Table of Contents

١. INTRODUCTION

As fossil fuel reserves are depleted, the cost of energy continues to rise. According to the US Department of Energy, the cost of energy used in US commercial buildings increased by more than ٢٠٠% between ١٩٧٩ and ١٩٩٥ and conservative DOE estimates predict an additional ٤٦% increase between ٢٠٠١ and ٢٠٢٥. Given this and the fact that, on average, HVAC systems consume ٣٩% of the energy used in commercial buildings, energy- efficient HVAC systems represent potentially significant savings in building operating costs. Increased ventilation rates, which are required to satisfy the ventilation standard ASHRAE ٦٢.١-٢٠٠٤, mean a greater expenditure of energy to condition outside air.

One way that savings can be realized in an HVAC system is through the use of exhaust air energy recovery. Exhaust air energy recovery can take many forms—rotary heat exchangers, heat-pipes, plate heat exchangers, etc., but all of the devices operate on the same principle—they use exhaust air to condition supply air through a transfer of energy. This application guide examines the rotary heat exchanger also called an energy wheel, or energy recovery wheel and the benefits of incorporating such a device in an air handling unit.

١.١. Summary

Exhaust air energy recovery technology provides a valuable opportunity for engineers to reduce the first costs and operating costs of buildings. Owners benefit not only in these initial and annual savings, but may also receive incentives for operating a “green” building in some areas. Finally, the use of energy recovery reduces the use of non-renewable resources and promotes a cleaner environment. Therefore, whether mandated by state or local building codes, or not, proper application of energy recovery wheels and heat recovery in general is a win-win proposition.

٢. 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.

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 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.

However, a high value of e 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 ١٠ or more independent design variables as well as a number of specified constraints and operating conditions.
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

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

$$= \frac{(C_{s,\,init} - \mathrm{ITC})}{C_e(1 - T_{inc})}\mathrm{CRF}(i'',n) \quad (١)$$

where
$C_{s,init}$ = initial system cost
ITC = investment tax credit for energy-efficient improvements
Ce = 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^2 = 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 ٥ years; values less than ٣ 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 ٢٥ L/s 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 (١) allowing less expensive filter bags to be used, (٢) improving the efficiency of electronic precipitators, or (٣) 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.

٣. 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.

٤. 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 (٢) is used to determine the rate of energy transfer when sensible (temperature) and latent (moisture) energy transfer occurs, while Equation (٣) is used for sensible-only energy transfer.

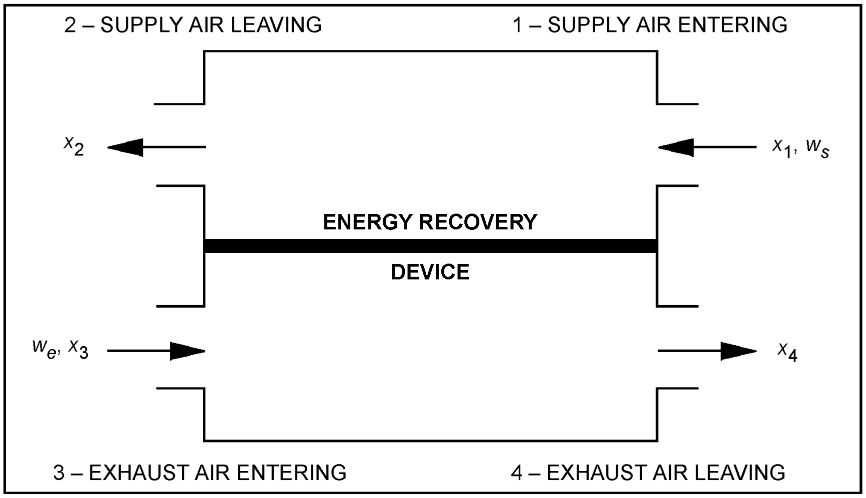


Fig. ١ Airstream Numbering Convention

$q_{total} = Q\rho(h_{in} - h_{out})$ (٢)

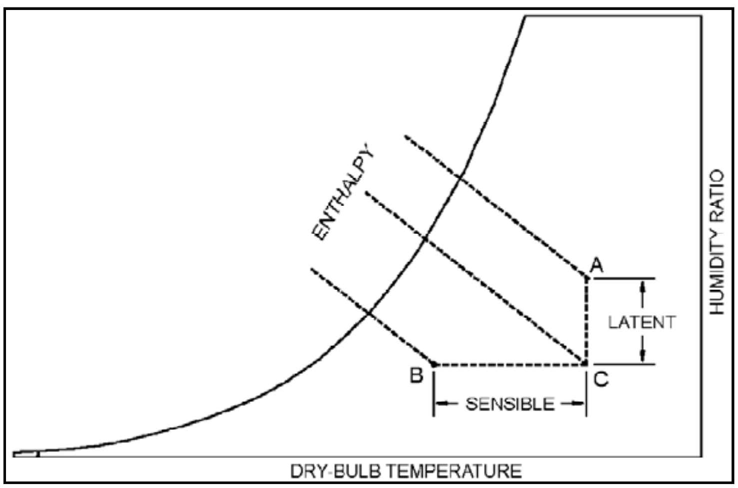


Fig. ٢ Maximum Sensible and Latent Heat from Process A-B

$q_{sensible} = Q\rho c_p(t_{in} - t_{out})$ (٣)

where

$q_{total} = q_{sensible} + q_{latent}$ = total energy transfer, kW
$q_{sensible}$ = sensible heat transfer, kW
Q = airflow rate, m٣/s
r = air density, kg/m٣
cp = specific heat of air = ١.٠٠ kJ/(kg·K)
t_{in} = dry-bulb temperature of air entering exchanger, °C
t_{out} = dry-bulb temperature of air leaving exchanger, °C
h_{in} = enthalpy of air entering heat exchanger, kJ/kg
h_{out} = enthalpy of air leaving heat exchanger, kJ/kg

The following general procedure may be used to determine energy recovered in air-to-air energy recovery applications.

Step ١. 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 with airflows stated at standard temperature and pressure conditions. 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 at standard temperature and pressure conditions. If necessary, convert flow rates to mass flow rates (e.g., L/s or m^{٣}/s at standard temperature and pressure or kg/s) 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 ١, is $\rho cpQ_{min}(t_{٣} - t_{١})$. If latent energy transfer occurs, the theoretical maximum energy transfer q_{max} is $rQ_{min}(h_{٣} - h_{١})$. The maximum moisture transfer rate $w_{m,max}$ is also implied by Equation (٢) and is $w_{min}(W_{٣} - W_{١})$, where $W_{٣}$ and $W_{١}$ are the humidity ratios at state ٣ and state ١.
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 ٢. 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 ٢. 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 (١) exchanger construction, including configuration, heat transfer material, moisture transfer properties, transfer surface area, airflow path, distance between heat transfer surfaces, and overall size; and (٢) 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:

a. Does the published sensible effectiveness result from tests with condensation in the exhaust airstream?
b. Are the published effectivenesses for sensible and total energy transfer different or are they assumed to be equal?
c. Are published airflow rates based on standard or actual temperature and barometric pressure at the fan?
d. Has the exchanger performance been verified by an independent laboratory to meet ASHRAE *Standard* ٨٤ 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 ٣. Calculate actual moisture and energy (sensible and total) transfer.
The amount of energy transferred is the product of the effectiveness
□ for the airstream with the lesser mass flow rate and the theoretical maximum heat transfer determined in Step ١ using Equation (٤):

$$\varepsilon = \frac{w_s(x_2 - x_1)}{w_{min}(x_3 - x_1)} = \frac{w_e(x_3 - x_4)}{w_{min}(x_3 - x_1)} \qquad (٤)$$

$$q_{total} = Q\rho(h_{in} - h_{out}) \qquad (٥)$$

$$q_{sensible} = Q\rho c_p(t_{in} - t_{out}) \qquad (٦)$$

$$w_m = \varepsilon_m \, w_{m,\,max} \qquad (٧)$$

$$q_{actual} = \varepsilon q_{max} \qquad (٨)$$

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

Step ٤. 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 ١, ٢, ٣, and ٤ in Equation (٤) 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 (٥) must be used to calculate the leaving air condition for airstreams in which inherent latent energy transfer occurs. Equation (٦) may be used for an airstream if only sensible energy transfer is involved.

Step ٥. 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 ٦. Plot entering and leaving conditions on psychrometric chart.
Examine the plotted information for each airstream to verify that the performance is reasonable and accurate.

٥. 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 ٣). 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 (١) 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 (٢) 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.

٥.١. 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. Directionally oriented media are available in various geometric configurations. The most common consist of small (١.٥ to ٢ mm) 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.

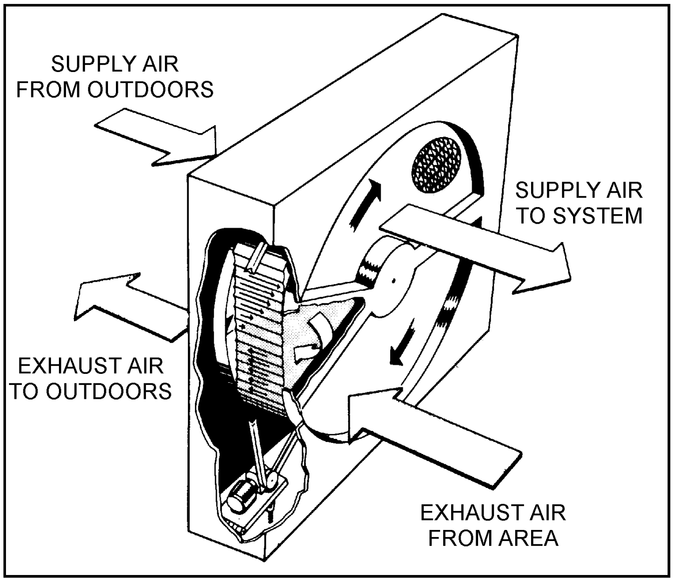


Fig. ٣ Rotary Air-to-Air Energy Exchanger

Stainless steel and ceramics are used for high temperatures and corrosive atmospheres.
Media surface areas exposed to airflow vary from ٣٠٠ to over ٣٣٠٠ m٢/m٣, 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, 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 ٠.١% of the exhaust airflow with a purge section (ASHRAE ١٩٧٤). 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 (٧٥ to ٩٥% void, depending on type and configuration).
For example, a ٣ m diameter, ٢٠٠ mm deep wheel with a ٩٠% void volume operating at ١٤ rpm has a carryover volumetric flow of

π (٣ / ٢)^{٢} (٠.٢)(٠.٩)(١٤ / ٦٠) = ٠.٣ m^{٣} / s

If the wheel is handling a ٩ m^{٣}/s balanced flow, the percentage carryover is
(٠.٣/٩) x١٠٠ = ٣.٣%

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

٦. PRINCIPLES OF ENERGY RECOVERY IN ROTARY AIR-TO-AIR ENERGY EXCHANGERS

Energy recovery involves a transfer of energy between an exhaust airstream and a supply airstream. Figure ٤ illustrates the heat transfer process of an energy recovery wheel where OA is outside air; SA is supply air; RA is return air from the conditioned space and EA is exhaust air.
As the two airstreams pass through the energy recovery wheel, the rotation of the wheel facilitates the transfer of energy from the higher energy airstream to the lower energy airstream. This means that the exhaust air preheats the supply air in the winter and precools the supply air in the summer. Some systems use energy recovery wheels to reheat supply air after it has been cooled—an effective means of humidity control. Some energy recovery wheels transfer only sensible energy, while others transfer sensible and latent (i.e. total) energy.

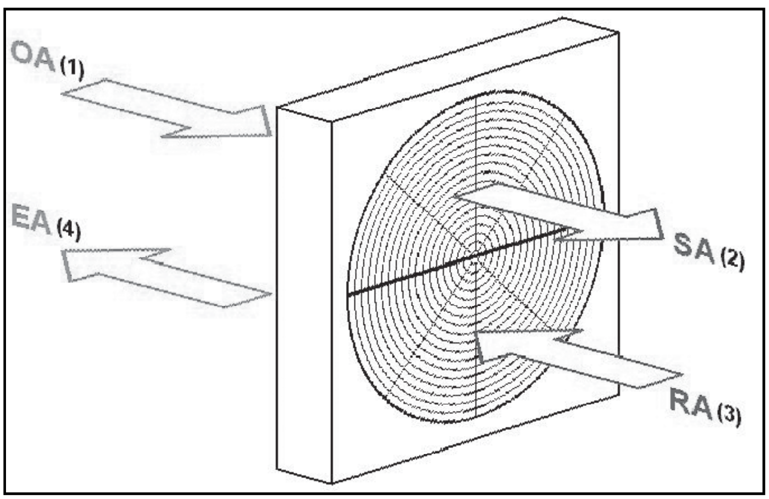


Figure ٤. Standard Airflow Conventions

٦.١. Sensible Heat Transfer

When sensible heat is transferred, the dry-bulb temperature of the colder airstream increases and the drybulb temperature of the warmer airstream decreases. No moisture is transferred, so the humidity ratio of the two airstreams remains unchanged unless the dry-bulb temperature of the warmer airstream is decreased below its dew point, allowing condensation to occur.

٦.٢. Total Heat Transfer

This process involves the transfer of sensible and latent heat energy. Latent heat energy is dependent on the amount of water vapor in the air and therefore total heat transfer can only occur when water vapor is transferred from one airstream to the other. In an energy recovery wheel this transfer is accomplished through the use of a desiccant which absorbs/adsorbs water vapor from the higher vapor pressure airstream and releases it to the lower vapor pressure airstream. Only energy recovery wheels and certain types of fixed-plate heat exchangers with permeable membranes can transfer latent, and therefore total, energy.

٦.٣. Effectiveness

The ratio of the amount of energy transferred by the energy recovery device to the difference in energy levels of the two incoming airstreams is called effectiveness.
The total amount of energy transferred by the wheel is a function of the effectiveness of the wheel, the airflow volumes of the two airstreams and the difference in energy levels between the two airstreams.
Equation ٩ shows the calculation of effectiveness as defined by ASHRAE Standard ٨٤-١٩٩١:

$$\square = [Vs \bullet (x_{١} - x_{٢})] / [V_{min} \bullet (x_{١} - x_{٣})] \qquad (٩)$$

Where:

$\square$ = Sensible, or total effectiveness
$x_{١}$ = OA temp (°Fdb) or enthalpy (btu/lb.)
$x_{٢}$ = SA temp (°Fdb) or enthalpy (btu/lb.)
$x_{٣}$ = RA temp. (°Fdb) or enthalpy (btu/lb.)
Vs = Supply (or outside) air volume (cfm)
V_{min} = the lower of the exhaust or supply air volume (cfm)

Amount of Heat Transferred
The sensible and total energy transferred by the energy recovery wheel can be calculated using Equations ١٠ and ١١:

$$Qs = \square \bullet ١.٠٨ \bullet V_{min} \bullet (t_{١} - t_{٣}) \qquad (١٠)$$
$$Qt = \square \bullet ٤.٥ \bullet V_{min} \bullet (h_{١} - h_{٣}) \qquad (١١)$$

Where:
Q_s = Sensible heat transferred (btu/hr)
Q_t = Total heat transferred (btu/hr)
$\square$ = Sensible, or total effectiveness
V_{min} = the lower of the exhaust or supply air volumes (cfm)

$t_{١}$ = Outside air temperature (°F)
$t_{٣}$ = Return air temperature (°F)
$h_{١}$ = Outside air enthalpy (btu/lb.)
$h_{٣}$ = Return air enthalpy (btu/lb.)
١.٠٨ = Conversion factor
٤.٥ = Conversion factor

If the effectiveness of the wheel is known, Equation ٩ can be solved for $x_{٢}$ to determine the supply air leaving enthalpy and/or temperature. In Example ١ the outside air and return air volumes are equal, or “balanced” ($Vs = V_{min}$). Optimal energy transfer occurs at “balanced flow” conditions, however building exfiltration and exhaust sources such as bathroom fans reduce the return air volume below the supply air volume. This unbalanced flow reduces heat transfer, even though it increases the effectiveness factor of the energy recovery wheel.

٦.٤. Examples:

Example ١: Calculate the supply air conditions leaving a *total* heat recovery wheel with ١٢,٠٠٠ cfm of outside air at ٩٤°Fdb and ٧٧°Fwb; and ١٢,٠٠٠ cfm of return air at ٧٥°Fdb and ٥٠% RH, if both the sensible & latent effectiveness of the wheel is ٦٠%. Calculate the reduction in required cooling energy.

At the conditions given above:
$x_{٣}$ = the return air enthalpy = ٢٨.١٥ btu/lb. = $h_{٣}$
$x_{١}$ = the outside air enthalpy = ٤٠.٣٨ btu/lb. = $h_{١}$

From Equation ٩:
$h_{٢} = h_{١} - [□ \bullet V_{min} \bullet (h_{١} - h_{٣})]/Vs$
$h_{٢} = ٤٠.٣٨ - [٠.٦ \bullet ١٢٠٠٠ \bullet (٤٠.٣٨ - ٢٨.١٥)] / ١٢٠٠٠$
$h_{٢}$ = ٣٣.٠٤ btu/lb.

Also from Equation ٩:
$t_{٢} = t_{١} - [□ \bullet Vmin \bullet (t_{١} - t_{٣})]/Vs$
$t_{٢} = ٩٤ - [٠.٦ \bullet ١٢٠٠٠ \bullet (٩٤ - ٧٥)] / ١٢٠٠٠$
$t_{٢}$ = ٨٢.٦°F.

Therefore, the supply air leaving the heat recovery wheel will be: ٨٢.٦°Fdb / ٦٨.٩°Fwb

From Equation ١١:
$Q_t = □ \bullet ٤.٥ \bullet V_{min} \bullet (h_{١} - h_{٣})$
$Q_t = ٠.٦ \bullet ٤.٥ \bullet ١٢٠٠٠ \bullet (٤٠.٣٨ - ٢٨.١٥)$
Q_t = ٣٩٦,٢٥٢ btu/hr = ٣٣ tons

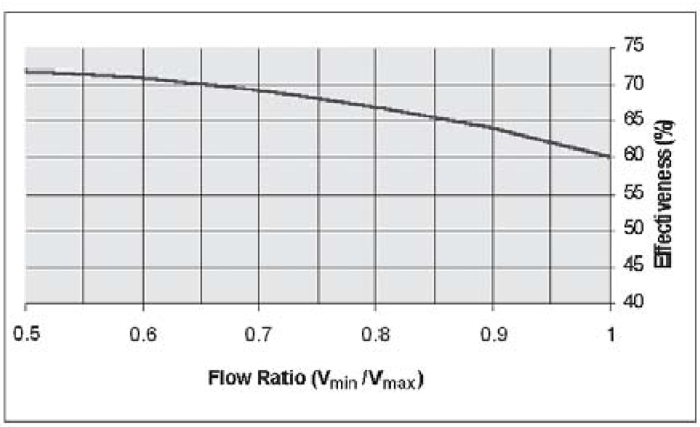


Figure ٥– Effect of Unbalanced Airflow

Example ٢ indicates that despite a ٧% increase in effectiveness, the overall heat transfer of the wheel decreased by ١٠% in relation to the balanced flow conditions of Example ١.

Example ٢: Consider the unit from Example ١, but instead of balanced flow, the return airflow is only ٩,٦٠٠ due to exfiltration and/or exhaust air. What are the conditions of the supply air leaving the energy recovery wheel in this case? What is the new heat transfer rate? (Assume the outside air and return air conditions remain the same.)

V_{min}/V_{max} = ٩,٦٠٠ cfm / ١٢,٠٠٠ cfm = ٠.٨.

From Figure ٥: At ٠.٨ flow ratio, □ = ٦٧%.
Solving Equation ١ for $x_{٢}$ ($h_{٢}$):
$h_{٢} = h_{١}$ - [□ • V_{min} • ($h_{١}$ - $h_{٣}$)]/Vs
$h_{٢}$ = ٤٠.٣٨ - [٠.٦٧ • ٩٦٠٠ • (٤٠.٣٨ - ٢٨.١٥)] / ١٢٠٠٠
$h_{٢}$ = ٣٣.٨٢ btu/lb.

And:

$t_{٢} = t_{١}$ - [□ • V_{min} • ($t_{١}$ - $t_{٣}$)]/Vs
$t_{٢}$ = ٩٤ - [٠.٦٧ • ٩٦٠٠ • (٩٤ - ٧٥)] / ١٢٠٠٠
$t_{٢}$ = ٨٣.٨°F.

Therefore, the supply air leaving the energy recovery wheel will be: ٨٣.٨°Fdb / ٦٩.٨°Fwb
From Equation ٣:

Qt = □ • ٤.٥ • V_{min} • ($h_{١}$ - $h_{٣}$)
Qt = ٠.٦٧ • ٤.٥ • ٩٦٠٠ • (٤٠.٣٨ - ٢٨.١٥)
Qt = ٣٥٣,٩٨٥ btu/hr = ٢٩.٥ tons

If a sensible-only energy recovery device, such as a heat pipe, had been used, the energy savings in Example ١ would have been:
Qs = ٠.٦ • ١.٠٨ • ١٢٠٠٠ • (٩٤-٧٥) = ١٤٧,٧٤٤ btu/hr
= ١٢.٣ tons
And for Example ٢:
Qs = ٠.٦٧ • ١.٠٨ • ٩٦٠٠ • (٩٤-٧٥) = ١٣١,٩٨٥ btu/hr
= ١١.٠ tons
The additional energy savings achieved by transferring total energy illustrate one of the advantages of using energy recovery wheels over a device that only transfers sensible energy.

٦.٥. Transfer of Air between Airstreams
Inherent in the operation of an energy recovery wheel is a direct transfer of air between the return and supply airstreams (See Figure ٦). This air transfer is due to leakage through the seals separating the airstreams as well as by the small amount of air carried over in the matrix of the wheel as it rotates from one airstream to the other.

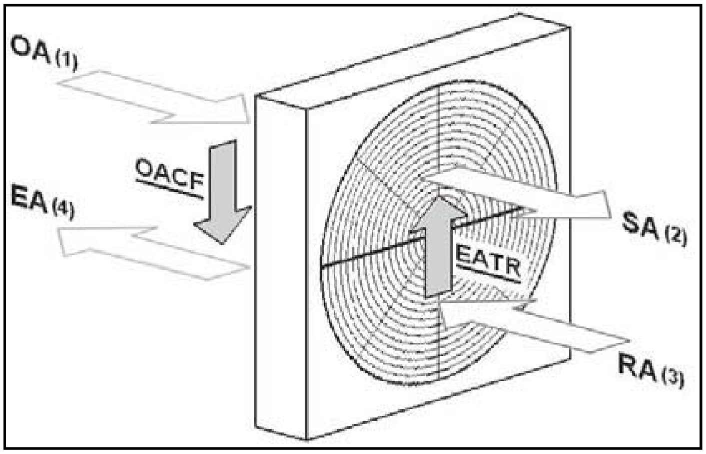


Figure ٦ – Air Transfer Paths

The air passing from the return airstream to the supply airstream is defined as the Exhaust Air Transfer Ratio (EATR). The EATR is the percentage of supply air that originated as return air. This ratio is determined by measuring the concentrations of a tracer gas in the RA, SA and OA airstreams. The air passing from the outside airstream to the exhaust airstream is defined as the Outside Air Correction Factor (OACF). The OACF is the OA volume at
Point ١ divided by SA volume at Point ٢. The OACF and EATR are determined for a given condition through testing in accordance with ARI Standard ١٠٦٠-٢٠٠٠. Figure ٧ illustrates the effect of this leakage on the airflow rates of an energy recovery wheel with a balanced, nominal flow rate of ١٢,٠٠٠ cfm. The EATR and OACF are typically calculated by software provided by the energy recovery wheel manufacturer, but it is important to understand how these factors affect, and are affected by, system design. The magnitude of the EATR and OACF affects fan sizing, while the positions of the supply and exhaust fans, and the air pressure drops they develop, affect the magnitude of the EATR and OACF. Figures ٨ through ١١ show the four possible fan arrangements along with the advantages and precautions associated with each.

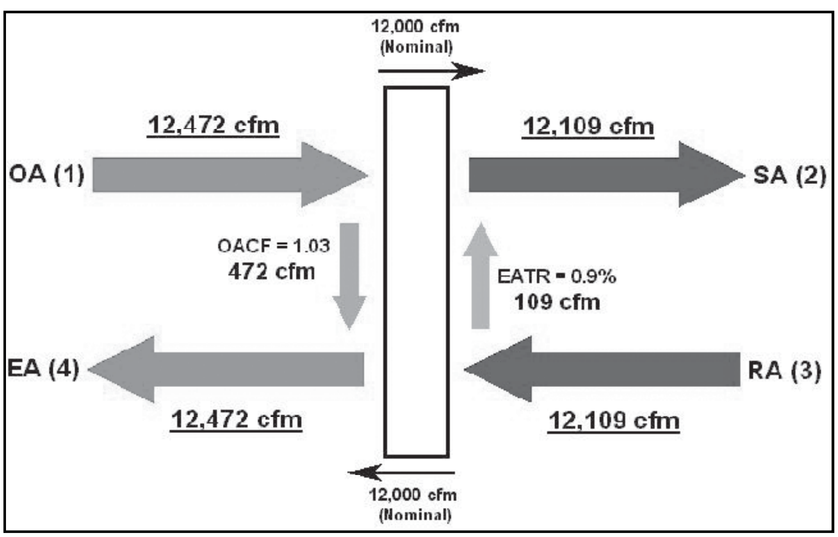


Figure ٧– Effect of EATR and OACF

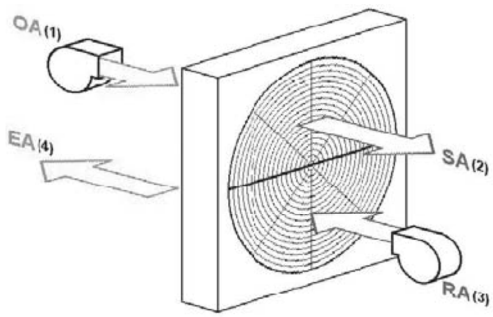


Figure ٨ – Blow-Thru Supply/Blow-Thru Exhaust

Advantages
• Minimal leakage when supply and exhaust path static pressures is nearly equal.
Precautions
• Direction of leakage depends on relative static pressures.

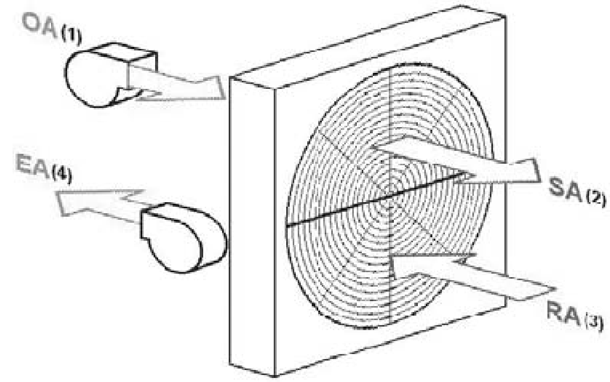


Figure ٩ – Blow-Thru Supply/Draw-Thru Exhaust

Advantages
• Minimizes leakage of air from exhaust path to supply path.
Precautions
• Leakage from supply path to exhaust path can be excessive if static pressure differences are not minimized.

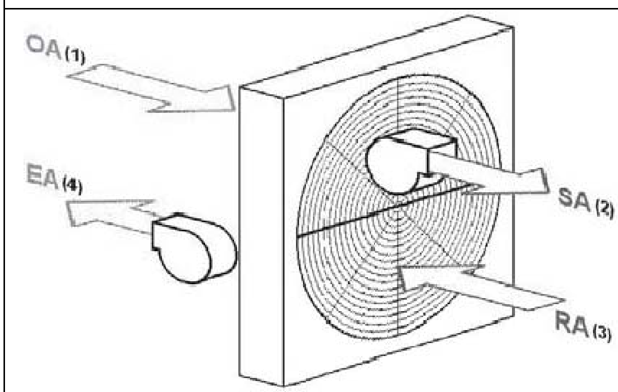


Figure ١٠ – Draw-Thru Supply/Draw-Thru Exhaust

Advantages
• Minimal leakage when supply and exhaust path static pressures are nearly equal.
• Good air distribution across the face of the wheel.
Precautions
• Direction of leakage depends on relative static pressure.

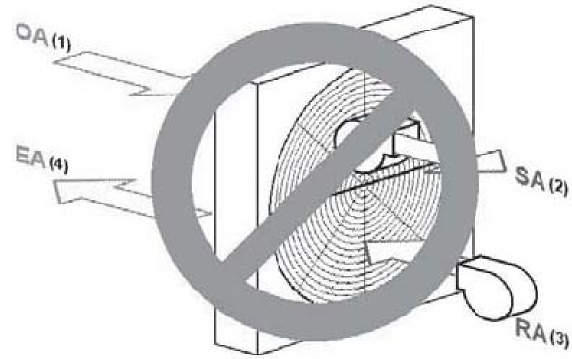


Figure ١١ – Draw-Thru Supply/Blow-Thru Exhaust

Advantages
• None.
Precautions
• Significant leakage from exhaust air path to supply air path.
• **DO NOT USE**.

Proper fan arrangement and control of the static pressure minimizes the air transfer caused by leakage through the gaskets and seals of the wheel, however these practices will not reduce the transfer of air caused by carryover. As Figure ١٢ shows, some of the air from the return/exhaust path remains in the matrix of the wheel as it rotates to the outside/supply airstream. This "carryover" air mixes with the incoming outside air and enters the supply air path. For most applications, such as comfort cooling, carryover is of little concern. However, if there were high concentrations of hazardous substances such as VOCs or carcinogens in the air, they would enter the supply airstream via the carryover air and may present a health hazard to occupants. For this reason, energy recovery wheels should not be used in applications where high concentrations of hazardous substances may be present in the exhaust air, for example laboratories or hospital operating rooms.

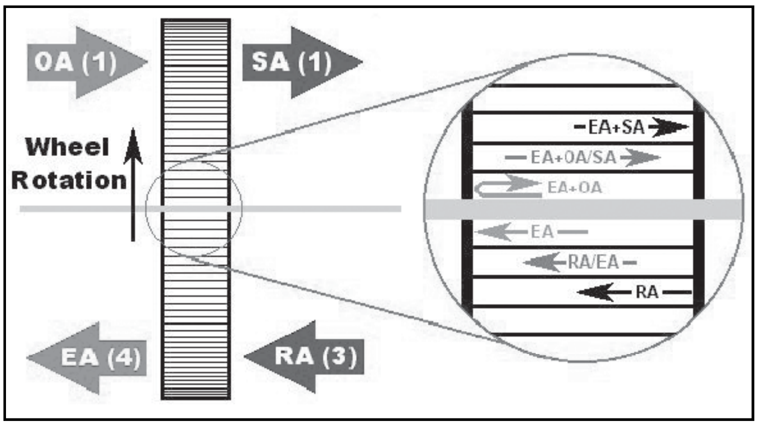


Figure ١٢ – Air Transfer Due To Carryover

Purge

A mechanical purge section can be used to reduce the volume of carryover air. (See Figure ١٣.) Mechanical purge isolates a section of the wheel on the boundary between the RA/EA path and the OA/SA path at the point where the wheel rotates from the RA/EA path into the OA/SA path. Placing a block-off over this section of the wheel on the RA/SA side forces outside air that has traveled through the wheel to flow back into it in the opposite direction. This prevents air from the RA/EA path from entering the last few degrees of the wheel before it rotates into the OA/SA path. The return air that entered the wheel prior to the purge section has time to exit the wheel on the exhaust side. The angle of the purge section - how large a "slice" of the wheel it covers determines how effective it is. The larger the angle, the greater is the reduction in carryover. Purge should not, however, be counted on to eliminate carryover completely; therefore, even energy recovery wheels equipped with purge sections should not be used when high concentrations of hazardous substances are likely in the exhaust airstream.

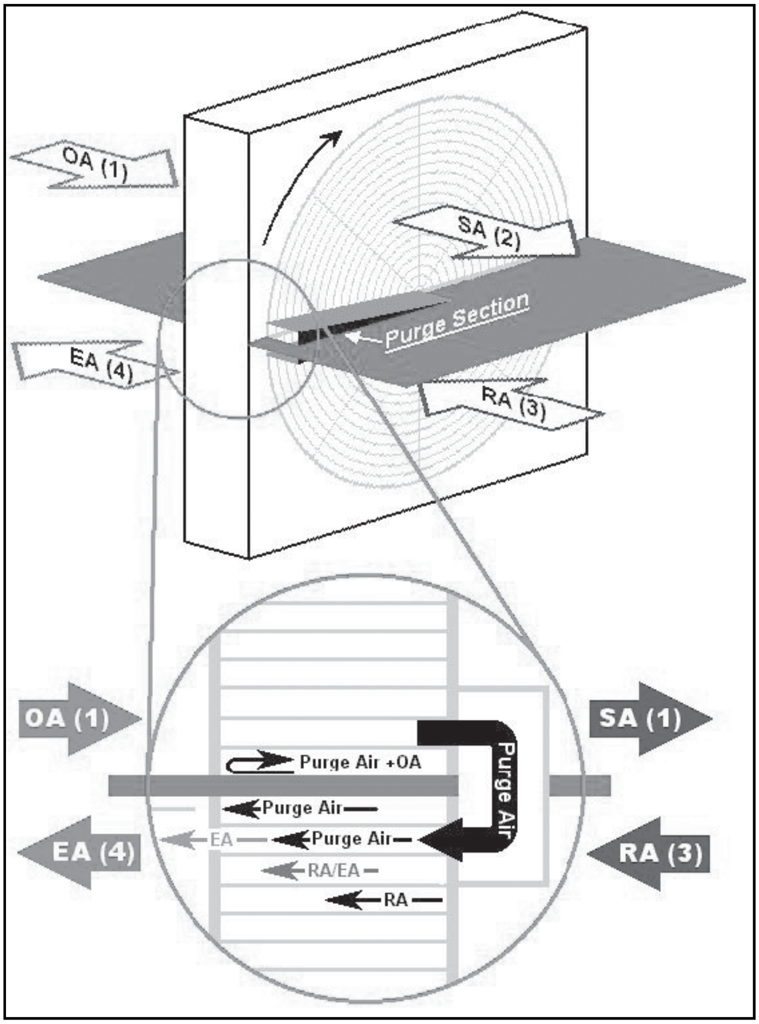


Figure ١٣ – Energy Recovery Wheel with Purge Section

٧. 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 (١) a silicon controlled rectifier (SCR) with variable-speed dc motor, (٢) a constant speed ac motor with hysteresis coupling, and (٣) an ac frequency inverter with an ac induction motor.

Figure ١٤ 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 ٤, and no leakage or cross-flow,

NTU = $(UA)avg / Cmin$ (١٢)
where
(UA)avg = product of modified overall heat transfer coefficient and heat exchange area, W/K
Cmin = minimum heat capacity rate of hot and cold airstreams, W/K
Cr = heat capacity rate for air mass within rotary wheel, W/K

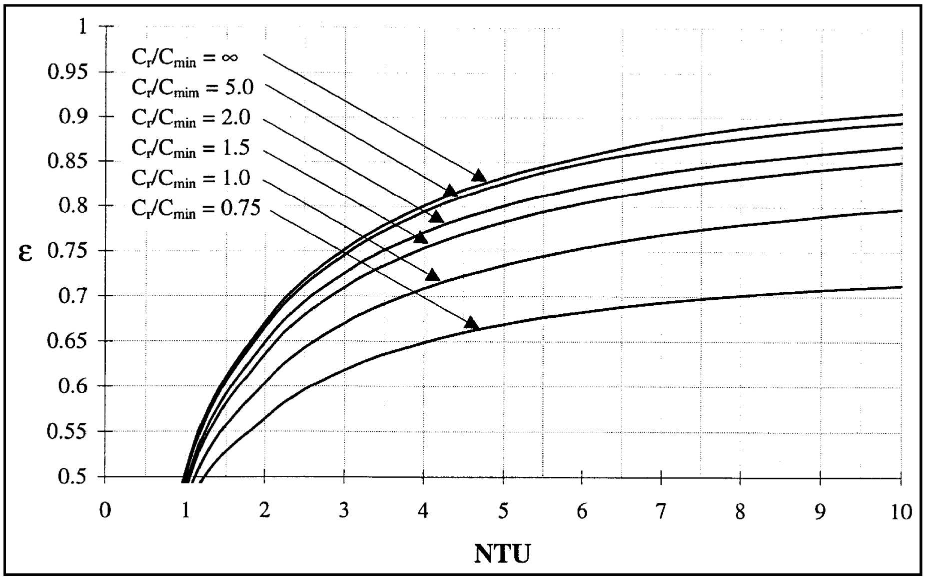


Fig. ١٤ Effectiveness of Counterflow Regenerator (Shah ١٩٨١)

Figure ١٤ also shows that regenerative counterflow rotary effectiveness increases with wheel speed (Cr is proportional to wheel speed); but there is no advantage in going beyond
Cr /Cmin = ٥ because the carryover of contaminants increases with wheel speed.
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 ٨٤.
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.

٧.١. Frost Control

Frost formation is always a concern when HVAC equipment operates in sub-freezing weather. Heat recovery equipment is no exception, but the likelihood of frost formation is greater on sensible-only heat transfer devices, including sensible-only energy wheels, than on total energy transfer devices. Consider both processes on the psychrometric chart:

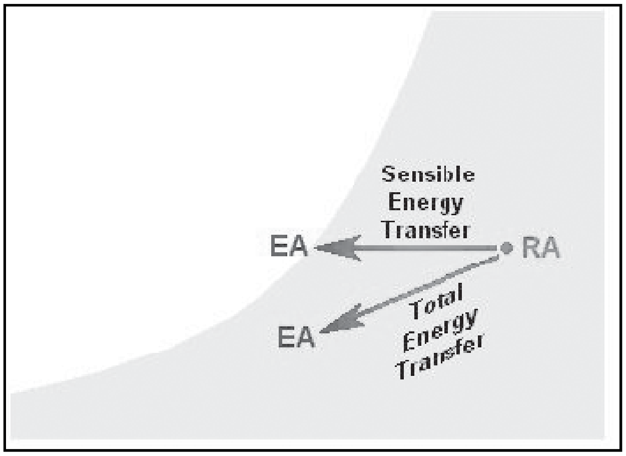


Figure ١٥ – Effect of Total Energy Transfer

During the heating season a total energy recovery device transfers heat and moisture from the warm, moist return air to the cold, dry outside air. This lowers the exhaust air dewpoint - the temperature at which condensation and frost formation occurs. Figure ١٦ compares typical frost threshold temperatures of total energy recovery wheels and sensible-only heat exchangers.
Condensation and frost begin to form to the left of the respective boundary lines. The dewpoint depression achieved with total energy devices can lower the frost formation threshold well below ٠°F when the indoor air relative humidity is low.
Total energy recovery wheels located in climates with extreme winter conditions, and/or where indoor air relative humidities are high may still require some means of frost prevention. There are four common methods of frost prevention - three of which are generally favored over the fourth.

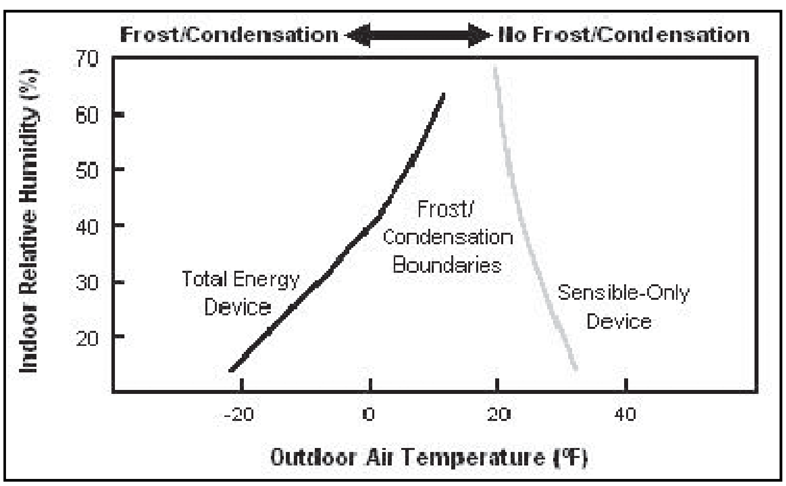


Figure ١٦ – Frost Threshold Temperatures

٧.٢. On-Off control

Is the least expensive and least complicated method of preventing frost. When the wheel is not rotating, no energy transfer takes place and the moisture in the exhaust airstream is in no danger of condensing and freezing. The drawbacks to this method are that when the energy recovery wheel is not operating, no energy savings are realized and heating elements must be sized for design winter conditions. On-Off frost prevention control is best suited for mixed air systems with a low minimum outside air requirement and in climates where the outdoor air temperature drops below the frost threshold mainly during the unoccupied period when ventilation is not required.

Bypassing the outside air or a portion of it is another common tactic for preventing frost formation. This method consists of mounting a bypass damper in the OA/SA path that opens to divert some of the outside air around the energy recovery wheel when outside air conditions are below the frost threshold. This reduces the heat transfer capacity of the wheel and prevents the leaving exhaust air from reaching saturation. Proper mixing is crucial to prevent stratification, which could freeze downstream coils in the air handler. The heating elements must be somewhat larger in this type of system than a system in which the wheel operates at ١٠٠% during winter design conditions, but do not necessarily have to be sized for design winter conditions. Bypass systems are best suited for climates with very few hours per year below the frost threshold and for systems that do not include humidifiers. Systems with airside economizers are required to have OA bypass to take full advantage of economizer operation.

Entering air preheat is a widely accepted method of frost control that has the additional advantage of allowing the wheel to operate at maximum air volume during design heating conditions. The approach to this frost control method is to mount a heating device on either the OA entering side of the wheel (position ١) or on the RA entering side of the wheel (position ٣); see Figure ١٧. When the heating device is mounted in position ١, the temperature of the outside air is raised above the frost threshold temperature of the wheel, preventing the exhaust air from getting cold enough to form frost. When the heating element is placed in position ٣, enough energy is added to the RA path to prevent the heat transfer to the OA/SA path from lowering the exhaust air temperature below the saturation point. This method of frost control is well suited to climates with extreme winter conditions, and for systems that include some form of mechanical humidification.

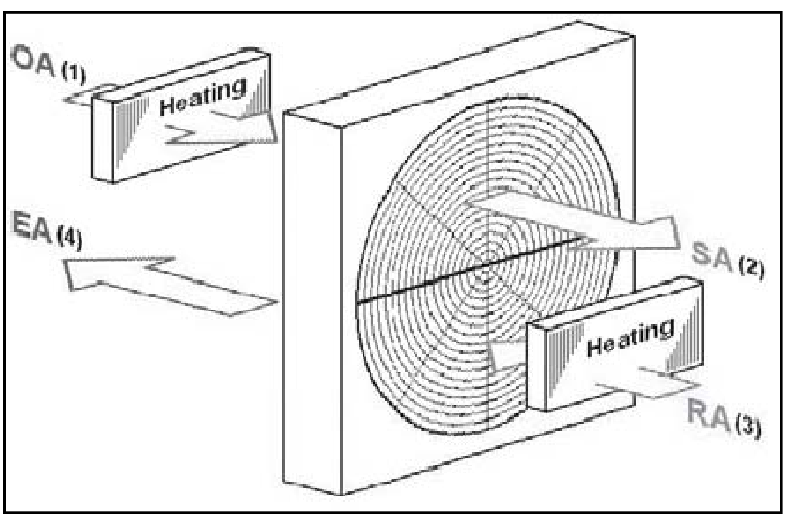


Figure ١٧ – Preheat Frost Control

If the preheat device consists of a steam or hydronic coil mounted in the outside airstream, some form of freeze protection for the coil must be included. No such precaution is necessary if a steam or hydronic coil is used to preheat the return air. Although a heating device mounted in position ٣ (RA) will typically require a greater capacity than a device mounted in position ١ (OA) to provide the same degree of frost prevention, adding heat to the return air path increases the sensible heat transfer to the supply air path. When a preheat element is used in position ٣, caution should be exercised so that the temperature of the entering return air does not exceed the recommended operating temperature of the energy recovery wheel. Variable speed control can be used to reduce the rotational speed of the wheel, which reduces its ability to transfer energy. Because of all the variables involved, this method of frost control requires a sophisticated control strategy, which if not followed correctly can actually increase the likelihood of frost formation. Reducing the wheel speed can also lead to significant reductions in performance and the need for supplementary freeze protection for downstream coils. For these reasons, the previous three methods of frost prevention are generally favored over variable wheel speed control.

٧.٣. Capacity Control

Maximum heat transfer is not always necessary, nor desirable. The two most common methods of controlling the energy transfer rate of the wheel are variable speed control and bypass of exhaust or outside air. Variable speed control consists of using a speed controller, such as a variable frequency drive (VFD) or silicon controlled rectifier (SCR), attached to the drive motor of the wheel. As the rotational speed of the wheel decreases, the heat transfer capacity also decreases. There are limits to this type of capacity control, however. The reduction in capacity is not proportional to the reduction in wheel speed; therefore, reducing the speed of the wheel by ٥٠% may only result in a ١٠% reduction in energy transfer. Exhaust or outside air bypass requires a bypass damper mounted in the OA/SA path or the RA/EA path and has several advantages over variable speed capacity control. Air bypass results in more linear control of the energy wheel capacity making the control strategy more reliable. Also, a greater overall reduction in capacity can be achieved by using bypass than by using variable speed control, and the pressure drop of the system is reduced as the volume of air passing through the wheel is reduced. Placing the bypass damper in a position to divert exhaust air in lieu of outside air will minimize the possibility of stratification in ١٠٠% outside air systems. This damper could also be placed in the exhaust airstream to control the capacity of the wheel by reducing the volume of exhaust air available to transfer energy to or from. However, in an air hander with an airside economizer, when the system modulates from minimum outside air operation to ١٠٠% economizer mode, some means of diverting the outside air around the wheel must be included to prevent excessive pressure drop and unwanted preheating. In this arrangement the economizer bypass damper could be used as a capacity control damper in the minimum outside air mode, although proper mixing of treated and bypassed air would be essential to prevent freezing of hydronic coils downstream of the energy recovery wheel.

٨. ENERGY RECOVERY APPLICATIONS

There are three application categories for air-to-air energy recovery based on the type of system the energy recovery device serves, and two subcategories based on how the device is used in the system. The application categories are process-to-process, process-to-comfort and comfort-to-comfort. Examples of these applications are given in Table ١. The subcategories are preconditioning of outside air and tempering of supply air.

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

Table ١ Applications for Air-to-Air Energy Recovery

٨.١. 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 ٨٧٠°C.

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.

٨.٢. 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.

٨.٣. 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 smallscale commercial applications. Air-to-air energy recovery devices available for comfort-tocomfort 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.

٨.٤. Preconditioning of Outside Air

A primary use of exhaust air energy recovery is to precondition outside air. Preheating/humidifying in winter and precooling/dehumidifying in summer reduce annual operating costs and may decrease first cost as the required heating and cooling capacity is reduced. Figure ١٨ illustrates one possible configuration of an outside air preconditioning system.

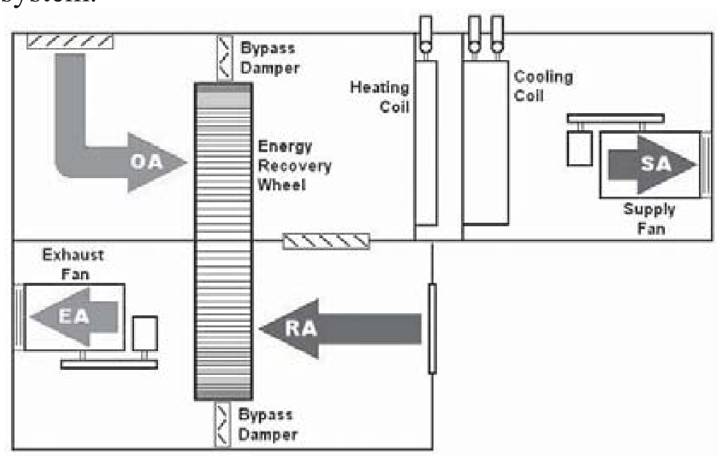
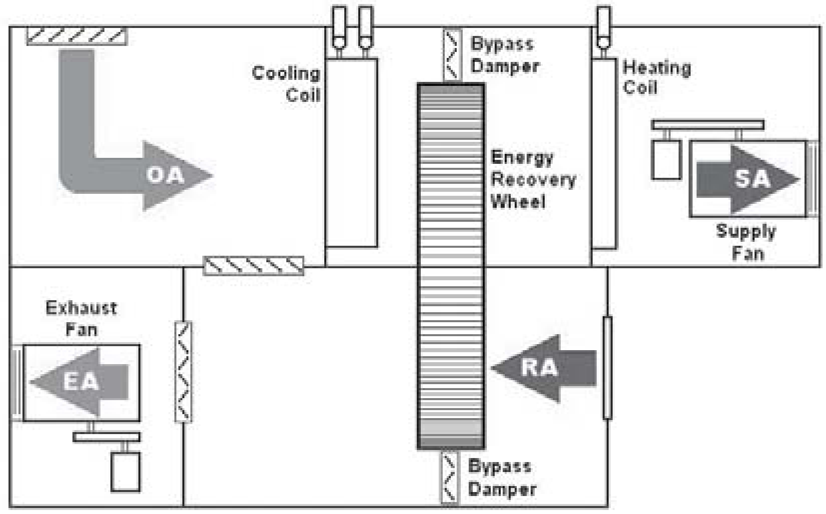


Figure ١٨ – Outside Air Preconditioning

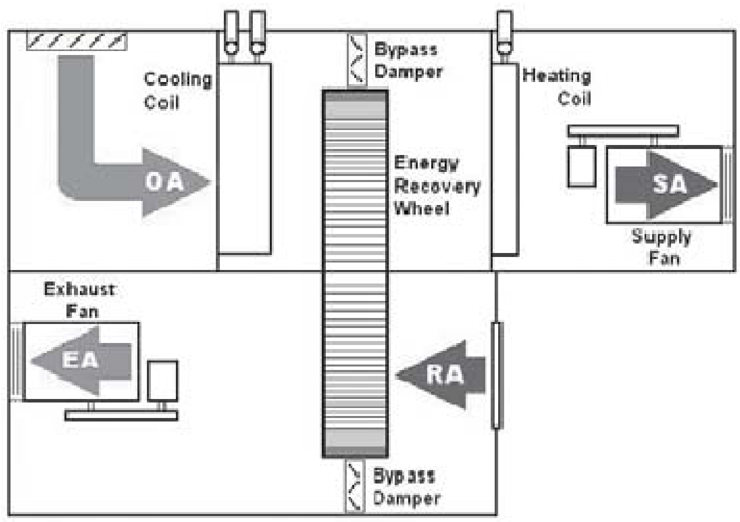
Example ١ on page ٢ illustrates a potential ٣٣ ton (٤٢%) equipment capacity reduction of an outside air preconditioning system in the cooling mode. Using the energy recovery wheel to preheat and humidify the outside air in the heating season can achieve similar savings and capacity reductions. In systems with airside economizers, the energy recovery wheel should be sized for minimum outside airflow only and provisions should be included to bypass or otherwise prevent operation of the wheel during economizer mode. Operation of an energy recovery wheel during economizer mode increases energy consumption.

٨.٥. Tempering of Supply Air

Using an energy recovery wheel to temper supply air is an energy efficient way to control humidity. In the past, humidity control involved cooling the air below the temperature required to satisfy the sensible load and then reheating it. While this simultaneous heating and cooling provided fine temperature and humidity control, it wasted a great deal of energy. ASHRAE Standard ٩٠.١-٢٠٠٤ prohibits simultaneous heating and cooling unless, among other specific exceptions, at least ٧٥% of the reheating energy is provided from a site recovered energy source. Two methods of tempering the supply air are shown in Figures ١٩ and ٢٠. The parallel arrangement uses the energy recovery wheel to reheat the supply air during the cooling mode by transferring energy from the return airstream to the supply airstream. During the heating mode, the wheel preheats the outside or mixed air by transferring energy to the supply airstream. This may help reduce the winter heating load, but any hydronic coils upstream of the wheel may require some of form of freeze protection, particularly in ١٠٠% outside air systems.



MIXED AIR

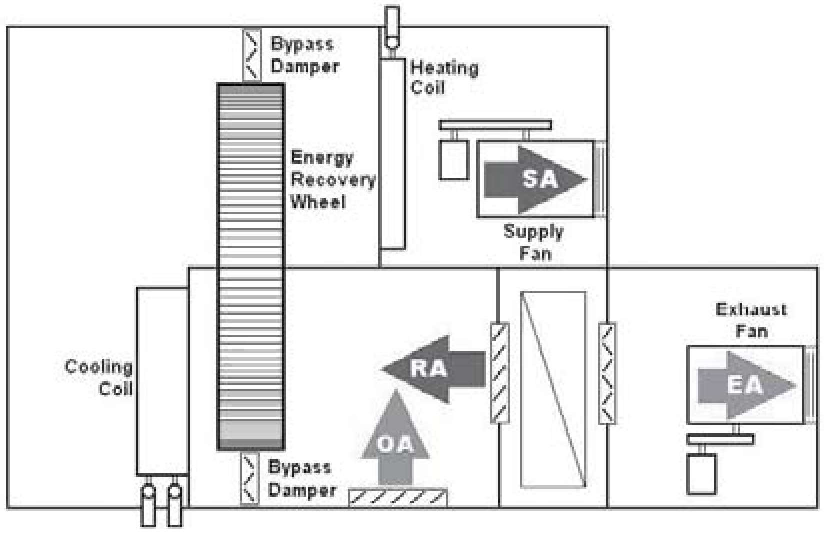


١٠٠% OUTSIDE AIR

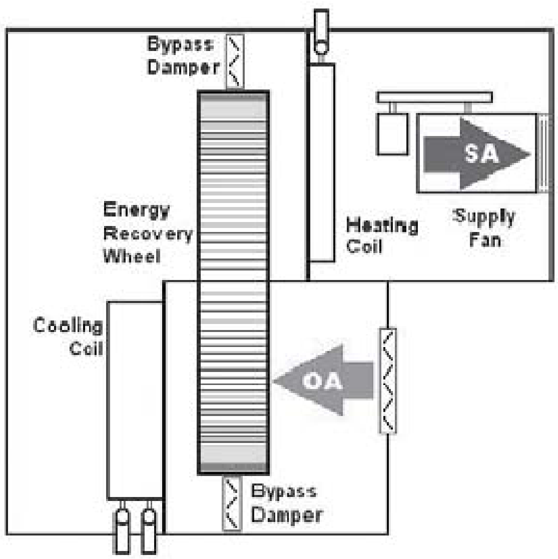
Figure ١٩ – Supply Air Tempering (Parallel)

In the series arrangement, energy is not actually "recovered", but simply transferred from the upstream to the downstream side of the cooling coil. This transfer reheats the supply air and preconditions the outside air simultaneously. Since the goal of a series arrangement supply air tempering system is dehumidified supply air, sensible-only energy recovery devices. Series type systems as shown in Figure ١٦ are not used during the heating season.

In both of the supply air tempering arrangements the cooling system is designed to accommodate the peak cooling (i.e. sensible) design load, which, as Figure ٢١ shows, occurs at a higher dry bulb temperature than the design dehumidifying condition. The energy recovery wheel is not used during peak sensible cooling conditions because it is not desirable to add heat to the supply air when sensible cooling is the main concern. Therefore, using an energy wheel for supply air tempering will not reduce the design capacity of the cooling system. Supply air tempering is used at the peak dehumidifying (i.e. latent) design load when the sensible load in the space is lower and the latent load is higher than the cooling design condition.



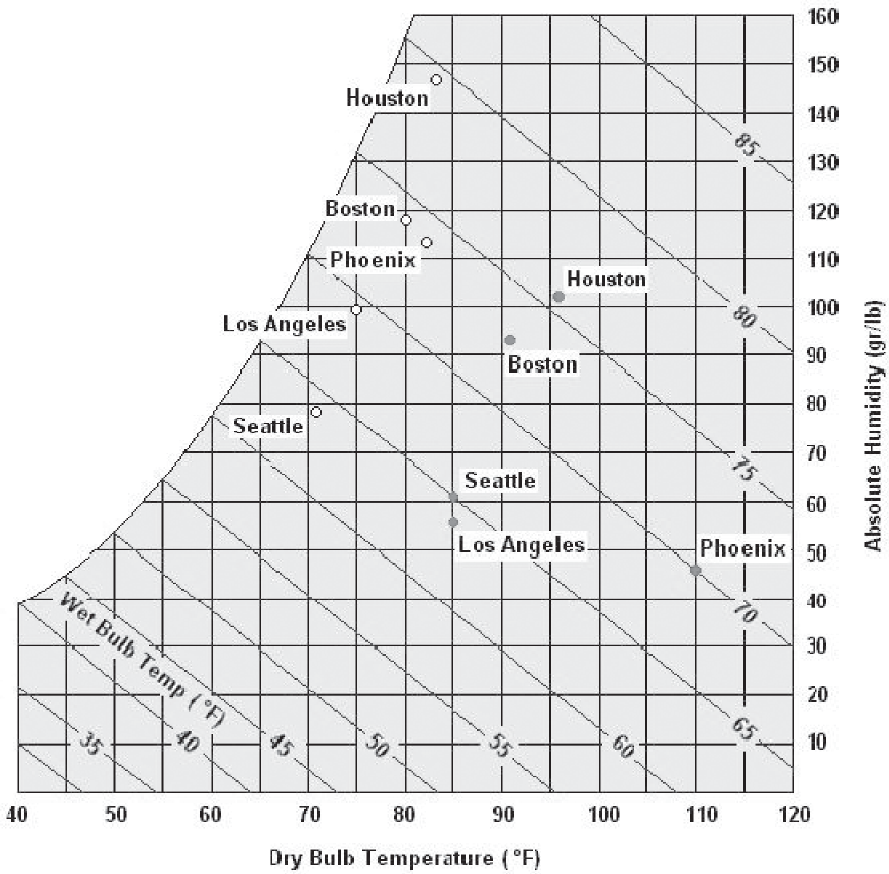
MIXED AIR



١٠٠% OUTSIDE AIR

Figure ٢٠ – Supply Air Temperating (Series)

Typically when the sensible cooling load decreases, the supply air temperature increases (in a constant volume system) or the supply air volume decreases (in a VAV system). However, due to the high latent load in the space, a temperature increase or airflow decrease results in unacceptable humidity levels in the space. To maintain acceptable humidity and temperature levels the supply air is “overcooled” to dehumidify it, and then reheated to the temperature required to satisfy the sensible load. Supply air tempering accomplishes this process while reducing or eliminating the need for mechanical reheat, which satisfies the requirements of ASHRAE Standard ٩٠.١-٢٠٠٤.



	Design (0.4%) Cooling Conditions			Design (0.4%) Dehumidifying Conditions		
Location	DB	WB	DP	DB	WB	DP
Phoenix, AZ	110	70	46.1	82	74.1	71
Los Angeles, CA	85	64	51	75	69.5	67
Boston, MA	91	73	65.2	80	74.2	72
Houston, TX	96	77	69.7	83	79.2	78
Seattle, WA	85	65	53.2	71	63.9	60

● Design Cooling (Sensible) Conditions

○ Design Dehumidifying (Latent) Conditions

Figure ٢١ – Cooling and Dehumidifying Design Conditions

٩. MAINTENANCE

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

١. 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.

٢. 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.
٣. Inspect wheels regularly for proper belt or chain tension.
٤. Refer to the manufacturer's recommendations for spare and replacement parts.

٩.١. Cleaning

Dry particle build-up in the heat exchange matrix is rare. Due to the laminar flow characteristics of the wheel, small particles that enter the matrix typically pass through while larger particles that lodge on the surface of the wheel are blown free when they pass into the counter-flow airstream. Build-up of oil or tar based aerosols on the surface of the energy wheel or within the matrix presents a greater concern. These contaminants can adhere to the surface of the wheel and reduce airflow by obstructing the air or, in the case of total-energy wheels, reducing the latent effectiveness of the wheel by clogging the water adsorbing pores on the surface of the desiccant.
The wheel should periodically (no less than once per year) be inspected for contamination. In environments such as cooking facilities, bars and restaurants with high smoking rates and industrial welding applications where oil or tar-based contamination is more likely, inspections should be conducted more frequently. Inspection and cleaning may be required as frequently as every three months in industrial applications.
Regardless of the application, the wheel should be cleaned whenever contamination is detected. Dry particles can generally be removed by vacuuming, however personnel should consult IOM manuals for instructions on the removal of oil or tar-based films. Different manufacturers use different desiccants and materials of construction, and what works well for one wheel may damage or destroy another. Many manufacturers construct their wheels of multiple, independently removable segments to facilitate the cleaning process. Care should be taken in the design of the system to allow sufficient access space for the inspection and cleaning processes.

١٠. REFERENCES

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٢. ASHRAE ٢٠٠٠ HVAC System and Equipment

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ENERGY WHEELS AND ENERGY RECOVERY VENTILATION TECHNOLOGY
Mark Rabbia, George Dowse, Carrier Corporation, Syracuse, New York, September ٢٠٠٠