AHRI Guideline V (I-P)

2011 Guideline for

# Calculating the Efficiency of Energy Recovery Ventilation and its Effect on Efficiency and Sizing of Building HVAC Systems

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Note:

This guideline supersedes AHRI Guideline V–2003.
For the SI version, see AHRI Guideline V (SI) - 2011

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# CALCULATING THE EFFICIENCY OF ENERGY RECOVERY VENTILATION AND ITS EFFECT ON EFFICIENCY AND SIZING OF BUILDING HVAC SYSTEMS

## Section 1. Purpose

**1.1** *Purpose.* The purpose of this guideline is to establish a method of calculating the energy efficiency of applied Energy Recovery Ventilation components and of heating, ventilating, and/or air-conditioning systems utilizing such components at selected operating conditions. It also provides guidance on proper sizing of cooling and heating equipment when such energy recovery components are applied.

**1.1.1** *Intent.* This guideline is intended for the guidance of the industry, including engineers, installers, contractors and users. It provides a means for calculating the impact of applied energy recovery equipment on the energy efficiency of the heating, ventilating and air-conditioning system at a single selected operating condition. The guideline is not a rating system for Energy Recovery Ventilation (ERV) Equipment, nor does it provide a means of estimating annual energy use.

**1.1.2** *Review and Amendment.* This guideline is subject to review and amendment as technology advances.

## Section 2. Scope

**2.1** *Scope.* This guideline applies to energy recovery ventilation component applications and combinations of energy recovery components with unitary heating, ventilating, and air-conditioning equipment incorporating mechanical ventilation with outside air.

**2.1.1** This guideline applies only to energy recovery applications utilizing components tested and rated in accordance with AHRI Standard 1060 (I-P).

**2.1.2** Because non-certified data is required for the calculations, the results should not be considered to be "certified".

## Section 3. Definitions

All terms in this document follow the standard industry definitions in the current edition of *ASHRAE Terminology of Heating, Ventilation, Air Conditioning and Refrigeration* and ASHRAE Standard 84, unless otherwise defined in this section.

**3.1** *Coefficient of Performance (COP).* A ratio of the cooling/heating capacity in watts to the power input values in watts at any given set of Rating Conditions expressed in watts/watts.

**3.2** *Combined Efficiency (CEF).* The efficiency of a system incorporating an ERV component with a unitary packaged air conditioner, heat pump, etc. Units vary according to the application. CEF is expressed in Btu/(W·h).

**3.3** *Effectiveness.* The measured energy recovery Effectiveness not adjusted to account for that portion of the psychrometric change in the leaving supply air (Figure 1, Station 2) that is the result of leakage of entering exhaust air (Figure 1, Station 3) rather than exchange of heat or moisture between the airstreams. The equation for determining Effectiveness is given in AHRI Standard 1060 (I-P), Appendix C.

**3.4** *Energy Efficiency Ratio (EER).* A ratio of the cooling capacity in Btu/h to the power input values in watts at any given set of Rating Conditions expressed in Btu/(W·h).

**3.5** *Energy Recovery Ventilation (ERV) Equipment.* Units which employ air-to-air heat exchangers to recover energy from exhaust air for the purpose of pre-conditioning outdoor air prior to supplying the conditioned air to the space, either directly or as part of an air-conditioning (to include air heating, air cooling, air circulating, air cleaning, humidifying and dehumidifying) system. Also referred to as the air-to-air heat exchanger (AAHX).

**3.5.1** *Heat Pipe Heat Exchanger*. A device employing tubes charged with a fluid for the purpose of transferring sensible energy from one air stream to another. Heat transfer takes place through the vaporization of the fluid exposed to the warmer air stream and condensation of the fluid in the cooler air stream.

**3.5.2** *Plate Heat Exchanger*. A device for the purpose of transferring energy (sensible or total) from one air stream to another with no moving parts. This exchanger may incorporate parallel, cross or counter flow construction or a combination of these to achieve the energy transfer.

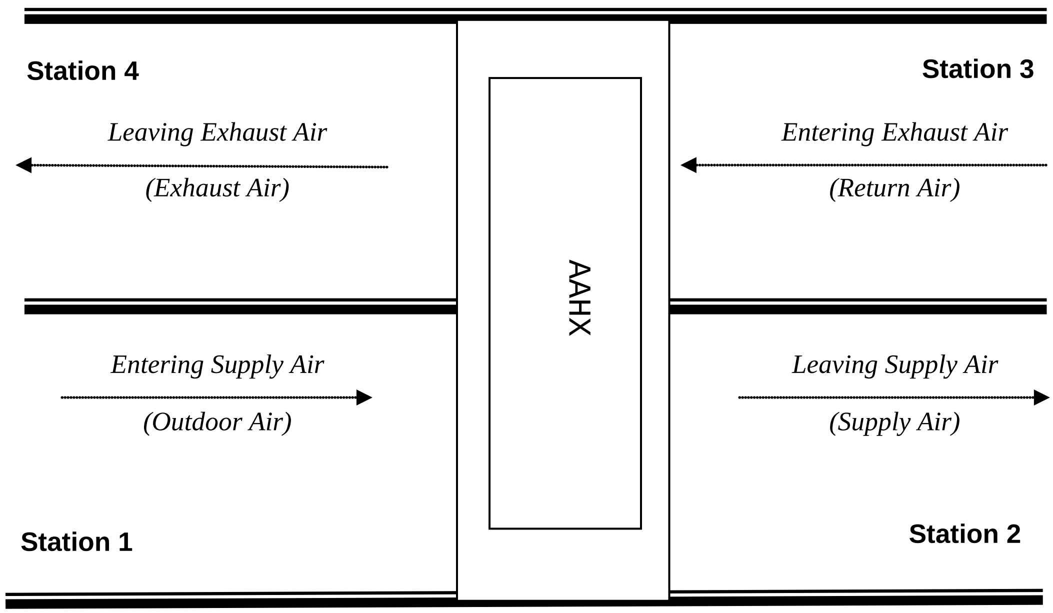


**Figure 1. Generic Configuration of an Air-to-Air Heat Exchanger Used for Energy Recovery in Ventilation Applications**

**3.5.3** *Rotary Heat Exchanger*. A device incorporating a rotating cylinder or wheel for the purpose of transferring energy (sensible or total) from one air stream to the other. It incorporates heat transfer material, a drive mechanism, a casing or frame, and includes any seals, which are provided to retard the bypassing and leakage of air from one air stream to the other.

**3.6** *Exhaust Air Transfer Ratio (EATR)*. The tracer gas concentration difference between the Leaving Supply Airflow and the Entering Supply airflow divided by the tracer gas concentration difference between the Entering Exhaust Airflow and the Entering Supply Airflow at the 100% rated airflows, expressed as a percentage. (Note: This guideline assumes that the tracer gas concentration difference is equal to the leakage of air from the Exhaust Airflow to the Supply Airflow. EATR, a ratio of the tracer gas, is used in the guideline formulae to represent a ratio of air flow.)

**3.7** *Fan/Motor Efficiency, $\eta_{Fan/Motor}$.* The product of the fan efficiency and the motor efficiency including drive losses (mechanical, electrical and/or electronic as applicable) for each airstream.

**3.8** Load Ratio, Y. The percentage of the system load (heating, cooling, humidification and/or dehumidification) met by the energy recovery component is designated as Y for the purposes of the calculations in this guideline.

**3.9** *Net Effectiveness*. The measured energy recovery Effectiveness adjusted to account for that portion of the psychrometric change in the leaving supply air (Figure 1, Station 2) that is the result of leakage of entering exhaust air (Figure 1, Station 3) rather than exchange of heat or moisture between the airstreams. The derivation of Net Effectiveness is given in AHRI Standard 1060 (I-P), Appendix C.

**3.10** *Net Supply Air Flow, $Q_{net\ supply}$*. That portion of the leaving supply air (Figure 1, Station 2) that originated as entering supply air (Figure 1, Station 1). The Net Supply Air Flow is determined by subtracting air transferred from the exhaust side of the AAHX from the gross air flow measured at the supply air leaving the heat exchanger and is given by the equation:

$$Q_{netsupply} = \text{Leaving supply air flow} \cdot (1 - \text{EATR}) \qquad 1$$

**3.11** *Outdoor Air Correction Factor (OACF)*. The entering supply air flow (Figure 1, Station 1) divided by the measured (gross) leaving supply air flow (Figure 1, Station 2).

**3.12** *Pressure Drop*. The difference in static pressure between the entering air and the leaving air for a given airstream.

**3.12.1** *Exhaust Pressure Drop.* The difference in static pressure between the entering exhaust air (Figure 1, Station 3) and the leaving exhaust air (Figure 1, Station 4).

**3.12.2** *Supply Pressure Drop.* The difference in static pressure between the entering supply air (Figure 1, Station 1) and the leaving supply air (Figure 1, Station 2).

**3.13** *Published Rating*. A statement of the assigned values of those performance characteristics at stated Rating Conditions, by which a unit may be chosen for its application. These values apply to all ERV Equipment of like size and type (identification) produced by the same manufacturer. The term Published Rating includes the rating of all performance characteristics shown on the unit or published in specifications, advertising or other literature controlled by the manufacturer, at stated Rating Conditions.

**3.13.1** *Application Rating*. A rating based on tests performed at application Rating Conditions (other than Standard Rating Conditions).

**3.13.2** *Standard Rating*. A rating based on tests performed at Standard Rating Conditions.

**3.14** *Rating Conditions*. Any set of operating conditions under which a single level of performance results, and which cause only that level of performance to occur.

**3.14.1** *Standard Rating Conditions*. Rating Conditions used as the basis of comparison for performance characteristics.

**3.15** *Recovery Efficiency Ratio (RER).* The efficiency of the energy recovery component in recovering energy from the exhaust airstream is defined as the energy recovered divided by the energy expended in the recovery process. Units vary according to the application. For Combined Efficiency with EER, the RER is expressed in Btu/(W·h). For Combined Efficiency with COP, the RER is expressed in W/W.

**3.16** *"Should."* "Should" is used to indicate provisions which are not mandatory but which are desirable as good practice.

**3.17** *Standard Air*. Air weighing 0.075 $lb/ft^3$, which approximates dry air at 70 ºF and at a barometric pressure of 29.92 in Hg.

**3.18** *Supply Air Flow*. The measured (gross) leaving supply air flow (Figure 1, Station 2). Also referred to as the rated air flow.

## Section 4. Information Requirements

**4.1** *Net Effectiveness.* Ratings of Net Effectiveness at application Rating Conditions and air flow rates are required to perform calculations of efficiency. AHRI certified ratings for Net Effectiveness are available at AHRI Standard 1060 (I-P) Standard Rating Conditions.

**4.2** *Blower Power.* A value for blower power input is required to perform the Combined Efficiency calculation. If manufacturer's data for blower power is not available, it may be calculated from component pressure loss and Fan/Motor Efficiency in accordance with this section and 6.1.

**4.2.1** *Pressure Drop.* Supply and Exhaust Pressure Drop values at application Rating Conditions and air flow rates are required to perform calculations of efficiency.

**4.2.2** *Fan/Motor Efficiency.* Values for Fan/Motor Efficiency may be required to calculate the RER of the component as applied. Fan/Motor Efficiency is used with the pressure loss of the energy recovery component to determine the blower power consumed in the process of recovering energy.

**4.2.3** *Determining Fan/Motor Efficiency.*

**4.2.3.1** When motor power is known:

$$\eta_{Fan/Motor} = \frac{Pwr_{Fan}}{Pwr_{Motor}} = \frac{P_{Fan} \cdot Q \cdot K_1}{K_2 \cdot {}^{\rho_A}\!/_{\rho_S} \cdot Pwr_{Motor}} \qquad 2$$

Where:

$\rho_A/\rho_S$ = Air density ratio (ratio of the air density to the density of Standard Air)
$\eta_{Fan/Motor}$ = Fan/Motor Efficiency
$K_1$ = 746 W/HP
$K_2$ = 6356 in $H_2O \cdot ft^3/HP \cdot min$
$P_{Fan}$ = Total static pressure across the fan, in $H_2O$
$Pwr_{Fan}$ = Fan Power, W
$Pwr_{Motor}$ = Motor Power, W
$Q$ = Air flow rate, cfm

**4.2.3.2** When the fan curve is available:

$$\eta_{Fan/Motor} = \left( \frac{P_{Fan} \cdot Q \cdot K_1}{K_2 \cdot {}^{\rho_A}\!/_{\rho_S} \cdot Pwr_{\ Fan}} \right) \cdot \eta_d \cdot \eta_m \qquad 3$$

Where:

$\eta_d$ = Drive efficiency
$\eta_m$ = Motor efficiency
$Pwr_{Fan}$ = Fan Power, HP

**4.2.3.3** When fan, motor and drive efficiencies are known:

$$\eta_{Fan/Motor} = \eta_f \cdot \eta_d \cdot \eta_m \qquad 4$$

Where:

$\eta_f$ = Fan efficiency

**4.3** *Unitary Equipment Efficiency.* The EER of the unitary equipment is required to perform calculations of CEF.

Calculations at Standard Rating Conditions may be used to provide an indication of comparative performance. To characterize actual performance, application Rating Conditions should be used.

System selection, fan configuration, energy recovery Effectiveness and outdoor air conditions can impact the applied EER of the unitary equipment. Changes in air flow rate, fan operating point or coil entering condition of the unitary equipment should be taken into account in calculating applied EER prior to completing the Combined Efficiency calculation.

Standard Ratings – EER at Standard Rating Conditions should be used when conditions (e.g. coil entering conditions and air flow rate) for the system match Standard Rating Conditions for the unitary equipment.

Application Ratings – EER at application Rating Conditions should be used if conditions (e.g. coil entering conditions and/or air flow rate) vary from Standard Rating Conditions for the unitary equipment.

## Section 5. General Principles

**5.1** *General Principle*. The general principle of all efficiency calculations is to determine the energy input or cost for a given useful energy output. In the case of ERV equipment, this is the recovered space conditioning energy divided by the power used to recover that energy. This can be expressed as a Recovery Efficiency Ratio (RER):

$$RER = \frac{Net\ conditioning\ energy\ recovered}{Total\ electric\ power\ consumed} \qquad 6$$

Where the net space conditioning energy can be either 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 AAHX as well as any additional power, such as the wheel drive motor in a Rotary Heat Exchanger.

The power required to move air through the AAHX is a function of the Supply and Exhaust Pressure Drop values through the AAHX, as well as the Fan/Motor Efficiency of the air-moving device. The power required to rotate a Rotary Heat Exchanger can be measured directly.

## Section 6. Calculating the Recovery Efficiency Ratio for the Energy Recovery Ventilation Component

**6.1** *Calculating the RER for the Energy Recovery Device.* Consult manufacturer's data for information on fan power consumption or pressure loss for the component. The RER is calculated in Equations 7, 8 and/or 9:

$$RER_{Total} = \frac{AAHX_{net\ total\ capacity}}{Pwr_{blwr} + Pwr_{comp}} = \frac{\varepsilon_{net\ total}\,\dot{m}_{min}\,(h_1 - h_3)}{Pwr_{blwr} + Pwr_{comp}} \qquad 7$$

$$RER_{Sensible} = \frac{AAHX_{net\ sensible\ capacity}}{Pwr_{blwr} + Pwr_{comp}} = \frac{\varepsilon_{net\ sensible}\,\dot{m}_{min}\,c_p(t_1 - t_3)}{Pwr_{blwr} + Pwr_{comp}} \qquad 8$$

$$RER_{Latent} = \frac{AAHX_{net\ latent\ capacity}}{Pwr_{blwr} + Pwr_{comp}} = \frac{\varepsilon_{net\ latent}\,\dot{m}_{min}\,h_{fg}(\omega_1 - \omega_3)}{Pwr_{blwr} + Pwr_{comp}} \qquad 9$$

Where:

| | | |
|---|---|---|
| $c_p$ | = | Specific heat of air, Btu/lb·°F |
| $h_1$ | = | Total enthalpy of the entering supply air, Btu/lb (Figure 1, Station 1) |
| $h_{fg}$ | = | Heat of condensation of water, Btu/lb |
| $h_3$ | = | Total enthalpy of the entering exhaust air, Btu/lb (Figure 1, Station 3) |

$\dot{m}_e$ = Mass flow rate of the entering exhaust air, lb/h (Figure 1, Station 3)

$\dot{m}_{min}$ = The lesser of $\dot{m}_e$ and $\dot{m}_s$, lb/h

$\dot{m}_s$ = Mass flow rate of leaving supply air, lb/h (Figure 1, Station 2)

$Pwr_{blwr}$ = Sum of the additional required blower power introduced by adding the energy recovery component to the system.

$Pwr_{comp}$ = Direct power input to the AAHX component, W

$t_1$ = Dry-bulb temperature of the entering supply air, °F (Figure 1, Station 1)

$t_3$ = Dry-bulb temperature of the entering exhaust air, °F (Figure 1, Station 3)

$\varepsilon_{net}$ = Net Effectiveness (sensible, latent, or total, as applicable), as defined in AHRI Standard 1060 (I-P) and determined in accordance with AHRI Standard 1060 (I-P)

$\omega_1$ = Humidity ratio of the entering supply air, lb (water)/lb (dry air) (Figure1, Station 1)

$\omega_3$ = Humidity ratio of the entering exhaust air, lb (water)/lb (dry air) (Figure 1, Station 3)

**6.2** *Determining value of $Pwr_{blwr}$, sum of the additional required blower power introduced by adding the energy recovery component to the system.* This includes both the supply and the exhaust airstreams.

**6.2.1** If blower power is known for the systems with and without the energy recovery component, $Pwr_{blwr}$ can be calculated as:

$$Pwr_{blwr} = Pwr_{bswer} + Pwr_{bewer} - Pwr_{bs} - Pwr_{be} \qquad 10$$

Where:
$Pwr_{be}$ = Power input to exhaust fan without energy recovery, W
$Pwr_{blwr}$ = Blower power, W
$Pwr_{bs}$ = Power input to supply fan without energy recovery, W
$Pwr_{bswer}$ = Power input to supply fan with energy recovery, W
$Pwr_{bewer}$ = Power input to exhaust fan with energy recovery, W

**6.2.2** If actual fan power is not known for the systems with and without the energy recovery component, the fan power associated with the additional pressure drop can be approximated by the following formula.

$$Pwr_{blwr} = \left[\frac{Q_{blower\ supply} \cdot \Delta P_{supply}}{C \cdot \eta_{Fan/Motor\ supply}}\right] + \left[\frac{Q_{blower\ exhaust} \cdot \Delta P_{exhaust}}{C \cdot \eta_{Fan/Motor\ exhaust}}\right] \qquad 11$$

Where:

C = Required unit conversion constant, 8.52 $ft^3 \cdot in\ H_2O./min \cdot W$
ΔP = Pressure drop of the component for the supply or exhaust airstreams, in $H_2O$
$Q_{blower\ supply}$ = supply fan airflow, cfm
$Q_{blower\ exhaust}$ = exhaust fan airflow, cfm

Note: Other alternatives (such as comparison of operating points on a fan curve) that accurately characterize the additional fan power required by the component are acceptable means of obtaining blower power.

**6.2.3** *Determination of fan airflow as a function of fan location.* $Q_{blower\ supply}$ and $Q_{blower\ exhaust}$ may be different from $Q_{supply}$ and $Q_{exhaust}$, depending on blower location with respect to the energy recovery component. See Figure 1.

If supply blower is located at Station 1, then:

$$Q_{blower\ supply} = Q_1 = Q_{supply} \times OACF \tag{12}$$

and:

$$Q_{blower\ supply} = Q_1 = \left[\frac{Q_{net\ supply} \cdot OACF}{(1 - EATR/100)}\right] \tag{13}$$

Where:

$Q_1$ = Airflow rate at Station 1, cfm

If supply blower is located at Station 2, then:

$$Q_{blower\_supply} = Q_2 = Q_{supply} \tag{14}$$

and:

$$Q_{blower\ supply} = Q_2 = \left[\frac{Q_{net\ supply}}{(1 - EATR/100)}\right] \tag{15}$$

Where:

$Q_2$ = Airflow rate at Station 2, cfm

If exhaust blower is located at station 3, then:

$$Q_{exhaust\_blower} = Q_3 = Q_{exhaust}$$

Where:

$Q_3$ = Airflow rate at Station 3, cfm

If exhaust blower is located at station 4, then:

$$Q_{exhaust\_blower} = Q_4 = Q_{exhaust} + Q_1 - Q_{supply} \tag{16}$$

and:

$$Q_{exhaust\ blower} = Q_4 = Q_{exhaust} + \left[\frac{Q_{net\ supply} \cdot OACF}{(1 - EATR/100)}\right] \tag{17}$$

Where:

$Q_4$ = Airflow rate at Station 4, cfm

**6.3** *Determining value of $Pwr_{comp}$, direct power input to the AAHX component.*

**6.3.1** Direct power input for a rotary exchanger is the measured drive motor power.

**6.3.2** Direct power input to the AAHX component for coil loops is the pump power. This can be obtained from actual pump power from manufacturers data. If not known, pump power can be estimated:

$$Pwr_{comp\,(pump)} = \frac{Q_p \cdot h_A \cdot SG}{C_{pump} \cdot \eta_{pump/motor}} \qquad 18$$

Where:

| | | |
|---|---|---|
| $Q_p$ | = | Fluid Flow Rate, gpm |
| $h_A$ | = | head added by pump, ft |
| SG | = | Specific Gravity of aqueous solution |
| $C_{pump}$ | = | Required unit conversion constant = 5.30 (gal · ft /min)/ W |
| $\eta_{pump/motor}$ | = | $\eta_{pump} \cdot \eta_{drive} \cdot \eta_{motor}$ |

Note: Specific gravity uses the density of fluid / density of water where density of water is 62.4 lbm/ft$^3$.

## Section 7. Integrating the Efficiency of the Energy Recovery Component with the Efficiency of Cooling and Heating Equipment

**7.1** CEF can be defined on a comparable basis to existing EER and COP ratings, based on the performance of the individual components. The basic principle (illustrated here for the cooling case) is:

$$CEF = \frac{\text{Net cooling delivered}}{\text{Total electric power consumed}} = \frac{cooling_1 + cooling_2 + cooling_{n-1} + cooling_n}{power_1 + power_2 + power_{n-1} + power_n} \qquad 19$$

When an AAHX is combined with a unitary air conditioner, the AAHX provides a portion of the system cooling capacity and the vapor compression cycle of the unitary air conditioner provides the rest. Consistent with the basic principle,

$$\text{EER} = \frac{\text{Net cooling capacity}}{\text{Total electric power consumption}} \qquad 20$$

The cooling system Combined Efficiency ($CEF_{cooling}$) of a unitary air conditioner with an AAHX cooling component can be defined as:

$$CEF_{cooling} = \frac{\text{net cooling capacity}_{AAHX} + \text{net cooling capacity}_{unitary}}{\text{electric power consumption}_{AAHX} + \text{electric power consumption}_{unitary}} \qquad 21$$

The heating system Combined Efficiency ($CEF_{heating}$) of a unitary air conditioner with an AAHX heating component can be defined as:

$$CEF_{heating} = \frac{\text{net heating capacity}_{AAHX} + \text{net heating capacity}_{unitary}}{\text{electric power consumption}_{AAHX} + \text{electric power consumption}_{unitary}} \qquad 22$$

## Section 8. Calculating the Effect of Energy Recovery Ventilation on Cooling System Efficiency

**8.1** *Calculating the Effect of the ERV on Cooling System CEF*. The $CEF_{cooling}$ may alternately be calculated from the RER, Btu/(W·h), of the AAHX ($RER_{AAHX}$) and the EER of the packaged equipment ($EER_{Unitary}$) according to the following expression:

$$CEF_{cooling} = \frac{1}{Y_c / RER_{AAHX} + (1 - Y_c) / EER_{Unitary}} \tag{23}$$

Where:

$$Y_c = \frac{net\ cooling\ capacity_{\ AAHX}}{net\ cooling\ capacity_{\ \mathrm{system}}}$$

**8.2** Note that RER, Btu/(W·h), 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.

## Section 9. Calculating the Effect of Energy Recovery Ventilation on Heating System Efficiency

**9.1** *Calculating the Effect of ERV on Heating System CEF*. The $CEF_{heating}$ is calculated from the RER, W/W, of the AAHX ($RER_{AAHX}$) and the COP of the packaged equipment ($COP_{Unitary}$) according to the following expression:

$$CEF_{heating} = \frac{1}{Y_h / RER_{AAHX} + (1 - Y_h) / COP_{Unitary}} \tag{24}$$

Where:

$$Y_H = \frac{net\ heating\ capacity_{\ AAHX}}{net\ heating\ capacity_{\ \mathrm{system}}} \quad \text{(from Section 4.4)}$$

**9.2** Note that RER can be calculated on the basis of sensible recovery, latent recovery or total energy recovery Effectiveness. The selection of the RER basis will depend on the analysis being conducted: use sensible for heating only, latent for humidification and total for heating and humidification.

## Section 10. Sizing

**10.1** *Sizing*. In evaluating the impact of energy recovery on CEF, it is important to recalculate the system size based on the load reduction provided by the energy recovery component at design conditions. Comparisons of systems with and without energy recovery should take this into account.

**10.2** *Methods*. Equipment should be sized with load reduction provided by energy recovery at design conditions. If not already accounted for in equipment selection, HVAC equipment should be reselected in accordance with 10.3.

**10.3** *HVAC Equipment Load Reduction Factor*. An estimate of the reduction in equipment size is provided by the capacity of the energy recovery component at design conditions and the load ratio Y, according to the expression:

$$\begin{pmatrix} \text{Required} \\ \text{equipment} \\ \text{capacity with} \\ \text{energy recovery} \end{pmatrix} = (1 - \mathrm{Y}) \begin{pmatrix} \text{Equipment capacity} \\ \text{without energy} \\ \text{recovery} \end{pmatrix} \tag{25}$$

## Section 11. Implementation

**11.1** *Conditions*. This guideline may be used to compare efficiencies of different systems at a set of standard conditions or for a specific set of conditions reflecting a specific application. The user should note that, like unitary EER values for Standard Rating Conditions, RER values for Standard Rating Conditions (for example, AHRI Standard 1060 (I-P) Standard Rating Conditions and a value for fan efficiency) can provide a rational comparison of different energy recovery components. Note that the RER for the energy recovery component as applied can vary with climate or conditions. This is due to the fact that the energy recovered is dependent on the difference between outdoor air and exhaust air conditions and thus varies widely, while the energy used (Pressure Drop · Fan/Motor Efficiency) is more consistent for a given air flow rate.

**11.2** *Blower Power*. The blower power calculations presented in the guideline are for the sole purpose of determining the incremental parasitic losses due to the addition of the energy recovery component to the airstreams. They do not describe the air-moving efficiency of a ventilation system in supplying outside air; nor do they describe the fan efficiency of unitary systems, which is included in unitary energy efficiency ratings. Fan placement, cabinet design and related system effects, while they can impact the efficiency of air delivery, are not addressed in this guideline.

**11.3** *Applications*. While the guideline provides a method of determining efficiency of the energy recovery and of systems incorporating energy recovery, it is not intended to be used to set minimum equipment efficiencies for heating or cooling equipment in general. It is only applicable where outside air is being introduced into the system; the benefit of energy recovery to the Combined Efficiency is directly dependent on the amount of outdoor air provided and the indoor and outdoor conditions.

**11.4** *Calculated Results*. The guideline provides a methodology for determining RER and CEF for a single point at specified design conditions. If it is desired to evaluate the seasonal impact of energy recovery, it is necessary to perform the guideline calculations for a series of representative conditions or, preferably, perform an energy analysis. See Appendix D for example results comparing CEF and energy analysis calculations for a variety of climates.

**11.5** *Accuracy*. The accuracy of the calculations is limited by the cumulative tolerances in testing and reporting of Standard and Application Ratings, estimates of Fan/Motor Efficiency, etc.

**11.6** *Sensible Heat Ratio*. Care should be exercised in selecting energy recovery components and cooling equipment to provide adequate moisture removal for humidity control in cooling. Combinations of equipment that result in a sensible heat ratio matching the load will provide improved humidity control over those that do not.

**11.7** *Additional Guidance*. Other guidelines or standards, such as local codes and ASHRAE Standard 90.1, may contain specific requirements for energy recovery.

# APPENDIX A. REFERENCES – NORMATIVE

**A1** Listed here are all standards, handbooks and other publications essential to the formation and implementation of the standard. All references in this appendix are considered as part of this standard.

**A1.1** ANSI/AHRI Standard 1061 (SI)-2011, *Performance Rating Air-To-Air Heat Exchangers For Energy Recovery Ventilation Equipment*, 2011, Air-Conditioning, Heating, and Refrigeration Institute, 2111 Wilson Boulevard, Suite 500, Arlington, VA 22201, U.S.A.

# APPENDIX B. REFERENCES – INFORMATIVE

**B1** Listed here are standards, handbooks and other publications which may provide useful information and background, but are not considered essential. References in this appendix are not considered part of the guideline.

**B1.1** AHRI Standard 330-98, *Water Source Heat Pumps*, 1998, Air-Conditioning, Heating, and Refrigeration Institute, 2111 Wilson Boulevard, Suite 500, Arlington, VA 22201, U.S.A.

**B1.2** ANSI/AHRI Standard 210/240-2008, *Unitary Air Conditioning and Air Source Heat Pump Equipment*, 2008 Air-Conditioning, Heating, and Refrigeration Institute, 2111 Wilson Boulevard, Suite 500, Arlington, VA 22201, U.S.A.

**B1.3** ANSI/AHRI Standard 310/380-2004, *Packaged Terminal Air-Conditioners and Heat Pumps (CSA-C744-93) (ANSI/AHRI 310/380-93)*, 2004, Air-Conditioning, Heating, and Refrigeration Institute, 2111 Wilson Boulevard, Suite 500, Arlington, VA 22201, U.S.A.

**B1.4** ANSI/AHRI Standard 340/360-2007 with Addendum 1,*Commercial and Industrial Unitary Air-Conditioning and Heat Pump Equipment*, 2007, Air-Conditioning, Heating, and Refrigeration Institute, 2111 Wilson Boulevard, Suite 500, Arlington, VA 22201, U.S.A.

**B1.5** ANSI/AHRI Standard 390-2001, *Single Package Vertical Air-Conditioners and Heat Pumps*, 2001, Air-Conditioning, Heating, and Refrigeration Institute, 2111 Wilson Boulevard, Suite 500, Arlington, VA 22201, U.S.A.

**B1.6** ANSI/AHRI Standard 430-2009, *Central Station Air Handling Units*, 2009, Air-Conditioning, Heating, and Refrigeration Institute, 2111 Wilson Boulevard, Suite 500, Arlington, VA 22201, U.S.A.

**B1.7** ANSI/AHRI Standard 1060 (I-P)-2011, *Performance Rating Air-To-Air Heat Exchangers For Energy Recovery Ventilation Equipment*, 2011, Air-Conditioning, Heating, and Refrigeration Institute, 2111 Wilson Boulevard, Suite 500, Arlington, VA 22201, U.S.A.

**B1.8** ANSI/ASHRAE Standard 84-1991, *Method of Testing Air-to-Air Heat Exchangers*, 1991, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., 1791 Tullie Circle N.E., Atlanta, GA 30329, U.S.A.

**B1.9** ANSI/AHRI/ASHRAE/ISO 13256-1, *Water-Source Heat Pumps – Testing and Rating for Performance – Part I: Water-to Air and Brine-to-Air Heat Pumps*, 1998, Air-Conditioning, Heating, and Refrigeration Institute/American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc./International Organization for Standardization, 2111 Wilson Boulevard, Suite 500, Arlington, VA 22201, U.S.A./1791 Tullie Circle N.E., Atlanta, GA 30329, U.S.A./Case Postale 56, CH-1211, Geneva 21 Switzerland.

**B1.10** ANSI/ASHRAE/IESNA Standard 90.1-2010, *Energy Standard for Buildings Except Low-Rise Residential Buildings*, 2010, American National Standards Institute/American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc./Illuminating Engineering Society of North America, 25 West 43rd Street, 4th Floor, New York, NY 10036 U.S.A/1791 Tullie Circle, N.E., Atlanta, GA 30329, U.S.A./120 Wall Street, Floor7 17, New York, NY 10005

**B1.11** ASHRAE Handbook, *Fundamentals*, 2009, American Society of Heating, Refrigerating and Air-Conditioning

Engineers, Inc., 1791 Tullie Circle N.E., Atlanta, GA 30329, U.S.A.

**B1.12** ASHRAE Handbook, *Systems and Equipment*, 2008, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., 1791 Tullie Circle N.E., Atlanta, GA 30329, U.S.A.

**B1.13** ASHRAE *Terminology of Heating, Ventilation, Air-Conditioning, and Refrigeration*, Second Edition, 1991, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., 1791 Tullie Circle, N.E., Atlanta, GA 30329, U.S.A.

**B1.14** *System Energy Efficiency Ratio, Establishing the Recovery Efficiency Ratio for Air-to-Air Energy Recovery Heat Exchangers and Their Effect on HVAC System Energy Efficiency*, 2002, Arthur D. Little, Inc., Acorn Park, Cambridge, MA 02140, U.S.A.

# APPENDIX C. SAMPLE CALCULATIONS – INFORMATIVE

**C1** Cooling example, enthalpy recovery and EER:

Where:

| | | |
|---|---|---|
| $\varepsilon_{\text{net total}}$ | = | 0.70 (70%) |
| Q | = | 1000 cfm |
| C | = | 8.52 $ft^3$ in $H_2O$./min·W |
| $\Delta p$ | = | 1 in $H_2O$ Exhaust and Supply Pressure Drop |
| $h_1$ - $h_3$ | = | 13.447 Btu/lb outdoor air at 95°F dry-bulb/78 °F wet-bulb, return air at 75°F dry-bulb/63 °F wet-bulb |
| $\rho_{\text{Air}}$ | = | 0.075 lb/$ft^3$ |
| $\eta_{\text{Fan/Motor}}$ | = | 0.84 motor efficiency x 0.50 fan efficiency = 0.42 |
| $Pwr_{\text{comp}}$ | = | 50 W for an enthalpy wheel ( = 0 for a plate or heat pipe heat exchanger) |

From Equation 7, $RER_{\text{Total}}$ is given by:

$$RER_{Total} = \frac{\varepsilon_{\text{net total}}\,\dot{m}_{\min}\,(h_1 - h_3)}{Pwr_{blwr} + Pwr_{\text{comp}}} \tag{C1}$$

Since actual Fan power is not known, $Pwr_{\text{blwr}}$ for supply and exhaust can be estimated by using equation 11. Note that if the component's OACF is greater then 1.0 then this will not include fan energy associated with the cross flow. See Example C4 for components with OACF greater than 1.

$$RER_{Total} = \frac{\varepsilon_{\text{net total}} \cdot Q_{\min} \cdot \rho_{air} \cdot (h_1 - h_3)}{\left[\left(\dfrac{Q_{\text{blower supply}} \cdot \Delta p_{\text{supply}}}{C \cdot \eta_{Fan/Motor_{\text{supply}}}}\right)\right] + \left[\left(\dfrac{Q_{\text{blower exhaust}} \cdot \Delta p_{exhaust}}{C \cdot \eta_{Fan/Motor_{exhaust}}}\right)\right] + Pwr_{comp}}$$

$$= \frac{0.70 \cdot \left(1000\dfrac{ft^3}{\min}\right) \cdot \left(0.075\dfrac{lb}{ft^3}\right) \cdot \left(60\dfrac{\min}{h}\right) \cdot \left(13.447\dfrac{Btu}{lb}\right)}{\left[\left(\dfrac{\left(1000\dfrac{ft^3}{\min}\right)\cdot(1\ in\ \text{H}_2\text{O})}{\left(8.52\dfrac{ft^3 \cdot in\ \text{H}_2\text{O}}{\min \cdot W}\right)\cdot(0.42)}\right)\right] + \left[\left(\dfrac{\left(1000\dfrac{ft^3}{\min}\right)\cdot(1\ in\ \text{H}_2\text{O})}{\left(8.52\dfrac{ft^3 \cdot in\ \text{H}_2\text{O}}{\min \cdot W}\right)\cdot(0.42)}\right)\right] + 50\ W}$$

$$= \frac{\left(42{,}358\dfrac{Btu}{h}\right)}{558.91\ W + 50\ W} = \frac{\left(42{,}358\dfrac{Btu}{h}\right)}{608.90\ W}$$

$$RER_{Total} = 69.57\ \text{Btu/(W·h)}$$

For a direct expansion system with EER=10 and where the ERV component (AAHX) is handling 30% of the system load at design conditions, the $CEF_{cooling}$ is given by Equation 23:

$$CEF_{cooling} = \frac{1}{Y_c/RER_{AAHX} + (1 - Y_c)/EER_{Unitary}}$$

$$= \frac{1}{0.3/69.58 + 0.7/10} \qquad C2$$

$$CEF_{cooling} = 13.46 \text{ Btu}/(W \cdot h)$$

**C2** Cooling example, sensible recovery and EER:

Where:

| | | |
|---|---|---|
| $\varepsilon_{net\ sensible}$ | = | 0.70 (70%) |
| Q | = | 1000 cfm |
| C | = | 8.52 $ft^3$ in $H_2O$./min·W |
| $\Delta p$ | = | 1 in $H_2O$. Exhaust and Supply Pressure Drop |
| $t_1 - t_3$ | = | 20°F (outdoor air at 50 °F, return air at 70°F) |
| $\rho_{Air}$ | = | 0.075 lb/$ft^3$ |
| $c_p$ | = | Specific heat of dry air = 0.24 Btu/lb· °F |
| $\eta_{Fan/Motor}$ | = | 0.84 motor efficiency x 0.50 fan efficiency = 0.42 |
| $Pwr_{comp}$ | = | 0 W for a plate or heat pipe heat exchanger (would be greater than 0 for an enthalpy wheel) |

From Equation 8, $RER_{sensible}$ is given by:

$$RER_{sensible} = RER_{sensible} = \frac{\varepsilon_{net\ sensible}\,\dot{m}_{min}\,c_p\,(t_1 - t_3)}{Pwr_{blwr} + Pwr_{comp}}$$

C3

Since actual Fan power is not known $Pwr_{blwr}$ for supply and exhaust can be estimated by equation 11. Note that if the component's OACF is greater then 1.0 then this will not include fan energy associated with the cross flow. See Example C4 for components with OACF greater than 1.

$$RER_{sensible} = \frac{\varepsilon_{net\ sensible} \cdot (Q \cdot \rho_{Air}) \cdot c_p \cdot (t_1 - t_3)}{\left[\left(\dfrac{Q_{supply} \cdot \Delta p_{supply}}{C \cdot \eta_{Fan/Motor_{supply}}}\right)\right] + \left[\left(\dfrac{Q_{exhaust} \cdot \Delta p_{exhaust}}{C \cdot \eta_{Fan/Motor_{exhaust}}}\right)\right] + Pwr_{comp}}$$

$$= \frac{0.70 \cdot \left(1000\dfrac{ft^3}{\min}\right) \cdot \left(0.075\dfrac{lb}{ft^3}\right) \cdot \left(60\dfrac{\min}{h}\right) \cdot \left(0.24\dfrac{Btu}{lb \cdot °F}\right) \cdot (20°F)}{\left[\left(\dfrac{\left(1000\dfrac{ft^3}{\min}\right) \cdot (1\ in\ H_2O)}{\left(8.52\dfrac{ft^3 \cdot in\ H_2O}{\min \cdot W}\right) \cdot (0.42)}\right)\right] + \left[\left(\dfrac{\left(1000\dfrac{ft^3}{\min}\right) \cdot (1\ in\ H_2O)}{\left(8.52\dfrac{ft^3 \cdot in\ H_2O}{\min \cdot W}\right) \cdot (0.42)}\right)\right] + 0W}$$

$$= \frac{15{,}120\ Btu/h}{558.9\,W}$$

$$RER_{sensible} = 27.1\ Btu/(W \cdot h)$$

For a direct expansion system with EER=10 and where the ERV component (AAHX) is handling 30% of the system load at design conditions, the $CEF_{cooling}$ is given by Equation 23:

$$CEF_{cooling} = \frac{1}{Y_c/RER_{AAHX} + (1 - Y_c)/EER_{Unitary}}$$

$$= \frac{1}{0.3/27.1 + 0.7/10} \qquad C4$$

$$CEF_{cooling} = 12.34\ Btu/(W \cdot h)$$

**C3** Heating example, sensible recovery and COP:

Where:

| | | |
|---|---|---|
| $\varepsilon_{net\ sensible}$ | = | 0.70 (70%) |
| Q | = | 1000 cfm |
| C | = | 8.52 $ft^3$ in $H_2O$./min·W |
| Z | = | 1 W/3.413 Btu/h |
| $\Delta p$ | = | 1 in $H_2O$. Exhaust and Supply Pressure Drop |
| $t_1 - t_3$ | = | 20°F (outdoor air at 50 °F, return air at 70°F) |
| $\rho_{Air}$ | = | 0.075 lb/$ft^3$ |
| COP | = | 2.93 (for the heat pump) |
| $c_p$ | = | Specific heat of dry air = 0.24 Btu/lb· °F |
| $\eta_{Fan/Motor}$ | = | 0.84 motor efficiency x 0.50 fan efficiency = 0.42 |
| $Pwr_{comp}$ | = | 0 W for a plate or heat pipe heat exchanger (would be greater than 0 for an enthalpy wheel) |

From Equation 8, $RER_{sensible}$ is given by:

$$RER_{sensible} = \frac{\varepsilon_{net\ sensible}\ \dot{m}_{min}\ c_p(t_1 - t_3)}{Pwr_{blwr} + Pwr_{comp}} \qquad C5$$

Since actual Fan power is not known $Pwr_{blwr}$ for supply and exhaust can be estimated by Equation 11. Note that if the component's OACF is greater then 1.0 then this will not include fan energy associated with the cross flow. See Example C4 for components with OACF greater than 1.

$$RER_{sensible} = \left( \frac{\varepsilon_{net\ sensible} \cdot (Q \cdot \rho_{Air}) \cdot c_p \cdot (t_1 - t_3)}{\left[\left(\frac{Q_{supply} \cdot \Delta p_{supply}}{C \cdot \eta_{Fan/Motor_{supply}}}\right)\right] + \left[\left(\frac{Q_{exhaust} \cdot \Delta p_{exhaust}}{C \cdot \eta_{Fan/Motor_{exhaust}}}\right)\right] + Pwr_{comp}} \right) \cdot (Z)$$

$$= \left( \frac{0.70 \cdot \left(1000 \frac{ft^3}{\min}\right) \cdot \left(0.075 \frac{lb}{ft^3}\right) \cdot \left(60 \frac{\min}{h}\right) \cdot \left(0.24 \frac{BTU}{lb \cdot °F}\right) \cdot (20°F)}{\left[\left(\frac{\left(1000 \frac{ft^3}{\min}\right) \cdot (1\, in\ \mathrm{H_2O})}{\left(8.52 \frac{ft^3 \cdot in\ \mathrm{H_2O}}{\min \cdot W}\right) \cdot (0.42)}\right)\right] + \left[\left(\frac{\left(1000 \frac{ft^3}{\min}\right) \cdot (1\, in\ \mathrm{H_2O})}{\left(8.52 \frac{ft^3 \cdot in\ \mathrm{H_2O}}{\min \cdot W}\right) \cdot (0.42)}\right)\right] + 0\,W} \right) \cdot \left( \frac{1\,W}{3.413\ Btu/h} \right)$$

$$= \left( \frac{15{,}120\ Btu/h}{558.9\,W} \right) \cdot \left( \frac{1\,W}{3.413\ Btu/h} \right)$$

$$RER_{Sensible} = 7.92$$

For a heat pump system with COP = 2.93 and where the ERV component (AAHX) is handling 30% of the system load at design conditions, the $CEF_{heating}$ is given by Equation 24:

$$CEF_{heating} = \frac{1}{Y_h / RER_{COP\ AAHX} + (1 - Y_h) / COP_{Unitary}}$$

$$= \frac{1}{0.3/7.92 + 0.7/2.93} \qquad \text{C6}$$

$$CEF_{heating} = 3.6$$

**C4** *Calculating RER Considering Fan Position for Components with OACF greater than 1:*

In this example we demonstrate that fan position must be correctly accounted for when calculating RER. We calculate RER for two cases, both with identical ventilation requirements and the same energy recovery component. However, the first case uses a draw through blower arrangement (see Figure C1) while the second cases uses a blow-through supply fan and draw-through exhaust fan arrangement (see Figure C2): the RERs for each case are different.

In both cases:

| | | |
|---|---|---|
| $Q_s$ | = | 10,000 cfm |
| $Q_e$ | = | 9,000 cfm |
| $T_1$ | = | 95°F Outside Air Temperature |
| $W_1$ | = | 0.01714 lb/lb Outside Air Humidity |
| $T_3$ | = | 75°F Return Air Temperature |
| $W_3$ | = | 0.00926 lb/lb Return Air Humidity |
| $E_{sensible}$ | = | 74.6% |
| $E_{latent}$ | = | 70.8% |
| $\eta_{Motor}$ | = | 0.92 motor efficiency · 0.50 fan efficiency |
| $Pwr_{comp}$ | = | 575W for the enthalpy wheel in this example |

Supply Fan Total Static Pressure <u>with</u> energy recovery component = 4.5 in. $H_2O$.
Exhaust Fan Total Static Pressure <u>with</u> energy recovery component = 3.0 in. $H_2O$.

$\Delta p_{supply}$ = 1.05 in $H_2O$
$\Delta p_{exhaust}$ = 0.90 in $H_2O$

Therefore:

Supply Fan Total Static Pressure <u>without</u> energy recovery component = 3.45 in. $H_2O$.
Exhaust Fan Total Static Pressure <u>without</u> energy recovery component = 2.1 in. $H_2O$.

For both cases, the first step is to find the recovered energy:

$$\dot{m}_e = \left(0.075\frac{lb}{ft^3}\right)\cdot\left(9{,}000\frac{ft^3}{\min}\right)\cdot\left(60\frac{\min}{h}\right) = 40{,}500\frac{lb}{h} \quad \text{C7}$$

$$\dot{m}_s = \left(0.075\frac{lb}{ft^3}\right)\cdot\left(10{,}000\frac{ft^3}{\min}\right)\cdot\left(60\frac{\min}{h}\right) = 47{,}250\frac{lb}{h} \quad \text{C8}$$

net sensible$_{\text{AAHX capacity}}$

$$= \varepsilon_{sensible}\cdot\dot{m}_{\min}\cdot c_{p}\cdot(t_1 - t_3) = 0.746\cdot\left(40{,}500\frac{lb}{h}\right)\cdot\left(0.24\frac{Btu}{lb}\right)\cdot(95°F - 75°F) = 145{,}022\frac{Btu}{h} \quad \text{C9}$$

net latent capacity $_{\text{AAHX}}$

$$= \varepsilon_{latent}\cdot\dot{m}_{\min}\cdot h_{fg}\cdot(w_1 - w_3) = 0.708\cdot\left(40{,}500\frac{lb}{h}\right)\cdot\left(1061\frac{Btu}{lb}\right)\cdot\left(0.01714 - 0.00925\left(\frac{lb}{lb}\right)\right) = 240{,}038\frac{Btu}{h} \quad \text{C10}$$

net total capacity$_{\text{AAHX}}$ = net sensible capacity$_{\text{AAHX}}$+ net latent capacity$_{\text{AAHX}}$ = 145,022 + 240,038 = 385,060 (Btu/hr) C11

The next step is to determine the blower power ($Pwr_{blower}$). The first case is a draw-through fan arrangement as shown in Figure C1:

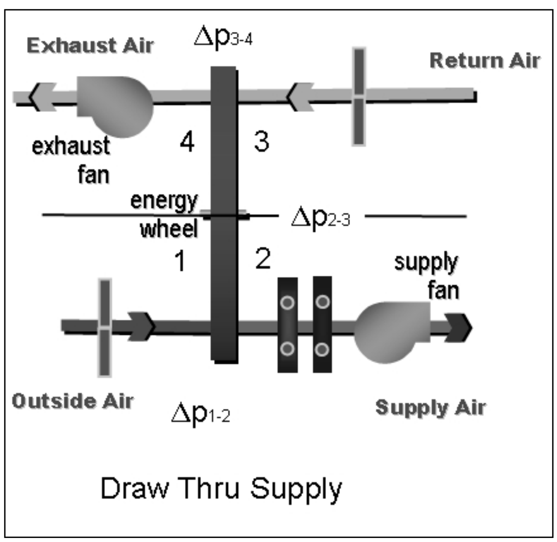


**Figure C1. Placement of Fans for a Draw-through Arrangement**

In this case:

$\Delta P_{2\text{-}3}$ = 0.5in. $H_2O$.

and:

EATR = 2.3%
OACF = 1.03

$$Q_{\text{supply blower}} = Q_2 = \left[\frac{Q_{net\ supply}}{(1-EATR/100)}\right] = \left[\frac{10{,}000\dfrac{ft^3}{\min}}{(1-2.3/100)}\right] = 10{,}235\frac{ft^3}{\min} \quad \text{C12}$$

$$Q_{\text{exhaust blower}} = Q_4 = Q_{exhaust} + \left[\frac{Q_{net\ supply}\cdot OACF}{(1-EATR/100)}\right] - Q_{net\ supply} = 9000 + \left[\frac{10{,}000\dfrac{ft^3}{\min}\cdot 1.03}{(1-2.3/100)}\right] - 10{,}000 = 9542\frac{ft^3}{\min} \quad \text{C13}$$

From fan selection software or fan curve find BHP for each fan, with energy recovery in the system. For this example:

Supply blower operating at 10,235cfm @ Total Static Pressure of 4.5 in. $H_2O$.: BHP = 10.87
Exhaust blower operating at 9,542cfm @ Total Static Pressure of 3.0 in. $H_2O$.: BHP = 6.98

Determine power input to blower motors with energy recovery in the system; direct drive assumed, with motor operating at $\eta_{motor}$ = 92%:

$$Pwr_{bswer} = \frac{10.87HP}{0.92}\cdot\left(745.7\frac{W}{HP}\right) = 8{,}811W \quad \text{C14}$$

$$Pwr_{bewer} = \frac{6.98HP}{0.92}\cdot\left(745.7\frac{W}{HP}\right) = 5{,}658W \quad \text{C15}$$

From fan selection software or fan curve find BHP for each fan, without energy recovery in the system. For this example:

Supply blower operating at 10,000cfm @ Total Static Pressure of 3.45 in. $H_2O$.: BHP = 8.36

Exhaust blower operating at 9,000cfm @ Total Static Pressure of 2.1 in. $H_2O$.: BHP = 4.87

Determine power input to blower motors without energy recovery in the system; direct drive assumed, with motor operating at $\eta_{motor}$ = 92%:

$$Pwr_{bs} = \frac{8.36HP}{0.92}\cdot\left(745.7\frac{W}{HP}\right) = 6{,}776W \quad \text{C16}$$

$$Pwr_{be} = \frac{4.87HP}{0.92}\cdot\left(745.7\frac{W}{HP}\right) = 3{,}947W \quad \text{C17}$$

Find the additional blower power needed when energy recovery component is added to the system, using Equation 10:

$Pwr_{blwr}$ = 8,811 + 5,658 - 6,776 - 3,974 = 3,745 W C18

Calculate RER for the example using draw-through blower arrangement, using Equation 7:

$$\text{RER}_{\text{Total}} = \left[ \frac{\text{Net Total Capacity}_{AAHX}}{Pwr_{blwr} + \text{Pwr}_{\text{comp}}} \right] = \left[ \frac{385{,}060 \dfrac{Btu}{h}}{3745W + 575\,W} \right] = 89.1 Btu/(W \cdot h) \qquad \text{C19}$$

Now we consider the second case, with a blow-through supply fan and draw-through exhaust fan arrangement (see Figure C2, below):

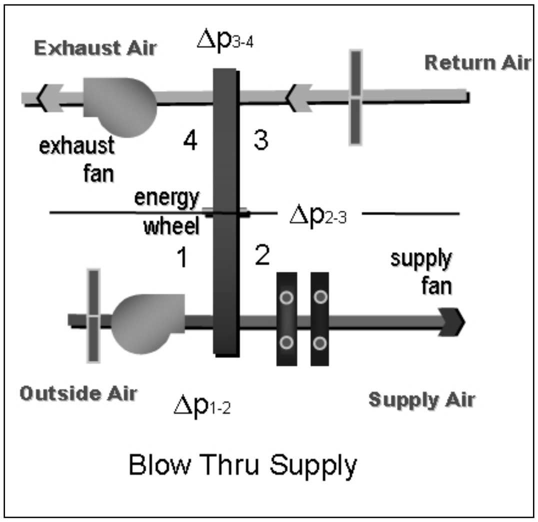


**Figure C2. Placement of Fans for a Blow-through Supply with Draw-Through Exhaust Arrangement**

In this case:

$\Delta P_{2\text{-}3} = 4.0$ in.$H_2O$

and:

EATR = 0.10%
OACF = 1.09

$$Q_{\text{supply blower}} = Q_1 = \left[ \frac{Q_{net\ supply} \cdot OACF}{(1 - EATR/100)} \right] = \left[ \frac{10{,}000 \dfrac{ft^3}{min} \cdot 1.09}{(1 - 0.1/100)} \right] = 10{,}910 \frac{ft^3}{min} \qquad \text{C20}$$

$$Q_{\text{exhaust blower}} = Q_4 = Q_{exhaust} + Q_1 - Q_{net\ supply} = 9000 + 10{,}910 - 10{,}000 = 9{,}910 \frac{ft^3}{min} \qquad \text{C21}$$

From fan selection software or fan curve find BHP for each fan, with energy recovery in the system. For this example:

Supply blower operating at 10,910 cfm @ Total Static Pressure of 4.5 in. $H_2O$.: BHP = 11.78

Exhaust blower operating at 9,910 cfm @ Total Static Pressure of 3.0 in. $H_2O$.: BHP = 7.7

Determine power input to blower motors with energy recovery in the system; direct drive assumed, with motor operating at $\eta_{motor}$ = 92%:

$$\mathrm{Pwr}_{\mathrm{bswer}} = \frac{11.78HP}{0.92} \cdot \left(745.7\frac{W}{HP}\right) = 9{,}548\,W \tag{C22}$$

$$\mathrm{Pwr}_{\mathrm{bewer}} = \frac{7.7HP}{0.92} \cdot \left(745.7\frac{W}{HP}\right) = 6{,}241\,W \tag{C23}$$

From fan selection software or fan curve find BHP for each fan, without energy recovery in the system. For this example:

Supply blower operating at 10,000 cfm @ Total Static Pressure of 3.45 in. $H_2O$.: BHP = 8.36

Exhaust blower operating at 9,000 cfm @ Total Static Pressure of 2.1 in. $H_2O$.: BHP = 4.87

Determine power input to blower motors without energy recovery in the system; direct drive assumed, with motor operating at $\eta_{motor}$ = 92%:

$$\mathrm{Pwr}_{\mathrm{bs}} = \frac{8.36HP}{0.92} \cdot \left(745.7\frac{W}{HP}\right) = 6{,}776\,W \tag{C24}$$

$$\mathrm{Pwr}_{\mathrm{be}} = \frac{4.87HP}{0.92} \cdot \left(745.7\frac{W}{HP}\right) = 3{,}947\,W \tag{C25}$$

Find the additional blower power needed when energy recovery component is added to the system, using Equation 10:

$\mathrm{Pwr}_{\mathrm{blwr}}$ = 9,548+ 6,241 - 6,776 - 3,947 = 5066W

Calculate RER for the example using a blow-through supply fan and draw-through exhaust fan arrangement, using Equation 7:

$$\mathrm{RER}_{\mathrm{Total}} = \left[\frac{AAHX_{\text{net Total Capacity}}}{Pwr_{blwr} + \mathrm{Pwr}_{\mathrm{comp}}}\right] = \left[\frac{385{,}060\frac{Btu}{h}}{5066W + 575\,W}\right] = 68.3Btu(W \cdot h) \tag{C26}$$

For this system the arrangement with both fans in the draw-through positions is 30.5% more efficient at recovering cooling energy at design condition than the arrangements with a blow-through supply fan and draw-through exhaust fan arrangement.

**C5** *Calculating RER for Coil Run Around Loop*

**Sample of Performance Ratings for two AHRI-410 Certified Coils ("Supply" and "Exhaust"):**

| Common Inputs | | Coil Entering Air | Summer | | Winter |
|---|---|---|---|---|---|
| Fluid Type: | Ethylene | Supply EDB (°F): | 95.00 | Supply EDB (°F): | 35.00 |
| Glycol (%): | 25.00 | Supply EWB (°F): | 78.00 | Supply EWB (°F): | 33.00 |
| Altitude (ft): | 0 | Exhaust EDB (°F): | 75.00 | Exhaust EDB (°F): | 70.00 |
| Fixed GPM: | 66.00 | Exhaust EWB (°F): | 63.00 | Exhaust EWB (°F): | 58.00 |

| Construction/Input Data | Summer / Winter Supply | Exhaust |
|---|---|---|
| Airflow (cfm): | 15000 | 15000 |
| Fin Height (in): | 54 | 54 |
| Fin Length (in): | 85 | 85 |
| Rows: | 6 | 6 |
| Fins Per Foot: | 144 | 144 |
| Fin Material: | Aluminum | Aluminum |
| Tube Material: | Copper | Copper |
| Tube Thickness (in): | .020 | .020 |
| Turbs: | Yes | Yes |

| Leaving Coil Data | Summer Supply | Exhaust | Winter Supply | Exhaust |
|---|---|---|---|---|
| TMBH: | 166.67 | 166.67 | 291.16 | 291.16 |
| SMBH: | 166.67 | 166.67 | 291.16 | 291.16 |
| Leaving Air DB (°F): | 84.76 | 85.53 | 52.90 | 52.30 |
| Leaving Air WB (°F): | 75.36 | 66.50 | 42.73 | 50.79 |
| Entering Fluid (°F): | 82.42 | 88.10 | 57.73 | 47.98 |
| Leaving Fluid (°F): | 88.10 | 82.42 | 47.98 | 57.73 |
| Fluid Flow (GPM): | 66.00 | 66.00 | 66.00 | 66.00 |
| Fluid Velocity (ft/s): | 2.03 | 2.03 | 2.03 | 2.03 |
| WPD (ft H2O): | 7.03 | 7.06 | 7.76 | 7.68 |
| Coil Area (ft^2): | 31.88 | 31.88 | 31.88 | 31.88 |
| Face Velocity (ft/min): | 471 | 471 | 471 | 471 |
| APD (in H2O): | 0.52 | 0.52 | 0.52 | 0.52 |

**Calculation of Effectiveness for the Coil Loop:**

For the purposes of this example, net sensible effectiveness can be calculated as follows:

$$\varepsilon_{net\ sensible} = \frac{\text{Net Sensible Capacity}_{\text{AAHX}}}{\dot{m}_{\min} \cdot c_p \cdot (t_1 - t_3)} \quad \text{C27}$$

Where:

$$\dot{m}_{\min} = \left(15{,}000\frac{ft^3}{\min}\right)\cdot\left(60\frac{\min}{h}\right)\cdot\left(0.075\frac{lb}{ft^3}\right) = 67{,}500\frac{lb}{h} \quad \text{C28}$$

and:

$$c_p = 0.24\frac{Btu}{lb\cdot{}^\circ F} \quad \text{C29}$$

Therefore, using the net appropriate sensible capacities from the coil performance ratings above:

$$\varepsilon_{net\ sensible\ cooling} = \frac{166{,}670\frac{Btu}{h}}{\left(67{,}500\frac{lb}{h}\right)\cdot\left(0.024\frac{Btu}{lb\cdot{}^\circ F}\right)\cdot\left(95\,{}^\circ F - 75\,{}^\circ F\right)} = 51.4\% \quad \text{C30}$$

and:

$$\varepsilon_{net\ sensible\ heating} = \frac{-291{,}1600\,\dfrac{Btu}{h}}{\left(67{,}500\,\dfrac{lb}{h}\right)\cdot\left(0.24\,\dfrac{BTU}{lb\cdot{}^{\circ}F}\right)\cdot\left(35\ {}^{\circ}F - 70\ {}^{\circ}F\right)} = 51.4\% \qquad \text{C31}$$

**Calculation of RER for the Coil Loop:**

From Equation 8, $RER_{sensible}$ is given by:

$$\mathrm{RER}_{\mathrm{Sensible}} = \frac{\text{Net Sensible Capacity}_{\mathrm{AAHX}}}{\mathrm{Pwr}_{blwr} + \mathrm{Pwr}_{\mathrm{comp}}} \qquad \text{C32}$$

The net sensible capacity is known from the coil performance ratings, but $Pwr_{blower}$ and $Pwr_{comp}$ must be calculated.

Calculation of $Pwr_{blwr}$ using Equation 11:

Q = 15,000 cfm
C = 8.52 $ft^3$ in $H_2O$./min·W
$\Delta$p = 0.52 in $H_2O$. Exhaust and Supply Pressure Drop
$\eta_{Fan/Motor}$ = 0.92 motor efficiency · 0.707 fan efficiency = 0.65

$$Pwr_{blwr} = \left[\frac{Q_{blower\ supply}\cdot\Delta P_{supply}}{C\cdot\eta_{Fan/Motor\,supply}}\right] + \left[\frac{Q_{blower\ exhaust}\cdot\Delta P_{exhaust}}{C\cdot\eta_{Fan/Motor\,exhaust}}\right] = \left[\frac{\left(15{,}000\,\dfrac{ft^3}{\min}\right)\cdot(0.52\,in.w.g.)}{\left(8.52\,\dfrac{ft^3\cdot in.H_2O}{\min\cdot W}\right)\cdot 0.65}\right] + \left[\frac{\left(15{,}000\,\dfrac{ft^3}{\min}\right)\cdot(0.52\,in.w.g.)}{\left(8.52\,\dfrac{ft^3\cdot in.H_2O}{\min\cdot W}\right)\cdot 0.65}\right] = 2817\ W \qquad \text{C33}$$

Calculation of $Pwr_{comp\ (pump)}$ using Equation 18:

Qp = 66 gpm
$h_A$ = Coils 15ft + Pipe loop 15 ft = 30 ft
SG = 1.03
$C_{pump}$ = 5.30 (gal·ft/min)/W
$\eta_{pump/Motor}$ = 60%

$$Pwr_{comp\,(pump)} = \frac{Q_p\cdot h_A\cdot SG}{C_{pump}\cdot\eta_{pump/motor}} = \frac{\left(66\,\dfrac{gal}{\min}\right)\cdot(30\,ft)\cdot 1.03}{\left(5.30\,\dfrac{gal\cdot ft}{\min\cdot W}\right)\cdot 0.60} = 641.3\,W \qquad \text{C34}$$

Now RER at Cooling Conditions can be calculated:

$$\mathrm{RER}_{\mathrm{sensible}} = \frac{\left(166{,}670\,\dfrac{Btu}{h}\right)}{2892\,W + 647\,W} = 47.1\,\frac{Btu}{W\cdot h} \qquad \text{C35}$$

and at Heating Conditions:

$$\mathrm{RER}_{\mathrm{COP\ sensible}} = \frac{291{,}106\,\frac{Btu}{h}}{2825\,W + 647\,W} \cdot \left( \frac{1\,W}{3.413\,\frac{Btu}{h}} \right) = 24.6 \qquad \text{C36}$$

# APPENDIX D. COMPARING TYPICAL COMBINED EFFICIENCY AND ENERGY ANALYSIS RESULTS IN A VARIETY OF CLIMATES – INFORMATIVE

As stated in the purpose, Combined Efficiency for cooling is calculated at a selected operating condition. As such, it is useful for determining the impact of energy recovery on system efficiency, equipment sizing and peak load at design conditions. It does not constitute a rating system for energy recovery, nor does it substitute for energy analysis in determining energy and/or economic savings. A 20% increase in Combined Efficiency for cooling may or may not represent a 20% savings in energy usage, depending on the climate and the percentage of the total load represented by the outside air. Table D1 below provides examples of how Combined Efficiency, equipment sizing and savings from energy analysis can vary differently with climate. These results are illustrative only; note that energy analysis can vary widely with assumptions, component selection, control strategy, etc. Users are advised to perform an energy analysis for the specific application in order to evaluate the impact of energy recovery on energy use or economics.

**Table D1. Sample Calculation Results for Five Climates**

| Location | Combined Efficiency, cooling, Btu/(W·h) | Annual Cooling Savings ($) | Annual Heating Savings ($) | Fan Energy Used ($) | Annual Net Savings ($) | System Sizing (1-Y) |
|---|---|---|---|---|---|---|
| Miami | 13.35 | 672 | 17 | 129 | 559 | 72% |
| Kansas City | 12.78 | 212 | 570 | 129 | 652 | 76% |
| Minneapolis | 12.19 | 82 | 845 | 129 | 798 | 79% |
| Tucson | 11.84 | 265 | 196 | 129 | 331 | 82% |
| Seattle | 10.60 | 9 | 455 | 129 | 334 | 91% |

Assumptions:

a. Unitary capacity of 10 tons and EER of 10.1 for cooling
b. Gas heat at 80% efficiency
c. Air flow rate of 1200 cfm outside air (approximately 30% outdoor air)
d. Energy recovery enthalpy effectiveness of 75%
e. Energy analysis with commercially available software and bin weather data from TMY-2
f. Office building schedule 8 a.m. to 8 p.m., six days per week, energy costs at $6.52/MMBtu, electricity at $0.079/kWh

# APPENDIX E. DERIVATION OF COEFFICIENTS – INFORMATIVE

### Use of Coefficient $K_2$

$K_2$ is a unit conversion constant in the I-P system. It is commonly used to calculate the static efficiency of blowers. It expresses in horsepower the power of an ideal blower (operating at 100% efficiency) that is pressurizing and moving a volume of air which is characterized in units of in w.g. (pressure change) and cubic feet per minute (volume of moving air). It is defined here as:

$$K_2 = 6356 \text{ in } H_2O.\cdot ft^3/HP\cdot min \qquad \text{E1}$$

### Derivation of Coefficient C

C is a unit conversion constant used to calculate in watts the power required to pressurize a volume of moving air characterized in units of in $H_2O$ (pressure change) and cubic feet per minute (volume of moving air). It is related directly to K2. Given that:

$K_2 = 6356$ (in $H_2O.\cdot ft^3/HP\cdot min$)
1 HP = 746 W

Then:

$$C \frac{in\ H_2O.\cdot\ ft^3}{W\cdot min} = \frac{6356\dfrac{in\ H_2O.\cdot\ ft^3}{HP\cdot min}}{746\dfrac{W}{HP}} = 8.52\,\frac{in\ H_2O\cdot\ ft^3}{W\cdot min} \qquad \text{E2}$$

# APPENDIX F. RATING CONVERSIONS – INFORMATIVE

**F1** Listed here are common conversion factors to accommodate mixed units.

**F1.1** Efficiency:

$$COP = EER / 3.413 \quad \text{F1}$$
$$EER = COP \cdot 3.413 \quad \text{F2}$$

**F1.2** Capacity:

$$kW = Tons \cdot 3515.97 \quad \text{F3}$$
$$kW = Btu/h / 3413 \quad \text{F4}$$
$$Btu/h = kW \cdot 3413 \quad \text{F5}$$
$$Tons = kW/ 3515.97 \quad \text{F6}$$