which may be used by manufacturers in support of petitions for waiver. These guidelines are not mandatory and the Department may determine that they do not apply to a particular model. Depending upon a manufacturer's approach for conducting field testing, additional data may be required. Manufacturers are encouraged to communicate with the Department prior to the commencement of field tests which may be used to support a petition for waiver. Section 6.3 of this Appendix provides an example of field testing for a clothes washer with an adaptive water fill control system. Other features, such as the use of various spin speed selections, could be the subject of field tests.

6.2 *Nonconventional Wash System Energy Consumption Test.* The field test may consist of a minimum of 10 of the nonconventional clothes washers (''test clothes washers'') and 10 clothes washers already being distributed in commerce (''base clothes washers''). The tests should include a minimum of 50 energy test cycles per clothes washer. The test clothes washers and base clothes washers should be identical in construction except for the controls or systems being tested. Equal numbers of both the test clothes washer and the base clothes washer should be tested simultaneously in comparable settings to minimize seasonal or consumer laundering conditions or variations. The clothes washers should be monitored in such a way as to accurately record the average total energy and water consumption per cycle, including water heating energy when electrically heated water is used, and the energy required to remove the remaining moisture of the test load. Standby and off mode energy consumption should be measured according to section 4.4 of this test procedure. The field test results should be used to determine the best method to correlate the rating of the test clothes washer to the rating of the base clothes washer.

6.3 *Adaptive water fill control system field test.* (1) Section 3.2.3.1 of this Appendix defines the test method for measuring energy consumption for clothes washers which incorporate both adaptive and alternate manual water fill control systems. Energy consumption calculated by the method defined in section 3.2.3.1 of this Appendix assumes the adaptive cycle will be used 50 percent of the time. This section can be used to develop field test data in support of a petition for waiver when it is believed that the adaptive cycle will be used more than 50 percent of the time. The field test sample size should be a minimum of 10 test clothes washers. The test clothes washers should be representative of the design, construction, and control system that will be placed in commerce. The duration of field testing in the user's house should be a minimum of 50 energy test cycles, for each unit. No special instructions as to cycle selection or product usage should be given to the field test participants, other than inclusion of the product literature pack which would be shipped with all units, and instructions regarding filling out data collection forms, use of data collection equipment, or basic procedural methods. Prior to the test clothes washers being installed in the field test locations, baseline data should be developed for all field test units by conducting laboratory tests as defined by section 1 through section 5 of this Appendix to determine the energy consumption, water consumption, and remaining moisture content values. The following data should be measured and recorded for each wash load during the test period: Wash cycle selected, the mode of the clothes washer (adaptive or manual), clothes load dry weight (measured after the clothes washer and clothes dryer cycles are completed) in pounds, and type of articles in the clothes load (*e.g.*, cottons, linens, permanent press). The wash loads used in calculating the in-home percentage split between adaptive and manual cycle usage should be only those wash loads which conform to the definition of the energy test cycle.

Calculate:

T=The total number of energy test cycles run during the field test.

T_a=The total number of adaptive control energy test cycles.

T_m=The total number of manual control energy test cycles.

The percentage weighting factors:

$P_a=(T_a/T) \times 100\%$ (the percentage weighting for adaptive control selection)

$P_m=(T_m/T) \times 100\%$ (the percentage weighting for manual control selection)

(2) Energy consumption (HE_T, ME_T, and D_E) and water consumption (Q_T), values calculated in section 4 of this Appendix for the manual and adaptive modes, should be combined using P_a and P_m as the weighting factors.

[77 FR 13939, Mar. 7, 2012]

APPENDIXES K–L TO SUBPART B OF PART 430 [RESERVED]

APPENDIX M TO SUBPART B OF PART 430—UNIFORM TEST METHOD FOR MEASURING THE ENERGY CONSUMPTION OF CENTRAL AIR CONDITIONERS AND HEAT PUMPS

NOTE: The procedures and calculations that refer to off mode energy consumption (*i.e.*, sections 3.13 and 4.2.8 of this appendix M) need not be performed to determine compliance with energy conservation standards for central air conditioners and heat pumps at this time. However, any representation related to standby mode and off mode energy consumption of these products made after corresponding revisions to the central air conditioners and heat pumps test procedure must be based upon results generated under this test procedure, consistent with the requirements of 42 U.S.C. 6293(c)(2). For residential central air conditioners and heat pumps manufactured on or after January 1, 2015, compliance with the applicable provisions of this test procedure is required in order to determine compliance with energy conservation standards.

3.2.2.1 Indoor fan capacity modulation that correlates with the outdoor dry bulb temperature.

3.2.2.2 Indoor fan capacity modulation based on adjusting the sensible to total (S/T) cooling capacity ratio.

3.2.3 Tests for a unit having a two-capacity compressor.

3.2.4 Tests for a unit having a variable-speed compressor.

3.3 Test procedures for steady-state wet coil cooling mode tests (the A, A_2, A_1, B, B_2, B_1, E_V, and F_1 Tests).

3.4 Test procedures for the optional steady-state dry coil cooling mode tests (the C, C_1, and G_1 Tests).

3.5 Test procedures for the optional cyclic dry coil cooling mode tests (the D, D_1, and I_1 Tests).

3.5.1 Procedures when testing ducted systems.

3.5.2 Procedures when testing non-ducted systems.

3.5.3 Cooling mode cyclic degradation coefficient calculation.

3.6 Heating mode tests for different types of heat pumps, including heating-only heat pumps.

3.6.1 Tests for a heat pump having a single-speed compressor that is tested with a fixed speed indoor fan installed, with a constant-air-volume-rate indoor fan installed, or with no indoor fan installed.

3.6.2 Tests for a heat pump having a single-speed compressor and a variable-speed, variable-air-volume-rate indoor fan: capacity modulation correlates with outdoor dry bulb temperature.

3.6.3 Tests for a heat pump having a two-capacity compressor (see Definition 1.45), including two-capacity, northern heat pumps (see Definition 1.46).

3.6.4 Tests for a heat pump having a variable-speed compressor.

3.6.5 Additional test for a heat pump having a heat comfort controller.

3.7 Test procedures for steady-state Maximum Temperature and High Temperature heating mode tests (the $H0_1$, H1, $H1_2$, $H1_1$, and $H1_N$ Tests).

3.8 Test procedures for the optional cyclic heating mode tests (the $H0C_1$, H1C, and $H1C_1$ Tests).

3.8.1 Heating mode cyclic degradation coefficient calculation.

3.9 Test procedures for Frost Accumulation heating mode tests (the H_2, $H2_2$, $H2_V$, and $H2_1$ Tests).

3.9.1 Average space heating capacity and electrical power calculations.

3.9.2 Demand defrost credit.

3.10 Test procedures for steady-state Low Temperature heating mode tests (the H_3, $H3_2$, and $H3_1$ Tests).

3.11 Additional requirements for the secondary test methods.

3.11.1 If using the Outdoor Air Enthalpy Method as the secondary test method.

3.11.1.1 If a preliminary test precedes the official test

3.11.1.2 If a preliminary test does not precede the official test.

3.11.1.3 Official test.

3.11.2 If using the Compressor Calibration Method as the secondary test method.

3.11.3 If using the Refrigerant Enthalpy Method as the secondary test method.

3.12 Rounding of space conditioning capacities for reporting purposes.

4. CALCULATIONS OF SEASONAL PERFORMANCE DESCRIPTORS

4.1 Seasonal Energy Efficiency Ratio (SEER) Calculations.

4.1.1 SEER calculations for an air conditioner or heat pump having a single-speed compressor that was tested with a fixed-speed indoor fan installed, a constant-air-volume-rate indoor fan installed, or with no indoor fan installed.

4.1.2 SEER calculations for an air conditioner or heat pump having a single-speed compressor and a variable-speed variable-air-volume-rate indoor fan.

4.1.2.1 Units covered by section 3.2.2.1 where indoor fan capacity modulation correlates with the outdoor dry bulb temperature.

4.1.2.2 Units covered by section 3.2.2.2 where indoor fan capacity modulation is used to adjust the sensible to total cooling capacity ratio.

4.1.3 SEER calculations for an air conditioner or heat pump having a two-capacity compressor.

4.1.3.1 Steady-state space cooling capacity at low compressor capacity is greater than or equal to the building cooling load at temperature T_j, $\dot{Q}_c^{k=1}(T_j) \geq BL(T_j)$.

4.1.3.2 Unit alternates between high (k=2) and low (k=1) compressor capacity to satisfy the building cooling load at temperature T_j, $\dot{Q}_c^{k=1}(T_j) < BL(T_j) < \dot{Q}_c^{k=2}(T_j)$.

4.1.3.3 Unit only operates at high (k=2) compressor capacity at temperature T_j and its capacity is greater than the building cooling load, $BL(T_j) < \dot{Q}_c^{k=2}(T_j)$.

4.1.3.4 Unit must operate continuously at high (k=2) compressor capacity at temperature T_j, $BL(T_j) \geq \dot{Q}_c^{k=2}(T_j)$.

4.1.4 SEER calculations for an air conditioner or heat pump having a variable-speed compressor.

4.1.4.1 Steady-state space cooling capacity when operating at minimum compressor speed is greater than or equal to the building cooling load at temperature T_j, $\dot{Q}_c^{k=1}(T_j) \geq BL(T_j)$.

4.1.4.2 Unit operates at an intermediate compressor speed (k=i) in order to match the building cooling load at temperature T_j, $\dot{Q}_c^{k=1}(T_j) < BL(T_j) < \dot{Q}_c^{k=2}(T_j)$.

4.1.4.3 Unit must operate continuously at maximum (k=2) compressor speed at temperature T_j, $BL(T_j) \geq \dot{Q}_c^{k=2}(T_j)$.

4.2 Heating Seasonal Performance Factor (HSPF) Calculations.

4.2.1 Additional steps for calculating the HSPF of a heat pump having a single-speed compressor that was tested with a fixed-speed indoor fan installed, a constant-air-volume-rate indoor fan installed, or with no indoor fan installed.

4.2.2 Additional steps for calculating the HSPF of a heat pump having a single-speed compressor and a variable-speed, variable-air-volume-rate indoor fan.

4.2.3 Additional steps for calculating the HSPF of a heat pump having a two-capacity compressor.

4.2.3.1 Steady-state space heating capacity when operating at low compressor capacity is greater than or equal to the building heating load at temperature T_j, $\dot{Q}_h^{k=1}(T_j) \geq BL(T_j)$.

4.2.3.2 Heat pump alternates between high (k=2) and low (k=1) compressor capacity to satisfy the building heating load at a temperature T_j, $\dot{Q}_h^{k=1}(T_j)$ BL $(T_j) < \dot{Q}_h^{k=2}(T_j)$.

4.2.3.3 Heat pump only operates at high (k=2) compressor capacity at temperature T_j and its capacity is greater than the building heating load, $BL(T_j) < \dot{Q}_h^{k=2}(T_j)$.

4.2.3.4 Heat pump must operate continuously at high (k=2) compressor capacity at temperature T_j, $BL(T_j) \geq \dot{Q}_h^{k=2}(T_j)$.

4.2.4 Additional steps for calculating the HSPF of a heat pump having a variable-speed compressor.

4.2.4.1 Steady-state space heating capacity when operating at minimum compressor speed is greater than or equal to the building heating load at temperature T_j, $\dot{Q}_h^{k=1}(T_j) \geq BL(T_j)$.

4.2.4.2 Heat pump operates at an intermediate compressor speed (k=i) in order to match the building heating load at a temperature T_j, $\dot{Q}_h^{k=1}(T_j) < BL(T_j) < \dot{Q}_h^{k=2}(T_j)$.

4.2.4.3 Heat pump must operate continuously at maximum (k=2) compressor speed at temperature T_j, $BL(T_j) \geq \dot{Q}_h^{k=2}(T_j)$.

4.2.5 Heat pumps having a heat comfort controller.

4.2.5.1 Heat pump having a heat comfort controller: Additional steps for calculating the HSPF of a heat pump having a single-speed compressor that was tested with a fixed-speed indoor fan installed, a constant-air-volume-rate indoor fan installed, or with no indoor fan installed.

4.2.5.2 Heat pump having a heat comfort controller: Additional steps for calculating the HSPF of a heat pump having a single-speed compressor and a variable-speed, variable-air-volume-rate indoor fan.

4.2.5.3 Heat pumps having a heat comfort controller: Additional steps for calculating the HSPF of a heat pump having a two-capacity compressor.

4.2.5.4 Heat pumps having a heat comfort controller: Additional steps for calculating the HSPF of a heat pump having a variable-speed compressor. [Reserved]

4.3 Calculations of the Actual and Representative Regional Annual Performance Factors for Heat Pumps.

4.3.1 Calculation of actual regional annual performance factors (APF_A) for a particular location and for each standardized design heating requirement.

4.3.2 Calculation of representative regional annual performance factors (APF_R) for each generalized climatic region and for each standardized design heating requirement.

4.4 Rounding of SEER, HSPF, and APF for reporting purposes.

1. Definitions

1.1 *Annual performance factor* means the total heating and cooling done by a heat pump in a particular region in one year divided by the total electric energy used in one year. Paragraph (m)(3)(iii) of §430.23 of the Code of Federal Regulations states the calculation requirements for this rating descriptor.

1.2 *ARI* means Air-Conditioning and Refrigeration Institute.

1.3 *ARI Standard 210/240–2006* means the test standard ''Unitary Air-Conditioning and Air-Source Heat Pump Equipment'' published in 2006 by ARI.

1.4 *ASHRAE* means the American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

1.5 *ASHRAE Standard 23–2005* means the test standard ''Methods of Testing for Rating Positive Displacement Refrigerant Compressors and Condensing Units'' published in 2005 by ASHRAE.

1.6 *ASHRAE Standard 37–2005* means the test standard ''Methods of Testing for Rating Unitary Air-Conditioning and Heat Pump Equipment'' published in 2005 by ASHRAE.

1.7 *ASHRAE Standard 41.1–86 (RA 01)* means the test standard ''Standard Method for Temperature Measurement'' published in 1986 and reaffirmed in 2001 by ASHRAE.

1.8 *ASHRAE Standard 41.2–87 (RA 92)* means the test standard ''Standard Methods for Laboratory Airflow Measurement'' published in 1987 and reaffirmed in 1992 by ASHRAE.

1.9 *ASHRAE Standard 41.6–94 (RA 01)* means the test standard ''Method for Measurement of Moist Air Properties'' published in 1994 and reaffirmed in 2001 by ASHRAE.

1.10 *ASHRAE Standard 41.9–00* means the test standard ''Calorimeter Test Methods for Mass Flow Measurements of Volatile Refrigerants'' published in 2000 by ASHRAE.

1.11 *ASHRAE Standard 51–99/AMCA Standard 210–1999* means the test standard ''Laboratory Methods of Testing Fans for

Aerodynamic Performance Rating'' published in 1999 by ASHRAE and the Air Movement and Control Association International, Inc.

1.12 ASHRAE Standard 116–95 RA(05) means the test standard ''Methods of Testing for Rating for Seasonal Efficiency of Unitary Air Conditioners and Heat Pumps'' published in 1995 and reaffirmed in 2005 by ASHRAE.

1.13 CFR means Code of Federal Regulations.

1.14 Constant-air-volume-rate indoor fan means a fan that varies its operating speed to provide a fixed air-volume-rate from a ducted system.

1.15 Continuously recorded, when referring to a dry bulb measurement, means that the specified temperature must be sampled at regular intervals that are equal to or less than the maximum intervals specified in section 4.3 part ''a'' of ASHRAE Standard 41.1–86 (RA 01). If such dry bulb temperatures are used only for test room control, it means that one samples at regular intervals equal to or less than the maximum intervals specified in section 4.3 part ''b'' of the same ASHRAE Standard. Regarding wet bulb temperature, dew point temperature, or relative humidity measurements, continuously recorded means that the measurements must be made at regular intervals that are equal to or less than 1 minute.

1.16 Cooling load factor (CLF) means the ratio having as its numerator the total cooling delivered during a cyclic operating interval consisting of one ON period and one OFF period. The denominator is the total cooling that would be delivered, given the same ambient conditions, had the unit operated continuously at its steady-state space cooling capacity for the same total time (ON + OFF) interval.

1.17 Coefficient of Performance (COP) means the ratio of the average rate of space heating delivered to the average rate of electrical energy consumed by the heat pump. These rate quantities must be determined from a single test or, if derived via interpolation, must be tied to a single set of operating conditions. COP is a dimensionless quantity. When determined for a ducted unit tested without an indoor fan installed, COP must include the section 3.7, 3.8, and 3.9.1 default values for the heat output and power input of a fan motor.

1.18 Cyclic Test means a test where the unit's compressor is cycled on and off for specific time intervals. A cyclic test provides half the information needed to calculate a degradation coefficient.

1.19 Damper box means a short section of duct having an air damper that meets the performance requirements of section 2.5.7.

1.20 Degradation coefficient (C_D) means a parameter used in calculating the part load factor. The degradation coefficient for cooling is denoted by C_D^c. The degradation coefficient for heating is denoted by C_D^h.

1.21 Demand-defrost control system means a system that defrosts the heat pump outdoor coil only when measuring a predetermined degradation of performance. The heat pump's controls monitor one or more parameters that always vary with the amount of frost accumulated on the outdoor coil (*e.g.*, coil to air differential temperature, coil differential air pressure, outdoor fan power or current, optical sensors, etc.) at least once for every ten minutes of compressor ON-time when space heating. One acceptable alternative to the criterion given in the prior sentence is a feedback system that measures the length of the defrost period and adjusts defrost frequency accordingly.[1] In all cases, when the frost parameter(s) reaches a predetermined value, the system initiates a defrost. In a demand-defrost control system, defrosts are terminated based on monitoring a parameter(s) that indicates that frost has been eliminated from the coil.

A demand-defrost control system, which otherwise meets the above requirements, may allow time-initiated defrosts if, and only if, such defrosts occur after 6 hours of compressor operating time.

1.22 Design heating requirement (DHR) predicts the space heating load of a residence when subjected to outdoor design conditions. Estimates for the minimum and maximum DHR are provided for six generalized U.S. climatic regions in section 4.2.

1.23 Dry-coil tests are cooling mode tests where the wet-bulb temperature of the air supplied to the indoor coil is maintained low enough that no condensate forms on this coil.

1.24 Ducted system means an air conditioner or heat pump that is designed to be permanently installed equipment and delivers conditioned air to the indoor space through a duct(s). The air conditioner or heat pump may be either a split system or a single-packaged unit.

1.25 Energy efficiency ratio (EER) means the ratio of the average rate of space cooling delivered to the average rate of electrical energy consumed by the air conditioner or heat pump. These rate quantities must be determined from a single test or, if derived via interpolation, must be tied to a single set of operating conditions. EER is expressed in units of

$$\frac{\text{Btu/h}}{\text{W}}.$$

When determined for a ducted unit tested without an indoor fan installed, EER must

[1] Systems that vary defrost intervals according to outdoor dry-bulb temperature are not demand defrost systems.

include the section 3.3 and 3.5.1 default values for the heat output and power input of a fan motor.

1.26 Heating load factor (HLF) means the ratio having as its numerator the total heating delivered during a cyclic operating interval consisting of one ON period and one OFF period. The denominator is the total heating that would be delivered, given the same ambient conditions, if the unit operated continuously at its steady-state space heating capacity for the same total time (ON plus OFF) interval.

1.27 Heating seasonal performance factor (HSPF) means the total space heating required during the space heating season, expressed in Btu's, divided by the total electrical energy consumed by the heat pump system during the same season, expressed in watt-hours. The HSPF used to evaluate compliance with the Energy Conservation Standards (see 10 CFR 430.32(c), subpart C) is based on Region IV, the minimum standardized design heating requirement, and the sampling plan stated in 10 CFR 430.24(m), subpart B.

1.28 Heat pump having a heat comfort controller means equipment that regulates the operation of the electric resistance elements to assure that the air temperature leaving the indoor section does not fall below a specified temperature. This specified temperature is usually field adjustable. Heat pumps that actively regulate the rate of electric resistance heating when operating below the balance point (as the result of a second stage call from the thermostat) but do not operate to maintain a minimum delivery temperature are not considered as having a heat comfort controller.

1.29 Mini-split air conditioners and heat pumps means systems that have a single outdoor section and one or more indoor sections. The indoor sections cycle on and off in unison in response to a single indoor thermostat.

1.30 Multiple-split air conditioners and heat pumps means systems that have two or more indoor sections. The indoor sections operate independently and can be used to condition multiple zones in response to multiple indoor thermostats.

1.31 Non-ducted system means an air conditioner or heat pump that is designed to be permanently installed equipment and directly heats or cools air within the conditioned space using one or more indoor coils that are mounted on room walls and/or ceilings. The unit may be of a modular design that allows for combining multiple outdoor coils and compressors to create one overall system. Non-ducted systems covered by this test procedure are all split systems.

1.32 Part-load factor (PLF) means the ratio of the cyclic energy efficiency ratio (coefficient of performance) to the steady-state energy efficiency ratio (coefficient of performance). Evaluate both energy efficiency ratios (coefficients of performance) based on operation at the same ambient conditions.

1.33 Seasonal energy efficiency ratio (SEER) means the total heat removed from the conditioned space during the annual cooling season, expressed in Btu's, divided by the total electrical energy consumed by the air conditioner or heat pump during the same season, expressed in watt-hours. The SEER calculation in section 4.1 of this appendix and the sampling plan stated in 10 CFR subpart B, 430.24(m) are used to evaluate compliance with the Energy Conservation Standards. (See 10 CFR 430.32(c), subpart C.)

1.34 Single-packaged unit means any central air conditioner or heat pump that has all major assemblies enclosed in one cabinet.

1.35 Small-duct, high-velocity system means a system that contains a blower and indoor coil combination that is designed for, and produces, at least 1.2 inches (of water) of external static pressure when operated at the full-loadair volume rate of 220–350 cfm per rated ton of cooling. When applied in the field, small-duct products use high-velocity room outlets (*i.e.*, generally greater than 1000 fpm) having less than 6.0 square inches of free area.

1.36 Split system means any air conditioner or heat pump that has one or more of the major assemblies separated from the others.

1.37 Standard Air means dry air having a mass density of 0.075 lb/ft^3.

1.38 Steady-state test means a test where the test conditions are regulated to remain as constant as possible while the unit operates continuously in the same mode.

1.39 Temperature bin means the 5 °F increments that are used to partition the outdoor dry-bulb temperature ranges of the cooling (≥ 65 °F) and heating (< 65 °F) seasons.

1.40 Test condition tolerance means the maximum permissible difference between the average value of the measured test parameter and the specified test condition.

1.41 Test operating tolerance means the maximum permissible range that a measurement may vary over the specified test interval. The difference between the maximum and minimum sampled values must be less than or equal to the specified test operating tolerance.

1.42 Time adaptive defrost control system is a demand-defrost control system (see definition 1.21) that measures the length of the prior defrost period(s) and uses that information to automatically determine when to initiate the next defrost cycle.

1.43 Time-temperature defrost control systems initiate or evaluate initiating a defrost cycle only when a predetermined cumulative compressor ON-time is obtained. This predetermined ON-time is generally a fixed value (*e.g.*, 30, 45, 90 minutes) although it may vary based on the measured outdoor dry-bulb temperature. The ON-time counter accumulates if controller measurements (*e.g.*, outdoor temperature, evaporator temperature) indicate that frost formation conditions are present, and it is reset/remains at zero at all other times. In one application of the control scheme, a defrost is initiated whenever the counter time equals the predetermined ON-time. The counter is reset when the defrost cycle is completed.

In a second application of the control scheme, one or more parameters are measured (*e.g.*, air and/or refrigerant temperatures) at the predetermined, cumulative, compressor ON-time. A defrost is initiated only if the measured parameter(s) falls within a predetermined range. The ON-time counter is reset regardless of whether a defrost is initiated. If systems of this second type use cumulative ON-time intervals of 10 minutes or less, then the heat pump may qualify as having a demand defrost control system (see definition 1.21).

1.44 Triple-split system means an air conditioner or heat pump that is composed of three separate components: An outdoor fan coil section, an indoor fan coil section, and an indoor compressor section.

1.45 Two-capacity (or two-stage) compressor means an air conditioner or heat pump that has one of the following:

(1) A two-speed compressor,

(2) Two compressors where only one compressor ever operates at a time,

(3) Two compressors where one compressor (Compressor #1) operates at low loads and both compressors (Compressors #1 and #2) operate at high loads but Compressor #2 never operates alone, or

(4) A compressor that is capable of cylinder or scroll unloading.

For such systems, low capacity means:

(1) Operating at low compressor speed,

(2) Operating the lower capacity compressor,

(3) Operating Compressor #1, or

(4) Operating with the compressor unloaded (*e.g.*, operating one piston of a two-piston reciprocating compressor, using a fixed fractional volume of the full scroll, etc.).

For such systems, high capacity means:

(1) Operating at high compressor speed,

(2) Operating the higher capacity compressor,

(3) Operating Compressors #1 and #2, or

(4) Operating with the compressor loaded (*e.g.*, operating both pistons of a two-piston reciprocating compressor, using the full volume of the scroll).

1.46 Two-capacity, northern heat pump means a heat pump that has a factory or field-selectable lock-out feature to prevent space cooling at high-capacity. Two-capacity heat pumps having this feature will typically have two sets of ratings, one with the feature disabled and one with the feature enabled. The indoor coil model number should reflect whether the ratings pertain to the lockout enabled option via the inclusion of an extra identifier, such as ''+LO.'' When testing as a two-capacity, northern heat pump, the lockout feature must remain enabled for all tests.

1.47 Wet-coil test means a test conducted at test conditions that typically cause water vapor to condense on the test unit evaporator coil.

2. Testing Conditions

This test procedure covers split-type and single-packaged ducted units and split-type non-ducted units. Except for units having a variable-speed compressor, ducted units tested without an indoor fan installed are covered.

a. Only a subset of the sections listed in this test procedure apply when testing and rating a particular unit. Tables 1–A through 1–C show which sections of the test procedure apply to each type of equipment. In each table, look at all four of the Roman numeral categories to see what test sections apply to the equipment being tested.

1. The first category, Rows I–1 through I–4 of the Tables, pertains to the compressor and indoor fan features of the equipment. After identifying the correct ''I'' row, find the table cells in the same row that list the type of equipment being tested: Air conditioner (AC), heat pump (HP), or heating-only heat pump (HH). Use the test section(s) listed above each noted table cell for testing and rating the unit.

2. The second category, Rows II–1 and II–2, pertains to the presence or absence of ducts. Row II–1 shows the test procedure sections that apply to ducted systems, and Row II–2 shows those that apply to non-ducted systems.

3. The third category is for special features that may be present in the equipment. When testing units that have one or more of the three (special) equipment features described by the Table legend for Category III, use Row III to find test sections that apply.

4. The fourth category is for the secondary test method to be used. If the secondary method for determining the unit's cooling and/or heating capacity is known, use Row IV to find the appropriate test sections. Otherwise, include all of the test sections referenced by Row IV cell entries—*i.e.*, sections 2.10 to 2.10.3 and 3.11 to 3.11.3—among those sections consulted for testing and rating information.

b. Obtain a complete listing of all pertinent test sections by recording those sections identified from the four categories above.

c. The user should note that, for many sections, only part of a section applies to the unit being tested. In a few cases, the entire section may not apply. For example, sections 3.4 to 3.5.3 (which describe optional dry coil tests), are not relevant if the allowed default value for the cooling mode cyclic degradation coefficient is used rather than determining it by testing.

Example for Using Tables 1–A to 1–C

Equipment Description: A ducted air conditioner having a single-speed compressor, a fixed-speed indoor fan, and a multi-speed outdoor fan.

Secondary Test Method: Refrigerant Enthalpy Method

Step 1. Determine which of four listed Row ''I'' options applies ==> Row I–2

Table 1–A: ''AC'' in Row I–2 is found in the columns for sections 1.1 to 1.47, 2.1 to 2.2, 2.2.4 to 2.2.4.1, 2.2.5, 2.3 to 2.3.1, 2.4 to 2.4.1, 2.5, 2.5.2 to 2.10, and 2.11 to 2.13.

Table 1–B: ''AC'' is listed in Row I–2 for sections 3 to 3.1.4, 3.1.5 to 3.1.8, 3.2.1, 3.3 to 3.5, 3.5.3, 3.11 and 3.12.

Table 1–C: ''AC'' is listed in Row I–2 for sections 4.1.1 and 4.4.

Step 2. Equipment is ducted ==> Row II–1

Table 1–A: ''AC'' is listed in Row II–1 for sections 2.4.2 and 2.5.1 to 2.5.1.2.

Table 1–B: ''AC'' is listed in Row II–1 for sections 3.1.4.1 to 3.1.4.1.1 and 3.5.1.

Table 1–C: no ''AC'' listings in Row II–1.

Step 3. Equipment Special Features include multi-speed outdoor fan ==> Row III, M

Table 1–A: ''M'' is listed in Row III for section 2.2.2

Tables 1–B and 1–C: no ''M'' listings in Row III.

Step 4. Secondary Test Method is Refrigerant Enthalpy Method ==> Row IV, R

Table 1–A: ''R'' is listed in Row IV for section 2.10.3

Table 1–B: ''R'' is listed in Row IV for section 3.11.3

Table 1–C: no ''R'' listings in Row IV.

Step 5. Cumulative listing of applicable test procedure sections 1.1 to 1.47, 2.1 to 2.2, 2.2.2, 2.2.4 to 2.4.1, 2.2.5, 2.3 to 2.3.1, 2.4 to 2.4.1, 2.4.2, 2.5, 2.5.1 to 2.5.1.2, 2.5.2 to 2.10, 2.10.3, 2.11 to 2.13, 3. to 3.1.4, 3.1.4.1 to 3.1.4.1.1, 3.1.5 to 3.1.8, 3.2.1, 3.3 to 3.5, 3.5.1, 3.5.3, 3.11, 3.11.3, 3.12, 4.1.1, and 4.4.

Table 1A. Selection of Test Procedure Sections: Section 1 (Definitions) and Section 2 (Testing Conditions)

Key Equipment Features and Secondary Test Method / Sections From the Test Procedure	1.1 to 1.47	2.1 to 2.2	2.2.1	2.2.2	2.2.3	2.2.4 to 2.2.4.1	2.2.4.2	2.2.5	2.3 to 2.3.1	2.3.2	2.4 to 2.4.1	2.4.2	2.5	2.5.1 to 2.5.1.2	2.5.2 to 2.10	2.10.1	2.10.2	2.10.3	2.11 to 2.13
I-1. Single-speed Compressor; Variable-Speed Variable Air Volume Indoor Fan	AC HP HH	AC HP HH	HP HH			AC HP	HP HH	AC HP HH	AC HP	HP HH	AC HP HH		AC HP HH		AC HP HH				AC HP HH
I-2. Single-speed Compressor Except as Covered by "I-1"	AC HP HH	AC HP HH	HP HH			AC HP	HP HH	AC HP HH	AC HP	HP HH	AC HP HH		AC HP HH		AC HP HH				AC HP HH
I-3. Two-capacity Compressor	AC HP HH	AC HP HH	HP HH			AC HP	HP HH	AC HP HH	AC HP	HP HH	AC HP HH		AC HP HH		AC HP HH				AC HP HH
I-4. Variable-speed Compressor	AC HP HH	AC HP HH	HP HH			AC HP	HP HH	AC HP HH	AC HP	HP HH	AC HP HH		AC HP HH		AC HP HH				AC HP HH
II-1. Ducted												AC HP HH		AC HP HH					
II-2. Non-Ducted																			
III. Special Features				M	G														
IV. Secondary Test Method																O	C	R	

Legend for Table Entries:

Categories I and II: AC = applies for an Air Conditioner that meets the corresponding Column 1 "Key Equipment . . ." criterion
HP = applies for a Heat Pump that meets the corresponding Column 1 "Key Equipment . . ." criterion
HH = applies for a Heating-only Heat pump that meets the corresponding Column 1 "Key Equipment . . ." criterion

Category III: G = ganged mini-splits or multi-splits;
H = heat pump with a heat comfort controller;
M = units with a multi-speed outdoor fan.

Category IV: O = Outdoor Air Enthalpy Method; C = Compressor Calibration Method; R = Refrigerant Enthalpy Method

Table 1B. Selection of Test Procedure Sections: Section 3 (Testing Procedures)

Key Equipment Features and Secondary Test Method \ Sections From the Test Procedure	3. to 3.1.4	3.1.4.1 to 3.1.4.1.1	3.1.4.1.2	3.1.4.2	3.1.4.3	3.1.4.4 to 3.1.4.4.2	3.1.4.4.3	3.1.4.4.4	3.1.4.5	3.1.4.6 to 3.1.4.7	3.1.5 to 3.1.8	3.1.9	3.2.1	3.2.2 to 3.2.2.2	3.2.3	3.2.4	3.3 to 3.5	3.5.1	3.5.2	3.5.3
I-1. Single-speed Compressor; Variable-speed, Variable Air Volume Indoor Fan	AC HP HH			AC HP					HP HH		AC HP HH	HP HH		AC HP			AC HP			AC HP
I-2. Single-speed Compressor Except as Covered by "I-1"	AC HP HH										AC HP HH	HP HH	AC HP				AC HP			AC HP
I-3. Two-capacity Compressor	AC HP HH			AC HP					HP HH		AC HP HH	HP HH			AC HP		AC HP			AC HP
I-4. Variable-speed Compressor	AC HP HH			AC HP	AC HP				HP HH	HP HH	AC HP HH	HP HH				AC HP	AC HP			AC HP
II-1. Ducted		AC HP				HP	HH											AC HP		
II-2. Non-Ducted			AC HP					HP HH											AC HP	
III. Special Features												H								
IV. Secondary Test Method																				

Legend for Table Entries:

Categories I and II: AC = applies for an Air Conditioner that meets the corresponding Column 1 "Key Equipment . . ." criterion

HP = applies for a Heat Pump that meets the corresponding Column 1 "Key Equipment . . ." criterion

HH = applies for a Heating-only Heat pump that meets the corresponding Column 1 "Key Equipment . . ." criterion

Category III: G = ganged mini-splits or multi-splits;

H = heat pump with a heat comfort controller;

M = units with a multi-speed outdoor fan.

Category IV: O = Outdoor Air Enthalpy Method; C = Compressor Calibration Method; R = Refrigerant Enthalpy Method

Table 1B. Selection of Test Procedure Sections: Section 3 (Testing Procedures) (continued)

Key Equipment Features and Secondary Test Method \ Sections From the Test Procedure	3.6.1	3.6.2	3.6.3	3.6.4	3.6.5	3.7 to 3.8.1	3.9 to 3.10	3.11	3.11.1 to 3.11.1.3	3.11.2	3.11.3	3.12
I-1. Single-speed Compressor; Variable-speed, Variable Air Volume Indoor Fan		HP HH				HP HH	HP HH	AC HP HH				AC HP HH
I-2. Single-speed Compressor Except as Covered by "I-1"	HP HH					HP HH	HP HH	AC HP HH				AC HP HH
I-3. Two-capacity Compressor			HP HH			HP HH	HP HH	AC HP HH				AC HP HH
I-4. Variable-speed Compressor				HP HH		HP HH	HP HH	AC HP HH				AC HP HH
II-1. Ducted												
II-2. Non-Ducted												
III. Special Features					H							
IV. Secondary Test Method									O	C	R	

Legend for Table Entries:

Categories I and II: AC = applies for an Air Conditioner that meets the corresponding Column 1 "Key Equipment . . ." criterion
HP = applies for a Heat Pump that meets the corresponding Column 1 "Key Equipment . . ." criterion
HH = applies for a Heating-only Heat pump that meets the corresponding Column 1 "Key Equipment . . ." criterion

Category III: G = ganged mini-splits or multi-splits;
H = heat pump with a heat comfort controller;
M = units with a multi-speed outdoor fan.

Category IV: O = Outdoor Air Enthalpy Method; C = Compressor Calibration Method; R = Refrigerant Enthalpy Method

Table 1C. Selection of Test Procedure Sections: Section 4 (Calculations of Seasonal Performance Descriptors)

Key Equipment Features and Secondary Test Method / Sections From the Test Procedure	4 to 4.1	4.1.1	4.1.2 to 4.1.2.2	4.1.3 to 4.1.3.4	4.1.4 to 4.1.4.3	4.2	4.2.1	4.2.2	4.2.3 to 4.2.3.4	4.2.4 to 4.2.4.3	4.2.5 to 4.2.5.4	4.3 to 4.3.2	4.4
I-1. Single-speed Compressor; Variable-speed Variable Air Volume Indoor Fan	AC HP		AC HP			HP HH		HP HH				HP	AC HP HH
I-2. Single-speed Compressor Except as Covered by "I-1"		AC HP				HP HH	HP HH					HP	AC HP HH
I-3. Two-capacity Compressor	AC HP			AC HP		HP HH			HP HH			HP	AC HP HH
I-4. Variable-speed Compressor	AC HP				AC HP	HP HH				HP HH		HP	AC HP HH
II-1. Ducted													
II-2. Non-Ducted													
III. Special Features						H					H		
IV. Secondary Test Method													

Legend for Table Entries:

Categories I and II: AC = applies for an Air Conditioner that meets the corresponding Column 1 "Key Equipment . . ." criterion

HP = applies for a Heat Pump that meets the corresponding Column 1 "Key Equipment . . ." criterion

HH = applies for a Heating-only Heat pump that meets the corresponding Column 1 "Key Equipment . . ." criterion

Category III: G = ganged mini-splits or multi-splits;

H = heat pump with a heat comfort controller;

M = units with a multi-speed outdoor fan.

Category IV: O = Outdoor Air Enthalpy Method; C = Compressor Calibration Method; R = Refrigerant Enthalpy Method

2.1 Test room requirements. a. Test using two side-by-side rooms, an indoor test room and an outdoor test room. For multiple-split air conditioners and heat pumps (see Definition 1.30), however, use as many available indoor test rooms as needed to accommodate the total number of indoor units. These rooms must comply with the requirements specified in sections 8.1.2 and 8.1.3 of ASHRAE Standard 37–2005 (incorporated by reference, see §430.22).

b. Inside these test rooms, use artificial loads during cyclic tests and frost accumulation tests, if needed, to produce stabilized room air temperatures. For one room, select an electric resistance heater(s) having a heating capacity that is approximately equal to the heating capacity of the test unit's condenser. For the second room, select a heater(s) having a capacity that is close to the sensible cooling capacity of the test unit's evaporator. When applied, cycle the heater located in the same room as the test unit evaporator coil ON and OFF when the test unit cycles ON and OFF. Cycle the heater located in the same room as the test unit condensing coil ON and OFF when the test unit cycles OFF and ON.

2.2 Test unit installation requirements. a. Install the unit according to section 8.2 of ASHRAE Standard 37–2005 (incorporated by reference, see §430.22). With respect to interconnecting tubing used when testing split systems, however, follow the requirements given in section 6.1.3.5 of ARI Standard 210/240–2006 (incorporated by reference, see §430.22). When testing triple-split systems (see Definition 1.44), use the tubing length specified in section 6.1.3.5 of ARI Standard 210/240–2006 (incorporated by reference, see §430.22) to connect the outdoor coil, indoor compressor section, and indoor coil while still meeting the requirement of exposing 10 feet of the tubing to outside conditions. When testing split systems having multiple indoor coils, connect each indoor fan-coil to the outdoor unit using: (a) 25 feet of tubing, or (b) tubing furnished by the manufacturer, whichever is longer. If they are needed to make a secondary measurement of capacity, install refrigerant pressure measuring instruments as described in section 8.2.5 of ASHRAE Standard 37–2005 (incorporated by reference, see §430.22). Refer to section 2.10 of this appendix to learn which secondary methods require refrigerant pressure measurements. At a minimum, insulate the low-pressure line(s) of a split-system with insulation having an inside diameter that matches the refrigerant tubing and a nominal thickness of 0.5 inch.

b. For units designed for both horizontal and vertical installation or for both up-flow and down-flow vertical installations, the manufacturer must specify the orientation used for testing. Conduct testing with the following installed:

(1) the most restrictive filter(s);

(2) supplementary heating coils; and

(3) other equipment specified as part of the unit, including all hardware used by a heat comfort controller if so equipped (see Definition 1.28). For small-duct, high-velocity systems, configure all balance dampers or restrictor devices on or inside the unit to fully open or lowest restriction.

c. Testing a ducted unit without having an indoor air filter installed is permissible as long as the minimum external static pressure requirement is adjusted as stated in Table 2, note 3 (see section 3.1.4). Except as noted in section 3.1.9, prevent the indoor air supplementary heating coils from operating during all tests. For coil-only indoor units that are supplied without an enclosure, create an enclosure using 1 inch fiberglass ductboard having a nominal density of 6 pounds per cubic foot. Or alternatively, use some other insulating material having a thermal resistance (''R'' value) between 4 and 6 hr·ft^2· °F/Btu. For units where the coil is housed within an enclosure or cabinet, no extra insulating or sealing is allowed.

2.2.1 Defrost control settings. Set heat pump defrost controls at the normal settings which most typify those encountered in generalized climatic region IV. (Refer to Figure 2 and Table 17 of section 4.2 for information on region IV.) For heat pumps that use a time-adaptive defrost control system (see Definition 1.42), the manufacturer must specify the frosting interval to be used during Frost Accumulation tests and provide the procedure for manually initiating the defrost at the specified time. To ease testing of any unit, the manufacturer should provide information and any necessary hardware to manually initiate a defrost cycle.

2.2.2 Special requirements for units having a multiple-speed outdoor fan. Configure the multiple-speed outdoor fan according to the manufacturer's specifications, and thereafter, leave it unchanged for all tests. The controls of the unit must regulate the operation of the outdoor fan during all lab tests except dry coil cooling mode tests. For dry coil cooling mode tests, the outdoor fan must operate at the same speed used during the required wet coil test conducted at the same outdoor test conditions.

2.2.3 Special requirements for multi-split air conditioners and heat pumps, and systems composed of multiple mini-split units (outdoor units located side-by-side) that would normally operate using two or more indoor thermostats. For any test where the system is operated at part load (i.e., one or more compressors ''off'', operating at the intermediate or minimum compressor speed, or at low compressor capacity), the manufacturer shall designate the particular indoor coils that are turned off during the test. For variable-speed systems, the manufacturer must designate at least one indoor unit that is turned off for all tests conducted at minimum compressor speed. For all other part-load tests, the manufacturer shall choose to turn off zero, one, two, or more indoor units. The chosen configuration shall remain unchanged for all tests conducted at the same compressor speed/capacity. For any indoor coil that is turned off during a test, take steps to cease forced airflow through this indoor coil and block its outlet duct. Because these types of systems will have more than one indoor fan and possibly multiple outdoor fans and compressor systems, references in this test procedure to a single indoor fan, outdoor fan, and compressor means all indoor fans, all outdoor fans, and all compressor systems that are turned on during the test.

2.2.4 Wet-bulb temperature requirements for the air entering the indoor and outdoor coils.

2.2.4.1 Cooling mode tests. For wet-coil cooling mode tests, regulate the water vapor content of the air entering the indoor unit to the applicable wet-bulb temperature listed in Tables 3 to 6. As noted in these same tables, achieve a wet-bulb temperature during dry-coil cooling mode tests that results in no condensate forming on the indoor coil. Controlling the water vapor content of the air entering the outdoor side of the unit is not required for cooling mode tests except when testing:

(1) Units that reject condensate to the outdoor coil during wet coil tests. Tables 3–6 list the applicable wet-bulb temperatures.

(2) Single-packaged units where all or part of the indoor section is located in the outdoor test room. The average dew point temperature of the air entering the outdoor coil during wet coil tests must be within ±3.0 °F of the average dew point temperature of the air entering the indoor coil over the 30-minute data collection interval described in section 3.3. For dry coil tests on such units, it may be necessary to limit the moisture content of the air entering the outdoor side of the unit to meet the requirements of section 3.4.

2.2.4.2 Heating mode tests. For heating mode tests, regulate the water vapor content of the air entering the outdoor unit to the applicable wet-bulb temperature listed in Tables 9 to 12. The wet-bulb temperature entering the indoor side of the heat pump must not exceed 60 °F. Additionally, if the Outdoor Air Enthalpy test method is used while testing a single-packaged heat pump where all or part of the outdoor section is located in the indoor test room, adjust the wet-bulb temperature for the air entering the indoor side to yield an indoor-side dew point temperature that is as close as reasonably possible to the dew point temperature of the outdoor-side entering air.

2.2.5 Additional refrigerant charging requirements. Charging according to the ''manufacturer's published instructions,'' as stated in section 8.2 of ASHRAE Standard 37–2005 (incorporated by reference, see §430.22), means the manufacturer's installation instructions that come packaged with the unit.If a unit requires charging but the installation instructions do not specify a charging procedure, then evacuate the unit and add the nameplate refrigerant charge. Where the manufacturer's installation instructions contain two sets of refrigerant charging criteria, one for field installations and one for lab testing, use the field installation criteria. For third-party testing, the test laboratory may consult with the manufacturer about the refrigerant charging procedure and make any needed corrections so long as they do not contradict the published installation instructions. The manufacturer may specify an alternative charging criteria to the third-party laboratory so long as the manufacturer thereafter revises the published installation instructions accordingly.

2.3 Indoor air volume rates. If a unit's controls allow for overspeeding the indoor fan (usually on a temporary basis), take the necessary steps to prevent overspeeding during all tests.

2.3.1 Cooling tests. a. Set indoor fan control options (*e.g.*, fan motor pin settings, fan motor speed) according to the published installation instructions that are provided with the equipment while meeting the airflow requirements that are specified in sections 3.1.4.1 to 3.1.4.3.

b. Express the Cooling Full-load Air Volume Rate, the Cooling Minimum Air Volume Rate, and the Cooling Intermediate Air Volume Rate in terms of standard air.

2.3.2 Heating tests. a. If needed, set the indoor fan control options (*e.g.*, fan motor pin settings, fan motor speed) according to the published installation instructions that are provided with the equipment. Do this set-up while meeting all applicable airflow requirements specified in sections 3.1.4.4 to 3.1.4.7.

b. Express the Heating Certified Air Volume Rate, the Heating Minimum Air Volume Rate, the Heating Intermediate Air Volume Rate, and the Heating Nominal Air Volume Rate in terms of standard air.

2.4 Indoor coil inlet and outlet duct connections. Insulate and/or construct the outlet plenum described in section 2.4.1 and, if installed, the inlet plenum described in section 2.4.2 with thermal insulation having a nominal overall sistance (R-value) of at least 19 hr·ft^{2}· °F/Btu.

2.4.1 Outlet plenum for the indoor unit. a. Attach a plenum to the outlet of the indoor coil. (NOTE: for some packaged systems, the indoor coil may be located in the outdoor test room.)

b. For systems having multiple indoor coils, attach a plenum to each indoor coil outlet. Connect two or more outlet plenums to a single common duct so that each indoor coil ultimately connects to an airflow measuring apparatus (section 2.6). If using more than one indoor test room, do likewise, creating one or more common ducts within each test room that contains multiple indoor coils. At the plane where each plenum enters a common duct, install an adjustable airflow damper and use it to equalize the static pressure in each plenum. Each outlet air temperature grid (section 2.5.4) and airflow measuring apparatus are located downstream of the inlet(s) to the common duct.

c. For small-duct, high-velocity systems, install an outlet plenum that has a diameter that is equal to or less than the value listed below. The limit depends only on the cooling Full-Load Air Volume Rate (see section 3.1.4.1.1) and is effective regardless of the flange dimensions on the outlet of the unit (or an air supply plenum adapter accessory, if installed in accordance with the manufacturer's installation instructions).

d. Add a static pressure tap to each face of the (each) outlet plenum, if rectangular, or at four evenly distributed locations along the circumference of an oval or round plenum. Create a manifold that connects the four static pressure taps. Figure 1 shows two of the three options allowed for the manifold configuration; the third option is the broken-ring, four-to-one manifold configuration that is shown in Figure 7a of ASHRAE Standard 37–2005 (incorporated by reference, see §430.22). See Figures 7a, 7b, 7c, and 8 of ASHRAE Standard 37–2005 (incorporated by reference, see §430.22) for the cross-sectional dimensions and minimum length of the (each) plenum and the locations for adding the static pressure taps for units tested with and without an indoor fan installed.

Cooling full-load air volume rate (scfm)	Maximum diameter* of outlet plenum (inches)
≤500	6
501 to 700	7
701 to 900	8
901 to 1100	9
1101 to 1400	10
1401 to 1750	11

*If the outlet plenum is rectangular, calculate its equivalent diameter using (4*A/P*,) where *A* is the area and *P* is the perimeter of the rectangular plenum, and compare it to the listed maximum diameter.

2.4.2 Inlet plenum for the indoor unit. Install an inlet plenum when testing a coil-only indoor unit or a packaged system where the indoor coil is located in the outdoor test room. Add static pressure taps at the center of each face of this plenum, if rectangular, or at four evenly distributed locations along the circumference of an oval or round plenum. Make a manifold that connects the four static-pressure taps using one of the three configurations specified in section 2.4.1. See Figures 7b, 7c, and Figure 8 of ASHRAE Standard 37–2005 (incorporated by reference, see §430.22) for cross-sectional dimensions, the minimum length of the inlet plenum, and the locations of the static-pressure taps. When testing a ducted unit having an indoor fan (and the indoor coil is in the indoor test room), the manufacturer has the option to test with or without an inlet plenum installed. Space limitations within the test room may dictate that the manufacturer choose the latter option. If used, construct the inlet plenum and add the four static-pressure taps as shown in Figure 8 of ASHRAE Standard 37–2005 (incorporated by reference, see §430.22). Manifold the four static-pressure taps using one of the three configurations specified in section 2.4.1. Never use an inlet plenum when testing a non-ducted system.

2.5 Indoor coil air property measurements and air damper box applications. a. Measure the dry-bulb temperature and water vapor content of the air entering and leaving the indoor coil. If needed, use an air sampling device to divert air to a sensor(s) that measures the water vapor content of the air. See Figure 2 of ASHRAE Standard 41.1–86 (RA 01) (incorporated by reference, see §430.22) for guidance on constructing an air sampling device. The sampling device may also divert air to a remotely located sensor(s) that measures dry bulb temperature. The air sampling device and the remotely located temperature sensor(s) may be used to determine the entering air dry bulb temperature during any test. The air sampling device and the remotely located leaving air dry bulb temperature sensor(s) may be used for all tests except:

(1) Cyclic tests; and

(2) Frost accumulation tests.

b. An acceptable alternative in all cases, including the two special cases noted above, is to install a grid of dry bulb temperature sensors within the outlet and inlet ducts. Use a temperature grid to get the average dry bulb temperature at one location, leaving or entering, or when two grids are applied as a thermopile, to directly obtain the temperature difference. A grid of temperature sensors (which may also be used for determining average leaving air dry bulb temperature) is required to measure the temperature distribution within a cross-section of the leaving airstream.

c. Use an inlet and outlet air damper box when testing ducted systems if conducting one or both of the cyclic tests listed in sections 3.2 and 3.6. Otherwise, install an outlet air damper box when testing heat pumps, both ducted and non-ducted, that cycle off the indoor fan during defrost cycles if no other means is available for preventing natural or forced convection through the indoor unit when the indoor fan is off. Never use an inlet damper box when testing a non-ducted system.

2.5.1 Test set-up on the inlet side of the indoor coil: for cases where the inlet damper box is installed. a. Install the inlet side damper box as specified in section 2.5.1.1 or 2.5.1.2, whichever applies. Insulate or construct the ductwork between the point where the air damper is installed and where the connection is made to either the inlet plenum (section 2.5.1.1 units) or the indoor unit (section 2.5.1.2 units) with thermal insulation that has a nominal overall resistance (R-value) of at least 19 hr·ft^2· °F/Btu.

b. Locate the grid of entering air dry-bulb temperature sensors, if used, at the inlet of the damper box. Locate the air sampling device, or the sensor used to measure the water vapor content of the inlet air, at a location immediately upstream of the damper box inlet.

2.5.1.1 If the section 2.4.2 inlet plenum is installed. Install the inlet damper box upstream of the inlet plenum. The cross-sectional flow area of the damper box must be equal to or greater than the flow area of the inlet plenum. If needed, use an adaptor plate or a transition duct section to connect the damper box with the inlet plenum.

2.5.1.2 If the section 2.4.2 inlet plenum is not installed. Install the damper box immediately upstream of the air inlet of the indoor unit. The cross-sectional dimensions of the damper box must be equal to or greater than the dimensions of the indoor unit inlet. If needed, use an adaptor plate or a short transition duct section to connect the damper box with the unit's air inlet. Add static pressure taps at the center of each face of the damper box, if rectangular, or at four evenly distributed locations along the circumference, if oval or round. Locate the pressure taps between the inlet damper and the inlet of the indoor unit. Make a manifold that connects the four static pressure taps.

2.5.2 Test set-up on the inlet side of the indoor unit: for cases where no inlet damper box is installed. If using the section 2.4.2 inlet plenum and a grid of dry bulb temperature sensors, mount the grid at a location upstream of the static pressure taps described in section 2.4.2, preferably at the entrance plane of the inlet plenum. If the section 2.4.2 inlet plenum is not used, but a grid of dry bulb temperature sensors is used, locate the grid approximately 6 inches upstream from the inlet of the indoor coil. Or, in the case of non-ducted units having multiple indoor coils, locate a grid approximately 6 inches upstream from the inlet of each indoor coil. Position an air sampling device, or the sensor used to measure the water vapor content of the inlet air, immediately upstream of the (each) entering air dry-bulb temperature sensor grid. If a grid of sensors is not used, position the entering air sampling device (or the sensor used to measure the water vapor content of the inlet air) as if the grid were present.

2.5.3 Indoor coil static pressure difference measurement. Section 6.5.2 of ASHRAE Standard 37–2005 (incorporated by reference, see §430.22) describes the method for fabricating static-pressure taps. Also refer to Figure 2A of ASHRAE Standard 51–99/AMCA Standard 210–99 (incorporated by reference, see §430.22). Use a differential pressure measuring instrument that is accurate to within ±0.01 inches of water and has a resolution of at least 0.01 inches of water to measure the static pressure difference between the indoor coil air inlet and outlet. Connect one side of the differential pressure instrument to the manifolded pressure taps installed in the outlet plenum. Connect the other side of the instrument to the manifolded pressure taps located in either the inlet plenum or incorporated within the air damper box. If an inlet plenum or inlet damper box are not used, leave the inlet side of the differential pressure instrument open to the surrounding atmosphere. For non-ducted systems that are tested with multiple outlet plenums, measure the static pressure within each outlet plenum relative to the surrounding atmosphere.

2.5.4 Test set-up on the outlet side of the indoor coil. a. Install an interconnecting duct between the outlet plenum described in section 2.4.1 and the airflow measuring apparatus described below in section 2.6. The cross-sectional flow area of the interconnecting duct must be equal to or greater than the flow area of the outlet plenum or the common duct used when testing non-ducted units having multiple indoor coils. If needed, use adaptor plates or transition duct sections to allow the connections. To minimize leakage, tape joints within the interconnecting duct (and the outlet plenum). Construct or insulate the entire flow section with thermal insulation having a nominal overall resistance (R-value) of at least 19 $hr \cdot ft^2 \cdot °F/Btu$.

b. Install a grid(s) of dry-bulb temperature sensors inside the interconnecting duct. Also, install an air sampling device, or the sensor(s) used to measure the water vapor content of the outlet air, inside the interconnecting duct. Locate the dry-bulb temperature grid(s) upstream of the air sampling device (or the in-duct sensor(s) used to measure the water vapor content of the outlet air). Air that circulates through an air sampling device and past a remote water-vapor-content sensor(s) must be returned to the interconnecting duct at a point:

(1) Downstream of the air sampling device;

(2) Upstream of the outlet air damper box, if installed; and

(3) Upstream of the section 2.6 airflow measuring apparatus.

2.5.4.1 Outlet air damper box placement and requirements. If using an outlet air damper box (see section 2.5), install it within the interconnecting duct at a location downstream of the location where air from the sampling device is reintroduced or downstream of the in-duct sensor that measures water vapor content of the outlet air. The leakage rate from the combination of the outlet plenum, the closed damper, and the duct section that connects these two components must not exceed 20 cubic feet per minute when a negative pressure of 1 inch of water column is maintained at the plenum's inlet.

2.5.4.2 Procedures to minimize temperature maldistribution. Use these procedures if necessary to correct temperature maldistributions. Install a mixing device(s) upstream of the outlet air, dry-bulb temperature grid (but downstream of the outlet plenum static pressure taps). Use a perforated screen located between the mixing device and the dry-bulb temperature grid, with a maximum open area of 40 percent. One or both items should help to meet the maximum outlet air temperature distribution specified in section 3.1.8. Mixing devices are described in sections 6.3—6.5 of ASHRAE Standard 41.1–86 (RA 01) (incorporated by reference, see §430.22) and section 5.2.2 of ASHRAE Standard 41.2–87 (RA 92) (incorporated by reference, see §430.22).

2.5.4.3 Minimizing air leakage. For small-duct, high-velocity systems, install an air damper near the end of the interconnecting duct, just prior to the transition to the airflow measuring apparatus of section 2.6. To minimize air leakage, adjust this damper such that the pressure in the receiving chamber of the airflow measuring apparatus is no more than 0.5 inch of water higher than the surrounding test room ambient. In lieu of installing a separate damper, use the outlet air damper box of sections 2.5 and 2.5.4.1 if it allows variable positioning. Also apply these steps to any conventional indoor blower unit that creates a static pressure within the receiving chamber of the airflow measuring apparatus that exceeds the test room ambient pressure by more than 0.5 inches of water column.

2.5.5 Dry bulb temperature measurement. a. Measure dry bulb temperatures as specified in sections 4, 5, 6.1–6.10, 9, 10, and 11 of ASHRAE Standard 41.1–86 (RA 01) (incorporated by reference, see §430.22). The transient testing requirements cited in section 4.3 of ASHRAE Standard 41.1–86 (RA 01) (incorporated by reference, see §430.22) apply if conducting a cyclic or frost accumulation test.

b. Distribute the sensors of a dry-bulb temperature grid over the entire flow area. The required minimum is 9 sensors per grid.

2.5.6 Water vapor content measurement. Determine water vapor content by measuring dry-bulb temperature combined with the air wet-bulb temperature, dew point temperature, or relative humidity. If used, construct and apply wet-bulb temperature sensors as specified in sections 4, 5, 6, 9, 10, and 11 of ASHRAE Standard 41.1–86 (RA 01) (incorporated by reference, see §430.22). As specified in ASHRAE 41.1–86 (RA 01) (incorporated by reference, see §430.22), the temperature sensor (wick removed) must be accurate to within ±0.2 °F. If used, apply dew point hygrometers as specified in sections 5 and 8 of ASHRAE Standard 41.6–94 (RA 01) (incorporated by reference, see §430.22). The dew point hygrometers must be accurate to within ±0.4 °F when operated at conditions that result in the evaluation of dew points above 35 °F. If used, a relative humidity (RH) meter must be accurate to within ±0.7% RH. Other means to determine the psychrometric state of air may be used as long as the measurement accuracy is equivalent to or better than the accuracy achieved from using a wet-bulb temperature sensor that meets the above specifications.

2.5.7 Air damper box performance requirements. If used (see section 2.5), the air damper box(es) must be capable of being completely opened or completely closed within 10 seconds for each action.

2.6 Airflow measuring apparatus. a. Fabricate and operate an Air Flow Measuring Apparatus as specified in section 6.6 of ASHRAE Standard 116–95 (RA05) (incorporated by reference, see §430.22). Refer to Figure 12 of ASHRAE Standard 51–99/AMCA Standard 210–99 (incorporated by reference, see §430.22) or Figure 14 of ASHRAE Standard 41.2–87 (RA 92) (incorporated by reference, see §430.22) for guidance on placing the static pressure taps and positioning the diffusion baffle (settling means) relative to the chamber inlet.

b. Connect the airflow measuring apparatus to the interconnecting duct section described in section 2.5.4. See sections 6.1.1, 6.1.2, and 6.1.4, and Figures 1, 2, and 4 of ASHRAE Standard 37–2005 (incorporated by reference, see §430.22), and Figures D1, D2, and D4 of ARI Standard 210/240–2006 (incorporated by reference, see §430.22) for illustrative examples of how the test apparatus may be applied within a complete laboratory set-up. Instead of following one of these examples, an alternative set-up may be used to handle the air leaving the airflow measuring apparatus and to supply properly conditioned air to the test unit's inlet. The alternative set-up, however, must not interfere with the prescribed means for measuring airflow rate, inlet and outlet air temperatures, inlet and outlet water vapor contents, and external static pressures, nor create abnormal conditions surrounding the test unit. (Note: Do not use an enclosure as described in section 6.1.3 of ASHRAE Standard 37–2005 (incorporated by reference, see §430.22) when testing triple-split units.)

2.7 Electrical voltage supply. Perform all tests at the voltage specified in section 6.1.3.2 of ARI Standard 210/240–2006 (incorporated by reference, see §430.22) for ''Standard Rating Tests.'' Measure the supply voltage at the terminals on the test unit using a volt meter that provides a reading that is accurate to within ±1.0 percent of the measured quantity.

2.8 Electrical power and energy measurements. a. Use an integrating power (watt-hour) measuring system to determine the electrical energy or average electrical power supplied to all components of the air conditioner or heat pump (including auxiliary components such as controls, transformers, crankcase heater, integral condensate pump on non-ducted indoor units, etc.). The watt-hour measuring system must give readings that are accurate to within ±0.5 percent. For cyclic tests, this accuracy is required during both the ON and OFF cycles. Use either two different scales on the same watt-hour meter or two separate watt-hour meters. Activate the scale or meter having the lower power rating within 15 seconds after beginning an

OFF cycle. Activate the scale or meter having the higher power rating active within 15 seconds prior to beginning an ON cycle. For ducted units tested with a fan installed, the ON cycle lasts from compressor ON to indoor fan OFF. For ducted units tested without an indoor fan installed, the ON cycle lasts from compressor ON to compressor OFF. For non-ducted units, the ON cycle lasts from indoor fan ON to indoor fan OFF. When testing air conditioners and heat pumps having a variable-speed compressor, avoid using an induction watt/watt-hour meter.

b. When performing section 3.5 and/or 3.8 cyclic tests on non-ducted units, provide instrumentation to determine the average electrical power consumption of the indoor fan motor to within ±1.0 percent. If required according to sections 3.3, 3.4, 3.7, 3.9.1, and/or 3.10, this same instrumentation requirement applies when testing air conditioners and heat pumps having a variable-speed constant-air-volume-rate indoor fan or a variable-speed, variable-air-volume-rate indoor fan.

2.9 Time measurements. Make elapsed time measurements using an instrument that yields readings accurate to within ±0.2 percent.

2.10 Test apparatus for the secondary space conditioning capacity measurement. For all tests, use the Indoor Air Enthalpy Method to measure the unit's capacity. This method uses the test set-up specified in sections 2.4 to 2.6. In addition, for all steady-state tests, conduct a second, independent measurement of capacity as described in section 3.1.1. For split systems, use one of the following secondary measurement methods: Outdoor Air Enthalpy Method, Compressor Calibration Method, or Refrigerant Enthalpy Method. For single packaged units, use either the Outdoor Air Enthalpy Method or the Compressor Calibration Method as the secondary measurement.

2.10.1 Outdoor Air Enthalpy Method. a. To make a secondary measurement of indoor space conditioning capacity using the Outdoor Air Enthalpy Method, do the following:

(1) Measure the electrical power consumption of the test unit;

(2) Measure the air-side capacity at the outdoor coil; and

(3) Apply a heat balance on the refrigerant cycle.

b. The test apparatus required for the Outdoor Air Enthalpy Method is a subset of the apparatus used for the Indoor Air Enthalpy Method. Required apparatus includes the following:

(1) An outlet plenum containing static pressure taps (sections 2.4, 2.4.1, and 2.5.3),

(2) An airflow measuring apparatus (section 2.6),

(3) A duct section that connects these two components and itself contains the instrumentation for measuring the dry-bulb temperature and water vapor content of the air leaving the outdoor coil (sections 2.5.4, 2.5.5, and 2.5.6), and

(4) On the inlet side, a sampling device and optional temperature grid (sections 2.5 and 2.5.2).

c. During the preliminary tests described in sections 3.11.1 and 3.11.1.1, measure the evaporator and condenser temperatures or pressures. On both the outdoor coil and the indoor coil, solder a thermocouple onto a return bend located at or near the midpoint of each coil or at points not affected by vapor superheat or liquid subcooling. Alternatively, if the test unit is not sensitive to the refrigerant charge, connect pressure gages to the access valves or to ports created from tapping into the suction and discharge lines. Use this alternative approach when testing a unit charged with a zeotropic refrigerant having a temperature glide in excess of 1 °F at the specified test conditions.

2.10.2 Compressor Calibration Method. Measure refrigerant pressures and temperatures to determine the evaporator superheat and the enthalpy of the refrigerant that enters and exits the indoor coil. Determine refrigerant flow rate or, when the superheat of the refrigerant leaving the evaporator is less than 5 °F, total capacity from separate calibration tests conducted under identical operating conditions. When using this method, install instrumentation, measure refrigerant properties, and adjust the refrigerant charge according to section 7.4.2 of ASHRAE Standard 37–2005 (incorporated by reference, see §430.22). Use refrigerant temperature and pressure measuring instruments that meet the specifications given in sections 5.1.1 and 5.2 of ASHRAE Standard 37–2005 (incorporated by reference, see §430.22).

2.10.3 Refrigerant Enthalpy Method. For this method, calculate space conditioning capacity by determining the refrigerant enthalpy change for the indoor coil and directly measuring the refrigerant flow rate. Use section 7.5.2 of ASHRAE Standard 37–2005 (incorporated by reference, see §430.22) for the requirements for this method, including the additional instrumentation requirements, and information on placing the flow meter and a sight glass. Use refrigerant temperature, pressure, and flow measuring instruments that meet the specifications given in sections 5.1.1, 5.2, and 5.5.1 of ASHRAE Standard 37–2005 (incorporated by reference, see §430.22).

2.11 Measurement of test room ambient conditions. a. If using a test set-up where air is ducted directly from the conditioning apparatus to the indoor coil inlet (see Figure 2, Loop Air-Enthalpy Test Method Arrangement, of ASHRAE Standard 37–2005 (incorporated by reference, see §430.22)), add instrumentation to permit measurement of the indoor test room dry-bulb temperature.

b. If the Outdoor Air Enthalpy Method is not used, add instrumentation to measure the dry-bulb temperature and the water vapor content of the air entering the outdoor coil. If an air sampling device is used, construct and apply the device as per section 6 of ASHRAE Standard 41.1–86 (RA 01) (incorporated by reference, see §430.22). Take steps (*e.g.*, add or re-position a lab circulating fan), as needed, to minimize the magnitude of the temperature distribution non-uniformity. Position any fan in the outdoor test room while trying to keep air velocities in the vicinity of the test unit below 500 feet per minute.

c. Measure dry bulb temperatures as specified in sections 4, 5, 6.1–6.10, 9, 10, and 11 of ASHRAE Standard 41.1–86 (RA 01) (incorporated by reference, see §430.22). Measure water vapor content as stated above in section 2.5.6.

2.12 Measurement of indoor fan speed. When required, measure fan speed using a revolution counter, tachometer, or stroboscope that gives readings accurate to within ±1.0 percent.

2.13 Measurement of barometric pressure. Determine the average barometric pressure during each test. Use an instrument that meets the requirements specified in section 5.2 of ASHRAE Standard 37–2005(incorporated by reference, see §430.22).

3. Testing Procedures

3.1 General Requirements. If, during the testing process, an equipment set-up adjustment is made that would alter the performance of the unit when conducting an already completed test, then repeat all tests affected by the adjustment. For cyclic tests, instead of maintaining an air volume rate, for each airflow nozzle, maintain the static pressure difference or velocity pressure during an ON period at the same pressure difference or velocity pressure as measured during the steady-state test conducted at the same test conditions.

3.1.1 Primary and secondary test methods. For all tests, use the Indoor Air Enthalpy Method test apparatus to determine the unit's space conditioning capacity. The procedure and data collected, however, differ slightly depending upon whether the test is a steady-state test, a cyclic test, or a frost accumulation test. The following sections described these differences. For all steady-state tests (*i.e.*, the A, A_2, A_1, B, B_2, B_1, C, C_1, EV, F_1, G_1, $H0_1$, H_1, $H1_2$, $H1_1$, HI_N, H_3, $H3_2$, and $H3_1$ Tests), in addition, use one of the acceptable secondary methods specified in section 2.10 to determine indoor space conditioning capacity. Calculate this secondary check of capacity according to section 3.11. The two capacity measurements must agree to within 6 percent to constitute a valid test. For this capacity comparison, use the Indoor Air Enthalpy Method capacity that is calculated in section 7.3 of ASHRAE Standard 37–2005 (incorporated by reference, see §430.22) (and, if testing a coil-only unit, do not make the after-test fan heat adjustments described in section 3.3, 3.4, 3.7, and 3.10 of this appendix). However, include the appropriate section 3.3 to 3.5 and 3.7 to 3.10 fan heat adjustments within the Indoor Air Enthalpy Method capacities used for the section 4 seasonal calculations.

3.1.2 Manufacturer-provided equipment overrides. Where needed, the manufacturer must provide a means for overriding the controls of the test unit so that the compressor(s) operates at the specified speed or capacity and the indoor fan operates at the specified speed or delivers the specified air volume rate.

3.1.3 Airflow through the outdoor coil. For all tests, meet the requirements given in section 6.1.3.4 of ARI Standard 210/240–2006(incorporated by reference, see §430.22) when obtaining the airflow through the outdoor coil.

3.1.4 Airflow through the indoor coil.

3.1.4.1 Cooling Full-load Air Volume Rate.

3.1.4.1.1 Cooling Full-Load Air Volume Rate for Ducted Units. The manufacturer must specify the Cooling Full-load Air Volume Rate. Use this value as long as the following two requirements are satisfied. First, when conducting the A or A_2 Test (exclusively), the measured air volume rate, when divided by the measured indoor air-side total cooling capacity must not exceed 37.5 cubic feet per minute of standard air (scfm) per 1000 Btu/h. If this ratio is exceeded, reduce the air volume rate until this ratio is equaled. Use this reduced air volume rate for all tests that call for using the Cooling Full-load Air Volume Rate. The second requirement is as follows:

a. For all ducted units tested with an indoor fan installed, except those having a variable-speed, constant-air-volume-rate indoor fan. The second requirement applies exclusively to the A or A_2 Test and is met as follows.

1. Achieve the Cooling Full-load Air Volume Rate, determined in accordance with the previous paragraph;

2. Measure the external static pressure;

3. If this pressure is equal to or greater than the applicable minimum external static pressure cited in Table 2, this second requirement is satisfied. Use the current air volume rate for all tests that require the Cooling Full-load Air Volume Rate.

4. If the Table 2 minimum is not equaled or exceeded,

4a. reduce the air volume rate until the applicable Table 2 minimum is equaled or

4b. until the measured air volume rate equals 95 percent of the air volume rate from step 1, whichever occurs first.

5. If the conditions of step 4a occur first, this second requirement is satisfied. Use the

step 4a reduced air volume rate for all tests that require the Cooling Full-load Air Volume Rate.

6. If the conditions of step 4b occur first, make an incremental change to the set-up of the indoor fan (e.g., next highest fan motor pin setting, next highest fan motor speed) and repeat the evaluation process beginning at above step 1. If the indoor fan set-up cannot be further changed, reduce the air volume rate until the applicable Table 2 minimum is equaled. Use this reduced air volume rate for all tests that require the Cooling Full-load Air Volume Rate.

b. For ducted units that are tested with a variable-speed, constant-air-volume-rate indoor fan installed.For all tests that specify the Cooling Full-load Air Volume Rate, obtain an external static pressure as close to (but not less than) the applicable Table 2 value that does not cause instability or an automatic shutdown of the indoor blower.

c. For ducted units that are tested without an indoor fan installed. For the A or A_2 Test, (exclusively), the pressure drop across the indoor coil assembly must not exceed 0.30 inches of water. If this pressure drop is exceeded, reduce the air volume rate until the measured pressure drop equals the specified maximum. Use this reduced air volume rate for all tests that require the Cooling Full-loadAir Volume Rate.

TABLE 2—MINIMUM EXTERNAL STATIC PRESSURE FOR DUCTED SYSTEMS TESTED WITH AN INDOOR FAN INSTALLED

Rated Cooling [1] or Heating [2] Capacity (Btu/h)	Minimum external resistance [3] (Inches of water)	
	All other systems	Small-duct, high-velocity systems [4,5]
Up Thru 28,800	0.10	1.10
29,000 to 42,500	0.15	1.15
43,000 and Above	0.20	1.20

[1] For air conditioners and heat pumps, the value cited by the manufacturer in published literature for the unit's capacity when operated at the *A* or A_2 Test conditions.

[2] For heating-only heat pumps, the value the manufacturer cites in published literature for the unit's capacity when operated at the *H1* or $H1_2$ Test conditions.

[3] For ducted units tested without an air filter installed, increase the applicable tabular value by 0.08 inch of water.

[4] See Definition 1.35 to determine if the equipment qualifies as a small-duct, high-velocity system.

[5] If a closed-loop, air-enthalpy test apparatus is used on the indoor side, limit the resistance to airflow on the inlet side of the indoor blower coil to a maximum value of 0.1 inch of water. Impose the balance of the airflow resistance on the outlet side of the indoor blower.

3.1.4.1.2 Cooling Full-load Air Volume Rate for Non-ducted Units. For non-ducted units, the Cooling Full-load Air Volume Rate is the air volume rate that results during each test when the unit is operated at an external static pressure of zero inches of water.

3.1.4.2 Cooling Minimum Air Volume Rate. a. For ducted units that regulate the speed (as opposed to the cfm) of the indoor fan,

$$\text{Cooling Minimum Air Vol. Rate} = \text{Cooling Full-load Air Vol. Rate} \times \frac{\text{Cooling Minimum Fan Speed}}{\text{A}_2\text{Test Fan Speed}},$$

where "Cooling Minimum Fan Speed" corresponds to the fan speed used when operating at low compressor capacity (two-capacity system), the fan speed used when operating at the minimum compressor speed (variable-speed system), or the lowest fan speed used when cooling (single-speed compressor and a variable-speed variable-air-volume-rate indoor fan). For such systems, obtain the Cooling Minimum Air Volume Rate regardless of the external static pressure.

b. For ducted units that regulate the air volume rate provided by the indoor fan, the manufacturer must specify the Cooling Minimum Air Volume Rate. For such systems, conduct all tests that specify the Cooling Minimum Air Volume Rate—(*i.e.*, the A_1, B_1, C_1, F_1, and G_1 Tests)—at an external static pressure that does not cause instability or an automatic shutdown of the indoor blower while being as close to, but not less than,

$$A_1,\ B_1,\ C_1,\ F_1,\ \&\ G_1\ \text{Test}\ \Delta P_{st} = \Delta P_{st,\ A_2} \times \left[\frac{\text{Cooling Minimum Air Volume Rate}}{\text{Cooling Full-load Air Volume Rate}}\right]^2,$$

where $\Delta P_{st,A_2}$ is the applicable Table 2 minimum external static pressure that was targeted during the A_2 (and B_2) Test.

c. For ducted two-capacity units that are tested without an indoor fan installed, the Cooling Minimum Air Volume Rate is the higher of (1) the rate specified by the manufacturer or (2) 75 percent of the Cooling Full-load Air Volume Rate. During the laboratory tests on a coil-only (fanless) unit, obtain this Cooling Minimum Air Volume Rate regardless of the pressure drop across the indoor coil assembly.

d. For non-ducted units, the Cooling Minimum Air Volume Rate is the air volume rate that results during each test when the unit operates at an external static pressure of zero inches of water and at the indoor fan setting used at low compressor capacity (two-capacity system) or minimum compressor speed (variable-speed system). For units having a single-speed compressor and a variable-speed variable-air-volume-rate indoor fan, use the lowest fan setting allowed for cooling.

3.1.4.3 Cooling Intermediate Air Volume Rate. a. For ducted units that regulate the speed of the indoor fan,

$$\text{Cooling Intermediate Air Vol. Rate} = \text{Cooling Full-load Air Vol. Rate} \times \frac{E_V \text{Test Fan Speed}}{A_2 \text{Test Fan Speed}},$$

For such units, obtain the Cooling Intermediate Air Volume Rate regardless of the external static pressure.

b. For ducted units that regulate the air volume rate provided by the indoor fan, the manufacturer must specify the Cooling Intermediate Air Volume Rate. For such systems, conduct the E_V Test at an external static pressure that does not cause instability or an automatic shutdown of the indoor blower while being as close to, but not less than,

$$E_V \text{Test } \Delta P_{st} = \Delta P_{st,\, A_2} \times \left[\frac{\text{Cooling Intermediate Air Volume Rate}}{\text{Cooling Full-load Air Volume Rate}}\right]^2,$$

where $\Delta P_{st,A_2}$ is the applicable Table 2 minimum external static pressure that was targeted during the A_2 (and B_2) Test.

c. For non-ducted units, the Cooling Intermediate Air Volume Rate is the air volume rate that results when the unit operates at an external static pressure of zero inches of water and at the fan speed selected by the controls of the unit for the E_V Test conditions.

3.1.4.4 Heating Full-load Air Volume Rate.

3.1.4.4.1 Ducted heat pumps where the Heating and Cooling Full-load Air Volume Rates are the same. a. Use the Cooling Full-load Air Volume Rate as the Heating Full-loadAir Volume Rate for:

1. Ducted heat pumps that operate at the same indoor fan speed during both the A (or A_2) and the H1 (or $H1_2$) Tests;

2. Ducted heat pumps that regulate fan speed to deliver the same constant air volume rate during both the A (or A_2) and the H1 (or $H1_2$) Tests; and

3. Ducted heat pumps that are tested without an indoor fan installed (except two-capacity northern heat pumps that are tested only at low capacity cooling—see 3.1.4.4.2).

b. For heat pumps that meet the above criteria "1" and "3," no minimum requirements apply to the measured external or internal, respectively, static pressure. For heat pumps that meet the above criterion "2," test at an external static pressure that does not cause instability or an automatic shutdown of the indoor blower while being as close to, but not less than, the same Table 2 minimum external static pressure as was specified for the A (or A_2) cooling mode test.

3.1.4.4.2 Ducted heat pumps where the Heating and Cooling Full-loadAir Volume Rates are different due to indoor fan operation. a. For ducted heat pumps that regulate the speed (as opposed to the cfm) of the indoor fan,

$$\text{Heating Full-load Air Volume Rate} = \text{Cooling Full-load Air Volume Rate} \times \frac{\text{H1 or H1}_2 \text{ Test Fan Speed}}{\text{A or A}_2 \text{ Test Fan Speed}},$$

For such heat pumps, obtain the Heating Full-loadAir Volume Rate without regard to the external static pressure.

b. For ducted heat pumps that regulate the air volume rate delivered by the indoor fan, the manufacturer must specify the Heating Full-load Air Volume Rate. For such heat pumps, conduct all tests that specify the Heating Full-load Air Volume Rate at an external static pressure that does not cause instability or an automatic shutdown of the indoor blower while being as close to, but not less than,

$$\text{Heating Full-load } \Delta P_{st} = \text{Cooling Full-load } \Delta P_{st} \times \left[\frac{\text{Heating Full-load Air Volume Rate}}{\text{Cooling Full-load Air Volume Rate}}\right]^2,$$

where the Cooling Certified ΔP_{st} is the applicable Table 2 minimum external static pressure that was specified for the A or A_2 Test.

c. When testing ducted, two-capacity northern heat pumps (see Definition 1.46), use the appropriate approach of the above two cases for units that are tested with an indoor fan installed. For coil-only (fanless) northern heat pumps, the Heating Full-load Air Volume Rate is the lesser of the rate specified by the manufacturer or 133 percent of the Cooling Full-load Air Volume Rate. For this latter case, obtain the Heating Full-load Air Volume Rate regardless of the pressure drop across the indoor coil assembly.

3.1.4.4.3 Ducted heating-only heat pumps. The manufacturer must specify the Heating Full-load Air Volume Rate.

a. For all ducted heating-only heat pumps tested with an indoor fan installed, except those having a variable-speed, constant-air-volume-rate indoor fan. Conduct the following steps only during the first test, the H1 or $H1_2$ Test.

1. Achieve the Heating Full-load Air Volume Rate.

2. Measure the external static pressure.

3. If this pressure is equal to or greater than the Table 2 minimum external static pressure that applies given the heating-only heat pump's rated heating capacity, use the current air volume rate for all tests that require the Heating Full-load Air Volume Rate.

4. If the Table 2 minimum is not equaled or exceeded,

4a. reduce the air volume rate until the applicable Table 2 minimum is equaled or

4b. until the measured air volume rate equals 95 percent of the manufacturer-specified Full-load Air Volume Rate, whichever occurs first.

5. If the conditions of step 4a occurs first, use the step 4a reduced air volume rate for all tests that require the Heating Full-load Air Volume Rate.

6. If the conditions of step 4b occur first, make an incremental change to the set-up of the indoor fan (e.g., next highest fan motor pin setting, next highest fan motor speed) and repeat the evaluation process beginning at above step 1. If the indoor fan set-up cannot be further changed, reduce the air volume rate until the applicable Table 2 minimum is equaled. Use this reduced air volume rate for all tests that require the Heating Full-load Air Volume Rate.

b.For ducted heating-only heat pumps that are tested with a variable-speed, constant-air-volume-rate indoor fan installed. For all tests that specify the Heating Full-load Air Volume Rate, obtain an external static pressure that does not cause instability or an automatic shutdown of the indoor blower while being as close to, but not less than, the applicable Table 2 minimum.

c. For ducted heating-only heat pumps that are tested without an indoor fan installed. For the H1 or $H1_2$ Test, (exclusively), the pressure drop across the indoor coil assembly must not exceed 0.30 inches of water. If this pressure drop is exceeded, reduce the air volume rate until the measured pressure drop equals the specified maximum. Use this reduced air volume rate for all tests that require the Heating Full-load Air Volume Rate.

3.1.4.4.4 Non-ducted heat pumps, including non-ducted heating-only heat pumps. For non-ducted heat pumps, the Heating Full-loadAir Volume Rate is the air volume rate that results during each test when the unit operates at an external static pressure of zero inches of water.

3.1.4.5 Heating Minimum Air Volume Rate. a. For ducted heat pumps that regulate the speed (as opposed to the cfm) of the indoor fan,

$$\text{Heating Minimum Air Vol. Rate} = \text{Heating Full-load Air Vol. Rate} \times \frac{\text{Heating Minimum Fan Speed}}{H1_2 \text{Test Fan Speed}},$$

where ''Heating Minimum Fan Speed'' corresponds to the fan speed used when operating at low compressor capacity (two-capacity system), the lowest fan speed used at any time when operating at the minimum compressor speed (variable-speed system), or the lowest fan speed used when heating (single-speed compressor and a variable-speed variable-air-volume-rate indoor fan). For such heat pumps, obtain the Heating Minimum Air Volume Rate without regard to the external static pressure.

b. For ducted heat pumps that regulate the air volume rate delivered by the indoor fan, the manufacturer must specify the Heating Minimum Air Volume Rate. For such heat pumps, conduct all tests that specify the Heating Minimum Air Volume Rate—(*i.e.*, the $H0_1$, $H1_1$, $H2_1$, and $H3_1$ Tests)—at an external static pressure that does not cause instability or an automatic shutdown of the indoor blower while being as close to, but not less than,

$$H0_1,\ H1_1,\ H2_1,\ H3_1,\ \text{Test } \Delta P_{st} = \Delta P_{st,\,H1_2} \times \left[\frac{\text{Htg Minimum Air Vol. Rate}}{\text{Htg Full-load Air Vol. Rate}}\right]^2,$$

where $\Delta P_{st,H1_2}$

is the minimum external static pressure that was targeted during the $H1_2$ Test.

c. For ducted two-capacity northern heat pumps that are tested with an indoor fan installed, use the appropriate approach of the above two cases.

d. For ducted two-capacity heat pumps that are tested without an indoor fan installed, use the Cooling Minimum Air Volume Rate as the Heating Minimum Air Volume Rate. For ducted two-capacity northern heat pumps that are tested without an indoor fan installed, use the Cooling Full-load Air Volume Rate as the Heating Minimum Air Volume Rate. For ducted two-capacity heating-only heat pumps that are tested without an indoor fan installed, the Heating Minimum Air Volume Rate is the higher of the rate specified by the manufacturer or 75 percent of the Heating Full-load Air Volume Rate. During the laboratory tests on a coil-only (fanless) unit, obtain the Heating Minimum Air Volume Rate without regard to the pressure drop across the indoor coil assembly.

e. For non-ducted heat pumps, the Heating Minimum Air Volume Rate is the air volume rate that results during each test when the unit operates at an external static pressure of zero inches of water and at the indoor fan setting used at low compressor capacity (two-capacity system) or minimum compressor speed (variable-speed system). For units having a single-speed compressor and a variable-speed, variable-air-volume-rate indoor fan, use the lowest fan setting allowed for heating.

3.1.4.6 Heating Intermediate Air Volume Rate. a. For ducted heat pumps that regulate the speed of the indoor fan,

$$\text{Heating Intermediate Air Volume Rate} = \text{Heating Full-load Air Volume Rate} \times \frac{H2_V \text{ Test Fan Speed}}{H1_2 \text{ Test Fan Speed}},$$

For such heat pumps, obtain the Heating Intermediate Air Volume Rate without regard to the external static pressure.

b. For ducted heat pumps that regulate the air volume rate delivered by the indoor fan, the manufacturer must specify the Heating Intermediate Air Volume Rate. For such heat pumps, conduct the $H2_V$ Test at an external static pressure that does not cause instability or an automatic shutdown of the indoor blower while being as close to, but not less than,

$$H2_V\ \text{ Test } \Delta P_{st} \Delta P_{st,\,H1_2} \times \left[\frac{\text{Heating Intermediate Air Volume Rate}}{\text{Heating Full-load Air Volume Rate}}\right]^2,$$

where $\Delta P_{st,H1_2}$

is the minimum external static pressure that was specified for the $H1_2$ Test.

c. For non-ducted heat pumps, the Heating Intermediate Air Volume Rate is the air volume rate that results when the heat pump operates at an external static pressure of zero inches of water and at the fan speed selected by the controls of the unit for the $H2_V$ Test conditions.

3.1.4.7 Heating Nominal Air Volume Rate. Except for the noted changes, determine the Heating Nominal Air Volume Rate using the approach described in section 3.1.4.6. Required changes include substituting "$H1_N$ Test" for $H2_V$ Test" within the first section 3.1.4.6 equation, substituting "$H1_N$ Test ΔP_{st}" for "$H2_V$ Test ΔP_{st}" in the second section 3.1.4.6 equation, substituting "$H1_N$ Test" for each "$H2_V$ Test", and substituting "Heating Nominal Air Volume Rate" for each "Heating Intermediate Air Volume Rate."

$$\text{Heating Nominal Air Volume Rate} = \text{Heating Full-load Air Volume Rate} \times \frac{H1_N \text{Test Fan Speed}}{H1_2 \text{Test Fan Speed}},$$

$$H1_N \text{ Test } \Delta P_{st} = \Delta P_{st,\, H1_2} \times \left[\frac{\text{Heating Nominal Air Volume Rate}}{\text{Heating Full-load Air Volume Rate}}\right]^2,$$

3.1.5 Indoor test room requirement when the air surrounding the indoor unit is not supplied from the same source as the air entering the indoor unit. If using a test set-up where air is ducted directly from the air reconditioning apparatus to the indoor coil inlet (see Figure 2, Loop Air-Enthalpy Test Method Arrangement, of ASHRAE Standard 37–2005) (incorporated by reference, see §430.22), maintain the dry bulb temperature within the test room within ±5.0 °F of the applicable sections 3.2 and 3.6 dry bulb temperature test condition for the air entering the indoor unit.

3.1.6 Air volume rate calculations. For all steady-state tests and for frost accumulation ($H2$, $H2_1$, $H2_2$, $H2_V$) tests, calculate the air volume rate through the indoor coil as specified in sections 7.7.2.1 and 7.7.2.2 of ASHRAE Standard 37–2005 (incorporated by reference, see §430.22). When using the Outdoor Air Enthalpy Method, follow sections 7.7.2.1 and 7.7.2.2 to calculate the air volume rate through the outdoor coil. To express air volume rates in terms of standard air, use:

$$\bar{\dot{V}}_s = \frac{\bar{\dot{V}}_{mx}}{0.075\,\frac{lbm_{da}}{ft^3}\cdot v_n^{'}\cdot\left[1+W_n\right]} = \frac{\bar{\dot{V}}_{mx}}{0.075\,\frac{lbm_{da}}{ft^3}\cdot v_n} \qquad (3\text{-}1)$$

where,

$\bar{\dot{V}}_s$ = air volume rate of standard (dry) air, $(ft^3/min)_{da}$

$\bar{\dot{V}}_{mx}$ = air volume rate of the air-water vapor mixture, $(ft^3/min)_{mx}$

v_n' = specific volume of air-water vapor mixture at the nozzle, ft^3 per lbm of the air-water vapor mixture

W_n = humidity ratio at the nozzle, lbm of water vapor per lbm of dry air

0.075 = the density associated with standard (dry) air, (lbm/ft^3)

v_n = specific volume of the dry air portion of the mixture evaluated at the dry-bulb temperature, vapor content, and barometric pressure existing at the nozzle, ft^3 per lbm of dry air.

(Note: In the first printing of ASHRAE Standard 37–2005, the second IP equation for Q_{mi} should read,

$$1097CA_n\sqrt{P_v v_n'}\,.)***$$

3.1.7 Test sequence. When testing a ducted unit (except if a heating-only heat pump), conduct the A or A_2 Test first to establish the Cooling Full-load Air Volume Rate. For ducted heat pumps where the Heating and Cooling Full-loadAir Volume Rates are different, make the first heating mode test one that requires the Heating Full-load Air Volume Rate. For ducted heating-only heat pumps, conduct the H1 or $H1_2$ Test first to establish the Heating Full-load Air Volume Rate. When conducting an optional cyclic test, always conduct it immediately after the steady-state test that requires the same test conditions. For variable-speed systems, the first test using the Cooling Minimum Air Volume Rate should precede the E_V Test if one expects to adjust the indoor fan control options when preparing for the first Minimum Air Volume Rate test. Under the same circumstances, the first test using the Heating Minimum Air Volume Rate should precede the $H2_V$ Test. The test laboratory makes all other decisions on the test sequence.

3.1.8 Requirement for the air temperature distribution leaving the indoor coil. For at least the first cooling mode test and the first heating mode test, monitor the temperature distribution of the air leaving the indoor coil using the grid of individual sensors described in sections 2.5 and 2.5.4. For the 30-minute data collection interval used to determine capacity, the maximum spread among the outlet dry bulb temperatures from any data sampling must not exceed 1.5 °F. Install the mixing devices described in section 2.5.4.2 to minimize the temperature spread.

3.1.9 Control of auxiliary resistive heating elements. Except as noted, disable heat pump resistance elements used for heating indoor air at all times, including during defrost cycles and if they are normally regulated by a heat comfort controller. For heat pumps equipped with a heat comfort controller, enable the heat pump resistance elements only during the below-described, short test. For single-speed heat pumps covered under section 3.6.1, the short test follows the H1 or, if conducted, the H1C Test. For two-capacity heat pumps and heat pumps covered under section 3.6.2, the short test follows the $H1_2$ Test. Set the heat comfort controller to provide the maximum supply air temperature. With the heat pump operating and while maintaining the Heating Full-loadAir Volume Rate, measure the temperature of the air leaving the indoor-side beginning 5 minutes after activating the heat comfort controller. Sample the outlet dry-bulb temperature at regular intervals that span 5 minutes or less. Collect data for 10 minutes, obtaining at least 3 samples. Calculate the average outlet temperature over the 10-minute interval, T_{CC}.

3.2 Cooling mode tests for different types of air conditioners and heat pumps.

3.2.1 Tests for a unit having a single-speed compressor that is tested with a fixed-speed indoor fan installed, with a constant-air-volume-rate indoor fan installed, or with no indoor fan installed. Conduct two steady-state wet coil tests, the A and B Tests. Use the two optional dry-coil tests, the steady-state C Test and the cyclic D Test, to determine the cooling mode cyclic degradation coefficient, C_D^c. If the two optional tests are conducted but yield a tested C_D^c that exceeds the default C_D^c or if the two optional tests are not conducted, assign C_D^c the default value of 0.25. Table 3 specifies test conditions for these four tests.

TABLE 3—COOLING MODE TEST CONDITIONS FOR UNITS HAVING A SINGLE-SPEED COMPRESSOR AND A FIXED-SPEED INDOOR FAN, A CONSTANT AIR VOLUME RATE INDOOR FAN, OR NO INDOOR FAN

Test description	Air entering indoor unit temperature (°F)		Air entering outdoor unit temperature (°F)		Cooling air volume rate
	Dry bulb	Wet bulb	Dry bulb	Wet bulb	
A Test—required (steady, wet coil)	80	67	95	[1] 75	Cooling full-load [2]
B Test—required (steady, wet coil)	80	67	82	[1] 65	Cooling full-load [2]
C Test—optional (steady, dry coil)	80	([3])	82	....	Cooling full-loadthnsp;[2]
D Test—optional (cyclic, dry coil)	80	([3])	82	....	([4])

[1] The specified test condition only applies if the unit rejects condensate to the outdoor coil.
[2] Defined in section 3.1.4.1.
[3] The entering air must have a low enough moisture content so no condensate forms on the indoor coil. (It is recommended that an indoor wet-bulb temperature of 57 °F or less be used.)
[4] Maintain the airflow nozzles static pressure difference or velocity pressure during the ON period at the same pressure difference or velocity pressure as measured during the C Test.

3.2.2 Tests for a unit having a single-speed compressor and a variable-speed variable-air-volume-rate indoor fan installed.

3.2.2.1 Indoor fan capacity modulation that correlates with the outdoor dry bulb temperature. Conduct four steady-state wet coil tests: The A_2, A_1 , B_2, and B_1 Tests. Use the two optional dry-coil tests, the steady-state C_1 Test and the cyclic D_1 Test, to determine the cooling mode cyclic degradation coefficient, C_D^c. If the two optional tests are conducted but yield a tested C_D^c that exceeds the default C_D^c or if the two optional tests are not conducted, assign C_D^c the default value of 0.25

3.2.2.2 Indoor fan capacity modulation based on adjusting the sensible to total (S/T) cooling capacity ratio. The testing requirements are the same as specified in section 3.2.1 and Table 3. Use a Cooling Full-load Air Volume Rate that represents a normal residential installation. If performed, conduct the steady-state C Test and the cyclic D Test with the unit operating in the same S/T capacity control mode as used for the B Test.

TABLE 4—COOLING MODE TEST CONDITIONS FOR UNITS HAVING A SINGLE-SPEED COMPRESSOR AND A VARIABLE AIR VOLUME RATE INDOOR FAN THAT CORRELATES WITH THE OUTDOOR DRY BULB TEMPERATURE (SEC. 3.2.2.1)

Test description	Air entering indoor unit temperature (°F)		Air entering outdoor unit temperature (°F)		Cooling air volume rate
	Dry bulb	Wet bulb	Dry bulb	Wet bulb	
A_2 Test—required (steady, wet coil)	80	67	95	[1] 75	Cooling full-load [2]
A_1 Test—required (steady, wet coil)	80	67	95	[1] 75	Cooling minimum [3]
B_2 Test—required (steady, wet coil)	80	67	82	[1] 65	Cooling full-load [2]
B_1 Test—required (steady, wet coil)	80	67	82	[1] 65	Cooling minimum [3]
C_1 Test [4]—optional (steady, dry coil)	80	([4])	82	....	Cooling minimum [3]
D_1 Test [4]—optional (cyclic, dry coil)	80	([4])	82	([5])	

[1] The specified test condition only applies if the unit rejects condensate to the outdoor coil.
[2] Defined in section 3.1.4.1.
[3] Defined in section 3.1.4.2.
[4] The entering air must have a low enough moisture content so no condensate forms on the indoor coil. (It is recommended that an indoor wet-bulb temperature of 57 °F or less be used.)
[5] Maintain the airflow nozzles static pressure difference or velocity pressure during the ON period at the same pressure difference or velocity pressure as measured during the C_1 Test.

3.2.3 Tests for a unit having a two-capacity compressor. (See Definition 1.45.) a. Conduct four steady-state wet coil tests: the A_2, B_2, B_1, and F_1 Tests. Use the two optional dry-coil tests, the steady-state C_1 Test and the cyclic D_1 Test, to determine the cooling-mode cyclic-degradation coefficient, C_D^c. If the two optional tests are conducted but yield a tested C_D^c that exceeds the default C_D^c or if the two optional tests are not conducted, assign C_D^c the default value of 0.25. Table 5 specifies test conditions for these six tests.

b. For units having a variable speed indoor fan that is modulated to adjust the sensible to total (S/T) cooling capacity ratio, use Cooling Full-load and Cooling Minimum Air Volume Rates that represent a normal residential installation. Additionally, if conducting the optional dry-coil tests, operate the unit in the same S/T capacity control mode as used for the B_1 Test.

c. Test two-capacity, northern heat pumps (see Definition 1.46) in the same way as a single speed heat pump with the unit operating exclusively at low compressor capacity (see section 3.2.1 and Table 3).

d. If a two-capacity air conditioner or heat pump locks out low-capacity operation at higher outdoor temperatures, then use the two optional dry-coil tests, the steady-state C_2 Test and the cyclic D_2 Test, to determine the cooling-mode cyclic-degradation coefficient that only applies to on/off cycling from high capacity, $C_D^c(k=2)$. If the two optional tests are conducted but yield a tested $C_D^c(k=2)$ that exceeds the default $C_D^c(k=2)$ or if the two optional tests are not conducted, assign $C_D^c(k=2)$ the default value. The default $C_D^c(k=2)$ is the same value as determined or assigned for the low-capacity cyclic-degradation coefficient, C_D^c [or equivalently, $C_D^c(k=1)$].

TABLE 5—COOLING MODE TEST CONDITIONS FOR UNITS HAVING A TWO-CAPACITY COMPRESSOR

Test description	Air entering indoor unit temperature (°F)		Air entering outdoor unit temperature (°F)		Compressor capacity	Cooling air volume rate
	Dry bulb	Wet bulb	Dry bulb	Wet bulb		
A_2 Test—required (steady, wet coil)	80	67	95	[1] 75	High	Cooling Full-Load. [2]
B_2 Test—required (steady, wet coil)	80	67	82	[1] 65	High	Cooling Full-Load. [2]
B_1 Test—required (steady, wet coil)	80	67	82	[1] 65	Low	Cooling Minimum. [3]
C_2 Test—optional (steady, dry-coil)	80	([4])	82	High	Cooling Full-Load. [2].	
D_2 Test—optional (cyclic, dry-coil)	80	([4])	82	High	([5]).	

TABLE 5—COOLING MODE TEST CONDITIONS FOR UNITS HAVING A TWO-CAPACITY COMPRESSOR—Continued

Test description	Air entering indoor unit temperature (°F)		Air entering outdoor unit temperature (°F)		Compressor capacity	Cooling air volume rate
	Dry bulb	Wet bulb	Dry bulb	Wet bulb		
C_1 Test—optional (steady, dry-coil)	80	(4)	82	Low	Cooling Minimum.[3]	
D_1 Test—optional (cyclic, dry-coil)	80	(4)	82	Low	(6).	
F_1 Test—required (steady, wet coil)	80	67	67	[1] 53.5	Low	Cooling Minimum.[3]

[1] The specified test condition only applies if the unit rejects condensate to the outdoor coil.
[2] Defined in section 3.1.4.1.
[3] Defined in section 3.1.4.2.
[4] The entering air must have a low enough moisture content so no condensate forms on the indoor coil. DOE recommends using an indoor air wet-bulb temperature of 57 °F or less.
[5] Maintain the airflow nozzle(s) static pressure difference or velocity pressure during the ON period at the same pressure or velocity as measured during the C_2 Test.
[6] Maintain the airflow nozzle(s) static pressure difference or velocity pressure during the ON period at the same pressure or velocity as measured during the C_1 Test.

3.2.4 Tests for a unit having a variable-speed compressor. a. Conduct five steady-state wet coil tests: The A_2, E_V, B_2, B_1, and F_1 Tests. Use the two optional dry-coil tests, the steady-state G_1 Test and the cyclic I_1 Test, to determine the cooling mode cyclic degradation coefficient,C_D^c. If the two optional tests are conducted but yield a tested C_D^c that exceeds the default C_D^c or if the two optional tests are not conducted, assign C_D^c the default value of 0.25.Table 6 specifies test conditions for these seven tests. Determine the intermediate compressor speed cited in Table 6 using:

$$\text{Intermediate speed} = \text{Minimum speed} + \frac{\text{Maximum speed} - \text{Minimum speed}}{3}$$

where a tolerance of plus 5 percent or the next higher inverter frequency step from that calculated is allowed.

b. For units that modulate the indoor fan speed to adjust the sensible to total (S/T) cooling capacity ratio, use Cooling Full-load, Cooling Intermediate, and Cooling Minimum Air Volume Rates that represent a normal residential installation. Additionally, if conducting the optional dry-coil tests, operate the unit in the same S/T capacity control mode as used for the F_1 Test.

c. For multiple-split air conditioners and heat pumps (except where noted), the following procedures supersede the above requirements: For all Table 6 tests specified for a minimum compressor speed, at least one indoor unit must be turned off. The manufacturer shall designate the particular indoor unit(s) that is turned off. The manufacturer must also specify the compressor speed used for the Table 6 E_V Test, a cooling-mode intermediate compressor speed that falls within ¼ and ¾ of the difference between the maximum and minimum cooling-mode speeds. The manufacturer should prescribe an intermediate speed that is expected to yield the highest EER for the given E_V Test conditions and bracketed compressor speed range. The manufacturer can designate that one or more indoor units are turned off for the E_V Test.

TABLE 6—COOLING MODE TEST CONDITION FOR UNITS HAVING A VARIABLE-SPEED COMPRESSOR

Test description	Air entering indoor unit temperature (°F)		Air entering outdoor unit temperature (°F)		Compressor speed	Cooling air volume rate
	Dry bulb	Wet bulb	Dry bulb	Wet bulb		
A_2 Test—required (steady, wet coil)	80	67	95	[1] 75	Maximum	Cooling Full-Load[2]
B_2 Test—required (steady, wet coil)	80	67	82	[1] 65	Maximum	Cooling Full-Load[2]
E_V Test—required (steady, wet coil)	80	67	87	[1] 69	Intermediate	Cooling Intermediate[3]

TABLE 6—COOLING MODE TEST CONDITION FOR UNITS HAVING A VARIABLE-SPEED COMPRESSOR—Continued

Test description	Air entering indoor unit temperature (°F)		Air entering outdoor unit temperature (°F)		Compressor speed	Cooling air volume rate
	Dry bulb	Wet bulb	Dry bulb	Wet bulb		
B_1 Test—required (steady, wet coil)	80	67	82	[1] 65	Minimum	Cooling Minimum [4]
F_1 Test—required (steady, wet coil)	80	67	67	[1] 53.5	Minimum	Cooling Minimum [4]
G_1 Test [5]—optional (steady, dry-coil)	80	([6])	67	Minimum	Cooling Minimum [4].	
I_1 Test [5]—optional (cyclic, dry-coil)	80	([6])	67	Minimum	([6]).	

[1] The specified test condition only applies if the unit rejects condensate to the outdoor coil.
[2] Defined in section 3.1.4.1.
[3] Defined in section 3.1.4.3.
[4] Defined in section 3.1.4.2.
[5] The entering air must have a low enough moisture content so no condensate forms on the indoor coil. DOE recommends using an indoor air wet bulb temperature of 57 °F or less.
[6] Maintain the airflow nozzle(s) static pressure difference or velocity pressure during the ON period at the same pressure difference or velocity pressure as measured during the G_1 Test.

3.3 Test procedures for steady-state wet coil cooling mode tests (the A, A_2, A_1, B, B_2, B_1, E_V, and F_1 Tests). a. For the pretest interval, operate the test room reconditioning apparatus and the unit to be tested until maintaining equilibrium conditions for at least 30 minutes at the specified section 3.2 test conditions. Use the exhaust fan of the airflow measuring apparatus and, if installed, the indoor fan of the test unit to obtain and then maintain the indoor air volume rate and/or external static pressure specified for the particular test. Continuously record (see Definition 1.15):

(1) The dry-bulb temperature of the air entering the indoor coil,

(2) The water vapor content of the air entering the indoor coil,

(3) The dry-bulb temperature of the air entering the outdoor coil, and

(4) For the section 2.2.4 cases where its control is required, the water vapor content of the air entering the outdoor coil.

Refer to section 3.11 for additional requirements that depend on the selected secondary test method.

b. After satisfying the pretest equilibrium requirements, make the measurements specified in Table 3 of ASHRAE Standard 37–2005 (incorporated by reference, see §430.22) for the Indoor Air Enthalpy method and the user-selected secondary method. Except for external static pressure, make the Table 3 measurements at equal intervals that span 10 minutes or less. Measure external static pressure every 5 minutes or less. Continue data sampling until reaching a 30-minute period (*e.g.*, four consecutive 10-minute samples) where the test tolerances specified in Table 7 are satisfied. For those continuously recorded parameters, use the entire data set from the 30-minute interval to evaluate Table 7 compliance. Determine the average electrical power consumption of the air conditioner or heat pump over the same 30-minute interval.

c. Calculate indoor-side total cooling capacity as specified in sections 7.3.3.1 and 7.3.3.3 of ASHRAE Standard 37–2005 (incorporated by reference, see §430.22). Do not adjust the parameters used in calculating capacity for the permitted variations in test conditions. Evaluate air enthalpies based on the measured barometric pressure. Assign the average total space cooling capacity and electrical power consumption over the 30-minute data collection interval to the variables $\dot{Q}_c^k(T)$ and $\dot{E}_c^k(T)$, respectively. For these two variables, replace the ''T'' with the nominal outdoor temperature at which the test was conducted. The superscript k is used only when testing multi-capacity units. Use the superscript k=2 to denote a test with the unit operating at high capacity or maximum speed, k=1 to denote low capacity or minimum speed, and k=v to denote the intermediate speed.

d. For units tested without an indoor fan installed, decrease $\dot{Q}_c^k(T)$ by

$$\frac{1250\ \text{Btu/h}}{1000\ \text{scfm}} \cdot \overline{\dot{V}}_s,$$

and increase $\dot{E}_c^k(T)$ by,

$$\frac{365\ \text{W}}{1000\ \text{scfm}} \cdot \overline{\dot{V}}_s,$$

where $\overline{\dot{V}}_s$ is the average measured indoor air volume rate expressed in units of cubic feet per minute of standard air (scfm).

TABLE 7—TEST OPERATING AND TEST CONDITION TOLERANCES FOR SECTION 3.3 STEADY-STATE WET COIL COOLING MODE TESTS AND SECTION 3.4 DRY COIL COOLING MODE TESTS

	Test operating tolerance [1]	Test condition tolerance [2]
Indoor dry-bulb, °F		
Entering temperature	2.0	0.5
Leaving temperature	2.0	
Indoor wet-bulb, °F		
Entering temperature	1.0	[3] 0.3
Leaving temperature	[3] 1.0	
Outdoor dry-bulb, °F		
Entering temperature	2.0	0.5
Leaving temperature	[4] 2.0	
Outdoor wet-bulb, °F		
Entering temperature	1.0	[5] 0.3
Leaving temperature	[4] 1.0	
External resistance to airflow, inches of water	0.05	[6] 0.02
Electrical voltage, % of rdg.	2.0	1.5
Nozzle pressure drop, % of rdg.	2.0	

[1] See Definition 1.41.
[2] See Definition 1.40.
[3] Only applies during wet coil tests; does not apply during steady-state, dry coil cooling mode tests.
[4] Only applies when using the Outdoor Air Enthalpy Method.
[5] Only applies during wet coil cooling mode tests where the unit rejects condensate to the outdoor coil.
[6] Only applies when testing non-ducted units.

d. For air conditioners and heat pumps having a constant-air-volume-rate indoor fan, the five additional steps listed below are required if the average of the measured external static pressures exceeds the applicable sections 3.1.4 minimum (or target) external static pressure (ΔP_{min}) by 0.03 inches of water or more.

1. Measure the average power consumption of the indoor fan motor ($\dot{E}_{fan,1}$) and record the corresponding external static pressure (ΔP_1) during or immediately following the 30-minute interval used for determining capacity.

2. After completing the 30-minute interval and while maintaining the same test conditions, adjust the exhaust fan of the airflow measuring apparatus until the external static pressure increases to approximately $\Delta P_1 + (\Delta P_1 - \Delta P_{min})$.

3. After re-establishing steady readings of the fan motor power and external static pressure, determine average values for the indoor fan power ($\dot{E}_{fan,2}$) and the external static pressure (ΔP_2) by making measurements over a 5-minute interval.

4. Approximate the average power consumption of the indoor fan motor at ΔP_{min} using linear extrapolation:

$$\dot{E}_{fan,min} = \frac{\dot{E}_{fan,2} - \dot{E}_{fan,1}}{\Delta P_2 - \Delta P_1}\left(\Delta P_{min} - \Delta P_1\right) + \dot{E}_{fan,1}\,.$$

5. Increase the total space cooling capacity, $\dot{Q}_c^k(T)$, by the quantity ($\dot{E}_{fan,1} - \dot{E}_{fan,min}$), when expressed on a Btu/h basis. Decrease the total electrical power, $\dot{E}_c^k(T)$, by the same fan power difference, now expressed in watts.

3.4 Test procedures for the optional steady-state dry-coil cooling-mode tests (the C, C_1, C_2, and G_1 Tests).

a. Except for the modifications noted in this section, conduct the steady-state dry coil cooling mode tests as specified in section 3.3 for wet coil tests. Prior to recording data during the steady-state dry coil test, operate the unit at least one hour after achieving dry coil conditions. Drain the drain pan and plug the drain opening. Thereafter, the drain pan should remain completely dry.

b. Denote the resulting total space cooling capacity and electrical power derived from the test as $\dot{Q}_{ss,dry}$ and $\dot{E}_{ss,dry}$. With regard to a section 3.3 deviation, do not adjust $\dot{Q}_{ss,dry}$ for duct losses (i.e., do not apply section 7.3.3.3 of ASHRAE Standard 37–2005 (incorporated by reference, see §430.22)). In preparing for the section 3.5 cyclic tests, record the average indoor-side air volume rate, $\bar{V}$, specific heat of the air, Cp,a (expressed on dry air basis), specific volume of the air at the nozzles, v'_n, humidity ratio at the nozzles, W_n, and either pressure difference or velocity pressure for the flow nozzles. For units having a variable-speed indoor fan (that provides either a constant or variable air volume rate) that will or may be tested during the cyclic dry coil cooling mode test with the indoor fan turned off (see section 3.5), include the electrical power used by the indoor fan motor among the recorded parameters from the 30-minute test.

3.5 Test procedures for the optional cyclic dry-coil cooling-mode tests (the D, D_1, D_2, and I_1 Tests). a. After completing the steady-state dry-coil test, remove the Outdoor Air Enthalpy method test apparatus, if connected, and begin manual OFF/ON cycling of the unit's compressor. The test set-up should otherwise be identical to the set-up used during the steady-state dry coil test. When testing heat pumps, leave the reversing valve during the compressor OFF cycles in the same position as used for the compressor ON cycles, unless automatically changed by the controls of the unit. For units having a variable-speed indoor fan, the manufacturer has the option of electing at the outset whether to conduct the cyclic test with the indoor fan enabled or disabled. Always revert to testing with the indoor fan disabled if cyclic testing with the fan enabled is unsuccessful.

b. For units having a single-speed or two-capacity compressor, cycle the compressor OFF for 24 minutes and then ON for 6 minutes ($\Delta\tau_{cyc,dry}$ = 0.5 hours). For units having a variable-speed compressor, cycle the compressor OFF for 48 minutes and then ON for 12 minutes ($\Delta\tau_{cyc,dry}$ = 1.0 hours). Repeat the OFF/ON compressor cycling pattern until the test is completed. Allow the controls of the unit to regulate cycling of the outdoor fan.

c. Sections 3.5.1 and 3.5.2 specify airflow requirements through the indoor coil of ducted and non-ducted systems, respectively. In all cases, use the exhaust fan of the airflow measuring apparatus (covered under section 2.6) along with the indoor fan of the unit, if installed and operating, to approximate a step response in the indoor coil airflow. Regulate the exhaust fan to quickly obtain and then maintain the flow nozzle static pressure difference or velocity pressure at the same value as was measured during the steady-state dry coil test. The pressure difference or velocity pressure should be within 2 percent of the value from the steady-state dry coil test within 15 seconds after airflow initiation. For units having a variable-speed indoor fan that ramps when cycling on and/or off, use the exhaust fan of the airflow measuring apparatus to impose a step response that begins at the initiation of ramp up and ends at the termination of ramp down.

d. For units having a variable-speed indoor fan, conduct the cyclic dry coil test using the pull-thru approach described below if any of the following occur when testing with the fan operating:

(1) The test unit automatically cycles off;

(2) Its blower motor reverses; or

(3) The unit operates for more than 30 seconds at an external static pressure that is 0.1 inches of water or more higher than the value measured during the prior steady-state test.

For the pull-thru approach, disable the indoor fan and use the exhaust fan of the airflow measuring apparatus to generate the specified flow nozzles static pressure difference or velocity pressure. If the exhaust fan cannot deliver the required pressure difference because of resistance created by the unpowered blower, temporarily remove the blower.

e. After completing a minimum of two complete compressor OFF/ON cycles, determine the overall cooling delivered and total electrical energy consumption during any subsequent data collection interval where the test tolerances given in Table 8 are satisfied. If available, use electric resistance heaters (see section 2.1) to minimize the variation in the inlet air temperature.

f. With regard to the Table 8 parameters, continuously record the dry-bulb temperature of the air entering the indoor and outdoor coils during periods when air flows through the respective coils. Sample the water vapor content of the indoor coil inlet air at least every 2 minutes during periods when air flows through the coil. Record external static pressure and the air volume rate indicator (either nozzle pressure difference or velocity pressure) at least every minute during the interval that air flows through the indoor coil. (These regular measurements of the airflow rate indicator are in addition to the required measurement at 15 seconds after flow initiation.) Sample the electrical voltage at least every 2 minutes beginning 30 seconds after compressor start-up. Continue until the compressor, the outdoor fan, and the indoor fan (if it is installed and operating) cycle off.

g. For ducted units, continuously record the dry-bulb temperature of the air entering (as noted above) and leaving the indoor coil. Or if using a thermopile, continuously record the difference between these two temperatures during the interval that air flows through the indoor coil. For non-ducted units, make the same dry-bulb temperature measurements beginning when the compressor cycles on and ending when indoor coil airflow ceases.

h. Integrate the electrical power over complete cycles of length $\Delta\tau_{cyc,dry}$. For ducted units tested with an indoor fan installed and operating, integrate electrical power from indoor fan OFF to indoor fan OFF. For all other ducted units and for non-ducted units, integrate electrical power from compressor OFF to compressor OFF. (Some cyclic tests will use the same data collection intervals to determine the electrical energy and the total space cooling. For other units, terminate data collection used to determine the electrical energy before terminating data collection used to determine total space cooling.)

TABLE 8—TEST OPERATING AND TEST CONDITION TOLERANCES FOR CYCLIC DRY COIL COOLING MODE TESTS

	Test Operating Tolerance [1]	Test Condition Tolerance [2]
Indoor entering dry-bulb temperature [3], °F	2.0	0.5
Indoor entering wet-bulb temperature, °F		[4]
Outdoor entering dry-bulb temperature [3], °F	2.0	0.5
External resistance to airflow [3], inches of water	0.05	
Airflow nozzle pressure difference or velocity pressure [3], % of reading	2.0	[5] 2.0
Electrical voltage [6], % of rdg.	2.0	1.5

[1] See Definition 1.41.
[2] See Definition 1.40.
[3] Applies during the interval that air flows through the indoor (outdoor) coil except for the first 30 seconds after flow initiation. For units having a variable-speed indoor fan that ramps, the tolerances listed for the external resistance to airflow apply from 30 seconds after achieving full speed until ramp down begins.
[4] Shall at no time exceed a wet-bulb temperature that results in condensate forming on the indoor coil.
[5] The test condition shall be the average nozzle pressure difference or velocity pressure measured during the steady-state dry coil test.
[6] Applies during the interval when at least one of the following—the compressor, the outdoor fan, or, if applicable, the indoor fan—are operating except for the first 30 seconds after compressor start-up.

i. If the Table 8 tolerances are satisfied over the complete cycle, record the measured electrical energy consumption as $e_{cyc,dry}$ and express it in units of watt-hours. Calculate the total space cooling delivered, $q_{cyc,dry}$, in units of Btu using,

$$q_{cyc,dry} = \frac{60 \cdot \bar{\dot{V}} \cdot C_{p,a} \cdot \Gamma}{\left[v'_n \cdot (1 + W_n)\right]} = \frac{60 \cdot \bar{\dot{V}} \cdot C_{p,a} \cdot \Gamma}{v_n} \qquad (3.5\text{-}1)$$

where $\bar{\dot{V}}$, $C_{p,a}$, v_n' (or v_n), and W_n are the values recorded during the section 3.4 dry coil steady-state test and,

$$\Gamma = \int_{\tau_1}^{\tau_2} \left[T_{al}(\tau) - T_{a2}(\tau)\right] d\tau\,,\ hr \cdot {}^\circ F.$$

$T_{al}(\tau)$ = dry bulb temperature of the air entering the indoor coil at time τ, °F.

$T_{a2}(\tau)$ = dry bulb temperature of the air leaving the indoor coil at time τ, °F.

τ_1 = for ducted units, the elapsed time when airflow is initiated through the indoor coil; for non-ducted units, the elapsed time when the compressor is cycled on, hr.

τ_2 = the elapsed time when indoor coil airflow ceases, hr.

3.5.1 Procedures when testing ducted systems. The automatic controls that are normally installed with the test unit must govern the OFF/ON cycling of the air moving equipment on the indoor side (exhaust fan of the airflow measuring apparatus and, if installed, the indoor fan of the test unit). For example, for ducted units tested without an indoor fan installed but rated based on using a fan time delay relay, control the indoor coil airflow according to the rated ON and/or OFF delays provided by the relay. For ducted units having a variable-speed indoor fan that has been disabled (and possibly removed), start and stop the indoor airflow at the same instances as if the fan were enabled. For all other ducted units tested without an indoor fan installed, cycle the indoor coil airflow in unison with the cycling of the compressor. Close air dampers on the inlet (section 2.5.1) and outlet side (sections 2.5 and 2.5.4) during the OFF period. Airflow through the indoor coil should stop within 3 seconds after the automatic controls of the test unit (act to) de-energize the indoor fan. For ducted units tested without an indoor fan installed (excluding the special case where a variable-speed fan is temporarily removed), increase $e_{cyc,dry}$ by the quantity,

$$\frac{365\ W}{1000\ scfm} \cdot \bar{\dot{V}}_s \cdot [\tau_2 - \tau_1], \qquad (3.5\text{-}2)$$

and decrease $q_{cyc,dry}$ by,

$$\frac{1250\ Btu/h}{1000\ scfm} \cdot \bar{\dot{V}}_s \cdot [\tau_2 - \tau_1], \qquad (3.5\text{-}3)$$

where $\bar{\dot{V}}_s$ is the average indoor air volume rate from the section 3.4 dry coil steady-state test and is expressed in units of cubic feet per minute of standard air (scfm). For units having a variable-speed indoor fan that is disabled during the cyclic test, increase $e_{cyc,dry}$ and decrease $q_{cyc,dry}$ based on:

a. The product of $[\tau_2 - \tau_1]$ and the indoor fan power measured during or following the dry coil steady-state test; or,

b. The following algorithm if the indoor fan ramps its speed when cycling.

1. Measure the electrical power consumed by the variable-speed indoor fan at a minimum of three operating conditions: at the speed/air volume rate/external static pressure that was measured during the steady-state test, at operating conditions associated with the midpoint of the ramp-up interval,

and at conditions associated with the mid-point of the ramp-down interval. For these measurements, the tolerances on the airflow volume or the external static pressure are the same as required for the section 3.4 steady-state test.

2. For each case, determine the fan power from measurements made over a minimum of 5 minutes.

3. Approximate the electrical energy consumption of the indoor fan if it had operated during the cyclic test using all three power measurements. Assume a linear profile during the ramp intervals. The manufacturer must provide the durations of the ramp-up and ramp-down intervals. If a manufacturer-supplied ramp interval exceeds 45 seconds, use a 45-second ramp interval nonetheless when estimating the fan energy.

The manufacturer is allowed to choose option a, and forego the extra testing burden of option b, even if the unit ramps indoor fan speed when cycling.

3.5.2 Procedures when testing non-ducted systems. Do not use air dampers when conducting cyclic tests on non-ducted units. Until the last OFF/ON compressor cycle, airflow through the indoor coil must cycle off and on in unison with the compressor. For the last OFF/ON compressor cycle—the one used to determine $e_{cyc,dry}$ and $q_{cyc,dry}$—use the exhaust fan of the airflow measuring apparatus and the indoor fan of the test unit to have indoor airflow start 3 minutes prior to compressor cut-on and end three minutes after compressor cutoff. Subtract the electrical energy used by the indoor fan during the 3 minutes prior to compressor cut-on from the integrated electrical energy, $e_{cyc,dry}$. Add the electrical energy used by the indoor fan during the 3 minutes after compressor cutoff to the integrated cooling capacity, $q_{cyc,dry}$. For the case where the non-ducted unit uses a variable-speed indoor fan which is disabled during the cyclic test, correct $e_{cyc,dry}$ and $q_{cyc,dry}$ using the same approach as prescribed in section 3.5.1 for ducted units having a disabled variable-speed indoor fan.

3.5.3 Cooling-mode cyclic-degradation coefficient calculation. Use the two optional dry-coil tests to determine the cooling-mode cyclic-degradation coefficient, C_D^c. Append ''(k=2)'' to the coefficient if it corresponds to a two-capacity unit cycling at high capacity. If the two optional tests are conducted but yield a tested C_D^c that exceeds the default C_D^c or if the two optional tests are not conducted, assign C_D^c the default value of 0.25. The default value for two-capacity units cycling at high capacity, however, is the low-capacity coefficient, i.e., $C_D^c(k=2)=C_D^c$. Evaluate C_D^c using the above results and those from the section 3.4 dry-coil steady-state test.

$$C_D^c = \frac{1 - \dfrac{EER_{cyc,dry}}{EER_{ss,dry}}}{1 - CLF}$$

where,

$$EER_{cyc,dry} = \frac{q_{cyc,dry}}{e_{cyc,dry}},$$

the average energy efficiency ratio during the cyclic dry coil cooling mode test, Btu/W·h

$$EER_{ss,dry} = \frac{\dot{Q}_{ss,dry}}{\dot{E}_{ss,dry}},$$

the average energy efficiency ratio during the steady-state dry coil cooling mode test, Btu/W·h

$$CLF = \frac{q_{cyc,dry}}{Q_{ss,dry} \cdot \Delta\tau_{cyc,dry}},$$

the cooling load factor dimensionless.

Round the calculated value for C_D^c to the nearest 0.01. If C_D^c is negative, then set it equal to zero.

3.6 Heating mode tests for different types of heat pumps, including heating-only heat pumps.

3.6.1 Tests for a heat pump having a single-speed compressor that is tested with a fixed speed indoor fan installed, with a constant-air-volume-rate indoor fan installed, or with no indoor fan installed. Conduct the optional High Temperature Cyclic (H1C) Test to determine the heating mode cyclic-degradation coefficient, C_D^h. If this optional test is conducted but yields a tested C_D^h that exceeds the default C_D^h or if the optional test is not conducted, assign C_D^h the default value of 0.25. Test conditions for the four tests are specified in Table 9.

TABLE 9—HEATING MODE TEST CONDITIONS FOR UNITS HAVING A SINGLE-SPEED COMPRESSOR AND A FIXED-SPEED INDOOR FAN, A CONSTANT AIR VOLUME RATE INDOOR FAN, OR NO INDOOR FAN

Test description	Air entering indoor unit Temperature (°F)		Air entering outdoor unit Temperature (°F)		Heating air volume rate
	Dry bulb	Wet bulb	Dry bulb	Wet bulb	
H1 Test (required, steady)	70	60(max)	47	43	Heating Full-load [1]
H1C Test (optional, cyclic)	70	60(max)	47	43	([2])
H2 Test (required)	70	60(max)	35	33	Heating Full-load [1]

TABLE 9—HEATING MODE TEST CONDITIONS FOR UNITS HAVING A SINGLE-SPEED COMPRESSOR AND A FIXED-SPEED INDOOR FAN, A CONSTANT AIR VOLUME RATE INDOOR FAN, OR NO INDOOR FAN—Continued

Test description	Air entering indoor unit Temperature (°F)		Air entering outdoor unit Temperature (°F)		Heating air volume rate
	Dry bulb	Wet bulb	Dry bulb	Wet bulb	
H3 Test (required, steady)	70	60(max)	17	15	Heating Full-load [1]

[1] Defined in section 3.1.4.4.
[2] Maintain the airflow nozzles static pressure difference or velocity pressure during the ON period at the same pressure difference or velocity pressure as measured during the H1 Test.

3.6.2 Tests for a heat pump having a single-speed compressor and a variable-speed, variable-air-volume-rate indoor fan: capacity modulation correlates with outdoor dry bulb temperature. Conduct five tests: two High Temperature Tests ($H1_2$ and $H1_1$), one Frost Accumulation Test ($H2_2$), and two Low Temperature Tests ($H3_2$ and $H3_1$). Conducting an additional Frost Accumulation Test ($H2_1$) is optional. Conduct the optional High Temperature Cyclic ($H1C_1$) Test to determine the heating mode cyclic-degradation coefficient, C_D^h. If this optional test is conducted but yields a tested C_D^h that exceeds the default C_D^h or if the optional test is not conducted, assign C_D^h the default value of 0.25. Test conditions for the seven tests are specified in Table 10. If the optional $H2_1$ Test is not performed, use the following equations to approximate the capacity and electrical power of the heat pump at the $H2_1$ test conditions:

$$\dot{Q}_h^{k=1}(35) = QR_h^{k=2}(35) \cdot \left\{\dot{Q}_h^{k=1}(17) + 0.6 \cdot \left[\dot{Q}_h^{k=1}(47) - \dot{Q}_h^{k=1}(17)\right]\right\}$$

$$\dot{E}_h^{k=1}(35) = PR_h^{k=2}(35) \cdot \left\{\dot{E}_h^{k=1}(17) + 0.6 \cdot \left[\dot{E}_h^{k=1}(47) - \dot{E}_h^{k=1}(17)\right]\right\}$$

where,

$$\dot{Q}R_h^{k=2}(35) = \frac{\dot{Q}_h^{k=2}(35)}{\dot{Q}^{k=2}(17) + 0.6 \cdot \left[\dot{Q}_h^{k=2}(47) - \dot{Q}_h^{k=2}(17)\right]}$$

$$PR_h^{k=2}(35) = \frac{\dot{E}_h^{k=2}(35)}{\dot{E}_h^{k=2}(17) + 0.6 \cdot \left[\dot{E}_h^{k=2}(47) - \dot{E}_h^{k=2}(17)\right]}.$$

The quantities $\dot{Q}_h^{k=2}(47)$, $\dot{E}_h^{k=2}(47)$, $\dot{Q}_h^{k=1}(47)$, and $\dot{E}_h^{k=1}(47)$ are determined from the $H1_2$ and $H1_1$ Tests and evaluated as specified in section 3.7; the quantities $\dot{Q}_h^{k=2}(35)$ and $\dot{E}_h^{k=2}(35)$ are determined from the $H2_2$ Test and evaluated as specified in section 3.9; and the quantities $\dot{Q}_h^{k=2}(17)$, $\dot{E}_h^{k=2}(17)$, $\dot{Q}_h^{k=1}(17)$, and $\dot{E}_h^{k=1}(17)$, are determined from the $H3_2$ and $H3_1$ Tests and evaluated as specified in section 3.10.

TABLE 10—HEATING MODE TEST CONDITIONS FOR UNITS HAVING A SINGLE-SPEED COMPRESSOR AND A VARIABLE AIR VOLUME RATE INDOOR FAN

Test description	Air entering indoor unit temperature (°F)		Air entering outdoor unit temperature (°F)		Heating air volume rate
	Dry bulb	Wet bulb	Dry bulb	Wet bulb	
$H1_2$ Test (required, steady) ..	70	60(max)	47	43	Heating Full-load. [1]
$H1_1$ Test (required, steady) ..	70	60(max)	47	43	Heating Minimum. [2]
$H1C_1$ Test (optional, cyclic)	70	60(max)	47	43	([3])
$H2_2$ Test (required)	70	60(max)	35	33	Heating Full-load. [1]

TABLE 10—HEATING MODE TEST CONDITIONS FOR UNITS HAVING A SINGLE-SPEED COMPRESSOR AND A VARIABLE AIR VOLUME RATE INDOOR FAN—Continued

Test description	Air entering indoor unit temperature (°F)		Air entering outdoor unit temperature (°F)		Heating air volume rate
	Dry bulb	Wet bulb	Dry bulb	Wet bulb	
$H2_1$ Test (optional)	70	60(max)	35	33	Heating Minimum.[2]
$H3_2$ Test (required, steady)	70	60(max)	17	15	Heating Full-load.[1]
$H3_1$ Test (required, steady)	70	60(max)	17	15	Heating Minimum.[2]

[1] Defined in section 3.1.4.4.
[2] Defined in section 3.1.4.5.
[3] Maintain the airflow nozzles static pressure difference or velocity pressure during the ON period at the same pressure difference or velocity pressure as measured during the $H1_1$ Test.

3.6.3 Tests for a heat pump having a two-capacity compressor (see Definition 1.45), including two-capacity, northern heat pumps (see Definition 1.46). a. Conduct one Maximum Temperature Test ($H0_1$), two High Temperature Tests ($H1_2$ and $H1_1$), one Frost Accumulation Test ($H2_2$), and one Low Temperature Test ($H3_2$). Conduct an additional Frost Accumulation Test ($H2_1$) and Low Temperature Test ($H3_1$) if both of the following conditions exist:

1. Knowledge of the heat pump's capacity and electrical power at low compressor capacity for outdoor temperatures of 37 °F and less is needed to complete the section 4.2.3 seasonal performance calculations; and

2.The heat pump's controls allow low-capacity operation at outdoor temperatures of 37 °F and less.

If the above two conditions are met, an alternative to conducting the $H2_1$ Frost Accumulation is to use the following equations to approximate the capacity and electrical power:

$$\dot{Q}_h^{k=1}(35) = 0.90 \cdot \left\{\dot{Q}_h^{k=1}(17) + 0.6 \cdot \left[\dot{Q}_h^{k=1}(47) - \dot{Q}_h^{k=1}(17)\right]\right\}$$
$$\dot{E}_h^{k=1}(35) = 0.985 \cdot \left\{\dot{E}_h^{k=1}(17) + 0.6 \cdot \left[\dot{E}_h^{k=1}(47) - \dot{E}_h^{k=1}(17)\right]\right\}$$

Determine the quantities $\dot{Q}_h^{k=1}$ (47) and $\dot{E}_h^{k=1}$ (47) from the $H1_1$ Test and evaluate them according to Section 3.7. Determine the quantities $\dot{Q}_h^{k=1}$ (17) and $\dot{E}_h^{k=1}$ (17) from the $H3_1$ Test and evaluate them according to Section 3.10.

b. Conduct the optional High Temperature Cyclic Test ($H1C_1$) to determine the heating-mode cyclic-degradation coefficient, C_D^h. If this optional test is conducted but yields a tested C_D^h that exceeds the default C_D^h or if the optional test is not conducted, assign C_D^h the default value of 0.25. If a two-capacity heat pump locks out low capacity operation at lower outdoor temperatures, conduct the optional High Temperature Cyclic Test ($H1C_2$) to determine the high-capacity heating-mode cyclic-degradation coefficient, C_D^h (k=2). If this optional test at high capacity is conducted but yields a tested C_D^h (k=2) that exceeds the default C_D^h (k=2) or if the optional test is not conducted, assign C_D^h the default value. The default C_D^h (k=2) is the same value as determined or assigned for the low-capacity cyclic-degradation coefficient, C_D^h [or equivalently, C_D^h (k=1)]. Table 11 specifies test conditions for these nine tests.

TABLE 11—HEATING MODE TEST CONDITIONS FOR UNITS HAVING A TWO-CAPACITY COMPRESSOR

Test description	Air entering indoor unit temperature (°F)		Air entering outdoor unit temperature (°F)		Compressor capacity	Heating air volume rate
	Dry bulb	Wet bulb	Dry bulb	Wet bulb		
$H0_1$ Test (required, steady)	70	60(max)	62	56.5	Low	Heating Minimum.[1]
$H1_2$ Test (required, steady)	70	60(max)	47	43	High	Heating Full-Load.[2]
$H1C_2$ Test (optional, cyclic)	70	60(max)	47	43	High	([3])
$H1_1$ Test (required)	70	60(max)	47	43	Low	Heating Minimum.[1]

TABLE 11—HEATING MODE TEST CONDITIONS FOR UNITS HAVING A TWO-CAPACITY COMPRESSOR—Continued

Test description	Air entering indoor unit temperature (°F)		Air entering outdoor unit temperature (°F)		Compressor capacity	Heating air volume rate
	Dry bulb	Wet bulb	Dry bulb	Wet bulb		
H1C₁ Test (optional, cyclic)	70	60(max)	47	43	Low	(4)
H2₂ Test (required)	70	60(max)	35	33	High	Heating Full-Load. [2]
H2₁ Test[5,6] (required)	70	60(max)	35	33	Low	Heating Minimum. [1]
H3₂ Test (required, steady)	70	60(max)	17	15	High	Heating Full-Load. [2]
H3₁ Test[5] (required, steady)	70	60(max)	17	15	Low	Heating Minimum. [1]

[1] Defined in section 3.1.4.5.
[2] Defined in section 3.1.4.4.
[3] Maintain the airflow nozzle(s) static pressure difference or velocity pressure during the ON period at the same pressure or velocity as measured during the $H1_2$ Test.
[4] Maintain the airflow nozzle(s) static pressure difference or velocity pressure during the ON period at the same pressure or velocity as measured during the $H1_1$ Test.
[5] Required only if the heat pump's performance when operating at low compressor capacity and outdoor temperatures less than 37 °F is needed to complete the section 4.2.3 *HSPF* calculations.
[6] If table note #5 applies, the section 3.6.3 equations for $\dot{Q}_h^{k=1}(35)$ and $\dot{E}_h^{k=1}(17)$ may be used in lieu of conducting the $H2_1$ Test.

3.6.4 Tests for a heat pump having a variable-speed compressor. a. Conduct one Maximum Temperature Test ($H0_1$), two High Temperature Tests ($H1_2$ and $H1_1$), one Frost Accumulation Test ($H2_V$), and one Low Temperature Test ($H3_2$). Conducting one or both of the following tests is optional: An additional High Temperature Test ($H1_N$) and an additional Frost Accumulation Test ($H2_2$). Conduct the optional Maximum Temperature Cyclic ($H0C_1$) Test to determine the heating mode cyclic-degradation coefficient, C_D^h. If this optional test is conducted but yields a tested C_D^h that exceeds the default C_D^h or if the optional test is not conducted, assign C_D^h the default value of 0.25. Test conditions for the eight tests are specified in Table 12. Determine the intermediate compressor speed cited in Table 12 using the heating mode maximum and minimum compressors speeds and:

$$\text{Intermediate speed} = \text{Minimum speed} + \frac{\text{Maximum speed} - \text{Minimum speed}}{3}$$

where a tolerance of plus 5 percent or the next higher inverter frequency step from that calculated is allowed. If the $H2_2$ Test is not done, use the following equations to approximate the capacity and electrical power at the $H2_2$ test conditions:

$$\dot{Q}_h^{k=2}(35) = 0.90 \cdot \left\{\dot{Q}_h^{k=2}(17) + 0.6 \cdot \left[\dot{Q}_h^{k=2}(47) - \dot{Q}_h^{k=2}(17)\right]\right\}$$

$$\dot{E}_h^{k=2}(35) = 0.985 \cdot \left\{\dot{E}_h^{k=2}(17) + 0.6 \cdot \left[\dot{E}_h^{k=2}(47) - \dot{E}_h^{k=2}(17)\right]\right\}.$$

b. Determine the quantities $\dot{Q}_h^{k=2}(47)$ and from $\dot{E}_h^{k=2}(47)$ from the $H1_2$ Test and evaluate them according to section 3.7. Determine the quantities $\dot{Q}_h^{k=2}(17)$ and $\dot{E}_h^{k=2}(17)$ from the $H3_2$ Test and evaluate them according to section 3.10. For heat pumps where the heating mode maximum compressor speed exceeds its cooling mode maximum compressor speed, conduct the $H1_N$ Test if the manufacturer requests it. If the $H1_N$ Test is done, operate the heat pump's compressor at the same speed as the speed used for the cooling mode A_2 Test. Refer to the last sentence of section 4.2 to see how the results of the $H1_N$ Test may be used in calculating the heating seasonal performance factor.

TABLE 12—HEATING MODE TEST CONDITIONS FOR UNITS HAVING A VARIABLE-SPEED COMPRESSOR

Test description	Air entering indoor unit temperature (°F)		Air entering outdoor unit temperature (°F)		Compressor speed	Heating air volume rate
	Dry bulb	Wet bulb	Dry bulb	Wet bulb		
$H0_1$ Test (required, steady)	70	$60^{(max)}$	62	56.5	Minimum	Heating Minimum.[1]
$H0C_1$ Test (optional, steady)	70	$60^{(max)}$	62	56.5	Minimum	([2])
$H1_2$ Test (required, steady)	70	$60^{(max)}$	47	43	Maximum	Heating Full-Load.[3]
$H1_1$ Test (required, steady)	70	$60^{(max)}$	47	43	Minimum	Heating Minimum.[1]
$H1_N$ Test (optional, steady)	70	$60^{(max)}$	47	43	Cooling Mode Maximum.	Heating Nominal.[4]
$H2_2$ Test (optional)	70	$60^{(max)}$	35	33	Maximum	Heating Full-Load.[3]
$H2_V$ Test (required)	70	$60^{(max)}$	35	33	Intermediate	Heating Intermediate.[5]
$H3_2$ Test (required, steady)	70	$60^{(max)}$	17	15	Maximum	Heating Full-Load.[3]

[1] Defined in section 3.1.4.5.
[2] Maintain the airflow nozzle(s) static pressure difference or velocity pressure during an ON period at the same pressure or velocity as measured during the $H0_1$ Test.
[3] Defined in section 3.1.4.4.
[4] Defined in section 3.1.4.7.
[5] Defined in section 3.1.4.6.

c. For multiple-split heat pumps (only), the following procedures supersede the above requirements. For all Table 12 tests specified for a minimum compressor speed, at least one indoor unit must be turned off. The manufacturer shall designate the particular indoor unit(s) that is turned off. The manufacturer must also specify the compressor speed used for the Table 12 $H2_V$ Test, a heating-mode intermediate compressor speed that falls within ¼ and ¾ of the difference between the maximum and minimum heating-mode speeds. The manufacturer should prescribe an intermediate speed that is expected to yield the highest COP for the given $H2_V$ Test conditions and bracketed compressor speed range. The manufacturer can designate that one or more specific indoor units are turned off for the $H2_V$ Test.

3.6.5 Additional test for a heat pump having a heat comfort controller. Test any heat pump that has a heat comfort controller (see Definition 1.28) according to section 3.6.1, 3.6.2, or 3.6.3, whichever applies, with the heat comfort controller disabled. Additionally, conduct the abbreviated test described in section 3.1.9 with the heat comfort controller active to determine the system's maximum supply air temperature. (Note: heat pumps having a variable speed compressor and a heat comfort controller are not covered in the test procedure at this time.)

3.7 Test procedures for steady-state Maximum Temperature and High Temperature heating mode tests (the $H0_1$, H1, $H1_2$, $H1_1$, and $H1_N$ Tests). a. For the pretest interval, operate the test room reconditioning apparatus and the heat pump until equilibrium conditions are maintained for at least 30 minutes at the specified section 3.6 test conditions. Use the exhaust fan of the airflow measuring apparatus and, if installed, the indoor fan of the heat pump to obtain and then maintain the indoor air volume rate and/or the external static pressure specified for the particular test. Continuously record the dry-bulb temperature of the air entering the indoor coil, and the dry-bulb temperature and water vapor content of the air entering the outdoor coil. Refer to section 3.11 for additional requirements that depend on the selected secondary test method. After satisfying the pretest equilibrium requirements, make the measurements specified in Table 3 of ASHRAE Standard 37–2005 (incorporated by reference, see §430.22) for the Indoor Air Enthalpy method and the user-selected secondary method. Except for external static pressure, make the Table 3 measurements at equal intervals that span 10 minutes or less. Measure external static pressure every 5 minutes or less. Continue data sampling until a 30-minute period (*e.g.*, four consecutive 10-minute samples) is reached where the test tolerances specified in Table 13 are satisfied. For those continuously recorded parameters, use the entire data set for the 30-minute interval when evaluating Table 13 compliance. Determine the average electrical power consumption of the heat pump over the same 30-minute interval.

TABLE 13—TEST OPERATING AND TEST CONDITION TOLERANCES FOR SECTION 3.7 AND SECTION 3.10 STEADY-STATE HEATING MODE TESTS

	Test operating tolerance [1]	Test condition tolerance [2]
Indoor dry-bulb, °F:		
Entering temperature	2.0	0.5
Leaving temperature	2.0	
Indoor wet-bulb, °F:		
Entering temperature	1.0	
Leaving temperature	1.0	
Outdoor dry-bulb, °F:		
Entering temperature	2.0	0.5
Leaving temperature	[2] 2.0	
Outdoor wet-bulb, °F:		
Entering temperature	1.0	0.3
Leaving temperature	[3] 1.0	
External resistance to airflow, inches of water	0.05	[4] 0.02
Electrical voltage, % of rdg	2.0	1.5
Nozzle pressure drop, % of rdg	2.0	

[1] See Definition 1.41.
[2] See Definition 1.40.
[3] Only applies when the Outdoor Air Enthalpy Method is used.
[4] Only applies when testing non-ducted units.

b. Calculate indoor-side total heating capacity as specified in sections 7.3.4.1 and 7.3.4.3 of ASHRAE Standard 37–2005 (incorporated by reference, see §430.22). Do not adjust the parameters used in calculating capacity for the permitted variations in test conditions. Assign the average space heating capacity and electrical power over the 30-minute data collection interval to the variables $\dot{Q}_h^k$ and $\dot{E}_h^k(T)$ respectively. The ‘‘T’’ and superscripted ‘‘k’’ are the same as described in section 3.3. Additionally, for the heating mode, use the superscript to denote results from the optional $H1_N$ Test, if conducted.

c. For heat pumps tested without an indoor fan installed, increase $\dot{Q}_h^k(T)$ by

$$\frac{1250\ \text{Btu/h}}{1000\ \text{scfm}} \cdot \overline{\dot{V}}_s,$$

and increase $\dot{E}_h^k(T)$ by,

$$\frac{365\ \text{W}}{1000\ \text{scfm}} \cdot \overline{\dot{V}}_s,$$

where $\overline{\dot{V}}_s$ is the average measured indoor air volume rate expressed in units of cubic feet per minute of standard air (scfm). During the 30-minute data collection interval of a High Temperature Test, pay attention to preventing a defrost cycle. Prior to this time, allow the heat pump to perform a defrost cycle if automatically initiated by its own controls. As in all cases, wait for the heat pump's defrost controls to automatically terminate the defrost cycle. Heat pumps that undergo a defrost should operate in the heating mode for at least 10 minutes after defrost termination prior to beginning the 30-minute data collection interval. For some heat pumps, frost may accumulate on the outdoor coil during a High Temperature test. If the indoor coil leaving air temperature or the difference between the leaving and entering air temperatures decreases by more than 1.5 °F over the 30-minute data collection interval, then do not use the collected data to determine capacity. Instead, initiate a defrost cycle. Begin collecting data no sooner than 10 minutes after defrost termination. Collect 30 minutes of new data during which the Table 13 test tolerances are satisfied. In this case, use only the results from the second 30-minute data collection interval to evaluate $\dot{Q}_h^k(47)$ and $\dot{E}_h^k(47)$.

d. If conducting the optional cyclic heating mode test, which is described in section 3.8, record the average indoor-side air volume rate, $\overline{\dot{V}}$, specific heat of the air, $C_{p,a}$ (expressed on dry air basis), specific volume of the air at the nozzles, v_n' (or v_n), humidity ratio at the nozzles, W_n, and either pressure difference or velocity pressure for the flow nozzles. If either or both of the below criteria apply, determine the average, steady-state, electrical power consumption of the indoor fan motor ($\dot{E}_{fan,1}$):

1. The section 3.8 cyclic test will be conducted and the heat pump has a variable-speed indoor fan that is expected to be disabled during the cyclic test; or

2. The heat pump has a (variable-speed) constant-air volume-rate indoor fan and during the steady-state test the average external static pressure (ΔP_1) exceeds the applicable section 3.1.4.4 minimum (or targeted) external static pressure (ΔP_{min}) by 0.03 inches of water or more.

Determine $\dot{E}_{fan,1}$ by making measurements during the 30-minute data collection interval, or immediately following the test and prior to changing the test conditions. When the above ‘‘2’’ criteria applies, conduct the following four steps after determining $\dot{E}_{fan,1}$ (which corresponds to ΔP_1):

i. While maintaining the same test conditions, adjust the exhaust fan of the airflow measuring apparatus until the external static pressure increases to approximately $\Delta P_1 + (\Delta P_1 - \Delta P_{min})$.

ii. After re-establishing steady readings for fan motor power and external static pressure, determine average values for the indoor fan power ($\dot{E}_{fan,2}$) and the external static pressure (ΔP_2) by making measurements over a 5-minute interval.

iii. Approximate the average power consumption of the indoor fan motor if the 30-minute test had been conducted at ΔP_{min} using linear extrapolation:

$$\dot{E}_{fan,min} = \frac{\dot{E}_{fan,2} - \dot{E}_{fan,1}}{\Delta P_2 - \Delta P_1}\left(\Delta P_{min} - \Delta P_1\right) + \dot{E}_{fan,1}.$$

iv. Decrease the total space heating capacity, $\dot{Q}_h^k(T)$, by the quantity $(\dot{E}_{fan,1} - \dot{E}_{fan,min})$, when expressed on a Btu/h basis. Decrease the total electrical power, $\dot{E}_h^k(T)$ by the same fan power difference, now expressed in watts.

3.8 Test procedures for the optional cyclic heating mode tests (the $H0C_1$, H1C, $H1C_1$ and $H1C_2$ Tests). a. Except as noted below, conduct the cyclic heating mode test as specified in section 3.5. As adapted to the heating mode, replace section 3.5 references to "the steady-state dry coil test" with "the heating mode steady-state test conducted at the same test conditions as the cyclic heating mode test." Use the test tolerances in Table 14 rather than Table 8. Record the outdoor coil entering wet-bulb temperature according to the requirements given in section 3.5 for the outdoor coil entering dry-bulb temperature. Drop the subscript "dry" used in variables cited in section 3.5 when referring to quantities from the cyclic heating mode test. Determine the total space heating delivered during the cyclic heating test, q_{cyc}, as specified in section 3.5 except for making the following changes:

(1) When evaluating Equation 3.5–1, use the values of $\bar{\dot{V}}$, $C_{p,a}$,v_n', (or v_n), and W_n that were recorded during the section 3.7 steady-state test conducted at the same test conditions.

(2) Calculate Γ using,

$$\Gamma = \int_{\tau_1}^{\tau_2} \left[T_{a2}(\tau) - T_{a1}(\tau)\right] \delta\tau, \ \text{hr} \cdot {}^\circ\text{F}.$$

b. For ducted heat pumps tested without an indoor fan installed (excluding the special case where a variable-speed fan is temporarily removed), increase q_{cyc} by the amount calculated using Equation 3.5–3. Additionally, increase e_{cyc} by the amount calculated using Equation 3.5–2. In making these calculations, use the average indoor air volume rate ($\bar{\dot{V}}_s$) determined from the section 3.7 steady-state heating mode test conducted at the same test conditions.

c. For non-ducted heat pumps, subtract the electrical energy used by the indoor fan during the 3 minutes after compressor cutoff from the non-ducted heat pump's integrated heating capacity, q_{cyc}.

d. If a heat pump defrost cycle is manually or automatically initiated immediately prior to or during the OFF/ON cycling, operate the heat pump continuously until 10 minutes after defrost termination. After that, begin cycling the heat pump immediately or delay until the specified test conditions have been re-established. Pay attention to preventing defrosts after beginning the cycling process. For heat pumps that cycle off the indoor fan during a defrost cycle, make no effort here to restrict the air movement through the indoor coil while the fan is off. Resume the OFF/ON cycling while conducting a minimum of two complete compressor OFF/ON cycles before determining q_{cyc} and e_{cyc}.

3.8.1 Heating mode cyclic-degradation coefficient calculation. Use the results from the optional cyclic test and the required steady-state test that were conducted at the same test conditions to determine the heating-mode cyclic-degradation coefficient C_D^h. Add "(k=2)" to the coefficient if it corresponds to a two-capacity unit cycling at high capacity. For the below calculation of the heating mode cyclic degradation coefficient, do not include the duct loss correction from section 7.3.3.3 of ASHRAE Standard 37–2005 (incorporated by reference, see § 430.22) in determining $\dot{Q}_h^k(T_{cyc})$ (or q_{cyc}). If the optional cyclic test is conducted but yields a tested C_D^h that exceeds the default C_D^h or if the optional test is not conducted, assign C_D^h the default value of 0.25. The default value for two-capacity units cycling at high capacity, however, is the low-capacity coefficient, i.e., C_D^h (k=2) = C_D^h. The tested C_D^h is calculated as follows:

$$C_D^h = \frac{1 - \dfrac{COP_{cyc}}{COP_{ss}\left(T_{cyc}\right)}}{1 - HLF}$$

where,

$$COP_{cyc} = \frac{q_{cyc}}{3.413 \dfrac{\text{Btu/h}}{\text{W}} \cdot e_{cyc}},$$

the average coefficient of performance during the cyclic heating mode test, dimensionless.

$$COP_{ss}\left(T_{cyc}\right) = \frac{\dot{Q}_h^k\left(T_{cyc}\right)}{3.413 \dfrac{\text{Btu/h}}{\text{W}} \cdot \dot{E}_h^k\left(T_{cyc}\right)},$$

the average coefficient of performance during the steady-state heating mode test conducted at the same test conditions—*i.e.*, same outdoor dry bulb temperature, T_{cyc}, and speed/capacity, k, if applicable—as specified for the cyclic heating mode test, dimensionless.

$$HLF = \frac{q_{cyc}}{\dot{Q}_h^k\left(T_{cyc}\right) \cdot \Delta\tau_{cyc}},$$

the heating load factor, dimensionless.

T_{cyc} = the nominal outdoor temperature at which the cyclic heating mode test is conducted, 62 or 47 °F.

$\Delta\tau_{cyc}$ = the duration of the OFF/ON intervals; 0.5 hours when testing a heat pump having a single-speed or two-capacity compressor and 1.0 hour when testing a heat pump having a variable-speed compressor.

Round the calculated value for C_D^h to the nearest 0.01. If C_D^h is negative, then set it equal to zero.

TABLE 14—TEST OPERATING AND TEST CONDITION TOLERANCES FOR CYCLIC HEATING MODE TESTS.

	Test operating tolerance [1]	Test condition tolerance [2]
Indoor entering dry-bulb temperature, [3] °F	2.0	0.5
Indoor entering wet-bulb temperature, [3] °F	1.0	
Outdoor entering dry-bulb temperature, [3] °F	2.0	0.5
Outdoor entering wet-bulb temperature, [3] °F	2.0	1.0
External resistance to air-flow, [3] inches of water	0.05	
Airflow nozzle pressure difference or velocity pressure,[3] % of reading	2.0	[4] 2.0
Electrical voltage, [5] % of rdg	2.0	1.5

[1] See Definition 1.41.
[2] See Definition 1.40.
[3] Applies during the interval that air flows through the indoor (outdoor) coil except for the first 30 seconds after flow initiation. For units having a variable-speed indoor fan that ramps, the tolerances listed for the external resistance to airflow shall apply from 30 seconds after achieving full speed until ramp down begins.
[4] The test condition shall be the average nozzle pressure difference or velocity pressure measured during the steady-state test conducted at the same test conditions.
[5] Applies during the interval that at least one of the following—the compressor, the outdoor fan, or, if applicable, the indoor fan—are operating, except for the first 30 seconds after compressor start-up.

3.9 Test procedures for Frost Accumulation heating mode tests (the H2, $H2_2$, $H2_V$, and $H2_1$ Tests). a. Confirm that the defrost controls of the heat pump are set as specified in section 2.2.1. Operate the test room reconditioning apparatus and the heat pump for at least 30 minutes at the specified section 3.6 test conditions before starting the ''preliminary'' test period. The preliminary test period must immediately precede the ''official'' test period, which is the heating and defrost interval over which data are collected for evaluating average space heating capacity and average electrical power consumption.

b. For heat pumps containing defrost controls which are likely to cause defrosts at intervals less than one hour, the preliminary test period starts at the termination of an automatic defrost cycle and ends at the termination of the next occurring automatic defrost cycle. For heat pumps containing defrost controls which are likely to cause defrosts at intervals exceeding one hour, the preliminary test period must consist of a heating interval lasting at least one hour followed by a defrost cycle that is either manually or automatically initiated. In all cases, the heat pump's own controls must govern when a defrost cycle terminates.

c. The official test period begins when the preliminary test period ends, at defrost termination. The official test period ends at the termination of the next occurring automatic defrost cycle. When testing a heat pump that uses a time-adaptive defrost control system (see Definition 1.42), however, manually initiate the defrost cycle that ends the official test period at the instant indicated by instructions provided by the manufacturer. If the heat pump has not undergone a defrost after 6 hours, immediately conclude the test and use the results from the full 6-hour period to calculate the average space heating capacity and average electrical power consumption.

For heat pumps that turn the indoor fan off during the defrost cycle, take steps to cease forced airflow through the indoor coil and block the outlet duct whenever the heat pump's controls cycle off the indoor fan. If it is installed, use the outlet damper box described in section 2.5.4.1 to affect the blocked outlet duct.

d. Defrost termination occurs when the controls of the heat pump actuate the first change in converting from defrost operation to normal heating operation. Defrost initiation occurs when the controls of the heat pump first alter its normal heating operation in order to eliminate possible accumulations of frost on the outdoor coil.

e. To constitute a valid Frost Accumulation test, satisfy the test tolerances specified in Table 15 during both the preliminary and official test periods. As noted in Table 15, test operating tolerances are specified for two sub-intervals: (1) When heating, except for the first 10 minutes after the termination of a defrost cycle (Sub-interval H, as described in Table 15) and (2) when defrosting, plus these same first 10 minutes after defrost termination (Sub-interval D, as described in Table 15). Evaluate compliance with Table 15 test condition tolerances and the majority of the test operating tolerances using the averages from measurements recorded only during Sub-interval H. Continuously record the dry bulb temperature of the air entering the indoor coil, and the dry bulb temperature and water vapor content of the air entering the outdoor coil. Sample the remaining parameters listed in Table 15 at equal intervals that span 10 minutes or less.

f. For the official test period, collect and use the following data to calculate average space heating capacity and electrical power. During heating and defrosting intervals when the controls of the heat pump have the indoor fan on, continuously record the dry-bulb temperature of the air entering (as noted above) and leaving the indoor coil. If using a thermopile, continuously record the difference between the leaving and entering dry-bulb temperatures during the interval(s) that air flows through the indoor coil. For heat pumps tested without an indoor fan installed, determine the corresponding cumulative time (in hours) of indoor coil airflow,

$\Delta\tau_a$. Sample measurements used in calculating the air volume rate (refer to sections 7.7.2.1 and 7.7.2.2 of ASHRAE Standard 37–2005 (incorporated by reference, see §430.22)) at equal intervals that span 10 minutes or less. (Note: In the first printing of ASHRAE Standard 37–2005, the second IP equation for Q_{mi} should read: .) Record the electrical energy consumed, expressed in watt-hours, from defrost termination to defrost termination, $e_{DEF}^k(35)$, as well as the corresponding elapsed time in hours, $\Delta\tau_{FR}$.

TABLE 15—TEST OPERATING AND TEST CONDITION TOLERANCES FOR FROST ACCUMULATION HEATING MODE TESTS.

	Test operating tolerance [1]		Test condition tolerance [2]
	Sub-interval H [3]	Sub-interval D [4]	Sub-interval H [3]
Indoor entering dry-bulb temperature, °F	2.0	[5] 4.0	0.5
Indoor entering wet-bulb temperature, °F	1.0		
Outdoor entering dry-bulb temperature, °F	2.0	10.0	1.0
Outdoor entering wet-bulb temperature, °F	1.5		0.5
External resistance to airflow, inches of water	0.05		0.02 [6]
Electrical voltage, % of rdg	2.0		1.5

[1] See Definition 1.41.
[2] See Definition 1.40.
[3] Applies when the heat pump is in the heating mode, except for the first 10 minutes after termination of a defrost cycle.
[4] Applies during a defrost cycle and during the first 10 minutes after the termination of a defrost cycle when the heat pump is operating in the heating mode.
[5] For heat pumps that turn off the indoor fan during the defrost cycle, the noted tolerance only applies during the 10 minute interval that follows defrost termination.
[6] Only applies when testing non-ducted heat pumps.

3.9.1 Average space heating capacity and electrical power calculations. a. Evaluate average space heating capacity, $\dot{Q}_h^k(35)$, when expressed in units of Btu per hour, using:

$$\dot{Q}_h^k(35) = \frac{60 \cdot \overline{\dot{V}} \cdot C_{p,a} \cdot \Gamma}{\Delta\tau_{FR}\left[v_n^{'} \cdot \left(1 + W_n\right)\right]} = \frac{60 \cdot \overline{\dot{V}} \cdot C_{p,a} \cdot \Gamma}{\Delta\tau_{FR} \cdot v_n}$$

where,

$\overline{\dot{V}}$ = the average indoor air volume rate measured during Sub-interval H, cfm.

$C_{p,a}$ = 0.24 + 0.444 · W_n, the constant pressure specific heat of the air-water vapor mixture that flows through the indoor coil and is expressed on a dry air basis, Btu / lbm_{da} · °F.

v_n' = specific volume of the air-water vapor mixture at the nozzle, ft^3 / lbm_{mx}.

W_n = humidity ratio of the air-water vapor mixture at the nozzle, lbm of water vapor per lbm of dry air.

$\Delta\tau_{FR}$ = $\tau_2 - \tau_1$, the elapsed time from defrost termination to defrost termination, hr.

$$\Gamma = \int_{\tau_1}^{\tau_2}\left[T_{a2}(\tau) - T_{a1}(\tau)\right]d\tau,\ \text{hr} \cdot {}^\circ\text{F}.$$

$T_{a1}(\tau)$ = dry bulb temperature of the air entering the indoor coil at elapsed time τ, °F; only recorded when indoor coil airflow occurs; assigned the value of zero during periods (if any) where the indoor fan cycles off.

$T_{a2}(\tau)$ = dry bulb temperature of the air leaving the indoor coil at elapsed time τ, °F; only recorded when indoor coil airflow occurs; assigned the value of zero during periods (if any) where the indoor fan cycles off.

τ_1 = the elapsed time when the defrost termination occurs that begins the official test period, hr.

τ_2 = the elapsed time when the next automatically occurring defrost termination occurs, thus ending the official test period, hr.

v_n = specific volume of the dry air portion of the mixture evaluated at the dry-bulb temperature, vapor content, and barometric pressure existing at the nozzle, ft^3 per lbm of dry air.

To account for the effect of duct losses between the outlet of the indoor unit and the section 2.5.4 dry-bulb temperature grid, adjust $\dot{Q}_h^k(35)$ in accordance with section 7.3.4.3 of ASHRAE Standard 37–2005 (incorporated by reference, see §430.22).

b. Evaluate average electrical power, $\dot{E}_h^k(35)$, when expressed in units of watts, using:

$$\dot{E}_h^k(35) = \frac{e_{def}(35)}{\Delta\tau_{FR}}.$$

For heat pumps tested without an indoor fan installed, increase $\dot{Q}_h^k(35)$ by,

$$\frac{1250\ \text{Btu/h}}{1000\ \text{scfm}} \cdot \overline{\dot{V}}_s \cdot \frac{\Delta\tau_a}{\Delta\tau_{FR}},$$

and increase $\dot{E}_h^k(35)$ by,

$$\frac{365\ \text{W}}{1000\ \text{scfm}} \cdot \overline{\dot{V}}_s \cdot \frac{\Delta\tau_a}{\Delta\tau_{FR}},$$

where $\overline{\dot{V}}_s$ is the average indoor air volume rate measured during the Frost Accumulation heating mode test and is expressed in units of cubic feet per minute of standard air (scfm).

c. For heat pumps having a constant-air-volume-rate indoor fan, the five additional steps listed below are required if the average of the external static pressures measured during sub-Interval H exceeds the applicable section 3.1.4.4, 3.1.4.5, or 3.1.4.6 minimum (or targeted) external static pressure (ΔP_{min}) by 0.03 inches of water or more:

1. Measure the average power consumption of the indoor fan motor ($\dot{E}_{fan,1}$) and record the corresponding external static pressure (ΔP_1) during or immediately following the Frost Accumulation heating mode test. Make the measurement at a time when the heat pump is heating, except for the first 10 minutes after the termination of a defrost cycle.

2. After the Frost Accumulation heating mode test is completed and while maintaining the same test conditions, adjust the exhaust fan of the airflow measuring apparatus until the external static pressure increases to approximately $\Delta P_1 + (\Delta P_1 - \Delta P_{min})$.

3. After re-establishing steady readings for the fan motor power and external static pressure, determine average values for the indoor fan power ($\dot{E}_{fan,2}$) and the external static pressure (ΔP_2) by making measurements over a 5-minute interval.

4. Approximate the average power consumption of the indoor fan motor had the Frost Accumulation heating mode test been conducted at ΔP_{min} using linear extrapolation:

$$\dot{E}_{fan,min} = \frac{\dot{E}_{fan,2} - \dot{E}_{fan,1}}{\Delta P_2 - \Delta P_1}\left(\Delta P_{min} - \Delta P_1\right) + \dot{E}_{fan,1}.$$

5. Decrease the total heating capacity, $\dot{Q}_h^k(35)$, by the quantity $[(\dot{E}_{fan,1} - \dot{E}_{fan,min}) \cdot (\Delta\tau_a/\Delta\tau_{FR}]$, when expressed on a Btu/h basis. Decrease the total electrical power, $E_h^k(35)$, by the same quantity, now expressed in watts.

3.9.2 Demand defrost credit. a. Assign the demand defrost credit, F_{def}, that is used in section 4.2 to the value of 1 in all cases except for heat pumps having a demand-defrost control system (Definition 1.21). For such qualifying heat pumps, evaluate F_{def} using,

$$F_{def} = 1 + 0.03 \cdot \left[1 - \frac{\Delta\tau_{def} - 1.5}{\Delta\tau_{max} - 1.5}\right],$$

where,

$\Delta\tau_{def}$ = the time between defrost terminations (in hours) or 1.5, whichever is greater.

$\Delta\tau_{max}$ = maximum time between defrosts as allowed by the controls (in hours) or 12, whichever is less.

b. For two-capacity heat pumps and for section 3.6.2 units, evaluate the above equation using the $\Delta\tau_{def}$ that applies based on the Frost Accumulation Test conducted at high capacity and/or at the Heating Full-loadAir Volume Rate. For variable-speed heat pumps, evaluate $\Delta\tau_{def}$ based on the required Frost Accumulation Test conducted at the intermediate compressor speed.

3.10 Test procedures for steady-state Low Temperature heating mode tests (the H3, $H3_2$, and $H3_1$ Tests). Except for the modifications noted in this section, conduct the Low Temperature heating mode test using the same approach as specified in section 3.7 for the Maximum and High Temperature tests. After satisfying the section 3.7 requirements for the pretest interval but before beginning to collect data to determine $\dot{Q}_h^k(17)$ and $\dot{E}_h^k(17)$, conduct a defrost cycle. This defrost cycle may be manually or automatically initiated. The defrost sequence must be terminated by the action of the heat pump's defrost controls. Begin the 30-minute data collection interval described in section 3.7, from which $\dot{Q}_h^k(17)$ and $\dot{E}_h^k(17)$ are determined, no sooner than 10 minutes after defrost termination. Defrosts should be prevented over the 30-minute data collection interval.

3.11 Additional requirements for the secondary test methods.

3.11.1 If using the Outdoor Air Enthalpy Method as the secondary test method. During the "official" test, the outdoor air-side test apparatus described in section 2.10.1 is connected to the outdoor unit. To help compensate for any effect that the addition of this test apparatus may have on the unit's performance, conduct a "preliminary" test where the outdoor air-side test apparatus is disconnected. Conduct a preliminary test prior to the first section 3.2 steady-state cooling mode test and prior to the first section 3.6 steady-state heating mode test. No other preliminary tests are required so long as the unit operates the outdoor fan during all cooling mode steady-state tests at the same speed and all heating mode steady-state tests at the same speed. If using more than one outdoor fan speed for the cooling mode steady-state tests, however, conduct a preliminary test prior to each cooling mode test where a different fan speed is first used. This same requirement applies for the heating mode tests.

3.11.1.1 If a preliminary test precedes the official test. a. The test conditions for the preliminary test are the same as specified for the official test. Connect the indoor air-side test apparatus to the indoor coil; disconnect the outdoor air-side test apparatus. Allow the test room reconditioning apparatus and the unit being tested to operate for at least one hour. After attaining equilibrium conditions, measure the following quantities at equal intervals that span 10 minutes or less:

1. The section 2.10.1 evaporator and condenser temperatures or pressures;

2. Parameters required according to the Indoor Air Enthalpy Method.

Continue these measurements until a 30-minute period (*e.g.*, four consecutive 10-minute samples) is obtained where the Table 7 or Table 13, whichever applies, test tolerances are satisfied.

b. After collecting 30 minutes of steady-state data, reconnect the outdoor air-side test apparatus to the unit. Adjust the exhaust fan of the outdoor airflow measuring apparatus until averages for the evaporator and condenser temperatures, or the saturated temperatures corresponding to the measured pressures, agree within ±0.5 °F of the averages achieved when the outdoor air-side test apparatus was disconnected. Calculate the averages for the reconnected case using five or more consecutive readings taken at one minute intervals. Make these consecutive readings after re-establishing equilibrium conditions and before initiating the official test.

3.11.1.2 If a preliminary test does not precede the official test. Connect the outdoor-side test apparatus to the unit. Adjust the exhaust fan of the outdoor airflow measuring apparatus to achieve the same external static pressure as measured during the prior preliminary test conducted with the unit operating in the same cooling or heating mode at the same outdoor fan speed.

3.11.1.3 Official test. a. Continue (preliminary test was conducted) or begin (no preliminary test) the official test by making measurements for both the Indoor and Outdoor Air Enthalpy Methods at equal intervals that span 10 minutes or less. Discontinue these measurement only after obtaining a 30-minute period where the specified test condition and test operating tolerances are satisfied. To constitute a valid official test:

(1) Achieve the energy balance specified in section 3.1.1; and,

(2) For cases where a preliminary test is conducted, the capacities determined using the Indoor Air Enthalpy Method from the official and preliminary test periods must agree within 2.0 percent.

b. For space cooling tests, calculate capacity from the outdoor air-enthalpy measurements as specified in sections 7.3.3.2 and 7.3.3.3 of ASHRAE Standard 37–2005 (incorporated by reference, see § 430.22). Calculate heating capacity based on outdoor air-enthalpy measurements as specified in sections 7.3.4.2 and 7.3.3.4.3 of the same ASHRAE Standard. Adjust the outdoor-side capacity according to section 7.3.3.4 of ASHRAE Standard 37–2005 (incorporated by reference, see § 430.22) to account for line losses when testing split systems. Use the outdoor unit fan power as measured during the official test and not the value measured during the preliminary test, as described in section 8.6.2 of ASHRAE Standard 37–2005 (incorporated by reference, see § 430.22), when calculating the capacity.

3.11.2 If using the Compressor Calibration Method as the secondary test method.

a. Conduct separate calibration tests using a calorimeter to determine the refrigerant flow rate. Or for cases where the superheat of the refrigerant leaving the evaporator is less than 5 °F, use the calorimeter to measure total capacity rather than refrigerant flow rate. Conduct these calibration tests at the same test conditions as specified for the tests in this appendix. Operate the unit for at least one hour or until obtaining equilibrium conditions before collecting data that will be used in determining the average refrigerant flow rate or total capacity. Sample the data at equal intervals that span 10 minutes or less. Determine average flow rate or average capacity from data sampled over a 30-minute period where the Table 7 (cooling) or the Table 13 (heating) tolerances are satisfied. Otherwise, conduct the calibration tests according to ASHRAE Standard 23–05 (incorporated by reference, see § 430.22), ASHRAE Standard 41.9–2000 (incorporated by reference, see § 430.22), and section 7.4 of ASHRAE Standard 37–2005 (incorporated by reference, see § 430.22).

b. Calculate space cooling and space heating capacities using the compressor calibration method measurements as specified in section 7.4.5 and 7.4.6 respectively, of ASHRAE Standard 37–2005 (incorporated by reference, see § 430.22).

b. Calculate space cooling and space heating capacities using the compressor calibration method measurements as specified in section 7.4.5 and 7.4.6 respectively, of ASHRAE Standard 37–2005 (incorporated by reference, see § 430.22).

3.11.3 If using the Refrigerant-Enthalpy Method as the secondary test method. Conduct this secondary method according to section 7.5 of ASHRAE Standard 37–2005 (incorporated by reference, see § 430.22). Calculate space cooling and heating capacities using the refrigerant-enthalpy method measurements as specified in sections 7.5.4 and 7.5.5, respectively, of the same ASHRAE Standard.

3.12 Rounding of space conditioning capacities for reporting purposes.

a. When reporting rated capacities, round them off as follows:

1. For capacities less than 20,000 Btu/h, round to the nearest 100 Btu/h.

2. For capacities between 20,000 and 37,999 Btu/h, round to the nearest 200 Btu/h.

3. For capacities between 38,000 and 64,999 Btu/h, round to the nearest 500 Btu/h.

b. For the capacities used to perform the section 4 calculations, however, round only to the nearest integer.

4. CALCULATIONS OF SEASONAL PERFORMANCE DESCRIPTORS

4.1 Seasonal Energy Efficiency Ratio (SEER) Calculations. SEER must be calculated as follows: For equipment covered under sections 4.1.2, 4.1.3, and 4.1.4, evaluate the seasonal energy efficiency ratio,

$$\mathrm{SEER} = \frac{\sum_{j=1}^{8} q_c(T_j)}{\sum_{j=1}^{8} e_c(T_j)} = \frac{\sum_{j=1}^{8} \frac{q_c(T_j)}{N}}{\sum_{j=1}^{8} \frac{e_c(T_j)}{N}} \qquad (4.1\text{-}1)$$

where,

$$\frac{q_c(T_j)}{N} =$$

the ratio of the total space cooling provided during periods of the space cooling season when the outdoor temperature fell within the range represented by bin temperature T_j to the total number of hours in the cooling season (N), Btu/h.

$$\frac{e_c(T_j)}{N} =$$

the electrical energy consumed by the test unit during periods of the space cooling season when the outdoor temperature fell within the range represented by bin temperature T_j to the total number of hours in the cooling season (N), W.

T_j = the outdoor bin temperature, °F. Outdoor temperatures are grouped or ''binned.'' Use bins of 5 °F with the 8 cooling season bin temperatures being 67, 72, 77, 82, 87, 92, 97, and 102 °F.

j = the bin number. For cooling season calculations, j ranges from 1 to 8.

Additionally, for sections 4.1.2, 4.1.3, and 4.1.4, use a building cooling load, $BL(T_j)$. When referenced, evaluate $BL(T_j)$ for cooling using,

$$BL(T_j) = \frac{(T_j - 65)}{95 - 65} \cdot \frac{\dot{Q}_c^{k=2}(95)}{1.1} \qquad (4.1\text{-}2)$$

where,

$\dot{Q}_c^{k=2}(95)$ = the space cooling capacity determined from the A_2 Test and calculated as specified in section 3.3, Btu/h.

1.1 = sizing factor, dimensionless.

The temperatures 95 °F and 65 °F in the building load equation represent the selected outdoor design temperature and the zero-load base temperature, respectively.

4.1.1 SEER calculations for an air conditioner or heat pump having a single-speed compressor that was tested with a fixed-speed indoor fan installed, a constant-air-volume-rate indoor fan installed, or with no indoor fan installed. a. Evaluate the seasonal energy efficiency ratio, expressed in units of Btu/watt-hour, using:

$SEER = PLF(0.5) \cdot EER_B$

where,

$$EER_B = \frac{\dot{Q}_c(82)}{\dot{E}_c(82)},$$

the energy efficiency ratio determined from the B Test described in sections 3.2.1, 3.1.4.1, and 3.3, Btu/h per watt.

$PLF(0.5) = 1 - 0.5 \cdot C_D^c$, the part-load performance factor evaluated at a cooling load factor of 0.5, dimensionless.

b. Refer to section 3.3 regarding the definition and calculation of $\dot{Q}_c(82)$ and $\dot{E}_c(82)$. If the optional tests described in section 3.2.1 are not conducted, set the cooling mode cyclic degradation coefficient, C_D^c, to the default value specified in section 3.5.3. If these optional tests are conducted, set C_D^c to the lower of:

1. The value calculated as per section 3.5.3; or

2. The section 3.5.3 default value of 0.25.

4.1.2 SEER calculations for an air conditioner or heat pump having a single-speed compressor and a variable-speed variable-air-volume-rate indoor fan.

4.1.2.1 Units covered by section 3.2.2.1 where indoor fan capacity modulation correlates with the outdoor dry bulb temperature. The manufacturer must provide information on how the indoor air volume rate or the indoor fan speed varies over the outdoor temperature range of 67 °F to 102 °F. Calculate SEER using Equation 4.1–1. Evaluate the quantity $q_c(T_j)/N$ in Equation 4.1–1 using,

$$\frac{q_c(T_j)}{N} = X(T_j) \cdot \dot{Q}_c(T_j) \cdot \frac{n_j}{N} \qquad (4.1.2\text{-}1)$$

where,

$$X(T_j) = \begin{Bmatrix} BL(T_j)/\dot{Q}_c(T_j) \\ \text{or} \\ 1 \end{Bmatrix};$$

whichever is less; the cooling mode load factor for temperature bin j, dimensionless.

$\dot{Q}_c(T_j)$ = the space cooling capacity of the test unit when operating at outdoor temperature, T_j, Btu/h.

n_j/N = fractional bin hours for the cooling season; the ratio of the number of hours during the cooling season when the outdoor temperature fell within the range represented by bin temperature T_j to the total number of hours in the cooling season, dimensionless.

a. For the space cooling season, assign n_j/N as specified in Table 16. Use Equation 4.1–2 to calculate the building load, $BL(T_j)$. Evaluate $\dot{Q}_c(T_j)$ using,

$$\dot{Q}_c(T_j) = \dot{Q}_c^{k=1}(T_j) + \frac{\dot{Q}_c^{k=2}(T_j) - \dot{Q}_c^{k=1}(T_j)}{FP_c^{k=2} - FP_c^{k=1}} \cdot \left[FP_c(T_j) - FP_c^{k=1}\right] \qquad (4.1.2\text{-}2)$$

where,

$$\dot{Q}_c^{k=1}(T_j) = \dot{Q}_c^{k=1}(82) + \frac{\dot{Q}_c^{k=1}(95) - \dot{Q}_c^{k=1}(82)}{95 - 82} \cdot (T_j - 82),$$

the space cooling capacity of the test unit at outdoor temperature T_j if operated at the Cooling Minimum Air Volume Rate, Btu/h.

$$\dot{Q}_c^{k=2}(T_j) = \dot{Q}_c^{k=2}(82) + \frac{\dot{Q}_c^{k=2}(95) - \dot{Q}_c^{k=2}(82)}{95 - 82} \cdot (T_j - 82),$$

the space cooling capacity of the test unit at outdoor temperature T_j if operated at the Cooling Full-load Air Volume Rate, Btu/h.

b. For units where indoor fan speed is the primary control variable, $FP_c^{k=1}$ denotes the fan speed used during the required A_1 and B_1 Tests (see section 3.2.2.1), $FP_c^{k=2}$ denotes the fan speed used during the required A_2 and B_2 Tests, and $FP_c(T_j)$ denotes the fan speed used by the unit when the outdoor temperature equals T_j. For units where indoor air volume rate is the primary control variable, the three FP_c's are similarly defined only now being expressed in terms of air volume rates rather than fan speeds. Refer to sections 3.2.2.1, 3.1.4 to 3.1.4.2, and 3.3 regarding the definitions and calculations of $\dot{Q}_c^{k=1}(82)$, $\dot{Q}_c^{k=1}(95)$, $\dot{Q}_c^{k=2}(82)$, and $\dot{Q}_c^{k=2}(95)$.

Calculate $e_c(T_j)/N$ in Equation 4.1–1 using,

$$\frac{e_c(T_j)}{N} = \frac{X(T_j) \cdot \dot{E}_c(T_j)}{PLF_j} \cdot \frac{n_j}{N} \qquad (4.1.2\text{-}3)$$

where,

$PLF_j = 1 - C_D^c \cdot [1 - X(T_j)]$, the part load factor, dimensionless.

$\dot{E}_c(T_j)$ = the electrical power consumption of the test unit when operating at outdoor temperature T_j, W.

c. The quantities $X(T_j)$ and n_j /N are the same quantities as used in Equation 4.1.2–1. If the optional tests described in section 3.2.2.1 and Table 4 are not conducted, set the cooling mode cyclic degradation coefficient, C_D^c, to the default value specified in section 3.5.3. If these optional tests are conducted, set C_D^c to the lower of:

1. The value calculated as per section 3.5.3; or

2 .The section 3.5.3 default value of 0.25.

d. Evaluate $\dot{E}_c(T_j)$ using,

$$\dot{E}_c(T_j) = \dot{E}_c^{k=1}(T_j) + \frac{\dot{E}_c^{k=2}(T_j) - \dot{E}_c^{k=1}(T_j)}{FP_c^{k=2} - FP_c^{k=1}} \cdot \left[FP_c(T_j) - FP_c^{k=1}\right] \qquad (4.1.2\text{-}4)$$

where

$$\dot{E}_c^{k=1}(T_j) = \dot{E}_c^{k=1}(82) + \frac{\dot{E}_c^{k=1}(95) - \dot{E}_c^{k=1}(82)}{95 - 82} \cdot (T_j - 82),$$

the electrical power consumption of the test unit at outdoor temperature T_j if operated at the Cooling Minimum Air Volume Rate, W.

$$\dot{E}_c^{k=2}(T_j) = \dot{E}_c^{k=2}(82) + \frac{\dot{E}_c^{k=2}(95) - \dot{E}_c^{k=2}(82)}{95 - 82} \cdot (T_j - 82),$$

the electrical power consumption of the test unit at outdoor temperature T_j if operated at the Cooling Full-load Air Volume Rate, W.

e. The parameters $FP_c^{k=1}$, and $FP_c^{k=2}$, and $FP_c(T_j)$ are the same quantities that are used when evaluating Equation 4.1.2–2. Refer to sections 3.2.2.1, 3.1.4 to 3.1.4.2, and 3.3 regarding the definitions and calculations of $\dot{E}_c^{k=1}(82)$, $\dot{E}_c^{k=1}(95)$, $\dot{E}_c^{k=2}(82)$, and $\dot{E}_c^{k=2}(95)$.

4.1.2.2 Units covered by section 3.2.2.2 where indoor fan capacity modulation is used to adjust the sensible to total cooling capacity ratio. Calculate SEER as specified in section 4.1.1.

4.1.3 SEER calculations for an air conditioner or heat pump having a two-capacity compressor. Calculate SEER using Equation 4.1–1. Evaluate the space cooling capacity, $\dot{Q}_c^{k=1}(T_j)$, and electrical power consumption, $\dot{E}_c^{k=1}(T_j)$, of the test unit when operating at low compressor capacity and outdoor temperature T_j using,

$$\dot{Q}_c^{k=1}(T_j) = \dot{Q}_c^{k=1}(67) + \frac{\dot{Q}_c^{k=1}(82) - \dot{Q}_c^{k=1}(67)}{82-67} \cdot (T_j - 67) \qquad (4.1.3\text{-}1)$$

$$\dot{E}_c^{k=1}(T_j) = E_c^{k=1}(67) + \frac{E_c^{k=1}(82) - E_c^{k=1}(67)}{82-67} \cdot (T_j - 67) \qquad (4.1.3\text{-}2)$$

where $\dot{Q}_c^{k=1}(82)$ and $\dot{E}_c^{k=1}(82)$ are determined from the B_1 Test, $\dot{Q}_c^{k=1}(67)$ and $\dot{E}_c^{k=1}(67)$ are determined from the F_1 Test, and all four quantities are calculated as specified in section 3.3. Evaluate the space cooling capacity, $\dot{Q}_c^{k=2}(T_j)$, and electrical power consumption, $\dot{E}_c^{k=2}(T_j)$, of the test unit when operating at high compressor capacity and outdoor temperature T_j using,

$$\dot{Q}_c^{k=2}(T_j) = \dot{Q}_c^{k=2}(82) + \frac{\dot{Q}_c^{k=2}(95) - \dot{Q}_c^{k=2}(82)}{95-82} \cdot (T_j - 82) \qquad (4.1.3\text{-}3)$$

$$\dot{E}_c^{k=2}(T_j) = \dot{E}_c^{k=2}(82) + \frac{\dot{E}_c^{k=2}(95) - \dot{E}_c^{k=2}(82)}{95-82} \cdot (T_j - 82) \qquad (4.1.3\text{-}4)$$

where $\dot{Q}_c^{k=2}(95)$ and $\dot{E}_c^{k=2}(95)$ are determined from the A_2 Test, $\dot{Q}_c^{k=2}(82)$, and $\dot{E}_c^{k=2}(82)$, are determined from the B_2 Test, and all are calculated as specified in section 3.3.

The calculation of Equation 4.1–1 quantities $q_c(T_j)/N$ and $e_c(T_j)/N$ differs depending on whether the test unit would operate at low capacity (section 4.1.3.1), cycle between low and high capacity (section 4.1.3.2), or operate at high capacity (sections 4.1.3.3 and 4.1.3.4) in responding to the building load. For units that lock out low capacity operation at higher outdoor temperatures, the manufacturer must supply information regarding this temperature so that the appropriate equations are used. Use Equation 4.1–2 to calculate the building load, $BL(T_j)$, for each temperature bin.

4.1.3.1 Steady-state space cooling capacity at low compressor capacity is greater than or equal to the building cooling load at temperature T_j, $\dot{Q}_c^{k=1}(T_j) \ge BL(T_j)$.

$$\frac{q_c(T_j)}{N} = X^{k=1}(T_j) \cdot \dot{Q}_c^{k=1}(T_j) \cdot \frac{n_j}{N}$$

$$\frac{e_c(T_j)}{N} = \frac{X^{k=1}(T_j) \cdot \dot{E}_c^{k=1}(T_j)}{PLF_j} \cdot \frac{n_j}{N}$$

where,

$X^{k=1}(T_j) = BL(T_j)/\dot{Q}_c^{k=1}(T_j)$, the cooling mode low capacity load factor for temperature bin j, dimensionless.

$PLF_j = 1 - C_D^c \cdot [1 - X^{k=1}(T_j)]$, the part load factor, dimensionless.

$$\frac{n_j}{N} =$$

fractional bin hours for the cooling season; the ratio of the number of hours during the cooling season when the outdoor temperature fell within the range represented by bin temperature T_j to the total number of hours in the cooling season, dimensionless.

Obtain the fractional bin hours for the cooling season, n_j/N, from Table 16. Use Equations 4.1.3–1 and 4.1.3–2, respectively, to evaluate $\dot{Q}_c^{k=1}(T_j)$ and $\dot{E}_c^{k=1}(T_j)$. If the optional tests described in section 3.2.3 and Table 5 are not conducted, set the cooling mode cyclic degradation coefficient, C_D^c, to the default value specified in section 3.5.3. If these optional tests are conducted, set C_D^c to the lower of:

a. The value calculated according to section 3.5.3; or

b. The section 3.5.3 default value of 0.25.

TABLE 16—DISTRIBUTION OF FRACTIONAL HOURS WITHIN COOLING SEASON TEMPERATURE BINS

Bin number, j	Bin temperature range °F	Representative temperature for bin °F	Fraction of of total temperature bin hours, n_j/N
1	65–69	67	0.214
2	70–74	72	0.231
3	75–79	77	0.216
4	80–84	82	0.161
5	85–89	87	0.104
6	90–94	92	0.052
7	95–99	97	0.018
8	100–104	102	0.004

4.1.3.2 Unit alternates between high (k=2) and low (k=1) compressor capacity to satisfy the building cooling load at temperature T_j, $\dot{Q}_c^{k=1}(T_j) < BL(T_j) < \dot{Q}_c^{k=2}(T_j)$.

$$\frac{q_c(T_j)}{N} = \left[X^{k=1}(T_j) \cdot \dot{Q}_c^{k=1}(T_j) + X^{k=2}(T_j) \cdot \dot{Q}_c^{k=2}(T_j)\right] \cdot \frac{n_j}{N}$$

$$\frac{e_c(T_j)}{N} = \left[X^{k=1}(T_j) \cdot \dot{E}_c^{k=1}(T_j) + X^{k=2}(T_j) \cdot \dot{E}_c^{k=2}(T_j)\right] \cdot \frac{n_j}{N}$$

where,

$$X^{k=1}(T_j) = \frac{\dot{Q}_c^{k=2}(T_j) - BL(T_j)}{\dot{Q}_c^{k=2}(T_j) - \dot{Q}_c^{k=1}(T_j)},$$

the cooling mode, low capacity load factor for temperature bin j, dimensionless.

$X^{k=2}(T_j) = 1 - X^{k=1}(T_j)$, the cooling mode, high capacity load factor for temperature bin j, dimensionless.

Obtain the fractional bin hours for the cooling season, n_j/N, from Table 16. Use Equations 4.1.3–1 and 4.1.3–2, respectively, to evaluate $\dot{Q}_c^{k=1}(T_j)$ and $\dot{E}_c^{k=1}(T_j)$. Use Equations 4.1.3–3 and 4.1.3–4, respectively, to evaluate $\dot{Q}_c^{k=2}(T_j)$ and $\dot{E}_c^{k=2}(T_j)$.

4.1.3.3 Unit only operates at high (k=2) compressor capacity at temperature T_j and its capacity is greater than the building cooling load, $BL(T_j) < \dot{Q}_c^{k=2}(T_j)$. This section applies to units that lock out low compressor capacity operation at higher outdoor temperatures.

$$\frac{q_c(T_j)}{N} = X^{k=2}(T_j) \cdot \dot{Q}_c^{k=2}(T_j) \cdot \frac{n_j}{N}$$

$$\frac{e_c(T_j)}{N} = \frac{X^{k=2}(T_j) \cdot \dot{E}_c^{k=2}(T_j)}{PLF_j} \cdot \frac{n_j}{N}$$

where,

$X^{k=2}(T_j) = BL(T_j)/\dot{Q}_c^{k=2}(T_j)$, the cooling mode high capacity load factor for temperature bin j, dimensionless.

$$PLF_j = 1 - C_D^c(k=2) \cdot \left[1 - X^{k=2}(T_j)\right],$$

the part load factor, dimensionless.

Obtain the fraction bin hours for the cooling season,

$$\frac{n_j}{N},$$

from Table 16. Use Equations 4.1.3–3 and 4.1.3–4, respectively, to evaluate $\dot{Q}_c^{k=2}(T_j)$ and $\dot{E}_c^{k=2}(T_j)$. If the optional C_2 and D_2 Tests described in section 3.2.3 and Table 5 are not conducted, set C_D^c (k=2) equal to the default value specified in section 3.5.3. If these optional tests are conducted, set C_D^c (k=2) to the lower of:

a. the C_D^c (k=2) value calculated as per section 3.5.3; or

b. the section 3.5.3 default value for C_D^c (k=2) .

4.1.3.4 Unit must operate continuously at high (k=2) compressor capacity at temperature T_j, $BL(T_j) \geq \dot{Q}_c^{k=2}(T_j)$.

$$\frac{q_c(T_j)}{N} = \dot{Q}_c^{k=2}(T_j) \cdot \frac{n_j}{N}$$

$$\frac{e_c(T_j)}{N} = \dot{E}_c^{k=2}(T_j) \cdot \frac{n_j}{N} \cdot$$

Obtain the fractional bin hours for the cooling season, n_j/N, from Table 16. Use Equations 4.1.3–3 and 4.1.3–4, respectively, to evaluate $\dot{Q}_c^{k=2}(T_j)$ and $\dot{E}_c^{k=2}(T_j)$.

4.1.4 SEER calculations for an air conditioner or heat pump having a variable-speed compressor. Calculate SEER using Equation 4.1–1. Evaluate the space cooling capacity, $\dot{Q}_c^{k=1}(T_j)$, and electrical power consumption, $\dot{E}_c^{k=1}(T_j)$, of the test unit when operating at minimum compressor speed and outdoor temperature T_j. Use,

$$\dot{Q}_c^{k=1}(T_j) = \dot{Q}_c^{k=1}(67) + \frac{\dot{Q}_c^{k=1}(82) - \dot{Q}_c^{k=1}(67)}{82-67} \cdot (T_j - 67) \qquad (4.1.4\text{-}1)$$

$$\dot{E}_c^{k=1}(T_j) = \dot{E}_c^{k=1}(67) + \frac{\dot{E}_c^{k=1}(82) - \dot{E}_c^{k=1}(67)}{82-67} \cdot (T_j - 67) \qquad (4.1.4\text{-}2)$$

where $\dot{Q}_c^{k=1}(82)$ and $\dot{E}_c^{k=1}(82)$ are determined from the B_1 Test, $\dot{Q}_c^{k=1}(67)$ and $\dot{E}_c^{k=1}(67)$ are determined from the F1 Test, and all four quantities are calculated as specified in section 3.3. Evaluate the space cooling capacity, $\dot{Q}_c^{k=2}(T_j)$, and electrical power consumption, $\dot{E}_c^{k=2}(T_j)$, of the test unit when operating at maximum compressor speed and outdoor temperature T_j. Use Equations 4.1.3–3 and 4.1.3–4, respectively, where $\dot{Q}_c^{k=2}(95)$ and $\dot{E}_c^{k=2}(95)$ are determined from the A_2 Test, $\dot{Q}_c^{k=2}(82)$ and $\dot{E}_c^{k=2}(82)$ are determined from the B_2 Test, and all four quantities are calculated as specified in section 3.3. Calculate the space cooling capacity, $\dot{Q}_c^{k=v}(T_j)$, and electrical power consumption, $\dot{E}_c^{k=v}(T_j)$, of the test unit when operating at outdoor temperature T_j and the intermediate compressor speed used during the section 3.2.4 (and Table 6) E_V Test using,

$$M_Q = \left[\frac{\dot{Q}_c^{k=1}(82) - \dot{Q}_c^{k=1}(67)}{82-67} \cdot (1 - N_Q)\right] + \left[N_Q \cdot \frac{\dot{Q}_c^{k=2}(95) - \dot{Q}_c^{k=2}(82)}{95-82}\right]$$

$$M_E = \left[\frac{\dot{E}_c^{k=1}(82) - \dot{E}_c^{k=1}(67)}{82-67} \cdot (1 - N_E)\right] + \left[N_E \cdot \frac{\dot{E}_c^{k=2}(95) - \dot{E}_c^{k=2}(82)}{95-82}\right]$$

4.1.4.1 Steady-state space cooling capacity when operating at minimum compressor speed is greater than or equal to the building cooling load at temperature T_j, $\dot{Q}_c^{k=1}(T_j) \geq BL(T_j)$.

$$\frac{q_c(T_j)}{N} = X^{k=1}(T_j) \cdot \dot{Q}_c^{k=1}(T_j) \cdot \frac{n_j}{N}$$

$$\frac{e_c(T_j)}{N} = \frac{X^{k=1}(T_j) \cdot \dot{E}_c^{k=1}(T_j)}{PLF_J} \cdot \frac{n_j}{N}$$

where,

$X^{k=1}(T_j) = BL(T_j) / \dot{Q}_c^{k=1}(T_j)$, the cooling mode minimum speed load factor for temperature bin j, dimensionless.

$PLF_j = 1 - C_D^c \cdot [1 - X^{k=1}(T_j)]$, the part load factor, dimensionless.

n_j/N = fractional bin hours for the cooling season; the ratio of the number of hours during the cooling season when the outdoor temperature fell within the range represented by bin temperature T_j to the total number of hours in the cooling season, dimensionless.

Obtain the fractional bin hours for the cooling season, n_j/N, from Table 16. Use Equations 4.1.3–1 and 4.1.3–2, respectively, to evaluate $\dot{Q}_c^{k=l}(T_j)$ and $\dot{E}_c^{k=l}(T_j)$. If the optional tests described in section 3.2.4 and Table 6 are not conducted, set the cooling mode cyclic degradation coefficient, C_D^c, to the default value specified in section 3.5.3. If these optional tests are conducted, set C_D^c to the lower of:

a. The value calculated according to section 3.5.3; or

b. The section 3.5.3 default value of 0.25.

4.1.4.2 Unit operates at an intermediate compressor speed (k=i) in order to match the building cooling load at temperature T_j, $\dot{Q}_c^{k=1}(T_j) < BL(T_j) < \dot{Q}_c^{k=2}(T_j)$.

$$\frac{q_c(T_j)}{N} = \dot{Q}_c^{k=i}(T_j) \cdot \frac{n_j}{N}$$

$$\frac{e_c(T_j)}{N} = \dot{E}_c^{k=i}(T_j) \cdot \frac{n_j}{N}$$

where,

$\dot{Q}_c^{k=i}(T_j) = BL(T_j)$, the space cooling capacity delivered by the unit in matching the building load at temperature T_j, Btu/h. The matching occurs with the unit operating at compressor speed k = i.

$$\dot{E}_c^{k=i}(T_j) = \frac{\dot{Q}_c^{k=i}(T_j)}{EER^{k=i}(T_j)},$$

the electrical power input required by the test unit when operating at a compressor speed of k = i and temperature T_j, W.

$EER^{k=i}(T_j)$ = the steady-state energy efficiency ratio of the test unit when operating at a compressor speed of k = i and temperature T_j, Btu/h per W.

Obtain the fractional bin hours for the cooling season, n_j/N, from Table 16. For each temperature bin where the unit operates at an intermediate compressor speed, determine the energy efficiency ratio $EER^{k=i}(T_j)$ using,

$EER^{k=i}(T_j) = A + B \cdot T_j + C \cdot T_j^2$.

For each unit, determine the coefficients A, B, and C by conducting the following calculations once:

$$D = \frac{T_2^2 - T_1^2}{T_v^2 - T_1^2}$$

$$B = \frac{EER^{k=1}(T_1) - EER^{k=2}(T_2) - D \cdot \left[EER^{k=1}(T_1) - EER^{k=v}(T_v)\right]}{T_1 - T_2 - D \cdot (T_1 - T_v)}$$

$$C = \frac{EER^{k=1}(T_1) - EER^{k=2}(T_2) - B \cdot (T_1 - T_2)}{T_1^2 - T_2^2}$$

$$A = EER^{k=2}(T_2) - B \cdot T_2 - C \cdot T_2^2$$

where,

T_1 = the outdoor temperature at which the unit, when operating at minimum compressor speed, provides a space cooling capacity that is equal to the building load ($\dot{Q}_c^{k=l}(T_l) = BL(T_l)$), °F. Determine T_l by equating Equations 4.1.3–1 and 4.1–2 and solving for outdoor temperature. T_v = the outdoor temperature at which the unit, when operating at the intermediate compressor speed used during the section 3.2.4 E_V Test, provides a space cooling capacity that is equal to the building load ($\dot{Q}_c^{k=v}(T_v) = BL(T_v)$), °F. Determine T_v by equating Equations 4.1.4–1 and 4.1–2 and solving for outdoor temperature.

T_2 = the outdoor temperature at which the unit, when operating at maximum compressor speed, provides a space cooling capacity that is equal to the building load ($\dot{Q}_c^{k=2}(T_2) = BL(T_2)$), °F. Determine

T_2 by equating Equations 4.1.3–3 and 4.1–2 and solving for outdoor temperature.

$$\mathrm{EER}^{k=1}(T_1) = \frac{\dot{Q}_c^{k=1}(T_1)\ [\text{Eqn. 4.1.4-1, substituting } T_1 \text{ for } T_j]}{\dot{E}_c^{k=1}(T_1)\ [\text{Eqn. 4.1.4-2, substituting } T_1 \text{ for } T_j]}, \text{ Btu/h per W.}$$

$$\mathrm{EER}^{k=v}(T_v) = \frac{\dot{Q}_c^{k=v}(T_v)\ [\text{Eqn. 4.1.4-3, substituting } T_v \text{ for } T_j]}{\dot{E}_c^{k=v}(T_v)\ [\text{Eqn. 4.1.4-4, substituting } T_v \text{ for } T_j]}, \text{ Btu/h per W.}$$

$$\mathrm{EER}^{k=2}(T_2) = \frac{\dot{Q}_c^{k=2}(T_2)\ [\text{Eqn. 4.1.3-3, substituting } T_2 \text{ for } T_j]}{\dot{E}_c^{k=2}(T_2)\ [\text{Eqn. 4.1.3-4, substituting } T_2 \text{ for } T_j]}, \text{ Btu/h per W.}$$

4.1.4.3 Unit must operate continuously at maximum (k=2) compressor speed at temperature Tj, $BL(T_j) \geq \dot{Q}_c^{k=2}(T_j)$. Evaluate the Equation 4.1–1 quantities

$$\frac{q_c(T_j)}{N} \text{ and } \frac{e_c(T_j)}{N}$$

as specified in section 4.1.3.4 with the understanding that $\dot{Q}_c^{k=2}(T_j)$ and $\dot{E}_c^{k=2}(T_j)$ correspond to maximum compressor speed operation and are derived from the results of the tests specified in section 3.2.4.

4.2 Heating Seasonal Performance Factor (HSPF) Calculations. Unless an approved alternative rating method is used, as set forth in 10 CFR 430.24(m), subpart B, HSPF must be calculated as follows: Six generalized climatic regions are depicted in Figure 2 and otherwise defined in Table 17. For each of these regions and for each applicable standardized design heating requirement, evaluate the heating seasonal performance factor using,

$$\mathrm{HSPF} = \frac{\sum_j^J n_j \cdot BL(T_j)}{\sum_j^J e_h(T_j) + \sum_j^J RH(T_j)} \cdot F_{def} = \frac{\sum_j^J \left[\frac{n_j}{N} \cdot BL(T_j)\right]}{\sum_j^J \frac{e_h(T_j)}{N} + \sum_j^J \frac{RH(T_j)}{N}} \cdot F_{def} \qquad (4.2\text{-}1)$$

where,

$e_h(T_j)/N=$

The ratio of the electrical energy consumed by the heat pump during periods of the space heating season when the outdoor temperature fell within the range represented by bin temperature T_j to the total number of hours in the heating season (N), W. For heat pumps having a heat comfort controller, this ratio may also include electrical energy used by resistive elements to maintain a minimum air delivery temperature (see 4.2.5).

$RH(T_j)/N=$

The ratio of the electrical energy used for resistive space heating during periods when the outdoor temperature fell within the range represented by bin temperature T_j to the total number of hours in the heating season (N), W. Except as noted in section 4.2.5, resistive space heating is modeled as being used to meet that portion of the building load that the heat pump does not meet because of insufficient capacity or because the heat pump automatically turns off at the lowest outdoor temperatures. For heat pumps having a heat comfort controller, all or part of the electrical energy used by resistive heaters at a particular bin temperature may be reflected in $e_h(T_j)/N$ (see 4.2.5).

T_j = the outdoor bin temperature, °F. Outdoor temperatures are ''binned'' such that calculations are only performed based one temperature within the bin. Bins of 5 °F are used.

$n_j/N=$

Fractional bin hours for the heating season; the ratio of the number of hours during the heating season when the outdoor temperature fell within the range represented by bin temperature T_j to the total number of hours in the heating season, dimensionless. Obtain n_j/N values from Table 17.

j = the bin number, dimensionless.

J = for each generalized climatic region, the total number of temperature bins, dimensionless. Referring to Table 17, J is the highest bin number (j) having a nonzero entry for the fractional bin hours for the generalized climatic region of interest.

F_{def} = the demand defrost credit described in section 3.9.2, dimensionless.

$BL(T_j)$ = the building space conditioning load corresponding to an outdoor temperature of T_j; the heating season building load also depends on the generalized climatic region's outdoor design temperature and the design heating requirement, Btu/h.

TABLE 17—GENERALIZED CLIMATIC REGION INFORMATION

		I	II	III	IV	V	VI
Region Number		I	II	III	IV	V	VI
Heating Load Hours, HLH		750	1250	1750	2250	2750	*2750
Outdoor Design Temperature, T_{OD}		37	27	17	5	−10	30
j	T_j (°F)	Fractional Bin Hours, n_j/N					
1	62	.291	.215	.153	.132	.106	.113
2	57	.239	.189	.142	.111	.092	.206
3	52	.194	.163	.138	.103	.086	.215
4	47	.129	.143	.137	.093	.076	.204
5	42	.081	.112	.135	.100	.078	.141
6	37	.041	.088	.118	.109	.087	.076
7	32	.019	.056	.092	.126	.102	.034
8	27	.005	.024	.047	.087	.094	.008
9	22	.001	.008	.021	.055	.074	.003
10	17	0	.002	.009	.036	.055	0
11	12	0	0	.005	.026	.047	0
12	7	0	0	.002	.013	.038	0
13	2	0	0	.001	.006	.029	0
14	−3	0	0	0	.002	.018	0
15	−8	0	0	0	.001	.010	0
16	−13	0	0	0	0	.005	0
17	−18	0	0	0	0	.002	0
18	−23	0	0	0	0	.001	0

*Pacific Coast Region.

Evaluate the building heating load using

$$BL(T_j) = \frac{(65 - T_j)}{65 - T_{OD}} \cdot C \cdot DHR \qquad (4.2\text{-}2)$$

where,

T_{OD} = the outdoor design temperature, °F. An outdoor design temperature is specified for each generalized climatic region in Table 17.

C = 0.77, a correction factor which tends to improve the agreement between calculated and measured building loads, dimensionless.

DHR = the design heating requirement (see Definition 1.22), Btu/h.

Calculate the minimum and maximum design heating requirements for each generalized climatic region as follows:

$$DHR_{min} = \left\{ \begin{array}{ll} \dot{Q}_h^k(47) \cdot \left[\dfrac{65 - T_{OD}}{60}\right], & \text{for Regions I, II, III, IV, \& VI} \\ \dot{Q}_h^k(47), & \text{for Region V} \end{array} \right\} \text{Rounded to the nearest standardized DHR given in Table 18.}$$

and

$$DHR_{max} = \begin{Bmatrix} 2 \cdot \dot{Q}_h^k(47) \cdot \left[\frac{65 - T_{OD}}{60}\right], \text{ for Regions I, II, III, IV, \& VI} \\ \\ 2.2 \cdot \dot{Q}_h^k(47), \qquad \text{for Region V} \end{Bmatrix} \text{Rounded to the nearest standardized DHR given in Table 18.}$$

where $\dot{Q}_h^k(47)$ is expressed in units of Btu/h and otherwise defined as follows:

1. For a single-speed heat pump tested as per section 3.6.1, $\dot{Q}_h^k(47) = \dot{Q}_h(47)$, the space heating capacity determined from the H1 Test.

2. For a variable-speed heat pump, a section 3.6.2 single-speed heat pump, or a two-capacity heat pump not covered by item 3, $\dot{Q}_h^k(47) = \dot{Q}_h^{k=2}(47)$, the space heating capacity determined from the $H1_2$ Test.

3. For two-capacity, northern heat pumps (see Definition 1.46), $\dot{Q}^k_h(47) = \dot{Q}^{k=1}_h(47)$, the space heating capacity determined from the $H1_1$ Test.

If the optional $H1_N$ Test is conducted on a variable-speed heat pump, the manufacturer has the option of defining $\dot{Q}^k_h(47)$ as specified above in item 2 or as $\dot{Q}^k_h(47)=\dot{Q}^{k=N}_h(47)$, the space heating capacity determined from the $H1_N$ Test.

For all heat pumps, HSPF accounts for the heating delivered and the energy consumed by auxiliary resistive elements when operating below the balance point. This condition occurs when the building load exceeds the space heating capacity of the heat pump condenser. For HSPF calculations for all heat pumps, see either section 4.2.1, 4.2.2, 4.2.3, or 4.2.4, whichever applies.

For heat pumps with heat comfort controllers (see Definition 1.28), HSPF also accounts for resistive heating contributed when operating above the heat-pump-plus-comfort-controller balance point as a result of maintaining a minimum supply temperature. For heat pumps having a heat comfort controller, see section 4.2.5 for the additional steps required for calculating the HSPF.

TABLE 18—STANDARDIZED DESIGN HEATING REQUIREMENTS (BTU/H)

5,000	25,000	50,000	90,000
10,000	30,000	60,000	100,000
15,000	35,000	70,000	110,000
20,000	40,000	80,000	130,000

4.2.1 Additional steps for calculating the HSPF of a heat pump having a single-speed compressor that was tested with a fixed-speed indoor fan installed, a constant-air-volume-rate indoor fan installed, or with no indoor fan installed.

$$\frac{e_h(T_j)}{N} = \frac{X(T_j) \cdot \dot{E}_h(T_j) \cdot \delta(T_j)}{PLF_j} \cdot \frac{n_j}{N} \qquad (4.2.1\text{-}1)$$

$$\frac{RH(T_j)}{N} = \frac{BL(T_j) - \left[X(T_j) \cdot \dot{Q}_h(T_j) \cdot \delta(T_j)\right]}{3.413 \frac{Btu/h}{W}} \cdot \frac{n_j}{N} \qquad (4.2.1\text{-}2)$$

where,

$$X(T_j) = \begin{Bmatrix} BL(T_J) / \dot{Q}_h(T_j) \\ \text{or} \\ 1 \end{Bmatrix}$$

whichever is less; the heating mode load factor for temperature bin j, dimensionless.

$\dot{Q}_h(T_j)$ = the space heating capacity of the heat pump when operating at outdoor temperature T_j, Btu/h.

$\dot{E}_h(T_j)$ = the electrical power consumption of the heat pump when operating at outdoor temperature T_j, W.

$\delta(T_j)$ = the heat pump low temperature cut-out factor, dimensionless.

$PLF_j = 1 - C_D^h \cdot [1 - X(T_j)]$ the part load factor, dimensionless.

Use Equation 4.2–2 to determine $BL(T_j)$. Obtain fractional bin hours for the heating season, n_j/N, from Table 17. If the optional H1C Test described in section 3.6.1 is not conducted, set the heating mode cyclic degradation coefficient, C_D^h, to the default value specified in section 3.8.1. If this optional test is conducted, set C_D^h to the lower of:

a. The value calculated according to section 3.8.1 or

b. The section 3.8.1 default value of 0.25.

Determine the low temperature cut-out factor using

$$\delta(T_j) = \begin{cases} 0, \text{ if } T_j \le T_{off} \text{ or } \dfrac{\dot{Q}_h(T_j)}{3.413 \cdot \dot{E}_h(T_j)} < 1 \\ 1/2, \text{ if } T_{off} < T_j \le T_{on} \text{ and } \dfrac{\dot{Q}_h(T_j)}{3.413 \cdot \dot{E}_h(T_j)} \ge 1 \\ 1, \text{ if } T_j > T_{on} \text{ and } \dfrac{\dot{Q}_h(T_j)}{3.413 \cdot \dot{E}_h(T_j)} \ge 1 \end{cases} \qquad (4.2.1\text{-}3)$$

where,

T_{off} = the outdoor temperature when the compressor is automatically shut off, °F. (If no such temperature exists, T_j is always greater than T_{off} and T_{on}).

T_{on} = the outdoor temperature when the compressor is automatically turned back on, if applicable, following an automatic shut-off, °F.

Calculate $\dot{Q}_h(T_j)$ and $\dot{E}_h(T_j)$ using,

$$\dot{Q}_h(T_j) = \begin{cases} \dot{Q}_h(17) + \dfrac{\left[\dot{Q}_h(47) - \dot{Q}_h(17)\right] \cdot (T_j - 17)}{47\text{-}17}, \text{ if } T_j \ge 45\ °\text{F or } T_j \le 17\ °\text{F} \\ \dot{Q}_h(17) + \dfrac{\left[\dot{Q}_h(35) - \dot{Q}_h(17)\right] \cdot (T_j - 17)}{35\text{-}17}, \text{ if } 17\ °\text{F} < T_j < 45\ °\text{F} \end{cases} \qquad (4.2.1\text{-}4)$$

$$\dot{E}_h(T_j) = \begin{cases} \dot{E}_h(17) + \dfrac{\left[\dot{E}_h(47) - \dot{E}_h(17)\right] \cdot (T_j - 17)}{47\text{-}17}, \text{ if } T_j \ge 45\ °\text{F or } T_j \le 17\ °\text{F} \\ \dot{E}_h(17) + \dfrac{\left[\dot{E}_h(35) - \dot{E}_h(17)\right] \cdot (T_j - 17)}{35\text{-}17}, \text{ if } 17\ °\text{F} < T_j < 45\ °\text{F} \end{cases} \qquad (4.2.1\text{-}5)$$

where $\dot{Q}_h(47)$ and $\dot{E}_h(47)$ are determined from the H1 Test and calculated as specified in section 3.7; $\dot{Q}_h(35)$ and $\dot{E}_h(35)$ are determined from the H2 Test and calculated as specified in section 3.9.1; and $\dot{Q}_h(17)$ and $\dot{E}_h(17)$ are determined from the H3 Test and calculated as specified in section 3.10.

4.2.2 Additional steps for calculating the HSPF of a heat pump having a single-speed compressor and a variable-speed, variable-air-volume-rate indoor fan. The manufacturer must provide information about how the indoor air volume rate or the indoor fan speed varies over the outdoor temperature range of 65 °F to −23 °F. Calculate the quantities

$$\frac{e_h(T_j)}{N} \text{ and } \frac{RH(T_j)}{N}$$

in Equation 4.2–1 as specified in section 4.2.1 with the exception of replacing references to the H1C Test and section 3.6.1 with the $H1C_1$ Test and section 3.6.2. In addition, evaluate the space heating capacity and electrical power consumption of the heat pump $\dot{Q}_h(T_j)$ and $\dot{E}_h(T_j)$ using

$$\dot{Q}_h(T_j) = \dot{Q}_h^{k=1}(T_j) + \frac{\dot{Q}_h^{k=2}(T_j) - \dot{Q}_h^{k=1}(T_j)}{FP_h^{k=2} - FP_h^{k=1}} \cdot \left[FP_h(T_j) - FP_h^{k=1}\right] \qquad (4.2.2\text{-}1)$$

$$\dot{E}_h(T_j) = \dot{E}_h^{k=1}(T_j) + \frac{\dot{E}_h^{k=2}(T_j) - \dot{E}_h^{k=1}(T_j)}{FP_h^{k=2} - FP_h^{k=1}} \cdot \left[FP_h(T_j) - FP_h^{k=1}\right] \qquad (4.2.2\text{-}2)$$

where the space heating capacity and electrical power consumption at both low capacity (k=1) and high capacity (k=2) at outdoor temperature Tj are determined using

$$\dot{Q}_h^k(T_j) = \begin{cases} \dot{Q}_h^k(17) + \dfrac{\left[\dot{Q}_h^k(47) - \dot{Q}_h^k(17)\right] \cdot (T_j - 17)}{47 - 17}, & \text{if } T_j \geq 45\ ^\circ\text{F or } T_j \leq 17\ ^\circ\text{F} \\ \dot{Q}_h^k(17) + \dfrac{\left[\dot{Q}_h^k(35) - \dot{Q}_h^k(17)\right] \cdot (T_j - 17)}{35 - 17}, & \text{if } 17\ ^\circ\text{F} < T_j < 45\ ^\circ\text{F} \end{cases} \qquad (4.2.2\text{-}3)$$

$$\dot{E}_h^k(T_j) = \begin{cases} E_h^k(17) + \dfrac{\left[\dot{E}_h^k(47) - \dot{E}_h^k(17)\right] \cdot (T_j - 17)}{47 - 17}, & \text{if } T_j \geq 45\ ^\circ\text{F or } T_j \leq 17\ ^\circ\text{F} \\ \dot{E}_h^k(17) + \dfrac{\left[\dot{E}_h^k(35) - \dot{E}_h^k(17)\right] \cdot (T_j - 17)}{35 - 17}, & \text{if } 17\ ^\circ\text{F} < T_j < 45\ ^\circ\text{F} \end{cases} \qquad (4.2.2\text{-}4)$$

For units where indoor fan speed is the primary control variable, $FP_h^{k=1}$ denotes the fan speed used during the required $H1_1$ and $H3_1$ Tests (see Table 10), $FP_h^{k=2}$ denotes the fan speed used during the required $H1_2$, $H2_2$, and $H3_2$ Tests, and $FP_h(T_j)$ denotes the fan speed used by the unit when the outdoor temperature equals T_j. For units where indoor air volume rate is the primary control variable, the three FP_h's are similarly defined only now being expressed in terms of air volume rates rather than fan speeds. Determine $\dot{Q}_h^{k=1}(47)$ and $\dot{E}_h^{k=1}(47)$ from the $H1_1$ Test, and $\dot{Q}_h^{k=2}(47)$ and $\dot{E}_h^{k=2}(47)$ from the $H1_2$ Test. Calculate all four quantities as specified in section 3.7. Determine $\dot{Q}_h^{k=1}(35)$ and $\dot{E}_h^{k=1}(35)$ as specified in section 3.6.2; determine $\dot{Q}_h^{k=2}(35)$ and $\dot{E}_h^{k=2}(35)$ and from the $H2_2$ Test and the calculation specified in section 3.9. Determine $\dot{Q}_h^{k=1}(17)$ and $\dot{E}_h^{k=1}$(17 from the $H3_1$ Test, and $\dot{Q}_h^{k=2}(17)$ and $\dot{E}_h^{k=2}(17)$ from the $H3_2$ Test. Calculate all four quantities as specified in section 3.10.

4.2.3 Additional steps for calculating the HSPF of a heat pump having a two-capacity compressor. The calculation of the Equation 4.2–1 quantities

$$\frac{e_h(T_j)}{N} \text{ and } \frac{RH(T_j)}{N}$$

differs depending upon whether the heat pump would operate at low capacity (section 4.2.3.1), cycle between low and high capacity (Section 4.2.3.2), or operate at high capacity (sections 4.2.3.3 and 4.2.3.4) in responding to the building load. For heat pumps that lock out low capacity operation at low outdoor temperatures, the manufacturer must supply information regarding the cutoff temperature(s) so that the appropriate equations can be selected.

a. Evaluate the space heating capacity and electrical power consumption of the heat pump when operating at low compressor capacity and outdoor temperature T_j using

$$\dot{Q}_h^{k=1}(T_j) = \begin{cases} \dot{Q}_h^{k=1}(47) + \dfrac{\left[\dot{Q}_h^{k=1}(62) - \dot{Q}_h^{k=1}(47)\right] \cdot (T_j - 47)}{62 - 47}, \text{if } T_j \geq 40\ °\text{F} \\ \dot{Q}_h^{k=1}(17) + \dfrac{\left[\dot{Q}_h^{k=1}(35) - \dot{Q}_h^{k=1}(17)\right] \cdot (T_j - 17)}{35 - 17}, \text{if } 17\ °\text{F} \leq T_j < 40\ °\text{F} \\ \dot{Q}_h^{k=1}(17) + \dfrac{\left[\dot{Q}_h^{k=1}(47) - \dot{Q}_h^{k=1}(17)\right] \cdot (T_j - 17)}{47 - 17}, \text{if } T_j < 17\ °\text{F} \end{cases}$$

$$\dot{E}_h^{k=1}(T_j) = \begin{cases} \dot{E}_h^{k=1}(47) + \dfrac{\left[\dot{E}_h^{k=1}(62) - \dot{E}_h^{k=1}(47)\right] \cdot (T_j - 47)}{62 - 47}, \text{ if } T_j \geq 40\ °\text{F} \\ \dot{E}_h^{k=1}(17) + \dfrac{\left[\dot{E}_h^{k=1}(35) - \dot{E}_h^{k=1}(17)\right] \cdot (T_j - 17)}{35 - 17}, \text{ if } 17\ °\text{F} \leq T_j < 40\ °\text{F} \\ \dot{E}_h^{k=1}(17) + \dfrac{\left[\dot{E}_h^{k=1}(47) - \dot{E}_h^{k=1}(17)\right] \cdot (T_j - 17)}{47 - 17}, \text{ if } T_j < 17\ °\text{F} \end{cases}$$

b. Evaluate the space heating capacity and electrical power consumption ($\dot{Q}_h^{k=2}(T_j)$ and $\dot{E}_h^{k=2}(T_j)$) of the heat pump when operating at high compressor capacity and outdoor temperature Tj by solving Equations 4.2.2–3 and 4.2.2–4, respectively, for k=2. Determine $\dot{Q}_h^{k=1}(62)$ and $\dot{E}_h^{k=1}(62)$ from the $H0_1$ Test, $\dot{Q}_h^{k=1}(47)$ and $\dot{E}_h^{k=1}(47)$ from the $H1_1$ Test, and $\dot{Q}_h^{k=2}(47)$ and $\dot{E}_h^{k=2}(47)$ from the $H1_2$ Test. Calculate all six quantities as specified in section 3.7. Determine $\dot{Q}_h^{k=2}(35)$ and $\dot{E}_h^{k=2}(35)$ from the $H2_2$ Test and, if required as described in section 3.6.3, determine $\dot{Q}_h^{k=1}(35)$ and $\dot{E}_h^{k=1}(35)$ from the $H2_1$ Test. Calculate the required 35 °F quantities as specified in section 3.9. Determine $\dot{Q}_h^{k=2}(17)$ and $\dot{E}_h^{k=2}(17)$ from the $H3_2$ Test and, if required as described in section 3.6.3, determine $\dot{Q}_h^{k=1}(17)$ and $\dot{E}_h^{k=1}(17)$ from the $H3_1$ Test. Calculate the required 17 °F quantities as specified in section 3.10.

4.2.3.1 Steady-state space heating capacity when operating at low compressor capacity is greater than or equal to the building heating load at temperature T_j, $\dot{Q}_h^{k=1}(T_j) \geq BL(T_j)$.

$$\frac{e_h(T_j)}{N} = \frac{X^{k=1}(T_j) \cdot \dot{E}_h^{k=1}(T_j) \cdot \delta'(T_j)}{PLF_j} \cdot \frac{n_j}{N} \qquad (4.2.3\text{-}1)$$

$$\frac{RH(T_j)}{N} = \frac{BL(T_j) \cdot \left[1 - \delta'(T_j)\right]}{3.413 \dfrac{\text{Btu/h}}{\text{W}}} \cdot \frac{n_j}{N} \qquad (4.2.3\text{-}2)$$

where,

$X^{k=1}(T_j) = BL(T_j) / \dot{Q}_h^{k=1}(T_j)$, the heating mode low capacity load factor for temperature bin *j*, dimensionless.

$PLF_j = 1 - C_D^h \cdot [1 - X^{k=1}(T_j)]$, the part load factor, dimensionless.

$\delta'(T_j)$ = the low temperature cutoff factor, dimensionless.

If the optional $H0C_1$ Test described in section 3.6.3 is not conducted, set the heating mode cyclic degradation coefficient, C_D^h, to the default value specified in section 3.8.1. If this optional test is conducted, set C_D^h to the lower of:

a. The value calculated according to section 3.8.1; or

b. The section 3.8.1 default value of 0.25.

Determine the low temperature cut-out factor using

$$\delta'(T_j) = \begin{cases} 0, & \text{if } T_j \le T_{off} \\ 1/2, & \text{if } T_{off} < T_j \le T_{on} \\ 1, & \text{if } T_j > T_{on} \end{cases} \qquad (4.2.3\text{-}3)$$

where T_{off} and T_{on} are defined in section 4.2.1. Use the calculations given in section 4.2.3.3, and not the above, if:

(a) The heat pump locks out low capacity operation at low outdoor temperatures and

(b) T_j is below this lockout threshold temperature.

4.2.3.2 Heat pump alternates between high (k=2) and low (k=1) compressor capacity to satisfy the building heating load at a temperature T_j, $\dot{Q}_h^{k=1}(T_j) < BL(T_j) < \dot{Q}_h^{k=2}(T_j)$. Calculate

$$\frac{RH(T_j)}{N}$$

using Equation 4.2.3–2. Evaluate

$$\frac{e_h(T_j)}{N}$$

using

$$\frac{e_h(T_j)}{N} = \left[X^{k=1}(T_j) \cdot \dot{E}_h^{k=1}(T_j) + X^{k=2}(T_j) \cdot \dot{E}_h^{k=2}(T_j)\right] \cdot \delta'(T_j) \cdot \frac{n_j}{N}$$

where,

$$X^{k=1}(T_j) = \frac{\dot{Q}_h^{k=2}(T_j) - BL(T_j)}{\dot{Q}_h^{k=2}(T_j) - \dot{Q}_h^{k=1}(T_j)}$$

$X^{k=2}(T_j) = 1 - X^{k=1}(T_j)$ the heating mode, high capacity load factor for temperature bin $_j$, dimensionless.

Determine the low temperature cut-out factor, $\delta'(T_j)$, using Equation 4.2.3–3.

4.2.3.3 Heat pump only operates at high (k=2) compressor capacity at temperature T_j and its capacity is greater than the building heating load, $BL(T_j) < \dot{Q}_h^{k=2}(T_j)$. This section applies to units that lock out low compressor capacity operation at low outdoor temperatures. Calculate

$$\frac{RH(T_j)}{N}$$

using Equation 4.2.3–2. Evaluate

$$\frac{e_h(T_j)}{N}$$

using

$$\frac{e_h(T_j)}{N} = \frac{X^{k=2}(T_j) \cdot \dot{E}_h^{k=2}(T_j) \cdot \delta'(T_j)}{PLF_j} \cdot \frac{n_j}{N}$$

where,

$X^{k=2}(T_j) = BL(T_j)/\dot{Q}_h^{k=2}(T_j)$.

$$PLF_j = 1 - C_D^h(k=2) \cdot \left[1 - X^{k=2}(T_j)\right].$$

If the optional $H1C_2$ Test described in section 3.6.3 and Table 11 is not conducted, set C_D^h (k=2) equal to the default value specified in section 3.8.1. If this optional test is conducted, set C_D^h (k=2) to the lower of:

a. the C_D^h (k=2) value calculated as per section 3.8.1; or

b. the section 3.8.1 default value for C_D^h (k=2).

Determine the low temperature cut-out factor, $\delta(T_j)$, using Equation 4.2.3–3.

4.2.3.4 Heat pump must operate continuously at high (k=2) compressor capacity at temperature T_j, $BL(T_j) \geq \dot{Q}_h^{k=2}(T_j)$.

$$\frac{e_h(T_j)}{N} = \dot{E}_h^{k=2}(T_j) \cdot \delta''(T_j) \cdot \frac{n_j}{N}$$

$$\frac{RH(T_j)}{N} = \frac{BL(T_j) - \left[\dot{Q}_h^{k=2}(T_j) \cdot \delta''(T_j)\right]}{3.413 \ \frac{Btu/h}{W}} \cdot \frac{n_j}{N}$$

Where

$$\delta''(T_j) = \begin{cases} 0, & \text{if } T_j \leq T_{off} \text{ or } \dfrac{\dot{Q}_h^{k=2}(T_j)}{3.413 \cdot \dot{E}_h^{k=2}(T_j)} < 1 \\ 1/2, & \text{if } T_{off} < T_j \leq T_{on} \text{ and } \dfrac{\dot{Q}_h^{k=2}(T_j)}{3.413 \cdot \dot{E}_h^{k=2}(T_j)} \geq 1 \\ 1, & \text{if } T_j > T_{on} \text{ and } \dfrac{\dot{Q}_h^{k=2}(T_j)}{3.413 \cdot \dot{E}_h^{k=2}(T_j)} \geq 1 \end{cases}$$

4.2.4 Additional steps for calculating the HSPF of a heat pump having a variable-speed compressor. Calculate HSPF using Equation 4.2–1. Evaluate the space heating capacity, $\dot{Q}_h^{k=1}(T_j)$, and electrical power consumption, $\dot{E}_h^{k=1}(T_j)$, of the heat pump when operating at minimum compressor speed and outdoor temperature T_j using

$$\dot{Q}_h^{k=1}(T_j) = \dot{Q}_h^{k=1}(47) + \frac{\dot{Q}_h^{k=1}(62) - \dot{Q}_h^{k=1}(47)}{62 - 47} \cdot (T_j - 47) \qquad (4.2.4\text{-}1)$$

$$\dot{E}_h^{k=1}(T_j) = \dot{E}_h^{k=1}(47) + \frac{\dot{E}_h^{k=1}(62) - \dot{E}_h^{k=1}(47)}{62 - 47} \cdot (T_j - 47) \qquad (4.2.4\text{-}2)$$

where $\dot{Q}_h^{k=1}(62)$ and $\dot{E}_h^{k=1}(62)$ are determined from the $H0_1$ Test, $\dot{Q}_h^{k=1}(47)$ and $\dot{E}_h^{k=1}(47)$ are determined from the $H1_1$ Test, and all four quantities are calculated as specified in section 3.7. Evaluate the space heating capacity, $\dot{Q}_h^{k=2}(T_j)$, and electrical power consumption, $\dot{E}_h^{k=2}(T_j)$, of the heat pump when operating at maximum compressor speed and outdoor temperature T_j by solving Equations 4.2.2–3 and 4.2.2–4, respectively, for k=2. Determine the Equation 4.2.2–3 and 4.2.2–4 quantities $\dot{Q}_h^{k=2}(47)$ and $\dot{E}_h^{k=2}(47)$ from the $H1_2$ Test and the calculations specified in section 3.7. Determine $\dot{Q}_h^{k=2}(35)$ and $\dot{E}_h^{k=2}(35)$ from the $H2_2$ Test and the calculations specified in section 3.9 or, if the $H2_2$ Test is not conducted, by conducting the calculations specified in section 3.6.4. Determine $\dot{Q}_h^{k=2}(17)$ and $\dot{E}_h^{k=2}(17)$ from the $H3_2$ Test and the calculations specified in section 3.10. Calculate the space heating capacity, $\dot{Q}_h^{k=v}(T_j)$, and electrical power consumption, $\dot{E}_h^{k=v}(T_j)$, of the heat pump when operating at outdoor temperature T_j and the intermediate compressor speed used during the section 3.6.4 $H2_V$ Test using

$$\dot{Q}_h^{k=v}(T_j) = \dot{Q}_h^{k=v}(35) + M_Q \cdot (T_j - 35) \qquad (4.2.4\text{-}3)$$

$$\dot{E}_h^{k=v}(T_j) = \dot{E}_h^{k=v}(35) + M_E \cdot (T_j - 35) \qquad (4.2.4\text{-}4)$$

where $\dot{Q}_h^{k=v}(35)$ and $\dot{E}_h^{k=v}(35)$ are determined from the H2$_V$ Test and calculated as specified in section 3.9. Approximate the slopes of the k=v intermediate speed heating capacity and electrical power input curves, M_Q and M_E, as follows:

$$M_Q = \left[\frac{\dot{Q}_h^{k=1}(62) - \dot{Q}_h^{k=1}(47)}{62-47} \cdot (1 - N_Q)\right] + \left[N_Q \cdot \frac{\dot{Q}_h^{k=2}(35) - \dot{Q}_h^{k=2}(17)}{35-17}\right]$$

$$M_E = \left[\frac{\dot{E}_h^{k=1}(62) - \dot{E}_h^{k=1}(47)}{62-47} \cdot (1 - N_E)\right] + \left[N_E \cdot \frac{\dot{E}_h^{k=2}(35) - \dot{E}_h^{k=2}(17)}{35-17}\right]$$

where,

$$N_Q = \frac{\dot{Q}_h^{k=v}(35) - \dot{Q}_h^{k=1}(35)}{\dot{Q}_h^{k=2}(35) - \dot{Q}_h^{k=1}(35)}, \text{ and}$$

$$N_E = \frac{\dot{E}_h^{k=v}(35) - \dot{E}_h^{k=1}(35)}{\dot{E}_h^{k=2}(35) - \dot{E}_h^{k=1}(35)}.$$

Use Equations 4.2.4–1 and 4.2.4–2, respectively, to calculate $\dot{Q}_h^{k=1}(35)$ and $\dot{E}_h^{k=1}(35)$.

The calculation of Equation 4.2–1 quantities

$$\frac{e_h(T_j)}{N} \text{ and } \frac{RH(T_j)}{N}$$

differs depending upon whether the heat pump would operate at minimum speed (section 4.2.4.1), operate at an intermediate speed (section 4.2.4.2), or operate at maximum speed (section 4.2.4.3) in responding to the building load.

4.2.4.1 Steady-state space heating capacity when operating at minimum compressor speed is greater than or equal to the building heating load at temperature T_j, $\dot{Q}_h^{k=1}(T_j \geq BL(T_j)$. Evaluate the Equation 4.2–1 quantities

$$\frac{e_h(T_j)}{N} \text{ and } \frac{RH(T_j)}{N}$$

as specified in section 4.2.3.1. Except now use Equations 4.2.4–1 and 4.2.4–2 to evaluate $\dot{Q}_h^{k=1}(T_j)$ and $\dot{E}_h^{k=1}(T_j)$, respectively, and replace section 4.2.3.1 references to "low capacity" and section 3.6.3 with "minimum speed" and section 3.6.4. Also, the last sentence of section 4.2.3.1 does not apply.

4.2.4.2 Heat pump operates at an intermediate compressor speed (k=i) in order to match the building heating load at a temperature T_j, $\dot{Q}_h^{k=1}(T_j) < BL(T_j) < \dot{Q}_h^{k=2}(T_j)$. Calculate

$$\frac{RH(T_j)}{N}$$

using Equation 4.2.3–2 while evaluating

$$\frac{e_h(T_j)}{N}$$

using,

$$\frac{e_h(T_j)}{N} = \dot{E}_h^{k=1}(T_j) \cdot \delta'(T_j) \cdot \frac{n_j}{N}$$

where,

$$\dot{E}_h^{k=i}(T_j) = \frac{\dot{Q}_h^{k=i}(T_j)}{3.413 \frac{Btu/h}{W} \cdot COP^{k=i}(T_j)}$$

and $\delta(T_j)$ is evaluated using Equation 4.2.3–3 while,

$\dot{Q}_h^{k=i}(T_j) = BL(T_j)$, the space heating capacity delivered by the unit in matching the building load at temperature (T_j), Btu/h. The matching occurs with the heat pump operating at compressor speed k=i.

$COP^{k=i}(T_j)$ = the steady-state coefficient of performance of the heat pump when operating at compressor speed k=i and temperature T_j, dimensionless.

For each temperature bin where the heat pump operates at an intermediate compressor speed, determine $COP^{k=i}(T_j)$ using,

$COP^{k=i}(T_j) = A + B \cdot T_j + C \cdot T_j^2$.

For each heat pump, determine the coefficients A, B, and C by conducting the following calculations once:

$$D = \frac{T_3^2 - T_4^2}{T_{vh}^2 - T_4^2}$$

$$B = \frac{COP^{k=2}(T_4) - COP^{k=1}(T_3) - D \cdot \left[COP^{k=2}(T_4) - COP^{k=v}(T_{vh})\right]}{T_4 - T_3 - D \cdot (T_4 - T_{vh})}$$

where,

T_3 = the outdoor temperature at which the heat pump, when operating at minimum compressor speed, provides a space heating capacity that is equal to the building load ($\dot{Q}_h^{k=1}(T_3) = BL(T_3)$), °F. Determine T_3 by equating Equations 4.2.4–1 and 4.2–2 and solving for:

$$C = \frac{COP^{k=2}(T_4) - COP^{k=1}(T_3) - B \cdot (T_4 - T_3)}{T_4^2 - T_3^2}$$

$$A = COP^{k=2}(T_4) - B \cdot T_4 - C \cdot T_4^2.$$

outdoor temperature.

T_{vh} = the outdoor temperature at which the heat pump, when operating at the intermediate compressor speed used during the section 3.6.4 $H2_V$ Test, provides a space heating capacity that is equal to the building load ($\dot{Q}_h^{k=v}(T_{vh}) = BL(T_{vh})$), °F. Determine T_{vh} by equating Equations 4.2.4–3 and 4.2–2 and solving for outdoor temperature.

T_4 = the outdoor temperature at which the heat pump, when operating at maximum compressor speed, provides a space heating capacity that is equal to the building load ($\dot{Q}_h^{k=2}(T_4) = BL(T_4)$), °F. Determine T_4 by equating Equations 4.2.2–3 (k=2) and 4.2–2 and solving for outdoor temperature.

$$COP^{k=1}(T_3) = \frac{\dot{Q}_h^{k=1}(T_3)\ \left[\text{Eqn. 4.2.4-1, substituting } T_3 \text{ for } T_j\right]}{3.413 \frac{Btu/h}{W} \cdot \dot{E}_h^{k=1}(T_3)\ \left[\text{Eqn. 4.2.4-2, substituting } T_3 \text{ for } T_j\right]}$$

$$COP^{k=v}(T_{vh}) = \frac{\dot{Q}_h^{k=v}(T_{vh})\ [\text{Eqn. 4.2.4-3, substituting } T_{vh} \text{ for } T_j]}{3.413 \frac{\text{Btu/h}}{\text{W}} \cdot \dot{E}_h^{k=v}(T_{vh})\ [\text{Eqn. 4.2.4-4, substituting } T_{vh} \text{ for } T_j]}$$

$$COP^{k=2}(T_4) = \frac{\dot{Q}_h^{k=2}(T_4)\ [\text{Eqn. 4.2.2-3, substituting } T_4 \text{ for } T_j]}{3.413 \frac{\text{Btu/h}}{\text{W}} \cdot \dot{E}_h^{k=2}(T_4)\ [\text{Eqn. 4.2.2-4, substituting } T_4 \text{ for } T_j]}$$

For multiple-split heat pumps (only), the following procedures supersede the above requirements for calculating $COP_h^{k=i}(T_j)$. For each temperature bin where $T_3 > T_j > T_{vh}$,

$$COP_h^{k=i}(T_j) = COP_h^{k=1}(T_3) + \frac{COP_h^{k=v}(T_{vh}) - COP_h^{k=1}(T_3)}{T_{vh} - T_3} \cdot (T_j - T_3).$$

For each temperature bin where $T_{vh} \geq T_j > T_4$,

$$COP_h^{k=i}(T_j) = COP_h^{k=v}(T_{vh}) + \frac{COP_h^{k=2}(T_4) - COP_h^{k=v}(T_{vh})}{T_4 - T_{vh}} \cdot (T_j - T_{vh}).”$$

4.2.4.3 Heat pump must operate continuously at maximum (k=2) compressor speed at temperature T_j, $BL(T_j) \geq \dot{Q}_h^{k=2}(T_j)$. Evaluate the Equation 4.2–1 quantities

$$\frac{e_h(T_j)}{N} \text{ and } \frac{RH(T_j)}{N}$$

as specified in section 4.2.3.4 with the understanding that $\dot{Q}_h^{k=2}(T_j)$ and $\dot{E}_h^{k=2}(T_j)$ correspond to maximum compressor speed operation and are derived from the results of the specified section 3.6.4 tests.

4.2.5 Heat pumps having a heat comfort controller. Heat pumps having heat comfort controllers, when set to maintain a typical minimum air delivery temperature, will cause the heat pump condenser to operate less because of a greater contribution from the resistive elements. With a conventional heat pump, resistive heating is only initiated if the heat pump condenser cannot meet the building load (*i.e.*, is delayed until a second stage call from the indoor thermostat). With a heat comfort controller, resistive heating can occur even though the heat pump condenser has adequate capacity to meet the building load (*i.e.*, both on during a first stage call from the indoor thermostat). As a result, the outdoor temperature where the heat pump compressor no longer cycles (*i.e.*, starts to run continuously), will be lower than if the heat pump did not have the heat comfort controller.

4.2.5.1 Heat pump having a heat comfort controller: additional steps for calculating the HSPF of a heat pump having a single-speed compressor that was tested with a fixed-speed indoor fan installed, a constant-air-volume-rate indoor fan installed, or with no indoor fan installed. Calculate the space heating capacity and electrical power of the heat pump without the heat comfort controller being active as specified in section 4.2.1 (Equations 4.2.1–4 and 4.2.1–5) for each outdoor bin temperature, T_j, that is listed in Table 17. Denote these capacities and electrical powers by using the subscript ‘‘hp’’ instead of ‘‘h.’’ Calculate the mass flow rate (expressed in pounds-mass of dry air per hour) and the specific heat of the indoor air (expressed in $Btu/lbm_{da} \cdot °F$) from the results of the H1 Test using:

$$\dot{m}_{da} = \bar{\dot{V}}_s \cdot 0.075\ \frac{1bm_{da}}{ft^3} \cdot \frac{60\ min}{hr} = \frac{\bar{\dot{V}}_{mx}}{v'_n \cdot [1 + W_n]} \cdot \frac{60\ min}{hr} = \frac{\bar{\dot{V}}_{mx}}{v_n} \cdot \frac{60\ min}{hr}$$

$$C_{p,da} = 0.24 + 0.444 \cdot W_n$$

where $\bar{\dot{V}}_s$, $\bar{\dot{V}}_{mx}$, v'_n (or v_n), and W_n are defined following Equation 3–1. For each outdoor bin temperature listed in Table 17, calculate the nominal temperature of the air leaving the heat pump condenser coil using,

$$T_o(T_j) = 70\ °F + \frac{\dot{Q}_{hp}(T_j)}{\dot{m}_{da} \cdot C_{p,da}}.$$

Evaluate $e_h(T_j/N)$, $RH(T_j)/N$, $X(T_j)$, PLF_j, and $\delta(T_j)$ as specified in section 4.2.1. For each bin calculation, use the space heating capacity and electrical power from Case 1 or Case 2, whichever applies.

Case 1. For outdoor bin temperatures where $T_o(T_j)$ is equal to or greater than T_{CC} (the maximum supply temperature determined according to section 3.1.9), determine $\dot{Q}_h(T_j)$ and $\dot{E}_h(T_j)$ as specified in section 4.2.1 (*i.e.*, $\dot{Q}_h(T_j)$ = $\dot{Q}_{hp}(T_j)$ and $\dot{E}_{hp}(T_j)$ = $\dot{E}_{hp}(T_j)$). Note: Even though $T_o(T_j) \geq T_{cc}$, resistive heating may be required; evaluate Equation 4.2.1–2 for all bins.

Case 2. For outdoor bin temperatures where $T_o(T_j) > T_{cc}$, determine $\dot{Q}_h(T_j)$ and $\dot{E}_h(T_j)$ using,

$$\dot{Q}_h(T_j) = \dot{Q}_{hp}(T_j) + \dot{Q}_{CC}(T_j)$$

$$\dot{E}_h(T_j) = \dot{E}_{hp}(T_j) + \dot{E}_{CC}(T_j)$$

where,

$$\dot{Q}_{CC}(T_j) = \dot{m}_{da} \cdot C_{p,da} \cdot [T_{CC} - T_o(T_j)]$$

$$\dot{E}_{CC}(T_j) = \frac{\dot{Q}_{CC}(T_j)}{3.413\ \frac{Btu}{W \cdot h}}$$

NOTE: Even though $T_o(T_j) < T_{cc}$, additional resistive heating may be required; evaluate Equation 4.2.1–2 for all bins.

4.2.5.2 Heat pump having a heat comfort controller: additional steps for calculating the HSPF of a heat pump having a single-speed compressor and a variable-speed, variable-air-volume-rate indoor fan. Calculate the space heating capacity and electrical power of the heat pump without the heat comfort controller being active as specified in section 4.2.2 (Equations 4.2.2–1 and 4.2.2–2) for each outdoor bin temperature, T_j, that is listed in Table 17. Denote these capacities and electrical powers by using the subscript "hp" instead of "h." Calculate the mass flow rate (expressed in pounds-mass of dry air per hour) and the specific heat of the indoor air (expressed in Btu/lbm$_{da}$ · °F) from the results of the H1$_2$ Test using:

$$\dot{m}_{da} = \bar{\dot{V}}_s \cdot 0.075\ \frac{1bm_{da}}{ft^3} \cdot \frac{60\ min}{hr} = \frac{\bar{\dot{V}}_{mx}}{v'_n \cdot [1 + W_n]} \cdot \frac{60\ min}{hr} = \frac{\bar{\dot{V}}_{mx}}{v_n} \cdot \frac{60\ min}{hr}$$

$$C_{p,da} = 0.24 + 0.444 \cdot W_n$$

where $\bar{\dot{V}}_S$, $\bar{\dot{V}}_{mx}$, v'_n (or v_n), and W_n are defined following Equation 3–1. For each outdoor bin temperature listed in Table 17, calculate the nominal temperature of the air leaving the heat pump condenser coil using,

$$T_o(T_j) = 70\ °F + \frac{\dot{Q}_{hp}(T_j)}{\dot{m}_{da} \cdot C_{p,da}}.$$

Evaluate $e_h(T_j)/N$, $RH(T_j)/N$, $X(T_j)$, PLF_j, and $\delta(T_j)$ as specified in section 4.2.1 with the exception of replacing references to the H1C Test and section 3.6.1 with the H1C$_1$ Test and section 3.6.2. For each bin calculation, use the space heating capacity and electrical power from Case 1 or Case 2, whichever applies.

Case 1. For outdoor bin temperatures where $T_o(T_j)$ is equal to or greater than T_{CC} (the maximum supply temperature determined according to section 3.1.9), determine $\dot{Q}_h(T_j)$ and $\dot{E}_h(T_j)$ as specified in section 4.2.2 (*i.e.* $\dot{Q}_h(T_j) = \dot{Q}_{hp}(T_j)$ and $\dot{E}_h(T_j) = \dot{E}_{hp}(T_j)$). Note: Even though $T_o(T_j) \geq T_{CC}$, resistive heating may be required; evaluate Equation 4.2.1–2 for all bins.

Case 2. For outdoor bin temperatures where $T_o(T_j) < T_{CC}$, determine $\dot{Q}_h(T_j)$ and $\dot{E}_h(T_j)$ using,

$\dot{Q}_h(T_j) = \dot{Q}_{hp}(T_j) + \dot{Q}_{CC}(T_j)$

$\dot{E}_h(T_j) = \dot{E}_{hp}(T_j) + \dot{E}_{CC}(T_j)$

where,

$\dot{Q}_{CC}(T_j) = \dot{m}_{da} \cdot C_{p,da} \cdot [T_{CC} - T_o(T_j)]$

$$\dot{E}_{CC}\left(T_j\right) = \frac{\dot{Q}_{CC}\left(T_j\right)}{3.413 \frac{\text{Btu}}{\text{W} \cdot \text{h}}}.$$

NOTE: Even though $T_o(T_j) < T_{cc}$, additional resistive heating may be required; evaluate Equation 4.2.1–2 for all bins.

4.2.5.3 Heat pumps having a heat comfort controller: additional steps for calculating the HSPF of a heat pump having a two-capacity compressor. Calculate the space heating capacity and electrical power of the heat pump without the heat comfort controller being active as specified in section 4.2.3 for both high and low capacity and at each outdoor bin temperature, T_j, that is listed in Table 17. Denote these capacities and electrical powers by using the subscript ''hp'' instead of ''h.'' For the low capacity case, calculate the mass flow rate (expressed in pounds-mass of dry air per hour) and the specific heat of the indoor air (expressed in Btu/lbm$_{da}$ · °F) from the results of the $H1_1$ Test using:

$$\dot{m}_{da}^{k=1} = \overline{\dot{V}}_s \cdot 0.075 \frac{1\text{bm}_{da}}{\text{ft}^3} \cdot \frac{60 \text{ min}}{\text{hr}} = \frac{\overline{\dot{V}}_{mx}}{v'_n \cdot \left[1 + W_n\right]} \cdot \frac{60 \text{ min}}{\text{hr}} = \frac{\overline{\dot{V}}_{mx}}{v_n} \cdot \frac{60 \text{ min}}{\text{hr}}$$

$$C_{p,da}^{k=1} = 0.24 + 0.444 \cdot W_n$$

where $\overline{\dot{V}}_s$, $\overline{\dot{V}}_{mx}$, v'_n (or v_n), and W_n are defined following Equation 3–1. For each outdoor bin temperature listed in Table 17, calculate the nominal temperature of the air leaving the heat pump condenser coil when operating at low capacity using,

$$T_o^{k=1}\left(T_j\right) = 70\ °\text{F} + \frac{\dot{Q}_{hp}^{k=1}\left(T_j\right)}{\dot{m}_{da}^{k=1} \cdot C_{p,da}^{k=1}}.$$

Repeat the above calculations to determine the mass flow rate ($\dot{m}_{da}^{k=2}$) and the specific heat of the indoor air ($C_{p,da}^{k=2}$) when operating at high capacity by using the results of the $H1_2$ Test. For each outdoor bin temperature listed in Table 17, calculate the nominal temperature of the air leaving the heat pump condenser coil when operating at high capacity using,

$$T_o^{k=2}\left(T_j\right) = 70\ °\text{F} + \frac{\dot{Q}_{hp}^{k=2}\left(T_j\right)}{\dot{m}_{da}^{k=2} \cdot C_{p,da}^{k=2}}.$$

Evaluate $e_h(T_j)/N$, $RH(T_j)/N$, $X^{k=1}(T_j)$, and/or $X^{k=2}(T_j)$, PLF_j, and $\delta'(T_j)$ or $\delta''(T_j)$ as specified in section 4.2.3.1. 4.2.3.2, 4.2.3.3, or 4.2.3.4, whichever applies, for each temperature bin. To evaluate these quantities, use the low-capacity space heating capacity and the low-capacity electrical power from Case 1 or Case 2, whichever applies; use the high-capacity space heating capacity and the high-capacity electrical power from Case 3 or Case 4, whichever applies.

Case 1. For outdoor bin temperatures where $T_o^{k=1}(T_j)$ is equal to or greater than T_{CC} (the maximum supply temperature determined according to section 3.1.9), determine $\dot{Q}_h^{k=1}(T_j)$ and $\dot{E}_h^{k=1}(T_j)$ as specified in section 4.2.3 (*i.e.*, $\dot{Q}_h^{k=1}(T_j) = \dot{Q}_{hp}^{k=1}(T_j)$ and $\dot{E}_h^{k=1}(T_j) = \dot{E}_{hp}^{k=1}(T_j)$).

NOTE: Even though $T_o^{k=1}(T_j) \geq T_{CC}$, resistive heating may be required; evaluate $RH(T_j)/N$ for all bins.

Case 2. For outdoor bin temperatures where $T_o^{k=1}(T_j) < T_{CC}$, determine $\dot{Q}_h^{k=1}(T_j)$ and $\dot{E}_h^{k=1}(T_j)$ using,

$\dot{Q}_h^{k=1}(T_j) = \dot{Q}_{hp}^{k=1}(T_j) + \dot{Q}_{CC}^{k=1}(T_j)$

$\dot{E}_h^{k=1}(T_j) = \dot{E}_{hp}^{k=1}(T_j) + \dot{E}_{CC}^{k=1}(T_j)$

where,

$$\dot{Q}_{CC}^{k=1}\left(T_j\right) = \dot{m}_{da}^{k=1} \cdot C_{p,da}^{k=1} \cdot \left[T_{CC} - T_o^{k=1}\left(T_j\right)\right]$$

$$\dot{E}_{CC}^{k=1}\left(T_j\right) = \frac{\dot{Q}_{CC}^{k=1}\left(T_j\right)}{3.413 \frac{\text{Btu}}{\text{W} \cdot \text{h}}}.$$

NOTE: Even though $T_o^{k=1}(T_j) \geq T_{cc}$, additional resistive heating may be required; evaluate $RH(T_j)/N$ for all bins.

Case 3. For outdoor bin temperatures where $T_o^{k=2}(T_j)$ is equal to or greater than T_{CC}, determine $\dot{Q}_h^{k=2}(T_j)$ and $\dot{E}_h^{k=2}(T_j)$ as specified in section 4.2.3 (*i.e.*, $\dot{Q}_h^{k=2}(T_j) = \dot{Q}_{hp}^{k=2}(T_j)$ and $\dot{E}_h^{k=2}(T_j) = \dot{E}_{hp}^{k=2}(T_j)$).

NOTE: Even though $T_o^{k=2}(T_j) < T_{CC}$, resistive heating may be required; evaluate $RH(T_j)/N$ for all bins.

Case 4. For outdoor bin temperatures where $T_o^{k=2}(T_j) < T_{CC}$, determine $\dot{Q}_h^{k=2}(T_j)$ and $\dot{E}_h^{k=2}(T_j)$ using,

$$\dot{Q}_h^{k=2}\left(T_j\right) = \dot{Q}_{hp}^{k=2}\left(T_j\right) + \dot{Q}_{CC}^{k=2}\left(T_j\right)$$

$$\dot{E}_h^{k=2}\left(T_j\right) = \dot{E}_{hp}^{k=2}\left(T_j\right) + \dot{E}_{CC}^{k=2}\left(T_j\right)$$

where,

$$\dot{Q}_{CC}^{k=2}\left(T_j\right) = \dot{m}_{da}^{k=2} \cdot C_{p,da}^{k=2} \cdot \left[T_{CC} - T_o^{k=2}\left(T_j\right)\right]$$

$$\dot{E}_{CC}^{k=2}\left(T_j\right) = \frac{\dot{Q}_{CC}^{k=2}\left(T_j\right)}{3.413 \frac{\text{Btu}}{\text{W} \cdot \text{h}}}.$$

NOTE: Even though $T_o^{k=2}(T_j) < T_{cc}$, additional resistive heating may be required; evaluate $RH(T_j)/N$ for all bins.

4.2.5.4 Heat pumps having a heat comfort controller: additional steps for calculating the HSPF of a heat pump having a variable-speed compressor. [Reserved]

4.3 Calculations of the Actual and Representative Regional Annual Performance Factors for Heat Pumps.

4.3.1 Calculation of actual regional annual performance factors (APF_A) for a particular location and for each standardized design heating requirement.

$$APF_A = \frac{CLH_A \cdot \dot{Q}_c^k(95) + HLH_A \cdot DHR \cdot C}{\frac{CLH_A \cdot \dot{Q}_c^k(95)}{SEER} + \frac{HLH_A \cdot DHR \cdot C}{HSPF}}$$

where,

CLH_A = the actual cooling hours for a particular location as determined using the map given in Figure 3, hr.

$\dot{Q}_c^k(95)$ = the space cooling capacity of the unit as determined from the A or A_2 Test, whichever applies, Btu/h.

HLH_A = the actual heating hours for a particular location as determined using the map given in Figure 2, hr.

DHR = the design heating requirement used in determining the HSPF; refer to section 4.2 and Definition 1.22, Btu/h.

C = defined in section 4.2 following Equation 4.2–2, dimensionless.

SEER = the seasonal energy efficiency ratio calculated as specified in section 4.1, Btu/W·h.

HSPF = the heating seasonal performance factor calculated as specified in section 4.2 for the generalized climatic region that includes the particular location of interest (see Figure 2), Btu/W·h. The HSPF should correspond to the actual design heating requirement (DHR), if known. If it does not, it may correspond to one of the standardized design heating requirements referenced in section 4.2.

4.3.2 Calculation of representative regional annual performance factors (APF_R) for each generalized climatic region and for each standardized design heating requirement.

$$APF_R = \frac{CLH_R \cdot \dot{Q}_c^k(95) + HLH_R \cdot DHR \cdot C}{\frac{CLH_R \cdot \dot{Q}_c^k(95)}{SEER} + \frac{HLH_R \cdot DHR \cdot C}{HSPF}}$$

where,

CLH_R = the representative cooling hours for each generalized climatic region, Table 19, hr.

HLH_R = the representative heating hours for each generalized climatic region, Table 19, hr.

HSPF = the heating seasonal performance factor calculated as specified in section 4.2 for the each generalized climatic region and for each standardized design heating requirement within each region, Btu/W.h.

The SEER, $\dot{Q}_c^k(95)$, DHR, and C are the same quantities as defined in section 4.3.1. Figure 2 shows the generalized climatic regions. Table 18 lists standardized design heating requirements.

TABLE 19—REPRESENTATIVE COOLING AND HEATING LOAD HOURS FOR EACH GENERALIZED CLIMATIC REGION

Region	CLH_R	HLH_R
I	2400	750
II	1800	1250
III	1200	1750
IV	800	2250
V	400	2750
VI	200	2750

4.4. Rounding of SEER, HSPF, and APF for reporting purposes. After calculating SEER according to section 4.1, round it off as specified in subpart B 430.23(m)(3)(i) of Title 10 of the Code of Federal Regulations. Round section 4.2 HSPF values and section 4.3 APF values as per §430.23(m)(3)(ii) and (iii) of Title 10 of the Code of Federal Regulations.

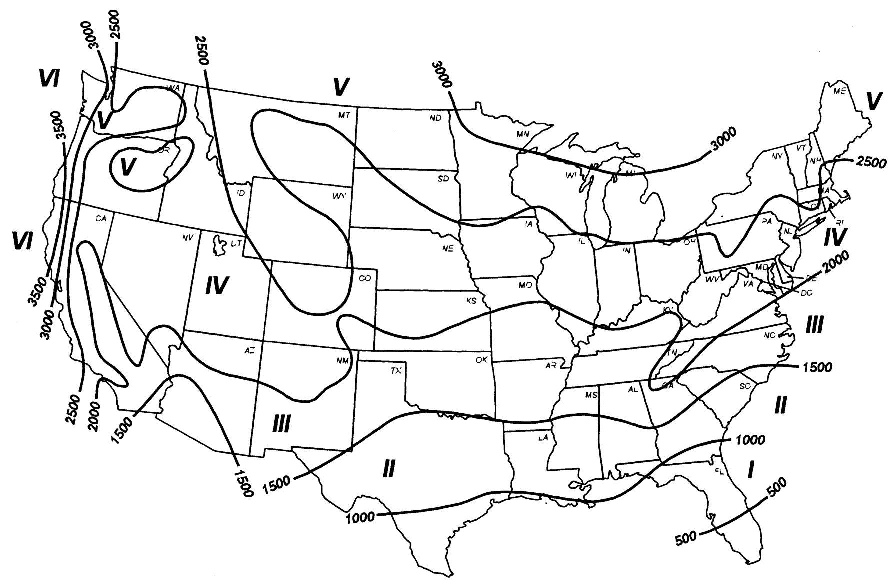

Figure 2 Heating Load Hours (HLH_A) for the United States

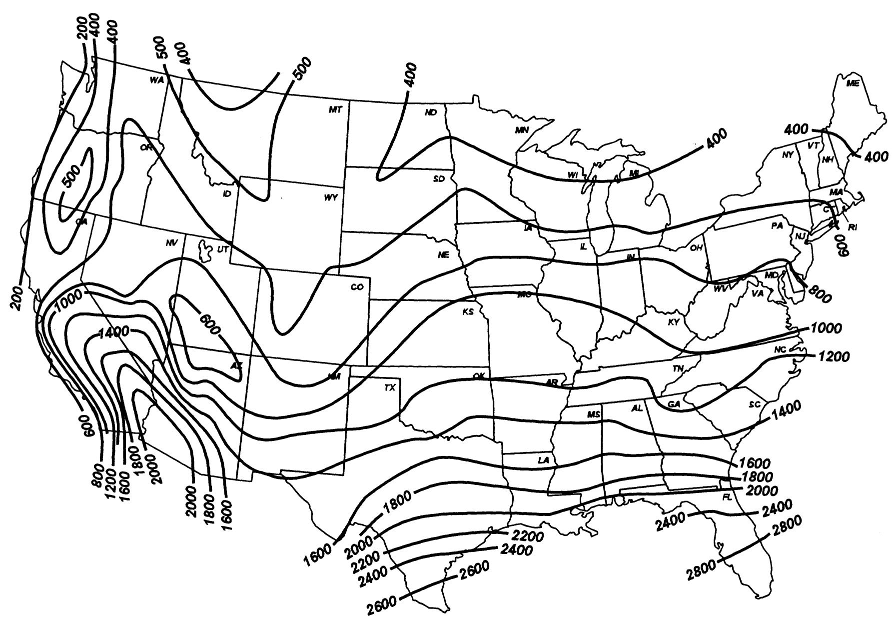

Figure 3 Cooling Load Hours (CLH_A) for the United States

[70 FR 59135, Oct. 11, 2005, as amended at 72 FR 59922, Oct. 22, 2007; 76 FR 37546, June 27, 2011]

EDITORIAL NOTE: At 72 FR 59922, Oct. 22, 2007, appendix M to subpart B of part 430 was amended; however, portions of the amendment could not be incorporated due to inaccurate amendatory instruction.