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**ANSI/ASHRAE Standard 84-2020**

Method of Testing Air-to-Air Heat/Energy Exchangers

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**NOTE**

Approved addenda, errata, or interpretations for this standard can be downloaded free of charge from the ASHRAE website at www.ashrae.org/technology.
FOREWORD

ASHRAE Standard 84 provides rules for the measurement and expression of values characterizing the energy-related performance of air-to-air heat/energy exchangers. The following changes are new in the 2020 edition:

a. Rules are provided for measurement of fixed-bed regenerator performance.

b. The metric "energy recovery ratio," first introduced in ASHRAE Standard 90.1-2016, is defined. This metric differs from the fundamental effectiveness equations in that it characterizes only an exchanger's ability to reduce the load associated with the supply air at a specified condition.

c. The standard was revised to comply with ASHRAE's mandatory language policy.

Versions of the standard prior to the 2008 edition were heavily prescriptive in measurement processes and yielded testing uncertainty within generally acceptable limits. This new edition, in keeping with the 2008 and 2013 editions, instead stipulates a maximum desired uncertainty while allowing laboratories the flexibility of selecting various testing apparatus as long as the uncertainty limits are satisfied. It should be noted that laboratories must evaluate their testing apparatus to ensure that their instrumentation achieves the required test uncertainty. Laboratories also may specify more rigorous uncertainty limits.

Application of the standard is determined by reviewing introductory Sections 1 through 3, which provide the scope and purpose of the standard and defines its terms. Section 4 presents the metrics that express performance of a heat/energy exchanger, the measurements that must be taken, and the equations used to calculate the metrics. This section also identifies the essential required instrumentation and requires pretest uncertainty analysis and appropriate instrument calibrations. Section 5 outlines the basic test procedures. The quality of test data is discussed in Section 6, which outlines the use of measurement inequalities to detect and reject invalid tests. Section 7 sets out the uncertainty levels that a performance test must satisfy to be acceptable. (Informative Appendix D shows how test conditions can be selected to meet specified uncertainty limits.) Section 8 describes acceptable instruments and measurement methods and the ASHRAE standards prescribing relevant measurement methods. The specialized measurement methods required for testing of fixed-bed regenerators, in which steady-state outlet conditions are not achieved, are introduced here for the first time. (Informative Appendix B provides examples from peer-reviewed literature of these measurement methods and testing of fixed-bed regenerators.) Performance calculations and test result reporting are presented in Sections 9 and 10, respectively. In an effort to both provide more flexibility in the design of test labs and to provide additional guidance, all discussion of test system layout has been moved from the normative body of the standard to Informative Appendix A. The special challenges in field testing of air-to-air heat/energy exchangers are discussed in Informative Appendix E. The extrapolation of test performance data is discussed in Informative Appendix F.

1. PURPOSE

The purpose of this standard is to

a. establish a uniform method of test for obtaining the effectiveness of air-to-air heat/energy exchangers;

b. specify the test conditions, data required, uncertainty analysis to be performed, calculations to be used, and reporting procedures for testing the performance of an air-to-air heat/energy exchanger; and

c. specify the types of test equipment for performing such tests.

2. SCOPE

2.1 This standard prescribes the laboratory methods for testing the performance of air-to-air heat and energy exchangers. In this standard, an air-to-air heat/energy exchanger is a device to
(This foreword is not a part of this standard. It is merely informative and does not contain requirements necessary for conformance to the standard. It has not been processed according to the ANSI requirements for a standard and may contain material that has not been subject to public review or a consensus process.)

This is a revision of Standard 84-2020. This standard was prepared under the auspices of the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE). It may be used, in whole or in part, by an association or government agency with due credit to ASHRAE. Adherence is strictly on a voluntary basis and merely in the interests of obtaining uniform standards throughout the industry.

The changes made for the 2023 revision were:

- References were updated
- Bibliography was updated
transfer heat, or heat and water vapor, from one airstream to another. The types of air-to-air heat/energy exchangers covered by this standard are as follows.

2.1.1 Regenerators:

a. Rotary regenerators (including heat wheels and total energy wheels)
b. Fixed-bed regenerators, including double-core regenerators, but excluding indoor fixed-bed regenerators that require connecting ductwork at Station 1 and 4 in which reversing airflows will occur in the field, and single-core fixed-bed regenerators

2.1.2 Exchangers with intermediate energy transfer medium arranged in a closed-loop circuit:

a. Heat pipe exchangers
b. Thermosiphon exchangers
c. Recovery loop exchangers (run-around loops)

2.1.3 Recuperators:

a. Fixed-plate exchangers (including sensible-only and enthalpic plates)

3. DEFINITIONS

adjustable purge: in a rotary regenerator, a segment, whose angle or area is adjustable, that directs outdoor air through a portion of the wheel medium into the exhaust air outlet to limit carryover of exhaust air.

airflow selector: a device consisting of one or multiple actuated dampers, control valves, or deflectors to maintain unidirectional flow for testing of fixed-bed regenerators.

air-to-air energy exchanger: a device to transfer heat and water vapor from one airstream to another.

air-to-air heat exchanger: a device to transfer heat from one airstream to another.

air leakage: air transferred from the exhaust to the supply airstream due to pressure differentials.

carryover: in regenerators, the amount of exhaust air that is moved to the supply from the exhaust by the mechanical operation of the exchanger. [Informative Note: The air trapped within the matrix of an energy wheel as it rotates from the exhaust to the supply airstream is an example of carryover. The exhaust air trapped in the matrix of a fixed-bed regenerator at switch-over is another example of carryover, although for fixed-bed regenerators it also is common to refer to this as displacement leakage. Carryover is distinct from air leakage.]

deviation: the difference between a single result and the mean of many results.

effectiveness: the actual energy transfer rate (sensible, latent, or total) in an exchanger at specific operating conditions divided by the maximum energy transfer rate in an ideal exchanger at the same operating conditions. [Informative Note: An ideal exchanger is a hypothetical exchanger achieving complete exchange—i.e., is 100% effective].

enthalpy recovery ratio: a ratio of the change in enthalpy of the outdoor air supply to the difference in enthalpy between the entering supply airflow and the entering exhaust airflow, with no adjustment to account for that portion of the psychrometric change in the leaving supply airflow that is the result of leakage of entering exhaust airflow rather than exchange of heat or moisture between the airstreams.

exhaust air transfer: the air quantity transferred from the exhaust to the supply.

exhaust air transfer ratio (EATR): the tracer gas concentration difference between the supply air outlet and the supply air inlet, divided by the tracer gas concentration difference between the exhaust air inlet and the supply air inlet, expressed as a percentage.

fixed-bed regenerator: an exchanger where one or more stationary matrices are charged with energy from and discharge energy to alternating flows of the supply and exhaust air. [Informative Note: Sometimes referred to as fixed-bed regenerators, single-core regenerators, and reversing-flow regenerators.]

fixed-plate exchanger: an exchanger with multiple alternate airflow channels, separated by plates capable of transferring heat or heat and water vapor, and connected to supply and exhaust airstreams.
heat pipe exchanger: an exchanger with an array of finned and sealed tubes that are placed in side-by-side supply and exhaust airstreams, fabricated with an internal capillary wick structure in each tube, and filled with a refrigerant heat transfer fluid.

maximum deviation: for a set of multiple measurements of a physical property, where statistical methods have been used to remove spurious data points, the greatest of the deviations of the set of indicated values from the mean of the set. [Informative Note: Guidance on the removal of spurious data points is given in ASHRAE Guideline 2-2010 (RA2014)\textsuperscript{G1}].

net effectiveness: the effectiveness adjusted to account for that portion of the psychrometric change in the leaving supply airflow that is the result of leakage from the entering exhaust airflow rather than exchange of heat or moisture between the airstreams.

net recovery efficiency ratio: the recovery efficiency ratio (RER) adjusted to account for that portion of the psychrometric change in the leaving supply airflow that is the result of leakage from the entering exhaust airflow rather than exchange of heat or moisture between the airstreams.

outdoor air correction factor (OACF): a factor defined as the entering supply airflow divided by the leaving supply airflow.

random error: the portion of the total error that varies randomly in repeated measurements of the true value throughout a test process.

recovery efficiency ratio (RER): a ratio of the energy recovered divided by the energy expended in the energy recovery process.

recovery period: the length of time that the matrix of the regenerator is charged by the entering supply or exhaust air.

recuperator: an exchanger where energy is directly transferred between the supply and exhaust airstream. The two airstreams flow through adjacent but separate channels.

regenerator: an exchanger where an energy transfer medium is exposed alternately to supply and exhaust airstreams for the purpose of recovering energy. The two airstreams pass through the same channels in the transfer medium, alternately.

rotary total energy wheel or enthalpy wheel: an exchanger with porous discs, fabricated from materials with heat and water-vapor retention capacity, that are regenerated by collocated supply and exhaust airstreams.

rotary heat wheel: an exchanger with porous discs, fabricated from materials with heat retention capacity, that are regenerated by collocated supply and exhaust airstreams.

run-around loop (coil recovery loop exchanger): an exchanger that has finned-tube coils with interconnecting piping placed in supply and exhaust airstreams and is filled with heat transfer fluid that is pumped between coils.

standard air: air weighing 1.204 kg/m\textsuperscript{3} (0.075 lb/ft\textsuperscript{3}), which approximates dry air at a temperature of 21.1°C (70°F) and a barometric pressure of 101.3 kPa (29.92 in. Hg).

standard error of estimate (SEE): the measure of dispersion of the dependent variable about a least-squares regression or curve as defined by ANSI/ASME PTC 19.1\textsuperscript{1}.

systematic error: the portion of total error that remains constant in repeated measurements of the true value throughout a test process.

thermosiphon exchanger: an exchanger that contains finned-tube coils with interconnecting piping placed in supply and exhaust airstreams and is filled with a refrigerant heat transfer fluid that moves by gravitational forces only. [Informative Note: Thermosiphons differ from heat pipe exchangers in that the thermosiphon tubes do not include a capillary wick structure.]

ttrue value: the actual physical quantity when the uncertainty is zero.

uncertainty: an expression of the ability of an instrument to indicate or record the true value of a measured quantity. [Informative Note: The error of indication, which is the difference between the indicated value and the true value of the measured quantity, expresses the uncertainty of an instrument when a large number of sample tests are averaged. Uncertainty is a combination of the effects from both systematic error and random error and is defined by ASME PTC 19.1\textsuperscript{1}].
Figure 1 Scheme of airflow for air-to-air heat/energy exchangers.

4. REQUIREMENTS FOR PERFORMANCE TESTING

4.1 Performance Determinations. The performance of an air-to-air heat/energy exchanger is primarily characterized by the following set of metrics: (a) effectiveness and net effectiveness, (b) recovery efficiency ratio and net recovery efficiency ratio, (c) pressure drop, (d) outdoor air correction factor (OACF), and (e) exhaust air transfer ratio (EATR). An additional parameter useful in application of these exchangers is (f) enthalpy recovery ratio. The values for each of these are dependent on mass flow rates and test conditions. [Informative Note: Refer to Informative Appendix C for a comprehensive discussion of effectiveness in performance.]

a. Determination of the effectiveness (ε) and net effectiveness (ε_{net}) of the exchanger, shown schematically in Figure 1, in transferring sensible, latent, and total energy from one airstream to another over the range of specified operating conditions is given by Equations 1 through 6:

\[
\varepsilon_{\text{sensible}} = \frac{\dot{m}_2 (c_{p,1} T_1 - T_2 c_{p,2})}{m_{\text{min}} (T_1 c_{p,1} - T_3 c_{p,3})}
\]  

\[
\varepsilon_{\text{sensible, net}} = \frac{\dot{m}_2 \left[ c_{p,1} T_1 - c_{p,2} T_2 - (\text{EATR}) c_{p,3} T_3 \right]}{m_{\text{min}} (T_1 c_{p,1} - c_{p,3} T_3) (1 - \text{EATR})}
\]

\[
\varepsilon_{\text{latent}} = \frac{\dot{m}_2 (h_{fg,1} W_1 - h_{fg,2} W_2)}{m_{\text{min}} (h_{fg,1} W_1 - h_{fg,3} W_3)}
\]

\[
\varepsilon_{\text{latent, net}} = \frac{\dot{m}_2 \left[ h_{fg,1} W_1 - h_{fg,2} W_2 - (\text{EATR}) h_{fg,3} W_3 \right]}{m_{\text{min}} (h_{fg,1} W_1 - h_{fg,3} W_3) (1 - \text{EATR})}
\]

\[
\varepsilon_{\text{total}} = \frac{\dot{m}_2 (h_1 - h_2)}{m_{\text{min}} (h_1 - h_3)}
\]

\[
\varepsilon_{\text{total, net}} = \frac{\left[ \dot{m}_2 (h_1 - h_2 - (\text{EATR}) h_3) \right]}{m_{\text{min}} (h_1 - h_3) (1 - \text{EATR})}
\]

where

\[\dot{m}_n = \text{mass flow rate at Station } n, \text{ kg/s [lbm/h]}\]

\[m_{\text{min}} = \text{minimum of } \dot{m}_2 \text{ and } \dot{m}_3, \text{ kg/s [lbm/h]}\]

\[c_{p,n} = \text{specific heat of dry air at Station } n, \text{ kJ/(kg·K) [Btu/(lbm·°F)]}\]

\[h_{fg} = \text{heat of vaporization of water, kJ/kg [Btu/lbm]}\]
\[ T_n = \text{dry-bulb temperature at Station } n \text{, } ^\circ\text{F} \]
\[ W_n = \text{humidity at Station } n, \text{ kg water/kg air} \text{ [lbm water/lbm air]} \]
\[ h_n = \text{enthalpy at Station } n, \text{ kJ/kg [Btu/lbm]} \]

EATR = exhaust air transfer ratio

**Informative Notes:**

1. In Equations 1 and 2, when measured temperatures are sufficiently close, uncertainty limits will not be satisfied. In Equations 3 and 4, when measured humidities are sufficiently close, uncertainty limits will not be satisfied. In Equations 5 and 6, when measured enthalpies are sufficiently close, uncertainty limits will not be satisfied.

2. In Equations 5 and 6, when enthalpies are very close, but temperatures or humidities are not, it is possible to calculate total effectiveness greater than 100% or less than 0%. In other cases, when the temperature and humidity transfer flows are opposite, the calculated total effectiveness can be outside the bounds of the sensible and latent effectiveness. In both of these cases, the sensible and latent effectivenesses have more utility and should be used; alternately, Equation 7, using the sensible and latent energy exchange rates, may be used.

\[
\varepsilon_{\text{total}} = \frac{\dot{m}_2 [c_{p,1} T_1 - c_{p,2} T_2] + \dot{m}_3 [W_1 (h_{fg,1} + c_{pv,1} T_1) - W_2 (h_{fg,2} + c_{pv,2} T_2)]}{\dot{m}_{\text{min}} [c_{p,1} T_1 - c_{p,3} T_3] + \dot{m}_{\text{min}} [W_1 (h_{fg,1} + c_{pv,1} T_1) - W_3 (h_{fg,3} + c_{pv,3} T_3)]} \quad (7)
\]

where

\[ \dot{m}_{\text{min}} = \text{lesser of } \dot{m}_2 \text{ and } \dot{m}_3, \text{ kg/s [lbm/s]} \]
\[ h_{fg,n} = \text{latent heat of vaporization of water at Station } n, \text{ kJ/kg [Btu/lbm]} \]
\[ c_{pv,n} = \text{specific heat of water vapor at Station } n, \text{ kJ/(kg-K) [Btu/(lbm-°F)}] \]
\[ c_{p,n} = \text{specific heat of dry air at Station } n, \text{ kJ/(kg-K) [Btu/(lbm-°F)}] \]

b. Determination of the recovery efficiency ratio (RER) and net recovery efficiency ratio (RER\(_{\text{net}}\)) of a heat/energy exchanger is given by Equations 8 and 9 (SI) and Equations 10 and 11 (I-P):

\[
\text{RER} = \frac{\dot{m}_2 (h_1 - h_2)}{\left(\frac{\Delta p_2 Q_2}{1000 \eta_{fe}} + \frac{\Delta p_3 Q_3}{1000 \eta_{fe}} + q_{aux}\right)} \quad \text{[SI]} \quad (8)
\]

\[
\text{RER}_{\text{net}} = \frac{\dot{m}_2 [h_1 - h_2 - (\text{EATR}) h_3]}{(1 - \text{EATR})} \quad \text{[SI]} \quad (9)
\]

where

\[ h_n = \text{enthalpy at Station } n, \text{ kJ/kg} \]
\[ \Delta p_s \text{ and } \Delta p_e = \text{pressure drops across the supply and exhaust sides of the exchanger, respectively, Pa} \]
\[ Q_2 \text{ and } Q_3 = \text{supply and exhaust side volume flow rates, respectively, m}^3/\text{s} \]
\[ \eta_{fe} \text{ and } \eta_{fe} = \text{supply and exhaust air fan and drive total efficiencies, respectively, ratio} \]
\[ q_{aux} = \text{auxiliary total power input to the exchanger, kW [Informative Note: The power used to rotate a regenerative wheel is one example of } q_{aux}] \]
\[ \rho_2 \text{ and } \rho_3 = \text{supply and exhaust air dry-air density, respectively, kg/m}^3 \]

\[
\text{RER} = \frac{\dot{m}_2 (h_1 - h_2)}{\left(\frac{\Delta p_2 Q_2}{2.494 \eta_{fe}} + \frac{\Delta p_3 Q_3}{2.494 \eta_{fe}} + 3413 q_{aux}\right)} \quad \text{[I-P]} \quad (10)
\]
\[ \text{RER}_{\text{net}} = \frac{\dot{m}_2 \left[ h_1 - h_2 - (\text{EATR}) h_3 \right]}{(1 - \text{EATR})} + \frac{\Delta p_2 \Omega_2}{2.494 \eta_{fs}} + \frac{\Delta p_3 \Omega_3}{2.494 \eta_{fe}} + 3413 q_{aux} \]  

[I-P]  

where

\( h_n \) = enthalpy at Station \( n \), Btu/lbm  
\( \Delta p_s \) and \( \Delta p_e \) = air friction pressure drops across the supply and exhaust sides of the exchanger, respectively, in. of water  
\( Q_2 \) and \( Q_3 \) = supply and exhaust side volume flow rates, ft\(^3\)/min  
\( \eta_{fs} \) and \( \eta_{fe} \) = supply and exhaust fan air and drive total efficiencies, ratio  
\( q_{aux} \) = auxiliary total power input to the exchanger, kW [Informative Note: The power used to rotate a regenerative wheel is one example of \( q_{aux} \)]  
\( \rho_2 \) and \( \rho_3 \) = supply and exhaust air dry-air density, respectively, lbm/ft\(^3\)  

c. Determination of the air friction pressure drops (\( \Delta p_s \) and \( \Delta p_e \)) through the heat/energy exchanger at specific operating conditions for both supply and exhaust airstreams is given by Equation 12. A method to determine a standardized air friction pressure drop is given in Section 9.3.  
\[ \Delta p_s = P_{s1} - P_{s2} \quad \text{and} \quad \Delta p_e = P_{s3} - P_{s4} \]  

(12)
d. Determination of the outdoor air correction factor (OACF) of a heat/energy exchanger is given by the equation  
\[ \text{OACF} = \frac{\dot{m}_1}{\dot{m}_2} \]  

(13)  
where \( \dot{m}_1 \) and \( \dot{m}_2 \) are the mass flow rates of dry air at Stations 1 and 2, respectively, kg/s [lbm/h].  
e. Determination of the EATR of a heat/energy exchanger is given by the equation  
\[ \text{EATR} = \frac{C_2 - C_1}{C_3 - C_1} \]  

(14)  
where \( C_1, C_2, \) and \( C_3 \) are the tracer gas concentrations at Stations 1, 2, and 3, respectively.  
f. Determination of the enthalpy recovery ratio of a heat/energy exchanger is given by the equation  
\[ \text{Enthalpy Recover Ratio} = \frac{h_1 - h_2}{h_1 - h_3} \]  

(15)  
where \( h_1, h_2, \) and \( h_3 \) are the enthalpies at Stations 1, 2, and 3, respectively, kJ/kg [Btu/lbm].  
Tests (a) through (f) shall be conducted on the same heat/energy exchanger with no adjustments to the unit equipment configuration between tests. Tests for OACF (d) and EATR (e) are performed at room temperature and humidity at inlet Stations 1 and 3 but with \( \dot{m}_2 = \dot{m}_3 \). Tests (a) and (b) must include a test for EATR (e) performed at the same mass flows and static pressure differentials, and this EATR is to be used in the respective net effectiveness and net recovery efficiency ratio calculations.  
[Informative Notes:  
1. Tests for effectiveness \( \varepsilon \) (a) and RER (b) may be performed at various inlet operating properties.  
2. Tests for air friction pressure drops (c) may be performed at various inlet operation properties or at room temperature and humidity.]  

4.2 Pretest Uncertainty Analysis. A pretest uncertainty analysis, defined in ASME PTC 19.1 \(^1\), shall be performed prior to any testing on all the parameters outlined in Section 4.2. Test points, procedures, and equipment shall be analyzed to confirm that the test conditions provide an accu-
rate measurement for those units. This pretest uncertainty analysis will indicate whether the test analysis will provide results within the uncertainty limits outlined in Section 7 and will also indicate the primary sources of uncertainty for each performance factor.

**Informative Note:** For the selection of test conditions that reduce the uncertainty to a specific value, refer to Informative Appendix D.

4.3 Apparatus. The test apparatus shall consist of four measurement stations. Three measurements shall be taken at each measurement station as follows:

a. Station 1—Supply Inlet: Temperature 1, Humidity 1, Dry-Air Mass Flow Rate 1
b. Station 2—Supply Outlet: Temperature 2, Humidity 2, Dry-Air Mass Flow Rate 2
c. Station 3—Exhaust Inlet: Temperature 3, Humidity 3, Dry-Air Mass Flow Rate 3
d. Station 4—Exhaust Outlet: Temperature 4, Humidity 4, Dry-Air Mass Flow Rate 4

4.3.1 Test Duct Leakage Requirements. Prior to first use of the system and periodically thereafter, the duct system, with no exchanger installed, shall be tested under the maximum negative pressure and flow rate that will be encountered under test or operating conditions. Flow rates shall be determined and must satisfy the mass flow inequality (Equation 23) in Section 6.

4.3.2 Equipment Installation. The equipment to be tested shall be installed in accordance with the manufacturer’s standard installation instructions using recommended installation procedures and accessories. The casing shall be sealed to prevent any infiltration or exfiltration of air.

4.4 Instrument Calibration. All measurement instruments shall be calibrated using sensors, transfer standards, and primary instruments that are traceable to NIST standards. Instrument uncertainty levels shall be shown by the pretest uncertainty analysis to result in expanded uncertainties no greater than those shown in Section 7. Calibration shall be performed prior to testing and after the testing has been completed, meeting the intent of ISO/IEC 17025 2 general requirements for the competence of testing and calibration laboratories (Section 5.5). The calibration curves associated with each instrument shall be available as a permanent record. Laboratories accredited under ISO 17025 2 shall be deemed to be in compliance with this section.

5. TEST PARAMETERS

**[Informative Note: This section applies only to laboratory test facilities. Field tests are addressed in Informative Appendix E.]**

5.1 Thermal Performance. Performance tests are subject to the following provisions.

5.1.1 The test duct, measuring equipment, and the equipment under test shall be operated within test conditions and test operating tolerances defined in Section 6 at all four air-conditioned measurement stations for not less than one-half hour.

**[Informative Note: See Informative Appendix E for a discussion of time constants for the heat/energy exchanger with changes to the operating conditions.]**

5.1.2 For rotary wheel exchangers, the wheel speed shall be measured before and after each group of tests intended to be performed at the same wheel speed.

5.1.3 For fixed-bed regenerators, the recovery period shall be measured before and after each group of tests intended to be performed at the same recovery period.

5.1.4 The dry-air mass flow rates used for calculation shall be those at the supply outlet and exhaust inlet airstreams.

5.1.5 Thermal effectiveness values shall be determined at temperature ranges representative of the application, and instruments used shall be calibrated for those temperature ranges.

5.1.6 Thermal effectiveness tests shall be performed with controlled static pressure conditions at the inlets and outlets, defined as either a

a. specific static pressure differential between the inlets (Stations 1 and 3) or
b. specific static pressure differential between the supply air outlet and the exhaust air inlet (Stations 2 and 3).

5.2 Leakage

5.2.1 Leakage tests shall be performed using a tracer gas test.
5.2.2 Leakage tests using tracer gases and tracer gas sensors shall be made with the heat/energy exchanger operating at controlled airflows and inlet static pressures. Tracer gas concentrations shall be taken in duct sections where the concentration is uniform. This shall include samples taken in the inlet supply (Station 1) and exhaust (Station 3) ducts upstream of the heat/energy exchanger as well as the supply outlet (Station 2) duct.

[Informative Note: For more information, see Shang et al. \[G3\] in Informative Appendix G.]

5.2.3 When leakage tests are made, static pressure differentials as described in Section 5.1.17 shall be recorded, and Station 2 and Station 3 airflow shall be recorded.

5.2.4 Leakage tests shall be made with no heat or moisture being added to the system.

5.3 Pressure Drop. Air friction pressure drop across the heat/energy exchanger from Stations 1 to 2 and Stations 3 to 4 shall be determined at the tested airflows.

5.3.1 For rotary regenerative wheels, the air friction pressure drop shall be obtained with the wheel rotating with no heat or moisture being added to or removed from the system. There shall be equal supply outlet and exhaust inlet dry-air mass flows \(m_1 = m_3\) and specified static pressure differences between the supply outlet and the exhaust inlet \(p_2 - p_3\) specified values greater or less than zero but not exactly zero.

6. OPERATING CONDITIONS, INEQUALITY CHECKS, AND CONDITIONS FOR REJECTION OF TEST DATA

6.1 The inlet air property variations during thermal performance testing shall satisfy the following inequalities:

For inlet temperatures,

\[
\frac{|\delta T_1|}{T_1 - T_3} < 0.02
\]

and

\[
\frac{|\delta T_3|}{T_1 - T_3} < 0.02
\]

where \(\delta T\) is the maximum deviation of any temperature reading from \(T\), its time-averaged mean value.

For inlet humidity ratios,

\[
\frac{|\delta W_1|}{W_1 - W_3} < 0.05 \text{ for } W_1 > W_3
\]

\[
< 0.1 \text{ for } W_1 < W_3
\]

and

\[
\frac{|\delta W_3|}{W_1 - W_3} < 0.05 \text{ for } W_1 > W_3
\]

\[
< 0.1 \text{ for } W_1 < W_3
\]

where \(\delta W\) is the maximum deviation of any humidity ratio from \(W\), its time-averaged mean value.

6.2 During thermal performance testing, the dry airflow mass measured flow rates shall satisfy the following inequality equation:

\[
\frac{|m_1 - m_2 + m_3 - m_4|}{m_{\text{minimum}(1, 3)}} < 0.05
\]

6.3 For all thermal performance tests where no condensate or frosting occurs, the dry airflow mass measured flow rates and the water vapor mass measured flow rates shall satisfy the following inequality equations:
\[
\frac{m_1 \left( \dot{m}_1 W_1 - \dot{m}_2 W_2 + \dot{m}_3 W_3 - \dot{m}_4 W_4 \right)}{m_{\text{minimum}(1,3)} W_1 - W_3} < 0.20
\]  
(21)

\[
\frac{\dot{m}_1 h_1 - \dot{m}_2 h_2 + \dot{m}_3 h_3 - \dot{m}_4 h_4}{m_{\text{minimum}(1,3)} h_1 - h_3} < 0.20
\]  
(22)

6.4 For the case of thermal performance tests of sensible-only devices where no phase change, condensate, or frosting occurs, the measured sensible energy flow rates shall satisfy the inequality of Equation 23 rather than that of Equation 22:

\[
\frac{\dot{m}_1 c_{p,1} t_1 - \dot{m}_2 c_{p,2} t_2 + \dot{m}_3 c_{p,3} t_3 - \dot{m}_4 c_{p,4} t_4}{(m \times c_{p})_{\text{minimum}} [t_1 - t_3]} < 0.20
\]  
(23)

where \((m \times c_{p})_{\text{minimum}}\) is the lesser of \(m_1 c_{p,1}\) and \(m_3 c_{p,3}\).

[Informative Note: In the case of thermal tests at conditions which result in the formation of condensate but not frost, the mass and energy flow rates associated with the condensate may be considered in which case the water vapor mass measured flow rates shall satisfy the inequality:

\[
\frac{\dot{m}_1 W_1 - \dot{m}_2 W_2 + \dot{m}_3 W_3 - \dot{m}_4 W_4 - m_{\text{condensate}}}{m_{\text{minimum}(1,3)} W_1 - W_3} < 0.20
\]  
(24)

and the energy exchange rates shall satisfy the inequality:

\[
\frac{\dot{m}_1 h_1 - \dot{m}_2 h_2 + \dot{m}_3 h_3 - \dot{m}_4 h_4 - \dot{Q}_{\text{condensate}}}{m_{\text{minimum}(1,3)} h_1 - h_3} < 0.20
\]  
(25)

where

- \(\dot{Q}_{\text{condensate}} = \dot{m}_{\text{condensate}} c_{p,t} \) \(c_{p,t}\text{condensate}\)
- \(\dot{m}_{\text{condensate}}\) = measured condensate flow rate at steady-state conditions during the test, kg/s (lb/h)
- \(\dot{m}_{1,2,3,4}\) = mass flow rate at Stations 1 through 4, kg/s (lb/h)
- \(t_{\text{condensate}}\) = measured temperature of the condensate °C (°F)]

6.5 During testing to determine OACF and EATR, the readings shall satisfy the airflow mass inequality:

\[
\frac{|\dot{m}_1 - \dot{m}_2 + \dot{m}_3 - \dot{m}_4|}{m_{\text{minimum}(1,3)}} < 0.05
\]  
(26)

and the tracer gas mass inequality:

\[
\frac{|m_1 C_1 - m_2 C_2 + m_3 C_3 - m_4 C_4|}{m_{\text{minimum}(1,3)} C_1 - C_3} < 0.15
\]  
(27)

The inequalities, Equations 19 to 27, allow for testing with airflow at Station 2 equal to or not equal to airflow at Station 3.

[Informative Note: At unequal airflows, the inequality limits become harder to achieve and in extreme cases, may not be achievable. Also, the uncertainty of the calculated inequalities is greatly increased when considering unequal airflows. Therefore, when testing at unequal airflows, it is important to consider the corresponding uncertainty in these inequalities and whether or not they still meet the tolerance limits when uncertainty is taken into account.]

7. PRE- AND POST-TEST UNCERTAINTY ANALYSIS

For testing of effectiveness, pretest and post-test uncertainties shall satisfy the following uncertainty inequalities:
\[ U(\varepsilon_2) < 5\% \]  
(28)

\[ U(\varepsilon_f) < 7\% \]  
(29)

\[ U(\varepsilon_i) < \frac{|\varepsilon_i - \varepsilon_i| 5\% + |\varepsilon_i - \varepsilon_i| 7\%}{|\varepsilon_i - \varepsilon_i|} \]  
(30)

Otherwise, the results do not satisfy this standard and shall not be reported.

When \( \varepsilon_i = 0 \), or when \( \varepsilon_i \) is not calculated because \( U(\varepsilon_i) \) is too large based on the findings of an uncertainty analysis, then Equation 31 replaces Equation 30.

\[ U(\varepsilon_i) < 5\% \]  
(31)

For RER testing, the pretest and post-test uncertainties shall satisfy the following inequality:

\[ \frac{U(\text{RER})}{\text{RER}} < 0.10 \]  
(32)

For pressure drop testing, the pretest and post-test uncertainties shall satisfy the following inequalities:

\[ \frac{U(\Delta P_x)}{\Delta P_x} < 0.10 \]  
(33)

\[ \frac{U(\Delta P_e)}{\Delta P_e} < 0.10 \]  
(34)

For OACF testing, the pretest and post-test uncertainties shall satisfy the following inequality:

\[ U(\text{OACF}) < 0.02 \]  
(35)

For EATR testing, the pretest and post-test uncertainties shall satisfy the following inequality:

\[ U(\text{EATR}) < 3\% \]  
(36)

8. INSTRUMENTS AND METHODS OF MEASUREMENT

8.1 Systematic and Random Uncertainty. The systematic uncertainty and random uncertainty for each measurement shall be such that the total uncertainty for the heat/energy exchanger effectiveness satisfy the limits in Section 7. When testing fixed-bed regenerators, outlet measurements vary with time. For operating checks and performance calculations, these instantaneous measurements are reduced to an average value. The standard error of estimate (SEE) method shall be used to evaluate the uncertainty related to this data reduction.

[Informative Note: For further information, see ASME PTC19.1 1.]

8.2 Instrumentation. Instrument specification and application shall be in accordance with ANSI/ASHRAE Standards 41.1 3, 41.2 4, 41.3 5, and 41.6 6, respectively, unless otherwise specified in this document.

[Informative Note: Temperature and pressure measuring instruments specified for air volume and density measurements for purposes of airflow determination may not require the same level of uncertainty as that required for air property measurements. Careful selection of instruments will be required to meet overall measurement uncertainties required in this standard.]

8.3 Temperature

8.3.1 Instruments. Temperature measurement shall be made with one of the following instruments in each airstream:

a. Resistance thermometer (RTD)
b. Calibrated thermistor
c. Calibrated thermocouple

8.3.2 Calibration. Calibration of the temperature sensors shall be traceable to NIST standards.

8.4 Humidity
8.4.1 Instruments. Humidity measurements shall be made with one or more of the following instruments in each airstream:

a. Chilled-mirror dew-point sensor
b. Wet-bulb temperature station sensor
c. Calibrated humidity sensor
d. Calibrated Dunmore-type relative humidity sensor
e. Aspirated psychrometer

8.4.2 Calibration. Calibration of the humidity sensors shall be traceable to NIST standards.

8.5 Pressure

8.5.1 Units. Pressure measurements shall be expressed in Pascals (Pa) when SI units are used and in inches of water (in. of water) when I-P units are used. The I-P units shall be based on a one-inch column of distilled water at 68°F (20°C) under standard gravity and a gas column balancing effect based on standard air.

8.5.2 Instruments. Pressure shall be measured by calibrated electrical pressure transducers.

8.5.3 Calibration. Each pressure measuring instrument shall be calibrated at both ends of the scale and at least nine equally spaced intermediate points. Calibration shall be made with an inclined manometer, a micromanometer, or a calibrated standard traceable to NIST Standards.

8.6 Static Pressure

8.6.1 Static Pressure Taps. The static pressure at a point shall be sensed with pressure taps on the wall of the duct as discussed in ASHRAE Standard 41.2-1987 (RA1992) or ASHRAE Standard 41.2-2018.4 The pressure signal shall then be transmitted to an indicator.

8.6.2 Static Pressure Tap Manifolding. At least four taps shall be manifolded outside of the airstream into a piezometer ring as discussed in ASHRAE Standard 41.2-1987 (RA1992) or ASHRAE Standard 41.2-2018.4

[Informative Note: An individual static pressure tap is sensitive only to the pressure in the immediate vicinity of the static pressure hole.]

8.6.3 The static pressure instrument shall have one side connected to four externally manifolded pressure taps in the inlet plenum. The other side of the instrument shall be connected to four externally manifolded pressure taps in the discharge plenum. The construction of the plenum and the location of the taps shall be as detailed in ASHRAE Standard 41.2-1987 (RA1992) or ASHRAE Standard 41.2-2018.4

8.6.4 Pressure taps in the supply outlet and the exhaust inlet shall be used to determine the pressure differential between the supply duct and the exhaust duct.

8.7 Barometric Pressure

8.7.1 Instruments. The barometric pressure shall be measured with uncertainty not to exceed ±0.03 in. Hg (100 Pa).

8.7.2 Calibration. Barometers shall be calibrated against a device traceable to NIST.

8.8 Airflow Measurement. Any flow measurement method (pressure differential device) used shall not exceed the uncertainty introduced by an appropriate flow nozzle or velocity sensor traverse method, as described by ASHRAE Standard 41.2-1987 (RA1992) or ASHRAE Standard 41.2-2018.4

8.9 Tracer Gas Measurement. To measure air transfer from the exhaust to the supply side of an exchanger, an inert tracer gas is injected into a turbulent region of the exhaust inlet. Air samples are then drawn from each of Stations 1 to 4. The sampling equipment required is as follows: (a) multipoint equal-length sampling grids or a single sampling tube with a mixing device 10 diameters or more upstream of measuring Stations 1, 2, 3, and 4 and (b) a means of collecting and transporting air samples from each station to a calibrated tracer gas analyzer. The inert tracer gas shall be injected and mixed into the airstream before the exhaust inlet (Station 3) fan. The uniformity of this tracer gas in the exhaust inlet duct shall be checked.

[Informative Notes:
1. For further information, see Shang et al.1 in Informative Appendix G.
2. The gas selected for this purpose shall not be significantly transferred by mechanisms other than bulk air leakage.]
A continuous airstream shall be drawn from each sampling grid. Samples shall be drawn from these streams by a laboratory-approved sampling procedure. Sample lines shall be short enough that dilution of the sample does not occur due to long sample lines and that the final recorded samples are not affected by the sample line transients.

Air conditions at Stations 1 and 3 shall be at the same temperature and humidity.

Tracer gas tests shall be conducted with equal dry-air mass flow rates in the supply outlet, Station 2, and exhaust inlet, Station 3.

The concentration of the tracer gas shall be high enough that a 0.04% exhaust air transfer rate results in readings above the lower detection limit of the tracer gas analyzer.

[Informative Note: For example, if SF6 is used for the tracer gas, and if the gas analyzer has a reliable lower detection limit of 10 parts per billion, a minimum concentration of 25,000 parts per billion (25 parts per million) must be used for the test.]

8.10 Fixed-Bed Regenerator Performance Testing

[Informative Note: Unlike other exchangers in the scope of this standard, a fixed-bed regenerator operating at steady state inlet conditions produces airstreams with periodically fluctuating air properties. This section provides methodologies to overcome difficulties in measurement posed by these alternating discharge and regeneration cycles when testing fixed-bed regenerators.]

8.10.1 Sampling Rate and Measurement Procedures. Temperature, pressure, and airflow measurements shall be taken at a constant rate and equally spaced throughout the duration of a recovery period. The sampling rate shall be less than or equal to 2 seconds to collect at least 30 samples per recovery period at each of the four measuring stations, using instruments with a response time shorter than the sampling rate. A moving average shall be used to compile these instantaneous measurements, as described in Section 8.10.4. If the recovery period is such that the response rate of the instruments is unable to produce at least 30 readings per recovery period, it must be shown that the results are within 2% of the bag method described in Section 8.10.2.1 or the heated tank method described in Section 8.10.2.2.

8.10.2 Humidity Sampling. Humidity at Station 2 and Station 4 shall be determined using either the bag sampling method in Section 8.10.2.1 or the heated tank method in Section 8.10.2.2. Humidity at Station 1 and Station 3 shall not be determined with these methods.

[Informative Notes:
1. Accurate humidity measurement of leaving airstreams is difficult when testing fixed-bed regenerators because of the periodic change in temperature and humidity.
2. Measurement of humidity at Station 1 and Station 3 does not require use of the bag sampling method in Section 8.10.2.1 or the heated tank method in Section 8.10.2.2 because these stations do not experience fluctuations in humidity during the recovery periods.]

8.10.2.1 Bag Sampling Method. Air samples shall be collected into a sampling bag inside a heated vacuum chamber equipped with a variable-flow gas pump to control fill time (see Figure 2). The filling of the bag shall be started midway through a recovery period. The pump flow rate shall be adjusted to fill no more than 80% of the bag over two recovery periods. The sample shall be fed to a sampling cell or humidity analyzer only after time has been allowed for the sample to reach equilibrium but no less than 5 minutes. The vacuum chamber, sampling lines, and any other component exposed to the air sample shall be insulated to a minimum RSI of 0.25 (m²·K)/W (R-value of 1.5 [ft²·°F·h]/Btu) and heated to a temperature at least 5°C (9°F) above dew-point temperature of the measured airstream to prevent condensation inside the sampling apparatus.

Sampling lines shall be sized to contain no more than 2% of the sampling bag volume. At least 6 bag samples shall be collected from each station only after the entire test system has reached steady state as described in Section 5.1.1. Bag sampling shall be located downstream of the respective airflow measurement location.

[Informative Note: With a one-minute recovery period, it would take 10 minutes to collect a bag sample: 2 minutes to fill, 5 minutes to mix and stabilize, and 2 to 3 minutes to discharge and analyze the sample. This yields 6 bag samples within the 1 hour data collection period. For fixed-bed regenerator testing with longer recovery periods, the data collection period needs to be extended to collect at least 6 bag samples from each station.]

8.10.2.2 Heated Tank Method. To obtain accurate humidity measurements of a fixed-bed regenerator during multiple recovery periods, wet-bulb or dew-point temperatures at Station 2
and Station 4 shall be measured using a heated tank sampling procedure to allow time for the sensor to attain equilibrium, and to ensure reading accuracy (see Figure 3).

The heated tank will act as a physical damper that will obtain stable readings over the course of multiple recovery periods. The tank, sampling lines, and any other component exposed to the air sample shall be insulated to a minimum RSI of 0.25 m²°C/W (R-value of 1.5 [ft²•°F•h]/Btu) and heated to a temperature at least 5°C (9°F) above dew-point temperature of the measured airstream to prevent condensation inside the sampling apparatus.

The air to be sampled shall be pulled through the heated lines and into the heated tank. The heated tank flow rate shall be set to evacuate the tank every 5 to 10 recovery periods. The same volume of air shall be exhausted from the heated tank and its temperature and humidity measured to determine the humidity content of the sampled air. The sampled air shall then be returned downstream of all other temperature measurements.

8.10.3 Tracer Gas Concentration. The bag sampling technique described in Section 8.10.2.1 shall be used for tracer gas testing with the following requirement. Sampling bags shall be replaced after each collected sample, or flushed with nitrogen at least three times if reused for subsequent measurements.

Alternately, the tracer gas concentrations shall be measured using the heated tank setup as described in Section 8.10.2.2.

The unit controls shall be adjusted to produce the same recovery period as used for thermal performance test in Section 5.1. The testing shall be run for a minimum of 1 hour.

8.10.4 Moving Average for Performance and Operating Checks. Outlet conditions are known to vary with time and shall be averaged with a simple moving average to capture the net performance of the unit.

The moving average shall be calculated from instantaneous data collected over a length of time equal to two recovery periods. The moving average is recalculated every time another data point is collected.
Figure 3 Schematic setup for heated tank equipment.

\[
\bar{X}_m = \frac{(X_1 + X_2 + X_3 + \ldots + X_B)}{B}
\]  

(37)

\[
B = \text{total number of measurements in two recovery periods}
\]

\[
\bar{X}_m = \text{moving average at latest measurement}
\]

\[
X_1 = \text{latest measurement}
\]

\[
X_2 = \text{previous measurement}
\]

\[
X_3 = \text{measurement before } X_2
\]

\[
X_B = \text{measurement taken } B \text{ samples before}
\]

This method creates a series of average values that shall be used to confirm equilibrium condition. The moving average updates continuously, allowing for continuous data collection without the need to establish the end of a recovery period.

8.10.5 Equilibrium Conditions. The test duct, instrumentation, and equipment under test shall be operated until equilibrium conditions are attained but for not less than 1 hour. A fixed-bed regenerator operating at steady state produces fluctuating outlet conditions despite steady inlets. To determine steady-state operating condition, the inlet deviations stated in Section 6 shall be applied to the moving average values at the outlets. The moving average values shall be used for operating conditions, inequality checks, and conditions for data rejection stated in Section 6.

8.10.6 Equipment Installation. One-directional airflow shall be maintained at all four measurement stations. Double-core units with reversing airflow shall be tested with an auxiliary airflow selector installed between Stations 1 and 4 and the test unit (see Figure 4). The auxiliary airflow selector(s) shall change position at the same time and speed as the dampers or reversing fans in the equipment under test. Duct length from end of heat exchanger to airflow selector shall not exceed 3 duct diameters.

[Informative Note: Double-core fixed-bed regenerators with reversing airflow require an auxiliary airflow selector to facilitate testing. The connecting ductwork and the airflow selector adversely affect EATR measurement due to the additional volume of trapped exhaust air. After each airflow switchover, this trapped exhaust air is carried over to the supply airstream. Duct layout for lab testing is important and needs to be optimized to obtain representative EATR values. The smaller the airflow selector and duct volume, the lower the additional EATR contribution. In the field, a roof top fixed-bed regenerator operates without auxiliary airflow selectors and connecting duct work. Therefore, EATR during operation might be slightly lower than lab test results.]

When testing with auxiliary airflow selectors, the EATR measured by tracer gas shall be corrected to account for seal leakage and exhaust air trapped in the auxiliary airflow selector.
Mode 1

Mode 2

Figure 4 Conceptual flow diagram showing function of airflow selectors with fixed-bed regenerators for operational or testing purposes.

First, the seal leakage rate of the auxiliary airflow selector shall be determined with the airflow leakage rate using ambient air test method described in ANSI/AMCA 500-D, Laboratory Methods of Testing Dampers for Rating, Section 6.2. A minimum of five measurements shall be taken at equal increments of pressure differential covering the range anticipated during thermal testing of the fixed-bed regenerator. The purpose of this test is to determine the relation between static pressure and seal leakage rate inside the auxiliary airflow selector.

Next, the volume of air in the auxiliary airflow selector and the ducts connecting it to the unit under test shall be calculated based on their physical dimensions. The transfer rate shall be obtained from trapped exhaust air volume and recovery period given by the following equation:

\[ \text{TR}_{EA(AAFS)} = \frac{V_{EA(AAFS)}}{2 \times \text{RP}} \quad (38) \]

where

- \( \text{TR}_{EA(AAFS)} \) = transfer rate of trapped exhaust air from the auxiliary airflow selector and connecting duct to the supply airstream, \( \text{m}^3/\text{s} \) [\( \text{ft}^3/\text{min} \)]
- \( V_{EA(AAFS)} \) = combined volume of trapped exhaust in auxiliary airflow selector and connecting duct, \( \text{m}^3 \) [\( \text{ft}^3 \)]
- \( \text{RP} \) = recovery period, s [minutes]

\( \text{ANSI/ASHRAE Standard 84-2020} \)
[Informative Note: Equation 38, $V_{EA(aofs)}$ is divided by 2 because in any one cycle only half of the air in the auxiliary airflow selector and connecting ducts is exhaust air. This equation holds true even if the connecting ducts are of different volume or if the location of the damper is asymmetrical within the selector volume. The average over any two recovery periods still will equal half of the total volume.]

Finally, the following equation provides the adjusted exhaust air transfer rate.

$$\text{EATR} = \text{EATR}_{FBR(AAFS)} - \frac{\text{TR}_{seals} + \text{TR}_{EA(AAFS)}}{Q_1}$$  \hspace{1cm} (39)

where

- $\text{EATR}_{FBR(AAFS)}$ = EATR determined with tracer gas measurement as per Section 8.10 for a fixed-bed regenerator connected to an auxiliary airflow selector for testing purposes
- $\text{TR}_{seals}$ = seal leakage rate obtained from AMCA 500-D \textsuperscript{8} damper leakage test, m\textsuperscript{3}/s [ft\textsuperscript{3}/min]
- $\text{TR}_{EA(AAFS)}$ = transfer rate of trapped exhaust from the auxiliary airflow selector and connecting duct, m\textsuperscript{3}/s [ft\textsuperscript{3}/min]
- $Q_1$ = supply airflow rate at Station 1, m\textsuperscript{3}/s [ft\textsuperscript{3}/min]

[Informative Note: When rectangular duct is used, calculate the equivalent duct diameter $\text{Dia}_{equiv}$ using the following formula:

$$\text{Dia}_{equiv} = \frac{1.3(a \times b)^{0.625}}{(a + b)^{0.25}}$$  \hspace{1cm} (40)

where

- $a$ = the major dimension of the duct, m (ft)
- $b$ = the minor dimension of the duct, m (ft)]

8.11 Adjustable Purge Setting. When a rotary regenerator with adjustable purge section is tested, purge angle or area setting shall be recorded for all tests.

9. CALCULATIONS

9.1 Airflow Rate. The airflow rate calculations shall be based on the measurements obtained in Section 8.8.

9.2 Total Enthalpy. The total enthalpy shall be calculated from the following equations:

$$h = 1.006 \times T + W \times h_g$$  \hspace{1cm} [SI]  \hspace{1cm} (41)

or

$$h = 0.240 \times T + W \times h_g$$  \hspace{1cm} [I-P]  \hspace{1cm} (42)

where $h$ is the specific enthalpy of moist air, kJ/kg [Btu/lbm], and

$$h_g = 2501 + 1.86 \times T$$  \hspace{1cm} [SI]  \hspace{1cm} (43)

or

$$h_g = 1061 + 0.444 \times T$$  \hspace{1cm} [I-P]  \hspace{1cm} (44)

where

- $T$ = dry-bulb temperature, C [°F]
- $W$ = humidity ratio of air, kg\textsubscript{water}/kg\textsubscript{dry air} [lb\textsubscript{m\textsuperscript{water}}/lb\textsubscript{m\textsuperscript{dry air}}]

For all temperatures, $h_g$ is the enthalpy for saturated water vapor and shall be obtained from psychrometric tables or software.

[Informative Note: One source of this data is 2017 ASHRAE Handbook—Fundamentals \textsuperscript{04}, Chapter 1, Table 3, "Thermodynamic Properties of Water Saturation."]
9.3 The Standardized Air Friction Pressure Drop

9.3.1 The standardized air-friction pressure drop ($\Delta P_s$) shall be determined by the following equations:

$$V_s = \frac{V_P}{\rho_s}$$

(45)

and

$$\Delta P_s = \Delta P \left( \frac{\rho}{\rho_s} \right)^{\frac{m}{s}}$$

(46)

where

$\Delta P$ = air friction pressure drop measured at the test conditions

$m$ = flow exponent determined in Section 9.3.2

$V$ = velocity, m/s [fpm]

$\rho$ = density, kg/m$^3$ [lbm/ft$^3$]

$\mu$ = air viscosity, kg/(m·s) [lbm/(ft·h)]

$s$ = standardized

9.3.2 The flow exponent shall be found as follows: The air friction pressure drop shall be obtained for a series of face velocities at a fixed temperature. The coefficients $a$ and $b$ will be calculated by fitting the results to the following equation:

$$\Delta P = a \times V^{2b}$$

(47)

The flow exponent $m$ in Equation 46 is then found from the following:

$$m = 2 - b$$

(48)

9.3.3 Alternatively, the results shall be presented as follows:

$$\Delta P = C_{dp} V^{2-m} \rho^{1-m} \mu^m$$

(49)

where

$$C_{dp} = \frac{a}{\rho^{1-m} \mu^m}$$

(50)

and where $a$ is determined as in Section 9.3.2 and $\rho$ and $\mu$ are evaluated at the test temperature in Section 9.3.2.

9.4 Outdoor Air Correction Factor

9.4.1 The outdoor air correction factor (OACF) shall be calculated using Equation 13.

9.4.2 An OACF shall be expressed as a ratio at the recorded pressure differential ($p_2 - p_3$) between leaving supply air and entering exhaust air.

[Informative Note: OACF characterizes mechanical air transfer from the entering supply airstream to the leaving exhaust airstream. OACF is used to determine the entering supply airflow required to provide the gross leaving supply airflow desired.]

9.5 Exhaust Air Transfer

9.5.1 Tracer gas measurements shall be used to calculate leakage. During tracer gas analysis, any effects due to tracer gas concentration in the ambient air (background) shall be minimized. The exhaust air transfer ratio (EATR) shall be calculated using Equation 14.

9.5.2 A minimum of six EATR readings shall be obtained and used for each pressure differential condition to obtain an average value of the EATR.

9.5.3 An EATR shall be expressed as a percentage, at the recorded pressure differential ($p_2 - p_3$) between leaving supply air and entering exhaust air.
10. REPORTING RESULTS

10.1 Reporting Requirements. Test results shall not be reported as meeting the requirements of this standard unless

a. the stability of the inlet and outlet conditions are within the limits defined by Equations 16 through 19;
b. the air and vapor mass and energy flows are within the relevant inequality limits defined by Equations 20 through 25;
c. the air and tracer gas mass flows are within the inequality limits defined by Equations 26 and 27; and

d. the applicable uncertainty limits specified in Equations 28 through 36 are satisfied.

10.2 Results of Test. Test results shall be reported at no less than two selected mass flow rates and no less than two ratios of mass flow rates for the following:

a. The effectivenesses, sensible ($\varepsilon_s$), latent ($\varepsilon_l$), and total ($\varepsilon_t$), and the net effectivenesses, sensible ($\varepsilon_{s,net}$), latent ($\varepsilon_{l,net}$), and total ($\varepsilon_{t,net}$)
b. RER and net RER
c. Air friction pressure drops ($\Delta p_d$ and $\Delta p_f$)
d. The mass flow rates $m_d$ and $m_e$ or mass flow rate $m_j$ and mass flow rate ratio, for each of the tests

Results shall be reported at no less than three selected static pressure differences ($p_2 - p_3$); one shall be negative, one shall be zero, and one shall be positive, at one or more selected mass flow rates and ratios of mass flow rates for the following:

a. OACF
b. EATR

10.3 Test Conditions. Reports of performance test results of air-to-air exchangers in a laboratory shall include the following data:

a. Operating test conditions: Flow rates, temperatures, humidities, and static pressures or pressure differentials at Stations 1 and 3 or Stations 2 and 3.
b. The variations in the inlet and outlet operating conditions as defined by Equations 16 through 19 for all the data used to calculate $\varepsilon_s$, $\varepsilon_l$, $\varepsilon_t$, and RER.
c. The relevant mass flow and energy inequalities given by Equations 20 through 25 for all the data used to calculate $\varepsilon_s$, $\varepsilon_l$, $\varepsilon_t$, and RER.
d. The mass flow inequalities of air and tracer gas as defined by Equations 26 and 27 for all the data used to calculate OACF and EATR.
e. For rotary regenerators with adjustable purge, the adjustable purge setting(s) must be reported for each test.
f. For fixed-bed regenerators, the recovery period must be reported for each test.

10.4 Uncertainties of Results. The uncertainties of each result in Section 10.2 shall be reported. Uncertainties shall be reported for all performance factors at the 95% data coverage level as described in ASME PTC 19.1.

11. NOMENCLATURE

11.1 Symbols (SI [1-P])

\[ \begin{align*}
A &= \text{cross-sectional area, } m^2 [\text{ft}^2] \\
B &= \text{number of recovery periods, fixed-bed regenerator testing} \\
C &= \text{tracer gas concentration} \\
C_{dp} &= \text{alternate coefficient for expression of pressure drop} \\
C_{her} &= \text{heat capacity rate} \\
c_p &= \text{specific heat of dry air, kJ/(kg} \cdot K) [\text{Btu/(lbm} \cdot \text{°F})] \\
c_{pv, n} &= \text{specific heat of water vapor at Station } n, \text{kJ/(kg} \cdot K) [\text{Btu/(lbm} \cdot \text{°F})] \\
D_{eq} &= \text{equivalent duct diameter, m [ft]} \\
EATR &= \text{exhaust air transfer ratio}
\end{align*} \]
\( \text{RER} \) = recovery efficiency ratio  
\( \text{RER}_{\text{net}} \) = net recovery efficiency ratio  
\( h \) = enthalpy, kJ/kg [Btu/lb \(_m\)]  
\( h_{fg} \) = heat of vaporization of water, kJ/kg [Btu/lb \(_m\)]  
\( h_g \) = enthalpy for saturated water vapor, kJ/kg [Btu/lb \(_m\)]  
\( m \) = flow exponent  
\( \dot{m} \) = mass flow rate of dry air, kg/s [lb \(_m\)/h]  
\( \text{OACF} \) = outdoor air correction factor  
\( p \) = pressure, Pa [in. of water]  
\( Q \) = volume flow rate of air, \( \text{m}^3/\text{s} \) [\( \text{ft}^3/\text{min} \)]  
\( q_{aux} \) = auxiliary total power input to the exchanger, kW  
\( R \) = specific gas constant, kJ/(kg·K) [Btu/(lb·R)]  
\( \text{RP} \) = recovery period s (h)  
\( S \) = standard deviations  
\( T \) = temperature, K [°F]  
\( \text{TR}_{\text{EA(AAFS)}} \) = transfer rate of trapped exhaust air from the auxiliary airflow selector and connecting duct to the supply airstream [\( \text{ft}^3/\text{min} \)]  
\( U \) = uncertainty in measured data or calculated result  
\( V \) = volume, \( \text{m}^3 \) (\( \text{ft}^3 \))  
\( v \) = air velocity, m/s [fpm]  
\( W \) = humidity ratio, kg water/kg dry air [lb \(_m\) water/lb \(_m\) dry air]  
\( X \) = variable representing temperature, humidity ratio, or enthalpy  
\( \bar{x} \) = mean value of the independent measurement, \( x \)  
\( X_m \) = moving average  
\( \Delta p \) = air friction pressure drop at test conditions due to airflow through the exchanger, Pa [in. of water]  
\( \Delta P_s \) = standardized air friction pressure drop, Pa [in. of water]  
\( \varepsilon \) = heat/energy transfer effectiveness  
\( \varepsilon_l \) = latent energy transfer effectiveness  
\( \varepsilon_{l,net} \) = net latent energy transfer effectiveness  
\( \varepsilon_s \) = sensible energy transfer effectiveness  
\( \varepsilon_{s,net} \) = net sensible energy transfer effectiveness  
\( \varepsilon_t \) = total energy transfer effectiveness  
\( \varepsilon_{t,net} \) = net total energy transfer effectiveness  
\( \mu \) = air viscosity, kg/(m·s) [lb \(_m\)/(ft·h)]  
\( \eta_f \) = overall fan efficiency  
\( \rho \) = dry-air density, kg/m\(^3\) [lb \(_m\)/ft\(^3\)]

### 11.2 Subscripts

\( a \) = air  
\( \text{atm} \) = atmospheric  
\( \text{aux} \) = auxiliary  
\( e \) = exhaust side  
\( HX \) = heat/energy exchanger  
\( lm \) = log mean
\( m \) = mass of water vapor
\( \text{min} \) = minimum
\( n \) = number of the variable in question
\( OC \) = operating condition
\( p \) = velocity probe
\( RSS \) = root - sum - square uncertainty limit (95%)
\( s \) = supply side, spatial variation, or standardized
1 = supply side inlet
2 = supply side outlet
3 = exhaust side inlet
4 = exhaust side outlet

12. NORMATIVE REFERENCES

INFORMATIVE APPENDIX A
LABORATORY TEST CONFIGURATIONS

Figure A-1 Generalized test system configuration.

1. Supply airflow to exchanger
2. Supply airflow from exchanger
3. Exhaust airflow to exchanger
4. Exhaust airflow from exchanger
5. Device under test (see detail figures)
6. Airflow device and static pressure controller
7. Static pressure measurement device
8. Airflow volume measurement device
9. Air mixer
10. Air conditioning apparatus
11. Relief inlet or outlet
12. Optional recycling duct
13. Temperature, humidity, and tracer-gas measuring stations

Figure A-2 Example test configuration with plenums for cross-flow plate exchanger.
Plenums shown shall meet the minimum requirements shown in Figures A-7 and A-8.
Figure A-3 Alternate test configuration for cross-flow plate exchanger.
When using this test configuration, the location of the static pressure taps must be shown to result in the same measurement values as with plenums as shown in Figures A-7 and A-8.

Figure A-4 Test configuration for rotary regenerator.
Plenums shown shall meet the minimum requirements shown in Figures A-7 and A-8.
Figure A-5 Double-core fixed-bed regenerator Test Configuration 1.
Single-damper regenerator with auxiliary airflow selector used to maintain one-directional flow at all measurement stations. Dashed line indicates device under test. Auxiliary airflow selector is required for testing purposes only and may be supplied by the manufacturer.

Notes:
1. Equipment of this type, when intended for installation indoors with ducts connecting Beds 1 and 2 to the outdoors such that airflows reverse in those ducts, is not in the scope of this standard.
2. Equipment of this type, when intended for installation outdoors with Beds 1 and 2 drawing directly from and discharging directly to the outdoor, is within the scope of this standard.
Figure A-6 Test configuration for double-core fixed-bed two-damper regenerators. Two-damper unit with one-directional airflow at all stations. Dashed line indicates device under test.
Figure A-7 Design for inlet and outlet plenums.
"H" is the larger of the height or width of the exchanger face. Depth of plenum indicated as H/2 shall be no less than H/2. Included angle shall be no greater than 40°.

Figure A-8 Splitter plate for inlet plenums.
For inlet plenums only, when included angle of transition section is greater than 20 degrees, a splitter plate must be provided, extending half the total plenum length L.
INFORMATIVE APPENDIX B
TRANSIENT TESTING OF ENERGY EXCHANGERS
USING A BAG SAMPLING METHOD

This appendix provides examples from the literature on (a) transient measurements using the bag sampling method and (b) transient testing of energy exchangers in support of the bag sampling method proposed in this standard.

B1. BAG SAMPLING METHOD

Polasek and Bullin\(^5\) compare the transient measurements of carbon monoxide (CO) measured with a continuous analyzer and a bag sampling method. They conclude that "when the samples to the bag samplers and continuous analyzers were drawn through a common header, the concentrations agreed to within ±1 ppm for at least 90% of the data points." Some details of the experiment and analysis are listed below.

a. Measurements were made along a road where CO concentrations were shown to have a cycle period as low as 30 seconds.
b. The data from the continuous analyzers were recorded every 10 seconds and averaged over 15 minutes.
c. The bags were continuously filled for 15 minutes, providing an average over 15 minutes.
d. The average CO concentration measured by the continuous analyzer and the bag sampler were compared.

Groves and Zellers\(^6\) present the effects of condensation of water vapor in Tedlar bags on the concentration of organic compounds in the bag. The study found that "differences between mean concentrations in wet and dry bags were significant only for methanol, which yielded a mean-wet-bag concentration approximately 10% lower than for dry bags." This shows that care must be taken to avoid condensation conditions in the sample bags.

B2. TRANSIENT TESTING OF ENERGY EXCHANGERS

Several studies\(^7\)–\(^14\) have been conducted at the University of Saskatchewan on the transient testing of fixed-bed regenerators. These have shown that the average outlet temperature and humidity can be determined when the fixed-bed exchanger is exposed to only a step change in temperature or a step change in humidity ratio but not both. The process in the studies was to deconvolute the sensor response from the measured response of the fixed-bed and sensor to get the response of the fixed-bed alone. Using the data in Abe et al.\(^13\) and Wang et al.\(^14\) and a cycle time of 1 minute (typical for fixed-bed regenerators), the average temperature and humidity ratio at the outlet of the tested beds are expected to be within 5% to 10% of the average that would be measured with an ideal sensor (i.e., a sensor that responds immediately to changes in temperature or humidity). However, Wang et al.\(^9\)\(^,\)\(^10\) show that if the bed is exposed to a simultaneous change in temperature and humidity ratio, common humidity sensors are not able to provide an accurate average outlet humidity ratio. Therefore, a bag sampling method is proposed in ASHRAE Standard 84 to measure the average outlet humidity when testing fixed-bed regenerators exposed to test conditions with a simultaneous change in temperature and humidity ratio (such as AHRI test conditions).
INFORMATIVE APPENDIX C
AN EXPLANATION FOR THE USE OF EFFECTIVENESSES TO CHARACTERIZE AIR-TO-AIR HEAT/ENERGY EXCHANGERS

This appendix provides an explanation of the origins for the definition of effectiveness for heat exchangers, how it is applied, and how it is generalized to include water vapor and total energy exchange between the supply and exhaust airstreams. Four different effectiveness dimensionless ratios are defined for air-to-air heat/energy exchangers: water vapor mass effectiveness ($\varepsilon_m$), latent energy effectiveness ($\varepsilon_I$), sensible energy effectiveness ($\varepsilon_s$), and total energy effectiveness ($\varepsilon_t$); however, only two need to be determined at each test condition because the other two may be deduced from the operating condition data and well-known property data for air. This is followed by some recent research findings on the measurement of the performance of air-to-air heat/energy exchangers as it relates to these effectiveness definitions. Finally, the method of testing and analysis is specified that will minimize uncertainty and reduce costs.

C1. DEVELOPMENT OF EFFECTIVENESS DEFINITIONS

Effectiveness has been widely accepted in the engineering literature as the best measure of thermal performance for the design of heat exchangers that exchange sensible energy between the two fluid streams inside a heat exchanger. For the design of heat exchangers with only sensible energy changes in the fluids, the designer needs to know only the inlet mass flow rates, specific heats, temperatures, and the effectiveness, which is defined as follows:

$$\varepsilon = \frac{\text{Heat Rate}}{\text{Thermodynamic Maximum Heat Rate}}$$

For sensible energy changes in heat exchangers, effectiveness is essentially constant for moderate changes in fluid temperatures and the specified flow rates. Effectiveness has a thermodynamic range between 0 and 1.0. ASME PTC 30 \textsuperscript{G15} test procedure is available to determine the effectiveness of heat exchangers of different thermal properties and geometric design. As a result, effectiveness has become a commonly used performance factor for heat exchangers that exchange sensible energy between two fluid flow streams operating at steady state such that there is no exchange of mass between the fluid flow streams. For the special cases of no phase changes or chemical reactions in the fluid flow streams and constant thermal fluid properties, equations and graphical effectiveness data have been developed for certain typical heat exchangers and their geometries. These effectiveness equations and graphs are presented as functions of two dimensionless independent operating factors: number of transfer units (NTU) and heat capacity rate ratio ($C_v$).

Air-to-air heat/energy exchangers can differ somewhat from the idealized model of a sensible heat exchanger by

a. leaking air from one airstream to another or between the exchanger and its surrounding air,
b. transferring water vapor from one airstream to another through a semipermeable membrane or by cyclic exposure of the airstreams to a desiccatant material within the exchanger,
c. permitting water vapor condensation and/or frosting in one of the airstreams, and
d. exchanging heat with the surroundings.

Each of these physical processes for air-to-air heat/energy exchangers are inconsistent with the above definition of heat exchanger effectiveness (i.e., with constant mass flow rates for each flow stream and steady state fluid thermal properties), so they all must have only a small impact on the effectiveness of the aforementioned effectiveness equations, and graphs are to be used for air-to-air exchangers. A few types of air-to-air heat exchangers may nearly fit these restrictions (e.g., run-around loop systems using finned-coil exchangers, heat pipes, plate exchangers with no leaks or permeable membranes) provided the steady-state operating condi-
tions do not include chemical reactions or phase changes in the fluid flows (e.g., combustion, condensation, or frosting) and the heat rate between the device or system and its energy exchange rate with the surroundings is negligible during a standard test. Even here we cannot assume these devices and systems will produce a unique effectiveness without knowing additional dimensionless independent factors (e.g., the relative heat capacity rate ratio for the pumped run-around loop heat exchanger system and the tilt angle and internal fluid properties and geometry of the heat pipe exchanger).

Shang and Besant G16,G17,G18, Simonson and Besant G19,G20, and Cieplinski et al. G21 also show that when phase change energy effects are properly accounted for and there is no condensation or frosting, the measured data for a Standard 84 test can be used to calculate each of the sensible, latent, and total effectiveness, as defined in Standard 84, at each of their uncertainties. However, unlike heat exchangers, these effectiveness values may, under special test conditions, exceed 100% and be less than zero.

Generally, the design of air-to-air heat/energy exchangers is such that air leakage and heat exchange with the surroundings are made small so that their effect on the thermal performance of the heat/energy exchanger is small or negligible. On the other hand, the transfer of water vapor from one airstream to another is, for some types of air-to-air heat/energy exchangers, a design feature that can substantially change the performance characteristics and cause the widely used effectiveness for sensible energy exchange to change significantly with the operating temperature and humidity and not just with the ratio of the flow rates. Condensation and frosting due to the exposure of airborne water vapor to surfaces below the dew point can occur in any type of air-to-air heat/energy exchanger. When this occurs, the conditions assumed for the development of effectiveness in heat exchangers with only sensible energy changes are no longer valid, and sensible-energy heat-exchanger effectiveness can change significantly with the operating conditions for two selected inlet airflow rates.

Despite the apparent conflicts between classical heat-exchanger theory for fluids that undergo only sensible energy changes and the physical characteristics of air-to-air heat/energy exchangers, the designer needs one or more well-defined performance factors for air-to-air heat/energy exchangers. These performance factors need to be (a) defined in an unambiguous or physically consistent manner, (b) easily understood and used by the engineering community, and (c) determined by a set of tests that can be completed at reasonable cost and to a specified uncertainty. With these needs in mind, the effectiveness, originally defined for heat exchangers in which only sensible energy exchange occurs, remains the best way to define performance factors that can fully characterize energy and water vapor exchanges within air-to-air heat/energy exchangers. At least two independent definitions of effectiveness are required to characterize the energy and water vapor transfer within air-to-air heat/energy exchangers: one to characterize the energy exchange and another for the water vapor exchange between the two airstreams—e.g., total and latent energy effectiveness or sensible and latent effectiveness. It should be noted that, although total energy and chemical species (i.e., dry air and water) will be conserved at steady state in an exchanger, the components of energy (i.e., sensible energy) will only be conserved when phase changes in the airstreams are zero. When water vapor is not exchanged, only one performance factor for energy effectiveness is required; however, it must be noted that this effectiveness may not be independent of the operating conditions when condensation and/or frosting occurs in one of the airstreams.

In an airstream flowing steadily at a local density (\(\rho'\)), temperature (\(T'\)), humidity ratio (\(W'\)), and speed (\(u'\)) into or out of a heat/energy exchanger through a connecting duct of area \(A\), we can define the flow of mass as follows:

\[
\int_0^A \rho'_i u' dA = \dot{m}_i
\]  

(C-2)

where \(\rho'_i\) is the density of species \(i\) (e.g., water vapor or dry air) and \(\dot{m}_i\) is the mass flow rate of species \(i\). For water vapor, we can write its mass flow rate in terms of the mass flow rate of dry air (\(\dot{m}\)) and the bulk average humidity ratio (\(W\)) as follows:

\[
\dot{m}_{wv} = \dot{m} W
\]  

(C-3)
For the flow of sensible energy in the stream of area \( A \), we can write the following:

\[
\int_0^A \rho' u' c'_p T \, dA = C_{hcr} T
\]  \hspace{1cm} (C-4)

where

\[
c'_p = \text{local specific heat of dry air}
\]

\[
C_{hcr} = \text{heat capacity rate} \left( C = \int_0^A \rho' u' C' \, dA \right)
\]

\[
T = \text{bulk average temperature of the flowing air}
\]

For total energy or enthalpy, which for a unit mass of air is defined as follows:

\[
h = C_p T + h_{fg} W = t + W(2501 + 1.805t) \quad \text{kJ/kg} \]  \hspace{1cm} (C-5)

when \( t = ^\circ C \). We can write the equation for sensible plus latent energy as follows:

\[
\int_0^A \rho' u' h' \, dA = \dot{m} h
\]  \hspace{1cm} (C-6)

where \( \dot{m} \) is the mass flow rate of dry air and \( h' \) is the bulk average enthalpy.

Using the effectiveness of sensible energy exchange (Equation C-1) as a template for air-to-air heat/energy exchangers that have no leakage or condensation, we can define the water vapor mass exchange effectiveness \( (\varepsilon_m) \) as the ratio:

\[
\varepsilon_m = \frac{\text{Water Vapor Mass Exchange Rate}}{\text{Maximum Mass Exchange Rate}}
\]  \hspace{1cm} (C-7)

\[
\varepsilon_m = \frac{\Delta m_{wv} (\text{in either the supply or exhaust airstream})}{\Delta m_{wv} (\text{from one airstream inlet to the other})}
\]  \hspace{1cm} (C-8)

which, for steady mass flow rates of dry air in the supply and exhaust airstreams \( \dot{m}_s \) and \( \dot{m}_e \), becomes the following:

\[
\varepsilon_m = \frac{\dot{m} \Delta W (\text{in either airstream})}{\dot{m}_{m_{min}} \Delta W (\text{from one airstream inlet to the other})}
\]  \hspace{1cm} (C-9)

where \( \dot{m}_{m_{min}} \) is the minimum of \( \dot{m}_s \) or \( \dot{m}_e \).

Note these differ from Equation 3 in the normative section of the standard in that Equations C-7, C-8, and C-9 refer to transfer of water vapor mass, whereas Equation 3 refers to the transfer of latent energy associated with that mass.

Either airstream can be used in the numerator because \( (\dot{m} \Delta W)_s \) will equal \( (\dot{m} \Delta W)_e \) when there is no condensation or continuous accumulation of water in the heat exchanger. During condensation and frosting conditions, the calculated supply and exhaust side effectiveness will be unequal.

Equation C-9, when multiplied by \( h_{fg} \) in the numerator and denominator, becomes an energy flow ratio as follows:

\[
\varepsilon_m = \frac{\dot{m} h_{fg} \Delta W (\text{in either airstream})}{\dot{m}_{m_{min}} h_{fg} \Delta W (\text{from one airstream inlet to the other})}
\]  \hspace{1cm} (C-10)

Here we have assumed that the latent heat of evaporation \( (h_{fg}) \) is constant or that temperature change effects are very small in the expression for latent energy change as can be seen from the properties of saturated water (see 2017 ASHRAE Handbook—Fundamentals 4.4). For \( t > 0^\circ C \), we can write the following:

\[
h_{fg} = (2501 - 2.387t) \quad \text{kJ/kg} \quad \text{when} \quad t = [^\circ C] \quad \text{and} \quad t > 0^\circ C
\]  \hspace{1cm} (C-11)
For $t < 0^\circ$C, the latent heat of sublimation is $h_{ig}$, and for this case we can write the following:

$$h_{ig} = 2834 - 0.226t \text{ [kJ/kg]} \quad \text{when} \quad t = [^\circ\text{C}] \quad \text{and} \quad t > 0^\circ\text{C}$$

(C-12)

Both Equations A-11 and A-12 show a very small sensitivity to temperature over the typical ranges of inlet temperatures found in air-to-air heat/energy exchangers. The difference between $h_{ig}$ and $h_{fg}$ is the latent heat of fusion ($h_{lf}$), which is also slightly dependent on temperature.

Any concerns $h_{ig}$ and $h_{fg}$ are not important because we have chosen $h_{fg}$ only in Equation C-10. Furthermore, the assumption of a constant value for $h_{fg}$ in this equation will introduce an error of much less than 1% relative to the total energy change in the calculation of latent heat rates for most typical operating conditions for air-to-air heat/energy exchangers for temperatures above $0^\circ$C. For temperatures below $0^\circ$C, $h_{ig}$ should be used, though as noted in this standard, testing under conditions in which frosting occurs is not recommended due to the great difficulty of measuring the mass and energy flows associated with frost buildup. To avoid confusion, we should not interpret Equation C-10 as the ratio of latent heats of sublimation when one or more of the airstream temperatures is below $0^\circ$C.

We define this latent heat ratio in Equation C-10 to be the latent heat of evaporative energy transfer effectiveness or latent energy effectiveness, $\varepsilon_{l}$; in brief, $\varepsilon_{l} = \varepsilon_{m}$.

Using the definition provided in Equation C-1 as a model, we can define a sensible and total energy effectiveness ($\varepsilon_{s}$ and $\varepsilon_{t}$) when there is a steady-state flow of air without leakage or heat exchange between the surroundings and the exchanger. The sensible energy effectiveness is defined as follows:

$$\varepsilon_{s} = \frac{\text{Sensible Energy Exchange Rate}}{\text{Maximum Sensible Energy Exchange Rate}}$$

(C-13)

$$\varepsilon_{s} = \frac{C_{hcr} \Delta T (\text{for one of the airstream flows})}{C_{hcr, min} \Delta T (\text{from one airstream inlet to the other})}$$

(C-14)

where $C_{hcr, min}$ is the minimum of the heat capacity rate $C_{hcr, s}$ and $C_{hcr, e}$ for the supply and exhaust airflows. When there is no condensation or frosting and no exchange of water vapor between airstreams, either airstream (supply or exhaust) can be used in the numerator, and the ratio becomes:

$$\varepsilon_{s} = \frac{C_{hcr} \Delta T (\text{for either airstream})}{C_{hcr, min} \Delta T (\text{from one airstream inlet to the other})}$$

(C-15)

The difference in the change of sensible energy between the two airstreams is not insignificant with respect to the maximum sensible energy change rate when the change in the latent energy differs between the two airstreams (supply and exhaust)—that is, when the ratio

$$\frac{\left\{ m \Delta[\mathcal{W}(2501 + 1.805t)] \right\}_{\text{supply}} - \left\{ m \Delta[\mathcal{W}(2501 + 1.805t)] \right\}_{\text{exhaust}}}{C_{hcr, min} \Delta T (\text{from one airstream inlet to the other})}$$

(C-16)

is not insignificant with respect to the sensible effectiveness ($\varepsilon_{s}$) in Equation C-15, and the sensible effectiveness calculated using the supply side will not equal the sensible effectiveness calculated using the exhaust side. This implies that sensible energy transfer in an air-to-air exchanger will only be conserved when the other forms of energy change in the airstreams are negligible (i.e., there are no significant changes in airstream humidity ratio, and the heat exchange between the exchanger and the surroundings is negligible).

It should be noted that this definition of $\varepsilon_{s}$ may result in a large error when a large energy change occurs as a result of condensation or frosting. The total energy effectiveness is preferred when condensation occurs.

The total energy effectiveness is defined as follows:

$$\varepsilon_{tot} = \frac{\text{Total (Enthalpy) Energy Exchange Rate}}{\text{Maximum Total (Enthalpy) Energy Exchange Rate}}$$

(C-17)
\[ \varepsilon_{\text{tot}} = \frac{(\dot{m}_1 \Delta h)(\text{for either airstream})}{\dot{m}_\text{min} \Delta h (\text{from one airstream to the other})} \quad (C-18) \]

It is noted that this definition of total effectiveness is not usually restricted like the sensible effectiveness because, if condensation occurs, the errors incurred will be small because the energy flow rate in the condensing stream of water will be much smaller than the energy flow rate in any other airstream. Frosting, however, requires special attention because it can substantially alter the airflow passages in a heat exchanger and reduce the heat rate. Frost must be periodically removed in a defrost cycle.

In summary, air-to-air heat/energy exchangers differ significantly from heat exchangers in which only sensible energy changes occur and there is no mass transfer between supply and exhaust airstreams. We have introduced several parameters to characterize the performance of air-to-air heat/energy exchangers where the leakage of air and the interaction with the surroundings are very small or negligible except for the supply and exhaust airstreams. These dimensionless performance factors or ratios are water vapor mass effectiveness \( \varepsilon_m \), latent energy effectiveness \( \varepsilon_l \), sensible energy effectiveness \( \varepsilon_s \), and total energy effectiveness \( \varepsilon_c \). Furthermore, we find that with only negligible errors \( \varepsilon_l = \varepsilon_m \). Each effectiveness is a convenient factor to use for heat/energy exchanger selection or design because, in the absence of significant temperature and phase changes in the air due to interactions with the exchanger surroundings, the designer need not know anything other than the inlet properties for the supply and exhaust airflows.

**C2. RESEARCH FINDINGS**

Research has shown several significant characteristics of these somewhat unconventional definitions for effectiveness. First, the three effectiveness values—\( \varepsilon_l \) (or \( \varepsilon_m \)), \( \varepsilon_s \), and \( \varepsilon_c \)—are not all independent. That is, we can find a relationship among \( \varepsilon_l, \varepsilon_s \), and \( \varepsilon_c \) such that

\[ F(\varepsilon_l, \varepsilon_s, \varepsilon_c) = 0 \quad (C-19) \]

and if only two of these are known, the third unknown effectiveness can be derived. For air-to-air heat exchangers that transfer only sensible energy and do not experience condensation or frosting, \( \varepsilon_l = 0 \), and only one independent effectiveness is needed: either \( \varepsilon_s \) or \( \varepsilon_c \). The relationships between \( \varepsilon_l, \varepsilon_s \), and \( \varepsilon_c \) will depend on the type of heat/energy exchanger and are not presented here but are available in the literature.

Research has also shown that, unlike heat exchangers that transfer only sensible energy, heat exchangers that transfer both heat and moisture do not have constant values for \( \varepsilon_l, \varepsilon_s \), or \( \varepsilon_c \) when the inlet properties for the supply and exhaust airstreams vary. That is, the designer cannot assume completely constant effectiveness values for the design and selection of those energy exchangers for applications in buildings with variable outdoor air properties. Another unique characteristic of these heat exchangers is that \( \varepsilon_l, \varepsilon_s \), and \( \varepsilon_c \) are not confined to the range of 0 to 1.0 as is the case for heat exchangers that transfer only sensible energy. That is, negative effectiveness and effectiveness greater than 100% are theoretically possible. Furthermore, these unexpected effectiveness values only occur when the energy transfer rates for the particular mode of energy transfer (sensible, latent, or total) are very small or negligible.

For this standard, the most important research finding is how we must account for the accuracy of measuring data to calculate effectiveness or, conversely, the uncertainty or probable range of experimental errors in determining each effectiveness. Each measurement with an instrument contains both systematic errors due to fixed calibration errors or environmental effects and random errors due to random variations in the test data. When several measurements are combined in an equation for the calculation of effectiveness, the systematic and random errors must be handled in a well-defined manner. Furthermore, because the main objective of this standard is to determine the performance characteristics with the least experimental uncertainty, even the question of exactly how we calculate each effectiveness is very important. That is, we can significantly reduce the magnitude of effectiveness uncertainty when all the mass flow rates for dry air are equal, and whenever the energy changes in the supply and exhaust airstreams are expected to be equal, by using the average value of the energy change in the supply and exhaust airstream. By averaging the mass flow rates of dry air and the supply and exhaust air effectiveness, we use all the available data to calculate each effectiveness and
its uncertainty. When this is done, research shows that the value of the dry-air mass flow rates will vary by less than 0.8% but its uncertainty will be reduced by more than 1.0%, and the calculated effectiveness will typically change by much less than 1% but its uncertainty will be increased by about 2%. This reduced uncertainty will allow the designer who wants to specify air-to-air heat/energy exchangers to compute the energy savings from these devices with greater confidence. Smaller effectiveness uncertainties also means that the HVAC system design engineer will have a greater confidence in the technology and permit testing of air-to-air energy exchangers over a wider range of operating conditions.

Finally, research studies on air-to-air heat/energy exchangers have resulted in accurate theoretical/numerical models for the performance of these devices. These models, combined with a few device-characteristic properties, allows the calculation of each effectiveness over a wide range of operating conditions. Conversely, using one or more experimentally determined effectiveness, one can extrapolate each experimentally determined effectiveness to predict their value at other operating conditions. For simple devices with no condensation or transfer of water vapor, the model for sensible effectiveness ($\varepsilon_s$) is derived from the classical heat exchanger theory. Of course, there may be some modifications for heat-pipe exchangers, run-around loop systems, and thermal siphons that depend on the fluids used and their configuration. For heat exchangers that transfer only sensible energy, the classical literature shows how they behave, but for energy exchangers that transfer water vapor as well as heat, models and correlations have recently been developed that will allow the accurate extrapolation of one or more sets of effectiveness data to a wide range of operating conditions.

In summary, research shows that, in general, only two independent effectiveness values need to be measured for air-to-air heat/energy exchangers, but these effectiveness parameters will vary with changes in the supply and exhaust air inlet properties. This range of variation can theoretically exceed 100%, but we are only interested in the measured performance when the measurement uncertainty is small, which always implies test conditions that will result in effectiveness values between 0 and 100%. In order to get the most accurate test results for each effectiveness, we must minimize the uncertainty by using all the test data for the exhaust as well as the supply side when the energy changes are expected to be equal for each airstream; this means averaging the effectiveness values for the supply and exhaust airstreams. Finally, it is impractical to try to test air-to-air heat/energy exchangers for every operating condition that may occur during applications. The manufacturer and HVAC designer can use theoretical/numerical models or correlations to extrapolate a limited set of test data for effectiveness to the wide range of operating conditions that may occur in practical application. The uncertainty bounds for such extrapolations are likely to be larger than those obtained when using this standard to obtain effectiveness data.

C3. EXHAUST AIR TRANSFER RATIO AND OUTDOOR AIR CORRECTION FACTOR

To fully characterize air-to-air exchangers, Standard 84 calls for tests in addition to the effectiveness tests. Exhaust air transfer ratio (EATR) and outdoor air correction factor (OACF) are to be determined using very different test conditions—i.e., at room temperature for each air inlet and unequal mass flow rates for the supply and exhaust air inlets and outlets. The uncertainties for these test data pertain to the uncertainties for each mass flow rate measurement and, for EATR, the inert tracer gas measurements that are exchanged as a bulk leakage flow or carryover. EATR is a measure of bulk leakage of air within a device (i.e., due to air pressure differences and carry over). It is not directly applicable for gaseous contaminate transfer by mechanisms other than bulk leakage flow. OACF is likewise a measure of bulk air mass flow differences between the outdoor air inlet and the outdoor air outlet. These performance factors (EATR and OACF) must be established by test for any heat/energy exchanger.

Where specific contaminants are of concern for cross contamination, the methodology presented in Equation 6 may be used to assess the transfer of the specific contaminant. In such an application, $C_1$, $C_2$, and $C_3$ are the concentrations of the specific contaminant at Stations 1, 2, and 3, respectively. Due to the complex nature of the mass transfer phenomenon that occurs in mass and energy exchangers, it is very difficult to adequately quantify contaminant transfer. Sorption, diffusion, condensation of vapor on surfaces, and mechanical transfer are among the possible mechanisms of contaminant transfer. The user of energy exchangers should be well aware of the potential areas and mechanisms for contaminant transfer prior to their testing.
C4. RECOVERY EFFICIENCY RATIO

The recovery efficiency ratio (RER) for an air-to-air energy recovery device is a ratio of the rate of thermal energy recovered divided by the total electrical power input to operate the device. Because this ratio includes the fan power, pump power, and mechanical rotary power for electrical inputs, which may be external to the device, a standardized calculation procedure must be specified. Note that a higher RER does not indicate more energy is being recovered but rather that more energy is being recovered for every unit of energy input. Because RER results depend on the operating conditions, they need to accompany the RER data.
INFORMATIVE APPENDIX D
SELECTION OF TEST CONDITIONS

D1. TESTING CONDITIONS

Good testing conditions may not always be achievable, especially when field testing. In the laboratory, the heat/energy exchanger should be tested under the manufacturer's complete design and range of steady-state operating conditions. However, in situations where the test conditions may not be steady state, the test must be controlled within the allowable limits set out by the pretest uncertainty analysis. Any variations in the operating conditions, whether they are design modifications or changes to the range of steady state conditions, must be documented.

Because it is difficult to measure condensate and/or frost mass accurately, and this will greatly increase the uncertainty of the mass and energy inequalities and the effectiveness determination, it is strongly preferred to test at conditions in which neither condensate or frosting occur.

D2. SELECTION OF OPERATING CONDITIONS

Two different methods can be used to select the operating conditions. First, given the instrumentation uncertainty characteristics, contours of equal effectiveness uncertainty on a psychrometric chart are drawn for certain typical operating conditions and instrumentation. Second, an operating condition uncertainty is defined that can be employed to check the maximum uncertainty limits of the selected operating conditions.

D2.1 The Graphical Selection Method. The psychrometric chart in Figure D-1 allows a pretest estimation of the uncertainty levels associated with any combination of supply condition with an exhaust condition of 24°C and 50% rh. Figure D-2 presents similar results to those in Figure D-1 except that the assumed systematic and random uncertainties for flow rate, temperature, and humidity are increased to represent conditions that are likely to occur during field testing. During laboratory testing, Figure D-1 can be used to select appropriate operating conditions, while Figure D-2 can be used to select appropriate weather conditions for field testing. Because the test conditions play such an important role in the final calculated test results for energy exchangers, it is important that they are clearly stated when test results are presented.

D2.2 The Calculation Method. It is convenient to define an operating condition uncertainty \( U^*[e_i(OC)] \) using only the denominator from the definition of the effectiveness:

\[
U^*[e_i(OC)] = \frac{U[m_{min} (X_1 - X_3)]}{m_{min} (X_1 - X_3)} \quad \text{(D-1)}
\]

This equation allows the experimenter to select operating conditions that will, for a given set of sensors, result in a small uncertainty for \( e_i \).
Figure D-1 Lines of constant uncertainty in measured effectiveness values for different supply inlet conditions.

The assumed systematic error and random error in mass flow rate, temperature, and humidity are ±4% and ±1%, ±0.2 K and ±0.1 K, and ±2% rh and 1% rh, respectively. In (a), \( \varepsilon_y = \varepsilon_f = \varepsilon_T = 60\% \), and in (b), \( \varepsilon_y = \varepsilon_f = \varepsilon_T = 70\% \).
Figure D-2 Lines of constant uncertainty in measured effectiveness values for different supply inlet conditions. The assumed systematic error and random error in mass flow rate, temperature, and humidity are ±5% and ±3%, ±0.5 K and ±0.1 K, and ±3% rh and 1% rh, respectively. In (a), \( \epsilon_x = \epsilon_f = \epsilon_T = 60\% \), and in (b), \( \epsilon_x = \epsilon_f = \epsilon_T = 70\% \).
INFORMATIVE APPENDIX E
FIELD TESTING

Field testing of air-to-air heat/energy exchangers using this standard is more complex than laboratory testing because flow measurements and air properties must be measured in ducts that are often noncircular, while the properties may be nonuniform spatially across one or more ducts and vary with time. Also, the time-varying test conditions likely will include data for which the uncertainty is excessive (see Informative Appendix C) and perhaps should be excluded from the analysis. New procedures are needed to measure duct properties over an extended time period and calculate the uncertainty of these properties and the performance factors. A data acquisition system with some capability to process sensor data from pressure transducers, thermocouples, and humidity sensors is normally used. Special in-situ tests are normally required to measure duct flow rates and leakage from the exhaust to the supply air ducts. Johnson et al. G22 provides some discussion of these problems for field testing using Standard 84-1991 as a guide for two different heat/energy exchangers. Significant spatial variations in outlet air properties for energy exchangers will be seen when property measurements are close to the exchanger. G23,G24 Field testing for exhaust air transfer ratio (EATR), outdoor air correction factor (OACF), and recovery efficiency ratio (RER) has not been reported in the literature.

For field testing, it is likely that the upper limits on inequalities of Equations 20 to 27 will need to be increased by a factor of 2.0. Likewise, the uncertainty limits in inequalities in Equations 28 through 36 will likely need to be increased by a factor of 1.5.

For field testing, the reporting requirements shall be the same as specified in Sections 10.1 to 10.4 after the above modifications to the inequalities and uncertainty limits are factored in for all the operating conditions, balance checks, and uncertainty analysis.

E1. MASS FLOW MEASUREMENT

Over an extended period of time (i.e., two or more weeks), it is convenient to determine mass flow rates by measuring the air friction pressure drop for air flowing through an obstruction that results in a significant pressure drop (e.g., the heat/energy exchanger). This pressure drop is monitored using several static pressure probes that are manifolded to give a good average pressure upstream and downstream of the heat/energy exchanger in the supply and exhaust airstreams. (Duct wall mounted static pressure taps often cause large errors in field applications.) The mass flow rate (m) is calculated using the following equation:

\[ \dot{m} = K(\Delta P)^n \pm U_m \]  

(E-1)

where \( \Delta P \) is the monitored pressure difference across either the heat/energy exchanger supply or exhaust ducts at any time, \( n \) is an exponent that equals 1/2 for fully developed turbulent flow, and \( K \) is the constant of proportionality that must be determined by a velocity traverse in each of the air ducts into and out of the heat/energy exchanger. During this traverse, both the mass flow rate of air (\( m \)) and the static pressure difference (\( \Delta P \)) must be determined so that the constant \( K \) may be calculated. The mass flow rate of dry air is calculated in each duct using the following equation:

\[ \dot{m} = \bar{\rho} \bar{v} A \]  

(E-2)

where \( A \) is the duct area, \( \bar{v} \) is the average air velocity in the duct and is obtained by the velocity traverse, and \( \bar{\rho} \) is the average dry-air density in the duct and is approximated by the ideal gas relationship as follows:

\[ \bar{\rho} = \frac{\rho_{atm}}{\frac{R_a T}{U_{\rho a} + U_{\rho a}}} \]  

(E-3)

where \( \rho_{atm} \) is the current barometric pressure, \( \bar{T} \) is the average air temperature in the duct, and \( U_{\rho a} \) is the uncertainty of dry-air density as determined from a knowledge of the barometric
pressure, static pressure in the duct, and vapor pressure due to humidity in the duct. This may be thought of as a systematic uncertainty in \( \rho \) for the calculation of \( K \). [Note: \( T \) should be measured in conjunction with the velocity traverse so that it represents the bulk mean value for the flow, because it is the bulk mean property values that must be used for the effectiveness calculations. See Shang and Besant\(^{G23,G24} \) for data and discussion of this problem for energy wheels.]

\( \bar{v} \) is the average air velocity in the duct and is obtained by the velocity traverse, giving

\[
\bar{v} = \frac{1}{N_v} \sum_{i=1}^{N_v} v_i \pm U_v
\]  

(E-4)

where \( N_v \) is the number of spatial points used to measure the velocity \( (v_i) \) in the duct using a calibrated velocity probe (e.g., hot film or hot-wire anemometer, pitot tube, vane anemometer). \( N_v \) should be greater than 8 for most ducts, but the number selected depends on the required uncertainty. In all cases, the locations for velocity or other property measurements should divide the duct into equal flow areas.

\( U_v \) is the uncertainty of average velocity measurement in the duct, which may also be thought of as a systematic error \( (B_v) \)

\[
B_v = \left( \frac{tS_v}{\sqrt{N_v}} \right)^2 + \left( \frac{S_{\psi}}{\sqrt{N_{\psi}}} \right)^2 + \left( \frac{S_{\Delta p}}{\sqrt{N_{\Delta p}}} \right)^2 \right)^{1/2}
\]  

(E-5)

where \( r \) is the “student \( r \)” which may be taken to be two for more than 30 samples; \( S_p \) is the sample standard deviation for the velocity probe obtained during \( N_p \) calibration points; and \( S_{\psi} \) is the sample standard deviation for spatial variations of velocity \( v_i \) about the mean, for \( N_s \) points in the duct.

\[
S_v = \left[ \frac{\sum_{i=-1}^{N_s} (\bar{v} - v_i)^2}{N_s - 1} \right]^{1/2}
\]  

(E-6)

\( S_v \) is the sample standard deviation for the temporal variations of velocity at each point for \( N_f \) measurements in time at each point and where

\[
S_{\psi} = \left[ \frac{\sum_{i=1}^{N_s} (S_{\psi}_i)^2}{N_f - 1} \right]^{1/2}
\]  

(E-7)

\[
S_{\Delta p} = \left[ \frac{\sum_{i=1}^{N_f} (\bar{\Delta p} - \Delta p)^2}{N_f - 1} \right]^{1/2}
\]  

(E-8)

Finally, the systematic error for the constant \( K \) is

\[
B_K = \left( B_{\rho}^2 + B_v^2 + B_A^2 + B_{\Delta p}^2 \right)^{1/2}
\]  

(E-9)

and the uncertainty for the data set becomes

\[
U_m = (B_K^2 + B_{\Delta p}^2)^{1/2}
\]  

(E-10)

See ANSI/ASME PTC 19.1\(^1 \) for further discussion of these terms and equations.
E2. TEMPERATURE AND HUMIDITY DETERMINATIONS

In order to measure $T$, two or more independent temperature sensors are monitored continuously in each duct. Each sensor should be centered in an area of equal flow area. For example, the average temperature in each duct is given by the following:

$$\bar{T} = \frac{1}{N_s} \sum_{i=1}^{N_s} T_i \pm U_T$$  \hspace{1cm} (E-11)

where $N_s$ is the number of spatial points where temperature is monitored and

$$U_T = \left[ P_T^2 + B_T^2 \right]^{\frac{1}{2}}$$  \hspace{1cm} (E-12)

where the random error of the average temperature is as follows:

$$P_T = \frac{tS_T}{\sqrt{N_s}}$$  \hspace{1cm} (E-13)

The sample standard deviation, when temporal variations are negligible, is as follows:

$$S_T = \left[ \frac{1}{N_s} \sum_{i=1}^{N_s} \left( \bar{T} - T_i \right)^2 \right]^{\frac{1}{2}}$$  \hspace{1cm} (E-14)

where $B_T$ is the systematic error in $T$ obtained during calibration of the temperature sensors.

Similar methods should be used for $W$, but this may prove to be impractical because of the high cost of accurate relative humidity probes. The experimenter is advised to bring extra temperature and humidity probes to the site of the field test so they can be installed and used for monitoring in ducts that show nonuniformity in temperature or relative humidity; the equations above show that extra sensors will reduce the uncertainty for each average property calculation. Once these average duct properties and their uncertainties are known, the calculations for field and laboratory testing are identical.

E3. QUASI-STEADY FIELD TEST CRITERIA

Due to changes in ambient air temperature, humidity, and airflow rate controllers, diurnal variations in the operating conditions of heat/energy exchangers installed in buildings always occur. These variations are negligible when the rate of change in the energy flow in the operating conditions is small with respect to the rate that energy is stored within the stationary or solid components of the heat/energy exchanger.

For sensible energy heat transfer, this restriction for changes in operating conditions can be expressed as a ratio of time constants; for sensible effectiveness, for example,

$$\frac{\tau_{HX,s}}{\varepsilon_s \tau_{OC,s}} \ll 1$$  \hspace{1cm} (E-15)

where

$$\varepsilon_s = \text{sensible effectiveness}$$

$$\tau_{HX,s} = \frac{M_s C_{Ps}}{UA}$$

the time constant of the solid components of the heat/energy exchanger that can change sensible energy due to small changes in operating conditions for sensible energy, and
\[ \tau_{OC,s} = \frac{U \Delta T_{lm}}{d} \left( \frac{d}{dt} (m C_p \Delta T_{OC}) \right) \]

the time constant of the sensible energy flow in the operating conditions with respect to the rate of sensible energy exchange in the heat/energy exchanger.

For latent energy transfer, the ratio is as follows:

\[ \frac{\tau_{HX,s}}{\tau_{OC,l}} = 1 \]

(E-16)

where

\[ \varepsilon_l = \text{latent energy effectiveness} \]

\[ \tau_{OC,l} = \frac{U_m A \Delta W_{lm}}{\frac{d}{dt} (m \Delta W_{OC})} \]

the time constant of the latent energy flow with respect to the rate of latent energy exchange in the energy exchanger, and

\[ \tau_{HX,s-l} = \frac{M_s C_p \Delta T_{lm}}{U_m A \Delta W_{lm} h_{fg}} \]

the time constant of the sensible energy flow within the solid components of the energy exchanger due to small changes in the operating conditions for latent energy flow through the energy exchanger.

For most air-to-air heat exchangers that transfer only sensible energy, \( \varepsilon_s = 0.5 \) to 0.7 and \( \tau_{HX,s} = 1 \) to 10 minutes, and for rotary heat wheels that transfer latent energy, \( \varepsilon_l = 0.5 \) to 0.8 and \( \tau_{HX,s-l} = 0.1 \) to 1 minutes.

Thus Equations E-15 and E-16 are satisfied when \( \tau_{OC,s} \approx 1 \) to 20 minutes for sensible air-to-air heat exchangers and \( \tau_{OC,s-l} \approx 0.1 \) to 2 minutes for latent energy changes for rotary total energy wheels. For typical diurnal variations in ambient temperature or humidity, the left-hand side of Equations E-15 and E-16 are about 0.01, implying that we can assume quasi-steady conditions.

Transient changes in ambient outdoor air temperature and humidity will almost always satisfy these conditions, but changes in airflow rates due to step changes in fan or damper set points may not. This suggests that field data for changes in the HVAC set points in buildings should be excluded for performance testing, especially if large changes to the airflow rates are involved. Within less than one hour, or even minutes before and after such changes, Equations D-13 and D-14 will be satisfied and the monitored field data can be included.

E4. REJECTION OF TEST DATA

Many field test conditions may result in poor quality data. Readings may need to be discarded based on the following criteria:

a. Weather conditions with high wind, rain, snow, or extreme temperatures
b. Atmospheric conditions with high concentrations of dust, organics, or chemicals
c. Site interference from unspecified terrain, buildings, or equipment
d. Equipment or instrumentation failures, improper operation, incorrect adjustments, or poor calibrations
e. Poor test operating conditions resulting in excessive sound or vibration, low temperature differentials, poor flow distributions, or leaks in connecting ducting
f. Post-test uncertainty analysis indicating unacceptable uncertainty limits
g. Balance checks of Section 6.1 that are not satisfied
INFORMATIVE APPENDIX F
EXTRAPOLATION OF TEST PERFORMANCE DATA

Testing air-to-air heat/energy exchangers can be expensive, especially in a laboratory. Testing for a wide range of operating conditions can be time consuming, especially if operating temperatures, humidities, and flow rates are to be independently varied in the laboratory. For field testing, the test engineer has little or no control over the operating conditions. These considerations point to the need for the test engineer to exercise good judgment about the selected number and operating conditions under which tests should be carried out.

If validated theoretical/numerical models and/or theoretical/empirical correlations are available for each type of heat/energy exchanger tested, then it will be unnecessary to require a very extensive and comprehensive set of test conditions. Rather, using the validated models or correlations for heat exchanger effectiveness, a limited test data set can be extrapolated from one operating condition to another with only a small added uncertainty. The magnitude of uncertainty associated with these models or correlations should be clearly stated in the literature. The extrapolation of test data should always include a statement on the expected uncertainty of the particular model or correlation used.

For sensible heat exchange with no water vapor phase change, classical heat exchanger theory can be used for these extrapolations provided the property changes are fully accounted for as shown in the paper by Guo et al. G25. In most cases, the changes in sensible effectiveness will be small over a wide range of test conditions. This result implies that only a few test data may be necessary for these types of heat exchangers. Condensation and frosting within air-to-air heat/energy exchangers can cause large changes in their performance, so testing will likely be necessary when operating conditions are such that these phase changes are significant.

For enthalpy wheels and some regenerators, water vapor sorption and desorption can cause changes in the performance effectiveness even when condensation or frosting do not occur. These characteristics imply that either very comprehensive sets of test data should be collected for energy wheels or that validated models or correlations should be used to extrapolate a limited amount of test data. Simonson and Besant G19,G20 present a theoretical/numerical model that is validated with laboratory test data G26,G27. Effectiveness correlations are presented in Simonson and Besant G28 and Simonson et al. G29,G30.
INFORMATIVE APPENDIX G
INFORMATIVE REFERENCES AND BIBLIOGRAPHY

G1. INFORMATIVE REFERENCES


G18. Shang, W., and R.W. Besant. 2009. Effectiveness of desiccant coated regenerative wheels from transient response characteristics and flow channel properties—Part II:
Predicting and comparing the latent effectiveness of dehumidifier and energy wheels using transient data and properties. HVAC&R Research 15(2):329–65.


G2. BIBLIOGRAPHY


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FOR THE ENVIRONMENTAL IMPACT OF ITS ACTIVITIES

ASHRAE is concerned with the impact of its members’ activities on both the indoor and outdoor environment. ASHRAE’s members will strive to minimize any possible deleterious effect on the indoor and outdoor environment of the systems and components in their responsibility while maximizing the beneficial effects these systems provide, consistent with accepted Standards and the practical state of the art.

ASHRAE’s short-range goal is to ensure that the systems and components within its scope do not impact the indoor and outdoor environment to a greater extent than specified by the Standards and Guidelines as established by itself and other responsible bodies.

As an ongoing goal, ASHRAE will, through its Standards Committee and extensive Technical Committee structure, continue to generate up-to-date Standards and Guidelines where appropriate and adopt, recommend, and promote those new and revised Standards developed by other responsible organizations.

Through its Handbook, appropriate chapters will contain up-to-date Standards and design considerations as the material is systematically revised.

ASHRAE will take the lead with respect to dissemination of environmental information of its primary interest and will seek out and disseminate information from other responsible organizations that is pertinent, as guides to updating Standards and Guidelines.

The effects of the design and selection of equipment and systems will be considered within the scope of the system’s intended use and expected misuse. The disposal of hazardous materials, if any, will also be considered.

ASHRAE’s primary concern for environmental impact will be at the site where equipment within ASHRAE’s scope operates. However, energy source selection and the possible environmental impact due to the energy source and energy transportation will be considered where possible. Recommendations concerning energy source selection should be made by its members.
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