BSR/ASHRAE Standard 215-2018 (RA 202X)

Public Review Draft

Method of Test to Determine Leakage of Operating HVAC Air Distribution Systems

First Public Review (May 202X)

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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.

(This foreword is not 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. Unresolved objectors on informative material are not offered the right to appeal at ASHRAE or ANSI.)

FOREWORD

As discussed in the 2016 and 2017 ASHRAE Handbook chapters "Duct Construction" (Chapter 19) and "Duct Design," (Chapter 21), heating, ventilating, and air-conditioning (HVAC) system air leakage significantly increases building energy consumption. For example, a leaky VAV system (e.g., 10% leakage upstream and 10% downstream of terminal box inlet dampers at operating conditions) can use 25% to 35% more fan energy than a tight system (e.g., 2.5% upstream and 2.5% downstream at operating conditions). For an exhaust system with 20% leakage, the fan has to move 25% more air to meet the specified flows at the grilles, which causes fan power to increase up to 95%. Leakage also reduces the system's ability to control and deliver intended flows and pressures and to manage the spread of contaminants. As such, there is a need to minimize leakage airflows during air-handling system operation.

Leakage test procedures currently used by industry focus on determining component airtightness (e.g., for ductwork located upstream of terminal box inlet dampers). Airtightness alone, however, is insufficient to determine leakage airflows. One must also then estimate system pressures during system operation to determine these flows. Determining the location of every leak and the pressure difference across each leak is practically impossible for most systems and can cause significant uncertainty in leakage airflow estimates using this approach.

To eliminate the uncertainty associated with estimating pressure differences across leaks, this standard provides a method of test for determining leakage airflows, either for the whole system or for selected parts. Flows into and out of the section being tested are measured at a repeatable reference operating condition: the difference is the leakage flow. The operating condition is not necessarily the design operating condition but corresponds to the greatest system inlet flow (outlet flow for exhaust systems) possible without being detrimental to the occupants of the building, the building structure, or the HVAC mechanical components, while maintaining the duct static pressure set point (where applicable) specified in design documents.

This standard does not mandate a calibration method for any instrument, nor does it dictate that the user employ a specific flow measurement technique. Informative Appendix A provides a general discussion of airflow measuring instrument technologies and their capabilities. Informative Appendix B provides recommended airflow measuring instrument calibration and verification procedures.

A simplified methodology for estimating leakage airflows downstream of terminal-box inlet dampers (Informative Appendix G) is also provided in this standard. This methodology can be used to distinguish leakage downstream of terminal-box inlet dampers from total leakage determined by this standard so that the user can determine where to focus potential sealing activities.

This is a reaffirmation of Standard 215-2018. This standard was prepared under the auspices of 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 interest of obtaining uniform guidelines throughout the industry. This version of the reaffirmation includes a clarification of the citation of ASHRAE Handbook—Fundamentals *in Section A1.1 and an update to Appendix H, "Informative References and Bibliography."*

1. PURPOSE

This standard specifies a method of test to determine leakage airflow and fractional leakage of operating HVAC air distribution systems and determines the uncertainty of the test results.

2. SCOPE

2.1 This standard is for field application in both new and existing buildings.

2.2 This standard can be applied to determine whole-system or sectional leakage airflow.

2.3 This standard provides

a. test procedures and requirements for measuring inlet and outlet airflows during system operation,
b. methods for calculating leakage airflows to/from system surroundings,

c. methods for calculating leakage test uncertainties,
d. methods for documenting the test plan, and
e. methods for reporting test results.

2.4 The test procedures in this standard are limited to single-duct supply and independent exhaust air systems.

2.5 This standard is not for determining return air leakage.

2.6 This standard is not for determining leakage involving ceiling and floor plenums, systems serving pressure-controlled spaces, or air dispersion systems.

2.7 This standard does not replace ductwork pressurization leakage testing.

2.8 This standard does not specify leakage acceptance criteria.

2.9 This standard shall not be used to override any safety, health, or critical process requirements.

3. DEFINITIONS

accuracy: the degree of conformity of an indicated value to an accepted standard value or true value. The degree of inaccuracy is known as "total measurement error" and is the sum of bias and precision errors.

air dispersion systems: any diffuser system designed to both convey air within a room, space, or area and diffuse air into that space while operating under positive pressure. These systems are commonly, but not exclusively, constructed of fabric, sheet metal, or plastic film.

air terminal: device other than an air valve that modulates the volume of air delivered to or removed from a conditioned space in response to an external demand.

bias error: the difference or offset between the true or actual value to be measured and the mean indicated value from the measuring system that persists and is usually due to the particular instrument or technique of measurement. This error is determined and minimized through calibration.

confidence level: the probability that a stated interval will include the true value. In analyzing measured data, a confidence level of 95% (approximately two standard deviations) is often used. The level used in this standard for error analyses is one standard deviation (approximately 68%).

error: the difference between the true value of the quantity measured and an observed value. All experimental errors can be classified as one of two types: bias error or precision error.

exhaust system, independent: air discharged from a space to the outdoors by a system not coupled to supply or return air systems.

fractional leakage: leakage airflow for the system or a section, whichever is applicable, divided by the associated inlet airflow at the reference operating condition.

independent exhaust system: see *exhaust system, independent*.

leakage airflow: the aggregate sum of all airflows through leaks either to or from an air-handling system or a section, whichever is applicable and regardless of whether the leaks are unintentional or not, due to pressure differences applied across the leaks during system operation at the reference operating condition. Calculated leakage airflows are referenced to standard air density.

mechanical room plenum: plenum between the air-handling unit (AHU) and the return or relief fan that make up the economizer cycle.

outlet: for a single-duct supply system, a grille or diffuser. For an independent exhaust system, the system discharge.

precision error: a statistical error that is caused by chance or by stochastic temporal or spatial variations of factors affecting the measurement process and that is not necessarily recurring. It causes readings to take varying values on either side of the mean value. This error can often be reduced by increasing the number of observations.

pressure-controlled space: an enclosed space within a building, such as a laboratory space or hospital isolation space, that has automatically controlled pressure relationships. Whole-building pressure control is not defined as a pressure-controlled space within this standard.

reference flow: airflow at which an HVAC system operates during a leakage test.

reference operating condition: configuration, as applicable, of system fans and all other system components and commands necessary to achieve the desired operating condition. It corresponds to the greatest system inlet flow (outlet flow for exhaust systems) possible without being detrimental to the occupants of the building, the building structure, or the HVAC mechanical components, while maintaining the duct static pressure set point (where applicable) specified in design documents.

reference static pressure: static pressure at which a system operates during a leakage test.

relative humidity: the ratio of the vapor pressure of a moist air sample to the saturation vapor pressure at the same mixture temperature and pressure.

single-duct supply air distribution system: system in which the air, having been conditioned, is distributed to various zones through a single-duct system, in some cases through an air terminal, but not through air valves.

surroundings: where applicable, system surroundings include outdoors and conditioned or unconditioned spaces within the building.

system: in this standard, (a) portions of HVAC systems or (b) independent exhaust systems.

The portions of the HVAC system covered by this standard are

a. the supply air distribution ductwork between the air-handling unit (AHU) discharge and the conditioned space, including variable-air-volume (VAV) terminal units, coils, dampers, and sound attenuators, or
b. a portion of that supply air distribution ductwork.

Exhaust air systems, independent of the HVAC system, are also covered by this standard. The AHU, return ductwork, return or exhaust/relief fan, and outdoor air and exhaust air plenums are not covered by this standard.

test plan: a document or other written form of communication that specifies the requirements for conducting the test procedure and reporting the test results.

test section: see *system*.

uncertainty: a measure of the dispersion of potential error relative to the true value. Uncertainty reflects doubt in a measurement to a specified confidence level. Although uncertainty is the result of both bias and precision errors, only precision errors are treated by statistical methods. Bias errors correspond to a mean error and can be minimized through calibration. In this standard, instrument readings are corrected by subtracting mean bias errors, and uncertainty is analyzed only as a precision error.

4. INSTRUMENTATION

4.1 Calibration Requirements. Measurements from the instruments shall be traceable to primary or secondary standards calibrated by the National Institute of Standards and Technology (NIST) or to the Bureau International des Poids et Mesures (BIPM) if a national metrology institute (NMI) other than NIST is used. Instruments shall be recalibrated on a regular schedule that is appropriate for each instrument but at least annually. Calibration records shall be maintained.

4.2 Precision Error. The required precision with a 95% confidence level for the instruments used shall be as follows.

4.2.1 Airflow instruments shall have a precision $P_{Q_{x,\,std}}$ equal to or better than 3% of measured airflow adjusted to standard air density or 1 L/s (2 cfm), whichever is greater, unless specified otherwise in the test plan. Precision shall be stated at standard air density: 1.204 kg/m^3 (0.075 lb$_m$/ft^3).

4.2.2 Barometric pressure instruments shall have a precision P_{pb} equal to or better than 0.5% of measured pressure or 500 Pa (0.073 psi, 0.148 in. of mercury), whichever is greater.

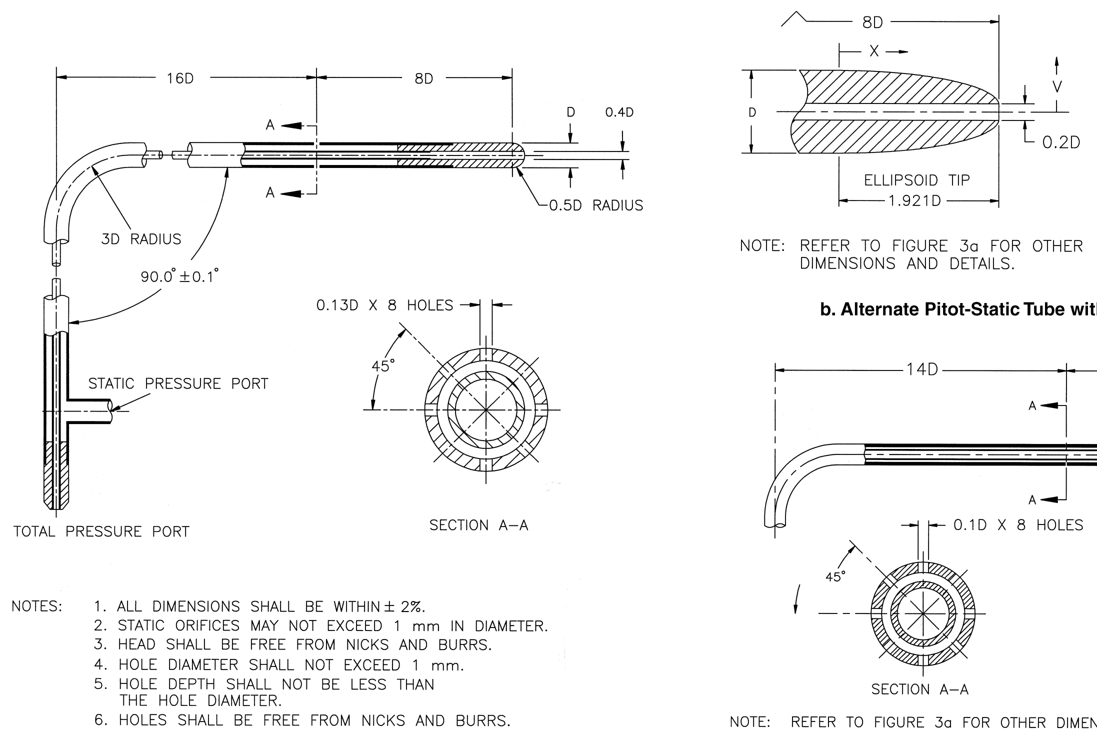
4.2.3 Duct static pressure instruments shall have a precision P_{ps} equal to or better than 2% of measured pressure or 1 Pa (0.004 in. of water), whichever is greater.

4.2.4 Temperature instruments shall have a precision P_t equal to or better than 1°C (2°F).

4.2.5 Relative humidity instruments shall have a precision P_φ equal to or better than 7%.

4.3 Bias Error. Mean bias errors for airflows shall be stated at standard air density as a *fractional* value $B_{Q_{x,\,std}\text{–}frac}$ of the measured airflow adjusted to standard air density or as an *absolute* value $B_{Q_{x,\,std}\text{–}abs}$, whichever is applicable.

4.4 Accuracy. When instrument accuracy is reported without separating out the precision and bias error components, it shall be assumed that “accuracy” means a precision error at a 95% confidence level and the bias error is zero.



X/D	V/D	X/D	V/D
0.000	0.500	1.602	0.314
0.237	0.496	1.657	0.295
0.336	0.494	1.698	0.279
0.474	0.487	1.730	0.266
0.622	0.477	1.762	0.250
0.741	0.468	1.796	0.231
0.936	0.449	1.830	0.211
1.025	0.436	1.858	0.192
1.134	0.420	1.875	0.176
1.228	0.404	1.888	0.163
1.313	0.388	1.900	0.147
1.390	0.371	1.910	0.131
1.442	0.357	1.918	0.118
1.506	0.343	1.920	0.109
1.538	0.333	1.921	0.100
1.570	0.323		

a. Static and Pitot-Static Tube with Spherical Head

b. Alternate Pitot-Static Tube with Ellipsoidal Head

c. Alternate Pitot-Static Tube with Blunt Head

Figure 1 Pitot-static tube.

4.5 Airflow Measuring Instruments

4.5.1 Test section airflows *not at grilles and diffusers* shall be measured using an airflow measuring station or another instrument capable of measuring the airflows with the precision stated in Section 4.

Test section airflows *at grilles and diffusers* shall be measured using a flow capture hood or other instrument capable of measuring the airflows with the precision stated in Section 4.

4.6 Pressure Measuring Instruments. Static pressure at a point in a duct shall be sensed with the static tap part of a pitot-static tube conforming to Figure 1.

5. TEST SETUP

5.1 General Requirements

5.1.1 The test setup operating parameters shall be described in sufficient detail in the test plan (Section 5.5) and the test report (Section 7) to enable achieving a repeatable reference operating condition.

5.1.2 The reference operating condition shall correspond to the greatest system inlet flow (outlet flow for exhaust systems) possible without being detrimental to the occupants of the building, the building structure, or the HVAC mechanical components, while maintaining the duct static pressure set point (where applicable) specified in design documents.

5.1.3 The description of the test setup shall include the following at a minimum:

a. Fan speeds, fan blade or vane positions, and fan variable frequency drive (VFD) output frequency, where applicable
b. All damper positions and commands (supply air, return air, outdoor air, relief air)
c. Static pressure reference locations and set points, where applicable
d. All other system and terminal unit control damper positions and commands necessary to achieve the reference operating condition
e. Diagram of test measurement locations

5.1.4 After the test is complete, the system configuration shall be returned to the configuration existing before the system set up.

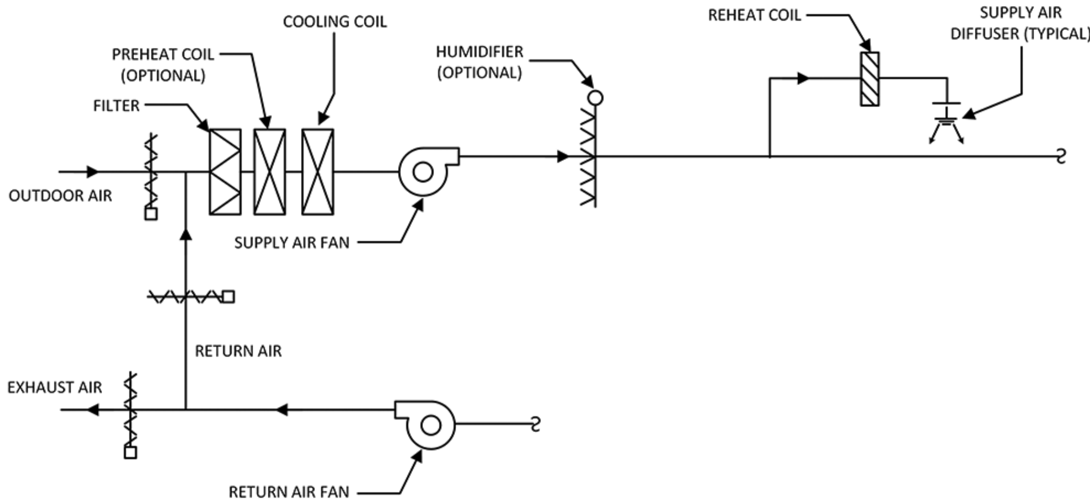


Figure 2 Typical constant-air-volume (CAV) system.

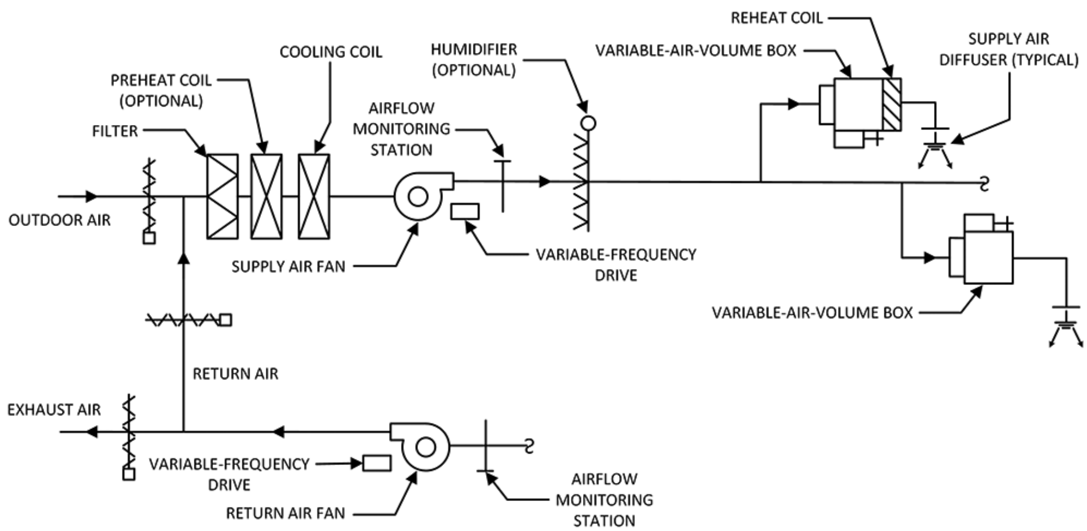


Figure 3 Typical variable-air-volume (VAV) system.

5.2 Single-Duct Constant-Air-Volume (CAV) Systems

5.2.1 The supply air fan shall operate at the reference operating condition. For a schematic of a typical CAV system, refer to Figure 2.

5.2.2 Other fans associated with the system (return or relief air fans) shall be operated to achieve the reference operating condition.

5.2.3 The dampers associated with the system shall be positioned as follows:

a. Return air damper: fully *open*
b. Exhaust air or relief air damper: fully *closed*
c. Outdoor air damper: fully *closed*

Exception to 5.2.3: The outdoor air damper shall remain fully *open* for 100% outdoor air systems or for systems where closing the outdoor air damper will damage the system by restricting airflow.

5.2.4 To prevent changes in the reference operating condition during the test, controls for the system heating, cooling, humidification, economizer cycle, temperature resets, unoccupied and occupied modes (including occupant overrides such as occupancy sensors and push-button temporary overrides and secondary temperature control in the system), and all other control modes that will cause unstable conditions during the test shall be overridden.

5.2.5 The system supply air temperature at the inlet measuring plane shall be maintained within ±1.7°C (3°F) of a fixed temperature set point.

5.3 Single-Duct Variable-Air-Volume (VAV) Systems

5.3.1 The supply air fan shall operate at the reference operating condition. For a schematic of a typical VAV system, refer to Figure 3.

5.3.2 The duct static pressure reference condition shall be measured at the system's installed duct static pressure sensor locations. The reported reference pressure shall be the pressure normally employed to control the supply air fan (e.g., the average of the pressures measured at the different sensor locations).

Exception to 5.3.2: Duct static pressure measurements shall be made at the inlet measuring plane to the test section if the test section does not contain the system's installed duct static pressure sensors.

5.3.3 Other fans associated with the system (return or relief air fans) shall be operated to achieve the reference operating condition.

5.3.4 The dampers associated with the air-handling unit (AHU) shall be positioned as follows:

a. Return air damper: fully *open*
b. Exhaust air or relief air damper: fully *closed*
c. Outdoor air damper: fully *closed*

Exception to 5.3.4: The outdoor air damper shall remain fully *open* for 100% outdoor air systems or for systems where closing the outdoor air damper will damage the system by restricting airflow.

5.3.5 For a system with direct digital control (DDC) of the VAV terminal boxes, the system shall be configured to operate at the reference operating condition using the following procedure:

a. The VAV terminal unit controls shall be overridden so that each terminal unit is set to its design maximum airflow.
b. With the system supply fan operating at the greatest speed possible without being detrimental to the occupants of the building, the building structure, or the HVAC mechanical components, if the duct static pressure at its control location is less than the maximum duct static pressure design set point, each terminal unit's airflow shall then be uniformly reduced by the same percentage of the terminal unit's design maximum airflow until the maximum duct static pressure design set point is achieved.
c. If, instead, the duct static pressure at its control location is greater than the maximum duct static pressure design set point, the fan speed shall be reduced until the maximum duct static pressure design set point is achieved.

The final fan speed, duct static pressure, and terminal unit settings that result are the reference operating condition.

5.3.6 For a system with pneumatically controlled terminal boxes, the system shall be configured to operate at the reference operating condition using the following procedure:

a. Each VAV terminal unit shall be set to its design maximum airflow by setting the thermostat to its minimum setting (e.g., 17°C [55°F]).
b. With the system supply fan operating at the greatest speed possible without being detrimental to the occupants of the building, the building structure, or the HVAC mechanical components, if the duct static pressure at its control location is less than the maximum duct static pressure design set point, starting at the zone located closest to the primary fan, override every other terminal box to its minimum airflow set point by setting the thermostat to its maximum setting (e.g., 35°C [95°F]) until the maximum duct static pressure design set point is achieved or just exceeded.
c. If, instead, the duct static pressure at its control location is greater than the maximum duct static pressure design set point, then the fan speed shall be reduced until the maximum duct static pressure design set point is achieved.

The final fan speed, duct static pressure, and thermostat settings that result are the reference operating condition.

5.3.7 To prevent changes in the reference operating condition, controls for the system heating, cooling, humidification, economizer cycle, static pressure resets, unoccupied and occupied modes (includ-

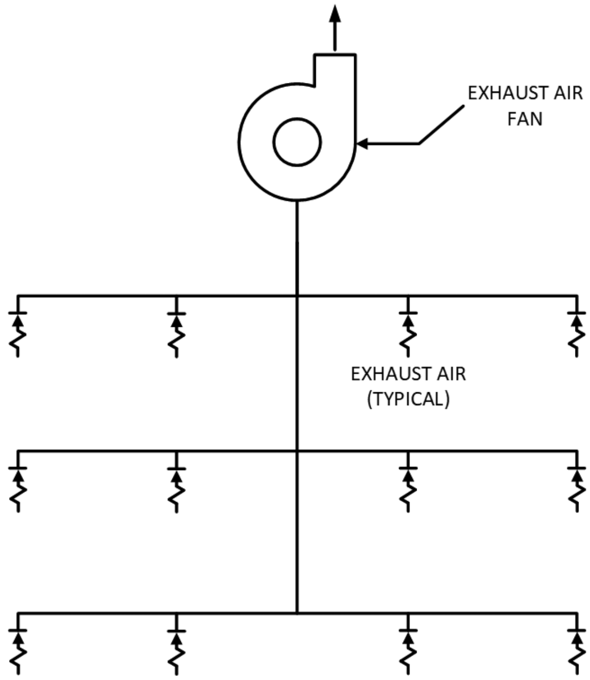


Figure 4 Typical independent exhaust system.

ing occupant overrides such as occupancy motion sensors and push-button temporary overrides), reheat coils, energy and heat recovery systems, and all other control modes that will cause unstable conditions during the test shall also be overridden.

5.3.8 The system supply air temperature at the inlet measuring plane shall be maintained within ±1.7°C (3°F) of a fixed temperature set point.

5.4 Independent Exhaust Systems

5.4.1 The exhaust air fans shall operate at a fixed speed at the reference operating condition. When the system has multiple fans serving the same exhaust system, all fans shall operate at a fixed speed. For a schematic of a typical independent exhaust system, refer to Figure 4.

5.4.2 All exhaust air dampers shall be set to fixed positions. The dampers shall be set to proportionally distribute airflow throughout the system (or section being tested) relative to the design flows so that no section of the system being tested is isolated from the outlet measurement plane.

5.4.3 The static pressure reference condition shall be measured at the static pressure sensors or with a static pressure probe located within 6 m (20 ft) of the exhaust fan if no sensor is present.

5.5 Test Plan

5.5.1 The test plan is a required document that references this standard and describes deviations from this standard along with associated rationale. The test plan shall include, but not be limited to, the following:

a. A text description or schematic diagram of the air distribution system features sufficient to specify the system sections to be tested and the system components involved in the leakage test.
b. A text description or schematic diagram sufficient to specify the instrumentation and any additional data required to be collected, including the following:
    1. Airflow instrumentation precision that deviates from the specifications in Section 4.
    2. Measurement procedures that deviate from published instructions provided by the instrumentation manufacturers. References to those instructions shall be provided.
    3. Flow measurement locations for the test that minimize the impacts of system features or operating conditions that degrade the precision of the measurements or introduce unknown bias errors.
    4. Specification for whether the same flow capture hood must be used for all outlets (inlets for exhaust systems) or if the use of multiple flow capture hoods is permitted.

c. Specification for system configurations that deviate from applicable requirements in Sections 5.1 through 5.4 that are necessary to achieve the reference operating condition.
d. Specification for the allowable deviation from the established reference operating condition during the data collection phase of the test procedure, and the protocol for determining the system deviation from the reference operating condition.
e. Text description or graphic examples sufficient to specify the format and information content for the leakage test report.

5.5.2 Sources of the test plan may include, but are not limited to, the following:

a. A person or organization that authorizes tests to be performed
b. Building design specifications
c. Building commissioning specifications
d. Regulation or code

Informative Note: Refer to Informative Appendix F for an example test plan.

6. TEST PROCEDURE

6.1 General Requirements

6.1.1 This procedure shall be used to determine the leakage airflow $Q_{leak,std}$, fractional leakage $L_{f,std}$, and the uncertainties $u_{Q_{leak,std}}$ and $u_{L_{f,std}}$ for the system or section under test.

6.1.2 Instrumentation that conforms to the requirements of Section 4 and the test plan shall be installed or applied to the system in accordance with applicable requirements specified in the test plan.

6.1.3 Precision and bias errors shall be determined by means of a previous calibration.

6.2 Measure and Record Data

6.2.1 Data collection for the leakage test shall be performed with the system at the reference operating condition that conforms to applicable requirements specified in Section 5 for establishing the reference operating condition.

6.2.2 Measure and record the following data:

a. Airflow into the supply system Q_{in} or into the exhaust system at each exhaust grille $Q_{in,i}$
b. Airflow out of the supply system at each supply grille or diffuser $Q_{out,i}$ or out of the exhaust system Q_{out}
c. Barometric pressure p_b at the same elevation as the airflow measurements
d. Dry-bulb temperature $t_{db,x}$, relative humidity φ_x, and duct static pressure $p_{s.x}$ at the inlet and outlet airflow measuring planes

Exception to 6.2.2: The duct static pressure at each supply grille or diffuser and at each exhaust grille shall be assumed to be zero and does not need to be measured at these locations.

Informative Note: The duct static gage pressure measured at these locations is insignificant compared to the barometric pressure (which is typically three orders of magnitude or more larger) and thus can be assumed to be zero for practical purposes when calculating air density in Section 6.4.2.

6.3 Time Averaging. Because airflows and pressures in fan-driven air-handling systems are never strictly steady, the airflow, pressure, temperature, or relative humidity indicated on an instrument fluctuates with time. Time averaging of measured parameters is used in this standard to account for such fluctuations. Instrument readings for supply system inlets and exhaust system outlets shall be based on averaged measurement samples recorded using multisample averaging instruments and analyzers designed for this purpose.

Averaging periods and the sample frequency in each period shall be selected to result in at least 20 samples for each measurement parameter (e.g., samples once every second over a 20 second period). The same averaging period shall be used for all measurements at a particular measurement plane.

The time-averaged value of the samples for a particular measurement parameter shall be calculated as follows:

$$\bar{X} = \frac{1}{M}\sum_{i=1}^{M} X_i \tag{1}$$

where

$\bar{X}$ = average value of measured parameter

X_i = i^{th} sample of measured parameter

M = number of samples for measured parameter

The averaging period and the sample frequency for each measurement parameter at each measurement plane shall be documented in the test report.

The time-averaged readings for all measurements at the inlet plane (supply system) or at the outlet plane (exhaust system) shall be determined at the start and end of the test, and the two time-averaged readings for each measurement parameter shall be averaged for use in Section 6.4.

6.4 Calculations

6.4.1 Adjustment for Bias Errors. Except for airflows, mean bias errors shall be subtracted from the measured values to remove the offsets from true values.

Mean bias errors for airflows shall be subtracted from the measured airflow after it is converted to standard air density (Section 6.4.3).

A *positive* mean bias error means that the measured value is greater than the true value, so the bias error is *subtracted* from the measured value. A *negative* mean bias error means that the measured value is less than the true value, so the bias error is *added* to the measured value (because a double negative results in a positive).

6.4.2 Measurement Plane Air Density. The air density ρ_x at each of the airflow measuring planes (duct location) shall be calculated as follows:

$$\rho_x = \frac{p_b + p_{s,x} - 0.378(\varphi_x/100)p_{ws,x}}{R_{da}T_x} \tag{SI 2}$$

$$\rho_x = \frac{144[p_b + 0.036p_{s,x} - 0.378(\varphi_x/100)p_{ws,x}]}{R_{da}T_x} \tag{I-P 2}$$

where

ρ_x = air density at plane x, kg/m^3 (lb$_m$/ft^3)

p_b = measured barometric pressure at same elevation as airflow measurement, Pa (psi)

$p_{s,x}$ = measured static pressure at plane x, Pa (in. of water)

$p_{ws,x}$ = saturation pressure at plane x, Pa (psi)

φ_x = measured relative humidity at plane x, %

R_{da} = gas constant of dry air, 287.042 J/(kg·K); (53.350 ft·lb$_f$/[lb$_m$·°R])

T_x = absolute temperature at plane x;

K = $t_{db,x}$ + 273.15; (°R = $t_{db,x}$ + 459.67)

$t_{db,x}$ = measured air temperature, °C (°F)

The saturation pressure $p_{ws,x}$ over ***liquid water*** for the temperature range 0°C to 200°C (32°F to 392°F) shall be calculated as follows:

$$p_{ws,x} = e^{(C_1/T_x + C_2 + C_3T_x + C_4T_x^2 + C_5T_x^3 + C_6\ln T_x)} \tag{3a}$$

where, in SI units,

C_1 = –5.8002206E+03

C_2 = 1.3914993E+00

C_3 = –4.8640239E-02

C_4 = 4.1764768E-05

C_5 = –1.4452093E-08

C_6 = 6.5459673E+00

and where, in I-P units,

C_1 = –1.0440397E+04

C_2 = –1.1294650E+01

C_3 = –2.7022355E-02

C_4 = 1.2890360E-05

C_5 = –2.4780681E-09

C_6 = 6.5459673E+00

For testing *during cold-weather system operation*, the saturation pressure $p_{ws,x}$ ***over ice*** for the temperature range –100°C to 0°C (–148°F to 32°F) shall be calculated as follows:

$$p_{ws,x} = e^{(C_7/T_x + C_8 + C_9 T_x + C_{10} T_x^2 + C_{11} T_x^3 + C_{12} T_x^4 + C_{13} \ln T_x)} \tag{3b}$$

where, in SI units,

C_7 = –5.6745359E+03

C_8 = 6.3925247E+00

C_9 = –9.6778430E-03

C_{10} = 6.2215701E-07

C_{11} = 2.0747825E-09

C_{12} = –9.4840240E-13

C_{13} = 4.1635019E+00

and where, in I-P units,

C_7 = –1.0214165E+04

C_8 = –4.8932428E+00

C_9 = –5.3765794E-03

C_{10} = 1.9202377E-07

C_{11} = 3.5575832E-10

C_{12} = –9.0344688E-14

C_{13} = 4.1635019E+00

Informative Note: Refer to Informative Appendix D for a derivation of Equation 2.

6.4.3 Airflow Adjustment to Standard Air Density and Correction for Bias Errors. Each airflow Q_x measured at the corresponding measurement plane *x* shall be adjusted to standard air density and then adjusted for bias errors at standard air density as follows:

$$Q_{x,std} = \left[\left(\frac{\rho_x}{\rho_{std}}\right) Q_x\right](1 - B_{Q_{x,std}\_frac}/100) - B_{Q_{x,std}\_abs} \tag{4}$$

where

$Q_{x,std}$	=	standard airflow at plane *x*, L/s (cfm)
ρ_{std}	=	standard air density, 1.204 kg/m^3 (0.075 lb$_m$/ft^3)
Q_x	=	measured airflow at plane *x*, L/s (cfm)
$B_{Q_{x,std}\_frac}$	=	*fractional* mean bias error for Q_x at standard air density at plane *x*, %
$B_{Q_{x,std}\_abs}$	=	*absolute* mean bias error for Q_x at standard air density at plane *x*, L/s (cfm)

6.4.4 Leakage Airflow. For *single-duct supply* systems, $Q_{leak,std}$ is a positive value representing the magnitude of the leakage airflow and shall be calculated as follows:

$$Q_{leak,std} = Q_{in,std} - \sum_{j=1}^{N'}\left(\sum_{i=1}^{n_j} Q_{out,i,std}\right) \tag{5a}$$

where

$Q_{leak,std}$	=	leakage airflow at standard air density, L/s (cfm)
$Q_{in,std}$	=	supply inlet airflow at standard air density, L/s (cfm)
$Q_{out,i,std}$	=	supply outlet *i* airflow at standard air density, L/s (cfm)
N'	=	number of flow capture hoods or measuring stations used to measure all $Q_{out,i,std}$ in test section
n_j	=	number of supply outlet airflows measured by flow capture hood *j* or measuring station *j*

For *independent exhaust* systems, $Q_{leak,std}$ is a positive value representing the magnitude of the leakage airflow and shall be calculated as follows:

$$Q_{leak,\,std} = Q_{out,\,std} - \sum_{j=1}^{N'}\left(\sum_{i=1}^{n_j} Q_{in,\,i,\,std}\right) \tag{5b}$$

where

$Q_{leak,std}$ = leakage airflow at standard air density, L/s (cfm)

$Q_{out,std}$ = exhaust outlet airflow at standard air density, L/s (cfm)

$Q_{in,i,std}$ = exhaust inlet i airflow at standard air density, L/s (cfm)

N' = number of flow capture hoods or measuring stations used to measure all $Q_{in,i,std}$ in test section

n_j = number of exhaust inlet airflows measured by flow capture hood j or measuring station j

6.4.5 Fractional Leakage. For *single-duct supply* systems, the fractional leakage $L_{f,std}$ shall be calculated as follows:

$$L_{f,\,std} = 100\left(\frac{Q_{leak,\,std}}{Q_{in,\,std}}\right) \tag{6a}$$

where

$L_{f.std}$ = leakage flow fraction, %

For *independent exhaust* systems, the fractional leakage $L_{f,std}$ shall be calculated as follows:

$$L_{f,\,std} = 100\left(\frac{Q_{leak,\,std}}{Q_{out,\,std}}\right) \tag{6b}$$

6.4.6 Standard Precision Errors. Fractional precision errors $P_{Q_{in,\,std}\_frac}$, $P_{Q_{out,\,std}\_frac}$ and absolute precision errors $P_{Q_{in,\,std}\_abs}$, $P_{Q_{out,\,std}\_abs}$ for airflow measurements, as applicable, shall be converted to standard precision errors (one standard deviation, which corresponds approximately to a 68% confidence level) as follows.

For *single-duct supply* systems:

$$S_{Q_{in,\,std}} = \frac{\max\left[\frac{(P_{Q_{in,\,std}\_frac}Q_{in,\,std})}{100},\, P_{Q_{in,\,std}\_abs}\right]}{k_p} \tag{7a}$$

$$S_{Q_{out,\,i,\,std}} = \frac{\max\left[\frac{(P_{Q_{out,\,i,\,std}\_frac}Q_{out,\,i,\,std})}{100},\, P_{Q_{out,\,i,\,std}\_abs}\right]}{k_p} \tag{7b}$$

where

$S_{Q_{in,\,std}}$ = standard precision error at standard air density for measuring station used to measure $Q_{in,std}$, L/s (cfm)

$P_{Q_{in,\,std}\_frac}$ = *fractional* precision error at standard air density for measuring station used to measure $Q_{in,std}$ at plane x, %

$P_{Q_{in,\,std}\_abs}$ = *absolute* precision error at standard air density for measuring station used to measure $Q_{in,std}$ at plane x, L/s (cfm)

$Q_{out,\,i,\,std}$ = standard precision error at standard air density for flow capture hood or measuring station j used to measure $Q_{out,i,std}$, L/s (cfm)

$P_{Q_{out,\,i,\,std}\_frac}$ = *fractional* precision error at standard air density for flow capture hood or measuring station j used to measure $Q_{out,i,std}$ at plane x, %

$P_{Q_{out,\,i,\,std}\_abs}$ = *absolute* precision error at standard air density for flow capture hood or measuring station j used to measure $Q_{out,i,std}$ at plane x, L/s (cfm)

k_p = coverage factor (from Table 1)

Table 1 Coverage Factors for Commonly Specified Confidence Levels

Instrument Precision Confidence Level, %	Coverage Factor (Number of Standard Deviations), k_p
68.27	1.000
90	1.645
95	**1.960**
95.45	2.000
99	2.576
99.73	3.000

Likewise, for *independent exhaust* systems,

$$s_{Q_{out,std}} = \frac{\max\left[\frac{(P_{Q_{out,std}\_frac} Q_{out,std})}{100}, P_{Q_{out,std}\_abs}\right]}{k_p} \tag{8a}$$

$$s_{Q_{in,i,std}} = \frac{\max\left[\frac{(P_{Q_{in,i,std}\_frac} Q_{in,i,std})}{100}, P_{Q_{in,i,std}\_abs}\right]}{k_p} \tag{8b}$$

where

$s_{Q_{out,std}}$ = standard precision error at standard air density for measuring station used to measure $Q_{out,std}$, L/s (cfm)

$P_{Q_{out,std}\_frac}$ = *fractional* precision error at standard air density for measuring station used to measure $Q_{out,std}$ at plane x, %

$P_{Q_{out,std}\_abs}$ = *absolute* precision error at standard air density for measuring station used to measure $Q_{out,std}$ at plane x, L/s (cfm)

$s_{Q_{in,i,std}}$ = standard precision error at standard air density for flow capture hood or measuring station j used to measure $Q_{in,i,std}$, L/s (cfm)

$P_{Q_{in,i,std}\_frac}$ = *fractional* precision error at standard air density for flow capture hood or measuring station j used to measure $Q_{in,i,std}$ at plane x, %

$P_{Q_{in,i,std}\_abs}$ = *absolute* precision error at standard air density for flow capture hood or measuring station j used to measure $Q_{in,i,std}$ at plane x, L/s (cfm)

6.4.7 Leakage Flow Uncertainty. For *single-duct supply* systems, the overall standard uncertainty in the leakage flow $u_{Q_{leak,std}}$ shall be calculated as follows:

$$u_{Q_{leak,std}} = \sqrt{s_{Q_{in,std}}^2 + \sum_{j=1}^{N'}\left[\left(\sum_{i=1}^{n_j} s_{Q_{out,i,std}}\right) / \sqrt{n_j}\right]^2} \tag{9a}$$

where

$u_{Q_{leak,std}}$ = overall standard uncertainty for $Q_{leak,std}$, L/s (cfm)

Likewise, for *independent exhaust* systems, the overall standard uncertainty in the leakage flow $u_{Q_{leak,std}}$ shall be calculated as follows:

$$u_{Q_{leak,std}} = \sqrt{s_{Q_{out,std}}^2 + \sum_{j=1}^{N'}\left[\left(\sum_{i=1}^{n_j} s_{Q_{in,i,std}}\right) / \sqrt{n_j}\right]^2} \tag{9b}$$

where

$u_{Q_{leak,std}}$ = overall standard uncertainty for $Q_{leak,std}$, L/s (cfm)

6.4.8 Fractional Leakage Uncertainty. For *single-duct supply* systems, the overall standard uncertainty in the fractional leakage $u_{L_{f,std}}$ shall be calculated as follows:

$$u_{L_{f,std}} = 100\left(\frac{u_{Q_{leak,std}}}{Q_{in,std}}\right) \tag{10a}$$

For *independent exhaust* systems, the overall standard uncertainty in the fractional leakage $u_{L_{f,std}}$ shall be calculated as follows:

$$u_{L_{f,std}} = 100\left(\frac{u_{Q_{leak,std}}}{Q_{out,std}}\right) \tag{10b}$$

Informative Note: Refer to Informative Appendix C for example calculations that include uncertainty analysis.

7. TEST REPORT

The test report shall document the results of the test as specified in the test plan and include, but not be limited to, the following information.

7.1 Date, time, and location of test.

7.2 Name of organization and technician who performed the test.

7.3 System identification or name and location or area served by the system.

7.4 The manufacturer's model number and serial number for the system fan or air-handling unit (AHU) serving the section under test.

7.5 A text description or schematic diagram of the system features sufficient to clarify the scope of the leakage test performed.

7.6 A text description or schematic diagram sufficient to clarify the test instrumentation used and data collected. The description shall include the instrument manufacturer's model number, serial number, instrument accuracy (precision and mean bias errors), and calibration status for instruments used for the test.

7.7 The test plan incorporated as an appendix to the test report.

7.8 A text description or schematic diagram sufficient to define the configured state of all system parameters used to achieve the reference operating condition specified in Sections 5.1 through 5.4 and Section 5.5.1. The configured state of all such parameters during the test, whether overridden or not, shall be reported to enable test repeatability.

7.9 The system's deviations from the reference operating condition during the data collection phase of the test.

7.10 The averaging period and the sample frequency for each measurement parameter at each measurement plane.

7.11 Test results and calculated values for leakage airflow $Q_{leak,std}$ (Equations 5a or 5b, as applicable), fractional leakage $L_{f,std}$ (Equations 6a or 6b, as applicable), leakage flow uncertainty $u_{Q_{leak,std}}$ (Equations 9a or 9b, as applicable), and the fractional leakage uncertainty $u_{L_{f,std}}$ (Equations 10a or 10b, as applicable).

7.12 A description of any unusual conditions present during the test, such as extreme weather conditions, that will affect the repeatability of the test results.

Informative Note: Refer to Informative Appendix F for an example test report.

(This appendix is not 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. Unresolved objectors on informative material are not offered the right to appeal at ASHRAE or ANSI.)

INFORMATIVE APPENDIX A
AIRFLOW MEASURING INSTRUMENT TECHNOLOGIES

A1. AIRFLOW MEASURING STATIONS

A1.1 Description. Airflow measuring stations (AFMS) are specialized devices that are permanently mounted in the airstream to provide a continuous output of the duct average velocities at the plane of measurement, typically for HVAC control or system monitoring. The most popular technologies used for airflow measuring stations are thermal dispersion and velocity pressure (e.g., pitot tubes, arrays, fan inlet piezo rings).

The primary technologies used with their effective operating velocity ranges can be compared to the airflow range for the three primary HVAC applications. Figure A-1 provides a visual representation of the suitability of measurement technologies for specific air measurement applications.

Measurement accuracy potential varies with the technology. Some airflow measuring stations are potentially sensitive to air pressure or velocity profiles, duct disturbances in the path of measurement, and duct turbulence intensity. Others are designed to be less sensitive to the causes of these errors. *ASHRAE Handbook—Fundamentals* (~~2017~~) provides instrument selection guidance regarding the recommended limits for velocity or volumetric airflow. 2025a

A1.1.1 Thermal Dispersion Stations. These devices are a means of measurement where thermal energy transfer is used to determine air velocity. The energy dispersed from a heated sensor in the airstream (typically a thermistor or resistance temperature detector [RTD]) is directly related to the air velocity passing the sensor element. Accurate velocity measurement requires precise and high-resolution temperature measurement at each point where velocity is sampled. Thermal methods are generally more sensitive to lower velocities than other measurement methods.

In most cases, one sensor is self-heated or heated indirectly from an external source, whereas the other sensor measures the ambient temperature of the flowing gas. The measured energy lost as a result of thermal transfer is directly proportional to velocity (not pressure). Conversion of the thermal sensors' raw analog voltage/current signal to an air volume indication is performed by microprocessor-based hardware and software that are proprietary to each station manufacturer.

Every manufacturer's conversion method is different but most solve for velocity by maintaining a constant temperature or a constant current between the two sensors. The relationship between the voltage, resistance, or current measured, and the temperature of the sensors are used to calculate the air velocity. These sensors are typically arrayed in a grid such that a thermal dispersion airflow measuring station can ultimately provide the velocity-weighted temperature average or the area-weighted mass airflow at the duct cross section being measured. With some methods, very precise analog to digital (A/D) conversion is required to avoid compounded measurement error. In others, very precise voltage measurement is more important.

Thermal dispersion devices have sources of potential error such as the quality of the components used to perform very precise functions, the design's resistance to environmental conditions, response time and stability of the components, the quality of a factory calibration versus field calibration references, and duct placement sensitivity.

A1.1.2 Velocity Pressure Stations. These in-duct devices are based on differential pressure or velocity pressure measurement and include the following

a. Fan inlet piezo rings
b. Self-averaging arrays or interconnected pitot probes
c. Fixed array of pitot-static tubes
d. Air measuring dampers used in combination with intake louvers
e. Flow-cross or flow-ring sensors used in variable-air-volume (VAV) terminal units.

For these airflow measuring stations, sampling ports are continuously exposed to changing total and static pressures. Pressures are accumulated or equalized in the manifolds connecting multiple elements arranged in an array and installed in a plane perpendicular to the direction of airflow in a duct. The resulting total differential or velocity pressure is measured by a single differential pres-

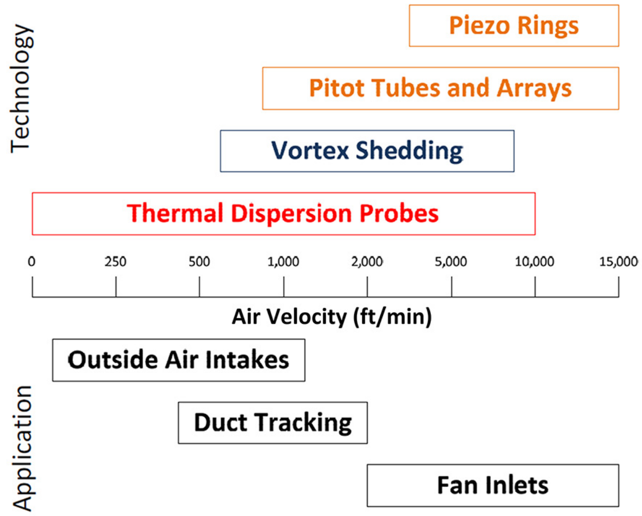


Figure A-1 Ranges of operation for airflow measuring stations.

sure transducer such that its output signal is proportional to the equalized nonlinear velocity pressure. The nonlinear electronic signal must then be made linear for use by the building automation system (BAS) or some other form of processing.

All pressure-based devices are limited by the errors contributed by the pressure sensing devices used. For some of these devices, accuracy is stated as a fraction of the full-scale reading, which results in significantly increased inaccuracy at lower airflows. When possible, instruments should be selected that have accuracy specifications as a function of the readings rather than full scale.

The nonlinear square root relationship between velocity pressure and velocity results in rapidly increasing velocity uncertainties at low velocities and strongly depends on the uncertainty of the pressure sensing device used. For example, consider a pressure sensing device with a precision equal to or better than 1% of measured pressure or 0.1 Pa (0.0004 in. of water), whichever is greater. At velocities above 3 m/s (600 fpm), the uncertainty of the corresponding velocity is about 1% or better. For lower velocities, the velocity uncertainty increases to about 2% at 2 m/s (400 fpm) and about 9% at 1 m/s (200 fpm). If the latter precision specification changes from 0.1 Pa (0.0004 in. of water) to 1 Pa (0.004 in. of water), the 1% velocity threshold occurs at about 9 m/s (1800 fpm). In this case, the velocity uncertainties at 2 m/s and 3 m/s increase to about 20% and 10%, respectively.

Velocity pressure arrays depend heavily on sensing the equalized pressures in their interconnected passageways. They must avoid potential averaging errors due to duct disturbances by seeking the best measurement conditions possible and airflow conditions without a significant pressure profile. Because these conditions are rarely encountered in operating duct systems, measurement performance can rarely be maximized in the field. Averaging errors that result from arithmetically averaging nonlinear values, together with poor placement conditions, can easily exceed 30% to 40% of a reading and many times can reach double that.

A1.2 Common-Sense Installation Advice. AFMS duct placement recommendations are as follows:

a. Follow the manufacturer's instructions for placement, installation, operation, calibration, and maintenance.
b. Provide more than minimum access for sensors to ensure that maintenance can be performed regularly and easily.
c. Do not measure downstream of modulating dampers.
d. Do not measure upstream of a fully modulating damper that is downstream of a fan.
e. Do not increase intake velocities near intake louvers to prevent a water carryover problem.
f. Do not measure directly downstream or immediately upstream of sound attenuators without a detailed application review by the instrument manufacturer.
g. Avoid overlubrication of fan bearings when in proximity to any type of sensors.

h. Avoid instrument damage by water exposure or accumulation.
i. Do not place the airflow station within the absorption distance of humidifiers.
j. Avoid placement locations with significant temperature changes.

A2. FLOW CAPTURE HOODS

Commercially available flow capture hoods for measuring grille and diffuser airflows typically use a fabric skirt that is fixed to a rigid frame that fits over the grille or over part or all of the diffuser (e.g., a slot diffuser). The hood has a larger open end that is usually sized to fit a commercial supply diffuser of 560 × 560 mm (22 × 22 inches), with other hood sizes offered by many manufacturers. The fabric hood directs the flow over a velocity or pressure-drop sensing element. Most of these devices have built-in electronic signal processing and information displays that include the ability to perform time averaging, temperature compensation, and variable insertion loss correction. However, some use analog gages to display the flow, have no temperature compensation, and have a fixed insertion loss correction.

Some flow capture hoods are more sensitive than others to flow nonuniformities at a grille/diffuser and to how the hood is positioned over the grille/diffuser, especially for supply grilles/diffusers. In some cases, as shown in Figures A-2 and A-3 for two commercially available hoods, simple linear correction factors independent of grille/diffuser type and airflow can be determined by calibration. In these plots, the relative error shown corresponds to the RMS error (approximately 68% confidence level). Figure A-2 shows the five different supply outlet diffusers that were used in the tests.

However, for some commercially available hoods, as shown in Figure A-4, the correction factors depend nonlinearly on grille/diffuser type and airflow such that a correction factor would need to be determined for every grille/diffuser and airflow, which is impractical.

Also, using hoods shorter than the full length of a slot diffuser means that hood insertion losses can cause flows to be reduced where the hood is placed and increased where it is not. Summing flows measured over the entire length of the diffuser can lead to substantial negative bias errors.

Because there is no industry standard for flow capture hood calibration, Section B2 in Informative Appendix B provides a laboratory method of test for determining the measurement accuracy (bias and precision errors) of flow capture hoods.

Powered flow capture hoods should be used whenever possible. They use a flow capture device connected to a fan and calibrated flowmeter. Some commercially available powered hoods (typically intended for residential applications) directly connect the flow capture device to the fan and flowmeter. Others use a length of flex duct and a flow straightener placed between the flow capture device and the fan and flowmeter to make this integrated flow measurement system less sensitive to nonuniform flows at the grille/diffuser. Adjusting the fan speed until there is no static pressure difference between the room and the hood compensates for the flow resistance of the capture hood, flexible duct, and flowmeter. This pressure balancing procedure ensures that placing the flow capture hood over the grille/diffuser does not reduce the grille flow.

The integrated device described above is not commercially available as a complete package for commercial building applications, but many practitioners already have flow capture hoods that can be adapted to use a fan and flowmeter by using flex duct to connect the hood to the fan and flowmeter. In this case, airflow is determined using the added flowmeter and not the flow capture hood's internal meter. The Energy Conservatory (TEC ~~2017)~~ 2023 provides instructions for constructing a powered flow hood using these components. Laboratory tests of a powered flow hood by Lawrence Berkeley National Laboratory in 2005 (unpublished) have shown that the accuracy of such a hood can be as good as 0.2% bias error and 0.6% precision error (at a 95% confidence level).

Because laboratory results (Walker et al. 2001; Wray et al. 2002; Etur 2003) showed that a powered flow capture hood is more accurate than other hoods, it also can be used as a reference hood to evaluate other hoods in field applications.

A3. TRACER GASES

Measurement of airflows in ducts using tracer gases is well established in the research community but is not regularly used in the HVAC testing, adjusting, and balancing industry. Historically, this has been due to the cost and complexity of tracer-gas concentration measurement equipment. However, costs and complexity have decreased significantly.

The details of how to measure duct airflows using this technique are provided in ASTM Standard E2029 (ASTM 2019). In brief, this technique involves injecting a known quantity of a specific tracer

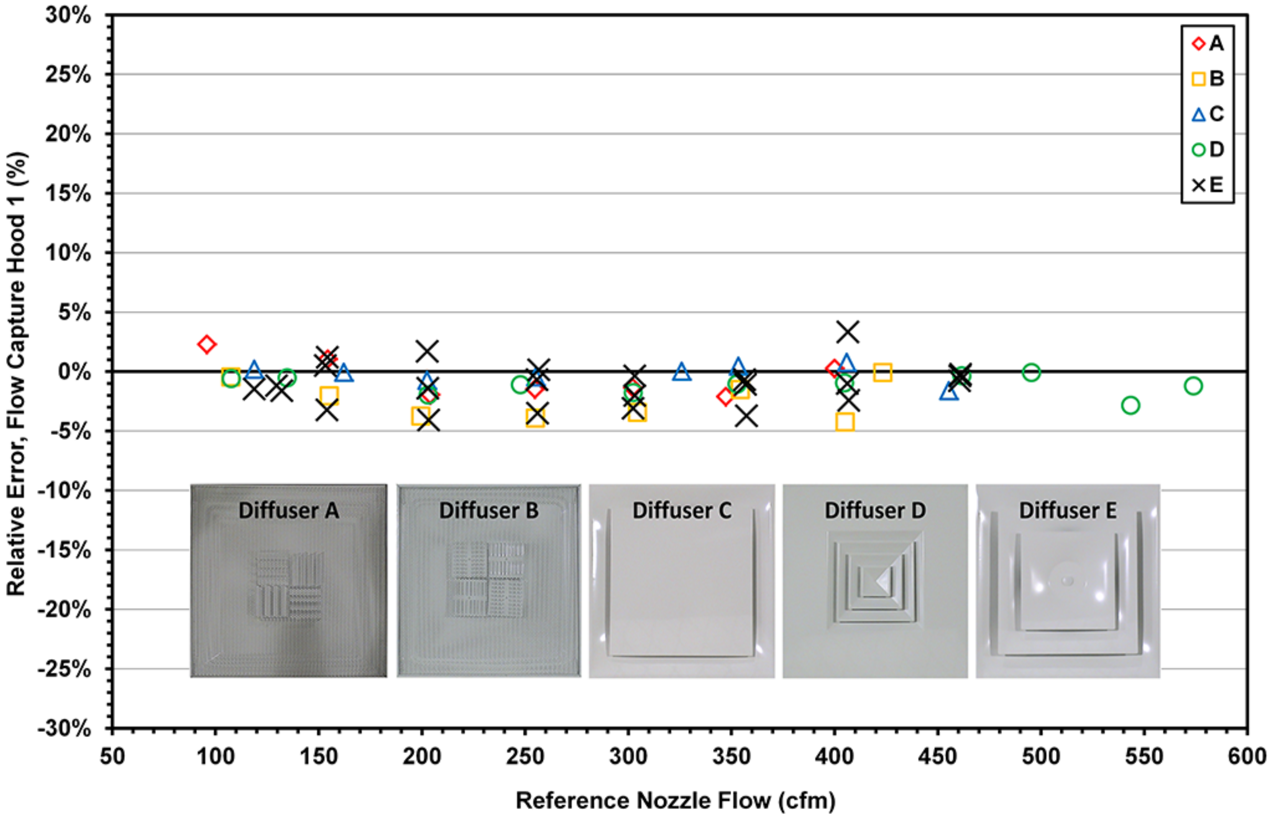


Figure A-2 Flow capture hood test data: small negative bias error.

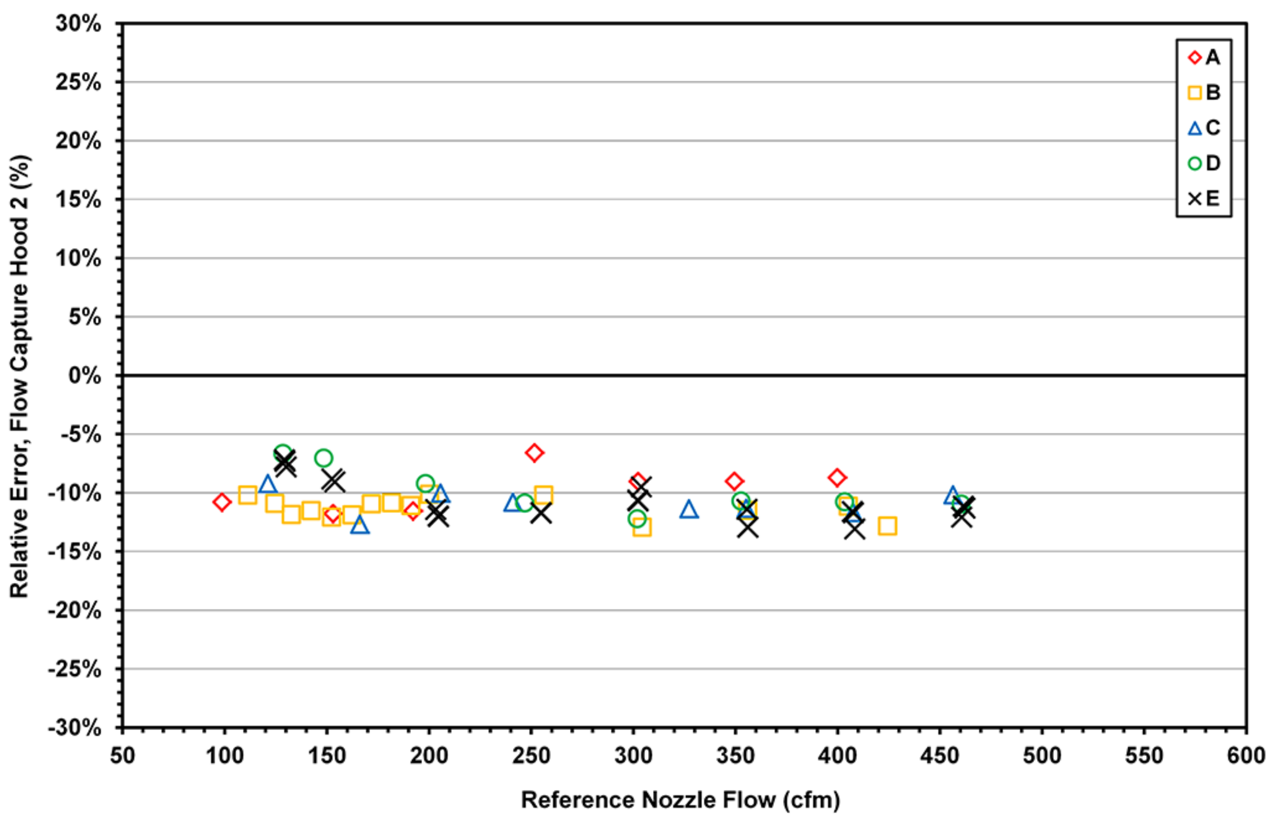


Figure A-3 Flow capture hood test data: larger negative bias error.

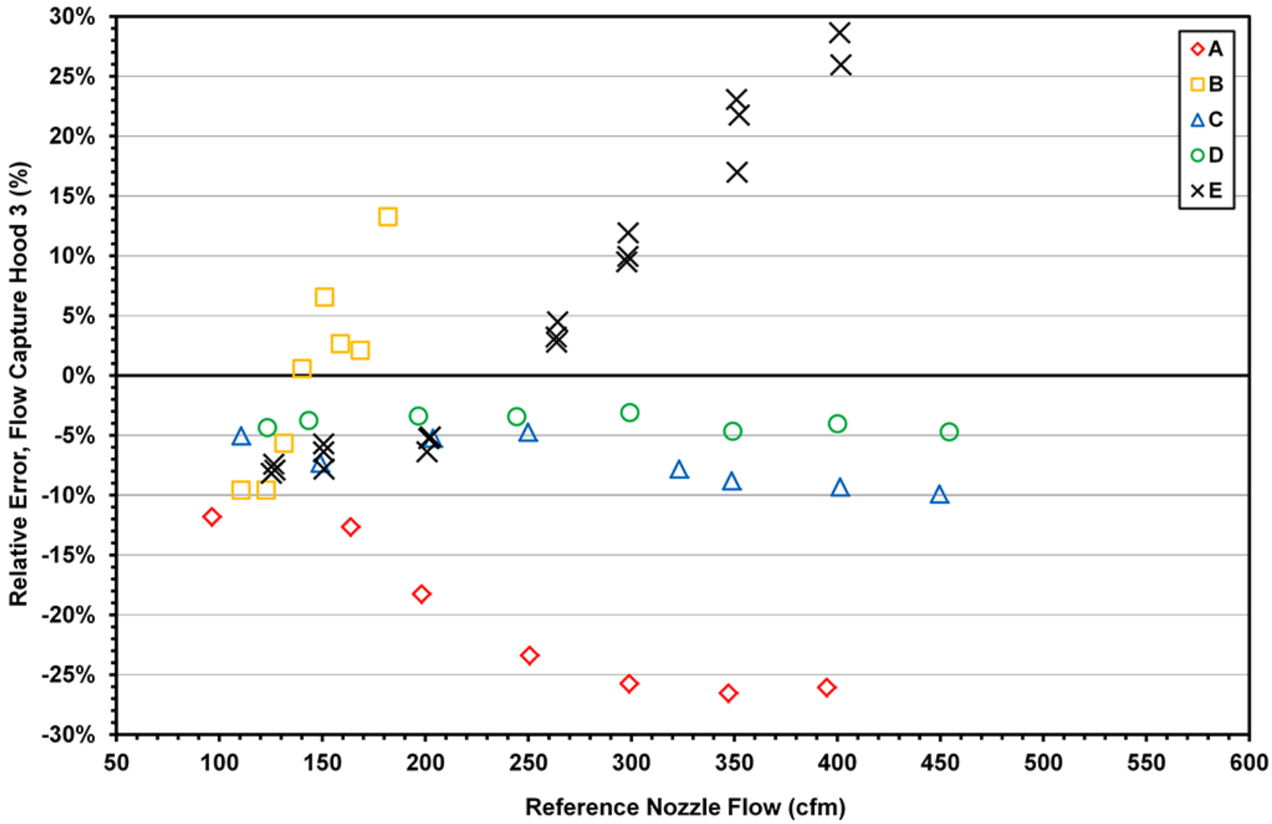


Figure A-4 Flow capture hood test data: scattered errors.

gas at a constant rate and then measuring the concentration of that gas downstream of the injection site. The duct airflow is determined by the relationship between the amount of tracer gas injected and the average measured concentration. In the case of steady-state injection (as documented in ASTM E2029), the mass flow rate of air is equal to the mass flow rate of tracer injection into the duct divided by the measured concentration of tracer gas in the air flowing through the duct.

One significant advantage of this technique is that it does not depend on having straight duct sections and well developed flows. The biggest challenge with this technique, however, is ensuring that the tracer gas is well mixed with the air flowing through the duct. This challenge is generally addressed by dispersed injection and forced mixing, as well as by checking the uniformity of the concentration. In addition, for supply air distribution systems, concentration can be measured well downstream of the injection plane, because the concentration is not impacted by supply duct leaks (except possibly in the case of gross short circuiting of the tracer gas injection to a leak) before full mixing has occurred. Turbulence induced by fittings helps achieve mixing. Also, the impact of background concentrations of the tracer gas (including changes with time) needs to be included in the analysis (detailed in ASTM E2029). With good mixing, tracer gases can be used to measure airflows to within 2% of the true flow at a 95% confidence level (Wang and Sherman 2004).

(This appendix is not 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. Unresolved objectors on informative material are not offered the right to appeal at ASHRAE or ANSI.)

INFORMATIVE APPENDIX B
AIRFLOW MEASURING INSTRUMENT CALIBRATION AND VERIFICATION PROCEDURES

B1. AIRFLOW/VELOCITY MEASURING STATIONS

B1.1 General. To help reduce uncertainty in test results, the committee offers the following recommendations regarding airflow measuring station (AFMS) performance verification and output adjustment, if and when necessary.

Factory calibration of some instruments is available, but the user must ensure that the calibration of the reference instrument is traceable to primary or secondary standards. Whenever possible, instruments should be applied in a manner that ensures compliance with the accuracy specified in the test plan.

Field calibration is more appropriately referred to as *verification and adjustment*. Here, performance is comparatively analyzed and the unit under test (UUT) output is adjusted to match a field reference measurement. These adjustments are useful but cannot provide the level of statistical confidence in measurement results available with a precision airflow reference system in a purpose-built laboratory. The effectiveness of field adjustments are determined by issues such as the level of quality in the field reference instrument or assembly, in the quality and design of the UUT being adjusted, the conditions at the measurement place, the test technician's qualifications, and the validity of the procedures used to perform the comparison.

For the calibration, the reference airflow measuring system should be certified to provide accuracy at least four times better than the specified tolerance at the measurement point and under all the conditions required for the intended use. For example, if the test tolerance is ±3% at a particular test point, then the total uncertainty in the reference airflow measuring system should be no more than ±0.75% at that operating point.

Some devices, when they meet specific physical design requirements, are considered primary instruments but are not useful without other devices such as pressure sensors, recorders, and loggers, which must be used together as an integrated lab measurement system. The accuracy of these combined devices in the field can be affected by unrecognized factors. Thus, the assembly and measurement process should be calibrated as an assembly.

Each installed airflow instrument assembly or UUT should be field verified in comparison to an airflow reference system at both ends of the UUT scale and at least four velocity points for a typical total of three intermediate velocities plus zero. The analog output or display resolution of the reference should be less than or equal to 0.01 m/s (2 fpm) for velocities from 0 to 25.4 m/s (5000 fpm). For digital outputs, there are no resolution issues. Displays used for this purpose should be capable of indicating velocity to a fixed four decimal places or eight significant digits in floating point calculations. Average velocity and temperatures are absolute numbers determined within the statistical uncertainty of the instrument and can be transmitted in eight significant digits or fixed decimal formats, depending on the system design.

B1.2 Airflow/Velocity Field Verification and Adjustment. When factory calibration is not possible, when not available for a specific meter design, or when duct conditions significantly affect measurement performance, the performance of an airflow measuring station still should be verified, and the output may need field adjustment.

The only two alternatives at that point, in order of performance superiority, are as follows:

a. Form a closed circuit or a direct path between a trusted reference and the unit needing adjustment, and then adjust the device in question to match the trusted unit. Duct or damper leakage must be accounted for in the results.
b. The method used by most testing, adjusting, and balancing (TAB) technicians and building operations staff is to use the best hand-held or permanent duct measurement reference available and compare the output of the airflow station to it or to a manual duct traverse at two or more operating points in the same airstream. By using only the best duct conditions available, the user reduces the uncertainty in the reference.

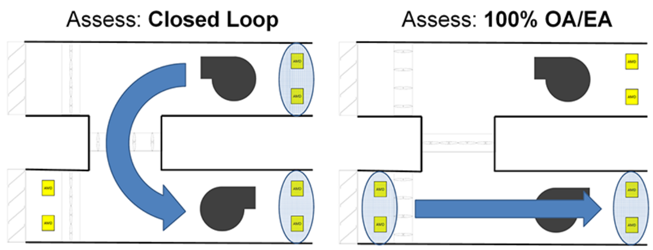


Figure B-1 AFMS assessment paths.

Field adjustment in the first case still employs a device having traceable performance in repeatable conditions. In the second and more common method, performance of the adjusted output is limited by several factors: the overall accuracy of the reference instrument, the method of measurement comparison that is used, the ability and care of the technician, the field conditions available, the duct conditions where the station is located, and the repeatability of a manual measurement. Refer to ISO 3966 (ISO ~~2020)~~ for a description of the primary sources of error in a pitot-static tube traverse.

B1.2.1 When Not to Adjust. The uncertainty of the method used to determine that the airflow station is reading in error should be analyzed thoroughly before an adjustment is made. Adjustment of factory calibrated airflow measuring stations is not recommended, but in certain conditions it may be necessary. Details of any output adjustment should be recorded in the test report and also filed with the operations and maintenance (O&M) documents held at the site. If it is determined that airflow measuring station errors are not the result of factors in the controls, mechanical system, zero drift, or duct conditions, only then should an adjustment to any device be made. Most methods of field adjustment do not possess the repeatability necessary to avoid compounding uncertainties from unnecessary adjustments. Changing the output from factory calibration unnecessarily diminishes the original accuracy of the device.

It should never be assumed that comparison to field measurements with simple handheld instruments (e.g., a rotating vane anemometer) indicates that the AFMS is responsible for the difference and must be adjusted to match the TAB reported actual duct average flow. There are not only wide ranges of quality in the field AFMS/UUT that are commercially available, but also a wide range of possible alternative sources for the differences reported. Therefore, it is recommended that one evaluate and eliminate alternative causes before concluding that adjustment is the only remedy. The potential impact from the adjustment should be considered by comparing the accuracy of the field reference being used to that of the UUT.

B1.3 Suggested Methods of AFMS Verification. The following are some of the methods and comparison references recommended for use in the verification and adjustment of factory calibrated airflow measuring stations after it is determined that measurement conditions or contractual requirements justify a field adjustment.

B1.3.1 Ducted Supply or Return/Exhaust AFMS

a. The best way to verify and adjust a precalibrated thermal dispersion array is to compare one AFMS to another in a closed loop as shown in Figure B-1. Low-leakage dampers are needed to block returning/recirculated or outdoor air from entering the isolated path. In cases where leakage is excessive, temporary blocks or seals should be added to minimize leakage. Regardless, duct or system leakage within the calibration path must be evaluated before instrument adjustments are made.
b. Another good way, using fan-inlet-mounted devices, is to follow these steps:
    1. Position precalibrated thermal fan inlet sensors to the flange of an inlet cone with factory-supplied mounting template.
    2. Record the UUT response throughout the fan range determined by a NIST-traceable lab tunnel with measurement uncertainty equal to or better than 0.75% of the reading at minimum flow.
    3. Develop a K-factor and use it with factory-calibrated units having the same size inlet cone and fan type.
    4. Use the mounting template to install other fan inlet AFMS units in the field.

5. Subsequent fan inlet units should not need additional adjustment. If it is found that conditions differ enough to justify adjustment, adjust subsequent AFMS units using the reference UUT/AFMS with the closed-loop or single-path methods previously described.

c. How to adjust the output:
   1. Most devices possess the means to change the gain (slope) and span (intercept) of a linear analog output signal.
   2. When the output response of a device intersects zero, make a one-point gain-only adjustment.
   3. When the output response does not intersect zero, make a two-point offset and gain adjustment.
   4. Some transmitters provide software or firmware to automate these adjustments and reduce the potential for human error.

B1.3.2 Fan Inlet AFMS Verification for Plenum Supply. Thermal stations of the same size can be compared to each other for diagnostics or periodic verification and adjustment if the equipment involved is stable and does not drift. Otherwise, the zero should first be verified and adjusted appropriately.

Velocity pressure fan inlet stations each require output adjustments to a secondary reference because small deviations in the calculation of the diameter in the inlet cone at the plane of measurement are critical to accuracy in the volume calculated.

B1.3.3 Adjustment of Ducted Velocity Pressure Stations. Because of the natural sensitivity of this technology to the presence of duct disturbances or pressure profiles, almost all of these devices are required to be set up and their outputs adjusted in the field. The issue then becomes how one can maximize the performance of these devices in the field and how they should be set up. The actions required include the following:

a. Place/install the device as far downstream of duct disturbances as possible so that the pressure profile can fully develop before the measurement plane, without intermediate ducts or devices.
b. Select a pressure transducer
   1. with a full-scale range that narrowly defines the velocity pressure operating range required, leaving headway of about 20% to avoid out-of-range readings;
   2. that uses only the highest-accuracy pressure sensor/transducer (lowest error) available and that automatically compensates for ambient temperature changes; and
   3. that autozeros (zero drift reset) its output at least once every 24 hours and preferably more often (e.g., once per minute).
c. Provide adequate accessibility for inspection and removal of the UUT for maintenance.
d. Use recently calibrated, NIST-traceable instruments to verify the velocity pressure UUT and reference source accuracy for initial set up and periodic adjustment.
e. Alternatively, send one UUT station with specific transducer and accessories to a National Voluntary Laboratory Accreditation Program (NVLAP) accredited and ISO/IEC 17025 (IEC 2018) certified calibration laboratory for testing as an assembly. The lab must have sufficient airmeter performance superiority/capabilities. Comparative testing should result in the output factors needed for the UUT response to reflect the corrected data. Once installed, use the tested UUT as the field reference to evaluate and adjust like-UUT in the same system, using a closed-loop or single-path method, as described for thermal dispersion instruments.

   NVLAP is a NIST-qualified voluntary program that uses the technical and management requirements from ISO/IEC 17025. The process described here may also be used for uncalibrated or non-NIST-traceable thermal flowmeters that have sufficient repeatability to reduce uncertainty.
f. If it is necessary to use handheld reference instruments, they should be located either upstream of the UUT or in the best possible other location within the same air path.
g. Temporarily block or remove disturbance sources if possible to improve set-up conditions.
h. Adjust UUT output to match the reference instrument or reference station.

B2. LABORATORY METHOD OF TEST FOR DETERMINING THE MEASUREMENT ACCURACY OF FLOW CAPTURE HOODS

B2.1 Purpose and Scope. The following provides a laboratory method of test to determine the measurement accuracy (bias and precision errors) of flow capture hoods used in field applications to measure airflows through user-selected air distribution system supply outlets and/or independent exhaust system inlets. It provides specifications for test equipment, bias and precision error calculations, and a format for reporting the test results. This method is not intended for field calibration of measurement devices.

B2.2 Summary of Test Method. The test method compares airflows determined by flow capture hoods to airflows through a reference flowmeter in a laboratory test apparatus. The test method includes a range of supply outlets or exhaust inlets, and airflows characteristic of nonresidential supply or exhaust air systems, as selected by the user. The testing includes the effects of hood placement relative to each outlet or inlet, where applicable. Tests are also included to evaluate hood insertion loss effects on the sum of flows through multi-branch systems (i.e., the tendency of hood flow resistance to reduce diffuser/grille airflows at the supply outlet or exhaust inlet being measured).

B2.3 Test Apparatus. Two different test apparatus configurations are required: one for single-branch testing (Section B.2.3.1) and the other for multibranch testing (Section B.2.3.2). The test apparatus components shall consist of the following:

a. A fan located at the test apparatus inlet (for testing supply outlets) or outlet (for testing exhaust inlets), as applicable, with an airflow control mechanism that can be adjusted to provide specified airflows through the test apparatus. The fan shall be capable of providing airflows through the test apparatus over the full range of the flow capture hood and supply outlets or exhaust inlets being tested.[1]
b. A reference flowmeter capable of measuring airflow with a precision equal to or better than 0.5% of reading. Straight ducts meeting the reference meter manufacturer's specifications, but at least 10 duct diameters (10 D) in length, shall be installed upstream and downstream of the reference flowmeter. A flow straightener shall be located within 2 D upstream of the reference meter inlet. The flowmeter shall be capable of measuring the flow through all of the supply outlets or exhaust inlets in the system.[1]
c. A set of supply outlets and/or exhaust inlets representative of the ones used in the actual systems to be tested in the field applications.
d. Ductwork to connect the fan, reference flowmeter, and supply outlets or exhaust inlets.[1]
e. A differential pressure gage capable of measuring air pressure with a precision equal to or better than 0.2 Pa (0.0008 in. of water) or 1% of reading, whichever is greater. The duct static pressure shall be measured 5 D downstream of the reference flowmeter exit when supply outlets are tested and 5 D upstream of the reference flowmeter entry when exhaust inlets are tested.
f. An air-temperature measuring device with a precision equal to or better than ±0.5°C (0.9°F). Air temperature in the duct shall be measured 5 D upstream of the reference flowmeter. Relative humidity shall be measured at the same location with a precision equal to or better than 5%.
g. A barometric pressure measuring device with a precision equal to or better than ±250 Pa (1.00 in. of water).

The required precision for each of the measuring instruments listed above corresponds to a 95% confidence level. Instrumentation (other than the flow capture hood being tested) shall comply with Section 4 of this standard unless stated otherwise here.

B2.3.1 Single-Branch Apparatus. Figures B-2 and B-3 are illustrations of a typical single-branch test apparatus. For evaluating performance with *supply outlets*, the single-branch test apparatus shall be constructed such that the fan or blower *blows air* through the duct that contains the reference flowmeter and terminates with the outlet used for each test configuration (Figure B-2). For evaluating performance with *exhaust inlets*, the apparatus shall be constructed such that the fan or blower draws air through the duct that contains the reference flowmeter and starts with the inlet used for each test configuration (Figure B-3).

The supply outlet or exhaust inlet being tested shall be mounted on a flat surface at least 0.3 m (1 ft) larger than the supply outlet or exhaust inlet. Each supply outlet or exhaust inlet shall be tested in this configuration using the duct diameter associated with the supply outlet or exhaust inlet. There shall be at least 10 duct diameters (10 D) between the reference flowmeter exit and the supply outlet or between the reference flowmeter entry and the exhaust inlet, as applicable.

1. The sizes of the fan, reference flowmeter, and associated ducts for the multibranch apparatus are larger than those for the single-branch apparatus to accommodate the sum of the flows through all three supply inlets or exhaust inlets, which have similar flows.

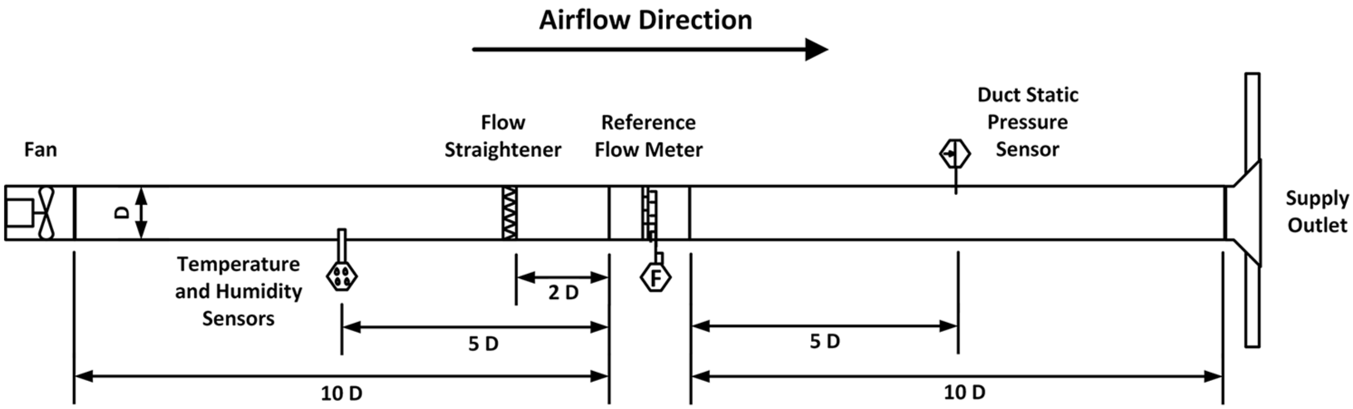


Figure B-2 Test apparatus for single-branch testing—supply outlet.

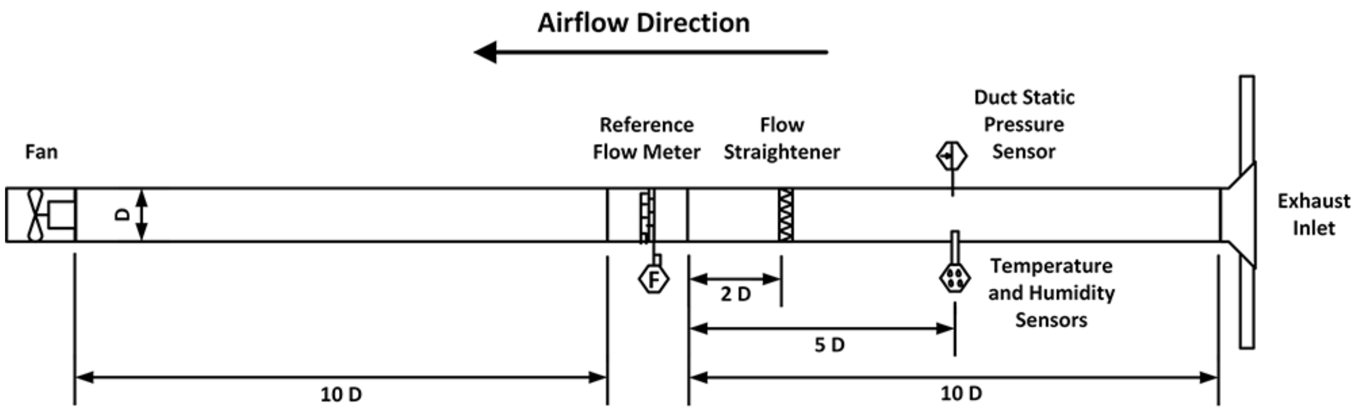


Figure B-3 Test apparatus for single-branch testing—exhaust inlet.

B2.3.2 Multibranch Apparatus. Figures B-4 and B-5 are illustrations of a typical multibranch test apparatus. The multibranch test apparatus shall be constructed such that the fan or blower moves air through the duct containing the reference flowmeter and through three equal length branch ducts.

The distance between the reference meter (entry when supply outlets are tested and exit when exhaust inlets are tested) and branch fitting entrance shall be at least 10 duct diameters (10 D). Each branch shall have a duct of the same diameter D_b, be at least 10 D long (10 D_b), and have the same supply outlet or exhaust inlet at its free end. Each supply outlet or exhaust inlet shall be mounted on a flat surface at least 0.3 m (1 ft) larger than the outlet or inlet. Supply outlets and exhaust inlets shall be tested in this configuration using the duct diameter D_b associated with the supply outlet or exhaust inlet. For multibranch testing, the air shall flow through the supply outlet or exhaust inlet being measured by the flow capture hood and also through the two other outlets or inlets that are not measured by the hood.

B2.4 Determining Test Apparatus Air Leakage. The test apparatus shall be tested for air leakage using the following procedure after initial construction and after any configuration change is made to the apparatus that could affect its airtightness, such as changing supply outlets or exhaust inlets or reference airflow meters.

B2.4.1 Seal all of the test apparatus supply outlets or exhaust inlets.

B2.4.2 A leakage test fan or blower and airflow measurement system meeting the following specifications shall be temporarily attached to the test apparatus immediately downstream of the reference flowmeter to pressurize the apparatus. The leakage test flowmeter shall measure leakage airflow with a precision of 3% or better with a confidence level of 95%. The leakage test fan shall be

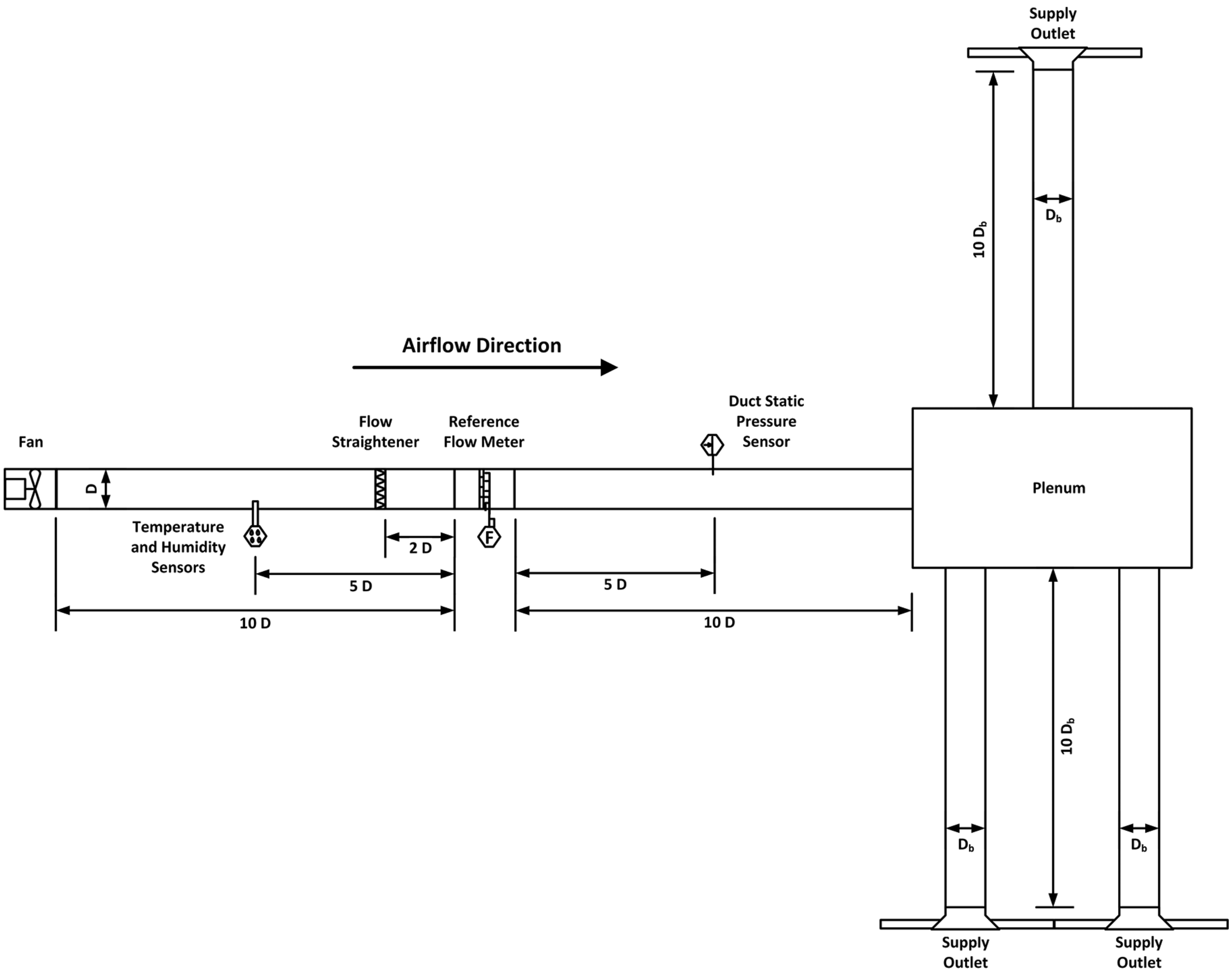


Figure B-4 Test apparatus for multibranch testing—supply outlets.

controlled using a flow control mechanism that can be adjusted to provide specified pressure differences inside the duct relative to its surroundings at the duct static pressure measurement location.

B2.4.3 The leakage test fan or blower shall be adjusted to maintain a positive 25 Pa (0.1 in. of water) pressure inside the test apparatus relative to its surroundings.

B2.4.4 The following shall be recorded: leakage airflow required to maintain the test pressure, temperature and relative humidity of air flowing through the test apparatus, and barometric pressure. All recorded flow, pressure, temperature, and humidity measurements shall be a one-minute average of points taken at least once every five seconds.

B2.4.5 Convert the measured leakage airflow to volumetric flow at standard air density (1.204 kg/m^3 [0.075 lb$_m$/ft^3]) using Equations 2 through 4 in Section 6 of this standard (or the flowmeter manufacturer's equations if they differ from Section 6) and the measured humidities, temperatures, and pressures.

B2.4.6 The maximum allowable leakage airflow shall be 0.5 L/s (1 cfm) at standard air density.

B2.4.7 If the air leakage is above the maximum allowed, then the apparatus shall be further air sealed and retested until the maximum allowable air leakage level in Section B2.4.6 is not exceeded.

B2.4.8 After leakage testing is complete, the supply outlet or exhaust inlet seals, as well as the leakage test fan or blower and its related airflow measurement system, shall be removed.

B2.5 Flow Capture Hood Test Procedure. For each single-branch and multibranch test of a flow capture hood, the test shall each be repeated for at least 30 airflow test points spaced equally over the measurement range of the hood.

For each airflow test point, the test shall be performed with the flow capture hood centered over the supply outlet or exhaust inlet. Additionally, if the entrance of the hood is larger than the supply outlet or exhaust inlet, then the tests shall be repeated for two relative locations of the supply outlet or exhaust inlet and hood: (a) with the supply outlet or exhaust inlet centered along one edge of the hood, and (b) with the supply outlet or exhaust inlet in one corner of the hood.

All recorded flow, pressure, humidity, and temperature measurements shall be a one-minute average of points taken at least once every five seconds.

B2.5.1 Single-Branch Testing. The single-branch test apparatus shall be used. The following procedure shall be followed for each supply outlet or exhaust inlet, airflow test point, and relative location of the supply outlet or exhaust inlet and flow capture hood.

B2.5.1.1 The flow capture hood shall be placed over the supply outlet or exhaust inlet.

B2.5.1.2 The test apparatus fan or blower shall be adjusted such that the airflow through the reference flowmeter is maintained within 1 L/s (2 cfm) of each test point at *standard* air density (as calculated in Section B2.6.1).

B2.5.1.3 The following shall be recorded: airflow through the reference flowmeter $Q_{ref,meas}$, airflow through the flow capture hood $Q_{hood,meas}$, temperature and humidity of air flowing through the test apparatus, static pressure inside the test apparatus relative to the apparatus surroundings P_{test}, and barometric pressure.

B2.5.2 Multi-Branch Testing. The multi-branch test apparatus shall be used. The following procedure shall be followed for each supply outlet or exhaust inlet, airflow test point, and relative location of supply outlet or exhaust inlet and flow capture hood.

B2.5.2.1 Without the flow capture hood in place, the fan or blower shall be adjusted such that the flow though the reference flowmeter is maintained within 1 L/s (2 cfm) of each test point at *standard* air density (as calculated in Section B2.6.1). Record the airflow through the reference flowmeter $Q_{ref,meas}$. The blower or fan shall be controlled to maintain the same airflow through the reference flowmeter when the hood is placed over the supply outlet or exhaust inlet.

B2.5.2.2 The hood shall be placed over the supply outlet or exhaust inlet.

B2.5.2.3 The following shall be recorded: airflow through the hood $Q_{hood,meas}$, temperature and humidity of air flowing through the test apparatus, static pressure inside the test apparatus relative to the apparatus surroundings P_{test}, and barometric pressure.

B2.6 Calculations. Analyses of the flow capture hood test data shall be performed separately for each set of tests, single-branch and multibranch.

B2.6.1 Convert the measured airflows $Q_{hood,meas}$ and $Q_{ref,meas}$ to volumetric flow $Q_{hood,std}$ and $Q_{ref,std}$ at standard air density (1.204 kg/m^3 [0.075 lb$_m$/ft^3]) using Equations 2 through 4 in Section 6 of this standard (or the manufacturer's equations if they differ from Section 6) and the measured humidities, temperatures, and pressures.

B2.6.2 The differences between the flow capture hood's standard airflows calculated in Section B2.6.1 relative to the reference standard airflows calculated in Section B2.6.1 shall be calculated for each combination *x* of apparatus, supply outlet or exhaust inlet, and position of the hood relative to the supply outlet or exhaust inlet. For the multibranch apparatus, the hood and reference airflows used in the following calculations shall each be the *sum* of the three supply outlet or exhaust inlet airflows and the *average* of the three reference airflows, respectively, all at standard air density.

$$Q_{diff,x,i} = (Q_{hood,std,x,i} - Q_{ref,std,x,i}) \tag{B-1}$$

where

$Q_{diff,x}$ = the difference between each i^{th} pair of hood and reference airflows for combination x of apparatus, supply outlet or exhaust inlet, and position of the hood relative to the supply outlet or exhaust inlet at standard air density, L/s (cfm)

The fractional differences in airflow $Q_{diff,x,i}/Q_{ref,std,x,i}$ as a function of the corresponding reference airflows $Q_{ref,std,x,i}$ for each combination of apparatus, supply outlet or exhaust inlet, and position of the flow capture hood relative to the supply outlet or exhaust inlet shall be plotted graphically. A polynomial curve shall be fitted to each set of data using statistical methods

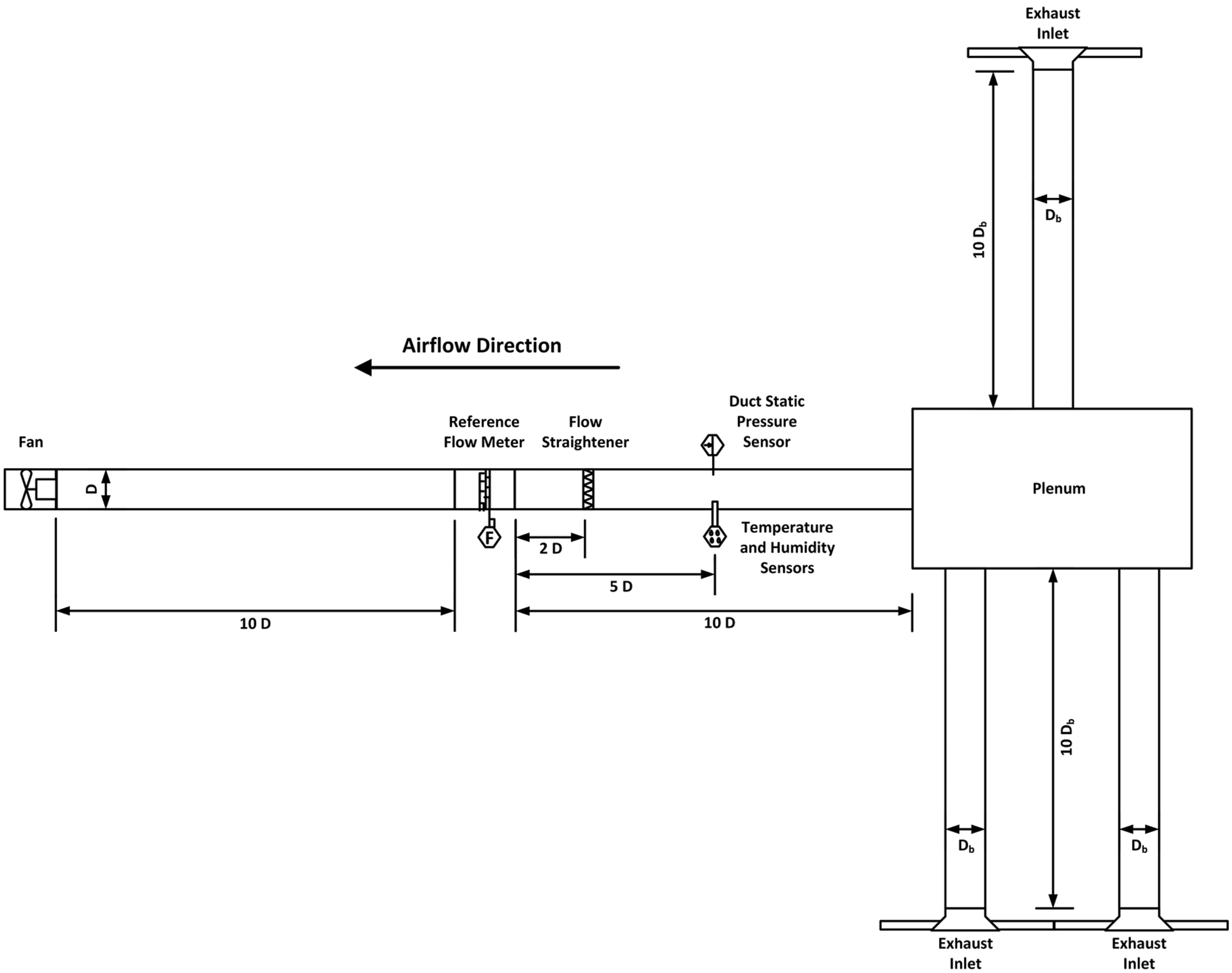


Figure B-5 Test apparatus for multibranch testing—exhaust inlets.

described in ASHRAE Guideline 2, Section 8, (ASHRAE 2014) to represent their relationship over the hood's operating range.

B2.6.3 If the polynomial fitted in Section B2.6.2 is a *linear function with zero slope*, the average airflow fractional bias error and the 95th percentile fractional precision error relative to the reference flow for combination *x* of apparatus, supply outlet or exhaust inlet, and position of the flow capture hood relative to the supply outlet or exhaust inlet at standard air density is *not* a function of the reading and shall be calculated as follows:

$$Q_{bias,\,x} = \frac{1}{N}\sum_{i=1}^{N}\left(\frac{Q_{diff,\,x,\,i}}{Q_{ref,\,std,x,i}}\right) \tag{B-2}$$

where

$Q_{bias,x}$ = average airflow fractional *bias* error for combination *x* of apparatus, supply outlet or exhaust inlet, and position of the hood relative to the supply outlet or exhaust inlet at standard air density, L/s (cfm).

N = number of airflow readings

$$Q_{prec,\,x} = 1.96\left[\frac{1}{N}\sum_{i=1}^{N}\left(\frac{Q_{diff,\,x,\,i}}{Q_{ref,\,std,x,\,i}}\right)^2\right]^{1/2} \tag{B-3}$$

where

$Q_{prec,x}$ = airflow fractional *precision* error for combination x of apparatus, supply outlet or exhaust inlet, and position of the hood relative to the supply outlet or exhaust inlet at standard air density, expanded to a 95% confidence level, L/s (cfm)

B2.6.4 If the polynomial fitted in Section B2.6.2 is *not* a linear function with zero slope, the bias and precision errors *depend on the reading*. In this case, use of the hood in field applications is not recommended. However, if it will be used in such applications, the bias and precision errors shall be determined using statistical methods described in ASHRAE Guideline 2, Annex A (ASHRAE 2014).

B2.7 Test Report. The test report shall contain the following information.

B2.7.1 The date, time, and location of the test.

B2.7.2 The make, model number, and calibration date for each of the measurement instruments other than the flow capture hood.

B2.7.3 A description of the test apparatus, including duct lengths and sizes, fan sizes, reference flowmeter characteristics, and the location of each measurement instrument.

B2.7.4 A list of the supply outlets or exhaust inlets used in the tests and the test airflows used for each supply outlet or exhaust inlet.

B2.7.5 The apparatus leakage airflows at standard air density.

B2.7.6 The make, model number, and serial number of the flow capture hood, along with the hood configuration during the tests (e.g., time averaging setting, diffuser screen use), as applicable.

B2.7.7 A list of the locations of the hood relative to the supply outlets or exhaust inlets used in the tests.

B2.7.8 Graphical plots of the fractional differences in airflow $Q_{diff,x,i}/Q_{ref,std,x,i}$ as a function of the corresponding reference airflows $Q_{ref,std,x,i}$ for each combination of apparatus, supply outlet or exhaust inlet, and position of the flow capture hood relative to the supply outlet or exhaust inlet, along with the polynomial curves fitted to the data to represent their relationship over the hood's operating range.

B2.7.9 A table of the test results $Q_{bias,x}$ and $Q_{prec,x}$ for each combination x of apparatus, supply outlet or exhaust inlet, and position of the flow capture hood relative to the supply outlet or exhaust inlet at standard air density.

(This appendix is not 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. Unresolved objectors on informative material are not offered the right to appeal at ASHRAE or ANSI.)

INFORMATIVE APPENDIX C
EXAMPLE CALCULATIONS INCLUDING UNCERTAINTY ANALYSIS

C1. UNCERTAINTY ANALYSIS BACKGROUND

For a single-duct supply system, as described in Section 6 of this standard, the system leakage flow estimate Q_{leak} is based on the following expression:

$$Q_{leak,\,std} = Q_{in,\,std} - \sum_{j=1}^{N'}\left(\sum_{i=1}^{n_j} Q_{out,\,i,\,std}\right) \tag{C-1}$$

where

$Q_{leak,std}$ = leakage airflow at standard air density, L/s (cfm)

$Q_{in,std}$ = supply inlet airflow at standard air density, L/s (cfm)

$Q_{out,i,std}$ = supply outlet i airflow at standard air density, L/s (cfm)

N' = number of flow capture hoods or measuring stations used to measure all $Q_{out,i,std}$ in test section

n_j = number of supply outlet airflows measured by flow capture hood j or measuring station j

The airflow measurements required to evaluate the system leakage are subject to bias and precision errors of the measurement equipment. To account for bias errors, each measured airflow in Equation C-1 is first adjusted to standard air density and then adjusted for bias errors (if any) at standard air density (Section 6, Equation 4).

To help understand the nature of bias errors and how they each represent an offset from the true or actual value, consider the following time analogy. If one travels from one time zone to another (e.g., from Chicago to Los Angeles), there is a difference in the time that is shown on one's watch compared to the actual time in the new time zone (two hours in this case, assuming the watch was correctly set in Chicago). So that the watch shows the correct time in the new time zone, one can either calibrate the watch by resetting its display to show the correct time (i.e., moving the hour display back two hours in this case), or the watch can remain as it was set in the first time zone and the user can subtract two hours from the watch reading each time it is read. The former approach is preferred to avoid the possible confusion caused by forgetting to subtract the time difference from the displayed time in the latter case.

Bias errors do not vary during repeated readings. They might be attributed to zero offset and calibration uncertainty, hysteresis, nonlinear behavior, limits on resolution, repeatability, sensor location error, and installation errors. Bias errors are independent of the sample size and are not amenable to statistical analysis. Although bias errors can be minimized by proper calibration of the measurement system, they are otherwise difficult to remove from the measurement process and must often be estimated by comparison of the instrument to a more accurate standard. It is assumed, however, that precision errors in determining bias errors combine in a root-sum-square manner, similar to precision errors, and are included in the overall precision specified by instrument manufacturers or determined by calibration.

To address precision errors, it is necessary to consider how the precision error for each measured value combines to produce the net error in the final result. When a quantity to be determined depends on more than one independent measurement, per ASHRAE Guideline 2 (ASHRAE 2014), the method of estimating experimental uncertainty is based on the chain rule of calculus. For example, consider a dependent variable u that is not measured directly and instead is a function of more than one independent variable x_i, such that

$$u = f(x_1, x_2, x_3, \ldots) \tag{C-2}$$

If allowance is made for the fact that precision errors tend to compensate for each other (being both positive and negative), and if all measurement precision errors are stated at the same confidence level, the root-sum-square precision error in the calculated parameter δu_r is estimated as follows:

$$\delta u_r = \pm\left[\sum_{i=1}^{n}\left(\frac{\partial f}{\partial x_i}\delta x_i\right)^2\right]^{1/2} \tag{C-3}$$

In this instance, δx_i is the precision error of each of the n measured independent variables.

Referring to Equation C-3, by replacing δx_i with the standard precision error for each airflow measurement plane, and evaluating derivatives (all equal to 1), the root-sum-square random uncertainty in the system leakage (as described in Section 6 of this standard) is given by

$$u_{Q_{leak,\,std}} = \sqrt{s^2_{Q_{in,\,std}} + \sum_{j=1}^{N'}\left[\left(\sum_{i=1}^{n_j} s_{Q_{out,\,i,\,std}}\right)/(\sqrt{n_j})\right]^2} \tag{C-4}$$

where

$u_{Q_{leak,\,std}}$ = overall standard uncertainty for $Q_{leak,std}$, L/s (cfm)

$s_{Q_{in,\,std}}$ = standard precision error at standard air density for measuring station used to measure $Q_{in,std}$,
L/s (cfm)

$s_{Q_{out,\,i,\,std}}$ = standard precision error at standard air density for flow capture hood or measuring station j used to measure $Q_{out,i,std}$, L/s (cfm)

Standard precision errors (one standard deviation, which corresponds approximately to a 68% confidence level) are used in this standard for consistency (each is expressed at a common confidence level) and for simplicity of reporting as recommended by the Joint Committee for Guides in Metrology (JCGM) (BIPM 2010). The overall standard uncertainty for $Q_{leak,std}$ that results from Equation C-4 could be subsequently expanded to other confidence levels (e.g., 95%), if desired, by multiplying by the coverage factor listed in Table 1 of this standard. The use of expanded uncertainty, however, is not recommended.

The term n_j included in Equation C-4 accounts for the compensating effects of positive and negative precision errors in measurements at multiple grilles or diffusers (as opposed to compensating errors for a single grille or diffuser). In particular, the precision error at each grille/diffuser, which can be assumed to be normally distributed, will be randomly dispersed about the central mean value that is measured. Sometimes, the error will be positive and sometimes it will be zero or negative. When airflow measurements are made at several grilles/diffusers, Monte Carlo statistical analyses can be used to show that the errors will tend to reduce the overall precision error of the sum of the grille/diffuser flows, with the reduction approximating $1/(\sqrt{n_j})$.

Due to this compensating effect, if the inlet and outlet flow measurements have similar precision errors, the inlet flow precision error will dominate the overall standard uncertainty for $Q_{leak,std}$. Therefore, it is imperative to measure the inlet flow of a supply system or the outlet flow of an independent exhaust system with the greatest possible precision to minimize the uncertainty when determining the leakage flow and leakage fraction.

C2. EXAMPLE CALCULATION—NO BIAS ERRORS

As a specific example, consider a variable-air-volume (VAV) system where the supply airflow is measured as 4195 L/s (8889 cfm) with a precision error of 3% at a 95% confidence level or 1 L/s (2 cfm), whichever is greater. There are a total of 18 supply diffusers in the system: the airflow from each of the first ten diffusers is measured as 189 L/s (400 cfm), and the airflow from each of the other eight diffusers is measured as 236 L/s (500 cfm). The diffuser airflows are measured using one flow capture hood with a precision error of 3% at a 95% confidence level or 1 L/s (2 cfm), whichever is greater.

At the same elevation as the test section, the barometric pressure is 98,100 Pa (14.228 psi). At the inlet airflow measurement plane, the static pressure is 2490 Pa (10.0 in. of water), the measured air dry-bulb temperature is 30.0ºC (86.0ºF), and the relative humidity is 50%. At the diffusers, the measured air dry-bulb temperature and relative humidity are the same as at the inlet plane. The static pressure at each diffuser discharge is zero. For simplicity in this example, the precision errors associated with these environmental measurements are ignored.

It is assumed in this example that all instruments have been calibrated or adjusted such that the readings represent the true values (i.e., there are no bias errors that need to be accounted for in subsequent calculations).

Converting the measured air temperature to absolute temperature,

$$T_{in} = T_{out,i} = 30.0 + 273.15 = 303.15 \text{ K} \qquad \text{(SI C-5)}$$

$$T_{in} = T_{out,i} = 86.0 + 459.67 = 545.67°\text{R} \qquad \text{(I-P C-5)}$$

The actual air density depends in part on the saturation pressure ($p_{ws,in} = p_{ws,out,i}$), which is calculated using Equation 3a:

$$p_{ws,\,in} = e^{\left\{\begin{array}{c}(-5.8002206\text{E+03}/303.15) + (1.3914993) - \\ (4.8640239\text{E-02} \times 303.15) + (4.1764768\text{E-05} \times 303.15^2) - \\ (1.4452093\text{E-08} \times 303.15^3) + [6.5459673 \times \ln(303.15)]\end{array}\right\}} = 4246 \text{ Pa} \qquad \text{(SI C-6)}$$

$$p_{ws,\,in} = e^{\left[\begin{array}{c}(-1.0440397\text{E+04}/545.67) - (11.29465) - \\ (2.7022355\text{E-02} \times 545.67) + (1.289036\text{E-05} \times 545.67^2) - \\ (2.4780681\text{E-09} \times 545.67^3) + (6.5459673 \times \ln(545.67))\end{array}\right]} = 0.616 \text{ psi} \qquad \text{(I-P C-6)}$$

Using Equation 2 and the saturation pressure from Equation C-6, the actual air density at the *inlet* plane is

$$\rho_{in} = \left[\frac{98{,}100 + 2490 - 0.378\,(50/100) \times 4246}{287.042 \times 303.15}\right] = 1.147 \text{ kg/m}^3 \qquad \text{(SI C-7)}$$

$$\rho_{in} = \left\{\frac{144[14.228 + 0.036 \times 10.0 - 0.378\,(50/100) \times 0.616]}{53.350 \times 545.67}\right\} = 0.0716 \text{ lb}_m/\text{ft}^3 \qquad \text{(I-P C-7)}$$

Again using Equation 2, the actual air density at the *outlet* planes (diffusers) is

$$\rho_{out,\,i} = \left[\frac{98{,}100 + 0 - 0.378\,(50/100) \times 4246}{287.042 \times 303.15}\right] = 1.118 \text{ kg/m}^3 \qquad \text{(SI C-8)}$$

$$\rho_{out,\,i} = \left\{\frac{144[14.228 + 0.036 \times 0 - 0.378\,(50/100) \times 0.616]}{53.350 \times 545.67}\right\} = 0.0698 \text{ lb}_m/\text{ft}^3 \qquad \text{(I-P C-8)}$$

The measured airflows can now be converted to standard airflows to facilitate comparing them. Using Equation 4 and the air density from Equation C-7, the *inlet* standard airflow (with zero bias errors) is

$$Q_{in,\,std} = \left[\left(\frac{1.147}{1.204}\right)4195\right](1 - 0/100) - 0 = 3996 \text{ L/s} \qquad \text{(SI C-9)}$$

$$Q_{in,\,std} = \left[\left(\frac{0.0716}{0.075}\right)8889\right](1 - 0/100) - 0 = 8484 \text{ cfm} \qquad \text{(I-P C-9)}$$

Similarly, using Equation 4 and the air density from Equation C-8, the *outlet* standard airflows are 176 L/s (372 cfm) for ten diffusers and 219 L/s (465 cfm) for eight diffusers. The sum of the 18 outlet flows is thus (10 × 176) + (8 × 219) = 3509 L/s, or (10 × 372) + (8 × 465) = 7446 cfm.

Using Equation 5a, the leakage airflow at standard air density is thus

$$Q_{leak,std} = 3996 - 3509 = 487 \text{ L/s} \qquad \text{(SI C-10)}$$

$$Q_{leak,std} = 8484 - 7446 = 1038 \text{ cfm} \qquad \text{(I-P C-10)}$$

From Equation 6a, the fractional leakage at standard air density is

$$L_{f,std} = 100(487/3996) = \mathbf{12.2\%} \qquad \text{(SI C-11)}$$

$$L_{f,std} = 100\,(1038/8484) = \mathbf{12.2\%} \qquad \text{(I-P C-11)}$$

To estimate the overall standard uncertainty in the leakage flow, one must first convert the airflow measuring instrument precision errors expressed at the 95% confidence level to standard precision errors (one standard deviation, which corresponds approximately to a 68% confidence level). Using Equation 7a, the stated fractional (0.03) and absolute (1 L/s [2 cfm]) precision errors, the *inlet* standard airflow from Equation C-9, and a coverage factor of 1.960 from Table 1, the standard precision error for the *inlet* airflow measurement is

$$s_{Q_{in,\,std}} = \frac{\max[(P_{Q_{in,\,std}\_frac}Q_{in,\,std})/100,\, P_{Q_{in,\,std}\_abs}]}{k_p} = \frac{\max[(0.03 \times 3996),\, 1]}{1.960} = 61 \text{ L/s} \qquad \text{(SI C-12)}$$

$$s_{Q_{in,\,std}} = \frac{\max[(P_{Q_{in,\,std}\_frac}Q_{in,\,std})/100,\, P_{Q_{in,\,std}\_abs}]}{k_p} = \frac{\max[(0.03 \times 8484),\, 2]}{1.960} = 130 \text{ cfm} \qquad \text{(I-P C-12)}$$

Similarly, the standard precision errors for the *outlet* airflow measurements are 2.7 and 3.4 L/s (5.7 and 7.1 cfm).

The overall standard uncertainty for the leakage flow as expressed by Equation C-5 (Equation 9a) is thus

$$u_{Q_{leak,\,std}} = \sqrt{61^2 + [(10 \times 2.7 + 8 \times 3.4)/(\sqrt{18})]^2} = \sqrt{61^2 + 12.7^2} = \sqrt{3721 + 160} = 62 \text{ L/s} \qquad \text{(SI C-13)}$$

$$u_{Q_{leak,\,std}} = \sqrt{130^2 + [(10 \times 5.7 + 8 \times 7.1)/(\sqrt{18})]^2} = \sqrt{130^2 + 26.9^2} = \sqrt{16{,}900 + 721} = 133 \text{ cfm} \qquad \text{(I-P C-13)}$$

Using Equation 10a, the overall standard uncertainty in the fractional leakage is

$$u_{L_{f,\,std}} = 100 \times (62/3996) = \mathbf{1.6\%} \qquad \text{(SI C-14)}$$

$$u_{L_{f,\,std}} = 100 \times (133/8484) = \mathbf{1.6\%} \qquad \text{(I-P C-14)}$$

In summary, the inlet airflow supplied by the fan at standard air density is 3996 ± 61 L/s (8484 ± 130 cfm), and the total airflow out through the diffusers at standard air density is 3509 ± 13 L/s (7446 ± 27 cfm). As a result, the leakage airflow and its overall standard uncertainty at standard air density is expressed as 487 ± 62 L/s (1038 ± 133 cfm). The corresponding *leakage fraction* at standard air density in this case is **12.2 ± 1.6%**, which represents a range of 10.6% to 13.8% leakage.

C3. EXAMPLE CALCULATION—INCLUDING AIRFLOW BIAS ERRORS

Consider again Section C2 but now including exemplar bias errors for the airflow measurements. Specifically, assume that the bias error for inlet airflow measurements is +4.0% and the bias errors for outlet airflow measurements are –5.0%, with all of the bias errors expressed as a function of airflows at standard air density. With these bias errors, the measured airflows in this case would be increased to 4370 L/s (9259 cfm) for the inlet flow, and decreased to 180 L/s (381 cfm) for the first ten diffusers and 225 L/s (476 cfm) for the other eight diffusers. Informative Appendix E provides the rationale for translating between the true (unbiased) airflows in Section C2 and the biased airflows described in this example.

As in Section C2, the measured airflows can be converted to standard airflows to facilitate comparing them. Again using Equation 4, the *inlet* standard airflow (with the bias errors converted to fractional values) is

$$Q_{in,\,std} = \left[\left(\frac{1.147}{1.204}\right)4370\right](1 - 4.0/100) - 0 = 3996 \text{ L/s} \qquad \text{(SI C-15)}$$

$$Q_{in,\,std} = \left[\left(\frac{0.0716}{0.075}\right)9259\right](1 - 4.0/100) - 0 = 8484 \text{ cfm} \qquad \text{(I-P C-15)}$$

Similarly, for the first ten *outlet* airflows (i = 1 to 10),

$$Q_{out,\,1,\,std} = \left[\left(\frac{1.118}{1.204}\right)180\right](1 - -5.0/100) - 0 = 176 \text{ L/s} \qquad \text{(SI C-16)}$$

$$Q_{out,\,1,\,std} = \left[\left(\frac{0.0698}{0.075}\right)381\right](1 - -5.0/100) - 0 = 372 \text{ cfm} \qquad \text{(I-P C-16)}$$

And, for the eight other *outlet* airflows (i = 9 to 18),

$$Q_{out,\,9,\,std} = \left[\left(\frac{1.118}{1.204}\right)225\right](1 - -5.0/100) - 0 = 219 \text{ L/s} \qquad \text{(SI C-17)}$$

$$Q_{out,\,9,\,std} = \left[\left(\frac{0.0698}{0.075}\right)476\right](1 - -5.0/100) - 0 = 465 \text{ cfm} \qquad \text{(I-P C-17)}$$

With the offsets due to the bias errors accounted for, the airflows at standard air density in Sections C2 and C3 are identical, as expected. Thus, the bias errors have no impact on the calculated leakage flow, leakage fraction, or their overall standard uncertainties at standard air density.

C4. UNCERTAINTY DUE TO CONVERSION TO STANDARD AIR

As noted previously, airflow measurements at every location in the duct system are subject to bias and precision errors. The calculation of actual air densities at those locations is also subject to bias and precision errors. In particular, the moist air density at any test section location is determined based on measurements of barometric pressure p_b, dry-bulb temperature $t_{db,x}$, relative humidity φ_x, and duct static pressure $p_{s.x}$, each of which may be subject to bias and precision errors. Bias errors can be removed by subtracting them as offsets from the measured values. Precision errors, however, must be addressed statistically.

Recall that the saturation pressure over liquid water at the measured dry-bulb temperature at any test location is calculated per Equation 3, where the constants C_n are provided in Section 6.4.2. For simplicity, consider for now only the SI version of Equation 3:

$$p_{ws,\,x} = e^{(C_1/T_x + C_2 + C_3T_x + C_4T_x^2 + C_5T_x^3 + C_6\ln T_x)} \qquad \text{(SI C-18)}$$

In this case, consider the following partial derivative; such partial derivatives are sometimes referred to as a sensitivity coefficient:

$$\frac{\partial p_{ws,\,x}}{\partial T_x} = e^{[(C_1/T_x + C_2 + C_3T_x + C_4T_x^2 + C_5T_x^3 + C_6\ln T_x)]} \times\left(-\frac{C_1}{T_x^2} + C_3 + 2C_4T_x + 3C_5T_x^2 + \frac{C_6}{T_x}\right) \qquad \text{(SI C-19)}$$

Therefore, the uncertainty in the saturation pressure with respect to the dry-bulb temperature is expressed as

$$\delta p_{ws,\,x} = \left\{\left(\frac{\partial p_{ws,\,x}}{\partial T_x}\delta T_x\right)^2\right\}^{1/2} = \left|\frac{\partial p_{ws,\,x}}{\partial T_x}\delta T_x\right| \qquad \text{(C-20)}$$

where δT_x is the precision error of the dry-bulb (absolute) temperature. The air density at any test section location is calculated by Equation 2:

$$\rho_x = \frac{p_b + p_{s,\,x} - 0.378(\varphi_x/100)p_{ws,\,x}}{R_{da}T_x} \qquad \text{(SI C-21)}$$

This implies the following partial derivatives:

$$\frac{\partial \rho_x}{\partial p_b} = \frac{1}{R_{da} T_x} \tag{SI C-22}$$

$$\frac{\partial \rho_x}{\partial p_{s,x}} = \frac{1}{R_{da} T_x} \tag{SI C-23}$$

$$\frac{\partial \rho_x}{\partial \varphi_x} = \frac{0.378 p_{ws,x}}{100 \times R_{da} T_x} \tag{SI C-24}$$

$$\frac{\partial \rho_x}{\partial p_{ws,x}} = \frac{0.378 \varphi_x}{100 \times R_{da} T_x} \tag{SI C-25}$$

$$\frac{\partial \rho_x}{\partial T_x} = -\frac{p_b + p_{s,x} - 0.378(\varphi_x/100) p_{ws,x}}{R_{da} T_x^2} \tag{SI C-26}$$

Hence, the uncertainty in the air density at any plane x within the duct system with respect to the measured/calculated variables is given by

$$\delta\rho_x = \begin{bmatrix} \left(\dfrac{\partial \rho_x}{\partial p_b}\delta p_b\right)^2 + \left(\dfrac{\partial \rho_x}{\partial p_{s,x}}\delta p_{s,x}\right)^2 + \left(\dfrac{\partial \rho_x}{\partial \varphi_x}\delta \varphi_x\right)^2 \\ + \left(\dfrac{\partial \rho_x}{\partial p_{ws,x}}\delta p_{ws,x}\right)^2 + \left(\dfrac{\partial \rho_x}{\partial T_x}\delta T_x\right)^2 \end{bmatrix}^{1/2} \tag{C-27}$$

In this instance, the quantity δp_b represents the precision error associated with the barometric pressure measured at the same elevation as the test section. Moreover, $\delta p_{s,x}$, $\delta\phi_x$, and δT_x refer to the precision errors associated with the static pressure, relative humidity, and temperature, respectively, at the test section location. Uncertainty in the saturation pressure over water $\delta\rho_{ws,x}$ is determined by Equation C-19.

The measured airflow determined at any plane x is converted to standard air density per Equation 4:

$$Q_{x,std} = \left[\left(\frac{\rho_x}{\rho_{std}}\right) Q_x\right](1 - B_{Q_{x,std\_frac}}/100) - B_{Q_{x,std-abs}} \tag{C-28}$$

The defined density of standard air ρ_{std} is 1.204 kg/m^3 (0.075 lb$_m$/ft^3). It is understood that this quantity is a constant and therefore not subject to measurement uncertainty. In this instance, assuming that the bias error components in Equation C-28 are also constant, the following partial derivatives are obtained:

$$\frac{\partial Q_{x,std}}{\partial \rho_x} = \frac{Q_x}{\rho_{std}}(1 - B_{Q_{x,std\_frac}}/100) \tag{C-29}$$

$$\frac{\partial Q_{x,std}}{\partial Q_x} = \frac{\rho_x}{\rho_{std}}(1 - B_{Q_{x,std\_frac}}/100) \tag{C-30}$$

Hence, by Equation C-3, the uncertainty in the standard airflow at any plane x within the duct system is given by

$$\delta Q_{x,std} = \left[\left(\frac{\partial Q_{x,std}}{\partial \rho_x}\delta\rho_x\right)^2 + \left(\frac{\partial Q_{x,std}}{\partial Q_x}\delta Q_x\right)^2\right]^{1/2} \tag{C-31}$$

where δQ_x represents the precision error expressed at standard air density that is associated with the measured airflow at plane x converted to standard air density.

Equation C-31 can be simplified algebraically such that the uncertainty in the standard airflow at any measurement location is expressed as

$$\delta Q_{x,std} = \frac{(1 - B_{Q_{x,std\_frac}}/100)}{\rho_{std}} \times [(Q_x \delta\rho_x)^2 + (\rho_x \delta Q_{x,std})^2]^{1/2} \tag{C-32}$$

Table C-1 Standard Airflow Sensitivity Coefficients and Uncertainty Values

$\frac{\partial p_{ws,x}}{\partial T_x}$ Pa/K (psi/°R)	2.4374E+02 (1.9640E-02)	$\delta p_{ws,x}$ Pa (psi)	243.7 (0.0354)
$\frac{\partial \rho_x}{\partial p_b}$ kg/m^3·Pa (lb$_m$/ft^3·psi)	1.1492E-05 (4.9465E-03)	$\delta \rho_x$ kg/m^3 (lbm/ft^3)	0.007 (0.0004)
$\frac{\partial \rho_x}{\partial p_{s,x}}$ kg/m^3·Pa (lb$_m$/ft^3·in. of water)	1.1492E-05 (1.7807E-04)		
$\frac{\partial \rho_x}{\partial \varphi_x}$ kg/m^3 (lb$_m$/ft^3)	–1.8445E-04 (–1.1515E-05)		
$\frac{\partial \rho_x}{\partial p_{ws,x}}$ kg/m^3·Pa (lb$_m$/ft^3·psi)	–2.1720E-06 (–3.1277E-04)		
$\frac{\partial \rho_x}{\partial T_x}$ kg/m^3·K (lb$_m$/ft^3·°R)	–3.7828E-03 (–1.3119E-04)		
$\frac{\partial Q_{x,std}}{\partial \rho_x}$ [(m^3·L)/(s·kg)] (ft^3·cfm/lb$_m$)	3.4843E+03 (1.1826E+05)	$\delta Q_{x,std}$ L/s (cfm)	116.8 (247.4)
$\frac{\partial Q_{x,std}}{\partial Q_x}$ [(m^3·L)/(s·kg)] (ft^3·cfm/lb$_m$)	9.5246E-01 (9.5241E-01)		

Consider again Section C2 (without airflow measurement bias errors). In this case, *converting* the measured airflow from actual air density (4195 L/s [8889 cfm]) to standard air density (3996 L/s [8466 cfm]) creates a –4.8% difference in flow and should not be ignored.

To understand the impacts of pressure, temperature, and relative humidity measurement precision errors on overall uncertainty if they were included in the analysis, assume that the precision errors at a 95% confidence level are 500 Pa (0.0725 psi) for barometric pressure, 49.8 Pa (0.2 in. of water) for static pressure, 1.0°C (1.8°F) for temperature, and 7% for relative humidity (based on the specifications in Section 4 of this standard). The corresponding sensitivity coefficients and uncertainty values are presented in Table C-1.

Recall that in Section C2 the precision error for the airflow measurement at a 95% confidence level is specified as 3.0% of the airflow at standard air density, which in this case corresponds to 119.9 L/s (254.0 cfm). However, including the precision errors for pressure, temperature, and relative humidity measurements results in an airflow measurement uncertainty of 116.8 L/s (247.4 cfm). The difference compared to the 3.0% precision of the airflow measurement is only –3.1 L/s (–6.6 cfm) or a 0.08 percentage point decrease. As a result, when the instrument precision specifications for pressure, temperature, and relative humidity measurements in Section 4 are not exceeded, the *uncertainty contributions* of the conversion to standard airflow can usually be ignored in the uncertainty analyses for leakage flow and leakage fraction.

(This appendix is not 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. Unresolved objectors on informative material are not offered the right to appeal at ASHRAE or ANSI.)

INFORMATIVE APPENDIX D
DERIVATION OF MOIST AIR DENSITY EQUATION

By the definition of "moist air density," for a mixture of dry air and water vapor,

$$\rho = \frac{m_{da} + m_w}{V} = \frac{m_{da}}{V}\left(1 + \frac{m_w}{m_{da}}\right) \tag{D-1}$$

where

ρ = moist air density, kg/m^3 (lb$_m$/ft^3)

m_{da} = mass of dry air, kg (lb$_m$)

m_w = mass of water vapor, kg (lb$_m$)

V = volume, m^3 (ft^3)

The humidity ratio is the ratio of the mass of water vapor to the mass of dry air:

$$\omega = \frac{m_w}{m_{da}} \tag{D-2}$$

where

ω = humidity ratio, dimensionless [kg$_w$/kg$_{da}$; (lb$_w$/lb$_{da}$)]

Thus, combining Equations D-1 and D-2, the moist air density is

$$\rho = \frac{m_{da}}{V}(1 + \omega) \tag{D-3}$$

The density of a mixture of dry air and water vapor is determined using the ideal gas law. Dalton's law of partial pressures states that the total pressure is equal to the sum of the partial pressures of the dry air and water vapor at the absolute temperature T and volume V of the mixture. Hence,

$$\frac{m_{da}}{V} = \frac{p_{da}}{R_{da}T} = \frac{p - p_w}{R_{da}T} \tag{SI D-4}$$

$$\frac{m_{da}}{V} = \frac{144 p_{da}}{R_{da}T} = \frac{p - p_w}{R_{da}T} \tag{I-P D-4}$$

where

p = thermodynamic pressure, Pa (psi)

p_{da} = partial pressure of dry air, Pa (psi)

p_w = partial pressure of water vapor at t_{db}, Pa (psi)

R_{da} = gas constant of dry air, 287.042 J/(kg·K); [53.350 ft·lb$_f$/(lb$_m$·°R)]

T = absolute temperature, K = t_{db} + 273.15 (°R = t_{db} + 459.67)

t_{db} = measured air temperature, °C (°F)

Therefore, combining Equations D-3 and D-4:

$$\rho = \frac{p - p_w}{R_{da}T}(1 + \omega) \tag{SI D-5}$$

$$\rho = 144\left(\frac{p - p_w}{R_{da}T}\right)(1 + \omega) \tag{I-P D-5}$$

For a mixture of dry and water vapor, the humidity ratio is expressed as

$$\omega = 0.622\frac{p_w}{p - p_w} \tag{D-6}$$

Substituting Equation D-6 into Equation D-5 and simplifying,

$$\rho = \frac{p - p_w}{R_{da}T}\left(1 + 0.622\frac{p_w}{p - p_w}\right) = \frac{1}{R_{da}T}(p - p_w + 0.622p_w) \tag{SI D-7}$$

$$\rho = \frac{144(p - p_w)}{R_{da}T}\left(1 + 0.622\frac{p_w}{p - p_w}\right) = \frac{144}{R_{da}T}(p - p_w + 0.622p_w) \tag{SI D-7}$$

The water vapor partial pressure is expressed in terms of the relative humidity and saturation pressure as follows:

$$p_w = (\varphi/100)p_{ws} \tag{D-8}$$

where

p_{ws} = saturation pressure, Pa (psi)

Therefore, combining Equations D-7 and D-8, the moist air density is given by

$$\rho = \frac{1}{R_{da}T}[p - 0.378(\varphi/100)p_{ws}] \tag{SI D-9}$$

$$\rho = \frac{144}{R_{da}T}[p - 0.378(\varphi/100)p_{ws}] \tag{I-P D-9}$$

In terms of the static pressure within the duct at any measurement plane and the barometric pressure at the same elevation as the test section, Equation D-9 can be expressed as

$$\rho_x = \frac{p_b + p_{s,x} - 0.378(\varphi_x/100)p_{ws,x}}{R_{da}T_x} \tag{SI D-10}$$

$$\rho_x = \frac{144[p_b + 0.036p_{s,x} - 0.378(\varphi_x/100)p_{ws,x}]}{R_{da}T_x} \tag{I-P D-10}$$

where

ρ_x = air density at plane x, kg/m^3 (lb$_m$/ft^3)

p_b = measured barometric pressure at same elevation as airflow measurement, Pa (psi)

$p_{s,x}$ = measured duct static pressure at plane x, Pa (in. of water)

$p_{ws,x}$ = saturation pressure at plane x, Pa (psi)

φ_x = measured relative humidity at plane x, %

R_{da} = gas constant of dry air, 287.042 J/(kg·K); [53.350 ft·lb$_f$/(lb$_m$·°R)]

T_x = absolute temperature at plane x, K= $t_{db,x}$ + 273.15; (°R = $t_{db,x}$ + 459.67)

$t_{db,x}$ = measured air temperature, °C (°F)

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INFORMATIVE APPENDIX E
EFFECT OF BIAS ERRORS ON MEASURED AIRFLOW

Equation 4 of this standard determines the standard airflow $Q_{x,std}$ at *standard air density* ρ_{std} by adjusting the measured airflow Q_x to account for air density at measurement conditions ρ_x and for bias errors $B_{Q_{x,std}\_frac}$ and $B_{Q_{x,std}\_abs}$, which are stated at standard air density:

$$Q_{x,std} = \left[\left(\frac{\rho_x}{\rho_{std}}\right)Q_x\right](1 - B_{Q_{x,std}\_frac}/100) - (B_{Q_{x,std}\_abs}) \tag{E-1}$$

The following determines, instead, the airflow that would be expected at measurement conditions when there are bias errors Q_x if the true airflow (zero bias error) at *measurement conditions* and the bias errors at standard air density are already known.[2]

Let $Q_{x,0}$ be the true airflow (no bias errors) at measurement conditions. From Equation E-1,

$$Q_{x,std} = \left[\left(\frac{\rho_x}{\rho_{std}}\right)Q_{x,0}\right] \tag{E-2}$$

Because the resulting standard airflow $Q_{x,std}$ is the same with and without bias errors, one can equate Equations E-1 and E-2:

$$\left[\left(\frac{\rho_x}{\rho_{std}}\right)Q_{x,0}\right] = \left[\left(\frac{\rho_x}{\rho_{std}}\right)Q_x\right](1 - B_{Q_{x,std}\_frac}/100) - B_{Q_{x,std}\_abs} \tag{E-3}$$

Rearranging Equation E-3 to solve for Q_x,

$$Q_x = \left[\frac{Q_{x,0} + \left(\frac{\rho_{std}}{\rho_x}\right)B_{Q_{x,std}\_abs}}{1 - B_{Q_{x,std}\_frac}/100}\right] \tag{E-4}$$

2. Usually, in a field application, the true airflow will not be known. However, this scenario is provided to support the example calculation that includes bias errors in Informative Appendix C, Section C3.

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INFORMATIVE APPENDIX F
EXAMPLE TEST PLAN AND TEST REPORT

F1. EXAMPLE TEST PLAN

F1.1 Scope. ASHRAE Standard 215 shall be used to determine the leakage airflow $Q_{leak,std}$, fractional leakage $L_{f,std}$, and uncertainties $u_{Q_{leak,std}}$ and $u_{L_{f,std}}$ for the system identified in this test plan.
Project Name: 2016-298-Georgia Business, Inc.
Project Location: Big City, GA
System Identification or Name: AHU-01
System Description: Single-duct variable-air-volume (VAV) supply air system with electric reheat coils and direct digital controls (DDC), which serves the first floor office space. The air distribution system schematic is shown in Figure F-1.

F1.2 Required Test Instrumentation. Instrumentation for the test shall conform to the following requirements (refer to Figure F-1):

a. Measurement instrumentation meeting the precision and calibration specifications in ASHRAE Standard 215, Section 4, shall be used for the test procedure. Measurements shall be performed in accordance with the instrumentation manufacturer's published instructions.
b. A thermal dispersion type of airflow measurement station (AFMS) shall be used to measure the inlet airflow. The AFMS shall be located at PL2 to avoid the impacts of system features or operating conditions that would degrade the accuracy of the measurements.
c. A powered flow capture hood shall be used for measuring the airflow at the diffusers served by the terminal units (1 through 7). The same flow capture hood shall be used for all of the diffusers (A through H).

F1.3 Requirements for Establishing Reference Operating Conditions. The reference operating condition shall be established according to the applicable specifications given in ASHRAE Standard 215, Section 5, for a single-duct VAV system. The reference operating condition shall also conform to the following requirements:

a. The system shall be tested at the airflow corresponding to full cooling (the system has no diversity) of 1888 L/s (4000 cfm) at standard air density (1.204 kg/m^3 [0.075 lb$_m$/ft^3]).
b. The system shall be tested at the design operating static pressure of 224 Pa (0.90 in. of water).

F1.4 Requirements for Measurement and Recording of Test Data. Measurement and recording of the test data shall be in accordance with the applicable requirements of ASHRAE Standard 215, Section 6.

a. During the data collection phase of the test procedure, the system inlet airflow Q_{in} shall not deviate more than 2% from the established reference operating condition determined by the specifications in this test plan.
b. The system inlet airflow Q_{in} shall be recorded at the start and at the end of the data collection phase of the leakage test. The percent difference between these two values shall be less than or equal to the maximum allowable deviation specified in Section F1.4(b).

F1.5 Requirements for Calculations. Data collected from the test shall be used to calculate the leakage airflow $Q_{leak,std}$, fractional leakage $L_{f,std}$, and uncertainties $u_{Q_{leak,std}}$ and $u_{L_{f,std}}$ for the system identified in this test plan using the equations in ASHRAE Standard 215, Section 6.4.

F1.6 Requirements for Test Report

a. The test report shall comply with all requirements given in ASHRAE Standard 215, Section 7.
b. The test report shall be signed by the technicians who conducted the test to certify that the information in the test report is true and accurate.
c. Data from the test procedure and the calculated results shall be presented in tabular format.
d. The test format shall be 8-1/2 × 11 in. page size made available as a PDF format file.

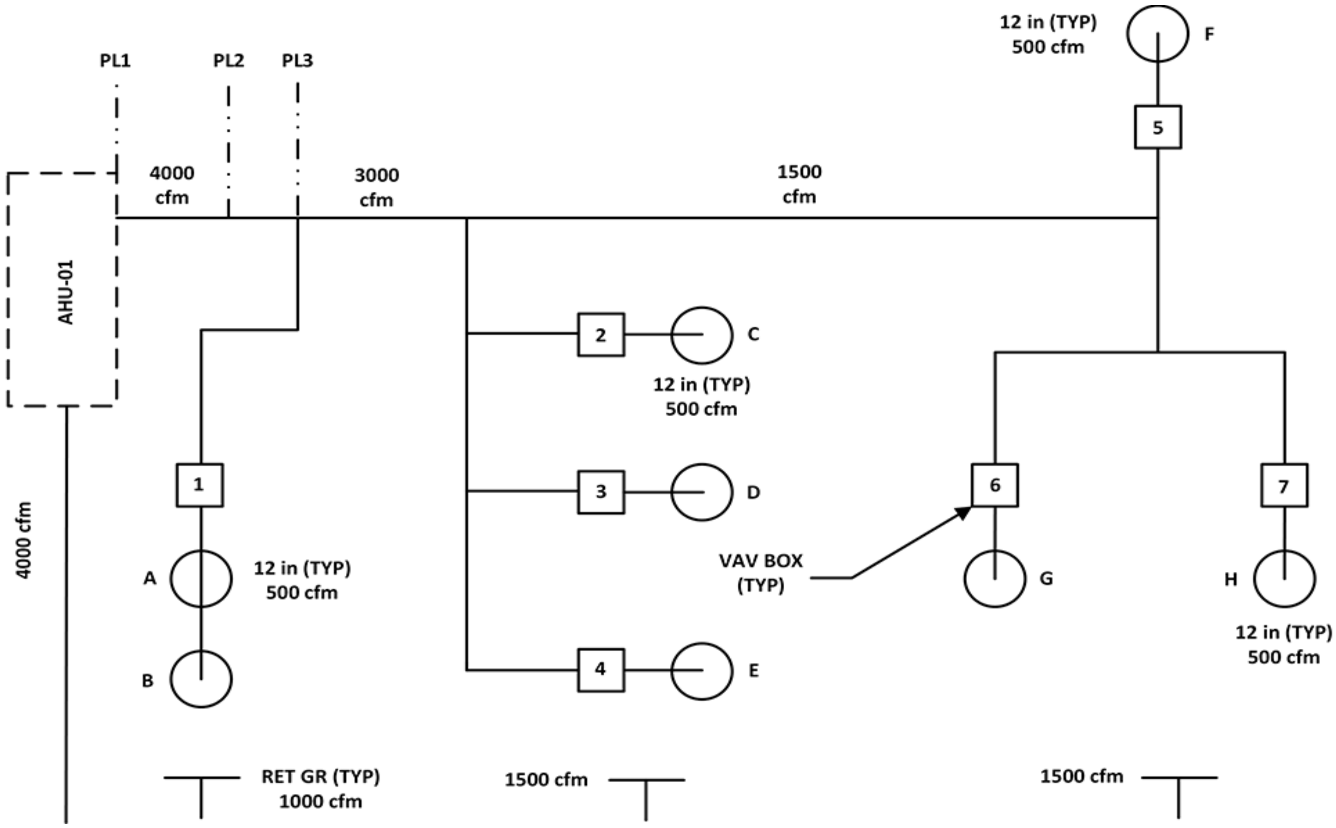

Figure F-1 AHU-01 air distribution system layout.

F2. EXAMPLE TEST REPORT

A copy of the test plan specified by the Engineer of Record for this system is attached to this report as an appendix (in this example, Informative Appendix F, Section F1).

Leakage Measurement Test Performed According to ASHRAE Standard 215

Table F-1 General Test Information

Date of test	June 26, 2017
Address of test	5328 Main Street, Big City, GA
System identification or name	AHU-01
System location or area served	First floor, office space
Air-handling unit (AHU) manufacturer's model number	CXSHW35HG200009
AHU manufacturer's serial number	30758372198
Name of organization that performed test	Air System Testing, Inc.
Field technician name	Tech Name 01
Field technician name	Tech Name 02

Table F-2 Instrumentation Used to Perform the Test Measurements

Instrument 01	Airflow measuring station
Type	Thermal dispersion air velocity array (with logging microcontroller)
Manufacturer's model number	AFMS2000
Manufacturer's serial number	7283940
Instrument bias error	Zero
Instrument precision error	3% of airflow reading
Calibration status	Within six months of test
Instrument 02	Powered flow capture hood
Type	Flow capture hood attached to fan and flowmeter device using 12 ft long, 10 in. diameter flex duct
Manufacturer's model number	PFH4800
Manufacturer's serial number	94837261
Instrument bias error	Zero
Instrument precision error	3% of airflow reading
Calibration status	Within six months of test
Instrument 03	Barometer
Type	Digital logging barometer
Manufacturer's model number	BAR567
Manufacturer's serial number	6865949
Instrument bias error	Zero
Instrument precision error	0.5% of pressure reading or 500 Pa, whichever is greater
Calibration status	Within six months of test
Type	Autozeroing digital logging micromanometer

Table F-2 Instrumentation Used to Perform the Test Measurements *(Continued)*

Instrument 04	Pressure transducer
Manufacturer's model number	MAN-2578
Manufacturer's serial number	324-984325
Instrument bias error	Zero
Instrument precision error	1.0% of pressure reading or 0.2 Pa, whichever is greater
Calibration status	Within six months of test
Instrument 05	Thermometer
Type	Digital logging thermometer with Type K air probe
Manufacturer's model number	THERM333
Manufacturer's serial number	10-9578
Instrument bias error	Zero
Instrument precision error	0.5°C
Calibration status	Within six months of test
Instrument 06	Relative humidity meter
Type	Digital logging hygrometer with polymer-film capacitive probe
Manufacturer's model number	RH8764
Manufacturer's serial number	268975
Instrument bias error	Zero
Instrument precision error	3%
Calibration status	Within six months of test

Table F-3 Description of Air Distribution System Features and Scope of the Leakage Test Performed

System Description
The system is a single-duct VAV supply air type (refer to Figure F-1 of test plan as needed).

a. **General:** AHU-01 serves the first floor office space. All eight of the supply diffusers are four-way-throw nominal 2 × 2 ft square types with perforated faces and neck-mounted blade deflectors (ABC Corp. Model 34589).
b. **Terminal Units:** The seven terminal units are single-duct VAV units with electric reheat coils and DDC.
c. **Air-Handling Unit:** AHU-01 consists of a blow-through unit including a hot-water preheat coil, a chilled-water cooling coil, and a steam humidifier. The fan speed is controlled by a variable-frequency drive (VFD). The unit includes an outdoor air airflow measuring station (AFMS), minimum outdoor air control, and economizer control.
d. **Control System:** The control system is an XYZ Inc. DDC system. The system controls the following parameters of AHU-01: fan speed based on duct static pressure, discharge air temperature, and outdoor airflow (through an AFMS) and also incorporates safeties for freeze and high static pressure. The system components and parameters can be manipulated through a main host computer located in the facilities and maintenance room, but the system was manipulated through the local control panel in the mechanical room for this test.
e. **Scope of the Leakage Test Performed:** The Engineer of Record for this project specified an ASHRAE Standard 215 leakage test to determine the leakage from the HVAC air distribution system served by AHU-01 while the system is operational. The portion of the system to be tested includes the medium-pressure and the low-pressure supply air distribution system from the discharge of the AHU to the supply air diffusers. This test was conducted after system installation and testing, adjusting, and balancing (TAB) were complete. During the construction phase of this project, and prior to operation, 25% of the medium-pressure ductwork was leak tested (using a pressurization test) by the contractor, and the AHU-01 casing leakage was tested by the TAB firm. Casing leakage on sample terminal units was also tested as part of the project submittal process prior to installation.

Table F-4 Establishing the Reference Operating Conditions and Conduct of the Test

a. As required by the Engineer of Record's specifications for this test, this system was set up according to the applicable specifications given in ASHRAE Standard 215, Section 5, "Test Setup." The Engineer of Record specified the following additional test conditions:
   1. System shall be tested at the airflow corresponding to full cooling (the system has no diversity) of 1888 L/s (4000 cfm) at standard air density (1.204 kg/m^3 [0.075 lb$_m$/ft^3]).
   2. System shall be tested at the design operating duct static pressure of 224 Pa (0.90 in. of water).
b. This test was conducted in the office space during unoccupied periods.
c. To obtain a stable, repeatable test reference operating condition, the following operating parameters were determined during testing and are provided for future testing. This system was forced to operate under the following conditions during the complete duration of the test:
   1. AHU-01 fan speed = 100% (VFD set to 60 Hz).
   2. The return fan speed was set to track the flow of the supply fan.
   3. The return air damper was fully opened.
   4. The exhaust air damper and outdoor air damper were fully closed.
   5. The hot-water-coil control valve was fully closed (outdoor conditions allowed this).
   6. The cooling-coil control valve set point (discharge air control) was adjusted to maintain a 22.2°C (72.0°F) supply air temperature to avoid overcooling the space during the measurement phase of the test.
   7. All VAV terminal-unit reheat coils were off, and the control damper positions were manually commanded through building automation system overrides to the following positions to achieve a duct static pressure set point of 224 Pa (0.90 in. of water):
      i. Terminal Unit 1 = 94%
      ii. Terminal Unit 2 = 94%
      iii. Terminal Unit 3 = 94%
      iv. Terminal Unit 4 = 94%
      v. Terminal Unit 5 = 94%
      vi. Terminal Unit 6 = 94%
      vii. Terminal Unit 7 = 94%
   8. The duct static pressure sensor location in the test plan is at PL2 in Figure F-1. The duct static pressure measured at this location was 224 Pa (0.90 in. of water) and was stable.
   9. All space occupancy sensors were overridden to prevent changes in the system control during the test.

Table F-5 Test Data

Barometric pressure P_b at same elevation as airflow measurements, Pa (psi)					**100,570 (14.586)**
Plan Tag (Figure F-1 of test plan)	**Measured Airflow, L/s (cfm)**	**Measured Dry-Bulb Temperature, °C (°F)**	**Measured Duct Static Pressure, Pa (in. of water)**	**Measured Relative Humidity, %**	**Notes**
Q_{in} at start of test	1909 (4045)	22.2 (72.0)	224 (0.90)	45	AFMS at PL2; averaging period 60 s with 1 sample per second
Q_{in} at end of test	1886 (3997)	22.2 (72.0)	224 (0.90)	45	AFMS at PL2; averaging period of 60 s with 1 sample per second
Diffuser A	215 (455)	22.2 (72.0)	0 (0)	45	Flow capture hood; averaging period 20 s with 1 sample per second
Diffuser B	224 (475)	22.2 (72.0)	0 (0)	45	Same as Diffuser A
Diffuser C	238 (505)	22.2 (72.0)	0 (0)	45	Same as Diffuser A
Diffuser D	219 (465)	22.2 (72.0)	0 (0)	45	Same as Diffuser A
Diffuser E	217 (460)	22.2 (72.0)	0 (0)	45	Same as Diffuser A
Diffuser F	212 (450)	22.2 (72.0)	0 (0)	45	Same as Diffuser A
Diffuser G	229 (485)	22.2 (72.0)	0 (0)	45	Same as Diffuser A
Diffuser H	215 (455)	22.2 (72.0)	0 (0)	45	Same as Diffuser A

Table F-6 Calculations

Saturation pressure $P_{ws,x}$, Pa (psi) [Equation 3]				**2677 (0.389)**	
Plan Tag (Figure F-1 of test plan)	**Measured Airflow, L/s (cfm)**	**Actual Air Density, kg/m^3 (lb$_m$/ft^3) [Equation 2]**	**Standard Airflow, L/s (cfm) [Equation 4]**	**Instrument Precision**	**Uncertainty, L/s (cfm)**
Q_{in} at start of test	1909 (4045)	1.1835 (0.0739)	1877 (3984)		
Q_{in} at end of test	1886 (3997)	1.1835 (0.0739)	1854 (3937)		
Q_{in} average			1865 (3961)	3%	29 (61)
Deviation from initial reference operating condition during test			1.2%		
Q_{out}, Diffuser A	215 (455)	1.1809 (0.0737)	211 (447)	3%	3.2 (6.8)
Q_{out}, Diffuser B	224 (475)	1.1809 (0.0737)	220 (467)	3%	3.4 (7.1)
Q_{out}, Diffuser C	238 (505)	1.1809 (0.0737)	233 (496)	3%	3.6 (7.6)
Q_{out}, Diffuser D	219 (465)	1.1809 (0.0737)	215 (457)	3%	3.3 (7.0)
Q_{out}, Diffuser E	217 (460)	1.1809 (0.0737)	213 (452)	3%	3.3 (6.9)
Q_{out}, Diffuser F	212 (450)	1.1809 (0.0737)	208 (442)	3%	3.2 (6.8)
Q_{out}, Diffuser G	229 (485)	1.1809 (0.0737)	225 (477)	3%	3.4 (7.3)
Q_{out}, Diffuser H	215 (455)	1.1809 (0.0737)	211 (447)	3%	3.2 (6.8)
Q_{out}, (sum Diffusers A through H)			1735 (3686)		
Leakage airflow $Q_{leak,std}$, L/s (cfm) (Equation 5)			130 (275)		
Fractional leakage $L_{f,std}$, % (Equation 6)			6.9%		
Leakage flow uncertainty $u_{Q_{leak}}$ ±, L/s (cfm) (Equation 9)			30 (64)		
Fractional leakage uncertainty $u_{L_{f,std}}$ ±, % (Equation 10)			1.6%		
Fractional leakage range, %			6.9% ± 1.6% (5.3% to 8.6%)		

(This appendix is not 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. Unresolved objectors on informative material are not offered the right to appeal at ASHRAE or ANSI.)

INFORMATIVE APPENDIX G
DIAGNOSTIC PROCEDURE FOR AIRTIGHTNESS TESTING OF LOW-PRESSURE SYSTEM SECTIONS DURING OPERATION

This test procedure determines leakage downstream of a variable-air-volume (VAV) terminal-unit inlet damper during normal system fan operation and simultaneously determines the leakage area of the inlet damper opening in its minimum or shut position (Modera et al. 2014). The test setup and procedure consists of the following eight steps:

1. With the VAV terminal-unit inlet damper in the fully open position, measure the normal operating static pressure downstream of the inlet damper $P_{down,damper\ open}$ and at the inlet of each supply outlet $P_{diffuser}$. Calculate the average of the $P_{diffuser}$ values $P_{diffuser,avg}$.
2. Close the inlet damper to its minimum position (either by means of the building HVAC control system or manually).
3. Measure the static pressure upstream of the inlet damper with the damper in its minimum position P_{up}.
4. Block all diffusers downstream of the VAV terminal unit except for one, and connect a portable pressurization/depressurization fan and flowmeter device to the remaining unblocked diffuser.
5. Measure the flow through the fan/flowmeter and the coincident static pressure differential at a midway point in the duct section downstream of the VAV terminal-unit inlet damper P_{duct}. Perform these measurements at several downstream pressure conditions, obtaining at least four points separated by nominally 5 to 10 Pa (0.020 to 0.040 in. of water).
6. Perform an iterative fit to the data, solving simultaneously for the downstream-section leakage flow coefficient K_{leak}, the downstream-section leakage pressure exponent n, and the effective leakage area of the VAV terminal-unit inlet damper in its minimum position A_{damper}.
7. Calculate the leakage flow downstream of the inlet damper Q_{leak} based on
   a. the average pressure difference across duct leaks with the damper open:

      $P_{leak,avg} = (P_{down,damper\ open} + P_{diffuser,avg})/2$, and

   b. the leakage flow coefficient and pressure exponent calculated in Step 6:

      $Q_{leak} = K_{leak}\,(P_{leak,avg})^n$
8. Calculate the percentage leakage by dividing the result obtained in Step 7 (Q_{leak}) by the flow through the VAV terminal unit in the fully open position ($Q_{normal\ operation}$).

Step 6 of the procedure involves an iterative linear-regression solution essentially to the conservation of mass equation with density ρ assumed to be constant:

$$Q_{damper} + Q_{fan} + Q_{leak} = 0 \tag{G-1}$$

where

Q_{damper} = flow through VAV terminal-unit inlet damper, m^3/s (cfm)

Q_{fan} = flow through portable fan, m^3/s (cfm)

Q_{leak} = flow through leaks downstream of terminal-unit inlet damper, m^3/s (cfm)

Step 5 can include testing in three different modes:

- **Mode 1.** The portable fan is *off* and the pressure in the section being tested is higher than the pressure surrounding the test section. The flow through the VAV terminal unit inlet damper is split between the flow exiting leaks and the flow exiting through the portable fan.
- **Mode 2.** The portable fan is *pressurizing* the duct section downstream of the inlet damper such that the fan flow and the damper flow exit *only* through leaks in the test section.
- **Mode 3.** The portable fan is *depressurizing* the duct section being tested. As a result, there is flow into the leaks from the test section surroundings, and all of the flow through the inlet damper and leaks is exiting through the portable fan. **Mode 3** is not required, but it can be used

to identify one-way-valve effects in leaks (e.g., pressurization-related lifting of diffusers [3]) based on comparing depressurization results with pressurization results (**Mode 2**).

In **Mode 1**, the inlet damper is modeled as an orifice, and Equation G-2 is used to represent the test section leakage flow:

$$K_{leak}(P_{duct})^n = A_{damper}\sqrt{\frac{2(P_{up}-P_{duct})}{\rho}} + Q_{fan,\,off,\,out} \qquad \text{(SI G-2)}$$

$$K_{leak}(P_{duct})^n = 7.6218 \times A_{damper}\sqrt{\frac{(P_{up}-P_{duct})}{\rho}} + Q_{fan,\,off,\,out} \qquad \text{(I-P G-2)}$$

where

K_{leak} = flow coefficient for leaks downstream of the inlet damper, $m^3/(s{\cdot}Pa^n)$ [cfm/(in. of water)^n]

P_{duct} = average static pressure difference across leaks, Pa (in. of water)

n = pressure exponent for these leaks, dimensionless

A_{damper} = leakage area of the inlet damper opening, m^2 (in^2)

P_{up} = static pressure upstream of the inlet damper, Pa (in. of water)

ρ = density of air entering through the damper opening, kg/m^3 (lb_m/ft^3)

$Q_{fan,off,out}$ = flow *exiting* the duct section through the portable fan, m^3/s (cfm)

In **Mode 2**,

$$K_{leak}(P_{duct})^n = A_{damper}\sqrt{\frac{2(P_{up}-P_{duct})}{\rho}} + Q_{fan,\,on,\,in} \qquad \text{(SI G-3)}$$

$$K_{leak}(P_{duct})^n = 7.6218 \times A_{damper}\sqrt{\frac{(P_{up}-P_{duct})}{\rho}} + Q_{fan,\,on,\,in} \qquad \text{(I-P G-3)}$$

where

$Q_{fan,on,in}$ = flow *entering* the duct section through the portable fan, m^3/s (cfm)

Taking the natural logarithm of both sides of Equation G-3 yields the following:

$$\ln(K_{leak}) + n \times \ln(P_{duct}) = \ln\left[A_{damper}\sqrt{\frac{2(P_{up}-P_{duct})}{\rho}} + Q_{fan,\,on,\,in}\right] \qquad \text{(SI G-4)}$$

$$\ln(K_{leak}) + n \times \ln(P_{duct}) = \ln\left[7.6218 \times A_{damper}\sqrt{\frac{(P_{up}-P_{duct})}{\rho}} + Q_{fan,\,on,\,in}\right] \qquad \text{(I-P G-4)}$$

In **Mode 3**,

$$K_{leak}(P_{duct})^n = Q_{fan,\,on,\,out} - A_{damper}\sqrt{\frac{2(P_{up}-P_{duct})}{\rho}} \qquad \text{(SI G-5)}$$

$$K_{leak}(P_{duct})^n = Q_{fan,\,on,\,out} - 7.6218 \times A_{damper}\sqrt{\frac{(P_{up}-P_{duct})}{\rho}} \qquad \text{(I-P G-5)}$$

where

$Q_{fan,on,out}$ = flow *exiting* the duct section through the portable fan, m^3/s (cfm)

Taking the natural logarithm of both sides of Equation G-5 yields the following:

3. Sometimes the "back-plate" or top of the diffuser is not firmly connected to the ceiling support, in which case the back-plate lifts up creating an opening between it and the ceiling support, which acts like a leak to the ceiling plenum. During normal operation, there is usually not a large enough pressure to overcome gravity and friction holding down the back-plate.

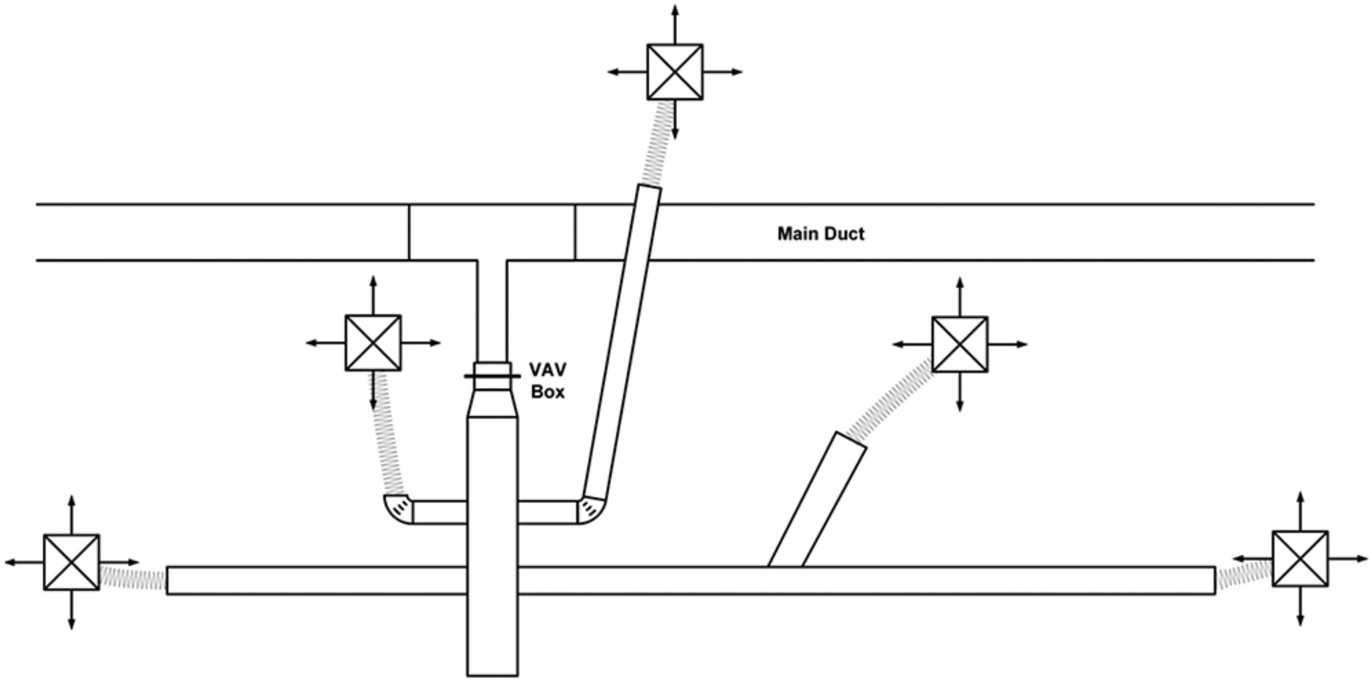

Figure G-1 Example system—five-diffuser duct section downstream of a VAV terminal unit.

$$\ln(K_{leak}) + n \times \ln(P_{duct}) = \ln\left[Q_{fan,\,on,\,out} - A_{damper}\sqrt{\frac{2(P_{up} - P_{duct})}{\rho}}\right] \quad \text{(SI G-6)}$$

$$\ln(K_{leak}) + n \times \ln(P_{duct}) = \ln\left[Q_{fan,\,on,\,out} - 7.6218 \times A_{damper}\sqrt{\frac{(P_{up} - P_{duct})}{\rho}}\right] \quad \text{(I-P G-6)}$$

The duct-section leakage flow coefficient K_{leak} and pressure exponent n, as well as the VAV inlet-damper opening leakage area A_{damper}, are calculated by an iterative solution of Equation G-2 plus Equation G-4 for *pressurization* (**Mode 2**), and Equation G-2 plus Equation G-6 for *depressurization* (**Mode 3**). These iterative solutions are based on measurements made at a series of pressure differentials P_{duct} that bracket the average pressure across the leaks $P_{leak,avg}$. Linear fits of Equation G-4 or G-6 are performed using different values of A_{damper}, iterating until A_{damper}, K_{leak}, and n also satisfy Equation G-2.

G1. EXAMPLE (PRESSURIZATION: MODES 1 AND 2)

Determine the leakage of the five-diffuser duct section downstream of the VAV terminal-unit inlet damper depicted in Figure G-1. The inlet airflow $Q_{normal\ operation}$ for the subject system from the TAB report is 567 L/s (1201 cfm). The air density ρ is 1.201 kg/m^3 (0.075 lb$_m$/ft^3).

Measured Data and Calculations
(See Example SI and I-P Spreadsheets on the ASHRAE Website)

- **Step 1.** Pressure downstream of VAV inlet damper in fully open position:

  $P_{down,damper\ open}$ = 63.0 Pa (0.253 in. of water)

  Pressures at inlet of each diffuser with VAV inlet damper in fully open position:

  $P_{diffuser}$ = 10.9, 16.9, 15.9, 18.9, 11.9 Pa (0.044, 0.068, 0.064, 0.076, 0.048 in. of water)

  $P_{diffuser,avg}$ = (10.9 + 16.9 + 15.9+ 18.9 + 11.9)/5 = 14.9 Pa (0.060 in. of water)

- **Steps 2 and 3.** Pressure upstream of VAV inlet damper at test conditions:

Table G-1 Example Measured Flows and Pressures (Modes 1 and 2)

Flow Mode	P_{duct}, Pa (in. of water)	$P_{up} - P_{duct}$, Pa (in. of water)	Q_{fan}, L/s (cfm)
1	17.9 (0.072)	94.1 (0.378)	–31 (–66)
2	48.0 (0.193)	64.0 (0.257)	37 (78)
2	53.0 (0.213)	59.0 (0.237)	47 (100)
2	61.0 (0.245)	51.0 (0.205)	62 (131)
2	66.9 (0.269)	45.1 (0.181)	75 (159)
2	74.9 (0.301)	37.1 (0.149)	91 (193)

$P_{up,damper\ closed} = 112.0$ Pa (0.450 in. of water)

- **Steps 4 and 5.** Measured flows and pressures with fan OFF (**Mode 1**) and during pressurization (**Mode 2**), as listed in Table G-1.
- **Step 6.** Iterative solution to Equations G-2 and G-4 (see example spreadsheets):

  $A_{damper} = 73.5\ \text{cm}^2 = 0.00735\ \text{m}^2$ (11.4 in.^{2})

  $K_{leak} = 0.01019\ \text{m}^3/(\text{s·Pa}^n) = 10.19\ \text{L}/(\text{s·Pa}^n)$ [663 cfm/(in. of water)^n]

  $n = 0.6207$

  Note: An explanation of the iterative solution of Equations G-2 and G-4 is given in the example spreadsheets. The iteratively determined values listed here, and their I-P equivalents, were determined using the SI spreadsheet. Corresponding values determined using the I-P spreadsheet are slightly different due to rounding errors.
- **Step 7.** Average leak pressure and leakage flow at full VAV terminal-unit flow:

  $P_{leak,avg} = (P_{down,damper\ open} - P_{diffuser,avg})/2$

  $P_{leak,avg} = (63.0 + 14.9)/2 = 39.0$ Pa (0.157 in. of water)

  $Q_{leak} = K_{leak} = (P_{leak,avg})^n = 0.01019(39.0)^{0.6207} = 0.099\ \text{m}^3/\text{s} = 99$ L/s (210 cfm)
- **Step 8.** Leakage fraction

$$\%\text{Leakage} = 100\left(\frac{Q_{leak}}{Q_{normal,\ operation}}\right) = 100\left(\frac{99}{567}\right) = 17\%$$

G2. EXAMPLE SPREADSHEETS

Example spreadsheets can be downloaded at **www.ashrae.org/Standard215-2018**.

(This appendix is not 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. Unresolved objectors on informative material are not offered the right to appeal at ASHRAE or ANSI.)

update references as shown on p.51-52 5 52

INFORMATIVE APPENDIX H
INFORMATIVE REFERENCES AND BIBLIOGRAPHY

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(This appendix is not 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. Unresolved objectors on informative material are not offered the right to appeal at ASHRAE or ANSI.)

INFORMATIVE APPENDIX H
INFORMATIVE REFERENCES AND BIBLIOGRAPHY

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POLICY STATEMENT DEFINING ASHRAE'S CONCERN 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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As an industry leader in research, standards writing, publishing, certification, and continuing education, ASHRAE and its members are dedicated to promoting a healthy and sustainable built environment for all, through strategic partnerships with organizations in the HVAC&R community and across related industries.

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