not involve any consideration of plant capacity, then any size of air-to-air exchanger will have the potential to reduce some utility costs. The problem is -once the size or performance of an air-to-air device has been selected; it is unlikely to be replaced for one with a different size or performance. Undersizing the selection may mean a very large loss in utility cost savings integrated over the life-cycle of the unit. Oversizing implies a slightly higher capital cost for the unit. This is a life-cycle cost optimization problem, not a maximum load problem for the selection of capacity.

The total life-cycle cost of an air-to-air heat/energy recovery device is given by

$$C_T = C_C + C_O \tag{25}$$

where C_C is the capital cost and C_O is the life-cycle cost of auxiliary energy used over the total life cycle of the device. The capital cost depends mostly on the size, mass, construction and type of materials used. The larger the heat/energy exchange area in the device, the larger this cost. The auxiliary energy rate, q_{aux}, on the other hand, is that energy demanded by the HVAC system to meet the load without the heat/energy recovery device. When an installed air-to-air device operates, it transfers q_t total energy rate giving the net operating cost for the year with t_{year} hours of operation.

$$C_O = P_{wef} C_{e,i} \int_{O}^{t_{year}} [q_{aux} - q_t]^{+} \, dt \tag{26}$$

where,

P_{wef} = the present worth escalation factor over the life-cycle of the unit (e.g. 10 to 20),
$C_{e,i}$ = the cost of energy for air heating $(i = h)$ and cooling $(i = c)$,
q_{aux} = the auxiliary energy rate with no exchanger,
q_t = the air-to-air exchanger energy rate which is controlled to never exceed q_{aux}.

The integral $\int_{O}^{t_{year}} [q_{aux} - q_t]^{+} dt$ is the total auxiliary energy required over the year which only has a positive value when the exchanger energy rate is less than the auxiliary energy rate (i.e., $q_t < q_{aux}$). For heating conditions, this energy demand operating condition will occur only during the coldest weather, but for cooling conditions, auxiliary cooling energy will be required for all the conditions that cannot be fully satisfied by an economizer airflow control.

The exchanger design problem is to size the device so that the total cost C_T is a minimum. Here the capital cost, C_C, increases directly with the exchanger surface area, A, and the operating cost, C_O, decreases with the exchanger effectiveness and this exchanger effectiveness increases with the exchanger surface area. At some exchanger surface area, A_{opt}, the total cost will be a minimum.

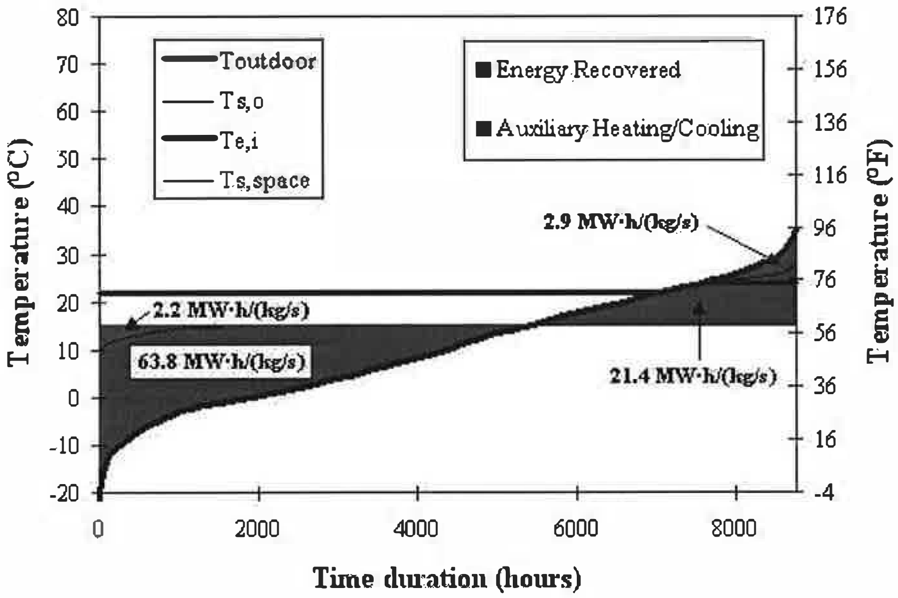


Figure 5. *Distribution of temperatures during the year showing the energy recovered with a sensible heat exchanger and the auxiliary heating and cooling required to meet the sensible load of a building with 100% outdoor supply air in Chicago, Illinois.*

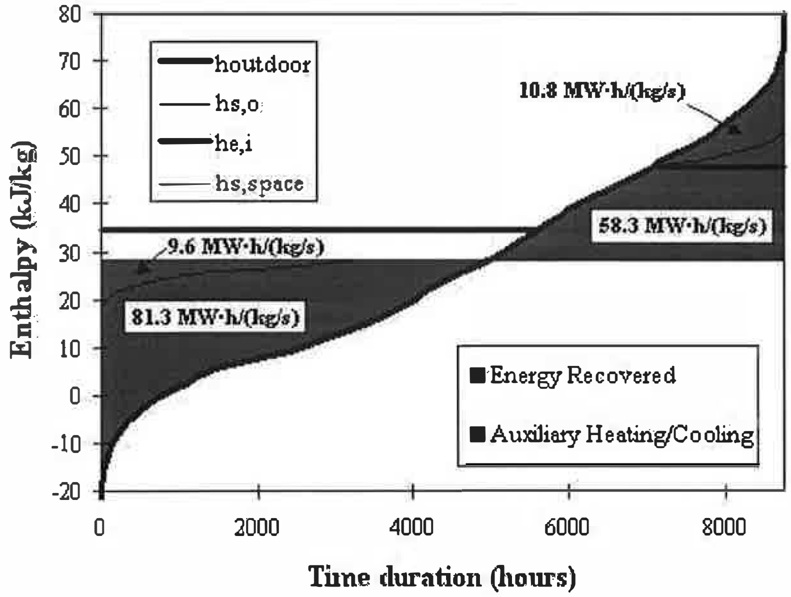


Figure 6. *Distribution of outdoor enthalpy during the year showing the energy recovered with an energy wheel and the auxiliary heating and cooling required to meet the load of a building with 100% outdoor supply air in Chicago, Illinois.*

This design problem can be better understood using actual weather data for one city. Figures 5 and 6 present all the weather data for temperature and enthalpy for Chicago IL. These data are not presented as a continuous time variation of air properties; rather, they are presented so that temperature in Figure 5 and enthalpy in Figure 6 monotonically increase with time duration, t. At $t = 0$, the temperature is the lowest in Figure 5 and it increases with t until 8,760 hours for a system that operates 24 hours a day , 365 days per year. Figure 6 shows a similar pattern for enthalpy. The indoor temperature and humidity are assumed to be 22°C (72°F) and 30% RH in winter and 24°C (75°F) and 50% RH in summer. These constant indoor operating conditions simplify this illustration.

The green shaded area in Figure 5 represents energy saved by a heat exchanger over one year and the red area represents the auxiliary heating or cooling energy required to condition the outdoor air to 15°C (59°F). In this figure, it is assumed that the mass flow rate of air is 1 kg/s (2.2 lbs/s) and the heat exchanger sensible effectiveness is 70%. It is clear from the areas of the graph in Figure 5 that an air-to-air heat exchanger can displace a large fraction of the total annual air heating load energy for ventilation air but only a small fraction of the annual air sensible cooling load energy. Here, only sensible energy changes are considered for the supply and exhaust airflows. As well, the fraction of time when the heat exchanger satisfies the total air heating load is large while the auxiliary cooling load is always present for ambient outdoor air temperatures greater than 15°C (59°F). For a building with humidity control, these auxiliary energy loads will be higher. The heat exchanger only provides a cooling benefit when the ambient air temperature exceeds the exhaust air temperature 24°C (75°F). Both the peak auxiliary heating and cooling loads for the supply air at $t = 0$ and $t = 8{,}760$ hours are substantially reduced by using the air-to-air heat exchanger.

For buildings which control humidity as well as temperature, the sensible energy analysis in Figure 5 omits the latent energy change requirements. The total energy requirements for air conditioning are more accurately shown by the enthalpy versus time duration curve in Figure 6. In Figure 6, the supply air enthalpy is assumed to be 28 kJ/kg, the exhaust air enthalpy is 35 kJ/kg, and the enthalpy threshold for energy recovery device (i.e., a rotary energy wheel) is 48 kJ/kg. Again, this figure shows energy areas for 1 kg/s (2.2 lbs/s) of supply airflow. These areas are much larger than for the sensible energy in Figure 5. The energy recovery areas for both heating and cooling for an energy recovery device of 70% total effectiveness are 28% and 270% larger than the sensible energy savings for heating and cooling in Figure 5. Although Figure 6 implies an increased fraction of time when there will be auxiliary heating load, in practice this will not occur for a building that relaxes its humidification requirements in winter. Then the duration of time for auxiliary heating will be more like Figure 5. Likewise, the peak auxiliary heating load at $t = 0$ in Figure 6 may be less than that implied when the humidity requirements at low outdoor air temperatures are relaxed. Both the peak auxiliary heating and cooling loads will be substantially reduced by using an energy wheel.

Now returning to the least life-cycle cost problem for sizing the air-to-air exchanger we can consider it to be one where the designer uses the red areas in Figure 5 and 6 to represent the auxiliary annual energy requirements in the equation for the present worth of this auxiliary energy, C_O. Then the designer sizes the exchanger so that the total cost ($C_T = C_C + C_O$) is a minimum. The assumptions behind Figure 5 and 6 were $\varepsilon_s = 70\%$ and $\varepsilon_t = 70\%$ respectively so these graphs cannot be used directly. Instead, the designer should use analytic expressions to represent these energy saving and auxiliary energy in Figure 5 and 6 for exchangers of any specified effectiveness. For energy wheels, this calculation should be done analytically (i.e., on a spreadsheet), not graphically, because the total effectiveness is not constant for all operating conditions.

Summary and Conclusion

In this paper, the design of HVAC systems employing air-to-air heat and /or moisture exchange between supply and exhaust airflows has been discussed. Factors that must be considered are: (1) capital costs, (2) climate, (3) HVAC system design, (4) auxiliary energy costs and (5) recovery system performance.

It is shown that heat exchangers are favored for heat recovery for many HVAC applications in cold dry climates but energy wheels are preferred for warm humid climates where supply air moisture removal is important. Furthermore, special HVAC design constraints may direct the selection towards certain types of devices.

Air-to-air heat and/or moisture recovery devices should be designed for the least life-cycle cost and trade-off costs should be considered for auxiliary HVAC heating and cooling plant capacity. When such trade-offs are included, air-to-air heat and/or moisture exchangers will likely be the least cost alternative for all large ventilation airflow rate systems. The least life-cycle cost exchanger design will maximize the annual savings for all air-to-air systems.

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