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EXECUTIVE SUMMARY

AAONaire energy recovery systems provide energy savings by recycling energy instead of losing energy through exhaust air streams. AAONaire systems also enhance indoor air quality by allowing larger amounts of outside air to be provided to the space and through improved humidity control. AAONaire systems save money through both an initial HVAC equipment reduction and ongoing lifecycle operating savings. For much of the country, the payback for the AAONaire system is less than one year. After the payback period, the system will continue to provide savings for the life of the product.

AAONaire units have been certified to perform according to AHRI Standard 1060. The certified ratings program requires testing, rating and independent verification of component performance at standard conditions and rated flow. Testing is in accordance with ASHRAE Standard 84. AHRI certified ratings include very complete information to allow designers to fully characterize thermal and airflow performance. In addition to separate sensible, latent, and total effectiveness at two airflows for both summer and winter test conditions, the standard requires information on pressure loss as well as air leakage.

In many climates, the AAONaire system eliminates three to four tons of air conditioning load for every 1000 cfm of ventilation air. When this benefit is claimed in the form of air conditioning equipment rightsizing, the initial cost of the system is typically on par with a system that does not utilize energy recovery. Payback is often immediate for the southeastern US and much of the midwest because the AAONaire energy recovery system actually has a lower initial cost than a system not using energy recovery. When payback is not immediate, most payback periods are less than one year.
WHY AAONAIRE AIR-TO-AIR ENERGY RECOVERY?

Building systems that do more with less are increasingly recognized not only as an ethical priority but as good business. Market demand for environmentally responsible heating and air conditioning (HVAC) systems and engineering design continues to grow rapidly. In addition, building code requirements for increased outdoor air ventilation rates have placed new demands on HVAC equipment and on building operating budgets.

Air-to-air energy recovery is the process of recovering energy from an airstream at a high temperature or humidity to an airstream at a low temperature or humidity. This process is important in maintaining acceptable indoor air quality (IAQ) while maintaining low energy costs and reducing overall energy consumption.

Energy can be recovered either in its sensible (temperature only) or latent (moisture) form, or a combination of both. Units that recover energy from temperature only are called sensible heat exchange devices or heat recovery ventilators (HRVs). Units that transfer both heat and moisture are known as energy or enthalpy devices or energy recovery ventilators (ERVs).

**Humidity Control**
Humidity directly affects the comfort level and health in the conditioned space as shown in Figure 2. The application of the AAONAIRE energy recovery device can affect the relative humidity and result in more comfortable conditions. Humidity that reaches excessive levels, for even short periods of time, can create an environment that promotes the growth of fungi and bacteria. Human exposure to fungi and bacteria can cause serious health issues.

**Energy Savings**
By recovering up to 80% of the energy of the exhaust air, as shown in Figure 1, far less energy is spent cooling and heating the outside air supplied to the building. This energy savings can typically reduce the operating cost by thousands of dollars per year for a single unit.
AAONAIRE energy recovery ventilators are perfectly suited to help control humidity. In the summer, when outside humidity is high, the energy wheel dehumidifies the outside air. This greatly reduces the latent load on the air conditioning equipment. It also eliminates the problems with high indoor moisture levels that can occur in hot, humid climates. Air conditioners use much energy to dehumidify moist airstreams. Excessive moisture in the air of a building can result in mold, allergies, and bacterial growth. ERVs can enhance dehumidification with packaged unitary air conditioners. Additionally during the winter, when outside air is dry, the energy wheel humidifies the incoming outside air. This result is increased comfort and reduced dehumidification and humidification requirements year-round.

**Air Conditioning Load Reduction**

The ERV effectively reduces the outside air design conditions. For example, if outside design is 95°F and 78°F wb and the return are is 75°F and 63°F wb, an 80% effective energy wheel would precondition the outside air to 79°F and 66°F wb. This effectively changes the design conditions of the air conditioning equipment to 79°F and 66°F wb, which results in a load reduction of 3.7 tons for each 1000 cfm of outside air. This can be seen in Figure 3a and 3b.

<table>
<thead>
<tr>
<th>OPERATING CONDITIONS</th>
<th>Outdoor Air = 95°F 78°F wb</th>
<th>Return Air = 75°F 63°F wb</th>
<th>No Return Air in Supply Air; EATR = 0</th>
<th>Outdoor Air equal Supply Air flow; OACF = 1</th>
</tr>
</thead>
</table>

**Figure 2. Humidity Level Effect**

**Figure 3a. Energy Savings vs. Airflow with AAONAIRE Energy Recovery**
The enthalpy difference between the outside air and the air entering the cooling coil is 10.3 Btu/lb. The average density of the air streams is 0.071 lb/cu.ft. At 1000 cu.ft./min the energy saved is:

\[
(10.3 \frac{Btu}{lb}) \left(0.071 \frac{lb}{ft^3}\right) \left(1000 \frac{ft^3}{min}\right) \left(60 \frac{min}{hr}\right) = 43,878 \frac{Btu}{hr} = 3.7 \text{ tons}
\]

*Figure 3b. Psychrometric Chart for 80% Effective Energy Recovery*
The psychrometric chart of Figure 4 shows the temperature and humidity moderating affect of the AAONAIRE energy recovery system. This Figure assumes a 75% effective total energy wheel.

The center point of the chart represents the return air conditions entering the AAONAIRE energy recovery system. The differing outside air conditions, represented by circles, are shown at various points on the psychrometric chart. The resultant air properties entering the HVAC equipment is represented by diamonds. In all cases, heating and cooling requirements are significantly reduced with the addition of the AAONAIRE energy recovery system.

![Figure 4. Energy Savings with AAONAIRE Energy Recovery](chart.png)

**APPLICATIONS**

In most comfort air conditioning applications, the AAONAIRE energy recovery ventilator 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.

The AAONAIRE energy recovery devices for comfort air conditioning applications may be sensible heat exchange devices (HRVs - transferring sensible energy only) or energy exchange devices (ERVs - transferring both sensible energy and moisture).

<table>
<thead>
<tr>
<th>TYPICAL COMFORT AIR CONDITIONING APPLICATIONS</th>
</tr>
</thead>
<tbody>
<tr>
<td>K-12 Schools</td>
</tr>
<tr>
<td>Universities</td>
</tr>
<tr>
<td>Hospitals</td>
</tr>
<tr>
<td>Assisted Care Facilities</td>
</tr>
<tr>
<td>Medical Office Buildings</td>
</tr>
<tr>
<td>Strip Malls</td>
</tr>
<tr>
<td>Department Stores</td>
</tr>
<tr>
<td>Restaurants</td>
</tr>
<tr>
<td>Animal ventilation</td>
</tr>
<tr>
<td>Plant ventilation</td>
</tr>
<tr>
<td>Convenience Stores</td>
</tr>
<tr>
<td>Swimming pools</td>
</tr>
<tr>
<td>Locker rooms</td>
</tr>
<tr>
<td>Residential</td>
</tr>
<tr>
<td>Lodging</td>
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<tr>
<td>Hotel/Motel</td>
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<tr>
<td>Dormitories</td>
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<tr>
<td>Manufacturing</td>
</tr>
<tr>
<td>Religious &amp; Worship</td>
</tr>
<tr>
<td>Federal Buildings</td>
</tr>
<tr>
<td>Libraries</td>
</tr>
<tr>
<td>Office Properties</td>
</tr>
</tbody>
</table>

Table 1. Typical comfort air conditioning applications
Heat Recovery Ventilators
AAONAIRe HRVs are suitable when outside air humidity is low and latent space loads are high for most of the year, and also for use with swimming pools, locker rooms and indirect evaporative coolers.

Energy Recovery Ventilators
AAONAIRe ERVs are suitable for applications in schools, offices, residences and other applications that require year-round economical preheating and/or precooling of outside supply air.

PERFORMANCE RATINGS
Standard laboratory rating tests and predictive computer models give AAONAIRe exchanger performance values for (1) heat transfer; (2) moisture transfer; (3) cross-stream air transfer; (4) average exhaust mass airflow, and (5) supply mass airflow leaving the exchanger. Effectiveness ratios for heat and mass water vapor transfer have been separately determined by rating tests in a laboratory that is staffed and instrumented to meet requirements of ASHRAE Standard 84 and AHRI Standard 1060.

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

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

AHRI Standard 1060 requires balanced airflow rates (see Figure 5) and the following conditions:

Winter:
Outside air at $t_1 = 35^\circ F$ and $t_{w_1} = 33^\circ F$
Inside (room) air at $t_3 = 67^\circ F$ and $t_{w_3} = 58^\circ F$ and $p_2 - p_3 = 0$

Summer:
Outside air at $t_1 = 95^\circ F$ and $t_{w_1} = 78^\circ F$
Inside (room) air at $t_3 = 75^\circ F$ and $t_{w_3} = 63^\circ F$ and $p_2 - p_3 = 0$

Figure 5. Air Leakage in Energy Recovery Units
Balanced mass airflows, as required for some ASHRAE and AHRI standard test methods, are rarely achieved in field operation for air handling systems. Fans are nearly constant-volume devices usually designed to run at a preset rpm. Significantly more mass airflow will be transported in cold (winter) conditions than in hot (summer) conditions.

For estimating changes in exchanger performance factors at each operating condition, ASHRAE Standard 84 specifies knowledge of seven performance factors (i.e., \( \Delta P_s \), \( \Delta P_e \), \( \varepsilon_s \), \( \varepsilon_e \), \( \varepsilon_{ETR} \), and OACF), but AHRI Standard 1060 certifies performance at only a few standard operating conditions. At other operating conditions, these performance factors must be extrapolated using accepted correlations.

Variables that can affect these performance factors for total energy transfer or sensible heat transfer devices include (1) water vapor partial-pressure differences; (2) heat transfer area; (3) air velocity or mass flow rates through the heat exchangers; (4) airflow arrangement or geometric configuration, or characteristic dimension of the flow passage through the recovery ventilator; and (6) method of frost control.

Current testing standards do not validate exchanger performance for testing conditions that require freezing or condensing temperatures, unbalanced airflow ratios, high pressure differentials, or air leakage rates based on varying the inputs.

**Guideline V**

Guideline V, Calculating the efficiency of energy recovery ventilation and its effect on building HVAC system efficiency, exists to establish a method of calculating the energy efficiency of applied energy recovery ventilation components and the energy efficiency of packaged heating, ventilating, and/or air conditioning systems utilizing such components. Guideline V uses the results of tests based on ASHRAE Standard 84 and AHRI Standard 1060. Guideline V provides a method of calculating the effect of energy recovery ventilation on the cooling and heating system overall efficiency.

**Recovery Energy Efficiency**

AHRI Guideline V provides a procedure for calculating the efficiency of energy recovery ventilation and its effect on efficiency and sizing of building HVAC systems for a given operating condition. Some additional information will be needed to find the Recovery Energy Efficiency (RER) and the Combined Energy Efficiency (CEF) of the complete AAONAIREE Energy Recovery Unit.

\[
RER = \frac{\text{Net conditioning delivered}}{\text{Total electric power consumed}}
\]

Where the net space conditioning energy can be heating, humidification, cooling, dehumidification or a combination thereof and the total electric power consumed includes the power required to move air through both sides of the heat exchanger as well as any additional power, such as the ERV or HRV drive motor.

The power required to move air through the heat exchanger is a function of both the pressure drop through the heat exchanger, as well as the motor and fan efficiency of the air-moving device. The power required to rotate an AAONAIREE heat exchanger can be measured directly from the small drive motor. Note that this energy is generally so small relative to the other energies as to be inconsequential.
The RER is calculated from the effectiveness and pressure drop for the heat exchanger and the air moving efficiency of the fan/motor combination according to the following expression:

\[
RER_{\text{Total}} = \frac{\epsilon_{\text{net}}\dot{m}_{\text{min}}(h_1 - h_2)}{P_{\text{wr blwr}} + P_{\text{wr comp}}}
\]

Where:

\(\epsilon_{\text{net}}\) = Net effectiveness (sensible, latent, or total, as applicable), as defined in AHRI Standard 1060 and determined in accordance with AHRI Standard 1060.

\(\dot{m}_s\) = Mass flow rate of supply air (at measurement station 2 as defined in ASHRAE Standard 84 and as illustrated in Figure 5), in units of mass of dry air per unit time.

\(\dot{m}_e\) = Mass flow rate of exhaust air (at measurement station 3 as defined in ASHRAE Standard 84 and as illustrated in Figure 5), in units of mass of dry air per unit time.

\(\dot{m}_{\text{min}}\) = The lesser of \(\dot{m}_s\) and \(\dot{m}_e\)

\(h_1\) = Total enthalpy of outdoor air entering the AAHX (measurement Station 1 as defined in ASHRAE Standard 84 and as illustrated in Figure 5).

\(h_2\) = Total enthalpy of return air entering the AAHX (measurement Station 3 as defined in ASHRAE Standard 84 and as illustrated in Figure 5).

\(P_{\text{wr blwr}}\) = P = Blower power

\(P_{\text{wr comp}}\) = Direct power input to the AAHX component

Where:

\[P = \frac{Q\Delta p}{635\eta_f}\]

**Note:** When the net effectiveness and pressure drops are determined according to AHRI-1060, it is for specific supply and exhaust air flow rates. In any air-to-air heat exchanger, the effectiveness decreases and the pressure drop increases as the air flow rates increase. Therefore, the preceding expressions for the RER are valid only when the effectiveness has been determined at the supply and exhaust air flow rates of interest.

<table>
<thead>
<tr>
<th>INFORMATION NEEDED FOR CALCULATIONS</th>
<th>INFORMATION FOR SAMPLE CALCULATION</th>
</tr>
</thead>
<tbody>
<tr>
<td>Energy Recovery Effectiveness ((\epsilon_{\text{cooling}}))</td>
<td>Sensible = 71%, Latent = 64%, Total=68%</td>
</tr>
<tr>
<td>Supply Air Flow</td>
<td>4000 ft³/min</td>
</tr>
<tr>
<td>Exhaust Air Flow</td>
<td>4000 ft³/min</td>
</tr>
<tr>
<td>(\Delta p) across Energy Recovery Component</td>
<td>1.25 in. w.c. (5.40 lb/ft²)</td>
</tr>
<tr>
<td>Fan/Motor Efficiency</td>
<td>0.60</td>
</tr>
<tr>
<td>Supply Air Enthalpy ((h_s))</td>
<td>92°F, 74°F wb = 37.9 Btu/lb</td>
</tr>
<tr>
<td>Exhaust Air Enthalpy ((h_e))</td>
<td>75°F, 63°F wb = 28.7 Btu/lb</td>
</tr>
<tr>
<td>(P_{\text{wr comp}})</td>
<td>50 W</td>
</tr>
</tbody>
</table>

Table 2. Combined Energy Efficiency Data
Combined Energy Efficiency

When an air-to-air heat exchanger is combined with a unitary air conditioner, (i.e. AAOAIRE) the air-to-air heat exchanger provides a portion of the total cooling capacity and the vapor compression cycle of the unitary air conditioner provides the rest. Consistent with the basic principle,

\[ EER = \frac{Net\ Cooling\ Capacity}{Total\ Power\ Consumed} \]

the cooling system EER of a unitary air conditioner with an Air-to-Air Heat eXchanger (AAHX) cooling component can be defined as:

\[ SysEER = \frac{AAHX\ Net\ Cooling\ Capacity + Unitary\ Net\ Cooling\ Capacity}{AAHX\ Electric\ Power\ Consumption + Unitary\ Electric\ Power\ Consumption} \]
The combined cooling system efficiency (CEF_{cooling}) is calculated from the efficiency of the air-to-air heat exchanger (RER) and the efficiency of the packaged equipment (EER_{unitary}) according to the following expression:

\[
CEF = \frac{1}{RER_{AAHX}} + \frac{(1-Y)}{EER_{unitary}}
\]

Where:

\[
Y = \frac{Net \, Cooling \, Capacity_{AAHX}}{Net \, Cooling \, Capacity_{AAHX} + Net \, Cooling \, Capacity_{unitary}}
\]

and RER is expressed in \(\frac{\text{Btu}}{\text{w} \cdot \text{h}}\).

**Note:** RER can be calculated on the basis of total energy recovery, latent recovery or sensible recovery effectiveness. The selection of the RER basis will depend on the analysis being conducted: use total for cooling and dehumidification, latent for dehumidification only and sensible for cooling without dehumidification.

For an AAONAIRE unit with a direct expansion system of EER=12 and where the ERV component (AAHX) is handling 30% of the system load at design conditions, the CEF_{cooling} is given by

\[
CEF_{cooling} = \frac{1}{RHR_{AAHX}} + \frac{(1-Y)}{EER_{unitary}}
\]

\[
CEF_{cooling} = \frac{1}{0.3 \cdot \frac{1}{67.14} + \frac{(1-0.3)}{12}}
\]

\[
CEF_{cooling} = 15.9 \frac{\text{Btu}}{\text{w} \cdot \text{h}}.
\]

**ECONOMIC CONSIDERATIONS**

Initial installed HVAC system cost is often lower for AAONAIRE 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 cost. The installed cost of heat recovery systems becomes lower per unit of flow as the amount of outside air used for ventilation increases.

With only that addition of an energy recovery wheel, the AAONAIRE unit is 33% more efficient than the standalone DX system.
AAONAIRE energy recovery cost benefits are best evaluated considering all capital, installation, operating, and energy-saving costs over the equipment life under normal operating conditions in terms of the life-cycle cost. As a rule, neither the most efficient nor the least expensive energy recovery device will be most economical. The absolute cost of energy and relative costs of various energy forms are major economic factors. High energy costs favor high levels of energy recovery. In regions where electrical costs are high relative to fuel prices, heat recovery devices with low pressure drops are preferable.

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.

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.

**Payback Analysis**

The annual energy savings and payback of the AAONAIRE energy recovery device can be demonstrated with a few simple calculations. Annual energy savings are based on climatic conditions. The analysis that will be done uses the format of “Bin and Degree Hour Weather Data for Simplified Energy Calculations”, Dry Bulb Temperature and Mean Coincident Wet Bulb (MCWB) from ASHRAE.

The recovered energy savings can be hand calculated, using equations discussed in this document, or selection software can be used to provide the recovered energy savings.

<table>
<thead>
<tr>
<th>INFORMATION NEEDED FOR CALCULATIONS</th>
<th>INFORMATION FOR SAMPLE CALCULATION</th>
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</thead>
<tbody>
<tr>
<td>Location</td>
<td>Chicago, IL</td>
</tr>
<tr>
<td>AAONAIRE Operating Schedule</td>
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</tr>
<tr>
<td>Net Outdoor Airflow Rate</td>
<td>4000 cfm</td>
</tr>
<tr>
<td>Net Indoor Airflow Rate</td>
<td>4000 cfm</td>
</tr>
<tr>
<td>Electric Utility Rate</td>
<td>$0.10 per kW-h</td>
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<tr>
<td>Gas Utility Rate</td>
<td>$10 per MMBtu</td>
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<tr>
<td>Electric Heating Efficiency</td>
<td>COP = 1</td>
</tr>
<tr>
<td>Gas Heating Efficiency</td>
<td>82% efficiency</td>
</tr>
<tr>
<td>Packaged Equipment Cooling Efficiency</td>
<td>COP = 2.9 (EER = 10)</td>
</tr>
<tr>
<td>Energy Recovery Effectiveness</td>
<td>Sensible = 71%, Latent = 64%, Total ($\epsilon_{cooling}$) = 68%</td>
</tr>
<tr>
<td>ΔP across Energy Recovery Component</td>
<td>1.25 in. w.c.</td>
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</tbody>
</table>

*Table 3. AAONAIRE Sample Payback Analysis*
### Table 4. Cooling Savings for Chicago, IL

<table>
<thead>
<tr>
<th>DRY BULB</th>
<th>MCWB</th>
<th>BIN OCCURRENCE</th>
<th>ENTHALPY</th>
<th>VENTILATION LOAD W/O AAONAIRES</th>
<th>AAONAIRES RECOVERED LOAD</th>
<th>NET VENTILATION LOAD</th>
<th>ANNUAL COOLING SAVINGS</th>
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<tr>
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<tr>
<td>F</td>
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### Table 5. Heating Savings for Chicago, IL

<table>
<thead>
<tr>
<th>DRY BULB</th>
<th>MCWB</th>
<th>BIN OCCURRENCE</th>
<th>ENTHALPY</th>
<th>VENTILATION LOAD W/O AAONAIRES</th>
<th>AAONAIRES RECOVERED LOAD</th>
<th>NET VENTILATION LOAD</th>
<th>ANNUAL HEATING SAVINGS</th>
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<td>41.87</td>
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<td>16.23</td>
<td>23,397</td>
<td>16,690</td>
<td>6,707</td>
<td>204</td>
</tr>
<tr>
<td>42</td>
<td>38.7</td>
<td>268</td>
<td>14.74</td>
<td>36,586</td>
<td>26,047</td>
<td>10,539</td>
<td>318</td>
</tr>
<tr>
<td>37</td>
<td>33.76</td>
<td>426</td>
<td>12.57</td>
<td>73,970</td>
<td>52,284</td>
<td>21,686</td>
<td>638</td>
</tr>
<tr>
<td>32</td>
<td>29.03</td>
<td>377</td>
<td>10.58</td>
<td>78,345</td>
<td>55,176</td>
<td>23,169</td>
<td>673</td>
</tr>
<tr>
<td>27</td>
<td>24.87</td>
<td>215</td>
<td>8.94</td>
<td>50,711</td>
<td>35,686</td>
<td>15,025</td>
<td>435</td>
</tr>
<tr>
<td>22</td>
<td>20.88</td>
<td>192</td>
<td>7.47</td>
<td>50,180</td>
<td>35,320</td>
<td>14,860</td>
<td>431</td>
</tr>
<tr>
<td>17</td>
<td>15.85</td>
<td>122</td>
<td>5.73</td>
<td>35,541</td>
<td>24,988</td>
<td>10,553</td>
<td>305</td>
</tr>
<tr>
<td>12</td>
<td>10.37</td>
<td>60</td>
<td>3.95</td>
<td>19,282</td>
<td>13,542</td>
<td>5,740</td>
<td>165</td>
</tr>
<tr>
<td>7</td>
<td>5.33</td>
<td>32</td>
<td>2.42</td>
<td>11,176</td>
<td>7,849</td>
<td>3,327</td>
<td>96</td>
</tr>
<tr>
<td>2</td>
<td>0.93</td>
<td>21</td>
<td>1.13</td>
<td>7,924</td>
<td>5,570</td>
<td>2,354</td>
<td>68</td>
</tr>
<tr>
<td>-2</td>
<td>-2.89</td>
<td>14</td>
<td>0.04</td>
<td>5,272</td>
<td>3,712</td>
<td>1,561</td>
<td>45</td>
</tr>
<tr>
<td>-7</td>
<td>-8.00</td>
<td>14</td>
<td>-1.36</td>
<td>5,598</td>
<td>3,942</td>
<td>1,656</td>
<td>48</td>
</tr>
<tr>
<td></td>
<td></td>
<td>3190</td>
<td></td>
<td>427,192</td>
<td>302,516</td>
<td>124,675</td>
<td>$3,689</td>
</tr>
</tbody>
</table>

Table 4. Cooling Savings for Chicago, IL

Table 5. Heating Savings for Chicago, IL
The annual savings in each table are found using the following equations:

| Cooling Savings | = | AAONAIRE Recovered Load | x | MBtu to kW-h conversion | x | Utility Rate | x | $\frac{1}{\text{Efficiency}}$ | = | $$$
|---|---|---|---|---|---|---|---|---|---|
| Cooling Savings | = | 30,303 MBtu | x | $0.2931 \frac{kW-h}{MBtu}$ | x | $0.10 \text{ per kW-h}$ | x | $\frac{1}{2.9}$ | = | $303$
| Heating Savings | = | AAONAIRE Recovered Load | x | MMBtu to MBtu conversion | x | Utility Rate | x | $\frac{1}{\text{Efficiency}}$ | = | $$$
| Heating Savings | = | 302,516 MBtu | x | $\frac{\text{MBtu}}{1000 \text{ MBtu}}$ | x | $10 \text{ per MMBtu}$ | x | $\frac{1}{0.82}$ | = | $3,689$

Table 6. Annual Energy Savings for Chicago, IL

For this example in Chicago, IL, the AAONAIRE would provide an annual savings of $303 + $3,689 = $3,992. If electric heat had been used instead of gas heat, the savings for electric heat could be found utilizing the same calculation method.

| Heating Savings | = | 302,516 MBtu | x | $0.2931 \frac{kW-h}{MBtu}$ | x | $0.10 \text{ per kW-h}$ | x | $\frac{1}{1.0}$ | = | $8,864$

The payback for the Chicago, IL example can be shown in the following worksheet.

**Payback Analysis**

<table>
<thead>
<tr>
<th>COOLING LOAD</th>
<th>Without AAONAIRE</th>
<th>With AAONAIRE</th>
</tr>
</thead>
<tbody>
<tr>
<td>Peak Ventilation Cooling Load (tons)</td>
<td>14.1</td>
<td>4.5</td>
</tr>
<tr>
<td>Space Cooling Load (tons)</td>
<td>50.0</td>
<td>50.0</td>
</tr>
<tr>
<td>Total Cooling Load (tons)</td>
<td>64.1</td>
<td>54.5</td>
</tr>
<tr>
<td>Cooling Equipment Reduction (tons)</td>
<td>9.6</td>
<td></td>
</tr>
</tbody>
</table>

Table 7. AAONAIRE Cooling Load Reduction

<table>
<thead>
<tr>
<th>HEATING LOAD</th>
<th>Without AAONAIRE</th>
<th>With AAONAIRE</th>
</tr>
</thead>
<tbody>
<tr>
<td>Peak Ventilation Heating Load (MBtu/hr)</td>
<td>400</td>
<td>118</td>
</tr>
<tr>
<td>Space Heating Load (MBtu/hr)</td>
<td>480</td>
<td>480</td>
</tr>
<tr>
<td>Total Heating Load (MBtu/hr)</td>
<td>880</td>
<td>598</td>
</tr>
<tr>
<td>Heating Equipment Reduction (MBtu/hr)</td>
<td>282</td>
<td></td>
</tr>
</tbody>
</table>

Table 8. AAONAIRE Heating Load Reduction
Energy recovery unit installed costs vary by application with ranges from $0.25/cfm to $5.00/cfm. Because the AAONAIRE is factory installed as an integral part of the rooftop unit it is always a cost effective alternative to separate, standalone ERV units.

The AAONAIRE energy recovery device would have a payback of less than one year and continue to save thousands of dollars per year for the life of the unit.

Figure 6 shows the energy savings for the building with and without the AAONAIRE energy recovery ventilator. Both figures show a monthly energy savings throughout the year.

**PAYBACK ANALYSIS**

<table>
<thead>
<tr>
<th>Description</th>
<th>Cost (Installed)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cooling cost ($/ton installed)</td>
<td>$1,000</td>
</tr>
<tr>
<td>Fossil fuel heating cost ($/MBtu/hr installed)</td>
<td>$4.50</td>
</tr>
<tr>
<td>AAONAIRE ($/cfm installed)</td>
<td>$2.25</td>
</tr>
<tr>
<td>Avoided Cooling Equipment Cost</td>
<td>+$9,600</td>
</tr>
<tr>
<td>Avoided Heating Equipment Cost</td>
<td>+$1,269</td>
</tr>
<tr>
<td>AAONAIRE First Cost</td>
<td>-$9,000</td>
</tr>
<tr>
<td>Net AAONAIRE First Cost</td>
<td>$1,869</td>
</tr>
<tr>
<td>Annual Savings</td>
<td>$3,992</td>
</tr>
</tbody>
</table>

Payback Period = \( \frac{1,869}{3,992} \) = .47 years

Table 9. AAONAIRE Payback Analysis for Chicago, IL

![Figure 6. AAONAIRE Heating and Cooling Energy Savings for Chicago, IL](image)
Using the same AAONAIRE unit and the following operating conditions, heating and cooling savings can be calculated for cities with differing climatic conditions.

<table>
<thead>
<tr>
<th>INFORMATION NEEDED FOR CALCULATIONS</th>
<th>INFORMATION FOR SAMPLE CALCULATION</th>
</tr>
</thead>
<tbody>
<tr>
<td>AAONAIRE Operating Schedule</td>
<td>4am – 8pm, 5 days per week</td>
</tr>
<tr>
<td>Net Outdoor Airflow Rate</td>
<td>4000 cfm</td>
</tr>
<tr>
<td>Net Indoor Airflow Rate</td>
<td>4000 cfm</td>
</tr>
<tr>
<td>Electric Utility Rate</td>
<td>$0.10 per kW-h</td>
</tr>
<tr>
<td>Fossil Fuel Utility Rate</td>
<td>$10 per MMBtu</td>
</tr>
<tr>
<td>Packaged Equipment Cooling Efficiency</td>
<td>COP = 2.9 (EER = 10.0)</td>
</tr>
<tr>
<td>Heat Pump Heating Efficiency</td>
<td>COP = 4.0 (EER = 13.6)</td>
</tr>
<tr>
<td>Fossil Fuel Heating Efficiency</td>
<td>82% efficiency</td>
</tr>
<tr>
<td>Electric Resistance Heating Efficiency</td>
<td>COP = 1</td>
</tr>
</tbody>
</table>

**Table 10. AAONAIRE Heating and Cooling Savings Inputs**

<table>
<thead>
<tr>
<th>City</th>
<th>Cooling Savings</th>
<th>Heat Pump Savings</th>
<th>Fossil Fuel Savings</th>
<th>Electric Resistance Savings</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chicago, IL</td>
<td>$303</td>
<td>$2,216</td>
<td>$3,689</td>
<td>$8,864</td>
</tr>
<tr>
<td>New York, NY</td>
<td>$310</td>
<td>$1,856</td>
<td>$3,091</td>
<td>$7,425</td>
</tr>
<tr>
<td>Miami, FL</td>
<td>$2,686</td>
<td>$80</td>
<td>$134</td>
<td>$322</td>
</tr>
<tr>
<td>Atlanta, GA</td>
<td>$682</td>
<td>$1,061</td>
<td>$1,767</td>
<td>$4,245</td>
</tr>
<tr>
<td>Minneapolis, MN</td>
<td>$286</td>
<td>$2,723</td>
<td>$4,534</td>
<td>$10,894</td>
</tr>
<tr>
<td>Dallas, TX</td>
<td>$1,190</td>
<td>$776</td>
<td>$1,292</td>
<td>$3,104</td>
</tr>
<tr>
<td>Houston, TX</td>
<td>$1,851</td>
<td>$499</td>
<td>$830</td>
<td>$1,995</td>
</tr>
<tr>
<td>Denver, CO</td>
<td>$231</td>
<td>$2,068</td>
<td>$3,444</td>
<td>$8,273</td>
</tr>
<tr>
<td>Los Angeles, CA</td>
<td>$88</td>
<td>$313</td>
<td>$521</td>
<td>$1,252</td>
</tr>
<tr>
<td>Seattle, WA</td>
<td>$29</td>
<td>$1,508</td>
<td>$2,512</td>
<td>$6,032</td>
</tr>
</tbody>
</table>

**Table 11. AAONAIRE Annual Savings for Select Cities**
The following chart shows an estimated payback period for the continental United States. The following assumptions were made in the creation of the chart:

- Office building with HVAC system operating 16 hr/day, 5 days/week
- AAONaire installed cost of $3.60 per cfm
- Air conditioning equipment installed cost of $1,000 per ton
- Energy costs of $0.10 per kW-hr and $10 per MMBtu
- Energy recovery effectiveness of 68%

**Figure 7. AAONaire Payback Map**

**RATE OF ENERGY TRANSFER**

The second law of thermodynamics states that heat energy always transfers from a region of high temperature to one of low temperature. This law can be extended to say that mass transfer always occurs from a region of high vapor pressure to one of low vapor pressure. The AAONaire ERV facilitates this transfer with a polymer energy transfer media that conducts heat and permanently bonded silica gel desiccant that is permeable to water vapor. Moisture is transferred when there is a difference in vapor pressure between the two airstreams.

The rate of energy transfer depends on the operating conditions and the intrinsic characteristics of the energy exchanger. As in a conventional heat exchanger, energy transfer between the airstreams is driven by cross-stream dry-bulb temperature differences. Energy is also transported piggyback-style between the streams by cross-stream mass transfer, which may include air, gases, and water vapor. In another mode of energy transfer, water vapor condenses into liquid in one of the two airstreams of the exchanger. The condensation process liberates the latent
heat of condensation, which is transferred to the other stream as sensible heat; this two-step process is also called latent heat transfer.

Latent energy transfer between airstreams occurs only when moisture is transferred from one airstream to another without condensation, thereby maintaining the latent heat of condensation. Once moisture has crossed from one airstream to the other, it may either remain in the vapor state or condense in the second stream, depending on the temperature of that stream.

AAONAI RE rotating energy recovery units are used because of their moisture exchange function. Passage of air or other gases across the exchanger is a negative consequence. As well, some cross-stream mass transfer may occur through leakage even when such transfer is unintended. This may alter exchanger performance from its design value, but for most HVAC applications with exhaust air from occupied spaces, small transfers to the supply air are not important.

Heat transfer differs in principle from mass transfer. Heat transfer only occurs when there is a temperature difference. In the case of air-to-air exchange between the supply and exhaust airflow, heat transfer by conduction and convection only occurs when there is a temperature difference between these airstreams. The following facts about heat/mass exchanger performance should be recognized:

The effectiveness for moisture transfer may not equal the effectiveness for heat transfer. The total energy effectiveness may not equal either the sensible or latent effectiveness.

Net total energy transfer and effectiveness need careful examination when the direction of sensible (temperature-driven) transfer is opposite to that of latent (moisture or water vapor) transfer.

Using the airstream numbering convention shown in Figure 8, on a typical summer day, supply air at temperature, humidity, or enthalpy of \( x_1 \) and mass flow rate \( m_s \) enters the AAONAI RE ERV, while exhaust air from the conditioned space enters at conditions \( x_3 \) and \( m_s \). Because conditions at \( x_3 \) are lower than conditions at \( x_1 \), heat and mass transfer from the supply airstream to the exhaust airstream because of differences in temperature and vapor pressures across the separating wall. Consequently, the supply air exit properties decrease, while those of the exhaust air increase. Exit properties of these two streams can be calculated, knowing the flow rates and the effectiveness of the heat exchanger.

ASHRAE Standard 84 defines effectiveness as

\[
\varepsilon = \frac{\text{Actual Transfer of Moisture or Energy}}{\text{Maximum Possible Transfer Between Airstreams}}
\]

**Heat Recovery Ventilators**

From Figure 8, the sensible effectiveness \( \varepsilon_s \) of a AAONAI RE heat recovery ventilator is given as

\[
\varepsilon_s = \frac{q_s}{q_{s, max}} = \frac{m_s c_p (t_2 - t_1)}{C_{min} (t_2 - t_1)} = \frac{m_s c_p (t_3 - t_4)}{C_{min} (t_3 - t_1)}
\]

where \( q_s \) is the actual sensible heat transfer rate given by

\[
q_s = \varepsilon s q_{s, max}
\]

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

\[
q_{s, max} = 60 C_{min} (t_3 - t_1)
\]
where

\[\varepsilon_s = \text{sensible effectiveness}\]
\[t_1 = \text{dry-bulb temperature at location 1 in Figure 8, } ^\circ F\]
\[m_s = \text{supply dry air mass flow rate, lb/min}\]
\[m_e = \text{exhaust dry air mass flow rate, lb/min}\]
\[C_{\text{min}} = \text{smaller of } c_{ps}m_s \text{ and } c_{pe}m_e\]
\[c_{ps} = \text{supply moist air specific heat at constant pressure, Btu/lb·} ^\circ F\]
\[c_{pe} = \text{exhaust moist air specific heat at constant pressure, Btu/lb·} ^\circ F\]

Assuming no water vapor condensation in the AAONAIRe HRV, the leaving supply air condition is

\[t_2 = t_1 - \varepsilon_s \frac{C_{\text{min}}}{m_s c_{ps}} (t_1 - t_3)\]

and the leaving exhaust air condition is

\[t_4 = t_3 - \varepsilon_s \frac{C_{\text{min}}}{m_e c_{pe}} (t_1 - t_3)\]

The preceding equations assume steady-state operating conditions; no heat or moisture transfer between the heat exchanger and its surroundings; no cross-leakage, and no energy gains from motors, fans, or frost control devices. Furthermore, condensation or frosting does not occur or is negligible. These assumptions are generally true for larger commercial AAONAIRe HRV applications. Note that the HRV only allows transfer of sensible heat energy associated with heat transfer because of a temperature difference between the airstreams. These equations apply even in winter, if there is no condensation in the HRV.

The sensible heat energy transfer \(q_s\) from the AAONAIRe heat recovery ventilator can be estimated from

where

\[q_s = 60m_s c_{ps} (t_2 - t_1) = 60Q_s \rho_s c_{ps} (t_2 - t_1)\]
\[q_s = 60m_e c_{pe} (t_4 - t_3) = 60Q_e \rho_e c_{pe} (t_4 - t_3)\]
\[q_s = 60\varepsilon_s m_{\text{min}} c_p (t_1 - t_3)\]

\[Q_s = \text{volume flow rate of supply air, cfm}\]
\[Q_e = \text{volume flow rate of exhaust air, cfm}\]
\[\rho_s = \text{density of dry supply air, lb/ft}^3\]
\[\rho_e = \text{density of dry exhaust air, lb/ft}^3\]
\[t_p, t_m, t_s, t_e = \text{inlet and exit temperatures of supply and exhaust airstreams, respectively}\]
\[m_{\text{min}} = \text{smaller of } m_s \text{ and } m_e\]
Sensible heat exchangers (HRVs) can be used in virtually all cases, especially for swimming pool, paint booth, and reheat applications.

**Energy Recovery Ventilators**

The AAONAIRE ERV allows the transfer of both sensible and latent heat, the latter due to the difference in water vapor pressures between the airstreams. (For a full explanation of the following latent transfer equations see Appendix.)

From Figure 8, assuming no condensation in the AAONAIRE ERV, the latent effectiveness $\varepsilon_L$ of an energy recovery ventilator is given as

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

where $q_L$ is the actual latent heat transfer rate given by

$$
q_L = \varepsilon_L q_{L,\text{max}}
$$

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

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

The total effectiveness $\varepsilon_t$ of an AAONAIRE energy recovery ventilator is given as

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

where $q_t$ is the actual total heat transfer rate given by

$$
q_t = \varepsilon_t q_{t,\text{max}}
$$

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

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

where

- $\varepsilon_t$ = total effectiveness
- $h$ = enthalpy at locations indicated in Figure 8, Btu/lb
- $m_s$ = supply dry air mass flow rate, lb/min
- $m_e$ = exhaust dry air mass flow rate, lb/min
- $m_{\text{min}}$ = smaller of $m_s$ and $m_e$

The leaving supply air enthalpy condition is

$$h_s = h_1 - \varepsilon_t \frac{m_{\text{min}}}{m_s} (w_1 - w_3)$$

and the leaving exhaust air enthalpy condition is

$$h_4 = h_3 - \varepsilon_t \frac{m_{\text{min}}}{m_e} (h_1 - h_3)$$

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

$$q_L = 60 \varepsilon_L m_{\text{min}} h_{fg}(w_1 - w_3)$$
where
\[ h_f = \text{enthalpy of vaporization or heat of vaporization of water vapor, Btu/lb} \]
\[ w_1, w_2, w_3, w_4 = \text{inlet and exit humidity ratios of supply and exhaust airstreams, respectively} \]

The total energy transfer \( q_t \) between the streams is given by
\[ q_t = 60 \varepsilon_i m_{\text{min}} (h_{1s} - h_{3e}) \]

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

AAONAIREDVs can be used where there is an opportunity to transfer heat and mass (water vapor) (e.g., humid areas, schools, offices with large occupancies). Latent energy transfer can be positive or negative depending on the direction of decreasing vapor pressure. An airstream flowing through an ERV may gain heat energy (+\( q_s \)) from the adjoining stream, but will lose the latent energy (-\( q_L \)) if it transfers the water vapor to the adjoining stream, because of transfer of moisture. The total net energy gain is the difference between \( q_s \) and \( q_L \), as shown in Example 1.

Example 1. Total Heat Recovery in Summer

Return air at 75°F and 63°F wb (\( \rho = 0.073 \text{ lb/ft}^3 \)) with a flow rate of 10,600 cfm is used to precool 8500 cfm of supply outdoor air at 95°F and 81°F wb (\( \rho = 0.069 \text{ lb/ft}^3 \)) using an AAONAIRED total energy exchanger. The sensible and total effectiveness for this heat exchanger are 70% and 56.7%, respectively. Assuming \( EATR = 0 \) and \( OACF \approx 1 \), determine the leaving supply air conditions and energy recovered.
Solution:
1. Calculate the theoretical maximum heat transfer. The outdoor airstream is a lesser or limiting airstream for energy and moisture transfer. Determine entering airstream enthalpies and humidity ratio from psychrometric chart.

   Outside air (95°F db, 81°F wb) \( h_1 = 44.6 \text{ Btu/lb} \) \( w_1 = 0.0198 \text{ lb/lb} \)

   Return air (75°F db, 63°F wb) \( h_3 = 28.5 \text{ Btu/lb} \) \( w_3 = 0.0096 \text{ lb/lb} \)

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

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

   \[ q_{\text{max}} \text{(total energy)} = (60)(0.069 \text{ lb/ft}^3)(8500 \text{ ft}^3/\text{min})(44.6 - 28.5) \]
   \[ = 567,000 \text{ Btu/h} \]

2. Determine supply sensible and total effectiveness.

   The data given for the design conditions provide the following effectiveness ratios:

   \( \varepsilon_s = 70\% \)

   \( \varepsilon_t = 56.7\% \)

3. Calculate energy transfer at design conditions.

   \( q_t = (0.567)(567,000 \text{ Btu/h}) = 321,000 \text{ Btu/h total recovered} \)

   \( q_s = - (0.7)(169,000 \text{ Btu/h}) = -118,000 \text{ Btu/h sensible recovered} \)

   \[ q_t = q_s + q_{\text{lat}} \]

   Therefore,

   \[ q_{\text{lat}} = 203,000 \text{ Btu/h latent recovered} \]
4. Calculate leaving air conditions.

**Supply air conditions**

\[
t_2 = 95^\circ F + \frac{-118,000 \text{ Btu/hr}}{60 \left( 0.069 \frac{\text{lb}}{\text{ft}^3} \right) \left( 8500 \frac{\text{ft}^3}{\text{min}} \right) \left( 0.24 \frac{\text{Btu}}{\text{lb} \cdot ^\circ F} \right)} = 81^\circ F
\]

\[
h_2 = 44.6 \frac{\text{Btu}}{\text{lb}} + \frac{-321,000 \text{ Btu/hr}}{60 \left( 0.069 \frac{\text{lb}}{\text{ft}^3} \right) \left( 8500 \frac{\text{ft}^3}{\text{min}} \right)} = 35.5 \frac{\text{Btu}}{\text{lb}}
\]

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

**Exhaust air conditions**

\[
t_4 = 75^\circ F + \frac{+118,000 \text{ Btu/hr}}{60 \left( 0.073 \frac{\text{lb}}{\text{ft}^3} \right) \left( 10,600 \frac{\text{ft}^3}{\text{min}} \right) \left( 0.24 \frac{\text{Btu}}{\text{lb} \cdot ^\circ F} \right)} = 86.2^\circ F
\]

\[
h_4 = 28.5 \frac{\text{Btu}}{\text{lb}} + \frac{+321,000 \text{ Btu/hr}}{60 \left( 0.073 \frac{\text{lb}}{\text{ft}^3} \right) \left( 10,600 \frac{\text{ft}^3}{\text{min}} \right)} = 35.4 \frac{\text{Btu}}{\text{lb}}
\]

From the psychrometric chart, the exhaust humidity ratio and wet bulb temperature are found to be \( w_4 = 0.0134 \) and \( t_{w4} = 71.6^\circ F \).
AAONAIRE rotary air-to-air energy exchangers

The AAONAIRE is a rotary air-to-air energy exchanger which has a revolving cylinder filled with an air-permeable medium having a large internal surface area. Adjacent supply and exhaust airstreams each flow through half the exchanger in a counterflow pattern (Figure 9). Heat transfer media may be selected to recover sensible heat only, HRVs, or total (sensible plus latent) heat, ERVs.

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

Engineered Polymer vs. Aluminum Wheel Material

Aluminum as a material for energy recovery wheels has been used since the early 1900’s. It is accepted as a rugged material with good heat transfer characteristics, however, aluminum wheels have a very heavy weight and associated cost, problematic corrosion in salt air applications and problems with maintaining the desiccant coating over time due to loss of the bond between the aluminum and the epoxy material used to hold the desiccant.

AAONAIRE technology, in contrast to the older aluminum technology, makes use of an engineered polymer with desiccant permanently imbedded. The AAONAIRE energy transfer matrix design overcomes the problems associated with traditional aluminum wheels by providing lower cost, lighter weight, cleanable, permanently bonded desiccant that is completely corrosion resistant in seacoast applications. To further the maintainability, AAONAIRE wheels are made in segments which can be easily removed from the unit for cleaning outside the unit.

This feature eliminates costly and troublesome drain pans under the wheel, a requirement of the aluminum application. Because of the robust design and the easily cleanable segments, a life of 20-30 years with original factory performance can be achieved.
In addition to being non-corrosive, polymer is also flexible. If, during maintenance, a tool would impact the media, it can pass through the media wraps without damaging the heat transfer matrix. This same robustness is not found in the aluminum wheel design. The face of an aluminum wheel is similar to the face of a aluminum heat exchange coil; the fluted design is very delicate. If impacted with a tool of any sort, the flutes bend and block air from passing through the wheel and reducing the efficiency and effectiveness of the wheel.

Even though many aluminum wheel designs are very heavy, few aluminum wheels are segmented. Therefore the entire hub assembly must be taken apart to remove the wheel from the cassette. On larger units, the wheel cannot be removed from the unit at all. By contrast the AONAIRe cassette containing the polymer wheel slides out of the air handling or rooftop unit for easy service. Polymer wheels larger than 30 inches are segmented, with the weight of each segment ranging from 4-23 pounds. The segments can easily be removed without the use of tools.

**Silica gel**

Silica gel is an inert, highly porous solid adsorbent material that structurally resembles a rigid sponge. It has a very large internal surface composed of myriad microscopic cavities and a vast system of capillary channels that provide pathways connecting the internal microscopic cavities to the outside surface of the “sponge”.

The characteristic curve for adsorption of water on silica gel is shown in Figure 11 as % weight adsorbed versus relative humidity of the airstream in contact with the silica gel. The amount of water adsorbed rises almost linearly with increasing relative humidity until relative humidity reaches about 60%. It then plateaus out at about 40% adsorbed as relative humidity approaches 100%. The curve for molecular sieves, by contrast, rises rapidly to plateau at about 20% adsorbed at 20% relative humidity. This helps to explain why the molecular sieve is an excellent choice for regenerated applications such as desiccant cooling and dehumidification systems which are designed to reduce processed airstreams to very low relative humidities. On the other hand, silica gel has superior characteristics for the recovery of space comfort conditioning energy.

The use of silica gel on AONAIRe energy recovery ventilation applications involves a process cycle where the silica gel is alternately exposed to airstreams having nearly equal relative humidity somewhere in the mid range of this curve (typically between 40 and 60% for comfort cooling applications). When the airstream with the higher relative humidity passes over the silica gel coated wheel, moisture is adsorbed from the airstream into the silica gel. Then when the airstream with the lower relative humidity contacts the silica gel, moisture is desorbed (removed) from the silica gel and put into the airstream.
Adsorption is different than absorption. Absorption occurs when one substance has fully entered or permeated into the other. Adsorption is a surface phenomenon and occurs when one substance sticks to the surface of the other and does not change the matrix property of the substrate. Adsorption would be analogous to static cling. When a fabric sticks to a person, the person does not absorb the fabric, the person adsorbs the fabric.

**Sensible Heat Exchangers (HRVs) vs. Total Energy Exchangers (ERVs)**

Both heat recovery ventilators and energy recovery ventilators transfer sensible heat. HRVs transfer sensible heat only. ERVs transfer latent energy, or moisture, as well. The transfer of moisture is important because the latent portion of the outdoor air load exceeds 60% of the total load for many climates. The best way to illustrate the impact of latent energy transfer is by examining cases for summer operation. Example 2 is at summer full load conditions; at these conditions the outdoor temperature and humidity require maximum cooling. Example 3 is at summer part load conditions; at these conditions the cooling requirement is below maximum, but outdoor humidity is still high. Both examples will use an energy recovery effectiveness of 75% for both the ERV and the HRV.

**Example 2: Sensible and Total Energy Recovery**

*Analysis of energy recovery at full load determines the amount of air conditioning reduction that can be realized. For this example, outdoor air at full load is 95°F and 78°F wb. The room air is 75°F and 63°F wb.*
Solution:
The psychrometric chart shows the leaving air conditions for both devices. Notice that the process line for the HRV is horizontal (no moisture transfer), while the process line for the ERV is directly toward room air conditions. While both products have a sensible effectiveness of 75%, the total effectiveness (sensible & latent) of each is determined by the transfer of enthalpy, not sensible temperature only. The maximum possible enthalpy transfer in this case is 13.2 Btu/lb of air, which is the difference between outdoor air and room air (41.4-28.2). The ERV transferred 9.9 Btu/lb (41.4-31.5), which translates into a total effectiveness of 75%. The HRV, on the other hand, only transferred 3.7 Btu/lb (41.4-37.7), for a total effectiveness of 28%.

The total effectiveness of the HRV is low because it transfers only sensible heat. In this example, the total outdoor air load is 37% sensible and 63% latent. The HRV is 75% effective on the sensible portion and 0% effective on the latent portion. As a result, the HRV's total effectiveness is only 28% [(0.37 x 75%) + (0.62 x 0%)]. The equipment reduction will be only 1.3 tons for every 1000 cfm of outdoor air.

The total effectiveness of the ERV is 75% [(0.37 x 75%) + (0.62 x 75%)] because the wheel transfers both sensible and latent energy with virtually identical efficiencies. As a result, the wheel will reduce the air conditioning load by 3.5 tons for every 1000 cfm of outdoor air.
Example 3: Part Load Sensible and Total Energy Recovery

Consider the part load condition of 75°F and 72°F wb.

<table>
<thead>
<tr>
<th>POINT NAME</th>
<th>DRY BULB (°F)</th>
<th>WET BULB (°F)</th>
<th>RELATIVE HUMIDITY (%)</th>
<th>ENTHALPY (Btu/lb)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Outdoor Air</td>
<td>75</td>
<td>72</td>
<td>87</td>
<td>35.8</td>
</tr>
<tr>
<td>Return Air</td>
<td>75</td>
<td>63</td>
<td>50</td>
<td>28.2</td>
</tr>
<tr>
<td>Leaving Air - ERV</td>
<td>75</td>
<td>65</td>
<td>59</td>
<td>30.0</td>
</tr>
<tr>
<td>Leaving Air - HRV</td>
<td>75</td>
<td>72</td>
<td>87</td>
<td>35.8</td>
</tr>
</tbody>
</table>

Solution:

As the psychrometric chart illustrates, the 75% effective ERV will create leaving air conditions that are three quarters of the distance between outdoor air and return air conditions. Even though the dry bulb temperature did not change, 75% of the total energy was transferred (the outdoor load is 0% sensible and 100% latent). The HRV would have an effectiveness of 0% in this case.

Transferring latent energy at part load is extremely important for maintaining proper indoor humidity levels. Most systems operate at part load conditions much of the time. If the outdoor air is not pre-conditioned by a total energy wheel, the indoor humidity levels would likely climb above 60% rh. This is a real problem building owners and consulting engineers face today; humidity control is far more difficult than temperature control.
Example 2 & 3 Summary:
The ability of ERVs to transfer latent energy translates into two key benefits that HRVs do not offer:

1. At full load, the ERV will reduce the outdoor air load by 2 1/2 times more than a HRV. For comparison, an ERV will reduce the load by 3.5 tons per 1000 cfm of outdoor air, where a HRV will reduce the load by 1.3 tons.

2. At part load conditions, the ERV always pre-conditions the outdoor air to near room air conditions. This function is very important to maintaining indoor humidity at desired levels. HRVs, on the other hand, have little effect if the outdoor dry bulb temperature is near (within 7°F) room air dry bulb temperature.

Humidity control is important to control in humid environments. However, humidity transfer is not always desired. In some applications, such as indirect evaporative cooling, a HRV is much better suited.

Example 4. Indirect Evaporative Cooling Recovery
Return air at 86°F and 63°F wb (ρ = 0.072 lb/ft³) with a flow rate of 32,000 cfm is used to precool 32,000 cfm of supply outdoor air at 102°F and 68°F wb (ρ = 0.070 lb/ft³) using an heat recovery ventilator and indirect evaporative cooling. The evaporative cooler increases the exhaust air to 90% rh before it enters the AAONAIRE HRV. The sensible effectiveness is given as 78%. Assuming EATR = 0 and OACF = 1, determine the leaving supply air conditions and energy recovered.
<table>
<thead>
<tr>
<th>POINT NAME</th>
<th>DRY BULB (°F)</th>
<th>WET BULB (°F)</th>
<th>RELATIVE HUMIDITY (%)</th>
<th>ENTHALPY (Btu/lb)</th>
<th>DENSITY (lb/cu.ft.)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Return Air</td>
<td>86.0</td>
<td>62.8</td>
<td>26.1</td>
<td>28.2</td>
<td>0.072</td>
</tr>
<tr>
<td>Outdoor Air</td>
<td>102.0</td>
<td>68.0</td>
<td>15.9</td>
<td>32.1</td>
<td>0.070</td>
</tr>
<tr>
<td>Return Air - After Evap Cooler</td>
<td>64.0</td>
<td>63.0</td>
<td>94.8</td>
<td>28.6</td>
<td>0.074</td>
</tr>
<tr>
<td>Exhaust Air</td>
<td>92.0</td>
<td>71.8</td>
<td>37.6</td>
<td>35.4</td>
<td>0.070</td>
</tr>
<tr>
<td>Supply Air - HRV</td>
<td>72.4</td>
<td>57.8</td>
<td>40.8</td>
<td>24.9</td>
<td>0.074</td>
</tr>
<tr>
<td>Supply Air - ERV</td>
<td>72.4</td>
<td>64.1</td>
<td>64.3</td>
<td>29.3</td>
<td>0.073</td>
</tr>
</tbody>
</table>

Solution:
First, determine the return air condition entering the exchanger (i.e., after it is adiabatically cooled). Air at 86°F and 63°F wb cools to 64°F and 63°F wb as shown by the process line from ‘return air’ point to the ‘return air – after evap cooled’ point in the psychrometric chart for example 4. In this problem the volumetric flows are equal, but the mass flows are not.

1. Calculate the theoretical maximum heat transfer. This would be the maximum temperature difference between the outside air entering the wheel and the air entering the wheel after leaving the indirect evaporative cooler:

\[ q_{\text{max}} (\text{sensible}) = (60)(0.070 \text{ lb/ft}^2)(32,000 \text{ ft}^3/\text{min})(0.24 \text{ Btu/lb} \cdot ^\circ F) \times (102 - 64 \text{ } ^\circ F) \]
\[ = 1,226,000 \text{ Btu/h} \]

2. Calculate actual energy transfer at the design conditions.

\[ q_{\text{actual}} = (0.78)(1,226,000 \text{ Btu/h}) = 956,000 \text{ Btu/h} \text{ recovered} \]

3. Calculate leaving air conditions.
   a. Leaving supply air temperature is

\[ t_2 = 102^\circ F + \frac{-956,000 \text{ Btu/hr}}{60(0.070 \text{ lb/ft}^3)(32,000 \text{ ft}^3/\text{min})(0.24 \text{ Btu/lb} \cdot ^\circ F)} = 72.4^\circ F \]
b. Leaving exhaust air temperature is

\[ t_4 = 64^\circ F + \frac{+956,000 \text{ Btu}}{60 \left( 0.074 \frac{\text{lb}}{\text{ft}^3} \right) \left( 32,000 \frac{\text{ft}^3}{\text{min}} \right) \left( 0.24 \frac{\text{Btu}}{\text{lb} \cdot ^\circ F} \right)} = 92^\circ F \]

It can be seen from the chart and corresponding table that the HRV provides a lower relative humidity to the air conditioning unit than the ERV and therefore provides an energy saving benefit to the building. The air entering the cooling coil will have a relative humidity of 40.8% with the HRV, versus 64.3% with the ERV, the air conditioning unit will be required to perform little or no latent cooling (dehumidification) and will save energy using the HRV instead of an ERV.

Both ERVs and HRVs save energy. HRVs are ideally suited to applications where problems may be created by extreme differences in interior and exterior moisture levels and in which heat transfer is desired but humidity transfer is not (e.g., swimming pools, kitchens, indirect evaporative cooling). ERVs are suitable for applications in schools, offices, residences and other applications that require year-round economical preheating and/or precooling of outside supply air.

**TECHNICAL CONSIDERATIONS**

*Air Leakage*

Air leakage refers to any air that enters or leaves the supply or exhaust airstreams. Zero air leakage in either airstream would require \( m_1 \) to equal \( m_2 \) and \( m_3 \) to equal \( m_4 \). External air leakage occurs when the ambient air surrounding the energy recovery system leaks into (or exits) either or both airstreams. Cross-flow air leakage results from inadequate sealing construction between ambient and cross-stream seal interfaces. Internal air leakage occurs when holes or passages are open to the other airstream. Internal air leakage occurs when heat or energy exchanger design allows (1) tangential air movement in the wheel’s rotational direction and (2) airflow movement through holes in the barrier between airstreams. Under some pressure differentials, air leaks in and out of each airstream in nearly equal amounts, giving the illusion that there is no air leakage. Heat and water vapor transfer could appear to be greater than it actually is. Air leakage is seldom zero because external and internal air pressures are usually different, causing air to leak from higher-pressure regions to lower-pressure regions.

![Figure 13. AAONAIRE Rotary Air-to-Air Energy Exchanger](#)
Cross-flow air leakage is usually caused by pressure differentials between airstreams. Carryover air leakage (specific to wheels) is caused by continuous rotation of trapped exhaust air in cavities in the heat transfer surface, which reverses airflow direction as the wheel rotates and spills this exhaust air into the supply airstream.

Cross-flow leakage occurs because the differential static pressure across the two airstreams drives air from a higher to a lower static pressure region. Leakage primarily occurs due to the difference in static pressures between states 2 and 3 and/or between states 1 and 4, as shown in Figure 13. This is a major cause of cross-flow leakage, and underscores the importance of fan locations that circulate the airstreams. Cross-flow can also be caused by factors such as provisions for surging, geometrical irregularities, and local velocity distribution of the airstreams. The AAONAIRED design uses an optimum draw-thru, draw-thru design where the supply fan draws the first air stream through the wheel and the exhaust fan draws second air stream through the wheel as shown in Figure 13. This configuration allows balanced pressures between the two air streams.

Carryover occurs as air is entrained within the rotation medium and is carried into the other 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 as shown in Figure 14.

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

The quantitative estimate of the air leakage is expressed by two dimensionless parameters: the Exhaust Air Transfer Ratio (EATR) and Outside Air Correction Factor (OACF).

\[
EATR = \frac{c_2 - c_1}{c_3 - c_1}
\]

where \(c_1\), \(c_2\), and \(c_3\) are the concentrations of inert gas at states 1, 2 and 3, respectively as shown in Figure 15. Note that EATR represents an exhaust air leakage based on observed relative concentration of inert gas in supply airflow.

\[
OACF = \frac{m_1}{m_2}
\]

where \(m_1\) and \(m_2\) are the mass flow rates of the incoming fresh airstream at state 1 and 2, respectively as shown in Figure 15. OACF helps estimate the extra quantity of outside air required at the inlet to compensate for the air that leaks into or out of the
exchanger, and to meet the required net supply airflow to the building space. Ideal airflow conditions exist when there is no air leakage between the streams; EATR is close to zero, and OACF approaches 1. Deviations from ideal conditions indicate air leakage between the airstreams, which complicates the determination of accurate values for pressure drop and effectiveness.

The theoretical carryover of a wheel without a purge section is directly proportional to the speed of the wheel and the void volume of the medium (75 to 95% void, depending on type and configuration). For example, a 6 ft diameter, 2 in. deep wheel with a 90% void volume operating at 14 rpm has a carryover volumetric flow of

\[
\pi \left( \frac{6}{2} \right)^2 \left( \frac{2}{12} \right) (0.9) \left( 14 \frac{\text{rev}}{\text{min}} \right) = 60 \text{ cfm}
\]

If the wheel is handling a 10,000 cfm balanced flow, the percentage carryover is

\[
\frac{60}{10,000} \times 100 = 0.6\%
\]

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

**Figure 15. Air Leakage in Energy Recovery Units**

**Condensation and Frost Formation**

Condensation, ice formation, and/or frosting may occur on heat exchange surfaces. If entrance and exit effects are neglected, four distinct air/moisture regimes may occur as the warm airstream cools from its inlet condition to its outlet condition. First, there is a dry region with no condensate. Once the warm airstream cools below its dew point, there is a condensing region, which wets the heat exchange surfaces. If the heat exchange surfaces fall below freezing, the condensation freezes. Finally, if the warm airstream temperature falls below its dew point, sublimation causes frost to form.

The locations of these regions and rates of condensation and frosting depend on the duration of frosting conditions; airflow rates; inlet air temperature and humidity; heat exchanger core temperature; heat exchanger effectiveness; the geometry, configuration, and orientation; and heat transfer coefficients.

Sensible heat exchangers, which are ideally suited to applications in which heat transfer is desired but humidity transfer is not (e.g., swimming pools, kitchens, indirect evaporative cooling), can benefit from the latent heat released by the exhaust gas when condensation occurs. One pound of moisture condensed transfers about 1050 Btu to the incoming air at room temperature.

Condensation increases the heat transfer rate and thus the sensible effectiveness; it can also significantly increase pressure drops in heat exchangers with narrow airflow passage spacing. Frosting fouls the heat exchanger surfaces, which initially improves energy transfer but subsequently restricts the exhaust airflow, which in turn reduces the energy transfer rate. In extreme cases, the exhaust airflow (and supply) can become blocked. Defrosting a fully blocked heat exchanger requires that the unit be shut down for an extended period.

For frosting or icing to occur, an airstream must be cooled to a dew point below 32°F. Total heat exchangers transfer moisture from the airstream with higher moisture
content (usually the warmer airstream) to the less humid one. As a result, frosting or icing occurs at lower supply air temperatures in enthalpy exchangers than in sensible heat exchangers. In comfort air conditioning applications, condensation may occur in the supply side in summer and in the exhaust side in winter.

**Frost Threshold Limits**

![Graph comparing frost threshold limits for ERVs and HRVs](image)

Figure 16. Comparison of Frosting Thresholds for ERVs and HRVs

Only in climates experiencing severe winter design conditions will energy recovery ventilation systems require frost protection or defrost means. The temperature below which frost will begin to accumulate on heat exchanger surfaces is referred to as the frost threshold temperature. It is a function of outdoor temperature and indoor relative humidity. Figure 16 compares the frost threshold of a typical sensible heat exchanger with that of a typical enthalpy wheel. Note that while frost forms between 22 and 30°F in a sensible heat exchanger, frost thresholds for total energy exchangers are generally 20 to 30 degrees lower. This is due to the removal of water from the exhaust airstream by the total energy exchangers, effectively lowering the dewpoint of the exhaust. The water removed is subsequently picked up (through desorption, re-evaporation and/or sublimation) by the entering outdoor supply air.

Thus for areas with winter design temperatures between -5°F and 22°F (depending on the indoor RH), the enthalpy wheel component enjoys a significant advantage over sensible units in that frost control is generally not required. Many regions, where sensible HRVs require frost control, will not need frost protection for an ERV application. Even in the northern United States, applications such as schools and office buildings can be designed without frost control because most of the frosting hours are at night when the building is unoccupied. Bin data, such as that provided by ASHRAE, may be consulted to qualify daytime applications in cold climates for frost-free operation.

Table 12 lists frost threshold temperatures for AAONAIRE rotary energy recovery ventilation wheels over a range of indoor air temperatures and relative humidities. Note that the total energy AAONAIRE wheel will tolerate limited hours of operation below the frost threshold without damage or significant reduction of airflow. Frost control is not required until entering air temperatures are below the threshold.

**Frost Control and Prevention**

Enthalpy wheels can provide continuous frost-free operation at severe winter temperatures.
conditions if provided with controls to avoid the frosting threshold. Two methods commonly employed with the AAONAIRE design: The first method, variable step control reduces the effectiveness of heat transfer and allows the return air to remove any frost formation on the portion of the wheel in the return air stream. The second method, preheating of the outdoor air, is the generally preferred strategy from an energy savings and design load reduction standpoint. Example 5 illustrates the psychrometric processes associated with preheat for avoiding frost formation in enthalpy wheels.

The preheat strategy has two process lines: first, the incoming air is preheated (see the lower left hand side of the psychrometric chart). Note that the new preheated outside air condition ‘preheat (min)’ moves away from the saturation curve. Now the entering wheel air condition of ‘preheat (min)’ allows the wheel to operate at full energy recovery effectiveness without encountering saturation, avoiding the frost threshold. The exhaust air condition with preheat, is represented as ‘exhaust (min)’. In any climate where frost control is required for longer than a few hours due to severity of climate and/or humidification of the space, the preheat strategy will represent significant operating savings when compared to variable step capacity control. Unless the number of hours below the frost control threshold is so small as to be insignificant, preheat is the preferred frost control strategy. In addition, preheat offers the greater reduction in heating plant design loads.

**Example 5: Preheat with Total Energy Recovery**

*Indoor air at 70°F and 55.8°F wb (ρ = 0.074 lb/ft³) with a flow rate of 1000 cfm and supply outdoor air at -15°F and -15°F wb (ρ = 0.089 lb/ft³) with a flow rate of 1000 cfm enter a rooftop unit. The sensible and latent effectiveness is given as 75%. Assuming EATR = 0 and OACF ≈ 1, determine the amount of preheat required to prevent frost formation.*
Solution:
The frosting point of an application can be found on a psychrometric chart by drawing a line between the indoor conditions and outdoor conditions. The point at which the line crosses the 100% rh value on the psychrometric chart is the frosting point of the application. The minimum amount of preheat required is the amount to move the outdoor conditions horizontally right on the psychrometric chart so that the line between the indoor conditions and outdoor conditions no longer intersects the 100% rh curve.

In this example with an indoor air condition of 70°F and 40% rh and an outdoor air condition of -15°F and 100% rh enough preheat must be applied to raise the outside air temperature, entering the energy recovery device, by 13.3°F or more. (Preheat temperature entering wheel -1.7°F or greater.)

<table>
<thead>
<tr>
<th>OUTDOOR WINTER DESIGN TEMP</th>
<th>RETURN AIR (INDOOR) CONDITIONS</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>70°F and 20% rh (Frost Threshold -14°F)</td>
</tr>
<tr>
<td>Preheat Temp at Design</td>
<td>Required Capacity ΔT</td>
</tr>
<tr>
<td>5</td>
<td>-</td>
</tr>
<tr>
<td>0</td>
<td>-</td>
</tr>
<tr>
<td>-5</td>
<td>-</td>
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<td>-35</td>
<td>-20.3</td>
</tr>
<tr>
<td>-40</td>
<td>-21.0</td>
</tr>
</tbody>
</table>

Table 13. Preheat Frost Control Temperatures and Capacity Requirements at Selected Indoor and Outdoor Conditions

The amount of heat required for preheat can be found using Table 13 and the equation:
For this example, the minimum preheat heater size is

\[ \text{Preheater Size} = \dot{m} c_p \Delta T \]

\[ \text{Preheater Size} = \left( \frac{x \text{ ft}^3}{\text{min}} \right) \left( \frac{60 \text{ min}}{1 \text{ hr}} \right) \left( \frac{y \text{ lb of air}}{\text{ft}^3} \right) \left( \frac{0.24 \text{ Btu}}{\text{lb} \cdot ^\circ \text{R}} \right) (\Delta T) \]

\[ \text{Preheater Size} = \left( \frac{1000 \text{ ft}^3}{\text{min}} \right) \left( \frac{60 \text{ min}}{1 \text{ hr}} \right) \left( \frac{0.088 \text{ lb of air}}{\text{ft}^3} \right) \left( \frac{0.24 \text{ Btu}}{\text{lb} \cdot ^\circ \text{R}} \right) (-1.7 - (-15)) \]

\[ \text{Preheater Size} = 16,900 \frac{\text{Btu}}{\text{hr}} = 5\text{kW} \]

By applying a 7 kW heater to the outside air the exhaust air relative humidity is reduced to 81% and the energy recovery device can operate without risk of frost formation.
**Controls**

Heat exchanger controls may control frost formation or regulate the amount of energy transferred between airstreams at specified operating conditions. For example, ventilation systems designed to maintain specific indoor conditions at extreme outdoor design conditions may require energy recovery modulation to provide an economizer function, to prevent overheating ventilation supply air during cool to moderate weather or to prevent over humidification. Modulation methods include changing rotational speeds of (or stopping) heat wheels or bypassing part of one airstream around the heat exchanger using dampers.

In supply air bypass control, the amount of supply air allowed to pass through the wheel establishes the supply air temperature. An air bypass damper, controlled by a wheel supply air discharge temperature sensor, regulates the proportion of supply air permitted to bypass the exchanger.

A dead band control, which stops or limits the exchanger, may be necessary when no recovery is desired (e.g., when outside air temperature is higher than the required supply air temperature but below the exhaust air temperature). When outside air temperature is above the exhaust air temperature, the equipment operates at full capacity to cool incoming air. During very cold weather, it may be necessary to heat the supply air; stop the wheel, or, in small systems, use a defrost cycle for frost control.

**Pressure Drop**

Pressure drop for each airstream through an energy recovery unit depends on many factors, including exchanger design, mass flow rate, temperature, moisture, and inlet and outlet air connections. The pressure drop must be overcome by fans or blowers. Because the power required for circulating airstreams through the recovery unit is directly proportional to the pressure drop, the pressure drop through the energy recovery unit should be known. The pressure drop may be used with the fan efficiency to characterize the energy used by the exchanger and in turn the efficiency (not effectiveness) of an application.

**Fan Power**

The fan power $P_s$ required by the supply air is estimated from

$$P_s = Q_s \Delta \rho_s / 6356 \eta_f$$

The fan power $P_e$ required by the exhaust air is estimated from

$$P_e = Q_e \Delta \rho_e / 6356 \eta_f$$

where

- $P_s = \text{fan power for supply air, hp}$
- $P_e = \text{fan power for exhaust air, hp}$
- $\Delta \rho_s = \text{pressure drop of supply air caused by fluid friction, in. of water}$
- $\Delta \rho_e = \text{pressure drop of exhaust air caused by fluid friction, in. of water}$
- $\eta_f = \text{overall efficiency of fan and motor or product of fan and motor efficiencies}$

The total pumping power $P$ of the ERV can be given as

$$P = P_s + P_e$$

**Cleaning**

The method used to clean a heat exchanger depends on the transfer medium or mechanism used in the energy recovery unit and on the nature of the material to be removed. AAONAIRe energy wheels are self-cleaning with respect to dry particles. Smaller particles pass through the wheel. Larger particles land on the surface and are blown clear as the wheel turns into the opposite airflow path. For this reason,
the primary cleaning need for AAONAIRE energy wheels is to remove oil-based aerosol films that have collected on the desiccant surfaces. Such films can close off micron-sized pores at the surface of the desiccant material, reducing its ability to adsorb and desorb moisture. Grease build-up from kitchen exhaust, for example, is often removed with an automatic water-wash system. Other kinds of dirt may be removed by vacuuming, blowing compressed air through the passages, soaking the units in soapy water or solvents, steam cleaning or manual spray cleaning.

AAONAIRE energy wheels use a silica gel desiccant permanently bonded without adhesives to the heat exchange surface; therefore, the desiccant will not be washed off in the cleaning process. AAONAIRE energy wheels larger than 30 inches in diameter are made with removable segments for easy cleaning. Proper cleaning of the energy wheel should restore latent effectiveness to near original performance.

Cleaning frequency depends on the quality of the exhaust airstream. Comfort air conditioning systems generally require only infrequent cleaning. In a reasonably clean indoor environment, such as a school or office building, experience shows that reductions of airflow or loss of sensible effectiveness may not occur for ten or more years. Where there is moderate tobacco smoke, or within cooking facilities, reduction in effectiveness can occur much faster. In applications experiencing high levels of tobacco smoke, such as smoking lounges, nightclubs, bars, and restaurants, the energy transfer surfaces may require cleaning as often as every six months. Similar washing cycles may also be appropriate for industrial applications, such as welding or machining operations, involving ventilation of high levels of smoke or oil-based aerosols.

**SUMMARY**

AAONAIRE energy recovery systems provide energy savings by recycling energy instead of losing energy through exhaust air streams. AAONAIRE systems also enhance indoor air quality by allowing larger amounts of outside air to be provided to the space and through improved humidity control. AAONAIRE systems save money through both an initial HVAC equipment reduction and ongoing lifecycle operating savings. The payback for the AAONAIRE system for much of the country will pay for itself in less than one year and provide savings for the life of the product.

AAONAIRE systems can be optimized for sensible or total energy transfer. Heat recovery ventilators are suitable when outside air humidity is low and latent space loads are high for most of the year; and also for use with swimming pools, locker rooms, kitchens and indirect evaporative coolers. Energy recovery ventilators are suitable for applications in schools, offices, residences and other applications that require year-round economical preheating or/and precooling of outside supply air.

AAONAIRE units have been certified to perform according to AHRI Standard 1060. The certified ratings program requires testing, rating and independent verification of component performance at standard conditions and rated flow. Testing is in accordance with ASHRAE Standard 84. AHRI certified ratings include very complete information, to allow designers to fully characterize thermal and airflow performance. In addition to separate sensible, latent, and total effectiveness at two airflows for both summer and winter test conditions, the standard requires information on pressure loss as well as air leakage.
In many climates, the energy wheel eliminates three to four tons of air conditioning load for every 1000 cfm of ventilation air. When this benefit is claimed in the form of air conditioning equipment rightsizing, the initial cost of the system is typically on par with a system that does not utilize energy recovery. Payback is often immediate for the southeastern US and much of the midwest because the AAONAIRE energy recovery system actually has a lower initial cost than the system not using energy recovery. When payback is not immediate, most payback periods are less than two years.

The polymer construction of the AAONAIRE wheel yields a highly efficient, easily cleanable, light weight, corrosion resistant and flexible energy recovery wheel. The AAONAIRE energy transfer matrix design overcomes the problems associated with traditional aluminum wheels by providing lower cost, lighter weight, cleanable, permanently bonded desiccant that is completely corrosion resistant in seacoast applications. To further the maintainability, AAONAIRE wheels are made in segments which can be easily removed from the unit for cleaning outside the unit. The AAONAIRE energy transfer method reduces required HVAC system capacity more than fixed plate devices because of the ability to transfer both sensible and latent heat.

Contact your local AAON representative to learn more about the AAONAIRE system and other ways that AAON can help you to conserve energy, improve indoor air quality and save money.
APPENDIX:

**ERV Latent Energy Transfer Calculations**

Assuming no condensation in the AAON AIRE ERV, the latent effectiveness \( \varepsilon_L \) of an energy recovery ventilator is given as

\[
\varepsilon_L = \frac{q_L}{q_{L,\text{max}}} = \frac{m_s h_f (w_1 - w_2)}{m_{\text{min}} h_f (w_1 - w_3)} = \frac{m_e h_f (w_4 - w_3)}{m_{\text{min}} h_f (w_1 - w_3)} \tag{1a}
\]

where \( q_L \) is the actual latent heat transfer rate given by

\[
q_L = \varepsilon_L q_{L,\text{max}} \tag{1b}
\]

where \( q_{L,\text{max}} \) is the maximum heat transfer rate given by

\[
q_{L,\text{max}} = 60 m_{\text{min}} h_f (w_1 - w_3) \tag{1c}
\]

where

- \( \varepsilon_L \) = latent effectiveness
- \( h_f \) = enthalpy of vaporization, Btu/lb
- \( w \) = humidity ratios
- \( m_s \) = supply dry air mass flow rate, lb/min
- \( m_e \) = exhaust dry air mass flow rate, lb/min
- \( m_{\text{min}} \) = smaller of \( m_s \) and \( m_e \)

Because the enthalpy of vaporization from Equation (1a) can be dropped out from numerator and denominator, Equation (1a) can be rewritten as

\[
\varepsilon_m = \frac{m_w}{m_{w,\text{max}}} = \frac{m_s (w_1 - w_2)}{m_{\text{min}} (w_1 - w_3)} = \frac{m_e (w_4 - w_3)}{m_{\text{min}} (w_1 - w_3)} \tag{1d}
\]

where \( \varepsilon_m \) is moisture effectiveness, numerically equal to latent effectiveness \( \varepsilon_L \), and \( m_w \) is actual moisture transfer rate given by

\[
m_w = \varepsilon_m m_{w,\text{max}} \tag{1e}
\]

where \( m_{w,\text{max}} \) is the maximum moisture transfer rate given by

\[
m_{w,\text{max}} = m_{w,\text{min}} (w_1 - w_3) \tag{1f}\
\]
Assuming no water vapor condensation in the ERV, the leaving humidity ratios can be given as follows. The supply air leaving humidity ratio is

\[ w_2 = w_1 - \varepsilon_L \frac{m_{s,\min}}{m_s} (w_1 - w_3) \]  

(2a)

and the leaving exhaust air humidity ratio is

\[ w_4 = w_3 + \varepsilon_L \frac{m_{s,\min}}{m_s} (w_1 - w_3) \]  

(2b)

The total effectiveness \( \varepsilon_t \) of an AAONAIRe energy recovery ventilator is given as

\[ \varepsilon_t = \frac{q_t}{q_{t,\text{max}}} = \frac{m_s (h_z - h_1)}{m_{\text{min}} (h_3 - h_1)} = \frac{m_e (h_1 - h_4)}{m_{\text{min}} (h_3 - h_1)} \]  

(3a)

where \( q_t \) is the actual total heat transfer rate given by

\[ q_t = \varepsilon_t q_{t,\text{max}} \]  

(3b)

where \( q_{t,\text{max}} \) is the maximum total heat transfer rate given by

\[ q_{t,\text{max}} = 60m_{\text{min}} (h_1 - h_3) \]  

(3c)

where

- \( \varepsilon_t \) = total effectiveness
- \( h \) = enthalpy
- \( m_s \) = supply dry air mass flow rate, lb/min
- \( m_e \) = exhaust dry air mass flow rate, lb/min
- \( m_{\text{min}} \) = smaller of \( m_s \) and \( m_e \)

The leaving supply air condition is

\[ h_z = h_1 - \varepsilon_t \frac{m_{\text{min}}}{m_s} (w_1 - w_3) \]  

(4a)

and the leaving exhaust air condition is

\[ h_4 = h_3 - \varepsilon_t \frac{m_{\text{min}}}{m_e} (h_1 - h_3) \]  

(4b)
Assuming the stream at state 1 is of higher humidity, the latent heat recovery \( q_L \) from the ERV can be estimated from

\[
q_L = 60m_i h_{fg} (w_1 - w_2) = 60Q_s \rho_s h_{fg} (w_1 - w_2)
\]  
(5a)

\[
q_L = 60m_e h_{fg} (w_4 - w_3) = 60Q_e \rho_e h_{fg} (w_4 - w_3)
\]  
(5b)

\[
q_L = 60\varepsilon_L m_{\text{min}} h_{fg} (w_1 - w_3)
\]  
(5c)

\( h_{fg} \) = enthalpy of vaporization or heat of vaporization of water vapor, Btu/lb
\( w_1, w_2, w_3, w_4 = \text{inlet and exit humidity ratios of supply and exhaust airstreams, respectively} \)

The total energy transfer \( q_t \) between the streams is given by

\[
q_t = q_s + q_L = 60m_i (h_{1i} - h_{2i}) = 60Q_s \rho_s (h_{1i} - h_{2i}) = 60\left[m_i c_{pi} (t_1 - t_2) + m_i h_{fg} (w_1 - w_2)\right]
\]  
(6)

\[
q_t = q_s + q_L = 60m_e (h_{4e} - h_{3e}) = 60Q_e \rho_e (h_{4e} - h_{3e}) = 60\left[m_e c_{pe} (t_4 - t_3) + m_e h_{fg} (w_4 - w_3)\right]
\]  
(7a)

\[
q_t = 60\varepsilon_L m_{\text{min}} (h_{1i} - h_{3e})
\]  
(7b)

\( h_{1i} = \text{enthalpy of supply air at inlet, Btu/lb} \)
\( h_{3e} = \text{enthalpy of exhaust air at inlet, Btu/lb} \)
\( h_{2i} = \text{enthalpy of supply air at outlet, Btu/lb} \)
\( h_{4e} = \text{enthalpy of exhaust air at outlet, Btu/lb} \)
Example A. Negative Latent Energy Transfer

Inlet supply air enters an AAONAIRE ERV with a flow rate of 9350 cfm at 95°F and 20% rh. Inlet exhaust air enters with a flow rate of 9050 cfm at 75°F and 50% rh. Assume that the energy exchanger was tested under ASHRAE Standard 84, which rated the sensible heat transfer effectiveness at 50% and the latent (water vapor) transfer effectiveness at 50%. Assuming the specific heat of air is 0.24 Btu/lb·°F and the latent heat of vaporization to be 1100 Btu/lb, determine the sensible, latent, and net energy gained by the exhaust air.

Solution:

From the psychrometric chart, the properties of air at 95°F and 20% rh are

\[ V_1 = 14.14 \text{ ft}^3/\text{lb} \quad h_1 = 30.6 \text{ Btu/lb} \quad w_1 = 0.0071 \text{lbf s}/\text{lb of dry air} \]

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

\[ V_3 = 13.68 \text{ ft}^3/\text{lb} \quad h_3 = 28.1 \text{ Btu/lb} \quad w_3 = 0.0093 \text{lbf s}/\text{lb of dry air} \]

The mass flow rate at state 1 is obtained from

\[ m_1 = \frac{Q_1}{V_1} = \frac{9350 \text{ ft}^3/\text{min}}{14.14 \text{ ft}^3/\text{lb}} = 660 \text{ lb/\text{min}} \]
Similarly, the mass flow rate at state 3 is obtained from

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

These equal mass flow rates conform with ASHRAE Standard 84. Exit temperatures of the airstreams can be obtained as follows:

\[
t_2 = 95^\circ F - 0.5 \left( \frac{660 \text{ lb/min}}{660 \text{ lb/min}} \right) \left( 0.24 \frac{\text{Btu}}{\text{lb} \cdot ^\circ F} \right) (95^\circ F - 75^\circ F) = 85^\circ F
\]

\[
t_4 = 75^\circ F + 0.5 \left( \frac{660 \text{ lb/min}}{660 \text{ lb/min}} \right) \left( 0.24 \frac{\text{Btu}}{\text{lb} \cdot ^\circ F} \right) (95^\circ F - 75^\circ F) = 85^\circ F
\]

The exit humidity of the airstreams is found as follows:

\[
w_2 = 0.0071 - 0.5 \left( \frac{660 \text{ lb/min}}{660 \text{ lb/min}} \right) (0.0071 - 0.0093) = 0.0082 \text{ lb/lb of dry air}
\]

\[
w_4 = 0.0093 + 0.5 \left( \frac{660 \text{ lb/min}}{660 \text{ lb/min}} \right) (0.0071 - 0.0093) = 0.0082 \text{ lb/lb of dry air}
\]

The sensible heat gained by the exhaust stream is found as

\[
q_2 = (660 \text{ lb/min}) (0.24 \frac{\text{Btu}}{\text{lb} \cdot ^\circ F}) (85^\circ F - 75^\circ F) = 1584 \text{ Btu/min}
\]

The latent heat gained by the exhaust stream is found as

\[
q_2 = (660 \text{ lb/min}) (1100 \frac{\text{Btu}}{\text{lb}}) (0.0082 - 0.0093) = -799 \text{ Btu/min}
\]

The net heat energy gained by the exhaust airstream is therefore

\[
q = q_s + q_L = 1584 - 799 = 785 \text{ Btu/min}
\]

If the incoming outdoor air conditions had been at 95°F and 14% relative humidity, the net energy gained by the exhaust airstream would have been zero. The outlet exhaust airstream enthalpy at 85°F and 0.0082 lb/lb of dry air is given in the psychrometric chart as 29.4 Btu/lb. The net heat gained by the exhaust airstream is approximately 945 Btu/min.
REFERENCES:

Air Conditioning Heating Refrigeration Institute (AHRI), Arlington, VA. www.ahrinet.org

AirXchange, Rockland, MA. www.airxchange.com

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