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Air-To-Air Energy Recovery

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Indoor air quality, ventilation airflow rates and HVAC capital and operating costs usually are closely related. Auxiliary energy loads for supply air heating or cooling sometimes can substantially reduce the need for dehumidifying or humidifying by using air-to-air heat or energy exchangers. The most important factors in reducing HVAC capital and operating costs are ventilation airflow requirements, climatic parameters, exchanger recovery system performance factors and duration of operation.

Several types of air-to-air exchanger recovery devices are commercially available. Most HVAC design engineers want to know: What are the types of heat or energy recovery devices and how do I go about the task of selecting a type of heat or energy recovery device? How should a heat or energy recovery device be integrated into an HVAC system? What can I expect for performance factors? What size should it be? How quickly may I expect a payback? This article provides guidance but not exact answers for particular design problems because that requires detailed calculations using all the data for airflow, climate, device performance and duration of operation.

Airflow Configuration & Devices

Over many decades, plate type, regenerative heat wheels, runaround and heat-pipe heat exchangers were developed and used to transfer sensible energy for a variety of air-to-air applications.[1] These nearly passive devices, with airflows as shown in *Figure 1*, have good performance and excellent maintenance and reliability characteristics. However, they may require several years to pay down the initial investment for typical HVAC applications because they may be used only when the ambient air temperatures drop below the supply air setpoint, e.g., 15°C (59°F).

Configured as in *Figure 1*, the units may not be cost effective in warm climates. Although, when used with evaporative coolers in dry climates[2] or as supply air pre-coolers and post-reheaters with auxiliary air chillers as in *Figure 2a* and *b*, they can be cost effective in warm climates.[3]

The system in *Figure 2a* allows for complete control of the supply air condition before the air enters the space whereas the system in *Figure 2b*, which does not include any recirculated air, relies on some auxiliary heating directly in the building space during cold weather. This system in *Figure 2b*, which uses an energy wheel along with a sensible heat exchanger in the exhaust airstream, is currently used in some rooftop units.

The energy wheel (also called the enthalpy wheel) exchanges both heat and moisture between airstreams. Its development has substantially altered the economic tradeoffs for heat and moisture exchange. This is because the same device and airflow configuration can be used for both cold and warm ambient air temperatures. Thus, the duration of operation for energy wheels usually is much longer than heat exchangers that transfer only sensible energy between the supply and exhaust airflows.

With a desiccant coating on the exchanger surfaces and a configuration similar to the heat wheel, energy wheels transfer both heat and water vapor and cause the supply air temperature and absolute humidity to change from the ambient air condition toward that of the exhaust air. That is, the supply air enthalpy downstream of the energy wheel can closely track the exhaust air enthalpy for all weather conditions so it only may be bypassed during moderate outside air temperatures when the economizer is used.

Here, we have used the term energy (or enthalpy) wheel even though enthalpy is not exchanged and energy is always conserved for all types of exchangers. Furthermore, the outlet air properties of the energy wheel need not lie on the straight line connecting the inlet air properties on the psychrometric chart. This implies that the moisture adsorption and desorption characteristics of a rotating wheel coated with a desiccant differ at different temperatures. Unlike sensible heat exchangers, characterization of en-

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ergy wheels is more complex because both heat and mass transfer occur and they are coupled.

Performance Factors

Aside from the case when condensation or frosting or other property changes are significant, the effectiveness of a heat exchanger will be independent of the inlet temperatures. Thus, the heat exchange rate for the air-to-air heat exchanger will equal the product of the effectiveness for sensible heat transfer (ε_s) and the maximum possible heat rate when the heat exchange surface area is infinite. The equation for this heat exchange rate is:

$$Q_{he}=C_s(T_{s,i}-T_{s,o})=\mathring{a}_s C_m(T_{s,i}-T_{e,i})=\mathring{a}_s C_m \Delta T \quad (1)$$

where,

C_s = the thermal capacitance rate for the supply air (i.e., mass flow rate times specific heat of air)
C_m = minimum value of C for the supply and exhaust
T_s = supply air temperature, i = inlet, o = outlet
T_e = exhaust air temperature, i = inlet

This means that by obtaining the effectiveness (ε_s) from the heat exchanger manufacturer and knowing the inlet temperature conditions and the airflow rates, the heat rate can be calculated for that operating condition. In principle, this design process is simple because the designer only needs the data for the operating conditions.

On the other hand, energy wheels are not so simple because two independent effectivenesses are needed to calculate the performance: one for the sensible energy transfer (ε_s) and another for moisture transfer or the latent energy transfer (ε_l). In this case, the total energy transfer rate is calculated as,

$$Q_{he}=\dot{m}_s(h_{s,i}-h_{s,o})=\mathring{a}_t\dot{m}_m(h_{s,i}-h_{e,i})=\mathring{a}_t\dot{m}_m\Delta h \quad (2)$$

where,

$\dot{m}$ = the mass flow rate of air ($\dot{m}_s$ = supply and $\dot{m}_m$ = minimum)
h = enthalpy
ε_t = total energy or enthalpy effectiveness

The total energy or enthalpy effectiveness (ε_t) can be calculated from the other two and an operating condition factor using

$$\varepsilon_t=\frac{\varepsilon_s+H^*\varepsilon_l}{1+H^*} \quad (3)$$

where, the operating condition factor (H*) is the latent to sensible heat ratio for the inlet conditions and can be calculated as

$$H^*=\frac{1}{SHR}-1=K\frac{\Delta W}{\Delta T}=K\frac{W_{s,i}-W_{e,i}}{T_{s,i}-T_{e,i}} \quad (4)$$

where, K = 2500°C/(kg/kg) (4,500°F [lb/lb]).

Another complicating factor is that all three effectivenesses vary with the operating condition factor. They are not constant for a given supply and exhaust flow rate. Rather, all three effectivenesses vary with the operating condition sensible to latent heat ratio. As a first approximation, the effectivenesses may be assumed constant, but for most applications the change in effectiveness due to changes in outdoor temperature and humidity will be on the order of 5% to 10%. As the outdoor temperature and humidity increase (H^* increases), the effectiveness values typically decrease.

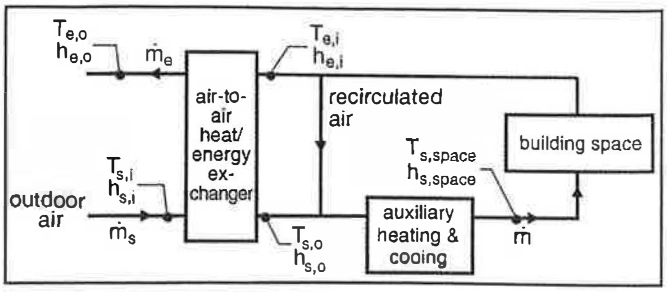

Figure 1: Schematic of a HVAC system with air-to-air heat/energy recovery.

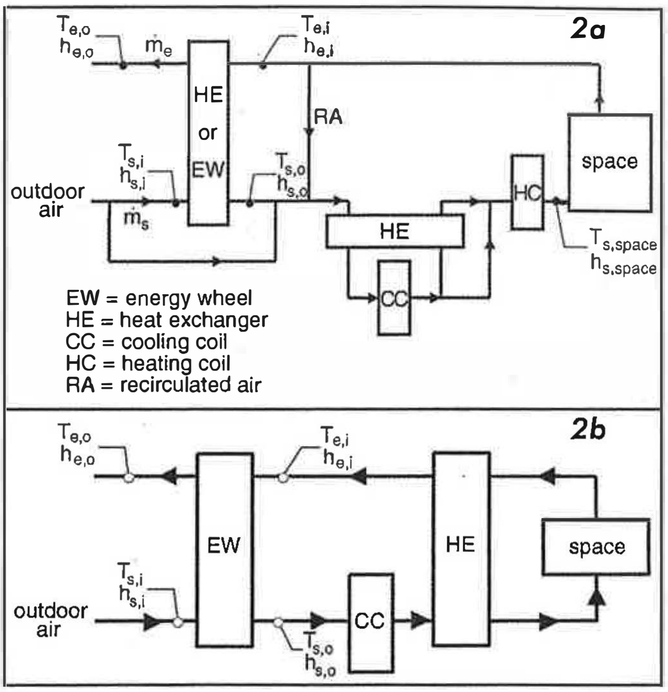

Figure 2: Schematics of HVAC systems with two air-to-air heat/energy recovery devices. 2a: Site built system with some recirculated air. 2b: components in a packaged air-conditioning and recovery unit.

These calculations are manageable on a spreadsheet for a limited range of desiccants but are more complex for each energy rate (sensible, latent and total energy) than is the case for sensible heat exchangers. The reader is referred to the recent literature for all the details on this calculation procedure.[4,5,6]

Selecting a Heat or Energy Recovery Device

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 (Equations 1 and 2). *Figure 3* presents the energy recovery rate per unit supply airflow rate for a sensible heat exchanger and an energy wheel as a function of the temperature difference

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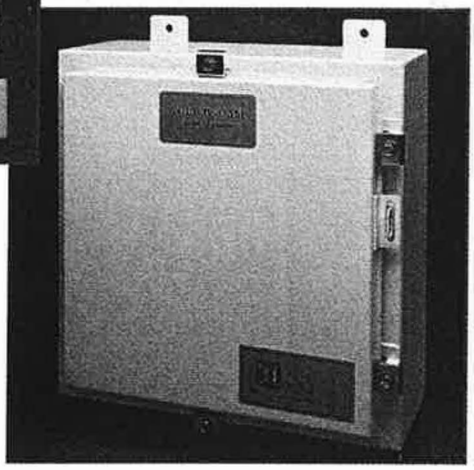

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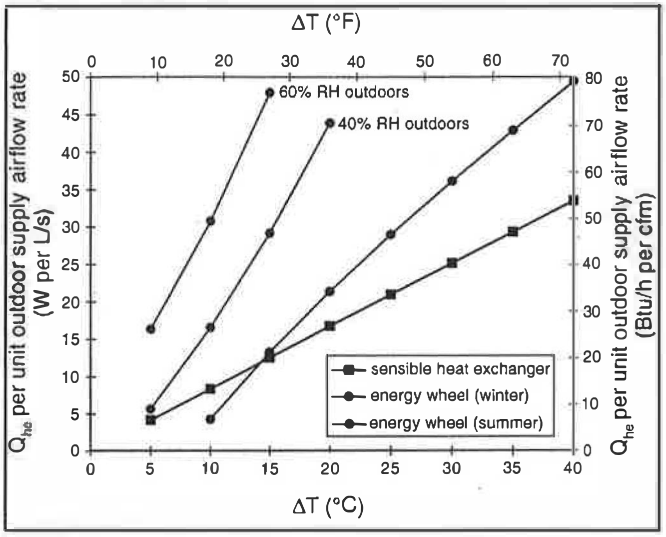

Figure 3: Energy recovery rate per unit outdoor supply airflow rate for sensible and total energy exchangers under different operating conditions.

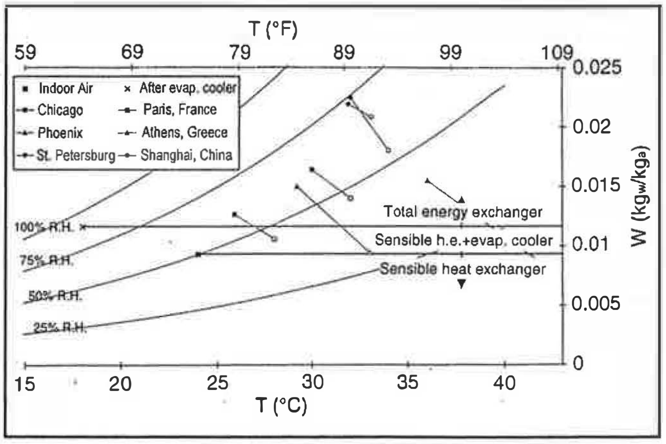

Figure 4: Psychrometric chart showing the regions where a sensible heat exchanger (with and without an evaporative cooler) and an energy wheel will provide the greatest energy recovery during cooling. For comparison, the 1% summer design conditions (WB/MDB: closed symbols and DB/MWB: open symbols) are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

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 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 airflow. If the design requires 1000 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.

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 $\Delta T = 10°C$ (18°F), energy wheels recover from two (40% RH) to four (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 $\Delta T > 15°C$ (27°F). When $\Delta T < 15°C$ (27°F) in large buildings, the sensible and total energy recovery devices likely will need to be controlled to prevent overheating. The energy recovery device may need to be controlled to prevent overhumidifying as well.

The choice between a sensible and total heat exchange 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[7] and shows that the region where a total energy exchanger is favored includes many typical summer design conditions. A sensible heat exchanger with evaporative cooler also is favored in some conditions, while a sensible heat exchanger 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.[8,9]

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 when selecting an energy recovery device. Since the temperature and humidity difference across the supply and exhaust ducts is a time variable, the designer should 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.[10] Then, the ability to control the heat and moisture transfer rates is essential. Each type of device uses a unique control scheme (e.g., plates use by-pass; heat wheels use wheel speed control or by-pass; runarounds use pumping rate bypass; and heat pipes use tilt control or by-pass). During extremely cold outdoor air temperatures, frost will cause blockage of the airflow passages—so it must be controlled.

Each heat/energy device manufacturer should recommend a frost control method for their device, but frosting can be expected for typical applications when the outdoor temperature is below about −10°C (≈15°F) for cross-flow plate heat exchangers and about −20°C (≈−5°F) for other devices operated with counter flow air directions. In some applications, the HVAC design may not permit the exhaust and supply air ducts to be side-by-side or, perhaps, the system must be designed with heat recovered from several exhaust stacks while one or more supply air ducts are not adjacent. In this case, a runaround heat recovery system likely is the only practical selection.

For some applications, toxic or hazardous contaminants in the exhaust gases will suggest that the heat or energy wheels may not be a good selection even when a purge section is used

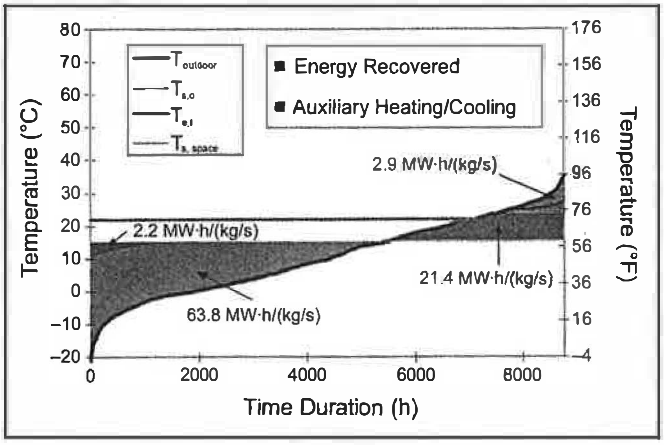

Figure 5: Distribution of temperatures during the year shows the energy recovered with a sensible heat exchanger and the auxiliary heating and cooling required to meet the sensible load of a building with 100% outdoor supply air in Chicago.

to reduce cross contamination to less than 1% of the flow rate. In other applications, large pressure differences between the exhaust and supply air ducts will infer that heat or energy wheels will experience excessive leakage and some plate type heat exchangers will mechanically deform causing excessive throttling of the airflow on the low pressure side. This list is only a guide to some of the possible constraints. The reader is referred to the literature for more examples.[1]

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., one to five years) without considering the cost savings that might be realized by downsizing the capacity of boilers or chillers. These payback periods usually are calculated without considering the operational benefit of better indoor air quality due to higher amounts of outdoor air supply compared to recirculated air.

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. 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 likely will prove to be the lowest capital cost design for many applications. Operating cost savings from the energy wheel are an added benefit. The data in *Figures 3* and *4* further demonstrates this.

The cost of air-to-air heat/energy recovery devices is about \$5 per L/s (\$2.50 per cfm) and the cost of heating and cooling equipment typically are \$100 to \$350 per kW (\$30 to \$105 per kBtu/h or \$1200 per ton of cooling), the results for many climates in *Figure 3* show that 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 airflow, which would reduce the capital investment for heating by \$5 per L/s and cooling by \$66 per L/s of ventilation air. 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, when the net capital cost of air-to-air heat and moisture exchanger is the least first-cost design, the energy savings that accrue during operation will result without any extra investment. Retrofitting air-to-air heat exchangers or energy wheels into existing HVAC facilities may be more expensive. Each one usually has some unique constraints that must be accounted for in the retrofit costs.

Many HVAC designs are functional rather than optimal designs. That is, the designer tries to ensure that each design satisfies the required design objectives or peak loads, is reliable, and once operational, requires little or no maintenance. As a consequence, fans, pumps, boilers, and chillers will be oversized while ducting may be undersized and/or ducting flow rates mal-distributed. This strategy minimizes the risk of system failure and possible lawsuits because no equipment replacements will be required. However, flow rate adjustments will be needed.

Size Optimization and Least Life-Cycle Cost

A new design strategy is needed for air-to-air heat or energy recovery devices because there is usually *no risk* of the HVAC system failure to meet the peak load due to undersizing or oversizing the heat or energy recovery device. A wide range of sizes can be functional and provide some savings. Since oversized devices result in higher capital costs, air-to-air recovery devices have tended to be undersized, which means lost operating cost savings. These lost savings, when integrated over the HVAC system life cycle, often exceed the capital cost of the device, sometimes by an order of magnitude. This means that some attempt at life-cycle cost optimization is essential for the design of HVAC systems using air-to-air heat or energy recovery devices. Taking all the cost factors into consideration for the selection of an air-to-air recovery device can be complex. First, we will consider a simple procedure for sizing an air-to-air heat recovery device for the least life-cycle cost. Later, some of the difficulties that may be experienced in more complex optimization problems will be discussed.

The total life-cycle cost of an HVAC air-to-air heat recovery device is

$$C_T = C_c + C_o \tag{5}$$

where, C_c is the capital or first cost of the device installed and made operational and C_o is the life-cycle cost of auxiliary energy used over the total life of the device. The capital cost of the heat exchanger depends on, among other factors, the effectiveness, heat transfer area and mass, and type of materials used in construction. Typically, the higher the effectiveness, heat transfer area and mass, the higher the cost of the heat exchanger. The auxiliary energy is the energy consumption of the HVAC system, including heating, cooling and fan energy. Increasing the effectiveness of the heat exchanger generally decreases the auxiliary energy consumption. However, at some point the decrease in heating and cooling energy will be offset by an increase in fan energy. In this simple analysis, C_o is assumed to include only the cost of air heating and cooling and is given by

$$C_o = P_{wef}\, C_{e,i} \int_0^{t_{year}} \left| Q_{aux} - Q_{he} \right| dt \tag{3}$$

where,

P_{wef} is the present worth escalation factor over the life cycle (e.g., 10 to 20)

$C_{e,i}$ is the cost of thermal energy for air heating ($i = h$) and cooling ($i = c$)

Q_{aux} is the thermal energy rate required when there is no heat exchanger present

Q_{he} is the energy transfer rate provided by the heat exchanger where $Q_{he} \leq Q_{max}$

t_{year} is the total number of hours of operation per year (e.g., 8,760 hours if airflows are steady 24 hours/day, 365 days per year).

The integral $\int_0^{t_{year}} \left| Q_{aux} - Q_{he} \right| dt$ is the energy required for auxiliary heating and cooling and is represented by the red shad-

ing in *Figures 5* and *6*, which contain weather data for Chicago. The outdoor ambient air temperature and enthalpy in *Figures 5* and *6* are arranged so that they monotonically increase with increasing duration, *t*, starting at the lowest ambient temperature and enthalpy at $t = 0$ and increasing to the highest temperature and enthalpy at t = 8,760 hours. The indoor temperature is assumed to be 22°C (72°F) and 30% RH during the winter and 24°C (75°F) and 50% RH during the summer. The temperature of the supply air to the building space is assumed to be constant at 15°C (59°F) and 50% RH year around and 100% outdoor supply air is assumed. In actual buildings, the indoor and supply air conditions will vary throughout the year, but here they are assumed to be constant to simplify this analysis.

Sensible Heat Exchangers

The green shaded area in *Figure 5* represents the energy recovered by a sensible heat exchanger and the red area represents the auxiliary heating or cooling required for conditioning the outdoor supply air to 15°C (59°F). These shaded areas can be converted from °C·h to kW·h by multiplying by the mass flow rate and specific heat capacity (1 kJ/[kg·K]) of the supply air. For a building with 1.2 kg/s (≈1000 L/s or [2,000 cfm]) of outdoor supply air, the energy recovered by the sensible heat exchanger is 76,600 kW·h in the winter and 3,500 kW·h in the summer, while the auxiliary heating is 2,700 kW·h and the auxiliary cooling is 25,700 kW·h. If the price of heating and cooling energy is \$0.05/kW·h and \$0.15/kW·h, the sensible heat exchanger will save \$4,400 per year.

These results are for an effectiveness of 70%, which may not represent the least life-cycle costs. Increasing the effectiveness will decrease the required heating and cooling energy and may decrease the capacity of the boiler and chiller, but will increase the capital costs of the heat exchanger (and maybe the fan energy consumption). Similarly, decreasing the effectiveness will increase the auxiliary heating and cooling, but decrease the capital costs of the heat exchanger.

For buildings with peak heating and cooling loads (at $t = 0h$ and $8{,}760h$ respectively), where the ventilation air load is a larger fraction of the building total, *Figure 5* shows that an air-to-air heat exchanger may alter the required capacity of the boiler and chiller significantly. Considering only the sensible energy rate in *Figure 5*, peak auxiliary heating and cooling loads for ventilation air will be reduced by about 85% and 40% respectively. The optimal effectiveness, which depends on energy prices, HVAC component costs, and building operation, can be found by a sensitivity analysis where the effectiveness is changed and the cost of air heating and cooling and the capital costs are calculated. A graph of the total costs versus effectiveness will reveal the design with the least life-cycle costs. Of course, this analytical optimum must be substituted back into Equation 5 for C_T to ensure that the design is cost effective; i.e.,

$$C_T \text{ (with he)} < C_T \text{ (with no he)} \qquad (7)$$

When it is cost effective, it is important to know how good this investment is, which is a given by the payback period or return on investment. The simple payback period (P_{PB}), or return on investment (*ROI*), for the air-to-air heat exchanger is given by

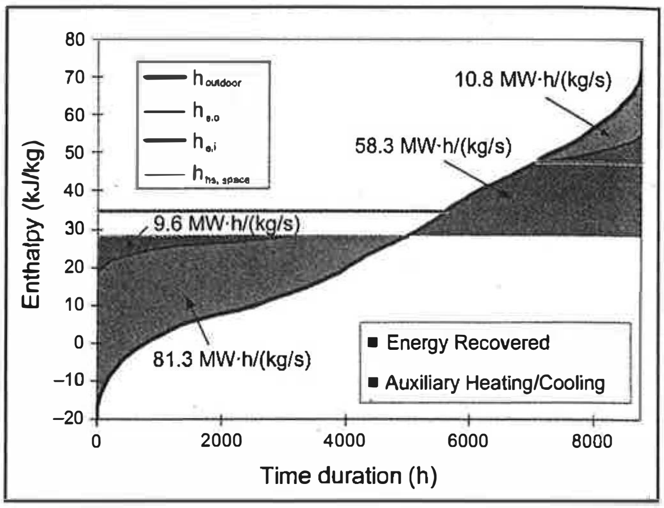

***Figure 6:** Distribution of outdoor enthalpy during the year shows the energy recovered with an energy wheel and the auxiliary heating and cooling required to meet the load of a building with 100% outdoor supply air in Chicago.*

$$P_{PB} = \frac{1}{ROI} = \frac{C_c P_{wef}}{\left[C_o \text{ (with no he)} - C_o \text{ (with he)}\right]} \qquad (8)$$

When P_{PB} is less than one year, a design with the optimal effectiveness still ensures the maximum cost savings for a design that is the least cost alternative.

In the earlier analysis (e.g., for many plate type or heat pipe air-to-air heat exchangers), a number of simplifying assumptions have been made that are nearly valid for air-to-air heat exchanger applications where the heat exchanger is used only to heat and cool the supply air. Frosting effects are assumed to be negligible. When an evaporative cooling system is used in the exhaust air before the air goes into the heat exchanger, $T_{e,i}$ can be lowered (e.g., up to 10°C [18°F]) which will change the contribution of air cooling, especially in warm dry weather conditions. Cooling energy required for condensing supply air water vapor is not included in this analysis.

For HVAC systems with an economizer where recycled air is mixed with the supply outside air, the value of the supply temperature, $T_{s,space}$, at which air is returned to the space should be adjusted to the lowest value when the economizer by-passes air around the heat exchanger. If no air passes through the heat exchanger when the economizer flow is present, then the time for which the air-to-air heat exchanger is providing a benefit may be shorter. Without an economizer, the supply air temperature will be about 15°C (59°F). From this outside air temperature to the exhaust air, $T_{e,i}$, the economizer will be used to decrease or eliminate the cooling load. If the building cooling load can be met with an economizer when the outdoor temperature is below 24°C (75°F), the auxiliary cooling energy will only be 1,500 kW·h in *Figure 4*.

Although *Figure 5* shows significant energy savings for the sensible heat exchanger in the winter, limited energy recovery is evident in the summer. Furthermore, the results in *Figure 5* only include sensible heat transfer and neglect the fact that the humidity in the space will be very low and high during parts

of the winter and summer respectively. An analysis with enthalpy will more accurately represent energy transfers for buildings with humidity control.

Energy Wheels

Sizing energy wheels is more complex than heat exchangers that transfer sensible energy because both heat and moisture are transferred. Using a similar approach to that for sensible heat exchangers, *Figure 6* shows the recovered and auxiliary heating and cooling energies and the implied reductions in the peak ventilation air heating ($t = 0$) and cooling ($t = 8,760h$) loads of about 80% and 40% respectively for an energy wheel in Chicago.

Comparing *Figures 5* and *6* it is evident that both the green and red shaded areas are larger in *Figure 6*. The reason for this is that *Figure 6* assumes that the building has both temperature and humidity control, which means humidification in the winter and dehumidification in the summer. Since humidity control is typical in the summer but atypical in the winter, both *Figures 5* and *6* deviate from reality, but give an indication of the potential savings and life-cycle cost analysis. *Figure 5* is likely more representative for winter conditions and *Figure 6* is likely more representative for summer conditions. Since the ordinate in *Figure 6* is enthalpy, the areas in *Figure 6* can be converted to energy simply by multiplying by the outdoor supply airflow rate. Again, assuming 1.2 kg/s (≈1000 L/s or 2,000 cfm), the energy recovered by the energy wheel in the winter and summer is 97,600 kW·h and 12,900 kW·h, while the auxiliary heating energy is 11,500 kW·h and the auxiliary cooling energy is 70,000 kW·h. For a heating and cooling energy cost of \$0.05/kW·h and \$0.15/kW·h, the energy wheel will save \$6,800 per year. Once again, these figures are based on effectiveness values of 70%. By altering this effectiveness value and calculating the total life-cycle costs, the designer can find the optimal life-cycle cost solution.

The main assumptions behind this energy wheel sizing problem are that supply air enthalpy difference is the correct measure of the supply air auxiliary energy requirements in both summer and winter and that $h_{e,i}$ and $h_{s,space}$ are known values of enthalpy and are constant. These will not be correct assumptions when the supply and exhaust air humidity changes over a wide range. Part load control of energy wheels is discussed in more detail elsewhere.[11] Again, frosting effects are assumed to be negligible. Condensation effects are unlikely to occur for energy wheels in most applications.

Maintenance and Reliability

It was stated earlier that air-to-air heat exchangers have excellent maintenance and reliability characteristics similar to the best HVAC equipment. The experience history of energy wheel applications is much shorter. There may be some doubts about their long-term performance. Nonetheless, there are long-term experiences with desiccant wheels used as supply air dryers. Provided the desiccant coatings of these desiccant dryer wheels are not contaminated by solvents or excessive amounts of organics or dust, or eroded by particulates, then they can have the same long-term performance characteristics as heat exchangers. Although energy wheels operate at much lower tempera-

tures than desiccant dryers and they may use slightly different desiccant coatings, we may expect the same long-term reliability and the same vulnerability to airborne contaminants. The risk of such exposures will be small for well-designed systems with the correct filters.

Conclusions

Applying air-to-air heat/energy exchangers in buildings is a cost-effective and reliable way of conditioning outside ventilation air. For many climates and applications an energy wheel is favored, while sensible heat exchangers are favored in others. This article shows that air-to-air heat/energy recovery can reduce significantly the capital costs and energy consumption of auxiliary heating and cooling equipment. For a retrofit, the payback on investment may be several years. In new applications the payback may be immediate because a carefully designed HVAC system, which includes energy recovery, will often have as low or lower initial costs as a system without energy recovery. In this case, the energy savings that accrue will result from essentially no investment.

Air-to-air heat and moisture exchange for ventilation air is important for HVAC design and operation because it can:

(1) reduce peak auxiliary energy rates and annual loads as well as capital and operating costs

(2) permit higher ventilation rates to create better IAQ at minimum auxiliary energy costs

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