DEPARTMENT OF ENERGY

10 CFR Parts 429 and 430

[Docket No. EERE–2016–BT–TP–0029]

RIN 1904–AD71

Energy Conservation Program: Test Procedures for Central Air Conditioners and Heat Pumps

AGENCY: Office of Energy Efficiency and Renewable Energy, Department of Energy.

ACTION: Final rule.

SUMMARY: On August 24, 2016, the U.S. Department of Energy (DOE) published a supplemental notice of proposed rulemaking (SNOPR) to amend the test procedure for central air conditioners and heat pumps. That SNOPR serves as the basis for this final rule. This final rule amends the test procedure and specific certification, compliance, and enforcement provisions related to this product. In this final rule, DOE makes two sets of amendments to the test procedure: Amendments to appendix M that would be required as the basis for making efficiency representations starting 180 days after final rule publication and a new appendix M1 that would be the basis for making efficiency representations as of the compliance date for any amended energy conservation standards. The new appendix M1 establishes new efficiency metrics SEER2, EER2, and HSPF2 that are based on the current efficiency metrics for cooling and heating performance, but generally have different numerical values than the current metrics. Broadly speaking, the amendments address off-mode test procedures, test set-up and fan delays, external static pressure conditions for testing, represented values for CAC/HP that are distributed in commerce with multiple refrigerants, the methodology for testing and calculating heating performance, and testing of variable-speed systems.

DATES: The effective date of this rule is February 6, 2017. The final rule changes of appendix M will be mandatory for representations of efficiency starting July 5, 2017. Representations using appendix M1 will be mandatory starting January 1, 2023. The incorporation by reference of certain publications listed in Appendix M1 is approved by the Director of the Federal Register on February 6, 2017 February 6, 2017. The incorporation by reference of certain publications listed in Appendix M was approved by the Director of the Federal Register as of July 8, 2016.

ADDRESSES: The docket, which includes **Federal Register** notices, public meeting attendee lists and transcripts, comments, and other supporting documents/materials, is available for review at *regulations.gov*. All documents in the docket are listed in the *regulations.gov* index. However, some documents listed in the index, such as those containing information that is exempt from public disclosure, may not be publicly available.

The docket Web page can be found at *https://www.regulations.gov/docket?D=EERE-2016-BT-TP-0029*. The docket Web page will contain simple instruction on how to access all documents, including public comments, in the docket.

FOR FURTHER INFORMATION CONTACT:

Ashley Armstrong, U.S. Department of Energy, Office of Energy Efficiency and Renewable Energy, Building Technologies Program, EE–2J, 1000 Independence Avenue SW., Washington, DC 20585–0121. Telephone: (202) 586–6590. Email: *Ashley.Armstrong@ee.doe.gov*.

Johanna Jochum, U.S. Department of Energy, Office of the General Counsel, GC–33, 1000 Independence Avenue SW., Washington, DC, 20585–0121. Telephone: (202) 287–6307. Email: *Johanna.Jochum@hq.doe.gov*.

For further information on how to review public comments and the docket contact the Appliance and Equipment Standards Program staff at (202) 586–6636 or by email: *CACHeatPump2016TP0029@ee.doe.gov*.

SUPPLEMENTARY INFORMATION: This final rule incorporates by reference into part 430 specific sections, figures, and tables in the following industry standards:

(1) ANSI/AHRI 210/240–2008 with Addenda 1 and 2, (''AHRI 210/240–2008''): 2008 Standard for Performance Rating of Unitary Air-Conditioning & Air-Source Heat Pump Equipment, ANSI approved October 27, 2011;

(2) ANSI/AHRI 1230–2010 with Addendum 2, (''AHRI 1230–2010''): 2010 Standard for Performance Rating of Variable Refrigerant Flow (VRF) Multi-Split Air-Conditioning and Heat Pump Equipment, ANSI approved August 2, 2010.

Copies of AHRI 210/240–2008 and AHRI 1230–2010 can be obtained from the Air-Conditioning, Heating, and Refrigeration Institute, 2111 Wilson Boulevard, Suite 500, Arlington, VA 22201, USA, 703–524–8800, or by going to *http://www.ahrinet.org/site/686/Standards/HVACR-Industry-Standards/Search-Standards.*

(3) ANSI/ASHRAE 23.1–2010, (''ASHRAE 23.1–2010''): Methods of Testing for Rating the Performance of Positive Displacement Refrigerant Compressors and Condensing Units that Operate at Subcritical Temperatures of the Refrigerant, ANSI approved January 28, 2010;

(4) ANSI/ASHRAE Standard 37–2009, (''ANSI/ASHRAE 37–2009''), Methods of Testing for Rating Electrically Driven Unitary Air-Conditioning and Heat Pump Equipment, ANSI approved June 25, 2009;

(5) ANSI/ASHRAE 41.1–2013, (''ANSI/ASHRAE 41.1–2013''): Standard Method for Temperature Measurement, ANSI approved January 30, 2013;

(6) ANSI/ASHRAE 41.6–2014, (''ASHRAE 41.6–2014''): Standard Method for Humidity Measurement, ANSI approved July 3, 2014;

(7) ANSI/ASHRAE 41.9–2011, (''ASHRAE 41.9–2011''): Standard Methods for Volatile-Refrigerant Mass Flow Measurements Using Calorimeters, ANSI approved February 3, 2011;

(8) ANSI/ASHRAE 116–2010, (''ASHRAE 116–2010''): Methods of Testing for Rating Seasonal Efficiency of Unitary Air Conditioners and Heat Pumps, ANSI approved February 24, 2010;

(9) ANSI/ASHRAE 41.2–1987 (Reaffirmed 1992), (''ASHRAE 41.2–1987 (RA 1992)''): ''Standard Methods for Laboratory Airflow Measurement'', ANSI approved April 20, 1992.

Copies of ASHRAE 23.1–2010, ANSI/ASHRAE 37–2009, ANSI/ASHRAE 41.1–2013, ASHRAE 41.6–2014, ASHRAE 41.9–2011, ASHRAE 116–2010, and ASHRAE 41.2–1987 (RA 1992) can be purchased from ASHRAE's Web site at *https://www.ashrae.org/resources--publications*.

(10) ANSI/AMCA 210–2007, ANSI/ASHRAE 51–2007, (''AMCA 210–2007'') Laboratory Methods of Testing Fans for Certified Aerodynamic Performance Rating, ANSI approved August 17, 2007.

Copies of AMCA 210–2007 can be purchased from AMCA's Web site at *http://www.amca.org/store/index.php*.

For a further discussion of these standards, see section IV.M.

Table of Contents

I. Authority and Background

A. Authority

Title III, Part B [1] of the Energy Policy and Conservation Act of 1975 (''EPCA'' or ''the Act''), Public Law 94–163 (42 U.S.C. 6291–6309, as codified) sets forth a variety of provisions designed to improve energy efficiency and established the Energy Conservation Program for Consumer Products Other Than Automobiles.[2] These products include central air conditioners and central air conditioning heat pumps,[3] (single-phase [4] with rated cooling capacities less than 65,000 British thermal units per hour (Btu/h)), which are the focus of this Final Rule. (42 U.S.C. 6291(1)–(2), (21) and 6292(a)(3))

Under EPCA, DOE's energy conservation program generally consists of four parts: (1) Testing; (2) labeling; (3) Federal energy conservation standards; and (4) certification, compliance, and enforcement. The testing requirements consist of test procedures that manufacturers of covered products must use as the basis of: (1) Certifying to DOE that their products comply with applicable energy conservation standards adopted pursuant to EPCA, and (2) making other representations about the efficiency of those products. (42 U.S.C. 6293(c); 42 U.S.C. 6295(s)) Similarly, DOE must use these test procedures to determine whether covered products comply with any relevant standards promulgated under EPCA. (42 U.S.C. 6295(s))

EPCA sets forth criteria and procedures DOE must follow when prescribing or amending test procedures for covered products. (42 U.S.C. 6293(b)(3)) EPCA provides, in relevant part, that any test procedures prescribed or amended under this section shall be reasonably designed to produce test results which measure the energy efficiency, energy use, or estimated annual operating cost of a covered product during a representative average use cycle or period of use, and shall not be unduly burdensome to conduct. *Id.*

In addition, if DOE determines that a test procedure amendment is warranted, it must publish proposed test procedures and offer the public an opportunity to present oral and written comments on them. (42 U.S.C. 6293(b)(2)) Finally, in any rulemaking to amend a test procedure, DOE must determine to what extent, if any, the amended test procedure would alter the measured energy efficiency of any covered product as determined under the existing test procedure. (42 U.S.C. 6293(e)(1))

The Energy Independence and Security Act of 2007 (EISA 2007), Public Law 110–140, amended EPCA to require that, at least once every 7 years, DOE must review test procedures for all covered products and either amend the test procedures (if the Secretary determines that amended test procedures would more accurately or fully comply with the requirements of 42 U.S.C. 6293(b)(3)) or publish a notice in the **Federal Register** of any determination not to amend a test procedure. (42 U.S.C. 6293(b)(1)(A))

DOE's existing test procedures for CAC/HP adopted pursuant to these provisions appear under Title 10 of the Code of Federal Regulations (CFR) part 430, subpart B, appendix M (''Uniform Test Method for Measuring the Energy Consumption of Central Air Conditioners and Heat Pumps''). These procedures establish the currently permitted means for determining energy efficiency and annual energy consumption for CAC/HP. The procedures established in the new appendix M1 include new efficiency metrics to represent cooling and heating performance whose values will be altered as compared to the current metrics. The new metrics include seasonal energy efficiency ratio 2 (SEER2), energy efficiency ratio 2 (EER2), and heating seasonal performance factor 2 (HSPF2). Use of the test procedures of appendix M1 will become mandatory to demonstrate compliance on the compliance date of revised energy conservation standards.

Section 310 of EISA 2007 established that the Department's test procedures for all covered products must account for standby mode and off mode energy consumption. (42 U.S.C. 6295(gg)(2)(A)) For CAC/HP, standby mode is

[1] For editorial reasons, Part B was codified as Part A in the U.S. Code.

[2] All references to EPCA in this document refer to the statute as amended through the Energy Efficiency Improvement Act of 2015, Public Law 114–11 (Apr. 30, 2015).

[3] This rulemaking uses the term ''CAC/HP'' to refer specifically to central air conditioners (which include heat pumps) as defined by EPCA. 42 U.S.C. 6291(21.)

[4] Where this rulemaking uses the term ''CAC/HP'', they are in reference specifically to central air conditioners and heat pumps as defined by EPCA.

incorporated into the SEER and HSPF metrics, while off mode power consumption is separately regulated. This final rule includes changes relevant to the determination of both SEER and HSPF (including standby mode) and off mode power consumption.

B. Background

DOE initiated a round of test procedure revisions for CAC/HP by publishing a notice of proposed rulemaking in the **Federal Register** on June 2, 2010 (June 2010 NOPR; 75 FR 31223). Subsequently, DOE published several supplemental notices of proposed rulemaking (SNOPRs) on April 1, 2011 (April 2011 SNOPR; 76 FR 18105), on October 24, 2011 (October 2011 SNOPR: 76 FR 65616), and on November 9, 2015 (November 2015 SNOPR; 80 FR 69277) in response to comments received and to address additional needs for test procedure revisions. The June 2010 NOPR and the subsequent SNOPRs addressed a broad range of test procedure issues. On June 8, 2016, DOE published a test procedure final rule (June 2016 final rule) that finalized test procedure amendments associated with many but not all of these issues. 81 FR 36991.

On November 5, 2014, DOE published a request for information for energy conservation standards (ECS) for CAC/HP (November 2014 ECS RFI). 79 FR 65603. In response, several stakeholders provided comments suggesting that DOE amend the current test procedure. The November 2015 SNOPR addressed those test procedure-related comments, but, as mentioned in this preamble, not all of the related issues were resolved in the June 2016 final rule.

On July 14, 2015, DOE published a notice of intent to form a Working Group to negotiate a NOPR for energy conservation standards for CAC/HP and requested nominations from parties interested in serving as members of the Working Group. 80 FR 40938. The Working Group, which ultimately consisted of 15 members in addition to one member from Appliance Standards and Rulemaking Federal Advisory Committee (ASRAC) and one DOE representative, identified a number of issues related to testing and certification. The term sheet summarizing the Working Group recommendations included several recommendations associated with test procedures. (CAC ECS: ASRAC Term Sheet, No. 76) [5]

On August 24, 2016 DOE published a SNOPR (August 2016 SNOPR) proposing several amendments to the test procedure and to certification, compliance, and enforcement provisions, including a proposal to establish a new appendix M1 to be used for testing under any new energy conservation standard. 81 FR 58164. That SNOPR addressed issues not resolved by the June 2016 final rule and also proposed test procedure amendments to implement several of the items summarized in the ASRAC Working Group Term Sheet.

II. Synopsis of the Final Rule

In this final rule, DOE revises the certification requirements and test procedure for CAC/HP based on public comment on various published materials and the ASRAC negotiation process discussed in section I.B. This final rule establishes two sets of test procedure changes: One set of changes to appendix M (effective 30 days after publication of a final rule and required for testing and determining compliance with current energy conservation standards); and another set of changes to create a new appendix M1 that would be used for testing to demonstrate compliance with any amended energy conservation standards (agreed compliance date of January 1, 2023, by the Working Group in the CAC rulemaking negotiations (CAC ECS: ASRAC Term Sheet, No. 76)). With the exceptions discussed in sections III.B.3 and III.B.7, the changes to appendix M do not alter measured efficiency. However, the new appendix M1 establishes new efficiency metrics for cooling and heating performance, SEER2, EER2, and HSPF2.

In this final rule, DOE makes the following changes to certification requirements:

(1) Codifying the CAC/HP ECS Working Group's recommendation regarding delayed implementation of testing to demonstrate compliance with amended energy conservation standards;

(2) Relaxing the requirement that a split system's tested combination be a high sales volume combination;

(3) Revising requirements for certification of multi-split systems in light of the adoption of multiple categories of duct pressure drop that the indoor units can provide;

(4) Making explicit certain provisions of the service coil definition;

(5) Revising the certification of separate individual combinations within the same basic model for each refrigerant that can be used in a model of split system outdoor unit and certification of details regarding the indoor units with which unmatched outdoor units are tested;

(6) Revising representation limitations for independent coil manufacturers;

(7) Revising the certification of low-capacity lockout for air conditioner and heat pumps with two capacity compressors;

(8) Revising the requirements for represented values of cooling and heating capacity; and

(9) Adding new efficiency metrics SEER2, EER2, and HSPF2 to reflect the changes in the test procedure that result in significant change in the efficiency metric values.

DOE implements the following changes to appendix M:

(1) Requiring a limit on the internal volume of lines and devices connected to measure pressure at refrigerant circuit;

(2) Revising the method to calculate EER and coefficient of performance (COP) for variable-speed units for calculating performance at intermediate compressor speeds;

(3) Requiring a 30-minute test without the outside-air apparatus connected (a ''free outdoor air'' test) to be the official test as part of all cooling and heating mode tests which use the outdoor air enthalpy method as the secondary measurement;

(4) Relaxing the requirement for secondary capacity checks, requiring instead use of a secondary capacity measurement that agrees with the primary capacity measurement to within 6 percent only for the cooling full load test and, for heat pumps, for the heating full load test;

(5) Revising the certification of the indoor fan off delay used for coil-only tests;

(6) Modifying the test procedure for variable-speed heat pumps; and

(7) Modifying the part load testing requirement of VRF multi-split systems and test unit installation requirement of cased coil insulation and sealing.

DOE adopts the following provisions for new appendix M1:

[5] This final rule addresses proposals and comments from two rulemakings: (1) Stakeholder comments and proposals regarding the CAC test procedure (CAC TP: Docket No. EERE–2009–BT–TP–0004); and (2) stakeholder comments and proposals regarding the CAC energy conservation standard from the Working Group (CAC ECS: Docket No. EERE–2014–BT–STD–0048). Comments received through documents located in the test procedure docket are identified by ''CAC TP'' preceding the comment citation. Comments received through documents located in the energy conservation standard docket (EERE–2014–BT–STD–0048) are identified by ''CAC ECS'' preceding the comment citation. Further, comments specifically received during the CAC/HP ECS Working Group meetings are identified by ''CAC ECS: ASRAC Public Meeting'' preceding the comment citation.

(1) New higher external static pressure requirements for all units, including unique minimum external static pressure requirements for mobile home systems, ceiling-mount and wall-mount systems, low- and mid-static multi-split systems, space-constrained systems, and small-duct, high-velocity systems;

(2) A unique default fan power for rating mobile home coil-only units and new default fan power for all other coil-only units;

(3) Revisions to the heating load line equation in the calculation of the heating mode efficiency metric, HSPF2;

(4) Amendments to the test procedures for variable-speed heat pumps that change speed at lower ambient temperatures and add a 5 °F heating mode test option for calculating full-speed performance below 17 °F; and

(5) Establishment of a 4-hour or 8-hour delay time before the power measurement for units that require the crankcase heating system to reach thermal equilibrium after setting test conditions.

The test procedure amendments to appendix M for subpart B to 10 CFR part 430 established in this final rule pertaining to the efficiency of CAC/HP will be effective 30 days after publication in the **Federal Register** (referred to as the ''effective date''). Pursuant to EPCA, manufacturers of covered products are required to use the applicable test procedure as the basis for determining that their products comply with the applicable energy conservation standards. (42 U.S.C. 6295(s)) 180 days after publication of a final rule, any representations made with respect to the energy use or efficiency of CAC/HPs are required to be made in accordance with the results of testing pursuant to the amended test procedures. (42 U.S.C. 6293(c)(2))

The test procedures established in this final rule for appendix M1 to subpart B of 10 CFR part 430 pertaining to the efficiency of CAC/HP are effective 30 days after publication in the **Federal Register**. The appendix M1 procedures will be required as the basis for determining that CAC/HP comply with any amended energy conservation standards (if adopted in the concurrent CAC/HP energy conservation standards rulemaking) and for representing efficiency as of the compliance date for those amended energy conservation standards.

DOE revises the test procedure and requirements for certification, compliance, and enforcement in this final rule effective on February 6, 2017. The amended test procedure of appendix M is mandatory for representations of efficiency as of July 5, 2017. The new test procedure of appendix M1 is mandatory for representations of efficiency as of January 1, 2023.

III. Discussion

This section discusses the revisions to the certification requirements and test procedure that DOE adopts in this final rule.

A. Testing, Rating, and Compliance of Basic Models of Central Air Conditioners and Heat Pumps

1. Representation Accommodation

In the August 2016 SNOPR, DOE proposed to implement the following recommendations from the CAC/HP ECS Working Group regarding representations for split systems in 10 CFR 429.16 and 429.70:

○ DOE will implement the following accommodation for representative values of split system air conditioners and heat pumps based on the M1 methodology:

○ By January 1, 2023, manufacturers of single-split systems must validate an AEDM that is representative of the amended M1 test procedure by:

▪ Testing a single-unit sample for 20-percent of the basic models certified.

▪ The predicted performance as simulated by the AEDM must be within 5 percent of the performance resulting from the test of each of the models.

▪ Although DOE will not require that a full complement of testing be completed by January 1, 2023, manufacturers are responsible for ensuring their representations are appropriate and that the models being distributed in commerce meet the applicable standards (without a 5% tolerance).

○ By January 1, 2023, manufacturers must either determine representative values for each combination of single-split-system CAC/HP based on the M1 test procedures using a validated AEDM or through testing and the applicable sampling plan.

○ By January 1, 2023, manufacturers of multi-split, multi-circuit, or multi-head mini-split systems must determine representative values for each basic model through testing and the applicable sampling plan.

○ By July 1, 2024, each model of condensing unit of split system CAC/HP must have at least 1 combination whose rating is based on testing using the M1 test procedure and the applicable sampling plan. 81 FR at 58167 (Aug. 24, 2016)

Lennox and AHRI commented that they supported DOE's proposal, although AHRI noted it supported DOE's proposal with certain exceptions. (Lennox, No. 25 at p. 2; AHRI, No. 27 at p. 1) While AHRI did not note the exceptions, DOE assumes these may be related to their comments regarding test requirements for two-stage air conditioners (Id at p. 2), effective dates for appendix M in the June 2016 Final Rule and this final rule (Id at p. 8), and AEDM options for multi-split systems (Id at p. 20). These issues are discussed separately in III.D and III.E. As these exceptions are tangential to the original proposal, DOE has adopted the accommodations as proposed.

2. Highest Sales Volume Requirement

In the August 2016 SNOPR, based on recommendations by the CAC/HP ECS Working Group, DOE proposed removing the requirement for single-split-system air conditioners that the individual combination required for testing be the highest sales volume combination (HSVC). Specifically, DOE proposed that for every basic model, a manufacturer must test the model of outdoor unit with a model of indoor unit.[6] 81 FR at 58202 (Aug. 24, 2016)

ACEEE, NRDC, ASAP, and NEEA supported DOE's proposal to adopt the CAC/HP ECS Working Group recommendations regarding removing the HSVC, as described in the SNOPR. (ACEEE, NRDC, and ASAP, No. 33 at p. 8; NEEA, No. 35 at p. 1) DOE received no other comment on this issue. Therefore, DOE adopts this proposal in this final rule. DOE notes that some stakeholders commented on related items that were finalized in the June 2016 Final Rule. These are discussed in section III.E.1.

3. Determination of Represented Values for Multi-Split, Multi-Circuit, and Multi-Head Mini-Split Systems

In the August 2016 SNOPR, DOE proposed that multi-split, multi-head mini-split, and multi-circuit systems could be tested and rated with five kinds of indoor units: Non-ducted, low-static ducted, mid-static ducted, conventional ducted, or small-duct, high velocity (SDHV). DOE proposed that when determining represented values (including certifying compliance with amended energy conservation standards), at a minimum, a manufacturer must test and rate a ''tested combination'' composed entirely of non-ducted units. Under the proposed rule, if a manufacturer were to offer the model of outdoor unit with

[6] As adopted in the June 2016 Final Rule, for single-split-system air conditioners with single-stage or two-stage compressors, the model of indoor unit must be coil-only.

models of low-static, mid-static, and/or conventional ducted indoor units, the manufacturer would be required, at a minimum, also to test and rate a second ''tested combination'' with the highest static variety of indoor unit offered. The manufacturer would also be allowed to choose to test and rate additional ''tested combinations'' composed of the lower static varieties. In each case, the manufacturer would test with the appropriate external static pressure. DOE did not propose use of AEDMs for these systems. 81 FR at 58169 (Aug. 24, 2016)

DOE also proposed to maintain its requirement from the June 2016 final rule that, if a manufacturer also sells a model of outdoor unit with SDHV indoor units, the manufacturer must test and rate the SDHV system (*i.e.,* test a combination with indoor units that all have SDHV pressure capability). DOE also proposed to continue to allow mix-match ratings across any two of the five varieties by taking a straight average of the ratings of the individual varieties, and to allow ratings of individual combinations through testing. 81 FR at 58169 (Aug. 24, 2016)

NEEA commented that they supported DOE's proposals regarding certification of multi-split, multi-circuit, and multi-head mini-split systems. (NEEA, No. 35 at p. 1–2) Lennox and Nortek commented that they supported DOE's proposals regarding tested combinations for multi-split, multi-head mini-split, and multi-circuit systems. (Lennox, No. 25 at p. 3–4; Nortek, No. 22 at p. 3) AHRI commented that they supported DOE's proposals regarding tested combinations for multi-split and multi-circuit systems. (AHRI, No. 27 at p. 2)

AHRI and Mitsubishi commented that they were concerned with DOE's proposal to add low-static and mid-static testing requirements to appendix M. They commented that the ''low-static'' and ''mid-static'' terminology and the associated testing requirements were negotiated for appendix M1, and implementing this requirement before the effective date of the 2023 standard would not be in alignment with the Working Group's recommendation. (AHRI, No. 27 at p. 2–3; Mitsubishi, No. 29 at p. 2)

DOE notes that it intended the low-static and mid-static requirements to apply to appendix M1 only. In the August 2016 SNOPR, 10 CFR 429.16(a)(1) and (b)(2)(i) included tables regarding determining represented values and minimum testing requirements. In both of these tables, DOE only discussed the static variety in regards to testing in accordance with M1 or making representations on and after January 1, 2023. In addition, the definitions for the static varieties are only found in appendix M1. However, DOE acknowledges that 10 CFR 429.16(c)(3) may have included unclear language on this topic. DOE has modified this language in this final rule.

AHRI and Mitsubishi commented that multi-head mini-split systems do not belong in the requirements for multi-split and multi-circuit systems because they operate as 1-to-1 combinations, and it is not possible to turn off one indoor unit for testing. In addition, they stated that these systems do not have multiple-ducted and non-ducted combinations. AHRI and Mitsubishi requested that DOE remove multi-head mini-split systems from non-applicable testing requirements and other sections and instead include multi-head mini-split in the same line as ''Single-Split-System'' in the table in 10 CFR 429.16(b)(2). (AHRI, No. 27 at p. 2; Mitsubishi, No. 29 at p. 1–2; Mitsubishi, Public Meeting Transcript, No. 20 at p. 113–114)

In response, DOE notes that, though the August 2016 SNOPR proposed additional requirements regarding tested combinations, the certification and testing requirements for multi-head mini-split systems became associated with the testing requirements for multi-split and multi-circuit systems in the June 2016 final rule, and were not proposed in the August 2016 SNOPR. The only related change proposed in the August 2016 SNOPR pertains to requirements for different static varieties. Furthermore, although multi-head mini-split systems are grouped with multi-split and multi-circuit systems in the certification requirements, appendix M and M1 do not require this equipment to turn off any indoor units during testing. In addition, DOE does not believe, based on the information provided by AHRI and Mitsubishi, that the proposed language in 10 CFR 429.16 presents a problem for multi-head mini-split systems. The certification and testing requirements allow only non-ducted representations if that is all that is sold, or representations of only one kind of ducted combination, if that is all that is sold. The fact that multi-head mini-split systems are sold in few combinations should not preclude manufacturers from meeting these requirements. For these reasons, DOE is not removing multi-head mini-splits from its grouping with multi-split and multi-circuit systems in 10 CFR 429.16.

DOE received no other comment on the proposals in the August 2016 SNOPR for determining represented values for multi-split, multi-circuit, and multi-head mini-split systems and DOE adopts all of the proposed requirements in this final rule. DOE also notes that in the August 2016 SNOPR, DOE omitted mention in 10 CFR 429.16(a)(1) that non-SDHV multi-split, multi-circuit, and multi-head mini-split systems may also include space-constrained units, so DOE has clarified that in this final rule.

4. Service Coil Definition

In the June 2016 final rule, to distinguish newly installed cased and uncased coils from replacement cased and uncased coils, DOE added a definition for service coils and explicitly excluded them from indoor units in the indoor unit definition.

In the August 2016 SNOPR, DOE proposed to modify the adopted definition of service coil to more explicitly define what ''labeled accordingly'' meant. Specifically, DOE proposed that a manufacturer must designate a service coil as ''for indoor coil replacement only'' on the nameplate and in manufacturer product and technical literature. In addition, DOE proposed that the model number for any service coil must include some mechanism (*e.g.,* an additional letter or number) for differentiating a service coil from a coil intended for an indoor unit. 81 FR at 58169–58170 (Aug. 24, 2016)

AHRI, Nortek, and Ingersoll Rand commented that they support DOE's proposal. (AHRI, No. 27 at p. 3, Nortek, No. 22 at p. 3, Ingersoll Rand, No. 38 at p. 2) DOE received no other comments on this issue. Therefore, DOE is adopting this proposal in this final rule.

5. Efficiency Representations of Split-Systems for Multiple Refrigerants

DOE made numerous proposals in the August 2016 SNOPR regarding efficiency representations for multiple refrigerants, and they elicited voluminous and multi-faceted responses. The proposals themselves can be divided into three broad categories, including (1) representations for multiple refrigerants, (2) certification report requirements for outdoor units with no match, and (3) clarifying what outdoor units must have no-match efficiency representations. By far most of the responses addressed the third category—discussion thereof has been divided up into the following sub-topics: DOE authority, altering the measured efficiency, specific no-match criteria, and normalized gross indoor fin surface (NGIFS) (addressed in sections III.A.5.c through III.A.5.f).

a. Representations for Multiple Refrigerants

In the August 2016 SNOPR, to address instances in which the manufacturer indicates that more than one refrigerant is acceptable for use in a unit, DOE proposed that a split-system air conditioner or heat pump, including an outdoor unit with no match, must be certified as a separate individual combination for every acceptable refrigerant. Specifically, each individual combination would be certified under the same basic model. DOE's existing requirements for basic models would continue to apply; therefore, if an individual combination or an outdoor unit with no match fails to meet DOE's energy conservation standards using any refrigerant indicated by the manufacturer to be acceptable, then the entire basic model would fail. DOE also proposed that manufacturers must certify the refrigerants for every individual combination that is distributed in commerce. For models where the manufacturer only indicates one acceptable refrigerant, this proposal would simply entail certifying to DOE the refrigerant for which the model is designed. Finally, DOE proposed that any outdoor unit model that has certain characteristics (*e.g.,* if it is distributed in commerce without a specific refrigerant), a manufacturer must determine the represented value as an outdoor unit with no match. For some outdoor units, the proposal called for representations both as an outdoor unit with no match and as part of a combination, both as part of the same basic model. 81 FR at 58170 (Aug. 24, 2016).

The August 2016 SNOPR proposed that a refrigerant's acceptability for use in an outdoor unit would be based on its being covered under the unit's warranty, either explicitly or based on refrigerant characteristics. *Id.* at 58201.

AHRI, Nortek, Ingersoll Rand, and Carrier/UTC supported DOE's proposal that manufacturers should be required to certify efficiency ratings for all refrigerants that they have designed their equipment to use. (AHRI, No. 27 at p. 3; Nortek, No. 22 at p. 3; Ingersoll Rand, No. 38 at p. 2; Carrier/UTC, No. 36 at p. 3) AHRI, Nortek, and JCI suggested that DOE revise the requirement so that, if a manufacturer approves an air conditioner or heat pump for multiple refrigerants by listing them on the nameplate, such a product is subject to DOE certification and enforcement requirements for each approved refrigerant. AHRI, Nortek, and JCI commented that manufacturers should have the option to rate all compatible refrigerants as one basic model with the same efficiency rating, or to list different efficiencies for different refrigerants as separate basic models. AHRI, Nortek, and JCI contend that the determination of different efficiency ratings for different refrigerants should be allowed based on testing, or the appropriate use of AEDMs. (AHRI, No. 27 at p. 6; Nortek, No. 22 at p. 6; JCI, No. 24 at p. 9) Ingersoll Rand commented similarly. (Ingersoll Rand, No. 38 at p. 2)

ACEEE, NRDC, and ASAP commented that they support the proposed requirement to assign separate model numbers to systems designed for more than one refrigerant. (ACEEE, NRDC, and ASAP, No. 33 at p. 4; Lennox, No. 25 at p. 5)

Goodman commented that they agreed with DOE's proposal in principle, but were concerned that clarification regarding the refrigerants that are approved for use in a product may not always be clear, and that a refrigerant may be used in the field if information about approved refrigerants is weak or not readily identifiable. Goodman proposed regulatory text to address this issue, emphasizing reliance on a product's nameplate to indicate which refrigerants are approved. Specifically, the suggestion was that any refrigerant listed on the unit nameplate of any portion of the basic model be considered to be approved. Further, Goodman's suggestion also includes as ''approved for use'' those non-zero ozone-depleting refrigerants with similar thermophysical properties to a refrigerant listed on the nameplate, (Goodman, No. 39, p. 2–3)

In response to these comments DOE has revised the requirements so that indication of which refrigerants require certification of performance is based on the unit nameplate that is required by safety standards (*e.g.,* UL 1995) to list all approved refrigerants (see newly designated paragraph (a)(3) of section 10 CFR 429.16).

DOE does not understand Goodman's reference to ''any portion of the basic model''. If an individual combination of a basic model includes an indoor unit whose nameplate lists a refrigerant that is not listed on the outdoor unit's nameplate, such listing on the indoor unit's nameplate would not make the refrigerant approved for use in the outdoor unit. The refrigerant would therefore not be approved for use with that individual combination and presumably would not be required for certification with the basic model. Hence, if listing on the unit's nameplate is a sufficiently strong indication of which refrigerants are approved for use, it is not clear that any refrigerant listed on the indoor unit's nameplate but not on the outdoor unit's nameplate should be considered approved for use with the outdoor unit. Consequently, DOE has not included the ''any portion of the basic model'' language in its requirements. DOE has not adopted this language due to manufacturers' representations that the refrigerant listings on the nameplate are respected sufficiently that installers would not use a refrigerant in a system if it is not listed on the outdoor unit's nameplate.

DOE also is not convinced that the ''approved refrigerants'' need to include any non-zero ozone depletion potential refrigerant that has similar thermophysical properties to a refrigerant approved for use on the unit nameplate. DOE is only aware of HCFC-22 as a non-zero ozone depletion refrigerant that is used for split system air conditioners—no such alternatives are approved in the EPA SNAP list for residential and light commercial air conditioning and heat pumps.[7] HCFC-22 and refrigerants with properties similar to HCFC-22, whether non-zero ozone depletion or not, are addressed separately in the no-match requirements (see section III.A.5.e).

Additionally, in the August 2016 SNOPR, DOE did not intend to require testing of each refrigerant. In this final rule, DOE is clarifying the requirement to allow the manufacturer to test the unit with one refrigerant and to use an AEDM for other refrigerants. This clarification appears in paragraph (a)(3) of § 429.16, but DOE has also modified paragraph (c)(2) of this section to emphasize this clarification for outdoor units with no match. Additionally, in this final rule, DOE is adding a provision in paragraph (a)(3) of § 429.16 to allow grouping of refrigerants in reporting provided that the representative values represent the least efficient refrigerant. In response to ACEEE, NRDC, and ASAP, DOE does not believe the additional reporting burden of requiring that each refrigerant have its own model number and efficiency representation is justified if the rating represents the least efficient refrigerant. In response to AHRI and Nortek, DOE is requiring that all of the refrigerants for the given model of outdoor unit be part of the same basic model. This is consistent with the basic model definition adopted in the June 2016 final rule, which groups all combinations with a given model of

[7] *https://www.epa.gov/snap/acceptable-substitutes-residential-and-light-commercial-air-conditioning-and-heat-pumps.*

outdoor unit into the same basic model. 81 FR at 37053 (June 8, 2016).

b. Certification Report Requirements for Outdoor Units With no Match

DOE proposed to require reporting of additional non-public information for the indoor unit that is tested with an outdoor unit with no match. This would include the indoor coil face area, depth in the direction of airflow, fin density (fins per inch), fin material, fin style (*e.g.,* wavy or louvered), tube diameter, tube material, and numbers of tubes high and deep. These additional requirements would apply to outdoor units with no match, whether or not the outdoor unit was also certified as part of an individual combination. 81 FR at 58172 (Aug. 24, 2016).

Unico, Goodman, ACEEE, NRDC, and ASAP supported DOE in requiring that specific indoor coil descriptions be specified for outdoor units with no match. (Unico, Inc., No. 30 at p. 2; Goodman, No. 39 at p. 5; ACEEE, NRDC, and ASAP, No. 33 at p. 4)

AHRI generally did not support DOE's proposals for outdoor units with no match, but noted that the following fin styles are available as options in the AHRI Directory: Flat corrugated, high performance, lanced, louvered, and N/A. (AHRI, No. 27 at p. 7) Rheem commented that the proposed list of indoor unit details are insufficient as a measure of indoor coil performance. Rheem opposed reporting of additional non-public information for the indoor unit that is tested with an outdoor unit with no match. (Rheem, No. 37 at p. 2) Nortek similarly commented that DOE's attempt to have manufacturers describe a fin style and tube diameter is obsolete and that with the varying materials and technologies in the market, the burden of characterizing fins as ''lanced, flat, corrugated'', etc. is of no value. (Nortek, No. 22 at p. 7)

In response to the comments from AHRI, DOE will include options noted by AHRI for fin style in the certification template. In response to the comments from Rheem and Nortek, DOE notes that the reporting of information on the indoor unit is necessary for DOE's assessment and enforcement testing. DOE notes that, although Rheem indicated that the listed information is insufficient, they provided no recommendations regarding alternative ways that DOE can verify performance claimed for outdoor units with no match. Therefore, DOE adopts this requirement in this final rule.

c. DOE Authority

Per DOE's regulations in Appendix M established in the June 2016 final rule, the model of outdoor unit must be tested with an indoor unit meeting specified criteria. 81 FR at 37051 (June 8, 2016). 81 FR at 58171 (Aug. 24, 2016). Under the certification requirements proposed in the August 2016 SNOPR, DOE expanded the scope of outdoor units that would be required to be tested as outdoor units with no match. The specific criteria proposed to require such a rating are discussed in greater detail in section III.A.5.e, but they include having no designated refrigerant, a warranty that specifies refrigerant properties similar to those of HCFC-22 to define refrigerant acceptability (rather than or in addition to specific refrigerants), shipping without refrigerant or with a charge that requires addition of more than a pound of charge during setup, and shipping with any amount of R-407C. As proposed, any such unit would need to be certified as an outdoor unit with no match.

Multiple stakeholders commented on various aspects of DOE's authority to establish such requirements.

AHRI and Nortek commented that DOE has authority over manufacturers, but that DOE cannot expand that authority to make the manufacturer selling a legal product liable for the conduct of a distributor, contractor or individual consumer. They emphasized that an objective standard that could be the basis of DOE's certification and enforcement requirements will capture the conduct through which the manufacturer is distributing in commerce and marketing the equipment. (AHRI, No. 27 at p. 4; Nortek, No. 22 at p. 3–4)

DOE agrees that DOE has authority over manufacturers but notes that EPCA defines manufacture as ''to manufacture, produce, assemble, or import.'' (42 U.S.C. 6291(10))

AHRI and Nortek commented that the test requirements for outdoor units with no match represent design requirements and that DOE does not have authority to impose design requirements for central air conditioners. They noted that EPCA clearly states for some products that a standard may be a design requirement or a performance standard, but not both, and that EPCA does not even give DOE the option of considering design requirements for central air conditioners. AHRI and Nortek commented that when the use of a component with specific design requirements is mandated by the test procedure, it is in fact a design requirement for the product, since that test procedure must be used to determine the product's efficiency. (AHRI, No. 27 at p. 4–5; Nortek, No. 22 at p. 4)

In response, DOE does not agree that the test procedure imposes a design requirement as DOE does not impose any design restrictions on the outdoor unit. However, DOE must establish test procedures that are reasonably designed to measure energy efficiency during a representative average use cycle as determined by DOE (42 U.S.C. 6293 (b)(3)), which is why the indoor unit characteristics are specified. This requirement is analogous to the requirement to use higher external static pressure (ESP) when testing an SDHV system. DOE also notes that its delineation of outdoor units with no match is for units that are predominantly used to replace failed HCFC-22 outdoor units. As such, DOE has developed a straightforward approach to defining the characteristics of an indoor unit which is representative of such applications in order to allow the test procedure for these units to be representative of field installation. The extension of this concept to additional categories of outdoor units with no match (other than those designed for HCFC-22) does not invalidate this premise. For example, DOE has no evidence that outdoor units designed for use with R-407C are installed to a significant extent with new indoor units. Further discussion regarding the specific criteria to identify outdoor units with no match is in section III.A.5.e.

AHRI and Nortek commented that DOE's proposal for outdoor units with no match would be an expansion into technical and policy issues that are outside of DOE's authority under EPCA, were not within Congress' intent in granting DOE authority over energy efficiency standards, and are the jurisdiction of the EPA. They assert that the proposed approach would effectively ban the sale of otherwise legal products by requiring the very restrictive no match testing. (AHRI, No. 27 at p. 5; Nortek, No. 22 at p. 4–5) Similarly, JCI commented that DOE's R-407C proposal effectively bans the use of R-407C in split-system CACs and HPs by proposing to burden R-407C units with more stringent testing requirements than units designed for use with any other EPA–SNAP approved refrigerant, requiring testing with an inefficient indoor unit, and thus requiring outdoor unit efficiency that is either technically impossible or economically inviable to meet. JCI commented that this refrigerant-specific test procedure requirement constitutes back-door regulation of R-407C by DOE even though R-407C is already subject to direct regulation by EPA under the Clean Air Act, and EPA has permitted the use of R-407C in split system CAC/HPs. In proposing to manipulate the CAC/HP test procedure in a way that would eliminate the use of R-407C in split-system CAC/HPs, JCI stated that DOE is acting beyond its legal authority under EPCA. (JCI, No. 24 at p. 3–4)

Ingersoll Rand agrees with AHRI's position that these proposed requirements exceed DOE's statutory authority. (Ingersoll Rand, No. 38 at p. 3)

On the other hand, ACEEE, NRDC, and ASAP commented that DOE regulates energy efficiency and has a legal obligation to ensure that manufacturers comply with its standards. According to ACEEE, NRDC, and ASAP, the August 2016 test procedure SNOPR does precisely that by ensuring that units intended as replacement units have to meet the same rules regardless of the refrigerant they are designed to use. ACEEE, NRDC, and ASAP commented that in the SNOPR, DOE clearly set out to close a loophole in its own regulations that, if left unaddressed, would result in the sale of units that do not meet existing standards, resulting in higher energy consumption. ACEEE, NRDC, and ASAP commented that closing that loophole is the purpose of DOE's ''no-match'' requirements for certifying these units. ACEEE, NRDC, and ASAP further commented that DOE is not banning the sale of R-407C units and that selling outdoor unit replacements using R-407C is and will continue to be perfectly legal—in fact, manufacturers may produce and sell outdoor units with no match using any refrigerant they want, including R-22 and R-407C. They commented that these units will need to meet the efficiency of DOE's existing minimum standards, rather than skate by with a certified value not achieved in the real world. They expressed the view that DOE's SNOPR effectively addresses the efficiency performance of products on the market today. (ACEEE, NRDC, and ASAP, No. 33 at p. 11) ACEEE, NRDC, and ASAP also indicated that some products, including the R-407C products introduced to the market in 2016, can only meet the existing standards by pairing the outdoor unit with an oversized indoor unit, even though the units are sold as replacements for outdoor units in which the existing indoor unit is not replaced. They further stated that other combinations in which the outdoor and indoor units are mismatched are unlikely to be sold in these combinations in any significant quantity. (ACEEE, NRDC, and ASAP, No. 33 at p. 4) Lennox also commented that ''a manufacturer'' rated an outdoor unit for R-407C by matching the outdoor unit with an unusually large indoor coil and sold it with one pound of refrigerant charge as a replacement for HCFC-22 units. (Lennox, No. 25 at p. 4)

Contrary to the comments of AHRI, JCI, Nortek, and Ingersoll Rand, EPCA requires DOE to establish appropriate test procedures with which to measure product efficiency for a representative average use cycle. (42 U.S.C. 6293(b)(3)) DOE's proposals regarding outdoor units with no match are based on efficiency considerations and supported by DOE's authority granted by EPCA to regulate product efficiency and to establish appropriate test procedures with which to measure product efficiency. JCI commented that when consumers are offered the option to use R-407C, as opposed to HCFC-22, they take advantage of it, citing that sales of R-407C are rising proportionately with JCI's sales of R-407C units, and pointing out that they are giving customers the opportunity to avoid HCFC-22 refrigerant without entirely replacing their CAC/HP systems. (JCI, No. 24 at p. 7) These statements support DOE's expectation that the sales of these R-407C units are primarily, if not entirely, for no-match installations in which the indoor unit is not replaced. Although JCI claims that DOE cannot extend its arguments made for HCFC-22 outdoor units (*i.e.,* that they are clearly no-match installations because there is no valid EPA-approved combination that includes an HCFC-22 outdoor unit (JCI, No. 24 at p. 5)), DOE asserts that the possibility that there are or could be a few valid R-407C combinations sold does not in itself make sales of combinations (rather than no-match sales) the representative efficiency value for R-407C.

JCI also claimed that DOE has no authority to regulate outdoor units with no match because they are not a central air conditioner or a heat pump as defined by EPCA. (JCI, No. 24 at p. 4) DOE notes that in the June 2016 Final Rule, DOE reasonably interpreted the statutory definition to specify the following: ''A central air conditioner or central air conditioning heat pump may consist of: a single-package unit; an outdoor unit and one or more indoor units; an indoor unit only; or an outdoor unit with no match. In the case of an indoor unit only or an outdoor unit with no match, the unit must be tested and rated as a system (combination of both an indoor and an outdoor unit).'' 81 FR at 37056 (June 8, 2016). In that rule, DOE noted that this interpretation did not change the scope of DOE's product coverage and is in line with the current certification requirements for CAC/HP. 81 FR at 36999.

d. Altering the Measured Efficiency

In the August 2016 public meeting, JCI commented that they offer a matched combination with R-407C, and that the tested combination is available in the AHRI database. JCI noted that the product has been available since spring 2016, and it is too early to say that there is no tested combination of this product. JCI also questioned how long after introduction of an outdoor unit product an assessment can be made whether there is or is not a highest sales volume combination. (JCI, Public Meeting Transcript, No. 20 at pp. 124–132) In written comments, JCI cited EPCA requirements that when amending test procedures, DOE must consider to what extent the amendments alter the measured efficiency of covered products, and then amend the applicable energy conservation standards if a determination is made that the test procedure amendment alters the measurement. (42 U.S.C. 6293(e)(1–2)) JCI commented that DOE has not done this for its amendments associated with no-match R-407C products. JCI explained that the no-match proposals would force manufacturers to re-test previously certified compliant products using a new testing standard that is technically impossible to meet, which would render the previously-compliant R-407C systems non-compliant. (JCI, No. 24 at p. 6)

This test procedure provides a mechanism of assessing the performance of no-match products, such as those that use R-407C, which can then be used to provide a reasonable level of assurance that all field-match combinations of the new, unmatched outdoor units will achieve the established efficiency levels. The current test procedure requires that single-stage split system air conditioners be tested using the highest sales volume tested combination. 10 CFR 429.16. It is DOE's understanding that condensing units utilizing R407C typically do not have a highest sales volume indoor unit that satisfy the requirements of the test procedure and thus, could not be tested under the current regulatory regime. Further, if the condensing units were to have a highest sales volume indoor unit for testing, DOE believes the results of such testing would overstate the performance of R407C systems as installed. DOE believes this is the case because R407C systems typically get installed with existing indoor units, which are not properly sized, in order to achieve the system efficiency that would result from a new matched pair system. Thus, DOE believes that manufacturers of R407C condensing units should have sought a waiver for the current test procedure requirements pursuant to the procedures at 10 CFR 430.27. EPCA requires DOE to adopt test procedures that are reasonably designed to produce test results which measure energy efficiency of a covered product during a representative average use cycle or period of use. (42 U.S.C. 6293(b)(3)) To meet this requirement for outdoor units with no match, DOE is now adopting an alternative approach similar to the proposal with modification for testing and determining represented values for no-match R407C products based on stakeholder comments. DOE notes that under the approach adopted in this final rule, the testing method for no-match systems does not consider HSVC. In this rulemaking, the only proposal regarding HSVC was to remove the requirement for single-split system air conditioners, which DOE adopts as discussed in section III.A.2. The application of HSVC to current applicable regulations is not within the scope of this rulemaking. Therefore, DOE will not address its application in this rule.

JCI also questioned whether DOE performed any analysis on how the new requirements for units with R-407C refrigerant impact consumers. (JCI, Public Meeting Transcript, No. 20 at pp. 137–139)

In response, DOE does not evaluate impacts on consumers for test procedure amendments. The test procedure amendments are developed to provide efficiency representations for representative average use cycles. (42 U.S.C. 6293(a)(3)) As discussed in section III.A.5.d, DOE developed the test approach for outdoor units with no match on this basis. Thus, the energy conservation standard rulemaking's consideration of consumer impacts accounts for the impacts that might be associated with specific test procedure changes.

e. Specific No-Match Criteria

DOE proposed in the August 2016 SNOPR that manufacturers must determine efficiency representations for outdoor units as outdoor units with no match if they meet any of the following criteria: Having no designated refrigerant, a warranty that specifies refrigerant properties similar to those of HCFC-22 to define refrigerant acceptability (rather than or in addition to specific refrigerants), shipping without refrigerant or with a charge that requires addition of more than a pound of charge during setup, and shipping with any amount of R-407C. 81 FR at 58170–58172 (Aug. 24, 2016).

JCI and Goodman commented that there are other refrigerants, including MO-99 and NU-22, that are used as replacements for HCFC-22. JCI questioned why those refrigerants were not specifically called out in the proposed test procedure as R-407C was, while Goodman indicated that the proposal would do nothing to address these other HCFC-22 replacement refrigerants. (JCI, Public Meeting Transcript, No. 20 at p. 140; Goodman, No. 39 at p. 3)

JCI also stated that they have competitors that have published guidelines around the application of R-410A units into existing indoor applications, and questioned why those units would not have to be held to the same test approach for outdoor units with no match.

In response, it has always been the case that some outdoor units are installed as replacements for failed outdoor units. However, in most cases an outdoor unit model would also be sold in substantial numbers as a combination with indoor units. This is in contrast to R-407C units, which are predominantly sold in scenarios in which the outdoor unit is replaced, and the indoor unit is not replaced. Hence the test procedure is representative of an average use cycle for R-410A units without requiring that it be tested as a unit with no match.

JCI also commented that the benefits of R-407C will increase over time if products designed for this refrigerant based on ''additional valid matches'' are allowed to be sold, but that the proposed requirements would significantly limit any such possibility. JCI asserted that it can create a larger market for complete R-407C systems and that DOE should not limit the potential for such innovation. (JCI, No. 24 at p. 7)

ACEEE, NRDC, and ASAP and Lennox supported the proposed requirement that an outdoor unit distributed without a designated refrigerant must be tested and certified as an outdoor unit with no match. (ACEEE, NRDC, and ASAP, No. 33 at p. 4; Lennox, No. 25 at p. 5)

AHRI and Nortek commented that DOE's categorization of dry-ship units is overly-broad and does not necessarily equate to outdoor units with no match. AHRI and Nortek commented that units with long line sets require more than one pound of charge to be added in the field. AHRI and Nortek contended that it is also very realistic that manufacturers will not be able to ship units with mildly flammable refrigerants factory charged which will require adding refrigerants in the field during installation. (AHRI, No. 27 at p. 6; Nortek, No. 22 at p. 6) JCI, Ingersoll Rand, Goodman, Carrier/UTC also disagreed with DOE's proposal for similar reasons. Ingersoll Rand, Goodman, and Carrier/UTC gave examples of situations in which the entire charge required for a system could not be contained within the outdoor unit by itself as shipped from the factory, and would require more than a pound of refrigerant to be added, including for MicroChannel Heat Exchangers and long line sets. (JCI, No. 24 at p. 7–8; Ingersoll Rand, No. 38 at p. 2; Goodman, No. 39 at p. 3–4; Carrier/UTC, No. 36 at p. 3; JCI and Ingersoll Rand, Public Meeting Transcript, No. 20 at pp. 140–141) Goodman further commented that the regulatory text should restrict the one pound rule to laboratory tests and suggested regulatory text to address this issue as well as the small diameter tubing issue. (Goodman, No. 39 at p. 3–4) Lennox supported the intent of DOE's proposal but found it to be too restrictive because of the existence of products in which the internal volume of the product does not allow it to be fully charged from the factory. (Lennox, No. 25 at p. 5) Goodman, Lennox, and JCI were particularly concerned with potential unintended consequences and potentially impeding innovation as the industry moves toward lower global warming potential (GWP) refrigerants, in which cases the manufacturer may choose to ship split-system units designed for use with A2L refrigerants without the refrigerant factory-installed. (Goodman, No. 39 at p. 4) Lennox commented that the safety requirements and codes and standards required for a transition to A2L [8] refrigerants are not developed and that there is a high probability that some form of mitigation to ensure product safety will be required, for example, requiring that such units be dry-shipped, *i.e.* with a dry nitrogen charge rather than with refrigerant. Lennox commented that DOE should maintain a path that allows dry-shipping products (DOE understands this to mean not requiring no-match testing for these products) to ensure the most efficient transition to low-GWP products with the least

[8] A2L is a safety classification for refrigerants that have low toxicity and lower flammability. See *https://www.epa.gov/snap/refrigerant-safety.* Most refrigerants in current use (*e.g.* R-410A) have an A1 classification, indicating both low toxicity and no flame propagation.

negative consumer impacts. (Lennox, No. 25 at p. 5)

First Co. objected to the requirement to test an outdoor unit as a no-match outdoor unit if more than a pound of refrigerant would have to be added during set up. First Co. commented that the proposals are based on a single charge value when there are multiple charge values for different coils. First Co. requested DOE drop this requirement entirely. (EERE–2016–BT–TP–0029, No. 21 at p. 5)

In response to these comments DOE has revised the criteria for outdoor units with no match. Specifically, manufacturers must determine efficiency representations, and certify such representations, for outdoor units as an outdoor unit with no match if:

- The outdoor unit is approved for use with, determined by listing on the outdoor unit nameplate, HCFC-22 or refrigerants with similar thermophysical properties, as specified in § 429.16(a)(3) (the discussion below addresses similarity);
- There are no designations of approved refrigerants on the outdoor unit nameplate; or.
- The outdoor unit is shipped requiring more than two pounds of charge when tested according to the test procedure (*e.g.,* with 25 feet of interconnecting lines), unless (a) an A2L refrigerant is listed as approved on the nameplate, or (b) the factory charge listed on the nameplate is 70 percent or more of the outdoor unit's internal refrigerant circuit volume times the density for 95 °F refrigerant liquid.

DOE agrees with JCI and Goodman that outdoor units approved for use with refrigerants similar to HCFC-22 (other than R-407C) are likely to be intended for no-match use in the field. Hence, DOE is changing the criteria so that approval for use of any such refrigerant similar to HCFC-22 would make the outdoor unit subject to the no-match requirements. DOE does not find it likely that a large market for complete systems based on R-407C or other refrigerants similar to HCFC-22 would likely emerge in the near future given the initial trends associated with introduction of R-407C products, as discussed section III.A.5.c. As suggested by ACEEE, NRDC, and ASAP (ACEEE, NRDC, and ASAP, No. 33 at p. 3), R-410A is nearly universally used as the refrigerant that has replaced HCFC-22 in CAC/HP systems. Other refrigerants approved by the EPA in its SNAP listing for acceptable substitutes in residential and light commercial air conditioning and heat pumps [9] are rarely used in new split systems. DOE considered the approved refrigerants in the SNAP list and refrigerants understood to be suitable for use in HCFC-22 systems (''Refrigerants for R-22 Retrofits'', No. 46 at p. 1) and developed an HCFC-22 similarity criterion that would apply for these likely replacement options. DOE determined that the HCFC-22 replacement refrigerants would be selected and no other refrigerant that is likely to be approved for use in new split systems would be selected if the saturation pressure associated with 95 °F refrigerant temperature is within 18 percent of the pressure for HCFC-22. Hence, DOE adopts this as a criterion for no-match status of an outdoor unit. DOE recognizes that there may be A2L refrigerants that would themselves have similar pressures that in future may be approved on EPA's SNAP list for these products. To ensure that transition from global warming refrigerants is not restricted, DOE acknowledges that some revisions to these requirements may need to be developed as manufactures start to adopt such refrigerants in new split systems. DOE will consider such testing and certification revisions and propose options in a future rulemaking.

DOE is also revising the no-match criteria regarding dry shipping and required refrigerant addition as indicated above in response to manufacturer comments and additional research. First, DOE recognizes that where an installation requires long line sets, that a higher quantity of refrigerant may have to be added. DOE agrees with Goodman's suggestion to base this limit on a standardized scenario, specifically the addition of charge in a DOE test, for which 25 feet of refrigerant lines are specified. Second, DOE has adopted the exception associated with small-volume outdoor coils (factory charge 70 percent or more than the coil internal volume times refrigerant density) suggested by Goodman. However, DOE reviewed its own available test data for CAC/HP systems and determined that, for tests in which the added charge quantities were clearly recorded, a large percentage of tests required addition of 1 pound or more of refrigerant. Review of the data showed that nearly all of the tests could be conducted with the addition of less than 2 pounds of refrigerant. Hence, DOE is revising the charge addition requirement accordingly. First Company's comments addressed differences in indoor coil volumes, but did not provide specific information regarding the potential differences in charge that could be associated with different coil sizes—the additional pound doubles the allowed charge addition for a unit before requiring a no-match test and, based on DOE test experience, is sufficient to address nearly all tested systems. Because these systems were charged without consideration of this new requirement and would likely have required less charge addition if pre-charged with the limit in mind, and also considering that at least one manufacturer (Goodman) agreed with the one-pound limit on the basis of additional clarifications that DOE has adopted (the low-coil-volume exclusion and clarification that the limit applies for ratings testing), DOE believes that the finalized criteria are sufficiently flexible to avoid requiring no-match testing for any outdoor units that should not be tested this way.

DOE also acknowledges the issues associated with A2L refrigerants and small-volume heat exchanger technologies. DOE agrees with Goodman's suggestions for providing exceptions to the no-match requirements in these cases and has adopted the suggestions in this final rule.

f. NGIFS

In the July 2016 final rule, DOE set requirements for the indoor units that are used in tests of outdoor units with no match. 81 FR at 37065 (June 8, 2016). The August SNOPR proposed extension of this requirement to additional types of outdoor units with no match. 81 FR at 58170 (Aug. 24, 2016).

AHRI and Nortek commented that it will not always be the case that outdoor units with no match are a result of the phase-out of R-22 refrigerant and that in the future there will be a transition between nonflammable and mildly flammable refrigerants. They further suggested that when higher GWP refrigerants, such as R-410A are phased out, there will likely be a period of time when R-410A condensing units will be sold as outdoor units with no match, and that they will likely be shipped dry. AHRI and Nortek commented that while a NGIFS no higher than 1.0 sq.in./Btu/hr may be representative of R-22 units circa 2006, NGIFS of 1.0 makes no sense for R-410A, resulting in energy measurements that are not representative of the unit in the field. (AHRI, No. 27 at p. 5–6; Nortek, No. 22 at p. 5) Ingersoll Rand commented similarly. (Ingersoll Rand, No. 38 at p. 2) Ingersoll Rand further commented that the NGIFS definition is only appropriate for ⅜″ tube coils and cannot be used for coils with smaller

[9] *https://www.epa.gov/snap/acceptable-substitutes-residential-and-light-commercial-air-conditioning-and-heat-pumps.*

diameter tubes or with microchannel heat exchangers. Ingersoll Rand commented that NGIFS does not account for fin design or tube pattern which affects heat transfer, and its adoption will create the potential for testing loopholes in the future. Ingersoll Rand commented that it would be better to set a limit on coil cabinet volume based on coils sold in the 5 years prior to the elimination of a refrigerant. (Ingersoll Rand, No. 38 at p. 2)

DOE acknowledges that the old indoor units that are matched with no-match outdoor units in field installations will not always be old HCFC-22 indoor units. DOE will consider adjustments to the no-match requirements consistent with available information in a future rulemaking. However, DOE does not necessarily agree that a phaseout of high GWP refrigerants will by itself mean a step change of the existing population of indoor units to characteristics typical of more recent R-410A systems. Consideration will have to be given to whether the NGIFS value is allowed to rise to reflect representative field conditions or whether there are alternative approaches that would be more effective in addressing issues associated with installation of no-match outdoor units.

In response to Ingersoll Rand's comment regarding applicability of NGIFS, DOE responds that the vast majority of indoor units that are field-matched with no-match outdoor units have 3⁄8-in OD tubing. Further, DOE selected the NGIFS value based on the assumption that manufacturers would use enhanced fin surfaces (*e.g.,* lanced, louvered, wavy) for such tests. DOE also notes that such surfaces were in general use during the time period before phaseout of HCFC–22 for new systems. (See, *e.g.,* page 1–11 of the 1997 technical support document for room air conditioners, which indicates that such surfaces were in use for central air conditioners at the time, *https://www1.eere.energy.gov/buildings/appliance_standards/pdfs/tsdracv2.pdf.*)

6. Representation Limitations for Independent Coil Manufacturers

In the June 2016 final rule, DOE adopted language in 10 CFR 429.16 specifying that a basic model may only be certified as compliant with a regional standard if all individual combinations within that basic model meet the regional standard for which that basic model would be certified and that an ICM cannot certify a basic model containing a representative value that is more efficient than any combination certified by an OUM containing the same outdoor unit. 81 FR at 37050 (June 8, 2016).

Based on letters submitted by several stakeholders (Docket No. EERE–2016–BT–TP–0029–0006, –0005, and –0003), in the August 2016 SNOPR, DOE proposed to remove the sentence: ''An ICM cannot certify a basic model containing a representative value that is more efficient than any combination certified by an OUM containing the same outdoor unit.'' and replace it with the following language in 10 CFR 429.16(a)(4)(i): An ICM cannot certify an individual combination with a rating that is compliant with a regional standard if the individual combination includes a model of outdoor unit that the OUM has certified with a rating that is not compliant with a regional standard. Conversely, an ICM cannot certify an individual combination with a rating that is not compliant with a regional standard if the individual combination includes a model of outdoor unit that an OUM has certified with a rating that is compliant with a regional standard. 81 FR at 58172 (Aug. 24, 2016)

AHRI, Nortek, Unico, First Co., ADP, ACEEE, NRDC, and ASAP, Ingersoll Rand, Rheem, Carrier, Lennox, and JCI supported DOE's proposal. (AHRI, No. 27 at p. 7; Nortek, No. 22 at p. 7; Unico, Inc., No. 30 at p. 2; First Co, No. 21 at p. 3; ADP, No. 23 at p. 3; ACEEE, NRDC, and ASAP, No. 33 at p. 5; Ingersoll Rand, No. 38 at p. 3; Rheem, No. 37 at p. 2; Carrier/UTC, No. 36 at p. 4; Lennox, No. 25 at p. 11; JCI, No. 24 at p. 9; ADP, Public Meeting Transcript, No. 20 at p. 143) Therefore, in this final rule, DOE is adopting this language as proposed.

7. Reporting of Low-Capacity Lockout for Air Conditioners and Heat Pumps With Two-Capacity Compressors

In the August 2016 SNOPR, DOE proposed to require that the lock-out temperatures for both cooling and heating modes for CAC/HPs with two-capacity compressors be provided in the certification report. 81 FR 58163, 58172 (Aug. 24, 2016).

NEEA commented that they strongly support the proposed reporting requirement. (NEEA, No. 35 at p. 2) AHRI, Nortek, Ingersoll Rand, JCI, and Carrier/UTC commented that low-capacity lockout for air conditioners and heat pumps with two-capacity compressors is considered intellectual property, and that they are concerned about the possibility of reverse engineering products if this information is publicly reported. (AHRI, Public Meeting Transcript, No. 20 at p. 101; AHRI, No. 27 at p. 7; Nortek, No. 22 at p. 8; Ingersoll Rand, No. 38 at p. 3; JCI, No. 24 at p. 17–18; Carrier/UTC, No. 36 at p. 3)

In the existing requirements and the requirements proposed in the August 2016 SNOPR, DOE lists product-specific items that needs to be included in certification reports in 10 CFR 429.16(e), with subsection (2) listing public items, and subsection (4) listing additional items that would not be posted to DOE's public certification database. DOE notes that it included the proposal to require reporting the outdoor temperature(s) at which the unit locks out low capacity operation (where applicable) in proposed § 429.16(e)(4) of the August 2016 SNOPR. Because, under the proposal, the item would not be posted to DOE's public certification database, DOE is maintaining this requirement in this final rule.

8. Represented Values of Cooling Capacity

In the August 2016 SNOPR, DOE proposed to revise the regulatory text in three locations (10 CFR 429.16(b)(3), 10 CFR 429.16(d), 10 CFR 429.70(e)(5)(iv)) to allow a one-sided tolerance on cooling and heating capacity that allows underrating of any amount, but only overrating up to 5 percent (*i.e.,* the certified capacity must be no greater than 105 percent of the mean measured capacity or the output of the AEDM), as intended in the June 2016 final rule. As adopted in the June 2016 final rule, DOE would still use the mean of the measured capacities in its enforcement provisions.

AHRI, Mitsubishi, Rheem, Carrier, JCI, Nortek, Ingersoll Rand, ADP, Lennox, and Goodman opposed DOE's proposal for tolerance on cooling capacity. They commented that the same rules that apply to efficiency should be applied to capacity, where manufacturers should be permitted to rate cooling and heating capacity only as high as the tested value or AEDM output. (AHRI, No. 27 at p. 7; Mitsubishi, No. 29 at p. 2; Rheem, No. 37 at p. 2; Carrier/UTC, No. 36 at p. 4; JCI, No. 24 at p. 9; Nortek, No. 22 at p. 8; Ingersoll Rand, No. 38 at p. 3; ADP, No. 23 at p. 3–4; Lennox, No. 25 at p. 6; Goodman, No. 39 at p. 12; Carrier/UTC and Lennox, Public Meeting Transcript, No. 20 at p. 145) Additionally, Carrier commented that de-rating capacity would result in a consumer getting more capacity than expected but that overrating capacity as suggested in this proposal would result in a loss to the consumer. In addition, the double sided tolerance would statistically result in much higher risk for manufacturers. (Carrier/UTC, No. 36

at p. 4; Carrier/UTC, Public Meeting Transcript, No. 20 at p. 144)

ACEEE, NRDC, ASAP supported the use of one-sided tolerance tests where possible, stating that there may be legitimate business reasons to label and sell units that are more efficient than their certified values and that consumers can only be pleased if a product does better than claimed. (ACEEE, NRDC, and ASAP, No. 33 at p. 5)

Unico commented that they strongly support one-sided tolerance for capacity, without which a manufacturer cannot rate conservatively. Unico stated that it recognizes that, for some product classes other than small-duct high-velocity, there is a very small chance that a manufacturer could conservatively rate a system with the express intent to avoid testing with a slightly higher external static pressure. Unico believes the advantage that this provides is insignificant. (Unico, Inc., No. 30 at p. 2)

NEEA commented that they do not necessarily support the proposal, stating that they were not able to ascertain if DOE's one-sided tolerance for capacity reporting would result in a system being rated with a lower building load as a result of reporting an overly conservative value, and thus an overrated cooling and/or heating performance. (NEEA, No. 35 at p. 2)

First Co. agreed with DOE's proposal to allow one sided tolerance on represented values of cooling and heating capacity, but commented that the proposed language in § 429.70(e)(5)(iv) does not accurately reflect DOE's intention. First Co. believes that in the first sentence after the words ''. . . by more than 5 percent'' the text should read ''or tests worse than its certified cooling capacity by more than 5 percent.'' (First Co, No. 21 at p. 3)

DOE understands that overrating capacity could result in a loss to the consumer and could put the manufacturer at risk. In response to the comments received, in this final rule DOE is revising the tolerance on cooling capacity to be similar to the tolerance on efficiency, where the cooling capacity should be less than or equal to the lower of: (1) The mean of the sample and (2) the lower 90 percent confidence limit of the true mean divided by 0.95; or less than or equal to the AEDM output. DOE agrees with Unico that conservatively rating to gain some advantage is not a significant risk. In response to NEEA, DOE notes that the building loads, calculated by sections 4.1 and 4.2 of both appendix M and appendix M1 of the August 2016 SNOPR, use the tested heating and cooling capacities, not the rated capacities. Therefore, there is no concern of overrating cooling or heating performance.

In response to First Co.'s comments, DOE notes that the August 2016 SNOPR, § 429.70(e)(5)(iv), regarding AEDM verification testing, inadvertently stated that DOE would notify a manufacturer that a unit fails to meet its certified rating if the tested cooling capacity is greater than 105 percent of its certified cooling capacity. In this final rule, the section has been revised to indicate DOE will notify a manufacturer that a unit fails to meet its certified rating if the tested cooling capacity is lower than its certified cooling capacity. This is consistent with DOE's revisions to its tolerance on cooling capacity.

9. New Efficiency Metrics

During the August 2016 Public Meeting, EEI, PG&E, Goodman, Rheem, and Unico recommended renaming the efficiency metrics whose values will be altered as compared to the current metrics, which includes HSPF, SEER, and EER. The purpose of this would be to help avoid confusion in the marketplace and to allow more relevant utility incentive programs. (EEI, PG&E, Goodman, Rheem, and Unico, Public Meeting Transcript, No. 20 at pp. 85–91)

Additionally, EEI submitted a written comment suggesting that a new efficiency acronym be used under the revised test procedure in order to avoid market confusion and to ensure that consumers are aware that significant changes have been made in how heat pumps are tested and rated. EEI suggested the use of several specific acronyms. (EEI, No. 34, page 6) The California IOUs similarly commented that the proposed changes to appendix M1 efficiency ratings are so substantial that they should be given new descriptors. The California IOUs stated that value changes will cause confusion in the marketplace unless they are re-labeled as ''EER2,'' ''SEER2,'' and ''HSPF2,'' or with other labels determined by DOE to be appropriate. (California IOUs, No. 32 at p. 5)

In response to the comments, in this final rule, DOE is creating new efficiency metrics to represent cooling and heating performance whose values will be altered as compared to the current metrics. The new metrics include seasonal energy efficiency ratio 2 (SEER2), which will replace seasonal energy efficiency ratio (SEER); energy efficiency ratio 2 (EER2), which will replace energy efficiency ratio (EER); and heating seasonal performance factor 2 (HSPF2), which will replace heating seasonal performance factor (HSPF). These labels are consistent with those used in the CAC/HP ECS Working Group Term Sheet. New efficiency metrics SEER2, EER2, and HSPF2 reflect the changes in the test procedure in appendix M1 that result in change in the measured efficiency values. The definitions for these metrics are identical to those for the original metrics except that they are determined in accordance with appendix M1 instead of in accordance with appendix M.

B. Amendments to Appendix M Testing To Determine Compliance With the Current Energy Conservation Standards

Under EPCA, any test procedure that DOE prescribes or amends shall be reasonably designed to produce test results which measure energy efficiency and energy use of a covered product during a representative average use cycle or period of use. (42 U.S.C. 6293(b)(3)) In the August 2016 SNOPR, DOE proposed several revisions to appendix M to subpart B of 10 CFR part 430 to improve the test representativeness and repeatability. 81 FR 58164 (Aug. 24, 2016) In addition, DOE held a public meeting at DOE headquarters in Washington, DC, on August 26, 2016 (Public Meeting Transcript, Docket No. EERE–2016–BT–TP–0029–0020). Based on the comments DOE received from the August 2016 Public Meeting and from the August 2016 SNOPR comment period, DOE is modifying its approach and adopting revisions to its procedures in Appendix M, which is independent of Appendix M1.

1. Measurement of Off Mode Power Consumption: Time Delay for Units With Self-Regulating Crankcase Heaters

In the August 2016 SNOPR, DOE proposed revisions to the off-mode test procedure imposing time delays to allow self-regulating crankcase heaters to approach equilibrium before making measurements. DOE proposed a 4-hour time delay for units without compressor sound blankets and an 8-hour time delay for units with compressor sound blankets. 81 FR at 58173 (Aug. 24, 2016)

In the SNOPR public meeting, JCI commented that adding four or eight hour time delays is a substantial testing burden and requested that DOE consider developing an approach to predict the final values without much extra test time. They reiterated this request in written comments and suggested that a time-based correlation developed by manufacturers could be built into the AEDM for the off-mode metric. (JCI, Public Meeting Transcript, No. 20 at p. 31; JCI, No. 24 at p. 10)

AHRI and Nortek commented that they generally support establishing delay time but were concerned that manufacturers would have to retest all units again within 180 days of the publication of the final rule so soon after initiating off-mode testing after the June 2016 final rule first established the off-mode test procedures. AHRI asserted that this revision represents a significant and unnecessary testing burden. AHRI suggested that DOE should either allow the off-mode rating to be based on appendix M modifications finalized in the June 2016 Final Rule (DOE assumes this is a request to clarify that products tested within 180 days of the June 8 final rule need not be retested again using the time delays) or move this revision to appendix M1 (AHRI, No. 27 at p. 8; Nortek, No. 22 at pp. 8–9). Carrier commented that the estimated time to implement this change would be at least six additional months (Carrier, No. 36 at p. 5). Rheem disagreed with the implementation time frame because this change will double the testing time and supported moving the change to appendix M1 (Rheem, No. 37 at p. 2). Ingersoll Rand commented that completing all the required testing would extend beyond the effective date (Ingersoll Rand, No. 38 at p. 3).

ACEEE, NRDC, and ASAP commented that DOE's approach to the thermal response delay issue for self-regulating crankcase heaters seems reasonable and responsive, but also sub-optimal considering that the measured self-regulating heater's power at the end of the specified delay times could be higher or lower with compressors having more or less thermal mass. ACEEE, NRDC, and ASAP recommended that DOE allow manufacturers to select alternative delay times if shorter or longer delays are required for specific models. (ACEEE, NRDC, and ASAP, No. 33 at p. 6).

Lennox, the CA IOUs and NEEA supported DOE's proposal. (Lennox, No. 25 at p. 11; CA IOU, No. 32 at p. 4; NEEA, No. 35 at p. 2)

DOE agrees that this additional delay time requirement could change the off-mode power measurement for some tested combinations that manufacturers may have already tested using the test procedure of the June 2016 Final Rule. DOE does not intend to introduce unnecessary test burden due to the close timing between the June 2016 Final Rule and this final rule. Therefore, DOE has decided to remove this requirement from appendix M and adopt it only in appendix M1. As for JCI's suggestion to develop a time-based correlation to allow prediction of the final measurement based on the trend in the measurement over a limited time period, DOE does not have sufficient test data to be confident that such an approach would provide a predictable result. In fact, depending on the equation used to fit the curve created by the first few data points, the details of the particular compressor design, and the history of testing just prior to conducting an off-mode test, DOE is concerned that a wide range of results might be obtained for any given unit, including a prediction of infinite wattage. DOE understands JCI's concern and agrees that such an approach could be considered in the future with more analysis and testing to validate an approach. Hence, DOE will not adopt a shortened test using curve fitting to predict ultimate off-mode power input. Regarding JCI's mention of an AEDM for off-mode, DOE does not regulate what analytic evaluation can be used in an AEDM—there is nothing in the AEDM requirements that would prevent a manufacturer from adopting an AEDM that uses the results of a shortened test as its input, as long as the requirements in 10 CFR 429.16 and 429.70 are satisfied. Thus, this notice does not adopt a shortened test procedure using curve fitting and prediction to determine off-cycle power input for systems with self-regulating crankcase heaters.

DOE received no comment suggesting different time delays than those proposed by DOE. Hence, DOE has adopted in appendix M1 the proposed time delays for measurement of off-mode power for units with self-regulating crankcase heaters or heater systems in which the crankcase heater control is affected by the heater's heat.

In addition, DOE notes that the August 2016 SNOPR inadvertently included in the regulatory text a certification requirement for the duration of the crankcase heater time delay for the shoulder season and heating season, if such time delay is employed. DOE does not actually require this information and has not adopted this requirement in the final rule.

2. Refrigerant Pressure Measurement Instructions for Cooling and Heating Heat Pumps

In the August 2016 SNOPR, DOE proposed limiting the internal volume of the pressure measurement system (*i.e.* the pressure gauge or transducer and the capillary tube and tube fittings connecting the transducer to the refrigerant lines) at pressure measurement locations that may switch from liquid to vapor state when changing operating modes and for all locations for systems undergoing cyclic tests for cooling/heating heat pumps. Specifically, DOE proposed the limit to be 0.25 cubic inch per 12,000 Btu/h. DOE also proposed the default internal volumes to be assigned to pressure transducers and gauges of 0.1 and 0.2 cubic inches, respectively, if transducer or gauge datasheets do not provide their internal volume. 81 FR at 58174 (Aug. 24, 2016)

During the 2016 August Public Meeting, Carrier commented that manufacturers typically test with up to six pressure transducers and the proposed limit would prohibit the level of testing during manufacturers' development stage and limit the number of pressure transducers to two. Carrier requested a reconsideration of the tolerance. (Carrier, Public Meeting Transcript, No. 20 at pp. 70–75)

AHRI requested clarification of ''locations where the refrigerant state changes from liquid to vapor for different parts of the test.'' AHRI commented that it is standard industry practice to place pressure taps with capillary tubes at six locations and advised that one of its members reported that, in their test chambers, the average internal volume of each pressure line is 0.91 cubic inches. Hence, AHRI asserts that DOE's proposed limit is too tight, such that the allowed number of pressure transducers would be zero for a unit that has a capacity less than 3 tons, and only one for larger-capacity units. In addition, AHRI commented that, for a cyclic test, the refrigerant state change occurs so quickly during transient startup that the effects (if any) will be within the tolerance of the measuring equipment. According to AHRI, for steady-state tests of units with the cooling mode restrictor located in the outdoor unit, there are at most two locations where the refrigerant state changes from liquid to two-phase between heating and cooling. AHRI's comment provided a table showing the refrigerant states at the six typical measurement locations for a cooling/heating heat pump having two expansion devices (one each in the indoor and outdoor units) for four test scenarios: Cooling steady-state, cooling transient start-up, heating steady-state, and heating transient start-up. The comment provided a similar table showing the refrigerant states for a heat pump with a single expansion device in the outdoor unit. In these tables, the transient startup scenario entries were all ''two-phase''. In addition, the only differences in refrigerant state between steady-state heating and steady-state cooling were highlighted in the single-expansion-device table for the liquid service valve and indoor coil inlet locations. AHRI commented that the refrigerant weight difference (*e.g.,* associated with transfer of refrigerant in and out of the pressure lines) is extremely small (particularly considering standard charging conditions in the field), and would have a negligible effect on the system performance. AHRI requested that DOE eliminate restrictions on pressure transducer internal volume or increase them significantly in order to ensure proper system analysis. (AHRI, No. 27 at pp. 8–11) JCI, Carrier, Ingersoll Rand and Goodman concurred with AHRI's comment. (JCI, No. 24 at p. 10–12; Carrier/UTC, No. 36 at p. 5–6; Ingersoll Rand, No. 38 at p. 3; Goodman, No. 39 at p. 9) Ingersoll Rand further requested that there be clarification that this requirement would apply only to assessment and enforcement testing, not for developmental testing. (Ingersoll Rand, No. 38 at p. 3)

Lennox commented that this proposal is not practical or in alignment with current practice for either manufacturer or audit testing, and requested DOE remove or extensively revise this requirement to align with current practices. (Lennox, No. 25 at p. 12) Rheem disagreed with DOE's proposal, and commented that the amount of refrigerant trapped in pressure measuring devices can be adequately accounted for through proper refrigerant charging instructions. (Rheem, No. 37 at p. 3) Unico agreed there should be volume limits but did not have a comment on the value. Unico commented that most systems have a high tolerance for charging while some systems, particularly systems with microchannel coils, have a very low tolerance. (Unico, No. 30 at p. 3) ACEEE, NRDC, and ASAP appreciated DOE's interest but stated that it could not judge whether the proposed volumetric limits are the right ones. (ACEEE, NRDC, and ASAP, No. 33 at p. 6) The CA IOUs agreed with DOE's proposal (CA IOU, No. 32 at p. 4)

DOE has considered all of the comments received and is making revisions based on those comments. First, DOE agrees that the transient startup phase of a cyclic test may be sufficiently short that any transfer of refrigerant in or out of the pressure lines at this time could have very little impact on measured cyclic performance. The scenario for cyclic test performance enhancement at the end of the on cycle discussed in the August 2016 SNOPR could still occur (see 81 FR at 58174 (Aug. 24, 2016)), but there is no data available to demonstrate that this effect is significant.

DOE notes that the tables provided in the AHRI comment showing refrigerant states at different refrigerant circuit locations represent states in the refrigerant lines and not in the pressure measurement systems, which could be different. For example, while the refrigerant state is always vapor at the discharge location during steady-state operation, the pressure measurement system is at a lower temperature than the saturation temperature associated with the prevailing pressure level. Hence, the vapor in the pressure line will condense. The condensed liquid may flow out of the capillary line back into the system, but this is unlikely if the pressure measurement system is lower than the measurement location. Also, it is somewhat unclear whether surface tension inside a small-diameter capillary tube would impede the flow of condensed liquid back into the system, or whether the vapor flowing into the system to replace the liquid would hold up the liquid's return flow. DOE considered the potential states within the pressure measurement systems rather than at the measurement locations when evaluating the potential for refrigerant transfer between steady-state operating modes. DOE made some reasonable assumptions for this assessment, making liberal assumptions where there is some doubt about what will occur—specifically, DOE did not assume that for the above scenario that liquid return flow to the system would be impeded. DOE's assessment of likely refrigerant states for a single-expansion-valve heat pump is summarized in Table III–1. The table adds a seventh potential refrigerant circuit location, between the outdoor coil and the expansion valve, which DOE expects that some manufacturers may monitor during developmental testing to determine subcooling achieved during cooling mode operation.

TABLE III–1—REFRIGERANT STATES IN PRESSURE MEASUREMENT SYSTEMS FOR A SINGLE-EXPANSION-VALVE HEAT PUMP

Operating mode	Steady-state cooling		Steady-state heating	
Pressure measurement system above or below tap location	Above	Below	Above	Below
1. Compressor Discharge	Vapor	Liquid **	Vapor	Liquid **.
2. Between Outdoor Coil and Expansion Valve	Liquid	Liquid	Vapor *	Two-phase.
3. Liquid Service Valve	Vapor *	Two-phase	Liquid	Liquid.
4. Indoor Coil Inlet	Vapor *	Two-phase	Liquid	Liquid.
5. Indoor Coil Outlet	Vapor	Vapor	Vapor	Liquid **.
6. Common Suction Port (*i.e.* vapor service valve)	Vapor	Vapor	Vapor	Liquid **.
7. Compressor Suction	Vapor	Vapor	Vapor	Vapor.

* Any liquid that enters the pressure measurement system will evaporate because the system is at a warmer temperature than the saturation temperature associated with the pressure.

** Liquid will condense in the pressure measurement system because the system is at a cooler temperature than the saturation temperature associated with the pressure, and will not drain back into the refrigeration circuit.

DOE notes that the liquid that might transfer out of one pressure measurement system as the operating mode switches from cooling to heating may transfer into another pressure measurement system and therefore not affect total charge operating within the refrigerant circuit. Also, because of the large density difference between liquid and vapor, DOE believes that the charge in the pressure measurement system would be negligible if the refrigerant within it is two-phase or vapor. Hence, the likely transfer of refrigerant out of the refrigeration circuit as the system switches from cooling to heating would be equal to the liquid density (calculated for 100 °F bubble point conditions) multiplied by the volume differential obtained by adding the volumes of the downward-run pressure measurement systems at locations 5 and 6 (as designated in Table III–1) to the volumes of any pressure measurement systems at locations 3 and 4 and subtracting the volume of any pressure measurement system at location 2. For a system with two expansion valves, the transferred refrigerant would represent only the volumes of downward-run pressure measurement systems at locations 5 and 6.

DOE realizes the refrigerant transfer could be mitigated by complex phenomena occurring within the pressure measurement systems, some of which, for example surface tension, are mentioned above. Another mitigating phenomenon would be the filling of the pressure measurement system with compressor oil, which would displace any refrigerant that might transfer into it. Hence, DOE is relaxing the requirement proposed in the August 2016 SNOPR in new section 2.2.g (see 81 FR at 58207 (Aug. 24, 2016)) that the volume differentials listed above represent no more than 0.5 percent of refrigerant charge. DOE is instead adopting a requirement in section 2.2.g that the volume differential represent no more than 2 percent of the charge listed on the outdoor unit nameplate. Basing the limit on the outdoor unit nameplate charge will provide more flexibility for pressure measurement systems for those heat pumps that have more charge and would hence be less sensitive to this issue. However, due to the uncertainty regarding the actual potential behavior regarding refrigerant transfer, DOE also is imposing a pressure measurement system volume limit of 1 cu. in. for location 2 for single-expansion-device heat pumps, in order to prevent a test laboratory from using a very large volume for this location to offset the volumes of locations 3, 4, 5, and 6.

For a two-expansion-device heat pump with pressure measurement systems at locations 5 and 6 above the pressure tap locations, this approach imposes no volume limits. Also, for single-expansion-valve heat pumps with pressure measurement systems at locations 5 and 6 above the pressure tap locations and the volume at locations 2 offsetting the volumes at locations 3 and 4, there will also be no volume limit, other than the 1 cu. in. limit at location 2. DOE believes that these revisions to the proposal will allow manufacturers to make pressure measurements at the locations typically used for development and ratings testing while also providing some assurance that unforeseen impacts associated with refrigerant transfer between operating modes will be mitigated. However, DOE notes that the test procedure is for determining the performance of the product for the purpose of efficiency representations, not for development testing. DOE does not require pressure measurements installed at all 7 locations indicated in Table III–1. If manufacturers require use of pressure lines for development testing that exceed the volume requirements, they have the option of using isolation valves to isolate the tap locations not needed for ratings tests as the test transitions from development to determination of ratings for purposes of certifying compliance with applicable standards. Another option is to use pressure transducers that are more resistant to the temperature changes that occur in the test chamber. In any case, DOE may consider revisions to the requirements in the future if testing shows that they can be revised further to both improve test repeatability and allow more flexibility in making pressure measurements.

3. Revised EER and COP Interpolation Method for Units Equipped With Variable-Speed Compressors

In the August 2016 SNOPR, DOE proposed to require use of bin-by-bin interpolations for all variable-speed units (including variable-speed multi-split and multi-head mini-split systems), to calculate performance when operating at an intermediate compressor speed to match the building cooling or heating load. This method consists of using interpolation of EER or COP for each temperature bin based on the estimates of capacity and power input for the specific bin temperature. (EER is equal to cooling capacity divided by power input, while COP is proportional to heating capacity divided by power input.) 81 FR at 58175 (Aug. 24, 2016)

Nortek, JCI, Mitsubishi, Carrier, Rheem, Ingersoll Rand and AHRI expressed support for DOE's proposal but stated concerns that it would impact ratings and would, as a result, be more appropriate for inclusion in appendix M1 as opposed to Appendix M. (Nortek, No. 22 at p. 9; JCI, No. 24 at p. 12; Mitsubishi, No. 29 at p. 2; Carrier, No. 36 at p. 6; Rheem, No. 37 at p. 3; Ingersoll Rand, No. 38 at p. 4; AHRI, No. 27 at p. 11) AHRI also commented that its members were in the process of collecting data on the impact this proposed change would have on ratings and committed to providing additional information to the Department within 30 days of the close of the comment period. (AHRI, No. 27 at p. 11) DOE notes that the additional data were not provided. Goodman also requested DOE implement this change as part of appendix M1. (Goodman, No. 39 at p. 6) Unico recommended that this proposal be moved to appendix M1, and if it remains as an appendix M change, DOE should allow that the higher rating of both methods be used, but only if the bin-by-bin method results in a failure. (Unico, No. 30 at p. 3–4) Lennox, CA IOU, ACEEE, NRDC, and ASAP, and NEEA all supported DOE's proposal. (Lennox, No. 25 at p. 12; CA IOU, No. 32 at p. 4; ACEEE, NRDC, and ASAP, No. 33 at p. 6; NEEA, No. 35 at p. 2)

Central air conditioning heat pumps include single-speed, two-speed, and variable-speed products, all within the same product class that when tested in accordance with the DOE test procedure will have different measured efficiencies. Pursuant to 42 U.S.C. 6293(e), DOE is required to determine to what extent, if any, the proposed test procedure would alter the measured efficiency of the covered product. DOE proposed changes to the interpolation method for variable speed units only. For single-speed and two-speed products there would be no change in measured efficiency because they would not be impacted by this change in test procedure. However, variable-speed products would be impacted by this change in test procedure, so the measured efficiency would change.

Where an amended test procedure would alter measured efficiency, EPCA requires DOE to amend an energy conservation standard by measuring, under the amended test procedure, a sample of representative products that minimally comply with the standard. In this case, minimally compliant units are those with single-speed technology. Consistent with the statute, DOE has tested a representative sample of covered products that minimally comply with the existing standard. EPCA requires that the amended standard should constitute the average of the energy efficiency of those units, determined under the amended test procedure. As a result of that testing, DOE has determined that there is no change in measured average energy efficiency for single-speed units between the current test procedure and the amended test procedure. Thus, under 42 U.S.C. 6293(e)(2), the amended standard applicable to the amended test procedure and the current standard applicable to the amended test procedure are the same. As a result, DOE does not need to amend the existing standard to require that representations of variable-speed heat pumps be based on the amended test procedure in appendix M.

If DOE were to include this change in appendix M, Goodman requested that DOE allow industry up to two years to re-test and re-calculate SEER and HSPF, by either modifying the implementation date for this provision or by issuing a policy of non-enforcement for this provision. (Goodman, No. 39 at p. 6) DOE notes that this proposal would not require additional testing. The proposed change only impacts how ratings are calculated based on the new interpolation method, not the data that is measured or how it is measured. If manufacturers have test data that is otherwise valid under the amended test procedure, there would be no reason to retest solely because of the change in the way represented values for variable speed heat pumps are calculated.

Several commenters suggested that because the change to bin-by-bin interpolation for variable speed heat pumps might cause changes in ratings, DOE should not require the new method in Appendix M. Commenters did not explain why a simple change in ratings would warrant a decision to postpone the change in method, but DOE has considered three possibilities. First, commenters may be concerned about the work to comply with the new method. However, as noted above, the new interpolation method is only a matter of calculation; it will require no new tests. DOE believes that the burden of recalculation using existing test data will be minimal; Appendix M will specify how to perform the bin-by-bin interpolation, and relatively simple revision to a spreadsheet would suffice to implement this method as a substitute for the quadratic method required under the prior test procedure. Second, commenters may be concerned about the cost of revising labels and other representation documents to reflect the new ratings. Third, some commenters may object because if the new method results in a decreased rating, that change will make the affected models appear less efficient to potential buyers.

With respect to these second and third concerns, DOE believes that the inaccuracy of the current method warrants the change. As the August 2016 SNOPR explained, the quadratic interpolation method can produce inaccurate results. For HSPF the quadratic method can produce a value up to 7.9% different from what the bin-by-bin method produces (and DOE regards the latter as more accurate). Thus, for some equipment the rated HSPF is overstated, with respect to a fair measure of efficiency, by as much as 7.9%. A buyer using such equipment would consume 7.9% more energy, at 7.9% more cost, than expected based on the rating. DOE believes that amount is a significant difference. By contrast, the regulation requires a represented cooling capacity to be within 5% of the average measured cooling capacities, and it permits rounding of figures to approximately 1% precision (200 Btu/h for a 20,000 Btu/h system). Using 1% and 5% as indicators of what amount of error in a rating is significant, DOE believes it is important to correct an interpolation method that generates, for some models, larger errors. Of course, if a rating based on the old method is still valid—including by being within the regulation's tolerances with respect to recalculated values—a manufacturer could choose whether or not to revise the rating.

For these reasons, DOE is adopting this proposal both in appendix M and appendix M1 in this final rule.

4. Outdoor Air Enthalpy Method Test Requirements

In the August 2016 SNOPR, DOE proposed modifications to requirements when using the outdoor air enthalpy method as the secondary test method, including that the official test be conducted without the outdoor air-side test apparatus connected. 81 FR at 58175–58176 (Aug. 24, 2016)

During the August 26, 2016 public meeting, Carrier suggested that the proposal to require a heat balance only for the full-load cooling test and, for a heat pump, the full-load heating test be extended to other secondary capacity measurement methods, including to use of the refrigerant enthalpy method. Carrier contended that it can be difficult to get an energy balance for some operating conditions, particularly for variable-speed systems, when there is insufficient subcooling or superheat.[10] (Carrier, Public Meeting Transcript, No. 20 at pp. 38–39) Ingersoll Rand agreed with this suggestion; Goodman also agreed and indicated that the issue applies for tests of single-stage, two-stage, and variable-speed systems for the heating mode test conducted in 17 °F outdoor temperature. (Ingersoll Rand, Public Meeting Transcript, No. 20 at p. 39; Goodman, Public Meeting Transcript, No. 20 at p. 40)

JCI, Lennox, Carrier, Ingersoll Rand, Goodman and AHRI agreed with DOE on this proposal but recommended that the ducted test be a 30-minute test. (JCI, No. 24 at p. 12; Lennox, No. 25 at p. 12; Carrier, No. 36 at p. 7; Ingersoll Rand, No. 38 at p. 4; Goodman, No. 39 at p. 13; AHRI, No. 27 at p. 11–12) Carrier, Ingersoll Rand, Goodman and AHRI also suggested DOE similarly only require balance checks for the A_2 and $H1_2$ (or $H1_N$) tests for the refrigerant enthalpy method. (Carrier, No. 36 at p. 7; Ingersoll Rand, No. 38 at p. 4; Goodman, No. 39 at p. 13; AHRI, No. 27 at p. 11–12) In addition, AHRI and Ingersoll Rand suggested DOE eliminate the five consecutive readings for verifying the primary capacity measurements. (AHRI, No. 27 at p. 11–12; Ingersoll Rand, No. 38 at p. 4) CA IOU and Rheem agreed with DOE's proposal. (CA IOU, No. 32 at p. 4; Rheem, No. 37 at p. 3)

DOE agrees that validation of proper capacity measurement for cooling and heating modes for full-load operation is sufficient to show that the indoor air enthalpy method is being applied properly and gives an accurate measurement. Hence, use of the secondary method and achieving an energy balance for all load levels in each operating mode is not necessary. DOE notes that systems with capacity greater than 135,000 Btu/h are tested without any requirement for a secondary capacity check. (American Society of Heating Refrigeration, and Air-Conditioning Engineers (''ASHRAE'') Standard 37–2009 (''ASHRAE 37–2009''), which is incorporated by reference into the DOE test procedures for both residential and commercial air conditioners, indicates in Table 1 that a single method is used for systems with a cooling capacity greater than 135,000 Btu/h.) Further, DOE believes this modification will help to reduce test burden. The situation discussed in the public meeting and written comments, in which, when using the refrigerant enthalpy method as the secondary test method, a heat balance cannot be calculated for some conditions due to subcooling or superheat being too low, would technically make completion of a valid test impossible, according to the current test procedure, without resorting to an alternative secondary method. DOE recognizes that use of different secondary methods for different parts of the test would significantly increase test burden. Hence, DOE is modifying the test procedure to require use of a secondary capacity measurement that agrees with the primary capacity measurement to within 6 percent only for the cooling full load test and, for heat pumps, for the heating full load test.

DOE has decided to change the names for ''ducted'' and ''non-ducted'' outdoor air enthalpy methods to avoid confusion with certain product types. Specifically, DOE is adopting the new name ''free outdoor air test'' for non-ducted outdoor air enthalpy test, and ''ducted outdoor air test'' for ducted outdoor air enthalpy test. In this final rule, DOE is also

[10] In this context, subcooling refers to the difference between the saturated temperature associated with the pressure of the refrigerant liquid exiting the outdoor unit (in cooling mode) and the temperature of the liquid. Similarly, superheat refers to the difference between the temperature of the refrigerant exiting the indoor unit (in cooling mode) and the saturated temperature associated with the pressure of this refrigerant. The enthalpy of the refrigerant at these locations generally cannot be determined if these values are zero.

adopting a 30-minute ducted outdoor air test with measurements at five-minute intervals, and eliminating from section 3.11.1.2 the requirement of five consecutive readings for verifying primary capacity measurements.

DOE's proposed changes to outdoor air enthalpy method requirements in the August 2016 SNOPR included revision to section 3.11.1.2 that removed the reference to section 8.6.2 of ASHRAE 37–2009. 81 FR at 58209 (Aug. 24, 2016). However, the key points of section 8.6.2 still apply for the revised approach for the outdoor air enthalpy method. The finalized test procedure retains the reference to this section.

5. Certification of Fan Delay for Coil-Only Units

In the August 2016 SNOPR DOE proposed to amend its certification report requirements to require coil-only ratings to specify whether a time delay is included, and if so, the duration of the delay used. DOE proposed to use the certified time delay for any testing to verify performance. 81 FR at 58176 (Aug. 24, 2016)

Nortek, Ingersoll Rand, Carrier, JCI, Rheem, Goodman and AHRI suggested that the certification of the indoor fan off delay should not be public information. (Nortek, No. 22 at p. 2; Ingersoll Rand, No. 38 at p. 3; Carrier, No. 36 at p. 7; JCI, No. 24 at p. 13; Rheem, No. 37 at p. 3; Goodman, No. 39 at p. 12; AHRI, No. 27 at p. 12) ADP agreed that the duration of the indoor fan time delay needs to be specified but should be a part of the public product-specific information. ADP commented that making this information public improves the accuracy of ICM AEDM ratings. (ADP, No. 23 at p. 4) Lennox and ACEEE, NRDC, and ASAP supported DOE's proposal. (Lennox, No. 25 at p. 12; ACEEE, NRDC, and ASAP, No. 33 at p. 6)

DOE understands that manufacturers want to keep fan delay setting information private. Given that DOE proposed to require this information in the section of additional product-specific information that would not be posted to DOE's public certification database, DOE has decided to adopt this proposal in this final rule. In response to ADP, DOE will address concerns regarding reporting for ICMs through a separate process.

6. Normalized Gross Indoor Fin Surface Area Requirements for Split Systems

To help ensure that the test procedure results in ratings that are representative of average use, in the August 2016 SNOPR DOE, proposed to include a provision that would prevent testing certain combinations that are not representative of single-split systems with coil-only indoor units that are commonly distributed in commerce. Specifically, DOE proposed to limit the normalized gross indoor fin surface (NGIFS) for the indoor unit used for single-split-system coil-only tests to no greater than 2.0 square inches per British thermal unit per hour (sq.in./Btu/hr). NGIFS is equal to total fin surface multiplied by the number of fins and divided by system capacity. 81 FR at 58177 (Aug. 24, 2016)

In the August 2016 Public Meeting, Ingersoll Rand commented that it did a rough calculation for a micro channel heat exchanger and determined the NGIFS to be 0.81. Ingersoll Rand commented that this indicates that there are problems with looking at today's technology and coming up with a value for NGIFS. Ingersoll Rand further commented that in coming up with a value for NGIFS, it needs to be ensured that doing so does not create issues or loopholes. (Ingersoll Rand, Public Meeting Transcript, No. 20 at p. 45) Rheem commented that there needs to be further study on the 2.0 value of NGIFS before making a decision in order to not limit future efficiencies. (Rheem, Public Meeting Transcript, No. 20 at p. 46) Carrier/UTC similarly commented that there may be unforeseen consequences of limiting design options that manufacturers will have to comply with the efficiency standards. (Carrier/UTC, Public Meeting Transcript, No. 20 at pp. 47–48) Rheem also commented that due to the complexity of the issue, the NGIFS criteria should go in appendix M1, not in appendix M. (Rheem, Public Meeting Transcript, No. 20 at p. 46) Johnson Controls commented that units that are above 2.0 today would need to be retested, and the ratings for these units would most likely change. JCI commented that for this reason, they believe that the proposal for NGIFS belongs in appendix M1, not in appendix M. (JCI, Public Meeting Transcript, No. 20 at pp. 50–51) Allied commented that the values that DOE is proposing are reasonable, but that there are further considerations associated with the different technologies that apply. Allied also commented that, based on their review, future standard levels could be even more stringent and still allow some latitude in design approaches. (Allied, Public Meeting Transcript, No. 20 at pp. 49–50) JCI also commented that usually normalized values do not have dimensions and questioned whether the proposal takes into account fin and tube spacing. (JCI, Public Meeting Transcript, No. 20 at pp. 56–59)

Nortek and AHRI opposed DOE's proposal and commented that DOE does not have the authority to regulate the design of residential central air-conditioners and heat pumps, so all NGIFS restrictions should be removed from both appendix M and M1. AHRI commented that AHRI would like to aid the Department to address this ''golden blower'' issue in a way which does not put restrictions on design and is both refrigerant and technology neutral. AHRI proposed to develop a solution within 30 days of the close of the August 2016 SNOPR comment period, but they did not provide additional input. (Nortek, No. 22 at p. 10; AHRI, No. 27 at p. 12)

JCI commented that while DOE stated in the SNOPR that the 2.0 limit of NGIFS does not affect 95% of tested combinations, this also showed there are current systems that will not be compliant. JCI expressed concern that if such changes are made to appendix M, standards adjustments would be required. JCI recommended that DOE limit NGIFS in M1 only and the DOE recommended value of 2.5 appears to be a valid target. (JCI, No. 24 at p. 13)

Lennox commented that while it is reasonable to use 3/8″ round tube, plate fin coil in the NGIFS definition for outdoor units with no match, DOE must revise the definition for other split system products because there are other tube diameters and technologies used across the industry. Lennox recommended that DOE expand the definition to include all tube types and fin surfaces. Lennox supported DOE's proposal on the NGIFS calculation and proposed limit. (Lennox, No. 25 at p. 6–8) Carrier opposed DOE's proposal to limit NGIFS for the indoor unit and preferred DOE not restrict design options as that could impact consumer choices when different refrigerants are used in the future or lessen a manufacturer's ability to optimize for hot dry climates. Additionally, Carrier commented that this proposal does not address microchannel coils or any other coil tube diameter besides 3/8″. (Carrier, No. 36 at p. 7)

Rheem objects to the limitation of a fixed value for NGIFS and proposed that indoor coil area should be determined by balancing with the outside coil area. (Rheem, No. 37 at p. 3–4) Ingersoll Rand opposed the proposed NGIFS limit because it is only appropriate for 3/8″ tube coils. Ingersoll Rand commented that it would be better to set a limit on coil cabinet volume based on coils sold in the 5 years prior to the elimination of a refrigerant. (Ingersoll Rand, No. 38

at p. 4) Goodman also expressed concern that this requirement on the tested combination may inhibit future designs and did not support the proposed restrictions. Goodman suggested that some requirements in cabinet width might be appropriate and that DOE and AHRI should work together to develop a reasonable restriction. (Goodman, No. 39 at p. 7–8)

ACEEE, NRDC, and ASAP supported DOE's proposal and also suggested DOE should consider the input of manufacturers who may have a few models designed for hot, dry climates where the apparent evaporator surface oversizing can improve rated performance. (ACEEE, NRDC, and ASAP, No. 33 at p. 6) CA IOU and NEEA agreed with DOE's proposal. (CA IOU, No. 32 at p. 4; NEEA, No. 35 at p. 3)

In response to JCI, valid normalized values may have units. For example, energy efficiency ratio is a normalized value representing capacity per electric power input with units of British thermal units (Btu) per Watt-hour (Btu/W-h). Additionally, the NGIFS does take into consideration the fin spacing—the number of fins, N_f, is a parameter in the equation to determine NGIFS. As an example, consider two indoor coils with the same finned length—the coil with the higher fin density will have more fins and thus a higher NGIFS. It is true, however, that NGIFS does not include the impact of tube spacing.

Addressing the Ingersoll Rand and Allied comments, DOE acknowledges that NGIFS does not provide as good a representation of the heat transfer performance of microchannel indoor coils as that of conventional tube-fin indoor coils, and the development of an appropriate equivalent value for this newer technology will be important in order to prevent loopholes in the requirement. However, DOE is not aware of any significant current market share of systems using microchannel indoor coils, and so good information to use as the basis for development of NGIFS limits for this technology is not yet available. Further, the likely lower value of NGIFS for microchannel coils will mean that imposing a limit based on conventional coil technology would not limit use of microchannel coils before a better approach is developed. DOE has not developed an appropriate approach at the moment, but could consider adopting an NGIFS approach for microchannel indoor coils in a future rulemaking.

Because DOE's NGIFS analysis for coil-only systems does not consider tube diameters other than ⅜ inches and fin types other than plate fins, as well as the units currently on the market that would not meet the 2.0 NGIFS limit (*e.g.* as indicated by the JCI comment), the proposed approach does not resolve DOE's concern while maintaining a reasonable test procedure for units with different designs. Accordingly, DOE is not adopting the NGIFS requirement in this final rule for either appendix M or appendix M1. DOE will consider how best to address this issue in the future.

7. Modification to the Test Procedure for Variable-Speed Heat Pumps

The August 2016 SNOPR proposed changes to the test procedure of appendix M for variable-speed heat pumps to allow more flexibility in the design and testing of these products. 81 FR at 58177–79 (Aug. 24, 2016). The June 2016 final rule imposed restrictions on the compressor speeds that could be used in testing, indicating that full speed must be the same speed for all heating mode operating conditions. DOE adopted this approach based on the observation that extrapolation of performance outside of the range of conditions used for testing can lead to unreasonable results if the speeds are allowed to be different for the different test conditions. 81 FR at 37029 (June 8, 2016). However, the final rule discussed stakeholder comments regarding heat pumps that improve heating mode performance by using different compressor speeds at lower ambient temperatures, and indicated that consideration would be given in the future to test procedure revisions that would better address their operation. *Id.* In the August SNOPR, DOE proposed a test procedure revision that would allow testing of heat pumps whose compressors operate at higher speeds in lower ambient temperatures. 81 FR at 58177–58179 (Aug. 24, 2016). Specifically, DOE proposed the following amendments for appendix M.

- A 47 °F full-speed test used to represent the heating capacity would be required and designated as $H1_N$. However, the 47 °F full-speed test would not have to be conducted using the same compressor speed (determined based on revolutions per minute (RPM) or power input frequency) as the full-speed tests conducted at 17 °F and 35 °F ambient temperatures, nor at the same compressor speeds used for the full-speed cooling test conducted at 95 °F. For appendix M, the compressor speed for the 47 °F full-speed test would be at the manufacturer's discretion, except that it would have to be no lower than the speed used in the 95 °F full-speed cooling test. Prior to the June 2016 final rule amendments, the heating capacity was represented either by the $H1_2$ test (for which the compressor speed guidance was not explicit), or, if a manufacturer chose to conduct what was then the optional $H1_N$ test, this latter test (using the same compressor speed as the full-speed cooling mode test) represented the heating capacity. Under the proposal in the August SNOPR, heating capacity would be represented only by the $H1_N$ test, which would be mandatory, while the compressor speed would be at the manufacturer's discretion within a range from the speed used for the 95 °F full-speed cooling test to the speed used for the full-speed 17 °F test.
- The full-speed tests conducted at 17 °F and 35 °F ambient temperatures would still have to use the same speed, which would be the maximum speed at which the system controls would operate the compressor in normal operation in a 17 °F ambient temperature, although the 35 °F full-speed test would remain optional.
- It would be optional to conduct a second full-speed test at 47 °F ambient temperature at the same compressor speed as used for the 17 °F test, if this speed is higher than the speed used for the $H1_N$ test described in this preamble. This test would be designated the $H1_2$ test. Because DOE does not expect that an $H1_N$ test would ever use a higher compressor speed than used for the full-speed 17 °F test, the proposed test procedure would not provide for this situation.
- If no 47 °F full-speed test were conducted at the same speed as used for the 17 °F full-speed test, standardized slope factors for capacity and power input would be used to estimate the performance of the heat pump for the 47 °F full-speed test point for the purpose of calculating HSPF.
- The capacity measured for the $H1_N$ test would be used in the calculation to determine the design heating requirement.

In addition, DOE proposed that the $H1_N$ test, at 47 °F ambient temperature, be conducted to represent nominal heat pump heating capacity, but that there would be no specific compressor speed requirement associated with it for appendix M, except that it be no lower than the speed used for the 95 °F full-speed cooling test. Under the proposal, if the $H1_N$ test did not use the same speed as is used for the 17 °F full-speed heating test, it would affect the HSPF calculation only through its influence on the design heating requirement, since the standardized slope factors would be used to represent full-speed heat pump performance. 81 FR at 58179 (Aug. 24, 2016)

A number of manufacturers and AHRI recommended the proposed changes should be part of appendix M1 rather than appendix M. (Rheem, Public Meeting Transcript, No. 20 at pp. 54–55; Rheem, No. 37 at p. 4; Carrier, No. 36 at p. 2; Nortek, No. 22 at p. 11; AHRI, No. 27 at p. 13; Mitsubishi, No. 29 at p. 2–3) Carrier commented at the public meeting that the proposals may be good, but that there had not been sufficient time to thoroughly review them, adding that a key concern is avoiding any potential need to retest products. (Carrier/UTC, Public Meeting Transcript, No. 20 at pp. 55). Unico recommended moving the slope factor change and the proposal for compressor speed at 47 °F test to appendix M1. (Unico, No. 30 at p. 4)

JCI recommended the proposal that the $H1_2$ test be conducted at maximum speed should be made optional, and the use of slope factors should be permitted if the test is not run. JCI commented that the standardized slope factors predict performance fairly closely, but can lower the HSPF by as much as 0.5 HSPF, and requested to move this change to M1. Additionally, JCI objected to DOE's proposal on $H1_N$ test and commented that if a manufacturer wishes to rate the heating capacity of their units at 47 °F at a speed above the A_2 speed, they should be permitted to do so. (JCI, No. 24 at p. 14)

Goodman supported DOE's proposal to require a full-speed test at 47 °F to be designated $H1_N$. However, Goodman does not support the proposal to mandate that the compressor speed for this test be equal to or higher than the cooling full compressor speed. In addition, although Goodman generally supported DOE's proposal regarding the standardized slope factors to be used if no 47 °F test is run using the same compressor speed as the $H3_2$ test, Goodman commented that the datasets DOE's contractor have used to set the standardized slopes are not appropriate. According to Goodman, developing ratios of capacity based on certified heating capacities can lead to errors because ratings might be conservative. Further, Goodman asserted that it would be possible for models to be counted more than once, or that a limited number of an appropriate cross section of representative models would be included. Additionally, according to Goodman, varying technologies could have different slopes. Goodman suggested that DOE work with AHRI and manufacturers to review real test data. Goodman also supported the optional 5 °F test and suggested DOE to take a further step to provide an optional 5 °F test for two-speed and single-speed heat pumps. (Goodman, No. 39 at p. 5–7)

AHRI suggested that a test procedure similar to triple-capacity heat pumps should be made an optional procedure for variable-speed heat pumps. (AHRI, No. 27 at p. 13)

EEI strongly recommended that the 5 °F test and any additional considered test should remain optional. EEI also suggested that DOE should require tests and information be published for all furnaces and boilers at the same temperatures as for heat pumps. (EEI, No. 34 at p. 2)

Carrier supported DOE's modification to allow the $H1_N$ speed to be any speed between the 17 °F full heating speed and 95 °F full cooling speed. (Carrier, No. 36 at p. 8)

Lennox, ACEEE, NRDC, and ASAP, and NEEA supported DOE's proposals for revising the variable-speed heat pump test methods in appendix M. (Lennox, No. 25 at p. 13; ACEEE, NRDC, and ASAP, No. 33 at p. 7; NEEA, No. 35 at p. 3)

DOE considered the requests to move the proposed variable-speed heat pump test method amendments to appendix M1 and other detailed comments regarding specific aspects of the amendments. DOE revised part of its proposal as discussed later in this section. DOE's intention with the changes to the variable-speed heat pump test procedure of appendix M was to allow the tests conducted previously (*i.e.,* prior to the effective date of the June 2016 final rule) to still be used to represent heat pump performance, while preventing use of extrapolation of the performance below 17 °F using the results of tests conducted at different speeds at 17 °F and 47 °F. For this reason, DOE is not finalizing some aspects of its proposal for appendix M, and instead is finalizing them only for appendix M1.

DOE believes that the standardized slope factors (or use of same-speed tests, if a manufacturer does prefer to retest rather than use the standardized slope factors) would provide more accurate representation of heat pump performance. As discussed in section III.B.3, pursuant to 42 U.S.C. 6293(e), DOE is required to determine to what extent, if any, the proposed test procedure would alter the measured efficiency of the covered product. DOE proposed changes to heating mode test procedure for variable speed units only. For single-speed and two-speed products there would be no change in measured efficiency because they would not be impacted by this change in test procedure. However, variable-speed products would be impacted by this change in test procedure, so the measured efficiency may change.

Where an amended test procedure would alter measured efficiency, EPCA requires DOE to amend an energy conservation standard by measuring, under the amended test procedure, a sample of representative products that minimally comply with the standard. In this case, minimally compliant units are those with single-speed technology. Consistent with the statute, DOE has tested a representative sample of covered products that minimally comply with the existing standard. EPCA requires that the amended standard should constitute the average of the energy efficiency of those units, determined under the amended test procedure. As a result of that testing, DOE has determined that there is no change in measured average energy efficiency for single-speed units between the current test procedure and the amended test procedure. Thus, under 42 U.S.C. 6293(e)(2), the amended standard applicable to the amended test procedure and the current standard applicable to the amended test procedure are the same. As a result, DOE does not need to amend the existing standard to require representations of variable-speed heat pumps to be based on the amended test procedure in appendix M. Therefore, DOE is finalizing aspects of its proposal for appendix M, including the use of standardized slope factors, which might require recalculation of HSPF for variable-speed unit.

DOE believes that Unico's comment about the ''changing the slope factors'' may have been a comment regarding the heating load line equation slope factor rather than the standardized slope factors associated with the appendix M variable speed heat pump proposal. If so, the change was proposed only for appendix M1. If not, DOE's discussion regarding the standardized slope factors in the above paragraph responds to Unico's comment.

Based on the comments received, DOE concluded that the proposal details that commenters believed would lead to a need to retest are (a) requiring the compressor speed for the $H3_2$ and $H2_2$ tests to be the maximum speed at which the system controls would operate the compressor in normal operation in a 17 °F ambient temperature, and (b) requiring the compressor speed for the $H1_N$ test to be no lower than the for the A_2 test.

To resolve the first of these issues, DOE is adopting this requirement in appendix M1, but not appendix M. However, for appendix M, DOE is amending the proposal to require that the compressor speeds used for the $H3_2$ and $H2_2$ tests be the same (if the optional $H2_2$ test is conducted), and will require that the compressor frequency that corresponds to maximum speed at which the system controls would operate the compressor in normal operation in a 17 °F ambient temperature be provided in the certification reports. However, DOE will not post this information to DOE's public certification database. DOE has added this reporting requirement in 10 CFR 429.16(e).

To resolve the second issue, DOE is revising its proposal to allow the compressor speed used for the $H1_N$ test to be lower than used for the A_2 test, provided that the $H1_N$ capacity is no lower than the A_2 cooling capacity. Goodman's comment regarding this issue states that it is normally the case that products on the market today have heating full compressor speed equal to or higher than the cooling full compressor speed, but Goodman believes this does not necessarily have to be the case. (Goodman, No. 39 at p. 6) While DOE agrees that such a possibility could exist, this is not a very strong statement regarding the existence of heat pumps with lower heating speed. Goodman's comment continues with an explanation that achieving roughly equivalent capacity in heating mode at 47 °F as in cooling mode at 95 °F would likely provide better performance at lower ambient temperatures. *Id.* These statements suggest that a reasonable compromise would be to allow lower $H1_N$ speed than A_2 speed as long as the $H1_N$ capacity is no lower, which is the approach that DOE has adopted in this final rule.

Similarly, JCI's comment that the compressor speed for the $H1_N$ test be allowed to be higher than the A_2 speed is consistent with the previously-stated approach that DOE is adopting in this final rule.

As for Goodman's suggestion regarding an optional 5 °F test for two-speed and single-speed heat pumps, DOE discusses this in section III.C.4, as part of its discussion of amendments to appendix M1.

With regard to AHRI's suggestion to add an optional test procedure for variable-speed heat pumps that is similar to the test for triple-capacity heat pumps, DOE considered this suggestion, but is declining to adopt these optional tests in this final rule because stakeholders have not been given an opportunity to comment on them. However, DOE may consider such an option in the future. In response to EEI's comment on making the proposed 5 °F test and any additional test points optional, DOE notes that it has not proposed nor adopted any new heating mode tests for heat pumps that are not optional, either in the June 2016 final rule, the August 2016 SNOPR, or this rulemaking.

In response to JCI's comment that conducting the $H1_2$ test at maximum speed should be made optional, DOE notes that this was optional as proposed and is optional in the test procedure adopted in this final rule.

In response to Goodman's comment about rigorous review of test data to develop the standardized slope factors, DOE requested data or suggestions regarding how they should be changed. 81 FR at 58179 (Aug. 24, 2016). However, such data were not provided. DOE notes that the standardized slope factors, which DOE derived from different data sources, some of which must have represented test data, were remarkably consistent. Further, if capacities reported for both 17 °F and 47 °F test points are conservative, it is not clear that there would be a dramatic difference in the calculated slope. Therefore, DOE has adopted the standardized slope factors proposed in the August 2015 SNOPR.

Regarding EEI's comment that furnace performance should be provided at the same temperatures and for at least two temperatures for both furnaces and CAC/HP, DOE is reluctant to impose that additional reporting burden at this time. The capacity and steady-state efficiency for furnaces does not vary significantly as a function of outdoor temperature. Thus, DOE is not convinced that the additional information would be of significant value to consumers.

8. Clarification of the Requirements of Break-In Periods Prior to Testing

In the August 2016 SNOPR, DOE proposed modifications to the test procedure to clarify the use of break-in, generalizing the requirement so that it applies regardless of who conducts the test, indicating that the break-in requirement applies for each compressor of the unit, and clarifying that the compressor(s) must undergo the certified break-in period (which may not exceed 20 hours) prior to any test period used to measure performance. 81 FR at 58179 (Aug. 24, 2016)

During the August 2016 Public Meeting, Ingersoll Rand commented that DOE's proposed rule was unclear about whether a compressor change-out is required if the compressor of a unit operates longer than the certified break-in period during product development or operation associated with test set-up prior to making the first measurement used to determine an efficiency representation. (Ingersoll Rand, Public Meeting Transcript, No. 20 at pp. 27–29).

Many stakeholders commented that changing out compressors during testing is a significant burden. Nortek suggested that DOE extend the break-in period to 50 hours and allow the break-in to be conducted at ambient conditions. (Nortek, No. 22 at p. 11) ADP and Lennox commented that the 20 hour maximum should remain in place for any verification, enforcement or other non-development testing. ADP also suggested that the break-in period should be part of the public product-specific information so that ICMs can use this information for more accurate AEDM ratings. (ADP, No. 23 at p. 4; Lennox, No. 25 at p. 13) JCI suggested DOE allow up to 72 hours of break-in time and recommended allowing break ins to be conducted before installing the compressor in the unit, or to break in a system outside of the test cell. (JCI, No. 24 at p. 14) AHRI provided data from two compressor manufacturers and suggested DOE extend the allowed break-in period to 72 hours and permit the break-in to be conducted at ambient conditions. Rheem supported AHRI. (AHRI, No. 27 at p. 13–15; Rheem, No. 37 at p. 4) Unico supported a 72-hour minimum break-in period and commented that it is easy to run the unit outside the test chamber. (Unico, No. 30 at p. 5) Emerson commented that longer break-in will ensure repeatability and improve stability of compressor performance. Emerson also included data for several compressors. (Emerson, No. 31 at pp. 1–2) Carrier suggested that DOE allow a 72-hour break-in period and allow break in outside of test chamber while running tests on other units. (Carrier, No. 36 at p. 8–9) Ingersoll Rand, Goodman and the Joint Advocates commented that there is no technical reason to establish an upper limit for break-in. Goodman suggested to permit 72 hours of break-in. (Ingersoll Rand, No. 38 at p. 4; Goodman, No. 39 at p. 8–9; Joint Advocates, No. 33 at p.7) NEEA supported DOE's proposed modification of the test procedure. (NEEA, No. 35 at p. 3)

DOE does not intend to require a compressor change-out in the development test. Rather, the establishment of the 20-hour limit is to maintain test repeatability among labs regardless of who conducts the test. DOE notes that there is no requirement in the test procedure that the break-in has to be conducted in the psychrometric chamber, so manufacturers and technicians have an option, if needed, as to where break-in is conducted. Finally, DOE adopted the 20-hour break-in limit in the June 2016 Final Rule, and the proposal in the August 2016 SNOPR was intended to clarify how this requirement applies for manufacturers and third party testing. Accordingly, DOE will not change the 20-hour limit in this final rule.

In response to ADP's comments, DOE will discuss concerns about reporting requirements for ICMs through a separate process.

9. Modification to the Part Load Testing Requirement of VRF Multi-Split Systems

In the August 2016 SNOPR, DOE proposed to remove the 5 percent tolerance for part load operation from section 2.2.3.a of appendix M when comparing the sum of nominal capacities of the indoor units and the intended system part load capacity for VRF multi-split units. 81 FR at 58179 (Aug. 24, 2016)

DOE received no objections on this proposal, and adopts it in this final rule.

10. Modification to the Test Unit Installation Requirement of Cased Coil Insulation and Sealing

In the August 2016 SNOPR, DOE proposed to remove the statement about insulating or sealing cased coils from appendix M, section 2.2.c, in order to avoid confusion regarding whether sealing of duct connections is allowed. 81 FR at 58180 (Aug. 24, 2016)

DOE received no objections on this proposal, and adopts it in this final rule.

11. Correction for the Calculation of the Low-Temperature Cut-Out Factor for Single-Speed Compressor Systems

Equation 4.2.1–3 in section 4.2.1 of appendix M, used for calculating the low-temperature cut-out factor for a blower coil system heat pump having a single-speed compressor and either a fixed-speed indoor blower or a constant-air-volume-rate indoor blower, or for a single-speed coil-only system heat pump, was incorrectly modified in the June 2016 final rule, in that the ''or'' initially in the equation was changed to an ''and''. 81 FR at 37107 (June 8, 2016). DOE was alerted to this issue in comments received in response to the notice of data availability (NODA) associated with the CAC/HP energy conservation standard rulemaking published October 27, 2016. 81 FR 74727. (Docket Number EERE–2014–BT–STD–0048, AHRI, No. 94 at p. 2; Unico, No. 95 at p. 1) The equation originally used ''or''. This modification could have changed the range of temperature bins for which it is assumed that the heat pump function has cut out. DOE has corrected this issue in this rulemaking in appendix M and also has adopted the correct equation in appendix M1.

12. Clarification of the Refrigerant Liquid Line Insulation

In the June 2016 Final Rule, DOE adopted clarifications for insulation requirements for the refrigerant lines in section 2.2(a) of appendix M. 81 FR at 37027 (June 8, 2016). In some cases, these requirements may indicate that the refrigerant lines should be uninsulated, exposed to the air. However, DOE notes that this requirement is not appropriate to apply for every inch of refrigerant line, particularly where it would conflict with the requirements in ASHRAE 41.1–1986 (RA 2006) (referenced in section 5.1.1 of ASHRAE 37–2009, which is incorporated by reference, see § 430.3). ASHRAE 41.1–1986 (RA 2006) requires in sections 8.2 and 8.3 that it is acceptable to use surface temperature measurement for the refrigerant liquid temperature, but that insulating material extending to at least 6 in. on each side of a surface temperature-measuring element should be installed on the line. The liquid temperature measurement may be essential, *e.g.* when the refrigerant enthalpy method is used as the secondary method (see section 2.10.3 of appendix M). Therefore, DOE has decided to clarify in the test procedure (in both appendices M and M1) that the refrigerant insulation requirement in section 2.2(a) does not apply for portions of the lines insulated according to the ASHRAE 41.1–1986 (RA 2006) requirements for temperature measurement.

Because this clarification simply addresses DOE's intention on how to correctly conduct the test procedure, DOE finds that there is good cause under 5 U.S.C. 553(b)(B) to not issue a separate notice to solicit public comment on this change.

C. Amendments to Appendix M1

The November 2015 SNOPR proposed to establish a new appendix M1 to Subpart B of 10 CFR part 430, which would be required to demonstrate compliance with any new energy conservation standards. 80 FR at 69397 (Nov. 9, 2015) In the August SNOPR, DOE continued to propose establishing a new appendix M1. Under DOE's proposal, the appendix would include all of the test procedure provisions in appendix M as finalized in the June 2016 final rule, all of the changes to appendix M that are finalized in this rulemaking as discussed in section III.B, and all of the additional changes discussed in this section III.C, which would be included only in the new appendix M1. DOE proposed to make appendix M1 mandatory for representations of efficiency starting on the compliance date of any amended energy conservation standards for CAC/HP (however, note the phase-in of testing requirements for certain proposed new requirements for split systems discussed in section III.A.1).

1. Minimum External Static Pressure Requirements

Most of the residential central air conditioners and heat pumps in the United States use ductwork to distribute air in a residence, using either a fan inside the indoor unit or housed in a separate component, such as a furnace, to move the air. External static pressure (ESP) for a CAC/HP is the static pressure rise between the inlet and outlet of the indoor unit that is needed to overcome frictional losses in the ductwork. The external static pressure imposed by the ductwork affects the power consumed by the indoor fan, and therefore also affects the SEER and/or HSPF of a CAC/HP.

a. Conventional Central Air Conditioners and Heat Pumps

The current DOE test procedure [11] stipulates that certification tests for ''conventional'' CACs and heat pump blower coil systems (*i.e.,* CACs and heat pump blower coil systems which are not small-duct, high-velocity systems) must be performed with an external static pressure at or above 0.10 in. wc. if cooling capacity is rated at 28,800 Btu/h or less; at or above 0.15 in. wc. if cooling capacity is rated from 29,000 Btu/h to 42,500 Btu/h; and at or above 0.20 in. wc. if cooling capacity is rated at 43,000 Btu/h or more.

DOE did not propose revisions to minimum external static pressure requirements for conventional blower coil systems in the June 2010 test procedure NOPR, stating that new values and a consensus standard were not readily available.[12] 75 FR 13223, 31228 (June 2, 2010). However, between the June 2010 test procedure NOPR and the November 2015 test procedure SNOPR, many stakeholders submitted comments citing data that suggested the minimum external static pressure requirements were too low and a value

[11] Table 3 of 10 CFR part 430 subpart B appendix M.

[12] In the June 2010 NOPR, DOE proposed lower minimum ESP requirements for ducted multi-split systems: 0.03 in. wc. for units less than 28,800 Btu/h; 0.05 in. wc. for units between 29,000 Btu/h and 42,500 Btu/h; and 0.07 in. wc. for units greater than 43,000 Btu/h. 75 FR at 31232 (June 2, 2010).

of 0.50 in. wc. would be more representative of field conditions. These comments are summarized in the November 2015 test procedure SNOPR. 80 FR at 69317–69318 (Nov. 9, 2015). Ultimately, in the November 2015 SNOPR, DOE proposed to adopt, for inclusion into 10 CFR part 430, subpart B, appendix M1, for systems other than multi-split systems and small-duct, high-velocity systems, minimum external static pressure requirements of 0.45 in. wc. for units with a rated cooling capacity of 28,800 Btu/h or less; 0.50 in. wc. for units with a rated cooling capacity from 29,000 Btu/h to 42,500 Btu/h; and 0.55 in. wc. for units with a rated cooling capacity of 43,000 Btu/h or more. DOE reviewed available field data to determine the external static pressure values it proposed in the November 2015 test procedure SNOPR. DOE gathered field studies and research reports, where publically available, to estimate field external static pressures. DOE previously reviewed most of these studies when developing test requirements for furnace fans. The 20 studies, published from 1995 to 2007, provided 1,010 assessments of location and construction characteristics of CAC and/or heat pump systems in residences, with the data collected varying by location, representation of system static pressure measurements, equipment's age, ductwork arrangement, and air-tightness.[13] 79 FR 500 (Jan. 3, 2014). DOE also gathered data and conducted analyses to quantify the pressure drops associated with indoor coil and filter foulants.[14] The November 2015 test procedure SNOPR provides a detailed overview of the analysis approach DOE used to determine an appropriate external static pressure value using these data. 80 FR at 69318–69319 (Nov. 9, 2015). DOE did not consider revising the minimum external static pressure requirements for SDHV systems in the November 2015 test procedure SNOPR. DOE did, however, propose to establish a new category of ducted systems, short duct systems, which would have lower external static pressure requirements for testing. DOE proposed to define ''short duct system'' to mean ducted systems whose indoor units can deliver no more than 0.07 in. wc. external static pressure when delivering the full load air volume rate for cooling operation. 80 FR at 69314. DOE proposed in the November 2015 SNOPR to require short duct systems to be tested using the minimum external static pressure previously proposed in the June 2010 NOPR for ''multi-split'' systems: 0.03 in. wc. for units less than 28,800 Btu/h; 0.05 in. wc. for units between 29,000 Btu/h and 42,500 Btu/h; and 0.07 in. wc. for units greater than 43,000 Btu/h. 75 FR at 31232 (June 2, 2010)

In response to the November 2015 SNOPR, the CAC/HP ECS Working Group members weighed in on appropriate minimum external static pressure requirements. (CAC ECS: CAC/HP ECS Working Group meeting, No. 86 at pp. 31–128) Recommendation #2 of the CAC/HP ECS Working Group Term Sheet states that the minimum required external static pressure for CAC/HP blower coil systems other than mobile home systems, ceiling-mount and wall-mount systems, low and mid-static multi-split systems, space-constrained systems, and small-duct, high-velocity systems should be 0.50 in. wc. for all capacities. (CAC ECS: ASRAC Term Sheet, No. 76 at p. 2)

In the August 2016 SNOPR, DOE proposed to adopt a minimum external static pressure requirement of 0.50 in. wc. for systems other than mobile home, ceiling-mount and wall-mount systems, low and mid-static multi-split systems, space-constrained systems, and small-duct, high-velocity systems based on DOE's analysis and consistent with the CAC/HP ECS Working Group Term Sheet. 81 FR at 58181 (Aug. 24, 2016)

During the August 2016 SNOPR public meeting and in written comments, many stakeholders expressed support for the new minimum external static requirements that DOE proposed. JCI, Goodman, Unico, AHRI, NEEA, Carrier/UTC, Lennox, Ingersoll Rand, and Nortek expressed support for DOE's proposal to require conventional systems to be tested at a minimum external static pressure of 0.5 in. wc. consistent with Recommendation #2 of the Term Sheet. (JCI, No. 24 at p. 15; Goodman, No. 39 at p. 13; Unico, No. 30 at p. 6; AHRI, No. 27 at p. 16; NEEA, No. 35 at p. 3; Carrier/UTC, No. 36 at p. 9; Lennox, No. 25 at p. 10; Ingersoll Rand, No. 38 at p. 5; Nortek, No. 22 at p. 11)

In light of DOE's analysis results, the Term Sheet recommendation, and support expressed in written comments, DOE is adopting a minimum external static pressure of 0.50 in. wc. for all capacities of conventional CAC/HP products in this final rule.

b. Non-Conventional Central Air Conditioners and Heat Pumps

In response to the November 2015 SNOPR and during the CAC/HP ECS Working Group negotiations, DOE also received comment regarding the minimum external static pressure requirements for mobile home systems, ceiling-mount and wall-mount systems, low and mid-static multi-split systems, space-constrained systems, and small-duct, high-velocity systems. 81 FR at 58181 (Aug. 24, 2016). The CAC/HP ECS Working Group included in its Final Term Sheet Recommendation #2, which is summarized in Table III–2. (CAC ECS: ASRAC Term Sheet, No. 76 at p. 2)

TABLE III–2—CAC/HP ECS WORKING GROUP RECOMMENDED MINIMUM EXTERNAL STATIC STATIC PRESSURE REQUIREMENT

Product description	Minimum external static pressure (in. wc.)
All central air conditioners and heat pumps except (2)–(7) below	0.50.
(2) Ceiling-mount and Wall-mount Blower Coil System	TBD by DOE.
(3) Manufactured Housing Air Conditioner Coil System	0.30.
(4) Low-Static System	0.10.

[13] DOE has included a list of citations for these studies in the docket for the furnace fan test procedure rulemaking. The docket number for the furnace fan test procedure rulemaking is EERE–2010–BT–TP–0010.

[14] Siegel, J., Walker, I., and Sherman, M. 2002. ''Dirty Air Conditioners: Energy Implications of Coil Fouling'' Lawrence Berkeley National Laboratory report, number LBNL–49757.

ACCA. 1995. Manual D: Duct Systems. Washington, DC, Air Conditioning Contractors of America.

Parker, D.S., J.R. Sherwin, et al. 1997. ''Impact of evaporator coil airflow in air conditioning systems'' ASHRAE Transactions 103(2): 395–405.

TABLE III–2—CAC/HP ECS WORKING GROUP RECOMMENDED MINIMUM EXTERNAL STATIC STATIC PRESSURE REQUIREMENT—Continued

Product description	Minimum external static pressure (in. wc.)
(5) Mid-Static System	0.30.
(6) Small Duct, High Velocity System	1.15.
(7) Space-Constrained	0.30.

Recommendation #1 of the CAC/HP ECS Working Group included suggested definitions for distinguishing the CAC/HP varieties included in Recommendation #2 (Table III–2) to enable the proper administration of the CAC/HP ECS Working Group's recommended minimum external static pressure requirements.

DOE agrees with the intent of Recommendation #1 and #2 of the CAC/HP ECS Working Group Term Sheet because DOE recognizes that the CAC/HP varieties included in these recommendations have unique installation characteristics that result in different field external static pressure conditions, and in turn, indoor fan power consumption in the field. Consequently, in the August 2016 test procedure SNOPR, DOE proposed to adopt definitions similar to those that the CAC/HP ECS Working Group recommended for space-constrained systems, low-static systems, and mid-static systems, as well as the recommended minimum external static pressure requirements for those products, to be more reflective of field conditions.

In the August 2016 SNOPR, DOE proposed to adopt the following definitions for the CAC/HP varieties included in Recommendations #1 and #2 in the CAC/HP ECS Working Group Term Sheet, which are slightly modified versions of those suggested in the Term Sheet, but reflect the same intent:

- *Ceiling-mount blower coil system* means a split system for which the outdoor unit has a certified cooling capacity less than or equal to 36,000 Btu/h and the indoor unit is shipped with manufacturer-supplied installation instructions that specify to secure the indoor unit only to the ceiling of the conditioned space, with return air directly to the bottom of the unit (without ductwork), having an installed height no more than 12 inches (not including condensate drain lines) and depth (in the direction of airflow) of no more than 30 inches, with supply air discharged horizontally.
- *Low-static blower coil system* means a ducted multi-split or multi-head mini-split system for which all indoor units produce greater than 0.01 in. wc. and a maximum of 0.35 in. wc. external static pressure when operated at the cooling full-load air volume rate not exceeding 400 cfm per rated ton of cooling.
- *Mid-static blower coil system* means a ducted multi-split or multi-head mini-split system for which all indoor units produce greater than 0.20 in. wc. and a maximum of 0.65 in. wc. when operated at the cooling full-load air volume rate not exceeding 400 cfm per rated ton of cooling.
- *Mobile home blower coil system* means a split system that contains an outdoor unit and an indoor unit that meet the following criteria: (1) Both the indoor and outdoor unit are shipped with manufacturer-supplied installation instructions that specify installation only in a mobile home with the home and equipment complying with HUD Manufactured Home Construction Safety Standard 24 CFR part 3280; (2) the indoor unit cannot exceed 0.40 in. wc. when operated at the cooling full-load air volume rate not exceeding 400 cfm per rated ton of cooling; and (3) the indoor unit and outdoor unit each must bear a label in at least ¼ inch font that reads ''For installation only in HUD manufactured home per Construction Safety Standard 24 CFR part 3280.''
- *Wall-mount blower coil system* means a split-system air conditioner or heat pump for which the outdoor unit has a certified cooling capacity less than or equal to 36,000 Btu/h and the indoor unit is shipped with manufacturer-supplied installation instructions that specify to secure the back side of the unit only to a wall within the conditioned space, with the capability of front air return (without ductwork) and not capable of horizontal airflow, having a height no more than 45 inches, a depth of no more than 22 inches (including tubing connections), and a width no more than 24 inches (in the direction parallel to the wall). 81 FR at 58181–58183 (Aug. 24, 2016)

In response to the August 2016 test procedure SNOPR, NEEA, Lennox, AHRI, Ingersoll Rand, Goodman, Nortek and UTC/Carrier expressed support for DOE's proposed minimum external static pressure requirements and definitions for all product types. (NEEA, No. 35 at p. 3; Lennox, No. 25 at p. 14; AHRI, No. 27 at p. 16; IR, No. 38 at p. 5; Goodman, No. 39 at p. 13; Nortek, No. 22 at p. 12; UTC/Carrier, No. 36 at p. 9)

In written comments, JCI, ADP and First Co. suggested that DOE modify its proposed definition for wall-mount blower coil system. JCI, ADP, and First Co. pointed out that these systems have common installations that do not meet DOE's proposed definition. JCI, ADP and First Co. stated that wall-mount units are not exclusively installed by securing the back of the unit to a wall within the conditioned space. Instead, wall-mount units are often mounted to adjacent wall studs or within an enclosure (*e.g.,* a closet) such that the front side of the unit is flush with the wall of the conditioned space. JCI, ADP, and First Co. recommended that DOE modify the definition of wall-mount blower coil system to allow for these types of installations. (JCI, No. 24 at p. 15; ADP, No. 23 at pp. 4–5; First Co., No. 21 at p. 4–5) ADP provided an example installation manual for an ADP wall-mount blower coil that provided instructions for the installation options mentioned. ADP suggested adding ''the ability'' and remove ''only'' from the proposed definition. (ADP, No. 23 at p 4) Mortex echoed ADP's suggested modifications to DOE's proposed definition for wall-mount blower-coil systems. (Mortex, No. 26 at p. 4)

DOE recognizes that wall-mount units are often installed as JCI, ADP, Mortex, and First Co. describe in their comments. In this final rule, DOE is modifying the definition proposed in the August 2016 test procedure SNOPR to maintain the intent of the Term Sheet but also allow for the ''flush-mount'' installations described by JCI, ADP, Mortex and First Co. DOE is adopting the following modified definition for ''wall-mount blower coil system'':

Wall-mount blower coil system means a split-system air conditioner or heat pump for which (a) the outdoor unit has a certified cooling capacity less than or equal to 36,000 Btu/h; (b) the indoor unit(s) is/are shipped with manufacturer-supplied installation instructions that specify mounting only by (1) securing the back side of the unit to a wall within the conditioned space, or (2) securing the unit to adjacent wall studs or in an enclosure, such as a closet, such that the indoor unit's front face is flush with a wall in the conditioned space; (c) has front air return without ductwork and is not capable of horizontal air discharge; and (d) has a height no more than 45 inches, a depth (perpendicular to the wall) no more than 22 inches (including tubing connections), and a width no more than 24 inches (parallel to the wall).

In response to the August 2016 test procedure SNOPR, DOE received comment on its proposed definition for ceiling-mount blower coil system. In its comments, First Co. stated that these systems have common installations that do not meet DOE's proposed definition. According to First Co., ceiling-mount indoor units are often installed in a furred down space, which requires that return air comes into the back of the unit either through a duct or through the furred down space. DOE understands a furred down space to be an area below ceiling level that is enclosed and finished (*e.g.,* using drywall and paint). First Co. also identified another common installation practice for ceiling-mount indoor units used in applications with dropped ceilings in which the indoor unit is equipped with an insulated box that is suspended such that the bottom of the unit is flush with the ceiling and return air comes into the bottom of the unit. First Co. recommended modifications to DOE's proposed definition for ceiling-mount blower coil system to allow for these other common installation types. (First Co., No. 21 at pp. 4–5)

DOE recognizes that ceiling-mount units are often installed as First Co. describes. In this final rule, DOE is modifying the definition proposed in the August 2016 test procedure SNOPR to maintain the intent of the Term Sheet but also allow for the installations described by First Co. DOE is adopting the following modified definition for ''ceiling-mount blower coil system'':

Ceiling-mount blower coil system means a split system for which (a) the outdoor unit has a certified cooling capacity less than or equal to 36,000 Btu/h; (b) the indoor unit(s) is/are shipped with manufacturer-supplied installation instructions that specify to secure the indoor unit only to the ceiling, within a furred-down space, or above a dropped ceiling of the conditioned space, with return air directly to the bottom of the unit without ductwork, or through the furred-down space, or optional insulated return air plenum that is shipped with the indoor unit; (c) the installed height of the indoor unit is no more than 12 inches (not including condensate drain lines) and the installed depth (in the direction of airflow) of the indoor unit is no more than 30 inches; and (d) supply air is discharged horizontally.

The CAC/HP ECS Working Group tasked DOE with determining the appropriate minimum external static pressure for ceiling-mount and wall-mount systems. During the CAC/HP ECS Working Group meetings, manufacturers of these systems suggested a minimum external static pressure requirement of 0.30 in. wc. (CAC ECS: CAC/HP ECS Working Group meeting, No. 88 at p. 31) However, the CAC/HP ECS Working Group did not adopt this as a recommendation primarily due to lack of time to thoroughly review the subject. In the August 2016 test procedure SNOPR, DOE proposed to specify a minimum external static pressure requirement of 0.30 in. wc. for ceiling-mount and wall-mount systems, consistent with manufacturers' recommendations.

In response to the August 2016 SNOPR, First Co. disagreed with DOE's proposed minimum external static pressure requirements for ceiling-mount and wall-mount blower coil systems. First Co. claimed that the minimum external static pressure requirement for these products should be no greater than 0.20 in. wc. According to First Co., ceiling-mount and wall-mount systems typically use limited length or short run duct work, which produces lower static pressure. First Co. contested that 0.30 in. wc. is unreasonably high for representing such ductwork and that the requirement will result in reductions in product ratings and negative impacts on small manufacturers and product availability. (First Co., No. 21 at pp. 3–4) NEEA, Lennox, AHRI, Ingersoll Rand, Goodman, and UTC/Carrier expressed support for DOE's proposed minimum external static pressure requirement of 0.30 in. wc. for these products. (NEEA, No. 35 at p. 3; Lennox, No. 25 at p. 14; AHRI, No. 27 at p. 16; IR, No. 38 at p. 5; Goodman, No. 39 at p. 13; UTC/Carrier, No. 36 at p. 9).

DOE recognizes that ceiling-mount and wall-mount systems use shorter duct runs than conventional systems, which will result in lower static pressure. For this reason, DOE proposed a lower minimum external static pressure requirement for these products relative to its proposed minimum external static pressure requirement for conventional systems. DOE disagrees with First Co. that 0.30 in. wc. is not representative of field-installed ceiling-mount and wall-mount systems because manufacturers of these products recommended 0.30 in. wc. during the CAC/HP ECS Working Group Negotiations. (Docket EERE–2014–BT–STD–0048, CAC/HP ASRAC Working Group Meeting, October 13, 2015, No. 88 at p. 21) In addition, publicly-available product literature for these products include airflow data tables that include performance at 0.30 in. wc. (Wall Mount Blower Coil Literature Example, No. 41 at p. 3) DOE understands that higher minimum external static pressure requirements will result in reductions to rated performance. These impacts will be considered and accounted for in the energy conservation standard levels set by the concurrent energy conservation standard rulemaking. Therefore, DOE is adopting 0.30 in. wc. as the minimum external static pressure requirement for ceiling-mount and wall-mount blower coil systems in this final rule.

Recommendation #2 of the Term Sheet includes a recommended minimum external static pressure for ''space-constrained'' products. The Term Sheet does not differentiate between space-constrained outdoor units paired with conventional indoor units from those paired with non-conventional indoor units. In the August 2016 SNOPR, DOE proposed that when space-constrained outdoor units are paired with conventional indoor units, the minimum external static pressure requirement for space-constrained systems recommended by the CAC/HP ECS Working Group, 0.30 in. wc., would not be appropriate. Consequently, DOE proposed to apply the minimum external static pressure requirement included for space-constrained products in the Term Sheet only to single- package space-constrained products or space-constrained outdoor units paired with space-constrained indoor units. 81 FR at 58163, 58182 (Aug. 24, 2016).

In written comments, AHRI and Nortek expressed concern with DOE's proposal to modify the external static pressure requirements when space-constrained outdoor units are paired with conventional indoor units. AHRI and Nortek stated that there is no definition of a ''space-constrained indoor unit'' (air handler). AHRI and Nortek added that a space-constrained condensing unit rated using a conventional air handler at 0.5 in. wc would not be able to meet existing efficiency standards. According to AHRI and Nortek, size restrictions of space-constrained products require rating with an efficient conventional air handler as a matched system to meet existing standards. AHRI and Nortek submit that, by definition, space-constrained condensing units are all under 30,000 Btu/h, with limited applications. AHRI and Nortek concluded that the minimum external static pressure requirement for space-constrained systems recommended by the CAC/HP Working Group, 0.30 in. wc., was not only appropriate for these installations; they are required in order for manufacturers to offer these niche products, *i.e.* that DOE should not require use of 0.5 in. wc. for space-constrained system combinations using conventional air handlers. (AHRI, No. 27 at pp. 16–17; Nortek, No. 22 at p. 13).

In response to AHRI's and Nortek's comments, DOE understands that split-system space-constrained systems that comprise a space-constrained outdoor unit and conventional indoor unit are typically installed in homes with size restrictions that are different than homes in which conventional split-systems (*i.e.,* conventional outdoor and indoor unit) are typically installed. Space-constrained systems (regardless of whether paired with a conventional or non-conventional indoor unit) are more commonly installed in homes in which the system is installed in closer proximity to the conditioned space. Ductwork is typically shorter and less restrictive as a result. As such, the CAC/HP ECS Working Group recommended minimum external static pressure of 0.30 in. wc. is more representative. DOE is adopting 0.30 in. wc. for all space-constrained products in this final rule. DOE is adopting this provision because it will result in a test procedure that produces test results that measure the energy efficiency, energy use, or estimated annual operating cost of space-constrained products during a representative average use cycle. DOE is adopting this provision irrespective of comments regarding its implications on products' ability to meet standards. DOE will account for impacts to rated values in the concurrent energy conservation standard rulemaking.

In the August 2016 SNOPR, DOE proposed to adopt the CAC/HP ECS Working Group recommendations for minimum external static pressure requirements for low-static and mid-static systems. 81 FR at 58182–58183 (Aug. 24, 2016).

As mentioned, many stakeholders agreed with DOE's proposed minimum external static pressures and definitions for all product types. (NEEA, No. 35 at p. 3; Lennox, No. 25 at p. 14; AHRI, No. 27 at p. 16; IR, No. 38 at p. 5; Goodman, No. 39 at p. 13; Nortek, No. 22 at p. 12; UTC/Carrier, No. 36 at p. 9) Unico supported DOE's proposal, but voiced one concern. Unico recommended that DOE eliminate the mid-static product class, change the range for low static from 0.01 to 0.49 in. wc. so as not to overlap with the range for normal ducted systems, and to test those products as low-static (unless DOE would plan to establish a separate standard for mid-static systems). According to Unico, the mid-static products would be able to meet the low-static requirements without difficulty, so Unico would not separate these products into a separate class. Unico recommended that DOE add a requirement to the test procedure that both low and mid-static products should be labeled as ''low static'' with the maximum static clearly written on the product rating label, so that a manufacturer would be able to list the mid-static pressure on their literature and labels, while the product would still considered a low-static system (Unico, No. 30 at p. 5).

DOE does not agree with Unico's recommendation. Based on discussions during the CAC/HP ASRAC Working Group Negotiations, and as reflected in the Term Sheet recommendations, DOE understands that there are ducted multi-split and multi-head mini-split systems that are designed and installed to produce between 0.20 in. wc. and 0.65 in. wc. Testing these systems at 0.10 in. wc., as Unico recommends, would not be representative of field performance because they are typically installed in more restrictive applications, which results in higher fan energy consumption. In addition, testing these ''mid-static'' systems, at the same external static pressure as ''low-static,'' would not produce results reflective of relative performance. In the field, a ''mid-static'' system, which is typically installed in more restrictive applications, is expected to have higher fan energy consumption than a ''low-static'' system. Testing both types of systems at the same external static pressure would ignore this difference and would not reflect the increased fan energy consumption of the ''mid-static'' system compared to the ''low-static'' system. DOE is not establishing a separate product class or a separate standard for ''mid-static'' systems, as Unico infers. DOE is only establishing a differing test conditions for ''low-static'' and ''mid-static'' systems to reflect the differences in their application and resulting differences in field performance.

The CAC/HP ECS Working Group did not recommend changing the current minimum external static pressure required (1.15 in. wc.) for SDHV systems with a cooling or heating capacity between 29,000 to 42,500 Btu/h. However, the CAC/HP ECS Working Group recommended that 1.15 in. wc. also be used as the minimum external static pressure requirement for SDHV systems of all other capacities. Using a single minimum external static pressure value for all capacities of a given CAC/HP variety is consistent with the approach recommended by the Working Group for all CAC/HP varieties. In the August 2016 SNOPR, DOE proposed to adopt the Working Group recommendation for the minimum external static pressure requirement for SDHV systems. 81 FR at 58183 (Aug. 24, 2016).

DOE did not receive any negative comments regarding its August 2016 test procedure SNOPR proposed minimum external static pressure requirements for SHDV systems, and DOE is adopting these requirements in this final rule.

Table III–3 summarizes the minimum external static pressure requirements that DOE is adopting in this final rule.

TABLE III–3—MINIMUM EXTERNAL STATIC PRESSURE REQUIREMENTS

CAC/HP variety	Minimum external static pressure (in. wc.)
Conventional (*i.e.*, all central air conditioners and heat pumps not otherwise listed in this table)	0.50
Ceiling-mount and Wall-mount	0.30
Mobile Home	0.30

TABLE III–3—MINIMUM EXTERNAL STATIC PRESSURE REQUIREMENTS—Continued

CAC/HP variety	Minimum external static pressure (in. wc.)
Low-Static	0.10
Mid-Static	0.30
Small Duct, High Velocity	1.15
Space-Constrained (indoor and single-package units only)	0.30

c. Certification Requirements

In the August 2016 SNOPR, DOE proposed to establish the certification requirements for appendix M1 to require manufacturers to certify the kind(s) of CAC/HP associated with the minimum external static pressure used in testing or rating (*i.e.,* ceiling-mount, wall-mount, mobile home, low-static, mid-static, small duct high velocity, space-constrained, or conventional/not otherwise listed). In the case of mix-match ratings for multi-split, multi-head mini-split, and multi-circuit systems, manufacturers would be allowed to select two kinds. In addition, models of outdoor units for which some combinations distributed in commerce meet the definition for ceiling-mount and wall-mount blower coil system, would still be required to have at least one coil-only rating (which uses the 441W/1000 scfm default fan power value) that is representative of the least efficient coil distributed in commerce with the particular model of outdoor unit. Mobile home systems would also be required to have at least one coil-only rating that is representative of the least efficient coil distributed in commerce with the particular model of outdoor unit. Further, DOE proposed to specify a default fan power value of 406W/1000 scfm, rather than 441W/1000 scfm, for mobile home coil-only systems. Details of this proposal are discussed in detail in section III.C.2. 81 FR at 58183 (Aug. 24, 2016).

DOE did not receive any comments on the certification requirements regarding minimum external static pressure or default fan power. Comments on the minimum external static pressure requirements and default fan power are included in sections III.C.1 and III.C.2, respectively.

d. External Static Pressure Reduction Related to Condensing Furnaces

In the November 2015 SNOPR, DOE requested comment on its proposal to implement a 0.10 in. wc. reduction in the minimum external static pressure requirement for air conditioning units tested in blower coil (or single-package) configuration in which a condensing furnace is in the airflow path during the test. This issue was also discussed as part of the CAC/HP ECS Working Group negotiation process. In response to the November 2015 SNOPR, stakeholders commented that they did not support DOE's proposed reduction in the minimum external static pressure requirement because it would result in test results that are less representative of field energy use. (CAC TP: ADP, No. 59 at p. 12; Lennox, No. 61 at p. 20; NEEA and NPCC, No. 64 at p. 8; California IOUs, No. 67 at p. 6; Rheem, No. 69 at p. 17; ACEEE, NRDC, ASAP, No. 72 at p. 4) Recommendation #2 of the CAC/HP ECS Working Group Term Sheet reflects this sentiment, stating that DOE should not adopt its proposed reduction in minimum external static pressure required for units paired with condensing furnaces. (CAC ECS: CAC/HP ECS Working Group Term Sheet, No. 76 at p. 2).

In the August 2016 SNOPR, in light of public comments and the consensus of the CAC/HP ECS Working Group, DOE did not propose to adopt a reduced minimum external static pressure requirement for air conditioning units tested in blower coil (or single-package) configuration in which a condensing furnace is in the airflow path during the test. 81 FR at 58184 (Aug. 24, 2016).

In response to the August 2016 SNOPR, ADP agreed with removing the reduced ESP as it is not representative of actual installed performance. ADP also commented there were other more suitable means to drive the adoption of condensing furnaces. (APD, No. 23 at p. 4) NEEA, the Joint Advocates, UTC, Goodman, JCI, and Ingersoll Rand also supported this proposal. (NEEA, No. 35 at p. 3; Joint Advocates, No. 33 at p. 7; UTC, No. 36 at p. 10; Goodman, No. 39 at p. 11; JCI, No. 24 at p. 15; Ingersoll Rand, No. 38 at p. 5) Rheem also agreed with removing the reduced ESP, stating that its use could cause the representation of cooling efficiency to become similar to that with a non-condensing furnace, which would not reflect how the system would operate in the field. (Rheem, No. 37 at p. 5).

DOE did not receive any comments in favor of a reduced minimum external static pressure for systems tested with a condensing furnace. In light of stakeholder comments, DOE did not include a reduced minimum external static pressure requirement for these products in this final rule.

2. Default Fan Power for Rating Coil-Only Units

The default fan power value (hereafter referred to as ''the default value'') is used to represent fan power input when testing coil-only air conditioners, which do not include their own indoor fans.[15] In the current test procedure, the default value is 365 Watts (W) per 1,000 cubic feet per minute of standard air (scfm) and there is an associated adjustment to measured capacity to account for the fan heat equal to 1,250 British Thermal Units per hour (Btu/h) per 1,000 scfm (10 CFR part 430, subpart B, appendix M, section 3.3.d). The default value was discussed in the June 2010 NOPR, in which DOE did not propose to revise it due to uncertainty on whether higher default values would better represent field installations. 75 FR 31227 (June 2, 2010). In the November 2015 SNOPR, DOE proposed to update the default value to be more representative of field conditions (*i.e.,* consistent with indoor fan power consumption at the minimum required external static pressures proposed in the November 2015 SNOPR). In the November 2015 SNOPR, DOE used indoor fan electrical power consumption data from product literature, testing, and exchanges with manufacturers collected for the furnace fan rulemaking (79 FR 506, January 3, 2014) to determine an appropriate default value for coil-only products.[16] (80 FR 69318) DOE calculated the adjusted default fan power to be 441 W/1000 scfm. In the November 2015

[15] See 10 CFR part 430, subpart B, appendix M, section 3.3.d.

[16] For a complete explanation of DOE's methodology, see 80 FR at 69319–69320 (Nov. 9, 2015).

SNOPR, DOE proposed to use this value in appendix M1, while keeping the current default fan power of 365 W/1000 scfm in appendix M.

In response to the November 2015 SNOPR, many stakeholders supported raising the coil-only test default fan power to 441 W/1000 scfm to allow for more representative ratings of units. (CAC TP: NEEA and NPCC, No. 64 at p. 8; ACEEE, NRDC, and ASAP, No. 72 at p. 4; California IOUs, No. 67 at p. 2)

The CAC/HP ECS Working Group also discussed the default value as part of the negotiation process. Ultimately, the Working Group came to a consensus on a recommendation for the default value. Recommendation #3 of the CAC/HP ECS Working Group Term Sheet states that the default fan power for rating the performance of all coil-only systems other than manufactured housing products be 441W/1000 scfm. (CAC ECS: ASRAC Working Group Term Sheet, No. 76 at p. 3)

Consistent with the CAC/HP ECS Working Group Term Sheet, DOE maintained its previous proposal to use a default value of 441 W/1000 scfm for split-system air conditioner, coil-only tests in the August 2016 SNOPR. DOE also proposed to adjust measured capacity to account for the fan heat by 1,505 Btu/h per 1,000 scfm, consistent with 441W/1000 scfm. 81 FR at 58184 (Aug. 24, 2016). DOE proposed to use these values in appendix M1 of 10 CFR part 430 subpart B in place of the default fan power of 365 W/1000 scfm that had been used previously in appendix M.

Recommendation #3 of the CAC/HP ECS Working Group Term Sheet also stated that DOE should calculate an alternative default fan power for rating mobile home air conditioner coil-only units based on the minimum external static pressure requirement for blower coil mobile home units (0.30 in. wc.) suggested in recommendation #2 of the Term Sheet. (CAC TP: ASRAC Working Group Term Sheet, No. 76 at p. 3) As discussed in section III.C.1, the CAC/HP ECS Working Group included this recommendation because HUD requires less restrictive ductwork for mobile homes than for other types of housing, which reduces electrical energy consumption of the indoor fan. The default value used to rate coil-only mobile home systems should reflect this difference in field energy consumption to improve the field representativeness of the test procedure.

In the August 2016 test procedure SNOPR, DOE used the same aforementioned furnace fan power consumption data and methodology to calculate the appropriate default value for mobile home fan power consumption, which DOE found to be 406 W/1000 scfm. DOE proposed to use 406 W/1000 scfm and adjust cooling capacity by 1,385 Btu/h per 1,000 scfm for mobile home coil-only tests in the August 2016 test procedure SNOPR. 81 FR at 58163, 58183 (Aug. 24, 2016).

In response to the August 2016 SNOPR, AHRI, Nortek, Lennox, Ingersoll Rand, JCI, ACEEE, NRDC, ASAP, and Rheem supported DOE's proposal to use a default value of 441 W/1000 scfm for split-system air conditioner, coil-only tests. These stakeholders also supported a unique default fan power of 406 W/1000 scfm for rating mobile home coil-only units. (AHRI, No. 27 at p. 17; Nortek, No. 22 at p. 13; Lennox, No. 25 at p. 8; Ingersoll Rand, No. 38 at p. 5; JCI, No. 24 at p. 16; ACEEE, NRDC, and ASAP, No. 33 at p7; Rheem, No. 37 at p. 3) Carrier/UTC also expressed support for a default fan power value of 441 W/1000 scfm for split-system air conditioner, coil-only tests. (Carrier/UTC, No. 36 at p. 10) ADP and Lennox also expressed support for 406 W/1000CFM as a default fan power value for coil-only mobile home applications. (ADP, No., 23 at p.5, Lennox, No. 25 at p. 8) DOE did not receive any negative comments regarding the use of 441 W/1000 scfm or 406 W/1000 scfm as the default fan power values for conventional split-system or mobile home coil-only tests, respectively. DOE also did not receive any additional data to validate these values.

In light of stakeholder support and no adverse comments, DOE is adopting a default fan power value of 441 W/1000 scfm and capacity adjustment of 1,505 Btu/h/1000 scfm for non-mobile home coil-only systems and a default fan power value of 406 W/1000 scfm and capacity adjustment of 1,385 Btu/h/1000 scfm for mobile home coil-only systems.

In the August 2016 test procedure SNOPR, DOE proposed a definition for a mobile home coil-only system to appropriately apply the proposed default value for these kinds of CAC/HP. DOE proposed the following:

• *Mobile home coil-only system* means a coil-only split system that includes an outdoor unit and coil-only indoor unit that meet the following criteria: (1) The outdoor unit is shipped with manufacturer-supplied installation instructions that specify installation only for mobile homes that comply with HUD Manufactured Home Construction Safety Standard 24 CFR part 3280, (2) the coil-only indoor unit is shipped with manufacturer-supplied installation instructions that specify installation only in a mobile home furnace, modular blower, or designated air mover that complies with HUD Manufactured Home Construction Safety Standard 24 CFR part 3280, and (3) the coil-only indoor unit and outdoor unit each has a label in at least ¼ inch font that reads, ''For installation only in HUD manufactured home per Construction Safety Standard 24 CFR part 3280.'' 81 FR at 58163, 58185 (Aug. 24, 2016).

In written comments, Rheem, JCI, ACEEE, NRDC, and ASAP expressed support for DOE's proposed definition for mobile home coil-only system. (Rheem, No. 37 at p.5; JCI, No. 24 at p.16, ACEEE, NRDC, and ASAP, No. 33 at p.7)

Some stakeholders offered suggested improvements to the definition to better differentiate mobile home coil-only systems from other types of systems. ADP explained that indoor units are often installed in attics, basements, closets and other areas of limited access, so most consumers would not see a label, limiting the usefulness of a label. (ADP, No., 23 at p.6) Lennox and ADP recommended that DOE add the following physical indoor coil characteristics to the definition of mobile home coil-only system in addition to labeling requirements to limit the definition to products exclusively manufactured for mobile homes:

(1) Downturned refrigerant connections
(2) refrigerant connections on left hand side of coil (when viewed from the front)
(3) down-flow capable
(4) maximum size of 20″ wide, 32″ high and 21″ deep (Lennox, No. 25 at p. 8; ADP, No., 23 at p.6)

ADP added that these features are shared by products marketed as mobile home coils and collectively are not present in coils marketed for other applications. Mortex commented that mobile home furnaces have a unique footprint and are only compatible with indoor coils that have a drain pan footprint of 18.5″ wide by 21″ long. Mortex suggests that the definition for mobile home coil-only should include these dimension restrictions for indoor coils. (Mortex, No. 26 at p. 3)

DOE appreciates the suggestions from ADP, Lennox, and Mortex. DOE agrees that a definition that includes descriptions of physical characteristics unique to indoor and outdoor units and combinations that are installed in mobile homes will better distinguish mobile home coil-only systems from other systems. DOE reviewed public product literature for mobile home indoor coils to evaluate the additional criteria suggested by stakeholders.

DOE's search confirmed many of the suggestions, but not all. DOE could not confirm with confidence that all mobile home indoor coils include downturned refrigerant connections on the left hand side when viewed from the front. DOE also found mobile home indoor units that slightly exceeded the height limit that ADP and Lennox recommend. For these reasons, DOE is modifying its proposed definition to include some, but not all, of the physical characteristics that interested parties recommend. In this final rule, DOE is adopting the following definition for mobile home coil-only system:

Mobile home coil-only system means a coil-only split system that includes an outdoor unit and coil-only indoor unit that meet the following criteria: (1) The outdoor unit is shipped with manufacturer-supplied installation instructions that specify installation only for mobile homes that comply with HUD Manufactured Home Construction Safety Standard 24 CFR part 3280, (2) the coil-only indoor unit is shipped with manufacturer-supplied installation instructions that specify installation only in or with a mobile home furnace, modular blower, or designated air mover that complies with HUD Manufactured Home Construction Safety Standard 24 CFR part 3280, and has dimensions no greater than 20″ wide, 34″ high and 21″ deep, and (3) the coil-only indoor unit and outdoor unit each has a label in at least ¼ inch font that reads ''For installation only in HUD manufactured home per Construction Safety Standard 24 CFR part 3280.''

As discussed in detail in section III.C.1.b, in response to stakeholder comment, DOE is adopting a lower minimum external static pressure requirement for space-constrained products to better reflect their field-installed conditions. Similar to mobile home coil-only units, space-constrained coil-only tests should use a default fan power value and capacity adjustment representative of operation at the minimum external static pressure. Recommendation #2 of the Term Sheet includes 0.30 in. wc. as the suggested minimum external static pressure for both mobile home and space-constrained products. As discussed earlier in this section, DOE has determined, with stakeholder support, that a default fan power value of 406 W/1000 scfm and capacity adjustment of 1,385 Btu/h/1000 scfm are consistent with operation at 0.30 in. wc. For this reason, DOE is adopting a default fan power value of 406 W/1000 scfm and capacity adjustment of 1,385 Btu/h/1000 scfm for space-constrained products in this final rule.

3. Revised Heating Load Line Equation

a. Revision of the Heating Load Line Analysis and Proposals

DOE initially proposed revisions to the heating load line equation used in the calculation of heating season performance factor (HSPF) in the November 2015 SNOPR. 80 FR at 69320–69322 (Nov. 9, 2015) The proposals were based on a 2015 Oak Ridge National Laboratory (ORNL) study [17] that examined the heating load line equation for cities representing the six climate regions of the HSPF test procedure in appendix M. DOE received comments on its heating load line equation proposals both in written form in response to the November 2015 SNOPR and verbally during the CAC/HP ECS Working Group meetings. DOE considered the comments received, worked with ORNL on re-examination of certain aspects of the analysis described in the 2015 study, and revised its proposals for revision of the heating load line equation. The revised proposal presented in the August 2016 SNOPR included the following test procedure amendments.

- The zero-load temperature would vary by climate region according to the values provided in Table III–4, but remain at 55 °F (as proposed in the November 2015 SNOPR) for Region IV;
- The heating load line equation slope factor for single- and two-stage heat pumps would vary by climate region, as shown in Table III–4, and be 1.15 for Region IV; and
- For variable-speed heat pumps, the heating load line equation slope factor would be 7 percent less than for single- and two-stage heat pumps. It would vary by climate region, as shown in Table III–4, and be 1.07 for Region IV; 81 FR at 58189 (Aug. 24, 2016)

DOE also revised the heating load hours based on the new zero-load temperatures of each climate region. The revised heating load hours are also given in Table III–4.

TABLE III–4—CLIMATE REGION INFORMATION PROPOSED IN THE AUGUST 2016 SNOPR NOTICE

	Region Number					
	I	II	III	IV	V	VI *
Heating Load Hours	493	857	1247	1701	2202	1842
Zero-Load Temperature, T_{zl}, °F	58	57	56	55	55	57
Heating Load Line Equation Slope Factor, C	1.10	1.06	1.30	1.15	1.16	1.11
Variable-speed Slope Factor, C_{VS}	1.03	0.99	1.21	1.07	1.08	1.03

* Pacific Coast Region.
Note: Some of the values in this table for Region III differ from those presented in the SNOPR. See discussion of these corrections below.

Following from this proposed heating load line equation change, DOE also proposed in the August 2016 SNOPR to require cyclic testing for variable-speed heat pumps be run at 47 °F, rather than using the 62 °F ambient temperature that is required by the current test procedure (see appendix M, section 3.6.4 Table 11). The test would still be conducted using minimum compressor speed. The modified heating load line cyclic test at 47 °F would be more representative of the conditions for which cycling operation is considered in the HSPF calculation. 81 FR at 58190 (Aug. 24, 2016)

In addition, for variable-speed heat pumps, the SEER would be calculated using a building load that is adjusted downwards by 7 percent, consistent with the heating load adjustment.

Heating Load Line Zero-Load Temperature and Slope Factor

A number of commenters disagreed with the zero-load temperature and/or the slope factor proposed for the heating load line equation.

EEI commented that the zero-load temperatures appeared to be too low in light of the predominance of older houses in the building stock, and that the approach may be missing many

[17] ORNL, Rice, C. Keith, Bo Shen, and Som S. Shrestha, 2015. *An Analysis of Representative Heating Load Lines for Residential HSPF Ratings,* ORNL/TM–2015/281, July. (Docket No. EERE–2009–BT–TP–0004–0046).

heating load hours between 55 °F and 65 °F outdoor temperatures. (EEI, Public Meeting Transcript, No. 20 and pp. 82–83) In written comments, EEI reiterated objection to a 55 °F zero-load temperature, asserting that house temperature would fall to 55 °F if the heating system provided no heat at warmer temperatures. They stated most houses are not insulated as well as newer houses, and assuming zero heating system operation between 55 °F and the indoor thermostat settings (*e.g.,* 68 °F) is not realistic and results in lowering the estimated seasonal efficiency of heat pumps. EEI suggested using a zero-load temperature of 65 °F. EEI suggested using a slope of 0.77 or 1.02. (EEI, No. 34 at pp. 2–6)

In its comments, AHRI did not agree with the zero load point of 55 °F. AHRI commented that DOE's proposal was solely based on computer modeling and that AHRI members had submitted real world data from across the entire country during the negotiations to support AHRI's position. AHRI recommended keeping the existing zero-load temperature as 65 °F, and a single heating load line for all products with a 1.02 slope. (AHRI, No. 27 at p. 18) Mitsubishi, Carrier, Lennox, Nortek, Ingersoll Rand, and Goodman all submitted comments agreeing with AHRI's recommendation to use a 65 °F zero-load temperature and a 1.02 slope factor. (Mitsubishi, No. 29 at pp. 3–4; Carrier, No. 36 at p. 10; Lennox, No. 25 at pp. 9–10; Nortek, No. 22 at p. 14; Ingersoll Rand, No. 38 at p. 5; Goodman, No. 39 at p. 9–10)

JCI commented that the ORNL analysis was flawed in that it did not measure the heating load in homes in which human occupants were present. JCI expressed belief that a good survey would find heating load occurring even into the 70 °F–75 °F range in certain regions of the country for certain demographics, and recommended DOE use the 65 °F value for the zero-load temperature.

Bruce Harley Energy Consulting (BHEC) provided field monitoring data and analysis of heating loads conducted at the request of PG&E. The work addressed heating load data of seven homes covering regions I/II,[18] IV, and V that were monitored to measure heating system operation. The data sets and analysis for these houses were not explained extensively in the BHEC comment, but DOE understands that average heating loads were determined by 5 °F-wide temperature bins for hours representing at least a full heating season for each location. Linear curve fits to the binned loads as a function of temperature were determined. The zero-load temperatures for the linear fits lie within a range between 57 °F and 61 °F. Based on this study, BHEC suggested DOE use a value of 60 °F for the zero-load temperature for all climate regions. BHEC also pointed out that these homes are likely to be somewhat less efficient than the 2006 IECC. (BHEC, No. 28 at pp. 2–3)

BHEC's initial comparison of the regional heating load lines with the load lines determined for the seven monitored locations led to the conclusion that the heating load line equation in the August 2016 SNOPR incorrectly included the term T_{OD} (the regional outdoor design temperature) in the denominator. The comment provided analysis showing that the value T_{OD} should be replaced with 5 °F, which is the outdoor design temperature for Region IV. (BHEC, No. 28 at pp. 3–6) With this change, and use of 60 °F as the zero-load temperature, the comment showed that the field data provided good agreement with the calculated heating load lines using the 1.15 slope factor proposed in the August 2016 SNOPR for all but one of the seven monitored locations. This location, ''Site W'', has an unusually high heating load, as indicated by the comment. BHEC concluded that DOE should consider adopting a heating load line with a 60 °F zero-load temperature and a 1.15 slope factor. (BHEC, No. 28 at pp. 6–8)

PG&E commented during the public meeting that the August 2016 SNOPR proposals were not consistent with recent field data not available during the CAC/HP ECS negotiations, and that more details would be provided later. (PG&E, Public Meeting Transcript, No. 20 at p. 84) These additional details presumably are provided by the BHEC comment. The CA–IOUs (which includes PG&E) reiterated some of the discussion of the BHEC comment and supported the 60 °F zero-load temperature and the 1.15 slope factor, although indicating that the selection of zero-load temperature has less impact on measured efficiency. (CA IOU, No. 32 at pp. 1–4)

NEEA supported BHEC's comments on zero-load temperature and slope factor. (NEEA, No. 35 at p. 3)

ACEEE, ASAP, and NRDC supported the heating load line equation proposal of the August 2016 SNOPR but suggested that a more thorough review and revision of the test method for determining heat pump efficiency should be conducted in future. (ACEEE, NRDC, and ASAP, No. 33 at pp. 2, 7–8)

DOE agrees with BHEC's comment regarding appearance of T_{OD} in the denominator of the heating load line equation. This is a mistake that initially appeared in the November 2015 SNOPR. The correct form of the equation, shown in the initial ORNL Report, indicates that the T_{OD} should be replaced with 5 °F (Docket No. EERE–2009–BT–TP–0004, An Analysis of Representative Heating Load Lines for Residential HSPF Ratings, No. 46 at p. B–1).

Regarding several comments pointing to operation of heating systems in temperatures well above 55 °F, DOE does not dispute that this occurs. The ORNL analysis, in fact, shows that heating loads exist at higher temperatures, as illustrated in Figure 2 of the initial report (Docket No. EERE–2009–BT–TP–0004, An Analysis of Representative Heating Load Lines for Residential HSPF Ratings, No. 46 at p. 5) The zero-load temperature is not intended to be the highest temperature at which the heating system would operate. Instead, it is the zero-load intercept of the best-fit line representing the average loads calculated for each bin. The field data that were provided during the CAC/HP ECS negotiations, cited in the AHRI comment and also provided in the Ingersoll Rand comment (Ingersoll Rand, No. 38 at p. 6), represent many locations and likely represent a wide range of house characteristics and occupancy patterns. DOE does not believe that this type of aggregation of the data of all of the monitored locations is very useful to provide an understanding of building heating loads. For example, much of the operation of the heating systems above 55 °F outdoor temperature could be associated with recovery from night setback. Also, it is not known how the supplemental electric resistance heat compares with the heat pump capacity, or whether any of the locations have supplemental heat other than the electric resistance heat built into the monitored heat pumps—to the extent that such alternative supplemental heating (*e.g.* supplied by a separate space heater, furnace, or wood stove) occurs at different temperatures than heating provided by the heat pump—would affect the results by flattening the apparent load line slope. DOE initially requested additional details of this

[18] The comment indicates that three of the monitored homes are located in Stockton, CA and are in Region ''I/II''. Based on comparison of the location of Stockton with the climate zone map (Figure 1 in Appendix M), it is not clear that ''I/II'' clearly represents Stockton's climate zone—it would appear to be more likely in Region III. In contrast, the other locations mentioned in the comment are much more clearly in their listed zones, *e.g.* V for Southern Vermont, and IV for New York/New Jersey.

study to allow more careful analysis, but such information was not readily available. DOE points out similar issues associated with the aggregation of the field data provided by Lennox. (Lennox, No. 25 at p. 9) In contrast, the data provided by BHEC provides a clearer indication of how the load varies with ambient temperature for specific locations, because the data were provided separately for each location and the heating loads were more directly measured than for the data sets provided by Ingersoll Rand and Lennox.

DOE reviewed the work by BHEC, and believe that, while these data suggest use of a zero-load temperature higher than 55 °F, they do not show that DOE's 55 °F proposal is inappropriate. First, the best-fit zero-load temperatures of the monitored locations ranges from 57 °F to 61 °F. However, the 61 °F value is associated with Site W, which has unusually high loads, suggesting that this location is an outlier not consistent with most houses. Second, as suggested by the comment, these homes are likely less efficient than the 2006 IECC housing characteristics used in the ORNL analysis. During the CAC/HP ECS negotiations, Working Group members commented that, in developing test procedures, DOE should be looking further towards the future than represented by IECC 2006 (see, *e.g.,* Docket EERE–2014–BT–STD–0048, 2015–09–28 Working Group Meeting Transcript; Ingersoll Rand, No. 86 at p. 187; Carrier, No. 85 at p. 112) Hence, DOE believes consideration of house models representative of earlier building codes is not appropriate and maintains its selection of the IECC 2006 building models. DOE notes that there is variation in the existing housing stock and that some houses may have higher zero load slopes than others. Also, when considering all of the locations of the BHEC comment other than Site W, the heating load calculated using the 60 °F zero temperature is slightly higher than the field-correlated line for 5 locations. For these locations, reducing the zero-load temperature to 60 °F would slightly improve the fit of the calculated heating load line to the field data. DOE also considered the impact of a 60 °F zero-load temperature as opposed to the proposed 55 °F zero-load temperature on the differentiation between variable-speed and two-stage products. Using data provided by AHRI during the CAC/HP Working Group meetings, DOE determined that use of 60 °F would make little change to the differences in HSPF values calculated for heat pumps with different characteristics. The HSPF is roughly 2.4 percent higher when using the 60 °F zero-load temperature, and there are no significant difference in trends for products with different characteristics. For all these reasons, DOE has decided not to revise the heating load line using a 60 °F zero-load temperature.

In response to JCI's comment suggesting that the heating loads of the ORNL study did not include the impacts of human occupants, this is not true—the load analysis did include load contributions for human occupants. In response to EEI's comment that the house temperature would fall to 55 °F if the heating system did not operate at warmer temperature, DOE reiterates that the 55 °F zero-load temperature does not imply that there is no heating system operation at warmer temperatures and that the EEI statement ignores the impacts of internal heat loads and solar gain that raise the internal temperature above the exterior temperature even when there is no heating system operation.

Heat Pump and Furnace Load Lines

Ingersoll Rand (p. 6) and EEI (p. 4) commented that the heating load line equation for heat pumps should not be different than the equation used for furnaces in order to maintain neutrality between different heating products in performance information provided to consumers. In response, DOE first notes that neither the capacity nor the steady-state efficiency for furnaces varies significantly for different outdoor air temperatures (see, *e.g.,* Investigation of High Efficiency Furnace SSE Measurements versus AFUE, No. 42 at p. 1), which is not true for capacity and COP of heat pumps. Consequently, the load line does not affect the furnace efficiency metric, AFUE; in other words, the AFUE would not be significantly different if calculated for any of the alternative load lines proposed in the CAC/HP rulemaking notices and discussed in stakeholder comments. In contrast, the capacity of a heat pump varies greatly with ambient temperature. For example, the heating capacity at 7 °F for a single speed heat pump is about 50% of its capacity at 47 °F. (Docket No. EERE–2009–BT–TP–0004, An Analysis of Representative Heating Load Lines for Residential HSPF Ratings, No. 46 at p. 21) The much greater sensitivity to outdoor temperature of a heat pump suggests strongly that use of representative load profiles for calculating seasonal efficiency is much more important for them than for furnaces. DOE has based its proposal and final rule on a recent comprehensive assessment of heating loads, *i.e.* the ORNL analysis. (*Id*) The furnace test procedure has not recently been reviewed from the perspective of a similar assessment of heating loads. DOE acknowledges that the proposed change to the heating load line for heat pumps does change the seasonal heating load that is the basis of the annual operating cost calculation. However, due to the greater importance of using a representative load line for heat pumps, DOE believes that modification of the furnace test procedure to align with the heat pump test procedure is the appropriate resolution. DOE may consider in a future rulemaking whether the seasonal heating load for the furnace test procedure should be adjusted to match that of the heat pump test procedure.

Variable-Speed Slope Factor

Numerous comments addressed the different slope factor proposed for variable-speed products. JCI disagreed with DOE's proposal to modify the heating load line slope such that it varies with technology type. JCI stated they would be willing to adopt a 1.02 slope for all product types as proposed by industry in the CAC/HP ECS negotiations. (JCI, No. 24 at p. 16)

AHRI asserted that a single heating load line equation slope factor is appropriate for all products, because the building load is independent of the installed system. (AHRI, No. 27 at pp. 17–18) Several other commenters made identical arguments. (Goodman, No. 39 at p. 9; Carrier/UTC, No. 36 at p. 10; Lennox, No. 25 at p. 9; Ingersoll-Rand, No. 38 at p. 6; Nortek, No. 22 at p. 12) Rheem commented that different slope factors should not be used for single and two-stage products, further commenting that building load is not determined by the installed HVAC equipment. (Rheem, No. 37 at p. 5) Although DOE has not proposed different slope factors for single and two-stage equipment, DOE understands that the same argument might apply to variable-speed products, for which DOE did propose a different slope factor.

Emerson commented that DOE did not support the different oversizing factor for variable-speed products with any field installation data and noted that the May 2016 workshop on residential CAC/HP installation highlighted field installation inconsistencies including improper sizing and lack of data. Emerson stated that a misrepresentation of HSPF in ''variable capacity'' systems should be corrected by modifying the HSPF calculation, for example, by changing the run time. (Emerson, No. 31 at p. 2) However, Emerson also stated that variable speed allows oversizing in installation and suggested that the variable-speed slope factor also be allowed for use with other technologies that modulate capacity, including two-stage, tandem, vapor injection, and digital. (Emerson, No. 31 at p. 2)

BHEC supported a lower slope factor for variable-speed products than for single-speed, indicating further that the proposal to use the ratio of allowed cooling oversize factors in ACCA Manual S for these types of equipment (leading to a proposed slope factor of 1.07 for Region IV) is reasonable, and in the current test procedure is likely to be a conservative adjustment.

The CA IOUs, NEEA, and ACEEE, NRDC, and ASAP supported the lower slope factor for variable-speed products. (CA IOU, No. 32 at p. 4; NEEA, No. 35 at p. 3; ACEEE, NRDC, and ASAP, No. 33 at pp. 7–8)

In response to comments that the building load does not change with selection of heat pump technology, DOE notes that the proposal does not suggest any difference in building load when using different technology. The slope factor represents the ratio of building load to heat pump capacity. DOE acknowledges that variable-speed products are a bit more oversized in comparison to the building heating load than are single-speed and two-stage products. Keeping the building load constant and increasing the variable-speed heat pump capacity reduces the building load/capacity ratio; hence DOE selected a lower slope factor. Given that publicly available data regarding sizing trends is not available, and in response to comments pointing out the lack of data to support the lower slope factor for variable-speed products, DOE understands that ACCA Manual S is the best available indication of what sizing guidelines contractors and others may be using to select heat pumps, due to widespread citation of the ACCA manuals for use in calculating loads and sizing HVAC systems, including required use of Manual S for sizing of systems in ENERGY STAR certified homes. (''Why ACCA Manual S Means Superior Equipment Sizing'', No. 40; ''HVAC Design Report, ENERGY STAR Certified Homes'', No 43; ''What Exactly is Manual S in HVAC Design and Why Is It Important?'', No. 44; ''Residential Mechanical Equipment Loads and Sizing'', No. 45)

In response to Emerson's comment that potential HSPF misrepresentation for variable-speed products should be addressed by adjusting run time, it is not clear what Emerson's suggested approach is. DOE notes that the lower slope factor for variable-speed products leads directly to calculation of lower percentage run time for variable-speed products in the HSPF calculation when meeting loads lower than the minimum-speed capacity. If Emerson's comment was intended to address the cycle times used for variable-speed products during the cyclic test, DOE notes that the cycle times for variable-speed products are longer for variable-speed than for single-speed or two-stage products (see, *e.g.,* appendix M, section 3.5.b).

In response to Emerson's comment that the test procedure should allow variable-capacity technologies other than variable speed to use the lower slope factor, DOE declines to adopt that approach in this final rule because there were no data either provided by Emerson, or found by DOE that show how such systems would be sized and/or differences in how such systems would operate. For example, two-stage products currently on the market do not allow as wide a range of capacity modulation as do variable-speed products, so it is not clear that similar oversizing is justified for them. In fact, ACCA manual S recommends only slightly more oversizing for two-stage products than for single-stage. The modulation range of vapor-injection compressors is also not as wide as for variable-speed. Finally, DOE is not aware of any CAC/HP products on the market that use digital technology, so it is not clear how the modulation range of future products using this technology will compare, and it is also not clear whether alternative sizing guidelines will be extended to them. DOE is not against consideration of use of the lower slope factor for other variable-capacity technologies, but prefers to consider such a step when more is known about the products using these technologies.

Therefore, DOE is adopting the appendix M1 test procedure with the heating load line equation slope factors (1.15 for single- and two-stage heat pumps and 1.07 for variable-speed heat pumps) and zero-load temperature (55 °F) proposed in the August 2016 SNOPR.

Corrections

In the August 2016 SNOPR, DOE inadvertently included the incorrect values for the representative heating load hours for each generalized climatic region in Table 20 of appendix M1. 81 FR at 58268 (Aug. 24, 2016) The preamble also provided incorrect values for heating load hours, the slope factor, and the variable-speed slope factor for Region III. 81 FR at 58189–90. The corrected values were determined as described and reported in the ORNL report addendum. (CAC TP: ORNL Report Addendum, No. 2 at p. 8) Therefore in this final rule, DOE is adopting the corrected values in the test procedure, including the correct heating load hours for all of the climatic regions in Table 20, which in this notice has become Table 21.

DOE also notes that, in the August 2016 SNOPR, the heating load hours depicted in Figure 1 are not consistent with the new heating load line analysis. 81 FR at 58267 (Aug. 24, 2016) However, the figure is still helpful for depicting the climate zones. Therefore, in this final rule, DOE is renaming Figure 1 to indicate that the figure depicts climate zones rather than heating load hours. In addition, Figure 2, which depicts cooling load hours, is not referenced by any part of the test procedure as modified by the June 2016 final rule and the August 2016 SNOPR proposals. 81 FR at 37119 (June 8, 2016) and 81 FR at 58267 Hence, DOE is removing this figure to reduce potential confusion regarding its applicability to the test procedure and calculations.

Clarification Regarding Negative Heating Loads

DOE's proposed changes to the test procedure did not include removing fractional bin hour data for the temperature bins with temperature higher or equal to the new zero-load temperatures—this included data in Table 19 (number as proposed in the August 2016 SNOPR) for the 62 °F bin for Region I and both the 57 °F and 62 °F bins for all other regions. 81 FR at 58254–55 (Aug. 24, 2016)

DOE notes that for these bins with temperatures higher than the zero-load temperatures, a negative heating load would be calculated according to equation 4.2–1 as proposed. Unico raised this issue in comments submitted in response to the notice of data availability (NODA) associated with the CAC/TP energy conservation standard rulemaking which was published October 27, 2016 (see 81 FR 74727). (Docket Number EERE–2014–BT–STD–0048, Unico, No. 95 at p. 1) However, these negative-load contributions were not intended to be included in HSPF calculation, because they would incorrectly reduce the calculated total seasonal heating load and heating season energy use. In order to exclude the negative-load contributions in the HSPF calculation, DOE has set the fractional bin hours to zero for the 62 °F bin for Region I and both the 57 °F and 62 °F bins for all other regions.

b. Impact of DOE Proposal on Current HSPF Ratings and Model Differentiation

DOE provided in the August 2016 SNOPR a summary of the impacts of the revised heating load line equation proposal on HSPF ratings based on test results provided by AHRI for 2, 3, and 5-ton two-stage and variable-speed heat pumps. 81 FR at 58190 (Aug. 24, 2016) These impacts are reproduced in Table III–5.

TABLE III–5—EFFECT OF REGION IV SLOPE FACTORS ON HSPF OF TWO-STAGE (TS) AND VARIABLE-SPEED (VS) MODELS

	Region IV slope factors				
	Current: 0.77	1.02	1.15	1.30	August 2016 SNOPR *
Avg. TS HSPF	9.49	8.47	8.17	7.80	8.17
Avg. VS HSPF	10.93	9.44	8.95	8.44	9.26
Avg. HSPF Differential	1.44	0.97	0.79	0.64	1.09

* Slope factor for two-stage equipment: 1.15. Slope factor for variable-speed equipment: 1.07.

EEI commented in the public meeting that the change in HSPF associated with the test procedure proposal was so great that there should be consideration of changing the name of the heating mode efficiency metric. (EEI, Public Meeting Transcript, No. 20 at p. 86) PG&E seconded this point. (PG&E, Public Meeting Transcript, No. 20 at p. 87, 88) Other stakeholders mentioned that the working group in the CAC/HP negotiations had settled on calling the new efficiency metric HSPF2 and voiced support for this term—Goodman also indicated that it would be beneficial to use both ''HSPF'' and ''HSPF2'' for a period of time before the new test procedure becomes mandatory, to help consumers understand the differences between the old and new ratings. (Goodman, Rheem, Public Meeting Transcript, No. 20 at p. 87–88)

Consistent with the comments, and as discussed in section III.A.1, DOE is renaming the heating mode efficiency metric ''HSPF2.''

EEI also commented that the new slope has a significant impact on estimated energy usage. EEI commented many two-speed units would not qualify for Energy Star or even meet the minimum DOE HSPF with the new slope. EEI contended that the revision could take many high efficiency units off of the market. (EEI, No. 34 at p. 4) DOE notes that these comments do not take into consideration the changes in the standard levels that would be made to account for the measurement changes. In response, DOE expects that the Energy Star program will set new levels for ''HSPF2'' consistent with the measurement change associated with the test procedure change, as DOE has proposed to do with the new HSPF standard levels selected based on the current test procedure by the CAC/HP ECS Working Group.

No stakeholders stated that the heating load line slope factors proposed in the August 2016 SNOPR result in overly diminished differentiation of variable-speed heat pumps as compared with two-stage heat pumps. Therefore, concerns regarding insufficient product differentiation that had been raised regarding the slope factors proposed in the November 2015 SNOPR appear to be removed, thus strengthening the arguments for heating load line slope factors proposed in the August 2016 SNOPR, which are adopted in this final rule. Thus, DOE is adopting the new heating load line slope factors for variable speed heat pumps in this final rule.

c. Translation of CAC/HP ECS Working Group Recommended HSPF Levels Using Proposed Heating Load Line Equation Changes

Recommendation #9 of the CAC/HP ECS Working Group Term Sheet included two sets of recommended national HSPF standard levels. The Working Group based these levels on heating load line equation slope factors of 1.02 and 1.30 to reflect the two factors primarily discussed during the negotiations. The Working Group designated these levels as ''HSPF2'' to indicate that they are not equivalent to current HSPF ratings. Table III–6 includes the Working Group's recommended HSPF levels:

TABLE III–6—CAC/HP ECS WORKING GROUP RECOMMENDED HSPF LEVELS BASED ON PREVIOUSLY PROPOSED HEATING LOAD LINE EQUATIONS

Product class	HSPF2–1.02	HSPF2–1.30
Split-System Heat Pumps	7.8	7.1
Single-Package Heat Pumps	7.1	6.5

Because the August 2016 SNOPR proposed a heating load line equation with a slope factor of 1.15 for baseline systems, DOE calculated the expected HSPF2 standard levels for this intermediate slope factor—these values are presented in Table III–7.

TABLE III–7—CAC/HP ECS WORKING GROUP RECOMMENDED HSPF LEVELS BASED ON HEATING LOAD LINE EQUATION PROPOSED IN THE AUGUST 2016 SNOPR

Product class	HSPF2–1.15
Split-System Heat Pumps	7.5
Single-Package Heat Pumps	6.8

DOE requested comment on the adjusted values of minimum HSPF2.

During the public meeting, Goodman expressed provisional support of the values but indicated that some analysis would be conducted to confirm. (Goodman, Public Meeting Transcript, No. 20 at pp. 89–90) However, several commenters indicated in written comments that the 6.8 HSPF2 value for single-package heat pumps was too high.

AHRI expressed concern with the HSPF2 value determined for single-package heat pumps, indicating that of six such products with current-test HSPF of 8.0 and slightly higher that were evaluated, the results for five indicate that the crosswalk from HSPF of 8.0 to HSPF2 of 6.8 is not accurate using the 1.15 slope factor. AHRI indicated that it was in the process of collecting additional data and will provide a suggestion for an appropriate crosswalk for this class within 30-days of the comment submittal deadline. (AHRI, No. 27 at p. 18) Nortek submitted a nearly identical comment, but claimed that three of the six evaluated units would not be compliant with the 6.8 HSPF2 level, and indicated that more data would be collected and provided within 30 days. (Nortek, No. 22 at p. 15)

Goodman performed simulation analysis, from which it concluded that the proposed HSPF2 values for split system heat pumps is realistic, but that the crosswalk value for single package heat pumps is higher than it should be. Goodman requested a crosswalk HSPF2 value of 6.6 or 6.7 but indicated they would be providing more information. (Goodman, No. 39 at p. 10)

Rheem commented that, based on initial analysis of the HSPF to HSPF2 crosswalk, some of their products would become obsolete if the cross-walk is adopted—however, they did not clarify which type of product. Rheem commented that it was working with AHRI to determine appropriate cross-walk metrics, which would be reported to DOE. (Rheem No. 37 at p.7)

Ingersoll Rand also expressed concerns about the HSPF to HSPF2 crosswalk, and indicated they would be providing data to AHRI. (Ingersoll RandNo. 38 at p. 7)

JCI commented that residential single-package units will be more severely affected than the crosswalk currently reflected and requested more time for the industry to evaluate and confirm the HSPF to HSPF2 crosswalk. (JCI, No. 24 at p. 17)

ACEEE, NRDC, and ASAP supported the values assigned, commenting that without better information, the linear interpolation is an appropriate way to determine the adjusted minimum HSPF2 values for the heating load line equation slope factor proposed in the August 2015 SNOPR. (ACEEE, NRDC, and ASAP, No. 33 at p.2) Carrier/UTC supported the adjusted values of minimum HSPF2 as they are consistent with the CAC/HP ECS Working Group term sheet recommendation. (Carrier/UTC, No. 36 at p. 11)

Lennox supported the 7.5 HSPF2 value determined by DOE for split systems but did not support the 6.8 HSPF2 value for single package products. Lennox commented that an HSPF2 level of 6.5 would be appropriate for single package heat pumps under the M1 Appendix test procedure proposed in the August 2015 SNOPR. Lennox indicated that it was working to expand the sample of the data used in this determination to provide DOE evidence that supports this recommendation. Lennox expected this data collection to be complete within 30 days of the end of the comment period. (Lennox, No. 25 at p. 10)

Unico requested that the DOE defer action until AHRI presents additional data, since the crosswalk is a complex issue and requires additional time to determine the effect that the proposed adjustments will have on HSPF. (Unico, No. 30 at p.6)

DOE will consider these recommendations and any additional data provided in a timely fashion when it considers the final HSPF2 values to be set for single-package heat pumps in the energy conservation standard rulemaking.

d. Consideration of Inaccuracies Associated With Minimum-Speed Extrapolation for Variable-Speed Heat Pumps

DOE discussed in the November 2015 SNOPR potential inaccuracies associated with the use of test data conducted at minimum speed in 47 °F and 62 °F ambient temperature to estimate heat pump performance below 47 °F. 80 FR at 69322–23 (Nov. 9, 2015). Specifically, for heat pumps that increase compressor speed as ambient temperature drops below 47 °F, the extrapolation of performance based on the 47 °F and 62 °F minimum-speed tests over-estimates efficiency. However, for the 1.3 slope factor proposed in the November 2015 SNOPR, DOE found that the impact on HSPF for the available heat pump data was too small to justify modifying the test procedure. The higher slope factor reduced the impact of the issue because the higher heating load reduced the weighting of the HSPF on minimum-speed performance. DOE did not propose a resolution but indicated that it might reconsider this possibility if a lower heating load line equation slope factor were adopted. Id. In the August 2016 SNOPR, DOE proposed to reduce the heating load line equation slope factor to 1.07 for variable-speed heat pumps. DOE's analysis suggested that, with the lower slope factor, the HSPF may be overestimated by as much as 16 percent as a result of the inaccuracy associated with the minimum-speed extrapolation. Hence, DOE also proposed revision to the estimation of minimum-speed performance to reduce the impact of the error. Specifically, for heat pumps that vary the minimum speed when operating in outdoor temperatures that are in a range for which the minimum-speed performance factors into the HSPF calculation, DOE proposed the following.

• Adoption of a definition, ''minimum-speed-limiting variable-speed heat pump,'' to refer to such heat pumps.

• Minimum-speed performance between 35 °F and 47 °F would be estimated using the intermediate-speed frosting-operation test at 35 °F and the minimum-speed test at 47 °F, and minimum-speed performance below 35 °F would be equal to intermediate-speed performance.

• Including in certification reports for such variable-speed heat pumps whether this alternative approach was used to determine the rating.

81 FR at 58191 (Aug. 24, 2016)

Rheem, Unico, Nortek, Mitsubishi, AHRI, Ingersoll Rand, ACEEE, NRDC, and ASAP, and Lennox supported DOE's proposal to use alternative HSFP rating approach as part of M1. (Rheem, No. 37 at p. 6; Unico, No. 30 at p. 7; Nortek, No. 22 at p. 16; Mitsubishi, No. 29 at p.4; AHRI, No. 27 at p.19; Ingersoll Rand, No. 38 at p. 7; ACEEE, NRDC, and ASAP, No. 33 at p. 8; Lennox, No. 25 at p. 15) Carrier/UTC supported the methodology to account for variable-speed heat pumps that limit the low stage speed at lower ambient conditions by not requiring additional testing. (Carrier/UTC, No. 36 at p. 12) JCI essentially agreed with the proposal, commenting that additional tests would offer minimal improvement in HSPF accuracy, and are not worth the additional test burden. JCI also commented that if DOE adopts this change, it should be in appendix M1 and not in appendix M. (JCI, No. 24 at p. 17)

Carrier also commented that DOE should invest in creating an alternative load based (or some other) test method that simplifies the test procedure and accounts for all of the benefits of variable-speed technology, allowing a true comparison to other technologies. (Carrier/UTC, No. 36 at p. 12)

Goodman did not specifically comment on the proposed test procedure change for variable-speed products, but instead suggested a significantly revised test procedure for these products that would include two tests each at two different outdoor temperatures for each of the relevant compressor speeds (low, intermediate, high, and boost), where boost speed would be optional for testing and would be used for very low temperatures, *e.g.* 17 °F and below. In Goodman's scheme, the manufacturer would determine at which speed the heat pump would be operating for each temperature bin, and would certify (a) the temperature bin at which the variable-speed heat pump begins to increase above minimum speed, (b) the temperature bin at which full speed is achieved, and (c) in which temperature bin the boost speed is achieved. (Goodman, No. 39 at p. 11)

In response to Carrier and Goodman, DOE would support development by the industry and interested stakeholders of a blank-slate revision of the test procedure for variable-speed products with consideration of load-based methods as suggested by Carrier, but since these alternative methods are not fully defined, and certainly have not be made available for public comment, DOE cannot finalize any such test procedure with this final rule.

In this final rule, DOE adopts the proposal for the alternative method for variable-speed heat pumps that raise the compressor speed above the minimum speed at ambient temperatures below 47 °F. In response to JCI, this alternative method was proposed only for appendix M1 and is adopted in this final rule only for appendix M1.

4. Revised Heating Mode Test Procedure for Units Equipped With Variable-Speed Compressors

In the November 2015 SNOPR, DOE revisited the heating season ratings procedure for variable-speed heat pumps found in section 4.2.4 of appendix M of 10 CFR part 430 subpart B. DOE proposed as part of appendix M1 an optional approach for testing variable-speed heat pumps that included a test conducted at 2 °F outdoor temperature (or at the low cutoff temperature, whichever is higher). The proposal would have allowed manufacturers to choose to conduct one additional steady-state test, at maximum compressor speed and at a low temperature of 2 °F or at a low cutoff temperature, whichever is higher. 80 FR at 69322–23 (Nov. 9, 2015).

DOE received comments on this proposal, both in written form in response to the November 2015 SNOPR, and in the CAC/HP ECS negotiations. Working group members ultimately agreed that the optional test should be conducted at 5 °F rather than 2 °F—this is Recommendation #5 in the Term Sheet. (CAC ECS: ASRAC Term Sheet, No. 76 at p. 3)

The revised variable-speed heat pump test procedure proposed in the August 2016 SNOPR included the following changes in appendix M1.

• If the optional 5 °F full-speed test (to be designated $H4_2$) is conducted, full-speed performance for ambient temperatures between 5 °F and 17 °F would be calculated using interpolation between full-speed test measurements conducted at these two temperatures, rather than the current approach, which uses extrapolation of performance measured at 17 °F and 47 °F ambient temperatures. For all heat pumps for which the 5 °F full-speed test is not conducted, the extrapolation approach would still be used to represent performance for all ambient temperatures below 17 °F.
• A target wet bulb temperature of 3.5 °F for the optional 5 °F test.
• If the optional 5 °F full-speed test is conducted, performance for ambient temperatures below 5 °F would be calculated using the same slopes (capacity vs. temperature and power input vs. temperature) as determined for the heat pump between 17 °F and 47 °F. Specifically, the extrapolation would be based on the 17 °F-to-47 °F slope rather than the 5 °F-to-17 °F slope. If the 47 °F full-speed test is conducted at a different speed than the 17 °F full-speed test, the extrapolation would be based on the standardized slope discussed in section III.B.7.
• Manufacturers would have to indicate in certification reports whether the 5 °F full-speed test was conducted.
• As proposed for appendix M and discussed in section III.B.7, a 47 °F full-speed test, designated the $H1_N$ test, would be used to represent the heating capacity. However, for appendix M1, this test would be conducted at the maximum speed at which the system controls would operate the compressor in normal operation in a 47 °F ambient temperature.
• If the heat pump limits the use of the minimum speed (measured in terms of RPM or power input frequency) of the heat pump when operating at ambient temperatures below 47 °F (*i.e.* does not allow use of speeds as low as the minimum speed used at 47 °F for any temperature below 47 °F), a modified calculation would be used to determine minimum-speed performance below 47 °F (this proposal is discussed in section III.C.3.d).

81 FR at 58192–93 (Aug. 24, 2016).

DOE also requested comment regarding whether the 2 °F test for triple-capacity northern heat pumps should be changed to a 5 °F test. 81 FR at 58193. (Aug. 24, 2016).

Carrier/UTC, Lennox, the Joint Advocates, JCI, Ingersoll Rand, Goodman, Nortek, Unico, NEEA, Rheem, CA IOU, AHRI, and Mitsubishi agreed with DOE's proposal to adopt a very low temperature test for heat pumps, at the 5 °F temperature agreed to by the CAC/HP ECS Working Group, rather than the 2 °F initially proposed. (Carrier/UTC, No. 36 at p. 12; Lennox, No. 25 at p. 15; ACEEE, NRDC, and ASAP, No. 33 at p. 8, JCI, No. 24 at p. 17; Ingersoll Rand, No. 38 at p. 7, Goodman, No. 39 at p. 11; Nortek, No. 22 at p. 16; Unico, No. 30 at p. 7; NEEA, No. 35 at p. 3; Rheem, No. 37 at p. 6; CA IOU, No. 32 at p.4; AHRI, No. 27 at p.19; Mitsubishi, No. 29 at p.4) Rheem and the Joint Advocates commented that if the 5 °F full-speed test is conducted, the full-speed performance should be calculated using interpolation, rather than extrapolation. (Rheem, No. 37 at p. 6; ACEEE, NRDC, and ASAP, No. 33 at p. 8)

Goodman further suggested that an optional 5 °F test also be allowed for two-stage and single-speed heat pumps. In addition, Goodman recommended that for all of these products for which the optional 5 °F test is conducted, performance for all ambient conditions below 17 °F be based on the 5 °F and 17 °F tests, using linear interpolation between these temperatures and linear extrapolation below 5 °F, explaining that the potential inaccuracy of the extrapolation below 5 °F is not so important because less than 1% of heating performance for the HSPF in Region IV occurs at temperatures less than 5 °F. Goodman clarified that its support for this approach, including extension to single-speed and two-stage products, is contingent on the 5 °F test being optional. (Goodman, No. 39 at pp. 5–6)

Unico suggested that DOE consider establishing a cold climate heat pump product class with different test methods both for heating and cooling performance and different energy conservation standards for both operating modes in order to incentivize development of such products, claiming that they do not rate well using the current HSPF and SEER metrics because they are optimized for heating in lower ambient temperatures. (Unico, No. 30 at p. 7)

In response to stakeholders' comments, DOE has adopted the optional 5 °F test for variable-speed heat pumps. DOE notes that the Joint Advocate's suggestion to require use of interpolation, rather than extrapolation based on tests conducted in 47 °F and 17 °F temperatures, when the 5 °F test is conducted, is fully consistent with the proposal and is how the test procedure is adopted in this rule.

In response to Goodman's comments, DOE has extended 5 °F testing as an optional test to HSPF rating for single-speed and two-stage heat pumps. For single-speed, two-stage, and variable-speed heat pumps that are tested using the optional 5 °F full-speed test (to be designated $H4_2$), full-speed performance for ambient temperatures between 5 °F and 17 °F will be calculated using interpolation based on full-speed test measurements conducted at these two temperatures, rather than the current approach, which uses extrapolation of performance measured at 17 °F and 47 °F ambient temperatures. Full speed-performance for temperatures lower than 5 °F will be calculated for single-speed and two-stage heat pumps using extrapolation based on the tests conducted at 5 °F and 17 °F, rather than using the 17 °F-to-47 °F slope that was proposed and is adopted for variable-speed heat pumps. DOE considers extrapolation below 5 °F for these products to be acceptable because the 5 °F and 17 °F tests will be conducted at the same compressor speed. For all heat pumps for which the 5 °F full-speed test is not conducted, the extrapolation approach using test results for 17 °F and 47 °F temperatures (or the standardized slope factors for variable-speed heat pumps which do not use the same speed for these tests) would be used to represent performance for all ambient temperatures below 17 °F.

DOE considered Unico's suggestion to create a separate product class with a different test standard and test procedure for products designed for cold climate. However, because other stakeholders have not had the opportunity to comment, DOE cannot adopt that suggestion in this final rule.

In response to DOE's proposal of a target wet bulb temperature of 3.5 °F for the optional 5 °F test, ACEEE, NRDC, and ASAP agreed with the proposed 3.5 °F target wet bulb temperature. (ACEEE, NRDC, and ASAP, No. 33 at p. 8) Carrier/UTC, Lennox, JCI, Ingersoll Rand, Goodman, Nortek, NEEA, Rheem, the CA IOUs, AHRI, and Mitsubishi all recommended that the target wet bulb temperature for the 5 °F test should be 3 °F or less, rather than the proposed 3.5 °F target. The commenters indicated that holding tight tolerances on the wet bulb temperature at such low temperatures is very challenging, but that the frost loading for this temperature is so low that the variation in moisture up to the 3 °F wet bulb temperature level would not affect the test significantly. Unico made a similar recommendation, but suggested a maximum of 4 °F wet bulb temperature. (Carrier/UTC, No. 36 at p. 12; Lennox, No. 25 at p. 15; JCI, No. 24 at p. 17; Ingersoll Rand, No. 38 at p. 7, Goodman, No. 39 at p. 11; Nortek, No. 22 at p. 16; Unico, No. 30 at p. 7; NEEA, No. 35 at p. 3; Rheem, No. 37 at p. 6; CA IOU, No. 32 at p.4; AHRI, No. 27 at p.19; Mitsubishi, No. 29 at p.4). DOE agrees that the amount of moisture in 5 °F air would be sufficiently low that imposing a maximum wet bulb temperature of 3 °F would be adequate to ensure test repeatability; hence, DOE adopts the suggestion to set a maximum level of 3 °F in this final rule.

JCI, Goodman, Unico, UTC, AHRI, ACEEE, NRDC, and ASAP supported testing triple-capacity northern heat pumps at 5 °F to be consistent with other heat pumps. In addition, AHRI suggests that DOE modify the test procedure for triple-capacity northern heat pumps, and allow variable speed heat pumps to be tested like the triple-capacity northern heat pumps in heating mode. Unico also suggested that triple-capacity systems should also be tested at 17 °F at the third (boost) capacity to allow for extrapolation (H33), thus adding a capacity curve at the third capacity. (JCI, No.24, at p 17; Goodman, No. 39, at p 14; Unico, No. 30 at p 7; Carrier/UTC No. 36 at p. 12; AHRI, No. 27 at p. 19; ACEEE, NRDC, and ASAP, No. 33 at p. 8)

In response to those comments, DOE adopts testing of triple-capacity northern heat pumps at 5 °F in both appendix M and appendix M1. DOE considered AHRI's suggestion of modifying the testing of triple-capacity northern heat pumps and allowing testing variable-speed heat pumps using the procedure, and decided not to make the changes in this final rule. More discussion regarding this issue is in section III.B.7. In response to Unico's suggestion on adding a 17 °F test at the 3rd capacity to allow for extrapolation (H33), DOE notes that the current triple-capacity test procedure already requires the requested test.

As discussed in section III.B.7, many stakeholders responded to DOE's proposal of modification to the test procedure for variable-speed heat pumps in appendix M, recommending that the proposed changes, if adopted, should be part of appendix M1 rather than appendix M. In response to these comments, DOE has removed from appendix M the requirement that the $H3_2$ test be conducted at the highest speed that would normally be used in 17 °F ambient conditions—this change is adopted, however, in appendix M1.

D. Effective Dates and Representations

1. Effective Dates

DOE finalized some appendix M requirements in the June 2016 Final Rule, and representations must be made in accordance with appendix M, as adopted in that Final Rule, starting 180 days after it was published (December 6, 2016). DOE proposed additional changes to appendix M in the August 2016 SNOPR, some of which are adopted in this final rule, and representations must be made in accordance with this revised version of appendix M 180 days after this final rule is published. Representations must be made in accordance with the adopted appendix M1 when compliance with amended energy conservation standards is required.

Carrier and Mortex requested that the effective date of appendix M, including the changes published in the June 2016 final rule, be made 180 days from when this rule is finalized. (Carrier/UTC, No. 36 at p. 2; Mortex Products, Inc, No. 26 at p. 2) Ingersoll Rand recommended that all changes to M be made effective at the same time. (Ingersoll Rand, No. 38 at p. 3)

Mortex commented that if that is not possible, then the appendix M changes in the August 2016 SNOPR should be moved to appendix M1. (Mortex Products, Inc, No. 26 at p. 2) AHRI commented similarly. (AHRI, No. 27 at p. 8) JCI also recommended that all of the proposed test procedure changes in the August 2016 SNOPR in appendix M and all updated sections of 10 CFR 429 become effective at the same time that appendix M1 and the corresponding standard revision become effective. (JCI, No. 24 at p. 18) Goodman requested for multiple changes to variable-speed heat pumps be moved from appendix M to appendix M1 and requested that for those changes not moved to appendix M1, DOE exercise its authority under 42 U.S.C. 6293(c)(3) to extend the effective date another 180 days, for a total of 360 days in order to permit manufacturers a more appropriate time period to address the required changes. (Goodman, No. 39 at p. 12)

DOE notes that appendix M, as adopted in the June 2016 Final Rule, is already effective, and that the date by which representations must be in accordance with appendix M, as so adopted, is mandated by statute. (42 U.S.C. 6293(c)(2)) DOE maintains that appendix M revisions adopted in the final rule do not require re-testing as compared with appendix M as adopted in the June 2016 Final Rule (*i.e.,* DOE does not expect the revisions to change the ratings). In certain cases where commenters expressed specific concern, such as for the time delay requirement for off mode power consumption, DOE has moved items to appendix M1. As noted by Goodman, 42 U.S.C. 6293(c)(3) does allow individual manufacturers to request an additional 180 days for representations. This request cannot be made through a rulemaking public comment submission and must be done through petition separately. (42 U.S.C. 6293(c)(3))

2. Comment Period Length

JCI commented that Under Section 323(b)(2) of EPCA, the public's opportunity to comment ''shall be not less than 60 days and may be extended for good cause shown to not more than 270 days.'' 42 U.S.C. 6293(b)(2) JCI commented that given the nature of the proposals in the August 2016 SNOPR, DOE is required to provide a minimum 60-day comment period. JCI commented that test procedure revisions are frequently complex and technical, and Section 323(b)(2) can only reasonably be read to provide a new comment period to ensure that the public has an adequate opportunity for public comment on each discrete test procedure proposal.

In response, DOE notes that this was the fifth round of comments on this particular test procedure rulemaking. Further, DOE made available the pre-publication notice to stakeholders 3 weeks in advance of the actual **Federal Register** publication, effectively allowing for almost a two-month review period. Third, DOE received comments on both sides of the issue both requesting an extension and urging the Secretary to finalize the test procedure as expeditiously as possible. Lastly, there is a statutory maximum comment period for which DOE must be mindful, which DOE was close to reaching. Consequently, DOE did not extend the comment period for the CAC/HP TP SNOPR.

3. Representations From Appendix M1 Before Compliance Date

Lennox recommended that representations in accordance with appendix M1 be permitted 12 months prior to the compliance date of the 2023 amended energy conservation standards. They stated that while there must be a clear differentiation between the current appendix M and new appendix M1 efficiency descriptors associated with the amended standards, permitting representations 12 months prior to adoption helps avoid market disruption on the compliance date. They added that one year allows contractors, distributors and manufacturers adequate time to plan and educate the supply chain in advance of the standard change. (Lennox, No. 25 at p. 2–3) ADP made a similar suggestion, except without setting a time limit on when the representations in accordance with appendix M1 could begin. (ADP, No. 23 at p. 3) Carrier strongly suggested that manufacturers not have any repercussion or penalties from DOE for choosing to comply early with appendix M1. (Carrier/UTC, No. 36 at p. 4)

DOE has guidance in place that allow manufacturers to use the appendix M1 test procedure early as long as they are following the guidelines outlined therein. More information regarding early compliance can be found at: *https://www1.eere.energy.gov/buildings/appliance_standards/pdfs/tp_earlyuse_faq_2014-8-25.pdf*.

E. Comments Regarding the June 2016 Final Rule

1. Determination of Represented Values for Single-Split Systems

In the June 2016 final rule DOE adopted provisions for determining the represented values of single-split system air conditioners based on recommendations from the CAC/HP ECS Working Group. The recommendations from the CAC/HP ECS Working Group (Recommendation #7 of the Term Sheet, see CAC ECS, No. 76 at p. 4) read as follows:

- Every combination distributed in commerce must be rated.
  - Every single-stage and two-stage condensing unit distributed in commerce (other than a condensing unit for a 1-to-1 mini split) must have at least 1 coil-only rating that is representative of the least efficient coil distributed in commerce with a particular condensing unit.
- Every condensing unit distributed in commerce must have at least 1 tested combination.
  - For single-stage and two-stage condensing units (other than condensing units for a 1-to-1 mini split), this must be a coil-only combination.
- All other combinations distributed in commerce for a given condensing unit may be rated based on the application of an AEDM or testing in accordance with the applicable sampling plan.

81 FR at 37002–03 (June 8, 2016)

In the June 2016 final rule, DOE adopted the first and third recommendations. DOE did not relax the HSVC requirement for tested combinations as intended as part of the second recommendation, but did explicitly codify the requirement to test a coil-only combinations for single-stage and two-stage condensing units (including SDHV and space-constrained systems).

AHRI commented that the CAC/HP ECS ASRAC Working Group's recommendations were made in the context of appendix M1, including the proposed requirement for two-stage condensing units (other than condensing units for a 1-to-1 mini split) to be a coil-only combination and have at least one tested combination. AHRI commented that implementing this requirement before the effective date of the 2023 standard would be contradictory to the Working Group's recommendation and that would be an excessive burden on manufacturers to retest products, specifically two-stage air conditioners, in a short period of time. AHRI requested that DOE modify the test procedure so this requirement would be implemented January 1, 2023. Nortek, Carrier/UTC, Lennox, and Ingersoll Rand commented similarly. (AHRI, No. 27, p. 2; Nortek, No. 22 at p. 2–3; Carrier/UTC, No. 36 at p. 2–3; Lennox, No. 25 at p. 3; Ingersoll Rand, No. 38 at p. 1–2)

Additionally, Nortek commented that the requirement that two-speed products be tested with a coil-only combination has the potential to change ratings derived previously using a blower coil or the ARM. Nortek commented that this was part of the consensus agreement of the negotiated rulemaking for the appendix M1 test procedure, and that implementing this in the appendix M test procedure may provide unintended consequences, namely that some high efficiency products may be removed from the market as a result of regional standards. Nortek suggested it would be best to implement this change in tested combination requirements with the appendix M1 test procedure. (Nortek, No. 22 at p. 19–20)

Nortek commented that it did not agree with DOE requiring a coil-only match for two-stage equipment, which they believed should be optional. Nortek commented that to provide the rated efficiency, multiple capacity systems require a matched indoor blower system to provide the correct air-flows at the different stages, and that a blower-coil match is appropriate for these systems. Nortek commented that they do not wish to market a match they believe is inconsistent with providing the rated efficiency. Nortek strongly encouraged DOE to reconsider requiring manufacturers to rate a hypothetical two-stage match that the manufacturer does not intend to market, and that it believes that unintended consequences will occur if they are forced to do so. (Nortek, No. 22 at p. 19–20)

First Co. commented that space-constrained thru-the-wall units are sold and designed for installation with indoor air handlers fitted with ECM motors, meeting the applicable 12 SEER standard when matched with blower coil units. If the ''coil only'' testing requirement is enforced, most of these units will be unable to meet the 12 SEER standard because the default value for wattage in ''coil only'' testing exceeds the actual wattage of the high efficiency motors used in the blower coils with First Co. products. First Co. commented that their understanding is that the Working Group did not include a member that manufactures space-constrained units, but includes members that may benefit from the elimination of these products. (First Co, No. 21 at p. 2–3)

Lennox recommended that DOE further define the requirements for single and two-stage AC systems to test the ''least efficient'' combination and recommended that the ''least efficient'' combination be defined as the up-flow coil match with the lowest NGIFS. Lennox commented that it is common practice for manufacturers to rate several coils of various geometries at the base (*i.e.,* the least efficient level) for that product with the up-flow configuration being the most common, and that requiring a test of the lowest NGIFS up-flow coil clarifies which coil is required as the basis for testing. (Lennox, No. 25 at p. 3)

All of these comments address language adopted in the June 2016 Final Rule and for which no proposals were made in the August 2016 SNOPR. DOE notes that numerous coil-only two-stage combinations have been listed in DOE's CCMS and AHRI's database for years. For example, DOE identified 2,400 such combinations of two-stage split system air conditioners in a version of the database dating to late 2014. DOE also notes that the test procedure has specific provisions for setting air volume rate when testing such units (*i.e.* section 3.1.4.2.c of Appendix M), which correspond to how these units are typically installed in the field. These observations counter claims that multiple capacity systems require a matched indoor blower system and render this assertion false.

In response to First Co.'s comment regarding the required coil-only test for testing of space constrained products, DOE asserts that an exclusion for coil-only testing of space-constrained products was never established. DOE notes that prior to the effective date of the June 2016 final rule, paragraph (a)(2)(ii) of 10 CFR 429.16 still included text that stated that an exclusion for the coil-only test requirement applied for through-the-wall units that were sold and installed with blower coil indoor units. On January 23, 2010, all of the products meeting the definition for the product class of through-the-wall class of split system air conditioners were reclassified as part of the space constrained product class, for which a 12–SEER standard was set for cooling mode and a 7.4 HSPF standard was set for heat pump heating mode in a final rule published August 17, 2004. 69 FR 50997, 51001. Subsequently, the American Energy Manufacturing Technical Corrections Act (AEMTCA), which was signed into law on December 8, 2012, reintroduced definitions of through-the-wall air conditioners and through-the-wall heat pumps, which DOE subsequently codified into its regulations in a final rule published April 11, 2014. As part of that final rule, DOE made clear that products that meet the definition of through-the-wall air conditioners and heat pumps would be considered part of the space constrained air conditioner product class for regulatory purposes, regardless of whether they also met the definition of through-the-wall air conditioner. 79 FR 20091. Thus in DOE's view, First Company's assertion that the coil-only testing requirement did not apply to its through-the-wall products is invalid. Notwithstanding the requirement of all space constrained split system air conditioners that are single stage must be tested as coil-only, First Company explains in their own comment that their space-constrained through-the-wall condensing units are sold and designed for installation with indoor air handlers fitted with ECM motors. However, DOE notes the exclusion previously in 10 CFR 429.16(a)(2)(ii) for units that were sold and installed with blower coil indoor units would not have encompassed the circumstances that First Company describes. Thus, First Company would have always been subject to the coil-only requirement. While the language being adopted in this final rule removes the exclusion for through-the-wall units that were sold and installed with blower coil units from the coil-only testing requirement, this should have no effect on First Company's ratings if rated in accordance with current regulations. If a manufacturer believes that coil-only testing of a product is not appropriate because the basic model is only sold and installed exclusively with blower coil indoor units, the manufacturer may petition DOE for a test procedure waiver showing that installation is exclusively blower coil and requesting a blower coil test. To date, DOE has not received any petitions of this kind.

2. Alternative Efficiency Determination Methods

In the June 2016 Final Rule, DOE adopted alternative efficiency determination method (AEDM) requirements for central air conditioner and heat pumps in place of the previously used alternative rating methods (ARMs). 81 FR at 37054 (June 8, 2016). DOE did not allow the use of AEDMs for multi-split systems. 81 FR at 37052.

First Co. commented that ICMs, including First Co., have used DOE approved Alternative Rating Methods (ARMs) for many years, and converting from using an ARM to an ADEM requires extensive engineering time and laboratory testing. First Co. contends that DOE's claim that it is not requiring ICMs to conduct additional testing for AEDM validation fails to recognize that additional testing beyond certification testing is necessary for ICMs to develop a new AEDM. First Co. commented that compliance by the deadline will be nearly impossible for ICMs that lack their own testing facility and that the extensive time and engineering that ICMs must devote to the meet the new regulations deprives them of the opportunity to innovate or improve existing product lines. (First Co, No. 21 at p. 1)

AHRI commented that the ''tested combination'' requirements for multi-split systems require manufacturers to test at least two samples of a ''tested combination'' for non-ducted indoor units and at least another two samples of a ''tested combination'' for ducted indoor units. AHRI commented that as an AEDM cannot be used to rate a Basic Model, this causes more burden on the multi-split manufacturer than the non-multi-split manufacturer, and is not in line with the fact that other products can have two samples of a single tested combination tested with unlimited number of non-tested combinations rated by AEDM. AHRI commented that performing all required tests in six months is not achievable by some manufacturers. AHRI requested that DOE reconsider the option to apply the AEDM for multi-splits <65,000 Btu/h in the same manner as applied for VRFs ≥65,000 Btu/h. (AHRI, No. 27 at p. 20)

All of these comments address language adopted in the June 2016 Final Rule and for which no proposals were made in the August 2016 SNOPR. As a result, DOE is declining to modify these requirements in this final rule.

3. NGIFS Limit for Outdoor Unit With No Match

In the June 2016 Final Rule, DOE adopted the required NGIFS for an indoor unit tested with an outdoor unit with no match to be 1.0. 81 FR at 37009–10 (June 8, 2016)

Nortek and AHRI commented that the NGIFS limitation of 1.0 as finalized in the June 2016 Final Rule is only applicable to coils with ⅜-inch diameter tubes and is not applicable to either microchannel, 5⁄16″, or 7mm diameter tubes, or any other diameter tubes. (Nortek, No. 22 at p. 5–6; AHRI, No. 27 at p. 6)

DOE responds that the vast majority of indoor units that are field-matched with no-match outdoor units have ⅜-in OD tubing, which was used almost exclusively for CAC/HP evaporators before 2010. Further, as stated previously, this requirement was not part of the August 2016 SNOPR, and as such, DOE cannot modify this requirement in this final rule. Section III.A.5.f addresses concerns about the applicability of the requirements (such as for tube styles) of indoor units to be tested with no-match outdoor units.

4. Definitions

In the June 2016 Final Rule, DOE adopted definitions for multi-split system. 81 FR at 37059 (June 8, 2016).

Mitsubishi, AHRI and Nortek commented that DOE had previously agreed to remove coil-only from the multi-split definition. (Mitsubishi, No. 29 at p. 5; AHRI, No. 27 at p. 22; Nortek, No. 22 at p. 19) Mortex commented that there will be applications for coil-only indoor units and thus there is no reason to remove coil-only from the proposed definition. (EERE–2016–BT–TP–0029, No. 26 at p. 3) As stated previously, this requirement was not part of the August 2016 SNOPR, and as such, DOE cannot modify this requirement in this final rule. Additionally, DOE agrees with Mortex that it is a possible application that coil-only indoor units are used in a multi-split system, so keeping coil-only in the multi-split definition is reasonable and there is no need to modify the definition.

5. Inlet Plenum Setup

In the June 2016 Final Rule, DOE clarified the indoor unit air inlet geometry and specifically made revision to avoid inlet plenum being installed upstream of the airflow prevention device. 81 FR at 37037 (June 8, 2016).

AHRI and Nortek commented that DOE's clarification of inlet plenum brings concern that an overall height will exceed the current height limit of many psychrometric rooms. AHRI and Nortek requested DOE to consider allowing an alternative approach, included in ASHRAE's research project 1581. Specifically, AHRI and Nortek requested that DOE approve the use of the 6″ skirt coupled with the 90° square vane elbow and the appropriate leaving duct as being an alternative to the configuration. ASHRAE Standards Policy Committee (SPC) is currently working to add the details of RP 1581 to the standard and has a Work Statement for a project investigating the damper box/inlet duct to provide an improved recommendation for that as well. (AHRI, No. 27 at p. 21; Nortek, No. 22 at p. 17–18)

As stated previously, this requirement was not part of the August 2016 SNOPR, and as such, DOE cannot modify this requirement in this final rule. However, DOE is willing to consider this change in a future rulemaking after ASHRAE Standards Policy Committee has published standard revision to reflect this recommendation.

6. Off-Mode Power Consumption

In the June 2016 Final Rule, DOE adopted the off-mode test procedure and the method of calculation. In addition, DOE required that the calculated P1 and P2 should be rounded to the nearest watt. 81 FR at 37095–97 (June 8, 2016).

AHRI and Nortek commented that the accuracy of 0.5% for all watt-hour measurement in section 2.8 is not feasible for off-mode power measurement because it can be very close to zero. So AHRI suggested that the accuracy requirement in section 2.8 be 0.5% or 0.5 W, whichever is greater. (AHRI, No. 27 at p. 22; Nortek, No. 22 at p. 18) Ingersoll Rand recommended that the accuracy for the off mode power consumption measurement be 0.5 watts. (Ingersoll Rand, No. 38 at p. 5)

As stated previously, this requirement was not part of the August 2016 SNOPR, and as such, DOE cannot modify this requirement in this final rule. Mitsubishi expressed concern that multi-split systems were not fully considered in the development of off-mode tests, and requested that DOE review the off-mode power requirements to ensure that multi-split systems are not inadvertently disadvantaged. (Mitsubishi, No. 29 at p. 5)

Although DOE cannot modify this requirement in this final rule, DOE has reviewed the off-mode requirements and believes that multi-split systems should follow the same procedure—thus no change to the test procedure to specifically address multi-split systems is needed. DOE understands that the off-mode testing for multi-split system may be more complicated, but manufacturers have the option to develop an AEDM for most off-mode ratings if additional test requirements are necessary.

IV. Procedural Issues and Regulatory Review

A. Review Under Executive Order 12866

The Office of Management and Budget (OMB) has determined that test procedure rulemakings do not constitute ''significant regulatory actions'' under section 3(f) of Executive Order 12866, Regulatory Planning and Review, 58 FR 51735 (Oct. 4, 1993). Accordingly, this action was not subject to review under the Executive Order by the Office of Information and Regulatory Affairs (OIRA) in the Office of Management and Budget.

B. Review Under the Regulatory Flexibility Act

The Regulatory Flexibility Act (5 U.S.C. 601 *et seq.*) requires preparation of a final regulatory flexibility analysis (FRFA) for any final rule, unless the agency certifies that the rule, if promulgated, will not have a significant economic impact on a substantial number of small entities. A required by Executive Order 13272, ''Proper Consideration of Small Entities in Agency Rulemaking,'' 67 FR 53461 (August 16, 2002), DOE published procedures and policies on February 19, 2003, to ensure that the potential impacts of its rules on small entities are properly considered during the DOE rulemaking process. 68 FR 7990. DOE has made its procedures and policies available on the Office of the General Counsel's Web site: *http://energy.gov/gc/office-general-counsel*.

DOE reviewed this final rule under the provisions of the Regulatory Flexibility Act and the procedures and policies published on February 19, 2003. This final rule establishes two sets of test procedure changes: One set of changes to DOE's already-existing test procedure, appendix M; and another set of changes to create a new appendix M1 that would be used for testing to demonstrate compliance with any amended energy conservation standards. DOE has estimated the impacts of both sets of test procedure changes on small business manufacturers.

1. Description and Estimate of the Number of Small Entities Affected

For the purpose of the regulatory flexibility analysis for this final rule, DOE adopts the Small Business Administration (SBA) definition of a small entity within this industry as a manufacturing enterprise with 1,250 employees or fewer. DOE used the SBA's size standards to determine whether any small entities would be required to comply with the rule. The size standards are codified at 13 CFR part 121. The standards are listed by North American Industry Classification System (NAICS) code and industry description are available at: *https://www.sba.gov/sites/default/files/files/Size_Standards_Table.pdf*. CAC/HP manufacturers are classified under NAICS 333415, ''Air Conditioning and Warm Air Heating Equipment and Commercial and Industrial Refrigeration Equipment Manufacturing.'' 70 FR 12395 (March 11, 2005)

To estimate the number of small business manufacturers of equipment affected by this rulemaking, DOE conducted a market survey using available public information. DOE's research involved industry trade association membership directories (including AHRI), individual company Web sites, and market research tools (*e.g.,* Hoovers reports) to create a list of companies that manufacture products applicable to this rulemaking. DOE presented its list to manufacturers in MIA interviews and asked industry representatives if they were aware of any other small manufacturers during manufacturer interviews and ASRAC Working Group meetings. DOE reviewed publicly-available data and contacted companies on its list, as necessary, to determine whether they met the SBA's definition of a small business manufacturer. DOE screened out companies that do not offer products applicable to this rulemaking, do not meet the definition of a small business, or are foreign-owned and operated.

DOE identified 22 manufacturers of residential central air conditioners and heat pumps that would be considered domestic small businesses with a total of less than 3 percent of the market sales.

2. Discussion of Testing Burden and Comments

a. Testing Burdens

Potential impacts of the amended test procedure on all manufacturers, including small businesses, come from impacts associated with the cost of additional testing. DOE expects that many of the provisions in this notice will result in no increase to test burden. DOE's mandate to use new heating load line equation provisions to calculate HSPF for heat pumps, new default values for indoor fan power consumption, and a new interpolation approach for COP of variable-speed heat pumps are changes to calculations and do not require any additional time or investment from manufacturers. Similarly, DOE's mandate to require certification of the time delay used when testing coil-only units does not affect testing. DOE's mandate to test at new minimum external static pressure conditions would require manufacturers to test at different, but not additional test points using the same equipment and methodologies required by the current test procedure. DOE's mandate for single-package units to make the official test the test that does not include the secondary outdoor air enthalpy method measurement also does not require any additional testing. Similarly, DOE's mandate to include an optional test at 5 °F for variable-speed heat pumps does not require manufacturers to do any additional testing. However, other provisions may increase test burden. DOE anticipates that changes to provisions for mini-split refrigerant pressure lines may cause labs and manufacturers to relocate pressure transducers or in a worst case scenario, build a separate satellite test instrumentation console for pressure measurements closer to the test samples. DOE estimates that building such a satellite console would constitute a one-time cost on the order of $1,000 per test room. DOE's mandate to modify the off mode test for units with self-regulated crankcase heaters could result in more significant increases to test burden, but for a small number of models. DOE estimates that the new provisions could add 8 hours per test for units with self-regulated crankcase heaters and an additional 8 hours for those units with self-regulated crankcase heaters that also have a compressor sound blanket. Sound blankets are premium features. DOE estimates that less than 25 percent of all units have self-regulated crankcase heaters and less than 5 percent have self-regulated crankcase heaters and sound blankets. DOE estimates the additional cost of testing to be $250 for units with self-regulating crankcase heaters and $500 for units with self-regulating crankcase heaters and sound blankets. DOE also estimates that testing of basic models may not have to be updated more than once every five years, and therefore the average incremental burden of testing one basic model may be one-fifth of these values when the cost is spread over several years.

DOE mandates labeling requirements for the indoor and outdoor units of mobile home blower coil and coil-only systems and is also requiring that manufacturers include a specific designation in the installation instructions for these units. DOE estimates the additional cost to manufacturers associated with meeting the labeling requirement to be marginal as compared to the total production cost and the overall impact to be small.

As discussed in this preamble, DOE identified 22 domestic small business manufacturers of residential central air conditioners and heat pumps. Of these, only OUMs that operate their own manufacturing facilities (*i.e.,* are not private labelers selling only models manufactured by other entities) and OUM importing private labelers would be subject to the additional requirements for testing required by this proposed rule. DOE identified 12 such small businesses but was able to estimate the number of basic models associated only with nine of these.

DOE requires that only one combination associated with any given outdoor unit be laboratory tested. 10 CFR 429.16(b). The majority of residential central air conditioners and heat pumps offered by a manufacturer are split-system combinations that are not required to be laboratory tested but can be certified using an AEDM that does not require DOE testing of these units. DOE reviewed available data for the nine small businesses to estimate the incremental testing cost burden those firms might experience due to the revised test procedure. These manufacturers had an average of 35 models requiring testing. DOE determined the numbers of models using the AHRI Directory of Certified Product Performance, *www.ahridirectory.org/ahridirectory/pages/home.aspx*. As discussed, DOE estimates that less than 25 percent of models have self-regulating crankcase heaters and less than 5 percent have self-regulating crankcase heaters with blankets. Applying these estimates to the average 35 models for each small manufacturer results in an estimated two models with $500 per model in additional test costs and nine models with $250 per model in additional test costs as a result of the proposed changes. The additional testing cost for final certification of these models was therefore estimated at $3,250. Meanwhile, these certifications would be expected to last the residential central air conditioner and heat pump life, estimated to be at least five years based on the time frame established in EPCA for DOE review of central air conditioner efficiency standards. Hence, average annual additional costs for these small business manufacturers to perform the tests is $650.

DOE does not expect ICMs to incur any additional burden as a result of the amended changes because the changes for which DOE estimates there will be increased burden do not apply to ICMs. Only outdoor units include self-regulating crankcase heaters with or without blankets, and DOE assumes that ICM manufacturers do not produce indoor units that have components with off mode power consumption. Consequently, ICMs would be able to use the off mode power measurements acquired and certified by OUMs to meet the test procedure requirements for off mode. Regarding the changes for mini-split refrigerant lines, DOE is not aware of any ICMs that maintain in-house test facilities. Consequently, the one-time cost associated with the amended changes for mini-split refrigerant lines would not be incurred by the ICM. DOE also anticipates that the one-time cost is low enough that the per-test cost charged by independent labs that provide testing services to ICMs would not increase as a result of this change.

b. Comments on the SNOPR Regulatory Flexibility Analysis

Manufacturers commented that DOE's analysis does not accurately address the negative impacts of M and M1 test procedure changes that small manufacturers and ICMs may face. Particularly, Advanced Distributor Products (ADP) noted that DOE's small business impacts focused solely on the cost of these test procedure changes and do not take cumulative regulatory burden into consideration. A few manufacturers stated that residential central air conditioner and heat pump regulations threaten their ability to compete in the market, which in turn will reduce competition and consumer choices. According to ADP, these negative impacts are primarily due to the requirement to report data that ICMs do not possess. (ADP, No. 23 at p. 6) Mortex attributes these negative impacts to cumulative regulatory burden. (Mortex, No. 26 at p. 4) First Co. cites excessive testing and unreasonable deadlines as drivers of disproportionate impacts that may reduce competition. First Co. attributes these negative impacts to the provisions finalized in the June 2016 test procedure final rule. (First Co., No. 21 at p. 5)

DOE acknowledges the commenters' concerns that manufacturers may face cumulative regulatory burdens and disproportionate impacts. As discussed throughout this notice, DOE recognizes ADP's concern related to data reporting for ICMs and will address these issues through a separate process. Regarding Mortex's concerns with cumulative regulatory burden, DOE conducts an analysis of cumulative regulatory burden as part of the concurrent energy conservation standards rulemaking. Regardless of the findings of that analysis, DOE concludes with this FRFA that the burdens associated only with this rulemaking are not significant. DOE also understands that not all manufacturers have equal access to the resources needed to meet with the requirements of this final rule. EPCA does allow individual manufacturers to request an additional 180 days for representations—such a request cannot be made through a rulemaking public comment period submission and must be done through petition. (42 U.S.C 6293(c)(3)) The majority of the factors cited by First Co. as contributing to threats to their ability to compete are provisions adopted in the June 2016 Final Rule and for which no proposals were made in the August 2016 SNOPR. As a result, DOE cannot modify these requirements in this final rule.

First Co. noted that the ASRAC Working Group did not include a manufacturer of space-constrained products, but rather included manufacturers that may benefit from the elimination of these products from the market. Prior to adopting the Working Group recommendations, First Co. said that DOE should have sought public comments on this matter. (First Co., No. 21 at p. 2) Additionally, Unico commented that small entities typically offer niche products, such as space-constrained and small duct high velocity products, that larger companies do not manufacture. Unico believes small entities, like itself, will be disproportionately impacted by this final rule because, for SDHV, half the system is duct work which is not tested as part of the equipment. Consequently, comparing the real-life performance of small duct systems with other systems is difficult. (Unico, No. 30 at p. 7)

In response, DOE acknowledges First Co.'s concerns regarding the lack of representation of space-constrained manufacturers in the Working Group. During the NOPR stage, DOE identified four manufacturers of space-constrained units. Of the four, two are AHRI members. Although these manufacturers were not present at Working Group meetings, AHRI served as a Working Group member. DOE assumes that AHRI represented all of their members' interests throughout the negotiations. During the NODA phase of the rulemaking, DOE invited space-constrained manufacturers to participate in interviews but none were conducted.

In regards to Unico's comment, many of the CAC/HP products subject to this test procedure are installed and used with duct work. The test procedure does not include duct work for these products either. Instead, the test conditions for this procedure include provisions for minimum external static pressure, which is intended to mimic the operating conditions consistent with field duct work for each product. These minimum external static pressure requirements differ by product because not all CAC/HP are installed with the same duct work. These differing external static pressure requirements ensure that test results are representative of field conditions and can provide reasonable comparisons of performance.

Based on its research and discussions presented in this section, DOE concludes that the cost burdens accruing from the residential central air conditioner and heat pump test procedure final rule will not constitute ''significant economic impact on a substantial number of small entities.''

C. Review Under the Paperwork Reduction Act of 1995

Manufacturers of central air conditioners and heat pumps must certify to DOE that their products comply with any applicable energy conservation standards. In certifying compliance, manufacturers must test their products according to the DOE test procedures for central air conditioners and heat pumps, including any amendments adopted for those test procedures. DOE has established regulations for the certification and recordkeeping requirements for all covered consumer products and commercial equipment, including central air conditioners and heat pumps. 76 FR 12422 (March 7, 2011); 80 FR 5099 (Jan. 30, 2015). The collection-of-information requirement for the certification and recordkeeping is subject to review and approval by OMB under the Paperwork Reduction Act (PRA). This requirement has been approved by OMB under OMB control number 1910–1400. Public reporting burden for the certification is estimated to average 30 hours per response, including the time for reviewing instructions, searching existing data sources, gathering and maintaining the data needed, and completing and reviewing the collection of information.

Notwithstanding any other provision of the law, no person is required to respond to, nor shall any person be subject to a penalty for failure to comply with, a collection of information subject to the requirements of the PRA, unless that collection of information displays a currently valid OMB Control Number.

D. Review Under the National Environmental Policy Act of 1969

In this final rule, DOE amends its test procedure amendments that it expects will be used to develop and implement future energy conservation standards for central air conditioners and heat pumps. DOE has determined that this rule falls into a class of actions that are categorically excluded from review under the National Environmental Policy Act of 1969 (42 U.S.C. 4321 *et seq.*) and DOE's implementing regulations at 10 CFR part 1021. Specifically, this final rule amends the existing test procedures without affecting the amount, quality or distribution of energy usage, and, therefore, will not result in any environmental impacts. Thus, this rulemaking is covered by Categorical Exclusion A5 under 10 CFR part 1021, subpart D, which applies to any rulemaking that interprets or amends an existing rule without changing the environmental effect of that rule. Accordingly, neither an environmental assessment nor an environmental impact statement is required.

DOE's CX determination for this final rule is available at *http://energy.gov/nepa/categorical-exclusion-cx-determinations-cx.*

E. Review Under Executive Order 13132

Executive Order 13132, ''Federalism,'' 64 FR 43255 (August 4, 1999) imposes certain requirements on agencies formulating and implementing policies or regulations that preempt State law or that have Federalism implications. The Executive Order requires agencies to examine the constitutional and statutory authority supporting any action that would limit the policymaking discretion of the States and to carefully assess the necessity for such actions. The Executive Order also requires agencies to have an accountable process to ensure meaningful and timely input by State and local officials in the development of regulatory policies that have Federalism implications. On March 14, 2000, DOE published a statement of policy describing the intergovernmental consultation process it will follow in the development of such regulations. 65 FR 13735. DOE has examined this final rule and has determined that it would not have a substantial direct effect on the States, on the relationship between the national government and the States, or on the distribution of power and responsibilities among the various levels of government. EPCA governs and prescribes Federal preemption of State regulations as to energy conservation for the products that are the subject of this final rule. States can petition DOE for exemption from such preemption to the extent, and based on criteria, set forth in EPCA. (42 U.S.C. 6297(d)) No further action is required by Executive Order 13132.

F. Review Under Executive Order 12988

Regarding the review of existing regulations and the promulgation of new regulations, section 3(a) of Executive Order 12988, ''Civil Justice Reform,'' 61 FR 4729 (Feb. 7, 1996), imposes on Federal agencies the general duty to adhere to the following requirements: (1) Eliminate drafting errors and ambiguity; (2) write regulations to minimize litigation; (3) provide a clear legal standard for affected conduct rather than a general standard; and (4) promote simplification and burden reduction. Section 3(b) of Executive Order 12988 specifically requires that Executive agencies make every reasonable effort to ensure that the regulation: (1) Clearly specifies the preemptive effect, if any; (2) clearly specifies any effect on existing Federal law or regulation; (3) provides a clear legal standard for affected conduct while promoting simplification and burden reduction; (4) specifies the retroactive effect, if any; (5) adequately defines key terms; and (6) addresses other important issues affecting clarity and general draftsmanship under any guidelines issued by the Attorney General. Section 3(c) of Executive Order 12988 requires Executive agencies to review regulations in light of applicable standards in sections 3(a) and 3(b) to determine whether they are met or it is unreasonable to meet one or more of them. DOE has completed the required review and determined that, to the extent permitted by law, this final rule meets the relevant standards of Executive Order 12988.

G. Review Under the Unfunded Mandates Reform Act of 1995

Title II of the Unfunded Mandates Reform Act of 1995 (UMRA) requires each Federal agency to assess the effects of Federal regulatory actions on State, local, and Tribal governments and the private sector. Public Law 104–4, sec. 201 (codified at 2 U.S.C. 1531). For a regulatory action likely to result in a rule that may cause the expenditure by State, local, and Tribal governments, in the aggregate, or by the private sector of $100 million or more in any one year (adjusted annually for inflation), section 202 of UMRA requires a Federal agency to publish a written statement that estimates the resulting costs, benefits, and other effects on the national economy. (2 U.S.C. 1532(a), (b)) The UMRA also requires a Federal agency to develop an effective process to permit timely input by elected officers of State, local, and Tribal governments on a proposed ''significant intergovernmental mandate,'' and requires an agency plan for giving notice and opportunity for timely input to potentially affected small governments before establishing any requirements that might significantly or uniquely affect small governments. On March 18, 1997, DOE published a statement of policy on its process for intergovernmental consultation under UMRA. 62 FR 12820; also available at *http://energy.gov/gc/office-general-counsel*. DOE examined this final rule according to UMRA and its statement of policy and determined that the rule contains neither an intergovernmental mandate, nor a mandate that may result in the expenditure of $100 million or more in any year, so these requirements do not apply.

H. Review Under the Treasury and General Government Appropriations Act, 1999

Section 654 of the Treasury and General Government Appropriations Act, 1999 (Public Law 105–277) requires Federal agencies to issue a Family Policymaking Assessment for any rule that may affect family well-being. This final rule will not have any impact on the autonomy or integrity of the family as an institution. Accordingly, DOE has concluded that it is not necessary to prepare a Family Policymaking Assessment.

I. Review Under Executive Order 12630

DOE has determined, under Executive Order 12630, ''Governmental Actions and Interference with Constitutionally Protected Property Rights'' 53 FR 8859 (March 18, 1988), that this regulation will not result in any takings that might require compensation under the Fifth Amendment to the U.S. Constitution.

J. Review Under Treasury and General Government Appropriations Act, 2001

Section 515 of the Treasury and General Government Appropriations Act, 2001 (44 U.S.C. 3516 note) provides for agencies to review most disseminations of information to the public under guidelines established by each agency pursuant to general guidelines issued by OMB. OMB's guidelines were published at 67 FR 8452 (Feb. 22, 2002), and DOE's guidelines were published at 67 FR 62446 (Oct. 7, 2002). DOE has reviewed this final rule under the OMB and DOE guidelines and has concluded that it is consistent with applicable policies in those guidelines.

K. Review Under Executive Order 13211

Executive Order 13211, ''Actions Concerning Regulations That Significantly Affect Energy Supply, Distribution, or Use,'' 66 FR 28355 (May 22, 2001), requires Federal agencies to prepare and submit to OMB, a Statement of Energy Effects for any significant energy action. A ''significant energy action'' is defined as any action by an agency that promulgated or is expected to lead to promulgation of a final rule, and that: (1) Is a significant regulatory action under Executive Order 12866, or any successor order; and (2) is likely to have a significant adverse effect on the supply, distribution, or use of energy; or (3) is designated by the Administrator of OIRA as a significant energy action. For any significant energy action, the agency must give a detailed statement of any adverse effects on energy supply, distribution, or use should the proposal be implemented, and of reasonable alternatives to the action and their expected benefits on energy supply, distribution, and use.

The regulatory action to amend the test procedure for measuring the energy efficiency of central air conditioners and heat pumps is not a significant regulatory action under Executive Order 12866. Moreover, it will not have a significant adverse effect on the supply, distribution, or use of energy, nor has it been designated as a significant energy action by the Administrator of OIRA. Therefore, it is not a significant energy action, and, accordingly, DOE has not prepared a Statement of Energy Effects.

L. Review Under Section 32 of the Federal Energy Administration Act of 1974

Under section 301 of the Department of Energy Organization Act (Pub. L. 95–91; 42 U.S.C. 7101), DOE must comply with section 32 of the Federal Energy Administration Act of 1974, as amended by the Federal Energy Administration Authorization Act of 1977. (15 U.S.C. 788; FEAA) Section 32 essentially provides in relevant part that, where a proposed rule authorizes or requires use of commercial standards, the notice of proposed rulemaking must inform the public of the use and background of such standards. In addition, section 32(c) requires DOE to consult with the Attorney General and the Chairman of the Federal Trade Commission (FTC) concerning the impact of the commercial or industry standards on competition.

The rule incorporates testing methods contained in the following commercial standards: AHRI 210/240–2008 with Addendum 1 and 2, Performance Rating of Unitary Air Conditioning & Air-Source Heat Pump Equipment; and ANSI/AHRI 1230–2010 with Addendum 2, Performance Rating of Variable Refrigerant Flow Multi-Split Air Conditioning and Heat Pump Equipment. While the proposed test procedure is not exclusively based on AHRI 210/240–2008 or ANSI/AHRI 1230–2010, one component of the test procedure, namely test setup requirements, adopts language from AHRI 210/240–2008 without amendment; and another component of the test procedure, namely test setup and test performance requirements for multi-split systems, adopts language from ANSI/AHRI 1230–2010 without amendment. DOE has evaluated these standards and consulted with the Attorney General and the Chairman of the FTC and has concluded that this final rule fully complies with the requirement of section 32(b) of the FEAA.

M. Description of Materials Incorporated by Reference

In this final rule, DOE incorporates by reference (IBR) into appendix M1 to subpart B of part 430 specific sections, figures, and tables of several test standards published by AHRI, ASHRAE, and AMCA that are already incorporated by reference into appendix M to subpart B of part 430: ANSI/AHRI 210/240–2008 with Addenda 1 and 2, titled ''Performance Rating of Unitary Air-Conditioning & Air-Source Heat Pump Equipment;'' ANSI/AHRI 1230–2010 with Addendum 2, titled ''Performance Rating of Variable Refrigerant Flow (VRF) Multi-Split Air-Conditioning and Heat Pump Equipment;'' ASHRAE 23.1–2010, titled ''Methods of Testing for Rating the Performance of Positive Displacement Refrigerant Compressors and Condensing Units that Operate at Subcritical Temperatures of the Refrigerant;'' ASHRAE Standard 37–2009, titled ''Methods of Testing for Rating Electrically Driven Unitary Air-Conditioning and Heat Pump Equipment;'' ASHRAE 41.1–2013, titled ''Standard Method for Temperature Measurement;'' ASHRAE 41.2–1987 (RA 1992), titled ''Standard Methods for Laboratory Airflow Measurement;'' ASHRAE 41.6–2014, titled ''Standard Method for Humidity Measurement;'' ASHRAE 41.9–2011, titled ''Standard Methods for Volatile-Refrigerant Mass Flow Measurements Using Calorimeters;'' ASHRAE 116–2010, titled ''Methods of Testing for Rating Seasonal Efficiency of Unitary Air Conditioners and Heat Pumps;'' and AMCA 210–2007, titled ''Laboratory Methods of Testing Fans for Certified Aerodynamic Performance Rating.''

ANSI/AHRI 210/240–2008 is an industry accepted test procedure that measures the cooling and heating performance of central air conditioners and heat pumps and is applicable to products sold in North America. The test procedure in this final rule references various sections of ANSI/AHRI 210/240–2008 that address test setup, test conditions, and rating requirements. ANSI/AHRI 210/240–2008 is readily available on AHRI's Web site at *http://www.ahrinet.org/site/686/Standards/HVACR-Industry-Standards/Search-Standards*.

ANSI/AHRI 1230–2010 is an industry accepted test procedure that measures the cooling and heating performance of variable refrigerant flow (VRF) multi-split air conditioners and heat pumps and is applicable to products sold in North America. The test procedure in this final rule for VRF multi-split systems references various sections of ANSI/AHRI 1230–2010 that address test setup, test conditions, and rating requirements. ANSI/AHRI 1230–2010 is readily available on AHRI's Web site at *http://www.ahrinet.org/site/686/Standards/HVACR-Industry-Standards/Search-Standards*.

ASHRAE 23.1–2010 is an industry accepted test procedure for rating the thermodynamic performance of positive displacement refrigerant compressors and condensing units that operate at subcritical temperatures. The test procedure in this final rule references sections of ASHRAE 23.1–2010 that address requirements, instruments, methods of testing, and testing procedure specific to compressor calibration. ASHRAE 23.1–2010 can be purchased from ASHRAE's Web site at *https://www.ashrae.org/resources-publications*.

ASHRAE Standard 37–2009 is an industry accepted standard that provides test methods for determining the cooling capacity of unitary air conditioning equipment and the cooling or heating capacities, or both, of unitary heat pump equipment. The test procedure in this final rule references various sections of ASHRAE Standard 37–2009 that address test conditions and test procedures. ASHRAE Standard 37–2009 can be purchased from ASHRAE's Web site at *https://www.ashrae.org/resources-publications*.

ASHRAE 41.1–2013 is an industry accepted method for measuring temperature in testing heating, refrigerating, and air conditioning equipment. The test procedure in this final rule references sections of ASHRAE 41.1–2013 that address requirements, instruments, and methods for measuring temperature. ASHRAE 41.1–2013 can be purchased from ASHRAE's Web site at *https://www.ashrae.org/resources-publications.*

ASHRAE 41.2–1987 (RA 1992) is an industry accepted test method for measuring airflow. The test procedure in this final rule references sections of ASHRAE 41.2–1987 (RA 1992) that address test setup and test methods. ASHRAE 41.2–1987 (RA 1992) can be purchased from ASHRAE's Web site at *https://www.ashrae.org/resources-publications.*

ASHRAE 41.6–2014 is an industry accepted test method for measuring humidity of moist air. The test procedure in this final rule references sections of ASHRAE 41.6–2014 that address requirements, instruments, and methods for measuring humidity. ASHRAE 41.6–2014 can be purchased from ASHRAE's Web site at *https://www.ashrae.org/resources-publications.*

ASHRAE 41.9–2011 is an industry accepted standard that provides recommended practices for measuring the mass flow rate of volatile refrigerants using calorimeters. The test procedure in this final rule references sections of ASHRAE 41.9–2011 that address requirements, instruments, and methods for measuring refrigerant flow during compressor calibration. ASHRAE 41.9–2011 can be purchased from ASHRAE's Web site at *https://www.ashrae.org/resources-publications.*

ANSI/ASHRAE Standard 116–2010 is an industry accepted standard that provides test methods and calculation procedures for determining the capacities and cooling seasonal efficiency ratios for unitary air-conditioning, and heat pump equipment and heating seasonal performance factors for heat pump equipment. The test procedure in this final rule references various sections of ANSI/ASHRAE 116–2010 that addresses test methods and calculations. ANSI/ASHRAE Standard 116–2010 can be purchased from ASHRAE's Web site at *https://www.ashrae.org/resources-publications.*

AMCA 210–2007 is an industry accepted standard that establishes uniform test methods for a laboratory test of a fan or other air moving device to determine its aerodynamic performance in terms of airflow rate, pressure developed, power consumption, air density, speed of rotation, and efficiency for rating or guarantee purposes. The test procedure in this final rule references various sections of AMCA 210–2007 that address test conditions. AMCA 210–2007 can be purchased from AMCA's Web site at *http://www.amca.org/store/index.php.*

N. Congressional Notification

As required by 5 U.S.C. 801, DOE will report to Congress on the promulgation of this rule before its effective date. The report will state that it has been determined that the rule is not a ''major rule'' as defined by 5 U.S.C. 804(2).

V. Approval of the Office of the Secretary

The Secretary of Energy has approved publication of this final rule.

List of Subjects

10 CFR Part 429

Administrative practice and procedure, Confidential business information, Energy conservation, Reporting and recordkeeping requirements.

10 CFR Part 430

Administrative practice and procedure, Confidential business information, Energy conservation, Energy conservation test procedures, Household appliances, Imports, Incorporation by reference, Intergovernmental relations, Small businesses.

Issued in Washington, DC, on November 30, 2016.

Kathleen B. Hogan,

Deputy Assistant Secretary for Energy Efficiency, Energy Efficiency and Renewable Energy.

For the reasons stated in the preamble, DOE amends parts 429 and 430 of chapter II of title 10, subpart B, Code of Federal Regulations, as set forth below:

PART 429—CERTIFICATION, COMPLIANCE, AND ENFORCEMENT FOR CONSUMER PRODUCTS AND COMMERCIAL AND INDUSTRIAL EQUIPMENT

■ 1. The authority citation for part 429 continues to read as follows:

Authority: 42 U.S.C. 6291–6317; 28 U.S.C. 2461 note.

■ 2. Section 429.11 is amended by revising paragraph (a) to read as follows:

§ 429.11 General sampling requirements for selecting units to be tested.

(a) When testing of covered products or covered equipment is required to comply with section 323(c) of the Act, or to comply with rules prescribed under section 324, 325, or 342, 344, 345 or 346 of the Act, a sample comprised of production units (or units representative of production units) of the basic model being tested must be selected at random and tested, and must meet the criteria found in §§ 429.14 through 429.62 of this subpart. Components of similar design may be substituted without additional testing if the substitution does not affect energy or water consumption. Any represented values of measures of energy efficiency, water efficiency, energy consumption, or water consumption for all individual models represented by a given basic model must be the same, except for central air conditioners and central air conditioning heat pumps, as specified in § 429.16 of this subpart.

* * * * *

■ 3. Section 429.16 is amended by:

■ a. Revising paragraph (a)(1);

■ b. Redesignating paragraphs (a)(3) and (4) as paragraphs (a)(4) and (5);

■ c. Adding new paragraph (a)(3);

■ d. Revising newly designated paragraph (a)(4)(i) and paragraph (b)(2)(i);

■ e. Revising paragraphs (b)(3) introductory text and (b)(3)(ii) and (iii);

■ f. Removing paragraph (b)(3)(iv); and

■ g. Revising paragraphs (c)(1)(i)(B), (c)(2) and (3), (d)(2) through (4), (e)(2) through (4), (f) introductory text, (f)(1) and (2), and (f)(4) and (5).

The revisions and addition read as follows:

§ 429.16 Central air conditioners and central air conditioning heat pumps.

(a) *Determination of Represented Value*—(1) *Required represented values.* Determine the represented values (including SEER, EER, HSPF, SEER2, EER2, HSPF2, $P_{W,OFF}$, cooling capacity, and heating capacity, as applicable) for the individual models/combinations (or ''tested combinations'') specified in the following table.

Category	Equipment subcategory	Required represented values
Single-Package unit	Single-Package AC (including Space-Constrained). Single-Package HP (including Space-Constrained).	Every individual model distributed in commerce.
Outdoor Unit and Indoor Unit (Distributed in Commerce by OUM).	Single-Split-System AC with Single-Stage or Two-Stage Compressor (including Space-Constrained and Small-Duct, High Velocity Systems (SDHV)).	Every individual combination distributed in commerce must be rated as a coil-only combination. For each model of outdoor unit, this must include at least one coil-only value that is representative of the least efficient combination distributed in commerce with that particular model of outdoor unit. Additional blower-coil representations are allowed for any applicable individual combinations, if distributed in commerce.
	Single-Split-System AC with Other Than Single-Stage or Two-Stage Compressor (including Space-Constrained and SDHV).	Every individual combination distributed in commerce, including all coil-only and blower coil combinations.
	Single-Split-System HP (including Space-Constrained and SDHV).	Every individual combination distributed in commerce.
	Multi-Split, Multi-Circuit, or Multi-Head Mini-Split Split System—non-SDHV (including Space-Constrained).	For each model of outdoor unit, at a minimum, a non-ducted "tested combination." For any model of outdoor unit also sold with models of ducted indoor units, a ducted "tested combination." When determining represented values on or after January 1, 2023, the ducted "tested combination" must comprise the highest static variety of ducted indoor unit distributed in commerce (*i.e.,* conventional, mid-static, or low-static). Additional representations are allowed, as described in paragraph (c)(3)(i) of this section.
	Multi-Split, Multi-Circuit, or Multi-Head Mini-Split Split System—SDHV.	For each model of outdoor unit, an SDHV "tested combination." Additional representations are allowed, as described in paragraph (c)(3)(ii) of this section.
Indoor Unit Only Distributed in Commerce by ICM).	Single-Split-System Air Conditioner (including Space-Constrained and SDHV). Single-Split-System Heat Pump (including Space-Constrained and SDHV).	Every individual combination distributed in commerce.
	Multi-Split, Multi-Circuit, or Multi-Head Mini-Split Split System—SDHV.	For a model of indoor unit within each basic model, an SDHV "tested combination." Additional representations are allowed, as described in section (c)(3)(ii) of this section.
Outdoor Unit with no Match		Every model of outdoor unit distributed in commerce (tested with a model of coil-only indoor unit as specified in paragraph (b)(2)(i) of this section).

* * * * *

(3) *Refrigerants.* (i) If a model of outdoor unit (used in a single-split, multi-split, multi-circuit, multi-head mini-split, and/or outdoor unit with no match system) is distributed in commerce and approved for use with multiple refrigerants, a manufacturer must determine all represented values for that model using each refrigerant that can be used in an individual combination of the basic model (including outdoor units with no match or "tested combinations"). This requirement may apply across the listed categories in the table in paragraph (a)(1) of this section. A refrigerant is considered approved for use if it is listed on the nameplate of the outdoor unit. If any of the refrigerants approved for use is HCFC-22 or has a 95 °F midpoint saturation absolute pressure that is +/− 18 percent of the 95 °F saturation absolute pressure for HCFC-22, or if there are no refrigerants designated as approved for use, a manufacturer must determine represented values (including SEER, EER, HSPF, SEER2, EER2, HSPF2, $P_{W,OFF}$, cooling capacity, and heating capacity, as applicable) for, at a minimum, an outdoor unit with no match. If a model of outdoor unit is not charged with a specified refrigerant from the point of manufacture or if the unit is shipped requiring the addition of more than two pounds of refrigerant to meet the charge required for testing per section 2.2.5 of appendix M or appendix M1 (unless either (a) the factory charge is equal to or greater than 70% of the outdoor unit internal volume times the liquid density of refrigerant at 95 °F or (b) an A2L refrigerant is approved for use and listed in the certification report), a manufacturer must determine represented values (including SEER, EER, HSPF, SEER2, EER2, HSPF2, $P_{W,OFF}$, cooling capacity, and heating capacity, as applicable) for, at a minimum, an outdoor unit with no match.

(ii) If a model is approved for use with multiple refrigerants, a manufacturer may make multiple separate representations for the performance of that model (all within the same individual combination or outdoor unit with no match) using the multiple approved refrigerants. In the alternative, manufacturers may certify the model (all within the same individual combination or outdoor unit with no match) with a single representation, provided that the represented value is no more efficient than its performance using the least-efficient refrigerant. If a manufacturer certifies a single model with multiple representations for the different approved refrigerants, it may use an AEDM to determine the represented values for all other refrigerants besides the refrigerant used for testing. A single representation made for multiple refrigerants may not include equipment in multiple categories or equipment subcategories listed in the table in paragraph (a)(1) of this section.

(4) * * *

(i) *Regional.* A basic model may only be certified as compliant with a regional standard if all individual combinations within that basic model meet the regional standard for which it is certified. A model of outdoor unit that is certified below a regional standard can only be rated and certified as compliant with a regional standard if the model of outdoor unit has a unique model number and has been certified as a different basic model for distribution in each region. An ICM cannot certify an individual combination with a rating that is compliant with a regional standard if the individual combination includes a model of outdoor unit that the OUM has certified with a rating that is not compliant with a regional standard. Conversely, an ICM cannot certify an individual combination with a rating that is not compliant with a regional standard if the individual combination includes a model of outdoor unit that an OUM has certified with a rating that is compliant with a regional standard.

* * * * *

(b) * * *

(2) *Individual model/combination selection for testing.* (i) The table identifies the minimum testing requirements for each basic model that includes multiple individual models/combinations; if a basic model spans multiple categories or subcategories listed in the table, multiple testing requirements apply. For each basic model that includes only one individual model/combination, test that individual model/combination. For single-split-system non-space-constrained air conditioners and heat pumps, when testing is required in accordance with 10 CFR part 430, subpart B, appendix M1, these requirements do not apply until July 1, 2024, provided that the manufacturer is certifying compliance of all basic models using an AEDM in accordance with paragraph (c)(1)(i)(B) of this section and paragraph (e)(2)(i)(A) of § 429.70.

Category	Equipment subcategory	Must test:	With:
Single-Package Unit	Single-Package AC (including Space-Constrained).	The individual model with the lowest SEER (when testing in accordance with appendix M to subpart B of part 430) or SEER2 (when testing in accordance with appendix M1 to subpart B of part 430).	N/A.
	Single-Package HP (including Space-Constrained).		
Outdoor Unit and Indoor Unit (Distributed in Commerce by OUM).	Single-Split-System AC with Single-Stage or Two-Stage Compressor (including Space-Constrained and Small-Duct, High Velocity Systems (SDHV)).	The model of outdoor unit	A model of coil-only indoor unit.
	Single-Split-System AC with Other Than Single-Stage or Two-Stage Compressor (including Space-Constrained and SDHV).	The model of outdoor unit	A model of indoor unit.
	Single-Split-System HP (including Space-Constrained and SDHV).		
	Multi-Split, Multi-Circuit, or Multi-Head Mini-Split Split System—non-SDHV (including Space-Constrained).	The model of outdoor unit	At a minimum, a "tested combination" composed entirely of non-ducted indoor units. For any models of outdoor units also sold with models of ducted indoor units, test a second "tested combination" composed entirely of ducted indoor units (in addition to the non-ducted combination). If testing under appendix M1 to subpart B of part 430, the ducted "tested combination" must comprise the highest static variety of ducted indoor unit distributed in commerce (*i.e.,* conventional, mid-static, or low-static).
	Multi-Split, Multi-Circuit, or Multi-Head Mini-Split Split System—SDHV.	The model of outdoor unit	A "tested combination" composed entirely of SDHV indoor units.
Indoor Unit Only (Distributed in Commerce by ICM).	Single-Split-System Air Conditioner (including Space-Constrained and SDHV).	A model of indoor unit	The least efficient model of outdoor unit with which it will be paired where the least efficient model of outdoor unit is the model of outdoor unit in the lowest SEER combination (when testing under appendix M to subpart B of part 430) or SEER2 combination (when testing under appendix M1 to subpart B of part 430) as certified by the OUM. If there are multiple models of outdoor unit with the same lowest SEER (when testing under appendix M to subpart B of part 430) or SEER2 (when testing under appendix M1 to subpart B of part 430) represented value, the ICM may select one for testing purposes.

Category	Equipment subcategory	Must test:	With:
	Single-Split-System Heat Pump (including Space-Constrained and SDHV).	Nothing, as long as an equivalent air conditioner basic model has been tested. If an equivalent air conditioner basic model has not been tested, must test a model of indoor unit.	
	Multi-Split, Multi-Circuit, or Multi-Head Mini-Split Split System—SDHV.	A model of indoor unit	A "tested combination" composed entirely of SDHV indoor units, where the outdoor unit is the least efficient model of outdoor unit with which the SDHV indoor unit will be paired. The least efficient model of outdoor unit is the model of outdoor unit in the lowest SEER combination (when testing under appendix M to subpart B of part 430) or SEER2 combination (when testing under appendix M1 to subpart B of part 430) as certified by the OUM. If there are multiple models of outdoor unit with the same lowest SEER represented value (when testing under appendix M to subpart B of part 430) or SEER2 represented value (when testing under appendix M1 to subpart B of part 430), the ICM may select one for testing purposes.
Outdoor Unit with No Match	....	The model of outdoor unit	A model of coil-only indoor unit meeting the requirements of section 2.2e of appendix M or M1 to subpart B of part 430.

* * * * *

(3) *Sampling plans and represented values.* For individual models (for single-package systems) or individual combinations (for split-systems, including "tested combinations" for multi-split, multi-circuit, and multi-head mini-split systems) with represented values determined through testing, each individual model/combination (or "tested combination") must have a sample of sufficient size tested in accordance with the applicable provisions of this subpart. For heat pumps (other than heating-only heat pumps), all units of the sample population must be tested in both the cooling and heating modes and the results used for determining all representations. The represented values for any individual model/combination must be assigned such that:

* * * * *

(ii) *SEER, EER, HSPF, SEER2, EER2, and HSPF2.* Any represented value of the energy efficiency or other measure of energy consumption for which consumers would favor higher values shall be less than or equal to the lower of:

(*A*) The mean of the sample, where:

$$\bar{x} = \frac{1}{n}\sum_{i=1}^{n} x_i$$

and, $\bar{x}$ is the sample mean; n is the number of samples; and x_i is the ith sample; or,

(*B*) The lower 90 percent confidence limit (LCL) of the true mean divided by 0.95, where:

$$LCL = \bar{x} - t_{.90}\left(\frac{s}{\sqrt{n}}\right)$$

And $\bar{x}$ is the sample mean; s is the sample standard deviation; n is the number of samples; and $t_{0.90}$ is the t statistic for a 90 percent one-tailed confidence interval with n-1 degrees of freedom (from appendix D). Round represented values of EER, SEER, HSPF, EER2, SEER2, and HSPF2 to the nearest 0.05.

(iii) *Cooling Capacity and Heating Capacity.* The represented values of cooling capacity and heating capacity must each be a self-declared value that is:

(A) Less than or equal to the lower of:

(*1*) The mean of the sample, where:

$$\bar{x} = \frac{1}{n}\sum_{i=1}^{n} x_i$$

and, $\bar{x}$ is the sample mean; n is the number of samples; and x_i is the ith sample; or,

(*2*) The lower 90 percent confidence limit (LCL) of the true mean divided by 0.95, where:

$$LCL = \bar{x} - t_{.90}\left(\frac{s}{\sqrt{n}}\right)$$

And $\bar{x}$ is the sample mean; s is the sample standard deviation; n is the number of samples; and $t_{0.90}$ is the t statistic for a 90 percent one-tailed confidence interval with n-1 degrees of freedom (from appendix D).

(B) Rounded according to:

(*1*) To the nearest 100 Btu/h if cooling capacity or heating capacity is less than 20,000 Btu/h,

(*2*) To the nearest 200 Btu/h if cooling capacity or heating capacity is greater than or equal to 20,000 Btu/h but less than 38,000 Btu/h, and

(*3*) To the nearest 500 Btu/h if cooling capacity or heating capacity is greater than or equal to 38,000 Btu/h and less than 65,000 Btu/h.

(c) * * *

(1) * * *

(i) * * *

(B) The represented values of the measures of energy efficiency or energy consumption through the application of an AEDM in accordance with paragraph (d) of this section and § 429.70. An AEDM may only be used to determine represented values for individual models or combinations in a basic model (or separate approved refrigerants within an individual combination) other than the individual model or combination(s) required for mandatory testing under paragraph (b)(2) of this section, except that, for single-split, non-space-constrained systems, when testing is required in accordance with 10 CFR part 430, subpart B, appendix M1, an AEDM may be used to rate the individual model or combination(s) required for mandatory testing under paragraph (b)(2) of this section until July

1, 2024, in accordance with paragraph (e)(2)(i)(A) of § 429.70.

* * * * *

(2) *Outdoor units with no match.* All models of outdoor units with no match within a basic model must be tested. No model of outdoor unit with no match may be rated with an AEDM, other than to determine the represented values for models using approved refrigerants other than the one used in testing.

(3) *For multi-split systems, multi-circuit systems, and multi-head mini-split systems.* The following applies:

(i) When testing in accordance with 10 CFR part 430, subpart B, appendix M1, for basic models that include additional varieties of ducted indoor units (*i.e.,* conventional, low-static, or mid-static) other than the one for which representation is required in paragraph (a)(1) of this section, if a manufacturer chooses to make a representation, the manufacturer must conduct testing of a tested combination according to the requirements in paragraph (b)(3) of this section.

(ii) When testing in accordance with 10 CFR part 430, subpart B, appendix M, for basic models composed of both non-ducted and ducted combinations, the represented value for the mixed non-ducted/ducted combination is the mean of the represented values for the non-ducted and ducted combinations as determined in accordance with paragraph (b)(3) of this section. When testing in accordance with 10 CFR part 430, subpart B, appendix M1, for basic models that include mixed combinations of indoor units (any two kinds of non-ducted, low-static, mid-static, and conventional ducted indoor units), the represented value for the mixed combination is the mean of the represented values for the individual component combinations as determined in accordance with paragraph (b)(3) of this section.

(iii) When testing in accordance with 10 CFR part 430, subpart B, appendix M, for basic models composed of both SDHV and non-ducted or ducted combinations, the represented value for the mixed SDHV/non-ducted or SDHV/ducted combination is the mean of the represented values for the SDHV, non-ducted, or ducted combinations, as applicable, as determined in accordance with paragraph (b)(3) of this section. When testing in accordance with 10 CFR part 430, subpart B, appendix M1, for basic models including mixed combinations of SDHV and another kind of indoor unit (any of non-ducted, low-static, mid-static, and conventional ducted), the represented value for the mixed SDHV/other combination is the mean of the represented values for the SDHV and other tested combination as determined in accordance with paragraph (b)(3) of this section.

(iv) All other individual combinations of models of indoor units for the same model of outdoor unit for which the manufacturer chooses to make representations must be rated as separate basic models, and the provisions of paragraphs (b)(1) through (3) and (c)(3)(i) through (iii) of this section apply.

(v) With respect to $P_{W,OFF}$ only, for every individual combination (or ''tested combination'') within a basic model tested pursuant to paragraph (b)(2) of this section, but for which $P_{W,OFF}$ testing was not conducted, the representative values of $P_{W,OFF}$ may be assigned through either:

(A) The testing result from an individual model or combination of similar off-mode construction, or

(B) Application of an AEDM in accordance with paragraph (d) of this section and § 429.70.

(d) * * *

(2) *Energy efficiency.* Any represented value of the SEER, EER, HSPF, SEER2, EER2, HSPF2 or other measure of energy efficiency of an individual model/combination for which consumers would favor higher values must be less than or equal to the output of the AEDM but no less than the standard.

(3) *Cooling capacity.* The represented value of cooling capacity of an individual model/combination must be no greater than the cooling capacity output simulated by the AEDM.

(4) *Heating capacity.* The represented value of heating capacity of an individual model/combination must be no greater than the heating capacity output simulated by the AEDM.

(e) * * *

(2) *Public product-specific information.* Pursuant to § 429.12(b)(13), for each individual model (for single-package systems) or individual combination (for split-systems, including outdoor units with no match and ''tested combinations'' for multi-split, multi-circuit, and multi-head mini-split systems), a certification report must include the following public product-specific information: When certifying compliance with January 1, 2015, energy conservation standards, the seasonal energy efficiency ratio (SEER in British thermal units per Watt-hour (Btu/W-h)) or when certifying compliance with January 1, 2023, energy conservation standards, seasonal energy efficiency ratio 2 (SEER2 in British thermal units per Watt-hour (Btu/W-h)); the average off mode power consumption ($P_{W,OFF}$ in Watts); the cooling capacity in British thermal units per hour (Btu/h); the region(s) in which the basic model can be sold; when certifying compliance with January 1, 2023, energy conservation standards, the kind(s) of air conditioner or heat pump associated with the minimum external static pressure used in testing or rating (ceiling-mount, wall-mount, mobile home, low-static, mid-static, small duct high velocity, space-constrained, or conventional/not otherwise listed); and

(i) For heat pumps, when certifying compliance with January 1, 2015, energy conservation standards, the heating seasonal performance factor (HSPF in British thermal units per Watt-hour (Btu/W-h)) or, when certifying compliance with January 1, 2023, energy conservation standards, heating seasonal performance factor 2 (HSPF2 in British thermal units per Watt-hour (Btu/W-h));

(ii) For central air conditioners (excluding space-constrained products), when certifying compliance with January 1, 2015, energy conservation standards, the energy efficiency ratio (EER in British thermal units per Watt-hour (Btu/W-h)) from the A or A_2 test, whichever applies, or when certifying compliance with January 1, 2023, energy conservation standards, the energy efficiency ratio 2 (EER2 in Btu/W-h);

(iii) For single-split-systems, whether the represented value is for a coil-only or blower coil system;

(iv) For multi-split, multiple-circuit, and multi-head mini-split systems (including VRF and SDHV), when certifying compliance with January 1, 2015, energy conservation standards, whether the represented value is for a non-ducted, ducted, mixed non-ducted/ducted system, SDHV, mixed non-ducted/SDHV system, or mixed ducted/SDHV system;

(v) For all split systems including outdoor units with no match, the refrigerant.

(3) *Basic and individual model numbers.* The basic model number and individual model number(s) required to be reported under § 429.12(b)(6) must consist of the following:

Equipment type	Basic model number	Individual model number(s)		
		1	2	3
Single-Package (including Space-Constrained).	Number unique to the basic model.	Package	N/A	N/A.
Single-Split System (including Space-Constrained and SDHV).	Number unique to the basic model.	Outdoor Unit	Indoor Unit	If applicable—Air Mover (could be same as indoor unit if fan is part of indoor unit model number).
Multi-Split, Multi-Circuit, and Multi-Head Mini-Split System (including Space-Constrained and SDHV).	Number unique to the basic model.	Outdoor Unit	When certifying a basic model based on tested combination(s): * * *. When certifying an individual combination: Indoor Unit(s).	If applicable—When certifying a basic model based on tested combination(s): * * *. When certifying an individual combination: Air Mover(s).
Outdoor Unit with No Match.	Number unique to the basic model.	Outdoor Unit	N/A	N/A.

(4) *Additional product-specific information.* Pursuant to § 429.12(b)(13), for each individual model/combination (including outdoor units with no match and ''tested combinations''), a certification report must include the following additional product-specific information: The cooling full load air volume rate for the system or for each indoor unit as applicable (in cubic feet per minute of standard air (scfm)); the air volume rates that represent normal operation for other test conditions including minimum cooling air volume rate, intermediate cooling air volume rate, full load heating air volume rate, minimum heating air volume rate, intermediate heating air volume rate, and nominal heating air volume rate (scfm) for the system or for each indoor unit as applicable, if different from the cooling full load air volume rate; whether the individual model uses a fixed orifice, thermostatic expansion valve, electronic expansion valve, or other type of metering device; the duration of the compressor break-in period, if used; whether the optional tests were conducted to determine the C_D^c value used to represent cooling mode cycling losses or whether the default value was used; the temperature at which the crankcase heater with controls is designed to turn on, if applicable; whether an inlet plenum was installed during testing; the duration of the indoor fan time delay, if used; and

(i) For heat pumps, whether the optional tests were conducted to determine the C_D^h value or whether the default value was used; and the maximum time between defrosts as allowed by the controls (in hours);

(ii) For multi-split, multiple-circuit, and multi-head mini-split systems, the number of indoor units tested with the outdoor unit; the nominal cooling capacity of each indoor unit and outdoor unit in the combination; and the indoor units that are not providing heating or cooling for part-load tests;

(iii) For ducted systems having multiple indoor fans within a single indoor unit, the number of indoor fans; the nominal cooling capacity of the indoor unit and outdoor unit; which fan(s) operate to attain the full-load air volume rate when controls limit the simultaneous operation of all fans within the single indoor unit; and the allocation of the full-load air volume rate to each operational fan when different capacity blowers are connected to the common duct;

(iv) For blower coil systems, the airflow-control settings associated with full load cooling operation; and the airflow-control settings or alternative instructions for setting fan speed to the speed upon which the rating is based;

(v) For models with time-adaptive defrost control, the frosting interval to be used during Frost Accumulation tests and the procedure for manually initiating the defrost at the specified time;

(vi) For models of indoor units designed for both horizontal and vertical installation or for both up-flow and down-flow vertical installations, the orientation used for testing;

(vii) For variable-speed models, the compressor frequency set points, and the required dip switch/control settings for step or variable components;

(viii) For variable-speed heat pumps, whether the $H1_N$ or $H1_2$ test speed is the same as the $H3_2$ test speed; the compressor frequency that corresponds to maximum speed at which the system controls would operate the compressor in normal operation in a 17 °F ambient temperature; and when certifying compliance with January 1, 2023, energy conservation standards, whether the optional 5 °F very low temperature heating mode test was used to characterize performance at temperatures below 17 °F (except for triple-capacity northern heat pumps, for which the very low temperature test is required,) and whether the alternative test required for minimum-speed-limiting variable-speed heat pumps was used;

(ix) For models of outdoor units with no match, the following characteristics of the indoor coil: The face area, the coil depth in the direction of airflow, the fin density (fins per inch), the fin material, the fin style, the tube diameter, the tube material, and the numbers of tubes high and deep; and

(x) For central air conditioners and heat pumps that have two-capacity compressors that lock out low capacity operation for cooling at higher outdoor temperatures and/or heating at lower outdoor temperatures, the outdoor temperature(s) at which the unit locks out low capacity operation.

(f) *Represented values for the Federal Trade Commission.* Use the following represented value determinations to meet the requirements of the Federal Trade Commission.

(1) *Annual Operating Cost—Cooling.* Determine the represented value of estimated annual operating cost for cooling-only units or the cooling portion of the estimated annual operating cost for air-source heat pumps that provide both heating and cooling by calculating the product of:

(i) The value determined in paragraph (f)(1)(i)(A) of this section if using appendix M to subpart B of part 430 or the value determined in paragraph (f)(1)(i)(B) of this section if using appendix M1 to subpart B of part 430;

(A) the quotient of the represented value of cooling capacity, in Btu's per hour as determined in paragraph (b)(3)(iii) of this section, divided by the represented value of SEER, in Btu's per watt-hour, as determined in paragraph (b)(3)(ii) of this section;

(B) the quotient of the represented value of cooling capacity, in Btu's per hour as determined in paragraph (b)(3)(i)(C) of this section, and multiplied by 0.93 for variable-speed heat pumps only, divided by the represented value of SEER2, in Btu's per watt-hour, as determined in paragraph (b)(3)(i)(B) of this section.

(ii) The representative average use cycle for cooling of 1,000 hours per year;

(iii) A conversion factor of 0.001 kilowatt per watt; and

(iv) The representative average unit cost of electricity in dollars per kilowatt-hour as provided pursuant to section 323(b)(2) of the Act.

(2) *Annual Operating Cost—Heating.* Determine the represented value of estimated annual operating cost for air-source heat pumps that provide only heating or for the heating portion of the estimated annual operating cost for air-source heat pumps that provide both heating and cooling, as follows:

(i) When using appendix M to subpart B of part 430, the product of:

(A) The quotient of the mean of the standardized design heating requirement for the sample, in Btu's per hour, nearest to the Region IV minimum design heating requirement, determined for each unit in the sample in section 4.2 of appendix M to subpart B of part 430, divided by the represented value of heating seasonal performance factor (HSPF), in Btu's per watt-hour, calculated for Region IV corresponding to the above-mentioned standardized design heating requirement, as determined in paragraph (b)(3)(ii) of this section;

(B) The representative average use cycle for heating of 2,080 hours per year;

(C) The adjustment factor of 0.77, which serves to adjust the calculated design heating requirement and heating load hours to the actual load experienced by a heating system;

(D) A conversion factor of 0.001 kilowatt per watt; and

(E) The representative average unit cost of electricity in dollars per kilowatt-hour as provided pursuant to section 323(b)(2) of the Act;

(ii) When using appendix M1 to subpart B of part 430, the product of:

(A) The quotient of the represented value of cooling capacity (for air-source heat pumps that provide both cooling and heating) in Btu's per hour, as determined in paragraph (b)(3)(i)(C) of this section, or the represented value of heating capacity (for air-source heat pumps that provide only heating), as determined in paragraph (b)(3)(i)(D) of this section, divided by the represented value of heating seasonal performance factor 2 (HSPF2), in Btu's per watt-hour, calculated for Region IV, as determined in paragraph (b)(3)(i)(B) of this section;

(B) The representative average use cycle for heating of 1,572 hours per year;

(C) The adjustment factor of 1.15 (for heat pumps that are not variable-speed) or 1.07 (for heat pumps that are variable-speed), which serves to adjust the calculated design heating requirement and heating load hours to the actual load experienced by a heating system;

(D) A conversion factor of 0.001 kilowatt per watt; and

(E) The representative average unit cost of electricity in dollars per kilowatt-hour as provided pursuant to section 323(b)(2) of the Act;

* * * * *

(4) *Regional Annual Operating Cost—Cooling.* Determine the represented value of estimated regional annual operating cost for cooling-only units or the cooling portion of the estimated regional annual operating cost for air-source heat pumps that provide both heating and cooling by calculating the product of:

(i) The value determined in paragraph (f)(4)(i)(A) of this section if using appendix M to subpart B of part 430 or the value determined in paragraph (f)(4)(i)(B) of this section if using appendix M1 to subpart B of part 430;

(A) the quotient of the represented value of cooling capacity, in Btu's per hour as determined in paragraph (b)(3)(iii) of this section, divided by the represented value of SEER, in Btu's per watt-hour, as determined in paragraph (b)(3)(ii) of this section;

(B) the quotient of the represented value of cooling capacity, in Btu's per hour as determined in paragraph (b)(3)(i)(C) of this section, and multiplied by 0.93 for variable-speed heat pumps only, divided by the represented value of SEER2, in Btu's per watt-hour, as determined in paragraph (b)(3)(i)(B) of this section;

(ii) The value determined in paragraph (f)(4)(ii)(A) of this section if using appendix M to subpart B of part 430 or the value determined in paragraph (f)(4)(ii)(B) of this section if using appendix M1 to subpart B of part 430;

(A) the estimated number of regional cooling load hours per year determined from Table 22 in section 4.4 of appendix M to subpart B of part 430;

(B) the estimated number of regional cooling load hours per year determined from Table 21 in section 4.4 of appendix M1 to subpart B of part 430;

(iii) A conversion factor of 0.001 kilowatts per watt; and

(iv) The representative average unit cost of electricity in dollars per kilowatt-hour as provided pursuant to section 323(b)(2) of the Act.

(5) *Regional Annual Operating Cost—Heating.* Determine the represented value of estimated regional annual operating cost for air-source heat pumps that provide only heating or for the heating portion of the estimated regional annual operating cost for air-source heat pumps that provide both heating and cooling as follows:

(i) When using appendix M to subpart B of part 430, the product of:

(A) The estimated number of regional heating load hours per year determined from Table 22 in section 4.4 of appendix M to subpart B of part 430;

(B) The quotient of the mean of the standardized design heating requirement for the sample, in Btu's per hour, for the appropriate generalized climatic region of interest (*i.e.,* corresponding to the regional heating load hours from ''A'') and determined for each unit in the sample in section 4.2 of appendix M to subpart B of part 430, divided by the represented value of HSPF, in Btu's per watt-hour, calculated for the appropriate generalized climatic region of interest and corresponding to the above-mentioned standardized design heating requirement, and determined in paragraph (b)(3)(ii);

(C) The adjustment factor of 0.77; which serves to adjust the calculated design heating requirement and heating load hours to the actual load experienced by a heating system;

(D) A conversion factor of 0.001 kilowatts per watt; and

(E) The representative average unit cost of electricity in dollars per kilowatt-hour as provided pursuant to section 323(b)(2) of the Act.

(ii) When using appendix M1 to subpart B of part 430, the product of:

(A) The estimated number of regional heating load hours per year determined from Table 21 in section 4.4 of appendix M1 to subpart B of part 430;

(B) The quotient of the represented value of cooling capacity (for air-source heat pumps that provide both cooling and heating) in Btu's per hour, as determined in paragraph (b)(3)(i)(C) of this section, or the represented value of heating capacity (for air-source heat pumps that provide only heating), as determined in paragraph (b)(3)(i)(D) of this section, divided by the represented value of HSPF2, in Btu's per watt-hour, calculated for the appropriate generalized climatic region of interest, and determined in paragraph (b)(3)(i)(B) of this section;

(C) The adjustment factor of 1.15 (for heat pumps that are not variable-speed) or 1.07 (for heat pumps that are variable-speed), which serves to adjust the calculated design heating requirement and heating load hours to the actual load experienced by a heating system;

(D) A conversion factor of 0.001 kilowatts per watt; and

(E) The representative average unit cost of electricity in dollars per kilowatt-hour as provided pursuant to section 323(b)(2) of the Act.

* * * * *

■ 4. Section 429.70 is amended by revising paragraphs (e)(1), (e)(2)(i), and (e)(5)(iv) to read as follows:

§ 429.70 Alternative methods for determining energy efficiency or energy use.

* * * * *

(e) * * *

(1) *Criteria an AEDM must satisfy.* A manufacturer may not apply an AEDM to an individual model/combination to determine its represented values (SEER, EER, HSPF, SEER2, EER2, HSPF2, and/or $P_{W,OFF}$) pursuant to this section unless authorized pursuant to § 429.16(d) and:

(i) The AEDM is derived from a mathematical model that estimates the energy efficiency or energy consumption characteristics of the individual model or combination (SEER, EER, HSPF, SEER2, EER2, HSPF2, and/or $P_{W,OFF}$) as measured by the applicable DOE test procedure; and

(ii) The manufacturer has validated the AEDM in accordance with paragraph (e)(2) of this section.

(2) * * *

(i) Follow paragraph (e)(2)(i)(A) of this section for requirements on minimum testing. Follow paragraph (e)(2)(i)(B) of this section for requirements on ensuring the accuracy and reliability of the AEDM.

(A) *Minimum testing.* (*1*) For non-space-constrained single-split system air conditioners and heat pumps rated based on testing in accordance with appendix M to subpart B of part 430, the manufacturer must test each basic model as required under § 429.16(b)(2). Until July 1, 2024, for non-space-constrained single-split-system air conditioners and heat pumps rated based on testing in accordance with appendix M1 to subpart B of part 430, the manufacturer must test a single-unit sample from 20 percent of the basic models distributed in commerce to validate the AEDM. On or after July 1, 2024, for non-space-constrained single-split-system air conditioners and heat pumps rated based on testing in accordance with appendix M1 to subpart B of part 430, the manufacturer must complete testing of each basic model as required under § 429.16(b)(2).

(*2*) For other than non-space-constrained single-split-system air conditioners and heat pumps, the manufacturer must test each basic model as required under § 429.16(b)(2).

(B) Using the AEDM, calculate the energy use or efficiency for each of the tested individual models/combinations within each basic model. Compare the represented value based on testing and the AEDM energy use or efficiency output according to paragraph (e)(2)(ii) of this section. The manufacturer is responsible for ensuring the accuracy and reliability of the AEDM and that their representations are appropriate and the models being distributed in commerce meet the applicable standards, regardless of the amount of testing required in paragraphs (e)(2)(i)(A) and (e)(2)(i)(B) of this section.

* * * * *

(5) * * *

(iv) *Failure to meet certified value.* If an individual model/combination tests worse than its certified value (*i.e.,* lower than the certified efficiency value or higher than the certified consumption value) by more than 5 percent, or the test results in cooling capacity that is lower than its certified cooling capacity, DOE will notify the manufacturer. DOE will provide the manufacturer with all documentation related to the test set up, test conditions, and test results for the unit. Within the timeframe allotted by DOE, the manufacturer may present any and all claims regarding testing validity.

* * * * *

PART 430—ENERGY CONSERVATION PROGRAM FOR CONSUMER PRODUCTS

■ 5. The authority citation for part 430 continues to read as follows:

Authority: 42 U.S.C. 6291–6309; 28 U.S.C. 2461 note.

■ 6. Section 430.2 is amended by revising the definition of ''central air conditioner or central air conditioning heat pump'' to read as follows:

§ 430.2 Definitions.

* * * * *

Central air conditioner or central air conditioning heat pump means a product, other than a packaged terminal air conditioner or packaged terminal heat pump, which is powered by single phase electric current, air cooled, rated below 65,000 Btu per hour, not contained within the same cabinet as a furnace, the rated capacity of which is above 225,000 Btu per hour, and is a heat pump or a cooling unit only. A central air conditioner or central air conditioning heat pump may consist of: A single-package unit; an outdoor unit and one or more indoor units; an indoor unit only; or an outdoor unit with no match. In the case of an indoor unit only or an outdoor unit with no match, the unit *must* be tested and rated as a system (combination of both an indoor and an outdoor unit). For all central air conditioner and central air conditioning heat pump-related definitions, see appendix M or M1 of subpart B of this part.

* * * * *

§ 430.3 [Amended]

■ 7. Section 430.3 is amended by removing in paragraphs (b)(2) introductory text, (c)(1) introductory text, (c)(3) introductory text, (g)(2) introductory text, (g)(4) introductory text, (g)(7) introductory text, (g)(8) introductory text, (g)(9) introductory text, (g)(10) introductory text, and (g)(13) ''appendix M'' and adding in its place ''appendices M and M1''.

■ 8. Section 430.23 is amended by revising paragraph (m) to read as follows:

§ 430.23 Test procedures for the measurement of energy and water consumption.

* * * * *

(m) *Central air conditioners and heat pumps.* See the note at the beginning of appendix M and M1 to determine the appropriate test method. Determine all values discussed in this section using a single appendix.

(1) Determine cooling capacity from the steady-state wet-coil test (A or A_2 Test), as described in section 3.2 of appendix M or M1 to this subpart, and rounded off to the nearest

(i) To the nearest 50 Btu/h if cooling capacity is less than 20,000 Btu/h;

(ii) To the nearest 100 Btu/h if cooling capacity is greater than or equal to 20,000 Btu/h but less than 38,000 Btu/h; and

(iii) To the nearest 250 Btu/h if cooling capacity is greater than or equal to 38,000 Btu/h and less than 65,000 Btu/h.

(2) Determine seasonal energy efficiency ratio (SEER) as described in section 4.1 of appendix M to this subpart or seasonal energy efficiency ratio 2 (SEER2) as described in section

4.1 of appendix M1 to this subpart, and round off to the nearest 0.025 Btu/W-h.

(3) Determine energy efficiency ratio (EER) as described in section 4.6 of appendix M or M1 to this subpart, and round off to the nearest 0.025 Btu/W-h. The EER from the A or A_2 test, whichever applies, when tested in accordance with appendix M1 to this subpart, is referred to as EER2.

(4) Determine heating seasonal performance factors (HSPF) as described in section 4.2 of appendix M to this subpart or heating seasonal performance factors 2 (HSPF2) as described in section 4.2 of appendix M1 to this subpart, and round off to the nearest 0.025 Btu/W-h.

(5) Determine average off mode power consumption as described in section 4.3 of appendix M or M1 to this subpart, and round off to the nearest 0.5 W.

(6) Determine all other measures of energy efficiency or consumption or other useful measures of performance using appendix M or M1 of this subpart.

* * * * *

■ 9. Appendix M to subpart B of part 430 is revised to read as follows:

Appendix M to Subpart B of Part 430—Uniform Test Method for Measuring the Energy Consumption of Central Air Conditioners and Heat Pumps

Note: Prior to July 5, 2017, any representations, including compliance certifications, made with respect to the energy use, power, or efficiency of central air conditioners and central air conditioning heat pumps must be based on the results of testing pursuant to either this appendix or the procedures in Appendix M as it appeared at 10 CFR part 430, subpart B, Appendix M, in the 10 CFR parts 200 to 499 edition revised as of January 1, 2017. Any representations made with respect to the energy use or efficiency of such central air conditioners and central air conditioning heat pumps must be in accordance with whichever version is selected.

On or after July 5, 2017 and prior to January 1, 2023, any representations, including compliance certifications, made with respect to the energy use, power, or efficiency of central air conditioners and central air conditioning heat pumps must be based on the results of testing pursuant to this appendix.

On or after January 1, 2023, any representations, including compliance certifications, made with respect to the energy use, power, or efficiency of central air conditioners and central air conditioning heat pumps must be based on the results of testing pursuant to appendix M1 of this subpart.

1. Scope and Definitions

1.1 Scope

This test procedure provides a method of determining SEER, EER, HSPF and $P_{W,OFF}$ for central air conditioners and central air conditioning heat pumps including the following categories:

(a) Split-system air conditioners, including single-split, multi-head mini-split, multi-split (including VRF), and multi-circuit systems
(b) Split-system heat pumps, including single-split, multi-head mini-split, multi-split (including VRF), and multi-circuit systems
(c) Single-package air conditioners
(d) Single-package heat pumps
(e) Small-duct, high-velocity systems (including VRF)
(f) Space-constrained products—air conditioners
(g) Space-constrained products—heat pumps

For purposes of this appendix, the Department of Energy incorporates by reference specific sections of several industry standards, as listed in § 430.3. In cases where there is a conflict, the language of the test procedure in this appendix takes precedence over the incorporated standards.

All section references refer to sections within this appendix unless otherwise stated.

1.2 Definitions

Airflow-control settings are programmed or wired control system configurations that control a fan to achieve discrete, differing ranges of airflow—often designated for performing a specific function (*e.g.,* cooling, heating, or constant circulation)—without manual adjustment other than interaction with a user-operable control (*i.e.,* a thermostat) that meets the manufacturer specifications for installed-use. For the purposes of this appendix, manufacturer specifications for installed-use are those found in the product literature shipped with the unit.

Air sampling device is an assembly consisting of a manifold with several branch tubes with multiple sampling holes that draws an air sample from a critical location from the unit under test (*e.g.* indoor air inlet, indoor air outlet, outdoor air inlet, etc.).

Airflow prevention device denotes a device that prevents airflow via natural convection by mechanical means, such as an air damper box, or by means of changes in duct height, such as an upturned duct.

Aspirating psychrometer is a piece of equipment with a monitored airflow section that draws uniform airflow through the measurement section and has probes for measurement of air temperature and humidity.

Blower coil indoor unit means an indoor unit either with an indoor blower housed with the coil or with a separate designated air mover such as a furnace or a modular blower (as defined in appendix AA to the subpart).

Blower coil system refers to a split system that includes one or more blower coil indoor units.

Cased coil means a coil-only indoor unit with external cabinetry.

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 determined at a single set of operating conditions. COP is a dimensionless quantity. When determined for a ducted coil-only system, COP must include the sections 3.7 and 3.9.1 of this appendix: Default values for the heat output and power input of a fan motor.

Coil-only indoor unit means an indoor unit that is distributed in commerce without an indoor blower or separate designated air mover. A coil-only indoor unit installed in the field relies on a separately-installed furnace or a modular blower for indoor air movement. *Coil-only system* refers to a system that includes only (one or more) coil-only indoor units.

Condensing unit removes the heat absorbed by the refrigerant to transfer it to the outside environment and consists of an outdoor coil, compressor(s), and air moving device.

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

Continuously recorded, when referring to a dry bulb measurement, dry bulb temperature used for test room control, wet bulb temperature, dew point temperature, or relative humidity measurements, means that the specified value must be sampled at regular intervals that are equal to or less than 15 seconds.

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, and as its denominator 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.

Crankcase heater means any electrically powered device or mechanism for intentionally generating heat within and/or around the compressor sump volume. Crankcase heater control may be achieved using a timer or may be based on a change in temperature or some other measurable parameter, such that the crankcase heater is not required to operate continuously. A crankcase heater without controls operates continuously when the compressor is not operating.

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.

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

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.

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 either:

(1) 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) at least once for every ten minutes of compressor ON-time when space heating or

(2) operate as a feedback system that measures the length of the defrost period and adjusts defrost frequency accordingly. 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. (**Note:** Systems that vary defrost intervals according to outdoor dry-bulb temperature are not demand-defrost systems.) 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.

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 of this appendix.

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

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-package unit.

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. Determine these rate quantities from a single test or, if derived via interpolation, determine at a single set of operating conditions. EER is expressed in units of

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

When determined for a ducted coil-only system, EER must include, from this appendix, the section 3.3 and 3.5.1 default values for the heat output and power input of a fan motor.

Evaporator coil means an assembly that absorbs heat from an enclosed space and transfers the heat to a refrigerant.

Heat pump means a kind of central air conditioner that utilizes an indoor conditioning coil, compressor, and refrigerant-to-outdoor air heat exchanger to provide air heating, and may also provide air cooling, air dehumidifying, air humidifying, air circulating, and air cleaning.

Heat pump having a heat comfort controller means a heat pump with controls that can regulate the operation of the electric resistance elements to assure that the air temperature leaving the indoor section does not fall below a specified temperature. 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.

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, and its denominator the heating capacity measured at the same test conditions used for the cyclic test, multiplied by the total time interval (ON plus OFF) of the cyclic-test.

Heating season means the months of the year that require heating, *e.g.,* typically, and roughly, October through April.

Heating seasonal performance factor (HSPF) means the total space heating required during the heating season, expressed in Btu, 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 10 CFR 430.32(c) is based on Region IV and the sampling plan stated in 10 CFR 429.16(a). HSPF is determined in accordance with appendix M.

Independent coil manufacturer (ICM) means a manufacturer that manufactures indoor units but does not manufacture single-package units or outdoor units.

Indoor unit means a separate assembly of a split system that includes—

(1) An arrangement of refrigerant-to-air heat transfer coil(s) for transfer of heat between the refrigerant and the indoor air,

(2) A condensate drain pan, and may or may not include

(3) Sheet metal or plastic parts not part of external cabinetry to direct/route airflow over the coil(s),

(4) A cooling mode expansion device,

(5) External cabinetry, and

(6) An integrated indoor blower (*i.e.* a device to move air including its associated motor). A separate designated air mover that may be a furnace or a modular blower (as defined in appendix AA to the subpart) may be considered to be part of the indoor unit. A service coil is not an indoor unit.

Multi-head mini-split system means a split system that has one outdoor unit and that has two or more indoor units connected with a single refrigeration circuit. The indoor units operate in unison in response to a single indoor thermostat.

Multiple-circuit (or multi-circuit) system means a split system that has one outdoor unit and that has two or more indoor units installed on two or more refrigeration circuits such that each refrigeration circuit serves a compressor and one and only one indoor unit, and refrigerant is not shared from circuit to circuit.

Multiple-split (or multi-split) system means a split system that has one outdoor unit and two or more coil-only indoor units and/or blower coil indoor units connected with a single refrigerant circuit. The indoor units operate independently and can condition multiple zones in response to at least two indoor thermostats or temperature sensors. The outdoor unit operates in response to independent operation of the indoor units based on control input of multiple indoor thermostats or temperature sensors, and/or based on refrigeration circuit sensor input (*e.g.,* suction pressure).

Nominal capacity means the capacity that is claimed by the manufacturer on the product name plate. Nominal cooling capacity is approximate to the air conditioner cooling capacity tested at A or A2 condition. Nominal heating capacity is approximate to the heat pump heating capacity tested in H12 test (or the optional H1N test).

Non-ducted indoor unit means an indoor unit that is designed to be permanently installed, mounted on room walls and/or ceilings, and that directly heats or cools air within the conditioned space.

Normalized Gross Indoor Fin Surface (NGIFS) means the gross fin surface area of the indoor unit coil divided by the cooling capacity measured for the A or A2 Test, whichever applies.

Off-mode power consumption means the power consumption when the unit is connected to its main power source but is neither providing cooling nor heating to the building it serves.

Off-mode season means, for central air conditioners other than heat pumps, the shoulder season and the entire heating season; and for heat pumps, the shoulder season only.

Outdoor unit means a separate assembly of a split system that transfers heat between the refrigerant and the outdoor air, and consists of an outdoor coil, compressor(s), an air moving device, and in addition for heat pumps, may include a heating mode expansion device, reversing valve, and/or defrost controls.

Outdoor unit manufacturer (OUM) means a manufacturer of single-package units, outdoor units, and/or both indoor units and outdoor units.

Part-load factor (PLF) means the ratio of the cyclic EER (or COP for heating) to the steady-state EER (or COP), where both EERs (or COPs) are determined based on operation at the same ambient conditions.

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 central air conditioner or heat pump during the same season, expressed in watt-hours. SEER is determined in accordance with appendix M.

Service coil means an arrangement of refrigerant-to-air heat transfer coil(s), condensate drain pan, sheet metal or plastic parts to direct/route airflow over the coil(s), which may or may not include external cabinetry and/or a cooling mode expansion device, distributed in commerce solely for replacing an uncased coil or cased coil that has already been placed into service, and that has been labeled ''for indoor coil replacement only'' on the nameplate and in manufacturer technical and product literature. The model number for any service coil must include some mechanism (*e.g.,* an additional letter or number) for differentiating a service coil from a coil intended for an indoor unit.

Shoulder season means the months of the year in between those months that require cooling and those months that require heating, *e.g.,* typically, and roughly, April through May, and September through October.

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

Single-split system means a split system that has one outdoor unit and one indoor unit connected with a single refrigeration circuit. *Small-duct, high-velocity system* means a split system for which all indoor units are blower coil indoor units that produce at least 1.2 inches (of water column) of external static pressure when operated at the full-load air volume rate certified by the manufacturer of at least 220 scfm per rated ton of cooling.

Split system means any air conditioner or heat pump that has at least two separate assemblies that are connected with refrigerant piping when installed. One of these assemblies includes an indoor coil that exchanges heat with the indoor air to provide heating or cooling, while one of the others includes an outdoor coil that exchanges heat with the outdoor air. Split systems may be either blower coil systems or coil-only systems.

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

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.

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.

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

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.

Tested combination means a multi-head mini-split, multi-split, or multi-circuit system having the following features:

(1) The system consists of one outdoor unit with one or more compressors matched with between two and five indoor units;

(2) The indoor units must:

(i) Collectively, have a nominal cooling capacity greater than or equal to 95 percent and less than or equal to 105 percent of the nominal cooling capacity of the outdoor unit;

(ii) Each represent the highest sales volume model family, if this is possible while meeting all the requirements of this section. If this is not possible, one or more of the indoor units may represent another indoor model family in order that all the other requirements of this section are met.

(iii) Individually not have a nominal cooling capacity greater than 50 percent of the nominal cooling capacity of the outdoor unit, unless the nominal cooling capacity of the outdoor unit is 24,000 Btu/h or less;

(iv) Operate at fan speeds consistent with manufacturer's specifications; and

(v) All be subject to the same minimum external static pressure requirement while able to produce the same external static pressure at the exit of each outlet plenum when connected in a manifold configuration as required by the test procedure.

(3) Where referenced, ''nominal cooling capacity'' means, for indoor units, the highest cooling capacity listed in published product literature for 95 °F outdoor dry bulb temperature and 80 °F dry bulb, 67 °F wet bulb indoor conditions, and for outdoor units, the lowest cooling capacity listed in published product literature for these conditions. If incomplete or no operating conditions are published, the highest (for indoor units) or lowest (for outdoor units) such cooling capacity available for sale must be used.

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

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 or not 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).

Triple-capacity, northern heat pump means a heat pump that provides two stages of cooling and three stages of heating. The two common stages for both the cooling and heating modes are the low capacity stage and the high capacity stage. The additional heating mode stage is the booster capacity stage, which offers the highest heating capacity output for a given set of ambient operating conditions.

Triple-split system means a split system that is composed of three separate assemblies: An outdoor fan coil section, a blower coil indoor unit, and an indoor compressor section.

Two-capacity (or two-stage) compressor system means a central air conditioner or heat pump that has a compressor or a group of compressors operating with only two stages of capacity. For such systems, low capacity means the compressor(s) operating at low stage, or at low load test conditions. The low compressor stage that operates for heating mode tests may be the same or different from the low compressor stage that operates for cooling mode tests. For such systems, high capacity means the compressor(s) operating at high stage, or at full load test conditions.

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 heat pump is a two-capacity northern heat pump only when this feature is enabled at all times. The certified indoor coil model number must 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.

Uncased coil means a coil-only indoor unit without external cabinetry.

Variable refrigerant flow (VRF) system means a multi-split system with at least three compressor capacity stages, distributing refrigerant through a piping network to multiple indoor blower coil units each capable of individual zone temperature control, through proprietary zone temperature control devices and a common communications network. **Note:** Single-phase VRF systems less than 65,000 Btu/h are central air conditioners and central air conditioning heat pumps.

Variable-speed compressor system means a central air conditioner or heat pump that has a compressor that uses a variable-speed drive to vary the compressor speed to achieve variable capacities.

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 Overview and Conditions

(A) Test VRF systems using AHRI 1230–2010 (incorporated by reference, see § 430.3) and appendix M. Where AHRI 1230–2010 refers to the appendix C therein substitute the provisions of this appendix. In cases where there is a conflict, the language of the test procedure in this appendix takes precedence over AHRI 1230–2010.

For definitions use section 1 of appendix M and section 3 of AHRI 1230–2010 (incorporated by reference, see § 430.3). For rounding requirements, refer to § 430.23(m). For determination of certified ratings, refer to § 429.16 of this chapter.

For test room requirements, refer to section 2.1 of this appendix. For test unit installation requirements refer to sections 2.2.a, 2.2.b, 2.2.c, 2.2.1, 2.2.2, 2.2.3(a), 2.2.3(c), 2.2.4, 2.2.5, and 2.4 to 2.12 of this appendix, and sections 5.1.3 and 5.1.4 of AHRI 1230–2010. The ''manufacturer's published instructions,'' as stated in section 8.2 of ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3) and ''manufacturer's installation instructions'' discussed in this appendix mean the manufacturer's installation instructions that come packaged with or appear in the labels applied to the unit. This does not include online manuals. Installation instructions that appear in the labels applied to the unit take precedence over installation instructions that are shipped with the unit.

For general requirements for the test procedure, refer to section 3.1 of this appendix, except for sections 3.1.3 and 3.1.4, which are requirements for indoor air volume and outdoor air volume. For indoor air volume and outdoor air volume requirements, refer instead to section 6.1.5 (except where section 6.1.5 refers to Table 8, refer instead to Table 4 of this appendix) and 6.1.6 of AHRI 1230–2010.

For the test method, refer to sections 3.3 to 3.5 and 3.7 to 3.13 of this appendix. For cooling mode and heating mode test conditions, refer to section 6.2 of AHRI 1230–2010. For calculations of seasonal performance descriptors, refer to section 4 of this appendix.

(B) For systems other than VRF, only a subset of the sections listed in this test procedure apply when testing and determining represented values for a particular unit. Table 1 shows the sections of the test procedure that apply to each system. This table is meant to assist manufacturers in finding the appropriate sections of the test procedure; the appendix sections rather than the table provide the specific requirements for testing, and given the varied nature of available units, manufacturers are responsible for determining which sections apply to each unit tested based on the unit's characteristics. To use this table, first refer to the sections listed under ''all units''. Then refer to additional requirements based on:

(1) System configuration(s),

(2) The compressor staging or modulation capability, and

(3) Any special features.

Testing requirements for space-constrained products do not differ from similar equipment that is not space-constrained and thus are not listed separately in this table. Air conditioners and heat pumps are not listed separately in this table, but heating procedures and calculations apply only to heat pumps.

Table 1 Informative Guidance for Using Appendix M

			Testing conditions	Testing procedures			Calculations		
			General	General	Cooling*	Heating**	General	Cooling*	Heating**
Requirements for all units (except VRF)			2.1; 2.2a-c; 2.2.1; 2.2.4; 2.2.4.1; 2.2.4.1 (1); 2.2.4.2; 2.2.5.1-5; 2.2.5.7-8; 2.3; 2.3.1; 2.3.2; 2.4; 2.4.1a,d; 2.5a-c; 2.5.1; 2.5.2 - 2.5.4.2; 2.5.5 – 2.13	3.1; 3.1.1-3; 3.1.5-9; 3.11; 3.12	3.3; 3.4; 3.5a-i	3.1.4.7; 3.1.10; 3.7a,b,d; 3.8a,d; 3.8.1; 3.9; 3.10	4.4; 4.5	4.1	4.2
Additional Requirements	System Configurations (more than one may apply)	Single-split system – blower coil	2.2a(1)		3.1.4.1.1; 3.1.4.1.1a,b; 3.1.4.2a-b; 3.1.4.3a-b	3.1.4.4.1; 3.1.4.4.2; 3.1.4.4.3a-b; 3.1.4.5.1; 3.1.4.5.2a-c; 3.1.4.6a-b			
		Single-split system - coil-only	2.2a(1); 2.2d,e; 2.4.2		3.1.4.1.1; 3.1.4.1.1c; 3.1.4.2c; 3.5.1	3.1.4.4.1; 3.1.4.4.2; 3.1.4.4.3c; 3.1.4.5.2d; 3.7c; 3.8b; 3.9f; 3.9.1b			
		Tri-split	2.2a(2)						
		Outdoor unit with no match	2.2c						
		Single-package	2.2.4.1(2); 2.2.5.6b; 2.4.2		3.1.4.1.1; 3.1.4.1.1a,b; 3.1.4.2a-b; 3.1.4.3a-b	3.1.4.4.1; 3.1.4.4.2; 3.1.4.4.3a-b; 3.1.4.5.1; 3.1.4.5.2a-c; 3.1.4.6a-b			
		Heat pump	2.2.5.6.a						
		Heating-only heat pump			3.1.4.1.1 Table 5	3.1.4.4.3			
		Two-capacity northern heat pump		3.1.4.4.2c; 3.1.4.5.2 c- d	3.2.3c	3.6.3			
		Triple-capacity northern heat pump			3.2.5	3.6.6			4.2.6

		SDHV (non-VRF)	2.2b; 2.4.1c; 2.5.4.3						
		Single- zone-multi-coil split and non-VRF multiple-split with duct	2.2a(1),(3); 2.2.3; 2.4.1b		3.1.4.1.1; 3.1.4.1.1a-b; 3.1.4.2a-b; 3.1.4.3a-b	3.1.4.4.1; 3.1.4.4.2; 3.1.4.4.3a-b; 3.1.4.5.1; 3.1.4.5.2a-c; 3.1.4.6a-b			
		Single-zone-multi-coil split and non-VRF multiple-split, ductless	2.2.a(1),(3); 2.2.3		3.1.4.1.2; 3.1.4.2d; 3.1.4.3c; 3.2.4c; 3.5c,g,h; 3.5.2; 3.8c	3.1.4.4.4; 3.1.4.5.2e; 3.1.4.6c; 3.6.4.c; 3.8c			
		VRF multiple-split† and VRF SDHV†	2.1; 2.2.a; 2.2.b; 2.2.c; 2.2.1; 2.2.2; 2.2.3(a); 2.2.3(c);, 2.2.4; 2.2.5; 2.4-2.12	3.1 (except 3.1.3, 3.1.4) 3.1.4.1.1c; 3.11-3.13	3.3-3.5	3.7–3.10	4.4; 4.5	4.1	4.2
	Modulation Capability	Single speed compressor, fixed air volume rate			3.2.1	3.6.1		4.1.1	4.2.1
		Single speed compressor, VAV fan			3.2.2	3.6.2		4.1.2	4.2.2
		Two-capacity compressor		3.1.10	3.2.3	3.6.3		4.1.3	4.2.3
		Variable speed compressor			3.2.4	3.6.4		4.1.4	4.2.4
	Special Features	Heat pump with heat comfort controller				3.6.5			4.2.5
		Units with a multi-speed outdoor fan	2.2.2						
		Single indoor unit having multiple indoor blowers			3.2.6	3.6.2; 3.6.7		4.1.5	4.2.7

*Does not apply to heating-only heat pumps.

**Applies only to heat pumps; not to air conditioners.

†Use AHRI 1230-2010 (incorporated by reference, see § 430.3), with the sections referenced in section 2(A) of this appendix, in conjunction with the sections set forth in the table to perform test setup, testing, and calculations for determining represented values for VRF multiple-split and VRF SDHV systems.

NOTE: For all units, use section 3.13 of this appendix for off mode testing procedures and section 4.3 of this appendix for off mode calculations. For all units subject to an EER standard, use section 4.6 of this appendix to determine the energy efficiency ratio.

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, single-zone-multi-coil or multi-circuit air conditioners and heat pumps, however, use as many 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 ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3).

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. 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 ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3), subject to the following additional requirements:

(1) When testing split systems, follow the requirements given in section 6.1.3.5 of AHRI 210/240–2008 (incorporated by reference, see § 430.3). For the vapor refrigerant line(s), use the insulation included with the unit; if no insulation is provided, use insulation meeting the specifications for the insulation in the installation instructions included with the unit by the manufacturer; if no insulation is included with the unit and the installation instructions do not contain provisions for insulating the line(s), fully insulate the vapor refrigerant line(s) with vapor proof insulation having an inside diameter that matches the refrigerant tubing and a nominal thickness of at least 0.5 inches. For the liquid refrigerant line(s), use the insulation included with the unit; if no insulation is provided, use insulation meeting the specifications for the insulation in the installation instructions included with the unit by the manufacturer; if no insulation is included with the unit and the installation instructions do not contain provisions for insulating the line(s), leave the liquid refrigerant line(s) exposed to the air for air conditioners and heat pumps that heat and cool; or, for heating-only heat pumps, insulate the liquid refrigerant line(s) with insulation having an inside diameter that

matches the refrigerant tubing and a nominal thickness of at least 0.5 inches. However, these requirements do not take priority over instructions for application of insulation for the purpose of improving refrigerant temperature measurement accuracy as required by sections 2.10.2 and 2.10.3 of this appendix. Insulation must be the same for the cooling and heating tests.

(2) When testing split systems, if the indoor unit does not ship with a cooling mode expansion device, test the system using the device as specified in the installation instructions provided with the indoor unit. If none is specified, test the system using a fixed orifice or piston type expansion device that is sized appropriately for the system.

(3) When testing triple-split systems (see section 1.2 of this appendix, Definitions), use the tubing length specified in section 6.1.3.5 of AHRI 210/240–2008 (incorporated by reference, see § 430.3) 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;

(4) When testing split systems having multiple indoor coils, connect each indoor blower coil unit to the outdoor unit using:

(a) 25 feet of tubing, or

(b) tubing furnished by the manufacturer, whichever is longer.

At least 10 feet of the system interconnection tubing shall be exposed to the outside conditions. If they are needed to make a secondary measurement of capacity or for verification of refrigerant charge, install refrigerant pressure measuring instruments as described in section 8.2.5 of ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3). Section 2.10 of this appendix specifies which secondary methods require refrigerant pressure measurements and section 2.2.5.5 of this appendix discusses use of pressure measurements to verify charge. 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, use the orientation for testing specified by the manufacturer in the certification report. 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 section 1 of this appendix, Definitions). 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 4, note 3 (see section 3.1.4 of this appendix). Except as noted in section 3.1.10 of this appendix, prevent the indoor air supplementary heating coils from operating during all tests. For uncased coils, create an enclosure using 1 inch fiberglass foil-faced ductboard having a nominal density of 6 pounds per cubic foot. Or alternatively, construct an enclosure using sheet metal or a similar material and insulating material having a thermal resistance (''R'' value) between 4 and 6 hr·ft^2· °F/Btu. Size the enclosure and seal between the coil and/or drainage pan and the interior of the enclosure as specified in installation instructions shipped with the unit. Also seal between the plenum and inlet and outlet ducts.

d. When testing a coil-only system, install a toroidal-type transformer to power the system's low-voltage components, complying with any additional requirements for the transformer mentioned in the installation manuals included with the unit by the system manufacturer. If the installation manuals do not provide specifications for the transformer, use a transformer having the following features:

(1) A nominal volt-amp rating such that the transformer is loaded between 25 and 90 percent of this rating for the highest level of power measured during the off mode test (section 3.13 of this appendix);

(2) Designed to operate with a primary input of 230 V, single phase, 60 Hz; and

(3) That provides an output voltage that is within the specified range for each low-voltage component. Include the power consumption of the components connected to the transformer as part of the total system power consumption during the off mode tests; do not include the power consumed by the transformer when no load is connected to it.

e. Test an outdoor unit with no match (*i.e.,* that is not distributed in commerce with any indoor units) using a coil-only indoor unit with a single cooling air volume rate whose coil has:

(1) Round tubes of outer diameter no less than 0.375 inches, and

(2) a normalized gross indoor fin surface (NGIFS) no greater than 1.0 square inches per British thermal unit per hour (sq. in./Btu/hr). NGIFS is calculated as follows:

$$NGIFS = 2 \times L_f \times W_f \times N_f \div \dot{Q}_c(95)$$

where:

L_f = Indoor coil fin length in inches, also height of the coil transverse to the tubes.
W_f = Indoor coil fin width in inches, also depth of the coil.
N_f = Number of fins.
$\dot{Q}_c(95)$ = the measured space cooling capacity of the tested outdoor unit/indoor unit combination as determined from the A2 or A Test whichever applies, Btu/h.

f. If the outdoor unit or the outdoor portion of a single-package unit has a drain pan heater to prevent freezing of defrost water, the heater shall be energized, subject to control to de-energize it when not needed by the heater's thermostat or the unit's control system, for all tests.

g. If pressure measurement devices are connected to a cooling/heating heat pump refrigerant circuit, the refrigerant charge M_t that could potentially transfer out of the connected pressure measurement systems (transducers, gauges, connections, and lines) between operating modes must be less than 2 percent of the factory refrigerant charge listed on the nameplate of the outdoor unit. If the outdoor unit nameplate has no listed refrigerant charge, or the heat pump is shipped without a refrigerant charge, use a factory refrigerant charge equal to 30 ounces per ton of certified cooling capacity. Use Equation 2.2–1 to calculate M_t for heat pumps that have a single expansion device located in the outdoor unit to serve each indoor unit, and use Equation 2.2–2 to calculate M_t for heat pumps that have two expansion devices per indoor unit.

$$\text{Equation 2.2-1} \qquad M_t = \rho * (V_5 * f_5 + V_6 * f_6 + V_3 + V_4 - V_2)$$

$$\text{Equation 2.2-2} \qquad M_t = \rho * (V_5 * f_5 + V_6 * f_6)$$

where:

V_i (i=2,3,4. . .) = the internal volume of the pressure measurement system (pressure lines, fittings, and gauge and/or transducer) at the location i (as indicated in Table 2), (cubic inches)
f_i (i=5,6) = 0 if the pressure measurement system is pitched upwards from the pressure tap location to the gauge or transducer, 1 if it is not.
ρ = the density associated with liquid refrigerant at 100 °F bubble point conditions (ounces per cubic inch)

TABLE 2—PRESSURE MEASUREMENT LOCATIONS

Location	
Compressor Discharge	1
Between Outdoor Coil and Outdoor Expansion Valve(s)	2
Liquid Service Valve	3
Indoor Coil Inlet	4
Indoor Coil Outlet	5

TABLE 2—PRESSURE MEASUREMENT LOCATIONS—Continued

Location	
Common Suction Port (i.e. vapor service valve)	6
Compressor Suction	7

Calculate the internal volume of each pressure measurement system using internal volume reported for pressure transducers and gauges in product literature, if available. If such information is not available, use the value of 0.1 cubic inches internal volume for each pressure transducer, and 0.2 cubic inches for each pressure gauge.

In addition, for heat pumps that have a single expansion device located in the outdoor unit to serve each indoor unit, the internal volume of the pressure system at location 2 (as indicated in Table 2) must be no more than 1 cubic inch. Once the pressure measurement lines are set up, no change should be made until all tests are finished.

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 1 and Table 20 of section 4.2 of this appendix for information on region IV.) For heat pumps that use a time-adaptive defrost control system (see section 1.2 of this appendix, Definitions), the manufacturer must specify in the certification report the frosting interval to be used during frost accumulation tests and provide the procedure for manually initiating the defrost at the specified time.

2.2.2 Special Requirements for Units Having a Multiple-Speed Outdoor Fan

Configure the multiple-speed outdoor fan according to the installation manual included with the unit by the manufacturer, 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 Ducted Systems Using a Single Indoor Section Containing Multiple Indoor Blowers That Would Normally Operate Using Two or More Indoor Thermostats

Because these systems will have more than one indoor blower and possibly multiple outdoor fans and compressor systems, references in this test procedure to a singular indoor blower, outdoor fan, and/or compressor means all indoor blowers, all outdoor fans, and all compressor systems that are energized during the test.

a. Additional requirements for multi-split air conditioners and heat pumps. 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), record the indoor coil(s) that are not providing heating or cooling during the test. For variable-speed systems, the manufacturer must designate in the certification report at least one indoor unit that is not providing heating or cooling for all tests conducted at minimum compressor speed.

b. Additional requirements for ducted split systems with a single indoor unit containing multiple indoor blowers (or for single-package units with an indoor section containing multiple indoor blowers) where the indoor blowers are designed to cycle on and off independently of one another and are not controlled such that all indoor blowers are modulated to always operate at the same air volume rate or speed. For any test where the system is operated at its lowest capacity—*i.e.,* the lowest total air volume rate allowed when operating the single-speed compressor or when operating at low compressor capacity—indoor blowers accounting for at least one-third of the full-load air volume rate must be turned off unless prevented by the controls of the unit. In such cases, turn off as many indoor blowers as permitted by the unit's controls. Where more than one option exists for meeting this ''off'' requirement, the manufacturer shall indicate in its certification report which indoor blower(s) are turned off. The chosen configuration shall remain unchanged for all tests conducted at the same lowest capacity configuration. For any indoor coil turned off during a test, cease forced airflow through any outlet duct connected to a switched-off indoor blower.

c. For test setups where the laboratory's physical limitations requires use of more than the required line length of 25 feet as listed in section 2.2.a(4) of this appendix, then the actual refrigerant line length used by the laboratory may exceed the required length and the refrigerant line length correction factors in Table 4 of AHRI 1230–2010 are applied to the cooling capacity measured for each cooling mode 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 so that the wet-bulb temperature is as listed in Tables 5 to 8. 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 5–8 list the applicable wet-bulb temperatures.

(2) Single-package 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 of this appendix. For dry coil tests on such units, it may be necessary to limit the moisture content of the air entering the outdoor coil of the unit to meet the requirements of section 3.4 of this appendix.

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 12 to 15. 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 (section 2.10.1 of this appendix) is used while testing a single-package 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

2.2.5.1 Instructions To Use for Charging

a. 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.

b. For systems consisting of an outdoor unit manufacturer's outdoor section and indoor section with differing charging procedures, adjust the refrigerant charge per the outdoor installation instructions.

c. For systems consisting of an outdoor unit manufacturer's outdoor unit and an independent coil manufacturer's indoor unit with differing charging procedures, adjust the refrigerant charge per the indoor unit's installation instructions. If instructions are provided only with the outdoor unit or are provided only with an independent coil manufacturer's indoor unit, then use the provided instructions.

2.2.5.2 Test(s) To Use for Charging

a. Use the tests or operating conditions specified in the manufacturer's installation instructions for charging. The manufacturer's installation instructions may specify use of tests other than the A or A_2 test for charging, but, unless the unit is a heating-only heat pump, the air volume rate must be determined by the A or A_2 test as specified in section 3.1 of this appendix.

b. If the manufacturer's installation instructions do not specify a test or operating conditions for charging or there are no manufacturer's instructions, use the following test(s):

(1) For air conditioners or cooling and heating heat pumps, use the A or A_2 test.

(2) For cooling and heating heat pumps that do not operate in the H1 or $H1_2$ test (*e.g.* due to shut down by the unit limiting devices) when tested using the charge determined at the A or A_2 test, and for heating-only heat pumps, use the H1 or $H1_2$ test.

2.2.5.3 Parameters To Set and Their Target Values

a. Consult the manufacturer's installation instructions regarding which parameters (*e.g.,* superheat) to set and their target values. If the instructions provide ranges of values, select target values equal to the midpoints of the provided ranges.

b. In the event of conflicting information between charging instructions (*i.e.,* multiple conditions given for charge adjustment where all conditions specified cannot be met), follow the following hierarchy.

(1) For fixed orifice systems:

(i) Superheat

(ii) High side pressure or corresponding saturation or dew-point temperature

(iii) Low side pressure or corresponding saturation or dew-point temperature

(iv) Low side temperature

(v) High side temperature

(vi) Charge weight

(2) For expansion valve systems:

(i) Subcooling

(ii) High side pressure or corresponding saturation or dew-point temperature

(iii) Low side pressure or corresponding saturation or dew-point temperature

(iv) Approach temperature (difference between temperature of liquid leaving condenser and condenser average inlet air temperature)

(v) Charge weight

c. If there are no installation instructions and/or they do not provide parameters and target values, set superheat to a target value of 12 °F for fixed orifice systems or set subcooling to a target value of 10 °F for expansion valve systems.

2.2.5.4 Charging Tolerances

a. If the manufacturer's installation instructions specify tolerances on target values for the charging parameters, set the values within these tolerances.

b. Otherwise, set parameter values within the following test condition tolerances for the different charging parameters:

1. Superheat: +/− 2.0 °F
2. Subcooling: +/− 2.0 °F
3. High side pressure or corresponding saturation or dew point temperature: +/− 4.0 psi or +/− 1.0 °F
4. Low side pressure or corresponding saturation or dew point temperature: +/− 2.0 psi or +/− 0.8 °F
5. High side temperature: +/−2.0 °F
6. Low side temperature: +/−2.0 °F
7. Approach temperature: +/− 1.0 °F
8. Charge weight: +/− 2.0 ounce

2.2.5.5 Special Charging Instructions

a. Cooling and Heating Heat Pumps

If, using the initial charge set in the A or A_2 test, the conditions are not within the range specified in manufacturer's installation instructions for the H1 or $H1_2$ test, make as small as possible an adjustment to obtain conditions for this test in the specified range. After this adjustment, recheck conditions in the A or A_2 test to confirm that they are still within the specified range for the A or A_2 test.

b. Single-Package Systems

Unless otherwise directed by the manufacturer's installation instructions, install one or more refrigerant line pressure gauges during the setup of the unit, located depending on the parameters used to verify or set charge, as described:

(1) Install a pressure gauge at the location of the service valve on the liquid line if charging is on the basis of subcooling, or high side pressure or corresponding saturation or dew point temperature;

(2) Install a pressure gauge at the location of the service valve on the suction line if charging is on the basis of superheat, or low side pressure or corresponding saturation or dew point temperature.

Use methods for installing pressure gauge(s) at the required location(s) as indicated in manufacturer's instructions if specified.

2.2.5.6 Near-Azeotropic and Zeotropic Refrigerants.

Perform charging of near-azeotropic and zeotropic refrigerants only with refrigerant in the liquid state.

2.2.5.7 Adjustment of Charge Between Tests.

After charging the system as described in this test procedure, use the set refrigerant charge for all tests used to determine performance. Do not adjust the refrigerant charge at any point during testing. If measurements indicate that refrigerant charge has leaked during the test, repair the refrigerant leak, repeat any necessary set-up steps, and repeat all tests.

2.3 Indoor Air Volume Rates.

If a unit's controls allow for overspeeding the indoor blower (usually on a temporary basis), take the necessary steps to prevent overspeeding during all tests.

2.3.1 Cooling Tests

a. Set indoor blower airflow-control settings (*e.g.,* fan motor pin settings, fan motor speed) according to the requirements that are specified in section 3.1.4 of this appendix.

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. Set indoor blower airflow-control settings (*e.g.,* fan motor pin settings, fan motor speed) according to the requirements that are specified in section 3.1.4 of this appendix.

b. Express the heating full-load 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 as described in section 2.4.1 of this appendix and, if installed, the inlet plenum described in section 2.4.2 of this appendix with thermal insulation having a nominal overall resistance (R-value) of at least 19 hr·ft²· °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, or multiple indoor blowers within a single indoor section, attach a plenum to each indoor coil or indoor blower outlet. In order to reduce the number of required airflow measurement apparati (section 2.6 of this appendix), each such apparatus may serve multiple outlet plenums connected to a single common duct leading to the apparatus. More than one indoor test room may be used, which may use one or more common ducts leading to one or more airflow measurement apparati 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 of this appendix) and airflow measuring apparatus are located downstream of the inlet(s) to the common duct. For multiple-circuit (or multi-circuit) systems for which each indoor coil outlet is measured separately and its outlet plenum is not connected to a common duct connecting multiple outlet plenums, the outlet air temperature grid and airflow measuring apparatus must be installed at each outlet plenum.

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 in Table 3. The limit depends only on the Cooling full-load air volume rate (see section 3.1.4.1.1 of this appendix) 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 9 of ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3) shows allowed options for the manifold configuration. The cross-sectional dimensions of plenum shall be equal to the dimensions of the indoor unit outlet. See Figures 7a, 7b, and 7c of ANSI/ASHRAE 37–2009 for the minimum length of the (each) outlet plenum and the locations for adding the static pressure taps for ducted blower coil indoor units and single-package systems. See Figure 8 of ANSI/ASHRAE 37–2009 for coil-only indoor units.

TABLE 3—SIZE OF OUTLET PLENUM FOR SMALL-DUCT HIGH-VELOCITY INDOOR UNITS

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 cross-sectional 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, a ducted blower coil indoor unit, or a single-package system. See Figures 7b and 7c of ANSI/ASHRAE 37–2009 for cross-sectional dimensions, the minimum length of the inlet plenum, and the locations of the static-pressure taps for ducted blower coil indoor units and single-package systems. See Figure 8 of ANSI/ASHRAE 37–2009 for coil-only indoor units. The inlet plenum duct size shall equal the size of the inlet opening of the air-handling (blower coil) unit or furnace. For a ducted blower coil indoor unit the set up may omit the inlet plenum if an inlet airflow prevention device is installed with a straight internally unobstructed duct on its outlet end with a minimum length equal to 1.5 times the square root of the cross-sectional area of the indoor unit inlet. See section 2.5.1.2 of this appendix for requirements for the locations of static pressure taps built into the inlet airflow prevention device. For all of these arrangements, make a manifold that connects the four static-pressure taps using one of the three configurations specified in section 2.4.1.d of this appendix. Never use an inlet plenum when testing non-ducted indoor units.

2.5 Indoor Coil Air Property Measurements and Airflow Prevention Devices

Follow instructions for indoor coil air property measurements as described in section 2.14 of this appendix, unless otherwise instructed in this section.

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 section 5.3 of ANSI/ASHRAE 41.1–2013 (incorporated by reference, see § 430.3) for guidance on constructing an air sampling device. No part of the air sampling device or the tubing transferring the sampled air to the sensor shall be within two inches of the test chamber floor, and the transfer tubing shall be insulated. The sampling device may also be used for measurement of dry bulb temperature by transferring the sampled air to a remotely located sensor(s). 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 sensor(s) may be used to determine the leaving air dry bulb temperature for all tests except:

(1) Cyclic tests; and

(2) Frost accumulation tests.

b. Install grids of temperature sensors to measure dry bulb temperatures of both the entering and leaving airstreams of the indoor unit. These grids of dry bulb temperature sensors may be used to measure average dry bulb temperature entering and leaving the indoor unit in all cases (as an alternative to the dry bulb sensor measuring the sampled air). The leaving airstream grid is required for measurement of average dry bulb temperature leaving the indoor unit for the two special cases noted above. The grids are also required to measure the air temperature distribution of the entering and leaving airstreams as described in sections 3.1.8 and 3.1.9 of this appendix. Two such grids may applied as a thermopile, to directly obtain the average temperature difference rather than directly measuring both entering and leaving average temperatures.

c. Use of airflow prevention devices. Use an inlet and outlet air damper box, or use an inlet upturned duct and an outlet air damper box when conducting one or both of the cyclic tests listed in sections 3.2 and 3.6 of this appendix on ducted systems. If not conducting any cyclic tests, an outlet air damper box is required when testing ducted and non-ducted heat pumps that cycle off the indoor blower during defrost cycles and there is no other means for preventing natural or forced convection through the indoor unit when the indoor blower is off. Never use an inlet damper box or an inlet upturned duct when testing non-ducted indoor units. An inlet upturned duct is a length of ductwork installed upstream from the inlet such that the indoor duct inlet opening, facing upwards, is sufficiently high to prevent natural convection transfer out of the duct. If an inlet upturned duct is used, install a dry bulb temperature sensor near the inlet opening of the indoor duct at a centerline location not higher than the lowest elevation of the duct edges at the inlet, and ensure that any pair of 5-minute averages of the dry bulb temperature at this location, measured at least every minute during the compressor OFF period of the cyclic test, do not differ by more than 1.0 °F.

2.5.1 Test Set-Up on the Inlet Side of the Indoor Coil: For Cases Where the Inlet Airflow Prevention Device Is Installed

a. Install an airflow prevention device as specified in section 2.5.1.1 or 2.5.1.2 of this appendix, whichever applies.

b. For an inlet damper box, locate the grid of entering air dry-bulb temperature sensors, if used, and 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. For an inlet upturned duct, locate the grid of entering air dry-bulb temperature sensors, if used, and the air sampling device, or the sensor used to measure the water vapor content of the inlet air, at a location at least one foot downstream from the beginning of the insulated portion of the duct but before the static pressure measurement.

2.5.1.1 If the Section 2.4.2 Inlet Plenum Is Installed

Construct the airflow prevention device having a cross-sectional flow area equal to or greater than the flow area of the inlet plenum. Install the airflow prevention device upstream of the inlet plenum and construct ductwork connecting it to the inlet plenum. If needed, use an adaptor plate or a transition duct section to connect the airflow prevention device with the inlet plenum. Insulate the ductwork and inlet plenum with thermal insulation that has a nominal overall resistance (R-value) of at least 19 hr · ft^2 · °F/Btu.

2.5.1.2 If the Section 2.4.2 Inlet Plenum Is Not Installed

Construct the airflow prevention device having a cross-sectional flow area equal to or greater than the flow area of the air inlet of the indoor unit. Install the airflow prevention device immediately upstream of the inlet of the indoor unit. If needed, use an adaptor plate or a short transition duct section to connect the airflow prevention device with the unit's air inlet. Add static pressure taps at the center of each face of a rectangular airflow prevention device, or at four evenly distributed locations along the circumference of an oval or round airflow prevention device. Locate the pressure taps at a distance from the indoor unit inlet equal to 0.5 times the square root of the cross sectional area of the indoor unit inlet. This location must be between the damper and the inlet of the indoor unit, if a damper is used. Make a manifold that connects the four static pressure taps using one of the configurations shown in Figure 9 of ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3). Insulate the ductwork with thermal insulation that has a nominal overall resistance (R-value) of at least 19 hr · ft^2 · °F/Btu.

2.5.2 Test Set-Up on the Inlet Side of the Indoor Unit: for Cases Where No Airflow Prevention Device 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 of this appendix, preferably at the entrance plane of the inlet plenum. If the section 2.4.2 inlet plenum is not used (*i.e.* for non-ducted units) locate a grid approximately 6 inches upstream of the indoor unit inlet. In the case of a system having multiple non-ducted indoor units, do this for each indoor unit. 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

Fabricate pressure taps meeting all requirements described in section 6.5.2 of ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3) and illustrated in Figure 2A of AMCA 210–2007 (incorporated by reference, see § 430.3), however, if adhering strictly to the description in section 6.5.2 of ANSI/ASHRAE 37–2009, the minimum pressure tap length of 2.5 times the inner diameter of Figure 2A of AMCA 210–2007 is waived. 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 airflow prevention device. For non-ducted indoor units 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 of this appendix and the airflow measuring apparatus described below in section 2.6 of this appendix. 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·ft^2· °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). Turn off the sampler fan motor during the cyclic tests. Air leaving an indoor unit that is sampled by an air sampling device for remote water-vapor-content measurement must be returned to the interconnecting duct at a location:

(1) Downstream of the air sampling device;

(2) On the same side of the outlet air damper as the air sampling device; 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 of this appendix), 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 of this appendix. Mixing devices are described in sections 5.3.2 and 5.3.3 of ANSI/ASHRAE 41.1–2013 and section 5.2.2 of ASHRAE 41.2–1987 (RA 1992) (incorporated by reference, see § 430.3).

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 of this appendix. 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. If applicable, in lieu of installing a separate damper, use the outlet air damper box of sections 2.5 and 2.5.4.1 of this appendix 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, 5.3, 6, and 7 of ANSI/ASHRAE 41.1–2013 (incorporated by reference, see § 430.3).

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, 7.2, 7.3, and 7.4 of ASHRAE 41.6–2014 (incorporated by reference, see § 430.3). The temperature sensor (wick removed) must be accurate to within ±0.2 °F. If used, apply dew point hygrometers as specified in sections 4, 5, 6, 7.1, and 7.4 of ASHRAE 41.6–2014 (incorporated by reference, see § 430.3). 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 of this appendix), 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 airflow measuring apparatus as specified in section 6.2 and 6.3 of ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3). Place the static pressure taps and position the diffusion baffle (settling means) relative to the chamber inlet as indicated in Figure 12 of AMCA 210–2007 and/or Figure 14 of ASHRAE 41.2–1987 (RA 1992) (incorporated by reference, see § 430.3). When measuring the static pressure difference across nozzles and/or velocity pressure at nozzle throats using electronic pressure transducers and a data acquisition system, if high frequency fluctuations cause measurement variations to exceed the test tolerance limits specified in section 9.2 and Table 2 of ANSI/ASHRAE 37–2009, dampen the measurement system such that the time constant associated with response to a step change in measurement (time for the response to change 63% of the way from the initial output to the final output) is no longer than five seconds.

b. Connect the airflow measuring apparatus to the interconnecting duct section described in section 2.5.4 of this appendix. See sections 6.1.1, 6.1.2, and 6.1.4, and Figures 1, 2, and 4 of ANSI/ASHRAE 37–2009; and Figures D1, D2, and D4 of AHRI 210/240–2008 (incorporated by reference, see § 430.3) 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 ANSI/ASHRAE 37–2009 when testing triple-split units.)

2.7 Electrical Voltage Supply

Perform all tests at the voltage specified in section 6.1.3.2 of AHRI 210/240–2008 (incorporated by reference, see § 430.3) for ''Standard Rating Tests.'' If either the indoor or the outdoor unit has a 208V or 200V nameplate voltage and the other unit has a 230V nameplate rating, select the voltage supply on the outdoor unit for testing. Otherwise, supply each unit with its own nameplate voltage. 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 within 15 seconds prior to beginning an ON cycle. For ducted blower coil systems, the ON cycle lasts from compressor ON to indoor blower OFF. For ducted coil-only systems, the ON cycle lasts from compressor ON to compressor OFF. For non-ducted units, the ON cycle lasts from indoor blower ON to indoor blower 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 blower motor to within ±1.0 percent. If required according to sections 3.3, 3.4, 3.7, 3.9.1 of this appendix, and/or 3.10 of this appendix, this same instrumentation requirement (to determine the average electrical power consumption of the indoor blower motor to within ±1.0 percent) applies when testing air conditioners and heat pumps having a variable-speed constant-air-volume-rate indoor blower or a variable-speed, variable-air-volume-rate indoor blower.

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 of this appendix. In addition, for all steady-state tests, conduct a second, independent measurement of capacity as described in section 3.1.1 of this appendix. For split systems, use one of the following secondary measurement methods: Outdoor air enthalpy method, compressor calibration method, or refrigerant enthalpy method. For single-package 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) On the outlet side, an outlet plenum containing static pressure taps (sections 2.4, 2.4.1, and 2.5.3 of this appendix),

(2) An airflow measuring apparatus (section 2.6 of this appendix),

(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 of this appendix), and

(4) On the inlet side, a sampling device and temperature grid (section 2.11.b of this appendix).

c. During the free outdoor air tests described in sections 3.11.1 and 3.11.1.1 of this appendix, 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, install pressure gages to the access valves or to ports created from tapping into the suction and discharge lines according to sections 7.4.2 and 8.2.5 of ANSI/ASHRAE 37–2009. 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 and measure refrigerant properties according to section 7.4.2 and 8.2.5 of ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3). If removing the refrigerant before applying refrigerant lines and subsequently recharging, use the steps in 7.4.2 of ANSI/ASHRAE 37–2009 in addition to the methods of section 2.2.5 of this appendix to confirm the refrigerant charge. Use refrigerant temperature and pressure measuring instruments that meet the specifications given in sections 5.1.1 and 5.2 of ANSI/ASHRAE 37–2009.

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 ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3) 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 ANSI/ASHRAE 37–2009. Refrigerant flow measurement device(s), if used, must be either elevated at least two feet from the test chamber floor or placed upon insulating material having a total thermal resistance of at least R–12 and extending at least one foot laterally beyond each side of the device(s)' exposed surfaces.

2.11 Measurement of Test Room Ambient Conditions

Follow instructions for setting up air sampling device and aspirating psychrometer as described in section 2.14 of this appendix, unless otherwise instructed in this section.

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 ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3)), add instrumentation to permit measurement of the indoor test room dry-bulb temperature.

b. On the outdoor side, use one of the following two approaches, except that approach (1) is required for all evaporatively-cooled units and units that transfer condensate to the outdoor unit for evaporation using condenser heat.

(1) Use sampling tree air collection on all air-inlet surfaces of the outdoor unit.

(2) Use sampling tree air collection on one or more faces of the outdoor unit and demonstrate air temperature uniformity as follows. Install a grid of evenly-distributed thermocouples on each air-permitting face on the inlet of the outdoor unit. Install the thermocouples on the air sampling device, locate them individually or attach them to a wire structure. If not installed on the air sampling device, install the thermocouple grid 6 to 24 inches from the unit. The thermocouples shall be evenly spaced across the coil inlet surface and be installed to avoid sampling of discharge air or blockage of air recirculation. The grid of thermocouples must provide at least 16 measuring points per face or one measurement per square foot of inlet face area, whichever is less. This grid must be constructed and used as per section 5.3 of ANSI/ASHRAE 41.1–2013 (incorporated by reference, see § 430.3). The maximum difference between the average temperatures measured during the test period of any two pairs of these individual thermocouples located at any of the faces of the inlet of the outdoor unit, must not exceed 2.0 °F, otherwise approach (1) must be used.

The air sampling devices shall be located at the geometric center of each side; the branches may be oriented either parallel or perpendicular to the longer edges of the air inlet area. The air sampling devices in the outdoor air inlet location shall be sized such that they cover at least 75% of the face area of the side of the coil that they are measuring.

Air distribution at the test facility point of supply to the unit shall be reviewed and may require remediation prior to the beginning of testing. Mixing fans can be used to ensure adequate air distribution in the test room. If used, mixing fans shall be oriented such that they are pointed away from the air intake so that the mixing fan exhaust does not affect the outdoor coil air volume rate. Particular attention should be given to prevent the mixing fans from affecting (enhancing or limiting) recirculation of condenser fan exhaust air back through the unit. Any fan used to enhance test room air mixing shall not cause air velocities in the vicinity of the test unit to exceed 500 feet per minute.

The air sampling device may be larger than the face area of the side being measured, however care shall be taken to prevent discharge air from being sampled. If an air sampling device dimension extends beyond the inlet area of the unit, holes shall be blocked in the air sampling device to prevent sampling of discharge air. Holes can be blocked to reduce the region of coverage of the intake holes both in the direction of the trunk axis or perpendicular to the trunk axis. For intake hole region reduction in the direction of the trunk axis, block holes of one or more adjacent pairs of branches (the branches of a pair connect opposite each other at the same trunk location) at either the outlet end or the closed end of the trunk. For intake hole region reduction perpendicular to the trunk axis, block off the same number of holes on each branch on both sides of the trunk.

A maximum of four (4) air sampling devices shall be connected to each aspirating psychrometer. In order to proportionately divide the flow stream for multiple air sampling devices for a given aspirating psychrometer, the tubing or conduit conveying sampled air to the psychrometer shall be of equivalent lengths for each air sampling device. Preferentially, the air sampling device should be hard connected to the aspirating psychrometer, but if space constraints do not allow this, the assembly shall have a means of allowing a flexible tube to connect the air sampling device to the aspirating psychrometer. The tubing or conduit shall be insulated and routed to prevent heat transfer to the air stream. Any surface of the air conveying tubing in contact with surrounding air at a different temperature than the sampled air shall be insulated with thermal insulation with a nominal thermal resistance (R-value) of at least 19 hr · ft^2 · °F/Btu. Alternatively the conduit may have lower thermal resistance if additional sensor(s) are used to measure dry bulb temperature at the outlet of each air sampling device. No part of the air sampling device or the tubing conducting the sampled air to the sensors shall be within two inches of the test chamber floor.

Pairs of measurements (*e.g.,* dry bulb temperature and wet bulb temperature) used to determine water vapor content of sampled air shall be measured in the same location.

2.12 Measurement of Indoor Blower 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 ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3).

2.14 Air Sampling Device and Aspirating Psycrhometer Requirements

Air temperature measurements shall be made in accordance with ANSI/ASHRAE 41.1–2013, unless otherwise instructed in this section.

2.14.1 Air Sampling Device Requirements

The air sampling device is intended to draw in a sample of the air at the critical locations of a unit under test. It shall be constructed of stainless steel, plastic or other suitable, durable materials. It shall have a main flow trunk tube with a series of branch tubes connected to the trunk tube. Holes shall be on the side of the sampler facing the upstream direction of the air source. Other sizes and rectangular shapes can be used, and shall be scaled accordingly with the following guidelines:

(1) Minimum hole density of 6 holes per square foot of area to be sampled

(2) Sampler branch tube pitch (spacing) of 6 ± 3 in

(3) Manifold trunk to branch diameter ratio having a minimum of 3:1 ratio

(4) Hole pitch (spacing) shall be equally distributed over the branch (½ pitch from the closed end to the nearest hole)

(5) Maximum individual hole to branch diameter ratio of 1:2 (1:3 preferred)

The minimum average velocity through the air sampling device holes shall be 2.5 ft/s as determined by evaluating the sum of the open area of the holes as compared to the flow area in the aspirating psychrometer.

2.14.2 Aspirating Psychrometer

The psychrometer consists of a flow section and a fan to draw air through the flow section and measures an average value of the sampled air stream. At a minimum, the flow section shall have a means for measuring the dry bulb temperature (typically, a resistance temperature device (RTD) and a means for measuring the humidity (RTD with wetted sock, chilled mirror hygrometer, or relative humidity sensor). The aspirating psychrometer shall include a fan that either can be adjusted manually or automatically to maintain required velocity across the sensors.

The psychrometer shall be made from suitable material which may be plastic (such as polycarbonate), aluminum or other metallic materials. All psychrometers for a given system being tested, shall be constructed of the same material. Psychrometers shall be designed such that radiant heat from the motor (for driving the fan that draws sampled air through the psychrometer) does not affect sensor measurements. For aspirating psychrometers, velocity across the wet bulb sensor shall be 1000 ± 200 ft/min. For all other psychrometers, velocity shall be as specified by the sensor manufacturer.

3. Testing Procedures

3.1 General Requirements

If, during the testing process, an equipment set-up adjustment is made that would have altered the performance of the unit during any 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.

Use the testing procedures in this section to collect the data used for calculating

(1) Performance metrics for central air conditioners and heat pumps during the cooling season;

(2) Performance metrics for heat pumps during the heating season; and

(3) Power consumption metric(s) for central air conditioners and heat pumps during the off mode season(s).

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 the full-capacity cooling-mode test and (for a heat pump) the full-capacity heating-mode test, use one of the acceptable secondary methods specified in section 2.10 of this appendix to determine indoor space conditioning capacity. Calculate this secondary check of capacity according to section 3.11 of this appendix. 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 ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3) (and, if testing a coil-only system, compare capacities before making 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 of this appendix.

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 blower 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 AHRI 210/240–2008 (incorporated by reference, see § 430.3) when obtaining the airflow through the outdoor coil.

3.1.3.1 Double-Ducted

For products intended to be installed with the outdoor airflow ducted, the unit shall be installed with outdoor coil ductwork installed per manufacturer installation instructions and shall operate between 0.10 and 0.15 in H_2O external static pressure. External static pressure measurements shall be made in accordance with ANSI/ASHRAE 37–2009 section 6.4 and 6.5.

3.1.4 Airflow Through the Indoor Coil

Airflow setting(s) shall be determined before testing begins. Unless otherwise specified within this or its subsections, no changes shall be made to the airflow setting(s) after initiation of testing.

3.1.4.1 Cooling Full-Load Air Volume Rate

3.1.4.1.1. Cooling Full-Load Air Volume Rate for Ducted Units

Identify the certified cooling full-load air volume rate and certified instructions for setting fan speed or controls. If there is no certified Cooling full-load air volume rate, use a value equal to the certified cooling capacity of the unit times 400 scfm per 12,000 Btu/h. If there are no instructions for setting fan speed or controls, use the as-shipped settings. Use the following procedure to confirm and, if necessary, adjust the Cooling full-load air volume rate and the fan speed or control settings to meet each test procedure requirement:

a. For all ducted blower coil systems, except those having a constant-air-volume-rate indoor blower:

Step (1) Operate the unit under conditions specified for the A (for single-stage units) or A_2 test using the certified fan speed or controls settings, and adjust the exhaust fan of the airflow measuring apparatus to achieve the certified Cooling full-load air volume rate;

Step (2) Measure the external static pressure;

Step (3) If this external static pressure is equal to or greater than the applicable minimum external static pressure cited in Table 4, the pressure requirement is satisfied; proceed to step 7 of this section. If this external static pressure is not equal to or greater than the applicable minimum external static pressure cited in Table 4, proceed to step 4 of this section;

Step (4) Increase the external static pressure by adjusting the exhaust fan of the airflow measuring apparatus until either

(i) The applicable Table 4 minimum is equaled or

(ii) The measured air volume rate equals 90 percent or less of the Cooling full-load air volume rate, whichever occurs first;

Step (5) If the conditions of step 4 (i) of this section occur first, the pressure requirement is satisfied; proceed to step 7 of this section. If the conditions of step 4 (ii) of this section occur first, proceed to step 6 of this section;

Step (6) Make an incremental change to the setup of the indoor blower (*e.g.*, next highest fan motor pin setting, next highest fan motor speed) and repeat the evaluation process beginning above, at step 1 of this section. If the indoor blower setup cannot be further changed, increase the external static pressure by adjusting the exhaust fan of the airflow measuring apparatus until the applicable Table 4 minimum is equaled; proceed to step 7 of this section;

Step (7) The airflow constraints have been satisfied. Use the measured air volume rate as the Cooling full-load air volume rate. Use the final fan speed or control settings for all tests that use the Cooling full-load air volume rate.

b. For ducted blower coil systems with a constant-air-volume-rate indoor blower. 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 4 value that does not cause automatic shutdown of the indoor blower or air volume rate variation Q_{Var}, defined as follows, greater than 10 percent.

$$Q_{Var} = \left[\frac{Q_{max} - Q_{min}}{\left(\frac{Q_{max} + Q_{min}}{2}\right)}\right] * 100$$

where:

Q_{max} = maximum measured airflow value
Q_{min} = minimum measured airflow value
Q_{Var} = airflow variance, percent

Additional test steps as described in section 3.3.(e) of this appendix are required if the measured external static pressure exceeds the target value by more than 0.03 inches of water.

c. For coil-only indoor units. 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-load air volume rate.

TABLE 4—MINIMUM EXTERNAL STATIC PRESSURE FOR DUCTED BLOWER COIL SYSTEMS

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

[1] For air conditioners and air-conditioning heat pumps, the value certified by the manufacturer for the unit's cooling capacity when operated at the *A* or A_2 Test conditions.
[2] For heating-only heat pumps, the value certified by the manufacturer for the unit's heating 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 inches of water.
[4] See section 1.2 of this appendix, Definitions, 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 blower coil indoor unit 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.

d. For ducted systems having multiple indoor blowers within a single indoor section, obtain the full-load air volume rate with all indoor blowers operating unless prevented by the controls of the unit. In such cases, turn on the maximum number of indoor blowers permitted by the unit's controls. Where more than one option exists for meeting this "on" indoor blower requirement, which indoor blower(s) are turned on must match that specified in the certification report. Conduct section 3.1.4.1.1 setup steps for each indoor blower separately. If two or more indoor blowers are connected to a common duct as per section 2.4.1 of this appendix, temporarily divert their air volume to the test room when confirming or adjusting the setup configuration of individual indoor blowers. The allocation of the system's full-load air volume rate assigned to each "on" indoor blower must match that specified by the manufacturer in the certification report.

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

Identify the certified cooling minimum air volume rate and certified instructions for setting fan speed or controls. If there is no certified cooling minimum air volume rate, use the final indoor blower control settings as determined when setting the cooling full-load air volume rate, and readjust the exhaust fan of the airflow measuring apparatus if necessary to reset to the cooling full load air volume obtained in section 3.1.4.1 of this appendix. Otherwise, calculate the target external static pressure and follow instructions a, b, c, d, or e below. The target external static pressure, ΔP_{st_i}, for any test "i" with a specified air volume rate not equal to the Cooling full-load air volume rate is determined as follows:

$$\Delta P_{st\_i} = \Delta P_{st\_full} \left[\frac{Q_i}{Q_{full}}\right]^2$$

where:

ΔP_{st_i} = target minimum external static pressure for test i;
ΔP_{st_full} = minimum external static pressure for test A or A_2 (Table 4);
Q_i = air volume rate for test i; and

Q_{full} = Cooling full-load air volume rate as measured after setting and/or adjustment as described in section 3.1.4.1.1 of this appendix.

a. For a ducted blower coil system without a constant-air-volume indoor blower, adjust for external static pressure as follows:

Step (1) Operate the unit under conditions specified for the B1 test using the certified fan speed or controls settings, and adjust the exhaust fan of the airflow measuring apparatus to achieve the certified cooling minimum air volume rate;

Step (2) Measure the external static pressure;

Step (3) If this pressure is equal to or greater than the minimum external static pressure computed above, the pressure requirement is satisfied; proceed to step 7 of this section. If this pressure is not equal to or greater than the minimum external static pressure computed above, proceed to step 4 of this section;

Step (4) Increase the external static pressure by adjusting the exhaust fan of the airflow measuring apparatus until either

(i) The pressure is equal to the minimum external static pressure computed above or

(ii) The measured air volume rate equals 90 percent or less of the cooling minimum air volume rate, whichever occurs first;

Step (5) If the conditions of step 4 (i) of this section occur first, the pressure requirement is satisfied; proceed to step 7 of this section. If the conditions of step 4 (ii) of this section occur first, proceed to step 6 of this section;

Step (6) Make an incremental change to the setup of the indoor blower (*e.g.,* next highest fan motor pin setting, next highest fan motor speed) and repeat the evaluation process beginning above, at step 1 of this section. If the indoor blower setup cannot be further changed, increase the external static pressure by adjusting the exhaust fan of the airflow measuring apparatus until it equals the minimum external static pressure computed above; proceed to step 7 of this section;

Step (7) The airflow constraints have been satisfied. Use the measured air volume rate as the cooling minimum air volume rate. Use the final fan speed or control settings for all tests that use the cooling minimum air volume rate.

b. For ducted units with constant-air-volume indoor blowers, 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 an automatic shutdown of the indoor blower or air volume rate variation Q_{Var}, defined in section 3.1.4.1.1.b of this appendix, greater than 10 percent, while being as close to, but not less than the target minimum external static pressure. Additional test steps as described in section 3.3(e) of this appendix are required if the measured external static pressure exceeds the target value by more than 0.03 inches of water.

c. For ducted two-capacity coil-only systems, the cooling minimum air volume rate is the higher of (1) the rate specified by the installation instructions included with the unit by the manufacturer or (2) 75 percent of the cooling full-load air volume rate. During the laboratory tests on a coil-only (fanless) system, 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 blower 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 blower, use the lowest fan setting allowed for cooling.

e. For ducted systems having multiple indoor blowers within a single indoor section, operate the indoor blowers such that the lowest air volume rate allowed by the unit's controls is obtained when operating the lone single-speed compressor or when operating at low compressor capacity while meeting the requirements of section 2.2.3.b of this appendix for the minimum number of blowers that must be turned off. Using the target external static pressure and the certified air volume rates, follow the procedures described in section 3.1.4.2.a of this appendix if the indoor blowers are not constant-air-volume indoor blowers or as described in section 3.1.4.2.b of this appendix if the indoor blowers are constant-air-volume indoor blowers. The sum of the individual ''on'' indoor blowers' air volume rates is the cooling minimum air volume rate for the system.

3.1.4.3 Cooling Intermediate Air Volume Rate

Identify the certified cooling intermediate air volume rate and certified instructions for setting fan speed or controls. If there is no certified cooling intermediate air volume rate, use the final indoor blower control settings as determined when setting the cooling full load air volume rate, and readjust the exhaust fan of the airflow measuring apparatus if necessary to reset to the cooling full load air volume obtained in section 3.1.4.1 of this appendix. Otherwise, calculate target minimum external static pressure as described in section 3.1.4.2 of this appendix, and set the air volume rate as follows.

a. For a ducted blower coil system without a constant-air-volume indoor blower, adjust for external static pressure as described in section 3.1.4.2.a of this appendix for cooling minimum air volume rate.

b. For a ducted blower coil system with a constant-air-volume indoor blower, conduct the E_V Test at an external static pressure that does not cause an automatic shutdown of the indoor blower or air volume rate variation Q_{Var}, defined in section 3.1.4.1.1.b of this appendix, greater than 10 percent, while being as close to, but not less than the target minimum external static pressure. Additional test steps as described in section 3.3(e) of this appendix are required if the measured external static pressure exceeds the target value by more than 0.03 inches of water.

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-load air volume rate for:

(1) Ducted blower coil system heat pumps that do not have a constant-air-volume indoor blower, and that operate at the same airflow-control setting during both the A (or A_2) and the H1 (or $H1_2$) Tests;

(2) Ducted blower coil system heat pumps with constant-air-flow indoor blowers that provide the same air flow for the A (or A_2) and the H1 (or $H1_2$) Tests; and

(3) Ducted heat pumps that are tested with a coil-only indoor unit (except two-capacity northern heat pumps that are tested only at low capacity cooling—see section 3.1.4.4.2 of this appendix).

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. Use the final indoor blower control settings as determined when setting the Cooling full-load air volume rate, and readjust the exhaust fan of the airflow measuring apparatus if necessary to reset to the cooling full-load air volume obtained in section 3.1.4.1 of this appendix. For heat pumps that meet the above criterion ''2,'' test at an external static pressure that does not cause an automatic shutdown of the indoor blower or air volume rate variation Q_{Var}, defined in section 3.1.4.1.1.b of this appendix, greater than 10 percent, while being as close to, but not less than, the same Table 4 minimum external static pressure as was specified for the A (or A_2) cooling mode test. Additional test steps as described in section 3.9.1(c) of this appendix are required if the measured external static pressure exceeds the target value by more than 0.03 inches of water.

3.1.4.4.2. Ducted Heat Pumps Where the Heating and Cooling Full-Load Air Volume Rates Are Different Due to Changes in Indoor Blower Operation, *i.e.* Speed Adjustment by the System Controls

Identify the certified heating full-load air volume rate and certified instructions for setting fan speed or controls. If there is no certified heating full-load air volume rate, use the final indoor blower control settings as determined when setting the cooling full-load air volume rate, and readjust the exhaust fan of the airflow measuring apparatus if necessary to reset to the cooling full load air volume obtained in section 3.1.4.1 of this appendix. Otherwise, calculate target minimum external static pressure as described in section 3.1.4.2 of this appendix and set the air volume rate as follows.

a. For ducted blower coil system heat pumps that do not have a constant-air-volume indoor blower, adjust for external static pressure as described in section 3.1.4.2.a of this appendix for cooling minimum air volume rate.

b. For ducted heat pumps tested with constant-air-volume indoor blowers installed, conduct all tests that specify the heating full-load air volume rate at an external static pressure that does not cause an automatic shutdown of the indoor blower or air volume rate variation Q_{Var}, defined in section 3.1.4.1.1.b of this appendix, greater than 10 percent, while being as close to, but not less than the target minimum external static pressure. Additional test steps as described in section 3.9.1(c) of this appendix are required if the measured external static pressure exceeds the target value by more than 0.03 inches of water.

c. When testing ducted, two-capacity blower coil system northern heat pumps (see section 1.2 of this appendix, Definitions), use the appropriate approach of the above two cases. For coil-only system northern heat pumps, the heating full-load air volume rate is the lesser of the rate specified by the manufacturer in the installation instructions included with the unit 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.

d. For ducted systems having multiple indoor blowers within a single indoor section, obtain the heating full-load air volume rate using the same ''on'' indoor blowers as used for the Cooling full-load air volume rate. Using the target external static pressure and the certified air volume rates, follow the procedures as described in section 3.1.4.4.2.a of this appendix if the indoor blowers are not constant-air-volume indoor blowers or as described in section 3.1.4.4.2.b of this appendix if the indoor blowers are constant-air-volume indoor blowers. The sum of the individual ''on'' indoor blowers' air volume rates is the heating full load air volume rate for the system.

3.1.4.4.3. Ducted Heating-Only Heat Pumps

Identify the certified heating full-load air volume rate and certified instructions for setting fan speed or controls. If there is no certified heating full-load air volume rate, use a value equal to the certified heating capacity of the unit times 400 scfm per 12,000 Btu/h. If there are no instructions for setting fan speed or controls, use the as-shipped settings.

a. For all ducted heating-only blower coil system heat pumps, except those having a constant-air-volume-rate indoor blower. Conduct the following steps only during the first test, the H1 or $H1_2$ Test:

Step (1) Adjust the exhaust fan of the airflow measuring apparatus to achieve the certified heating full-load air volume rate.

Step (2) Measure the external static pressure.

Step (3) If this pressure is equal to or greater than the Table 4 minimum external static pressure that applies given the heating-only heat pump's rated heating capacity, the pressure requirement is satisfied; proceed to step 7 of this section. If this pressure is not equal to or greater than the applicable Table 4 minimum external static pressure, proceed to step 4 of this section;

Step (4) Increase the external static pressure by adjusting the exhaust fan of the airflow measuring apparatus until either (i) the pressure is equal to the applicable Table 4 minimum external static pressure or (ii) the measured air volume rate equals 90 percent or less of the heating full-load air volume rate, whichever occurs first;

Step (5) If the conditions of step 4(i) of this section occur first, the pressure requirement is satisfied; proceed to step 7 of this section. If the conditions of step 4(ii) of this section occur first, proceed to step 6 of this section;

Step (6) Make an incremental change to the setup of the indoor blower (*e.g.,* next highest fan motor pin setting, next highest fan motor speed) and repeat the evaluation process beginning above, at step 1 of this section. If the indoor blower setup cannot be further changed, increase the external static pressure by adjusting the exhaust fan of the airflow measuring apparatus until it equals the applicable Table 4 minimum external static pressure; proceed to step 7 of this section;

Step (7) The airflow constraints have been satisfied. Use the measured air volume rate as the heating full-load air volume rate. Use the final fan speed or control settings for all tests that use the heating full-load air volume rate.

b. For ducted heating-only blower coil system heat pumps having a constant-air-volume-rate indoor blower. For all tests that specify the heating full-load air volume rate, obtain an external static pressure that does not cause an automatic shutdown of the indoor blower or air volume rate variation Q_{Var}, defined in section 3.1.4.1.1.b of this appendix, greater than 10 percent, while being as close to, but not less than, the applicable Table 4 minimum. Additional test steps as described in section 3.9.1(c) of this appendix are required if the measured external static pressure exceeds the target value by more than 0.03 inches of water.

c. For ducted heating-only coil-only system heat pumps in 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-load 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.

3.1.4.5 Heating Minimum Air Volume Rate

3.1.4.5.1. Ducted Heat Pumps Where the Heating and Cooling Minimum Air Volume Rates Are the Same

a. Use the cooling minimum air volume rate as the heating minimum air volume rate for:

(1) Ducted blower coil system heat pumps that do not have a constant-air-volume indoor blower, and that operate at the same airflow-control setting during both the A_1 and the $H1_1$ tests;

(2) Ducted blower coil system heat pumps with constant-air-flow indoor blowers installed that provide the same air flow for the A_1 and the $H1_1$ Tests; and

(3) Ducted coil-only system heat pumps.

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. Use the final indoor blower control settings as determined when setting the cooling minimum air volume rate, and readjust the exhaust fan of the airflow measuring apparatus if necessary to reset to the cooling minimum air volume rate obtained in section 3.1.4.2 of this appendix. For heat pumps that meet the above criterion ''2,'' test at an external static pressure that does not cause an automatic shutdown of the indoor blower or air volume rate variation Q_{Var}, defined in section 3.1.4.1.1.b of this appendix, greater than 10 percent, while being as close to, but not less than, the same target minimum external static pressure as was specified for the A_1 cooling mode test. Additional test steps as described in section 3.9.1(c) of this appendix are required if the measured external static pressure exceeds the target value by more than 0.03 inches of water.

3.1.4.5.2. Ducted Heat Pumps Where the Heating and Cooling Minimum Air Volume Rates Are Different Due to Changes in Indoor Blower Operation, *i.e.* Speed Adjustment by the System Controls

Identify the certified heating minimum air volume rate and certified instructions for setting fan speed or controls. If there is no certified heating minimum air volume rate, use the final indoor blower control settings as determined when setting the cooling minimum air volume rate, and readjust the exhaust fan of the airflow measuring apparatus if necessary to reset to the cooling minimum air volume obtained in section 3.1.4.2 of this appendix. Otherwise, calculate the target minimum external static pressure as described in section 3.1.4.2 of this appendix.

a. For ducted blower coil system heat pumps that do not have a constant-air-volume indoor blower, adjust for external static pressure as described in section 3.1.4.2.a of this appendix for cooling minimum air volume rate.

b. For ducted heat pumps tested with constant-air-volume indoor blowers installed, 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 an automatic shutdown of the indoor blower while being as close to, but not less than the air volume rate variation Q_{Var}, defined in section 3.1.4.1.1.b of this appendix, greater than 10 percent, while being as close to, but not less than the target minimum external static pressure. Additional test steps as described in section 3.9.1.c of this appendix are required if the measured external static pressure exceeds the target value by more than 0.03 inches of water.

c. For ducted two-capacity blower coil system northern heat pumps, use the appropriate approach of the above two cases.

d. For ducted two-capacity coil-only system heat pumps, use the cooling minimum air volume rate as the heating minimum air volume rate. For ducted two-capacity coil-only system northern heat pumps, use the cooling full-load air volume rate as the heating minimum air volume rate.

For ducted two-capacity heating-only coil-only system heat pumps, the heating minimum air volume rate is the higher of the rate specified by the manufacturer in the test setup instructions included with the unit or 75 percent of the heating full-load air volume rate. During the laboratory tests on a coil-only system, 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 blower 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 blower, use the lowest fan setting allowed for heating.

f. For ducted systems with multiple indoor blowers within a single indoor section, obtain the heating minimum air volume rate using the same "on" indoor blowers as used for the cooling minimum air volume rate. Using the target external static pressure and the certified air volume rates, follow the procedures as described in section 3.1.4.5.2.a of this appendix if the indoor blowers are not constant-air-volume indoor blowers or as described in section 3.1.4.5.2.b of this appendix if the indoor blowers are constant-air-volume indoor blowers. The sum of the individual "on" indoor blowers' air volume rates is the heating full-load air volume rate for the system.

3.1.4.6 Heating Intermediate Air Volume Rate

Identify the certified heating intermediate air volume rate and certified instructions for setting fan speed or controls. If there is no certified heating intermediate air volume rate, use the final indoor blower control settings as determined when setting the heating full-load air volume rate, and readjust the exhaust fan of the airflow measuring apparatus if necessary to reset to the cooling full load air volume obtained in section 3.1.4.2 of this appendix. Calculate the target minimum external static pressure as described in section 3.1.4.2 of this appendix.

a. For ducted blower coil system heat pumps that do not have a constant-air-volume indoor blower, adjust for external static pressure as described in section 3.1.4.2.a of this appendix for cooling minimum air volume rate.

b. For ducted heat pumps tested with constant-air-volume indoor blowers installed, conduct the $H2_V$ Test at an external static pressure that does not cause an automatic shutdown of the indoor blower or air volume rate variation Q_{Var}, defined in section 3.1.4.1.1.b of this appendix, greater than 10 percent, while being as close to, but not less than the target minimum external static pressure. Additional test steps as described in section 3.9.1(c) of this appendix are required if the measured external static pressure exceeds the target value by more than 0.03 inches of water.

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

The manufacturer must specify the heating nominal air volume rate and the instructions for setting fan speed or controls. Calculate target minimum external static pressure as described in section 3.1.4.2 of this appendix. Make adjustments as described in section 3.1.4.6 of this appendix for heating intermediate air volume rate so that the target minimum external static pressure is met or exceeded.

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 ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3)), 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. Dew point shall be within 2 °F of the required inlet conditions.

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 ANSI/ASHRAE 37–2009. When using the outdoor air enthalpy method, follow sections 7.7.2.1 and 7.7.2.2 of ANSI/ASHRAE 37–2009 to calculate the air volume rate through the outdoor coil. To express air volume rates in terms of standard air, use:

Equation 3-1

$$\overline{\dot{V}_s} = \frac{\overline{\dot{V}_{mx}}}{0.075\,\frac{lbm_{da}}{ft^3} * v_n' * [1+W_n]} = \frac{\overline{\dot{V}_{mx}}}{0.075\,\frac{lbm_{da}}{ft^3} * v_n}$$

Where:

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

$\overline{V}_{mx}$ = 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 ANSI/ASHRAE 37–2009, the second IP equation for Q_{mi} should read

$$Q_{mi} = 1097CA_n\sqrt{P_V v_n'}$$

3.1.7 Test Sequence

Before making test measurements used to calculate performance, operate the equipment for the "break-in" period specified in the certification report, which may not exceed 20 hours. Each compressor of the unit must undergo this "break-in" period. 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-load air 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 a 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, and the first test using the heating minimum air volume rate must 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 of this appendix. 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 of this appendix to minimize the temperature spread.

3.1.9 Requirement for the Air Temperature Distribution Entering the Outdoor Coil

Monitor the temperatures of the air entering the outdoor coil using air sampling devices and/or temperature sensor grids, maintaining the required tolerances, if applicable, as described in section 2.11 of this appendix.

3.1.10 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 of this appendix, 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 of this appendix, 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-load air 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 System Having a Single-Speed Compressor and Fixed Cooling Air Volume Rate

This set of tests is for single-speed-compressor units that do not have a cooling minimum air volume rate or a cooling intermediate air volume rate that is different than the cooling full load air volume rate. 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 (for outdoor units with no match) or 0.20 (for all other systems). Table 5 specifies test conditions for these four tests.

TABLE 5—COOLING MODE TEST CONDITIONS FOR UNITS HAVING A SINGLE-SPEED COMPRESSOR AND A FIXED COOLING AIR VOLUME RATE

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-load.[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 of this appendix.
[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 Where the Indoor Section Uses a Single Variable-Speed Variable-Air-Volume Rate Indoor Blower or Multiple Indoor Blowers

3.2.2.1 Indoor Blower Capacity Modulation That Correlates With the Outdoor Dry Bulb Temperature or Systems With a Single Indoor Coil but Multiple Indoor Blowers

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_Dc that exceeds the default C_Dc or if the two optional tests are not conducted, assign C_Dc the default value of 0.20.

3.2.2.2 Indoor Blower 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 of this appendix and Table 5. Use a cooling full-load air volume rate that represents a normal 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 6—COOLING MODE TEST CONDITIONS FOR UNITS WITH A SINGLE-SPEED COMPRESSOR THAT MEET THE SECTION 3.2.2.1 INDOOR UNIT REQUIREMENTS

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 of this appendix.
[3] Defined in section 3.1.4.2 of this appendix.
[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 5 °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 Section 1.2 of This Appendix, Definitions)

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_Dc that exceeds the default C_Dc or if the two optional tests are not conducted, assign C_Dc the default value of 0.20. Table 6 specifies test conditions for these six tests.

b. For units having a variable speed indoor blower 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 installation. Additionally, if conducting the 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 section 1.2 of this appendix, Definitions) in the same way as a single speed heat pump with the unit operating exclusively at low compressor capacity (see section 3.2.1 of this appendix and Table 5).

d. If a two-capacity air conditioner or heat pump locks out low-capacity operation at higher outdoor temperatures, then use the two 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 CD^c (k = 2) that exceeds the default CD^c (k = 2) or if the two optional tests are not conducted, assign CD^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 7—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]).
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 of this appendix.
[3] Defined in section 3.1.4.2 of this appendix.
[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_Dc that exceeds the default C_Dc or if the two optional tests are not conducted, assign C_Dc the default value of 0.25. Table 8 specifies test conditions for these seven tests. The compressor shall operate at the same cooling full speed, measured by RPM or power input frequency (Hz), for both the A_2 and B_2 tests. The compressor shall operate at the same cooling minimum speed, measured by RPM or power input frequency (Hz), for the B_1, F_1, G_1, and I_1 tests. Determine the cooling intermediate compressor speed cited in Table 8 using:

$$\text{Cooling intermediate speed} = \text{Cooling minimum speed} + \frac{\text{Cooling full speed} - \text{Cooling minimum speed}}{3}$$

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 blower 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 installation. Additionally, if conducting the 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 8 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 8 E_V Test, a cooling-mode intermediate compressor speed that falls within ¼ and ¾ of the difference between the full 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 8—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	Cooling Full	Cooling Full-Load.[2]
B_2 Test—required (steady, wet coil).	80	67	82	[1] 65	Cooling Full	Cooling Full-Load.[2]
E_V Test—required (steady, wet coil).	80	67	87	[1] 69	Cooling Intermediate	Cooling Intermediate.[3]
B_1 Test—required (steady, wet coil).	80	67	82	[1] 65	Cooling Minimum	Cooling Minimum.[4]
F_1 Test—required (steady, wet coil).	80	67	67	[1] 53.5	Cooling Minimum	Cooling Minimum.[4]
G_1 Test[5]—optional (steady, dry-coil).	80	([6])	67		Cooling Minimum	Cooling Minimum.[4]
I_1 Test[5]—optional (cyclic, dry-coil).	80	([6])	67		Cooling 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 of this appendix.
[3] Defined in section 3.1.4.3 of this appendix.
[4] Defined in section 3.1.4.2 of this appendix.
[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.

3.2.5 Cooling Mode Tests for Northern Heat Pumps With Triple-Capacity Compressors

Test triple-capacity, northern heat pumps for the cooling mode in the same way as specified in section 3.2.3 of this appendix for units having a two-capacity compressor.

3.2.6 Tests for an Air Conditioner or Heat Pump Having a Single Indoor Unit Having Multiple Indoor Blowers and Offering Two Stages of Compressor Modulation

Conduct the cooling mode tests specified in section 3.2.3 of this appendix.

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 blower 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 section 1.2 of this appendix, Definitions):

(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 of this appendix cases where its control is required, the water vapor content of the air entering the outdoor coil.

Refer to section 3.11 of this appendix 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 ANSI/ASHRAE 37–2009 for the indoor air enthalpy method and the user-selected secondary method. Make said Table 3 measurements at equal intervals that span 5 minutes or less. Continue data sampling until reaching a 30-minute period (*e.g.,* seven consecutive 5-minute samples) where the test tolerances specified in Table 9 are satisfied. For those continuously recorded parameters, use the entire data set from the 30-minute interval to evaluate Table 9 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 and sensible cooling capacity as specified in sections 7.3.3.1 and 7.3.3.3 of ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3). To calculate capacity, use the averages of the measurements (*e.g.* inlet and outlet dry bulb and wet bulb temperatures measured at the psychrometers) that are continuously recorded for the same 30-minute interval used as described above to evaluate compliance with test tolerances. 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. Use the values of the specific heat of air given in section 7.3.3.1 of ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3) for calculation of the sensible cooling capacities. Assign the average total space cooling capacity, average sensible cooling capacity, and electrical power consumption over the 30-minute data collection interval to the variables $\dot{Q}_c^k(T)$, $\dot{Q}_{sc}^k(T)$ and $\dot{E}_c^k(T)$, respectively. For these three 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 full speed, k=1 to denote low capacity or minimum speed, and k=v to denote the intermediate speed.

d. For coil-only system tests, decrease $\dot{Q}_c^k(T)$ by

$$\frac{1250\ Btu/h}{1000\ scfm} * \overline{\dot{V}}_s$$

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

$$\frac{365\ W}{1000\ scfm} * \overline{\dot{V}}_s$$

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 9—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 [1]
Indoor dry-bulb, °F		
Entering temperature	2.0	0.5
Leaving temperature	2.0	
Indoor wet-bulb, °F		
Entering temperature	1.0	[2] 0.3
Leaving temperature	[2] 1.0	
Outdoor dry-bulb, °F		
Entering temperature	2.0	0.5
Leaving temperature	[3] 2.0	
Outdoor wet-bulb, °F		
Entering temperature	1.0	[4] 0.3
Leaving temperature	[3] 1.0	
External resistance to airflow, inches of water	0.05	[5] 0.02
Electrical voltage, % of rdg.	2.0	1.5
Nozzle pressure drop, % of rdg.	2.0	

[1] See section 1.2 of this appendix, Definitions.
[2] Only applies during wet coil tests; does not apply during steady-state, dry coil cooling mode tests.
[3] Only applies when using the outdoor air enthalpy method.
[4] Only applies during wet coil cooling mode tests where the unit rejects condensate to the outdoor coil.
[5] Only applies when testing non-ducted units.

e. For air conditioners and heat pumps having a constant-air-volume-rate indoor blower, 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 blower 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 blower 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 blower motor at ΔP_{min} using linear extrapolation:

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

(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 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 of this appendix 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 ANSI/ASHRAE 37–2009). In preparing for the section 3.5 cyclic tests of this appendix, record the average indoor-side air volume rate, $\dot{V}$, 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 blower (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 blower turned off (see section 3.5 of this appendix), include the electrical power used by the indoor blower motor among the recorded parameters from the 30-minute test.

c. If the temperature sensors used to provide the primary measurement of the indoor-side dry bulb temperature difference during the steady-state dry-coil test and the subsequent cyclic dry-coil test are different, include measurements of the latter sensors among the regularly sampled data. Beginning at the start of the 30-minute data collection period, measure and compute the indoor-side air dry-bulb temperature difference using both sets of instrumentation, ΔT (Set SS) and ΔT (Set CYC), for each equally spaced data sample. If using a consistent data sampling rate that is less than 1 minute, calculate and record minutely averages for the two temperature differences. If using a consistent sampling rate of one minute or more, calculate and record the two temperature differences from each data sample. After having recorded the seventh (i=7) set of temperature differences, calculate the following ratio using the first seven sets of values:

$$F_{CD} = \frac{1}{7}\sum_{i-6}^{i}\frac{\Delta T(Set\ SS)}{\Delta T(Set\ CYC)}$$

Each time a subsequent set of temperature differences is recorded (if sampling more frequently than every 5 minutes), calculate F_{CD} using the most recent seven sets of values. Continue these calculations until the 30-minute period is completed or until a value for F_{CD} is calculated that falls outside the allowable range of 0.94–1.06. If the latter occurs, immediately suspend the test and identify the cause for the disparity in the two temperature difference measurements. Recalibration of one or both sets of instrumentation may be required. If all the values for F_{CD} are within the allowable range, save the final value of the ratio from the 30-minute test as F_{CD}*. If the temperature sensors used to provide the primary measurement of the indoor-side dry bulb temperature difference during the steady-state dry-coil test and the subsequent cyclic dry-coil test are the same, set F_{CD}*= 1.

3.5 Test Procedures for the Cyclic Dry-Coil Cooling-Mode Tests (the D, D_1, D_2, and I_1 Tests)

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 blower, the manufacturer has the option of electing at the outset whether to conduct the cyclic test with the indoor blower enabled or disabled. Always revert to testing with the indoor blower disabled if cyclic testing with the fan enabled is unsuccessful.

a. For all cyclic tests, the measured capacity must be adjusted for the thermal mass stored in devices and connections located between measured points. Follow the procedure outlined in section 7.4.3.4.5 of ASHRAE 116–2010 (incorporated by reference, see § 430.3) to ensure any required measurements are taken.

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. If an upturned duct is used, measure the dry-bulb temperature at the inlet of the device at least once every minute and ensure that its test operating tolerance is within 1.0 °F for each compressor OFF period.

c. Sections 3.5.1 and 3.5.2 of this appendix specify airflow requirements through the indoor coil of ducted and non-ducted indoor units, respectively. In all cases, use the exhaust fan of the airflow measuring apparatus (covered under section 2.6 of this appendix) along with the indoor blower 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 blower 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 blower, 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 blower 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 indoor blower, temporarily remove the indoor blower.

e. Conduct three complete compressor OFF/ON cycles with the test tolerances given in Table 10 satisfied. Calculate the degradation coefficient C_D for each complete cycle. If all three C_D values are within 0.02 of the average C_D then stability has been achieved, and the highest C_D value of these three shall be used. If stability has not been achieved, conduct additional cycles, up to a maximum of eight cycles total, until stability has been achieved between three consecutive cycles. Once stability has been achieved, use the highest C_D value of the three consecutive cycles that establish stability. If stability has not been achieved after eight cycles, use the highest C_D from cycle one through cycle eight, or the default C_D, whichever is lower.

f. With regard to the Table 10 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 blower (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 blower coil systems tested with the unit's indoor blower operating for the cycling test, integrate electrical power from indoor blower OFF to indoor blower 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 10—TEST OPERATING AND TEST CONDITION TOLERANCES FOR CYCLIC DRY COIL COOLING MODE TESTS

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

[1] See section 1.2 of this appendix, Definitions.

[2] 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 blower that ramps, the tolerances listed for the external resistance to airflow apply from 30 seconds after achieving full speed until ramp down begins.

[3] Shall at no time exceed a wet-bulb temperature that results in condensate forming on the indoor coil.

[4] The test condition shall be the average nozzle pressure difference or velocity pressure measured during the steady-state dry coil test.

[5] Applies during the interval when at least one of the following—the compressor, the outdoor fan, or, if applicable, the indoor blower—are operating except for the first 30 seconds after compressor start-up.

If the Table 10 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,

$$q_{cyc,dry} = \frac{60 * \overline{\dot{V}} * C_{p,a} * \Gamma}{[v_n' * (1+W_n)]} = \frac{60 * \overline{\dot{V}} * C_{p,a} * \Gamma}{v_n} \quad \text{and} \quad \Gamma = F_{CD}^* \int_{\tau_1}^{\tau_2} [T_{a1}(\tau) - T_{a2}(\tau)] \delta\tau, \ hr * {}^\circ\mathrm{F}$$

Where,

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

$T_{a1}(\tau)$ = 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.

Adjust the total space cooling delivered, $q_{cyc,dry}$, according to calculation method outlined in section 7.4.3.4.5 of ASHRAE 116–2010 (incorporated by reference, see § 430.3).

3.5.1 Procedures When Testing Ducted Systems

The automatic controls that are installed in 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 the indoor blower of the test unit). For ducted coil-only systems rated based on using a fan time-delay relay, control the indoor coil airflow according to the OFF delay listed by the manufacturer in the certification report. For ducted units having a variable-speed indoor blower 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 coil-only systems, cycle the indoor coil airflow in unison with the cycling of the compressor. If air damper boxes are used, close them on the inlet and outlet side 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 blower. For ducted coil-only systems (excluding the special case where a variable-speed fan is temporarily removed), increase $e_{cyc,dry}$ by the quantity,

$$\text{Equation 3.5-2.} \qquad \frac{365\ W}{1000\ scfm} * \overline{\dot{V}}_s * [\tau_2 - \tau_1]$$

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

$$\text{Equation 3.5-3.} \qquad \frac{1250\ Btu/h}{1000\ scfm} * \overline{\dot{V}}_s * [\tau_2 - \tau_1]$$

where $\overline{\dot{V}}_s$ 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 blower 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 blower power measured during or following the dry coil steady-state test; or,

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

(1) Measure the electrical power consumed by the variable-speed indoor blower 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 midpoint 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 blower 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 the test setup instructions included with the unit by the manufacturer specifies a ramp interval that exceeds 45 seconds, use a 45-second ramp interval nonetheless when estimating the fan energy.

3.5.2 Procedures When Testing Non-Ducted Indoor Units

Do not use airflow prevention devices when conducting cyclic tests on non-ducted indoor 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 blower 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 blower 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 blower during the 3 minutes after compressor cutoff to the integrated cooling capacity, $q_{cyc,dry}$. For the case where the non-ducted indoor unit uses a variable-speed indoor blower 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 of this appendix for ducted units having a disabled variable-speed indoor blower.

3.5.3 Cooling-Mode Cyclic-Degradation Coefficient Calculation

Use the two 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 CD^c that exceeds the default CD^c or if the two optional tests are not conducted, assign CD^c the default value of 0.25 for variable-speed compressor systems and outdoor units with no match, and 0.20 for all other systems. 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.

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

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

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

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 and Fixed Heating Air Volume Rate

This set of tests is for single-speed-compressor heat pumps that do not have a heating minimum air volume rate or a heating intermediate air volume rate that is different than the heating full load air volume rate. 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 10.

TABLE 11—HEATING MODE TEST CONDITIONS FOR UNITS HAVING A SINGLE-SPEED COMPRESSOR AND A FIXED-SPEED INDOOR BLOWER, A CONSTANT AIR VOLUME RATE INDOOR BLOWER, OR NO INDOOR BLOWER

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]
H3 Test (required, steady)	70	60 (max)	17	15	Heating Full-load.[1]

[1] Defined in section 3.1.4.4 of this appendix.

[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 Single Indoor Unit Having Either (1) a Variable Speed, Variable-Air-Rate Indoor Blower Whose Capacity Modulation Correlates With Outdoor Dry Bulb Temperature or (2) Multiple Indoor Blowers

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 12. 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:

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

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

where:

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

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

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 of this appendix; 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 of this appendix; 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 of this appendix.

TABLE 12—TABLE HEATING MODE TEST CONDITIONS FOR UNITS WITH A SINGLE-SPEED COMPRESSOR THAT MEET THE SECTION 3.6.2 INDOOR UNIT REQUIREMENTS

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]
$H2_1$ 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 of this appendix.
[2] Defined in section 3.1.4.5 of this appendix.
[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 section 1.2 of this appendix, Definitions), Including Two-Capacity, Northern Heat Pumps (see section 1.2 of this appendix, Definitions)

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 of this appendix 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:

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

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

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 of this appendix. 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 of this appendix.

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 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 13 specifies test conditions for these nine tests.

TABLE 13—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[7], cyclic)	70	60 (max)	47	43	High	([3])
$H1_1$ Test (required)	70	60 (max)	47	43	Low	Heating Minimum.[1]
$H1C_1$ Test (optional, cyclic)	70	60 (max)	47	43	Low	([4])
$H2_2$ Test (required)	70	60 (max)	35	33	High	Heating Full-Load.[2]
$H2_1$ Test[5 6] (required)	70	60 (max)	35	33	Low	Heating Minimum.[1]
$H3_2$ Test (required, steady)	70	60 (max)	17	15	High	Heating Full-Load.[2]
$H3_1$ Test[5] (required, steady)	70	60 (max)	17	15	Low	Heating Minimum.[1]

[1] Defined in section 3.1.4.5 of this appendix.
[2] Defined in section 3.1.4.4 of this appendix.
[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.

[7] Required only if the heat pump locks out low capacity operation at lower outdoor temperatures.

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 ($H1_N$ 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 ($H1_2$) and an additional frost accumulation test ($H2_2$). If desired, 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 14. The compressor shall operate at the same heating full speed, measured by RPM or power input frequency (Hz), for the $H1_2$, $H2_2$ and $H3_2$ tests. For a cooling/heating heat pump, the compressor shall operate for the $H1_N$ test at a speed, measured by RPM or power input frequency (Hz), no lower than the speed used in the A_2 test if the tested $H1_2$ heating capacity is less than the tested cooling capacity in A_2 test. The compressor shall operate at the same heating minimum speed, measured by RPM or power input frequency (Hz), for the $H0_1$, $H1C_1$, and $H1_1$ tests. Determine the heating intermediate compressor speed cited in Table 14 using the heating mode full and minimum compressors speeds and:

$$\text{Heating intermediate speed} = \text{Heating minimum speed} + \frac{\text{Heating full speed} - \text{Heating minimum speed}}{3}$$

Where a tolerance on speed of plus 5 percent or the next higher inverter frequency step from the calculated value is allowed.

b. If the $H1_2$ test is conducted, set the 47 °F capacity and power input values used for calculation of HSPF equal to the measured values for that test:

$$\dot{Q}_{hcalc}^{k=2}(47) = \dot{Q}_h^{k=2}(47);\ \dot{E}_{hcalc}^{k=2}(47) = \dot{E}_h^{k=2}(47)$$

Where:

$\dot{Q}_{hcalc}^{k=2}(47)$ and $\dot{E}_{hcalc}^{k=2}(47)$ are the capacity and power input representing full-speed operation at 47 °F for the HSPF calculations,

$\dot{Q}_h^{k=2}(47)$ is the capacity measured in the $H1_2$ test, and

$\dot{E}_h^{k=2}(47)$ is the power input measured in the $H1_2$ test.

Evaluate the quantities $\dot{Q}_h^{k=2}(47)$ and from $\dot{E}_h^{k=2}(47)$ according to section 3.7.

Otherwise, if the $H1_N$ test is conducted using the same compressor speed (RPM or power input frequency) as the $H3_2$ test, set the 47 °F capacity and power input values used for calculation of HSPF equal to the measured values for that test:

$$\dot{Q}_{hcalc}^{k=2}(47) = \dot{Q}_h^{k=N}(47);\ \dot{E}_{hcalc}^{k=2}(47) = \dot{E}_h^{k=N}(47)$$

Where:

$\dot{Q}_{hcalc}^{k=2}(47)$ and $\dot{E}_{hcalc}^{k=2}(47)$ are the capacity and power input representing full-speed operation at 47 °F for the HSPF calculations,

$\dot{Q}_h^{k=N}(47)$ is the capacity measured in the $H1_N$ test, and

$\dot{E}_h^{k=N}(47)$ is the power input measured in the $H1_N$ test.

Evaluate the quantities $\dot{Q}_h^{k=N}(47)$ and from $\dot{E}_h^{k=N}(47)$ according to section 3.7.

Otherwise (if no high temperature test is conducted using the same speed (RPM or power input frequency) as the $H3_2$ test), calculate the 47 °F capacity and power input values used for calculation of HSPF as follows:

$$\dot{Q}_{hcalc}^{k=2}(47) = \dot{Q}_h^{k=2}(17) * (1 + 30^\circ F * CSF);$$

$$\dot{E}_{hcalc}^{k=2}(47) = \dot{E}_h^{k=2}(17) * (1 + 30^\circ F * PSF)$$

Where:

$\dot{Q}_{hcalc}^{k=2}(47)$ and $\dot{E}_{hcalc}^{k=2}(47)$ are the capacity and power input representing full-speed operation at 47 °F for the HSPF calculations,

$\dot{Q}_h^{k=2}(17)$ is the capacity measured in the $H3_2$ test,

$\dot{E}_h^{k=2}(17)$ is the power input measured in the $H3_2$ test,

CSF is the capacity slope factor, equal to 0.0204/°F for split systems and 0.0262/°F for single-package systems, and

PSF is the Power Slope Factor, equal to 0.00455/°F.

c. 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:

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

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

Where:

$\dot{Q}_{hcalc}^{k=2}(47)$ and $\dot{E}_{hcalc}^{k=2}(47)$ are the capacity and power input representing full-speed operation at 47 °F for the HSPF calculations,calculated as described in section b above.

$\dot{Q}_h^{k=2}(17)$ and $\dot{E}_h^{k=2}(17)$ are the capacity and power input measured in the $H3_2$ test.

d. Determine the quantities $\dot{Q}_h^{k=2}(17)$ and $\dot{E}_h^{k=2}(17)$ from the $H3_2$ test, determine the quantities $\dot{Q}_h^{k=2}(5)$ and $\dot{E}_h^{k=2}(5)$ from the $H4_2$ test, and evaluate all four according to section 3.10.

TABLE 14—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	Heating minimum.	Heating minimum.[1]
$H1_2$ test (optional, steady)	70	60(max)	47	43	Heating full[4]	Heating full-load.[3]
$H1_1$ test (required, steady)	70	60(max)	47	43	Heating minimum.	Heating minimum.[1]
$H1_N$ test (required, steady)	70	60(max)	47	43	Heating full	Heating full-load.[3]
$H1C_1$ test (optional, cyclic)	70	60(max)	47	43	Heating minimum.	([2])
$H2_2$ test (optional)	70	60(max)	35	33	Heating full[4]	Heating full-load.[3]
$H2_V$ test (required)	70	60(max)	35	33	Heating intermediate.	Heating intermediate.[5]
$H3_2$ test (required, steady)	70	60(max)	17	15	Heating full	Heating full-load.[3]

[1] Defined in section 3.1.4.5 of this appendix.
[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 $H1_1$ test.
[3] Defined in section 3.1.4.4 of this appendix.
[4] The same compressor speed used in the $H3_2$ test. The $H1_2$ test is not needed if the $H1_N$ test uses this same compressor speed.
[5] Defined in section 3.1.4.6 of this appendix.

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 section 1.2 of this appendix, Definitions) 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 3.1.10 of this appendix 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.)

3.6.6 Heating Mode Tests for Northern Heat Pumps With Triple-Capacity Compressors.

Test triple-capacity, northern heat pumps for the heating mode as follows:

a. Conduct one maximum-temperature test ($H0_1$), two high-temperature tests ($H1_2$ and $H1_1$), one frost accumulation test ($H2_2$), two low-temperature tests ($H3_2$, $H3_3$), and one minimum-temperature test ($H4_3$). 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 4.2.6 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 test to determine $\dot{Q}_h^{k=1}(35)$ and $\dot{E}_h^{k=1}(35)$ is to use the following equations to approximate this capacity and electrical power:

$$\dot{Q}_h^{k=1}(35) = 0.90 * \left\{\dot{Q}_h^{k=1}(17) + 0.6 * \left\lfloor \dot{Q}_h^{k=1}(47) - \dot{Q}_h^{k=1}(17) \right\rfloor \right\}$$

$$\dot{E}_h^{k=1}(35) = 0.985 * \left\{\dot{E}_h^{k=1}(17) + 0.6 * \left\lfloor \dot{E}_h^{k=1}(47) - \dot{E}_h^{k=1}(17) \right\rfloor \right\}$$

In evaluating the above equations, determine the quantities $\dot{Q}_h^{k=1}(47)$ from the $H1_1$ test and evaluate them according to section 3.7 of this appendix. 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 of this appendix. Use the paired values of $\dot{Q}_h^{k=1}(35)$ and $\dot{E}_h^{k=1}(35)$ derived from conducting the $H2_1$ frost accumulation test and evaluated as specified in section 3.9.1 of this appendix or use the paired values calculated using the above default equations, whichever contribute to a higher Region IV HSPF based on the DHRmin.

b. Conducting a frost accumulation test ($H2_3$) with the heat pump operating at its booster capacity is optional. If this optional test is not conducted, determine $\dot{Q}_h^{k=3}(35)$ and $\dot{E}_h^{k=3}(35)$ using the following equations to approximate this capacity and electrical power:

$$\dot{Q}_h^{k=3}(35) = QR_h^{k=2}(35) * \left\{\dot{Q}_h^{k=3}(17) + 1.20 * \left\lfloor \dot{Q}_h^{k=3}(17) - \dot{Q}_h^{k=3}(5) \right\rfloor \right\}$$

$$\dot{E}_h^{k=3}(35) = PR_h^{k=2}(35) * \left\{\dot{E}_h^{k=3}(17) + 1.20 * \left\lfloor \dot{E}_h^{k=3}(17) - \dot{E}_h^{k=3}(5) \right\rfloor \right\}$$

Where:

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

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

Determine the quantities $\dot{Q}_h^{k=2}(47)$ and $\dot{E}_h^{k=2}(47)$ from the H1$_2$ test and evaluate them according to section 3.7 of this appendix. Determine the quantities $\dot{Q}_h^{k=2}(35)$ and $\dot{E}_h^{k=2}(35)$ from the H2$_2$ test and evaluate them according to section 3.9.1 of this appendix. Determine the quantities $\dot{Q}_h^{k=2}(17)$ and $\dot{E}_h^{k=2}(17)$ from the H3$_2$ test, determine the quantities $\dot{Q}_h^{k=3}(17)$ and $\dot{E}_h^{k=3}(17)$ from the H3$_3$ test, and determine the quantities $\dot{Q}_h^{k=3}(5)$ and $\dot{E}_h^{k=3}(5)$ from the H4$_3$ test. Evaluate all six quantities according to section 3.10 of this appendix. Use the paired values of $\dot{Q}_h^{k=3}(35)$ and $\dot{E}_h^{k=3}(35)$ derived from conducting the H2$_3$ frost accumulation test and calculated as specified in section 3.9.1 of this appendix or use the paired values calculated using the above default equations, whichever contribute to a higher Region IV HSPF2 based on the DHRmin.

c. Conduct the optional high-temperature cyclic test (H1C$_1$) to determine the heating mode cyclic-degradation coefficient, C_D^h. A default value for C_D^h may be used in lieu of conducting the cyclic. The default value of C_D^h is 0.25. If a triple-capacity heat pump locks out low capacity operation at lower outdoor temperatures, conduct the high-temperature cyclic test (H1C$_2$) to determine the high-capacity heating mode cyclic-degradation coefficient, C_D^h (k=2). 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)]. Finally, if a triple-capacity heat pump locks out both low and high capacity operation at the lowest outdoor temperatures, conduct the low-temperature cyclic test (H3C$_3$) to determine the booster-capacity heating mode cyclic-degradation coefficient, C_D^h (k=3). The default C_D^h (k=3) is the same value as determined or assigned for the high-capacity cyclic-degradation coefficient, C_D^h [or equivalently, C_D^h (k=2)]. Table 15 specifies test conditions for all 13 tests.

TABLE 15—HEATING MODE TEST CONDITIONS FOR UNITS WITH A TRIPLE-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,[8] cyclic)	70	$60^{(max)}$	47	43	High	([3]).
H1$_1$ Test (required)	70	$60^{(max)}$	47	43	Low	Heating Minimum.[1]
H1C$_1$ Test (optional, cyclic)	70	$60^{(max)}$	47	43	Low	([4]).
H2$_3$ Test (optional, steady)	70	$60^{(max)}$	35	33	Booster	Heating Full-Load.[2]
H2$_2$ Test (required)	70	$60^{(max)}$	35	33	High	Heating Full-Load.[2]
H2$_1$ Test (required)	70	$60^{(max)}$	35	33	Low	Heating Minimum.[1]
H3$_3$ Test (required, steady)	70	$60^{(max)}$	17	15	Booster	Heating Full-Load.[2]
H3C$_3$ Test[5][6] (optional, cyclic)	70	$60^{(max)}$	17	15	Booster	([7]).
H3$_2$ Test (required, steady)	70	$60^{(max)}$	17	15	High	Heating Full-Load.[2]
H3$_1$ Test[5] (required, steady)	70	$60^{(max)}$	17	15	Low	Heating Minimum.[1]
H4$_3$ Test (required, steady)	70	$60^{(max)}$	5	$3^{(max)}$	Booster	Heating Full-Load.[2]

[1] Defined in section 3.1.4.5 of this appendix.
[2] Defined in section 3.1.4.4 of this appendix.
[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 4.2.6 HSPF2 calculations.
[6] If table note [5] applies, the section 3.6.6 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.
[7] 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 H3$_3$ test.
[8] Required only if the heat pump locks out low capacity operation at lower outdoor temperatures.

3.6.7 Tests for a Heat Pump Having a Single Indoor Unit Having Multiple Indoor Blowers and Offering Two Stages of Compressor Modulation

Conduct the heating mode tests specified in section 3.6.3 of this appendix.

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 blower 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 of this appendix 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 ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3) for the indoor air enthalpy method and the user-selected secondary method. Make said Table 3 measurements at equal intervals that span 5 minutes or less. Continue data sampling until a 30-minute period (*e.g.,* seven consecutive 5-minute samples) is reached where the test tolerances specified in Table 16 are satisfied. For those continuously recorded parameters, use the entire data set for the 30-minute interval when evaluating Table 16 compliance. Determine the average electrical power consumption of the heat pump over the same 30-minute interval.

TABLE 16—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 [1]
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	[2] 1.0	
External resistance to airflow, inches of water	0.05	[3] 0.02
Electrical voltage, % of rdg	2.0	1.5
Nozzle pressure drop, % of rdg	2.0	

[1] See section 1.2 of this appendix, Definitions.
[2] Only applies when the Outdoor Air Enthalpy Method is used.
[3] 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 ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3). To calculate capacity, use the averages of the measurements (*e.g.* inlet and outlet dry bulb temperatures measured at the psychrometers) that are continuously recorded for the same 30-minute interval used as described above to evaluate compliance with test tolerances. 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 of this appendix. Additionally, for the heating mode, use the superscript to denote results from the optional $H1_N$ test, if conducted.

c. For coil-only system heat pumps, increase $\dot{Q}_h^k(T)$ by

$$\frac{1250\,BTU/h}{1000\,scfm} * \bar{\dot{V}}_s$$

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

$$\frac{365\,W}{1000\,scfm} * \bar{\dot{V}}_s$$

where $\bar{\dot{V}}_s$ 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 16 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 cyclic heating mode test, which is described in section 3.8 of this appendix, record the average indoor-side air volume rate, $\dot{V}$, 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 blower motor ($\dot{E}_{fan,1}$):

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

(2) The heat pump has a (variable-speed) constant-air volume-rate indoor blower 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 blower 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 blower motor if the 30-minute test had been conducted at ΔP_{min} using linear extrapolation:

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

(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.

e. If the temperature sensors used to provide the primary measurement of the indoor-side dry bulb temperature difference during the steady-state dry-coil test and the subsequent cyclic dry-coil test are different, include measurements of the latter sensors among the regularly sampled data. Beginning at the start of the 30-minute data collection period, measure and compute the indoor-side air dry-bulb temperature difference using both sets of instrumentation, ΔT (Set SS) and ΔT (Set CYC), for each equally spaced data sample. If using a consistent data sampling rate that is less than 1 minute, calculate and record minutely averages for the two temperature differences. If using a consistent sampling rate of one minute or more, calculate and record the two temperature differences from each data sample. After having recorded the seventh (i=7) set of temperature differences, calculate the following ratio using the first seven sets of values:

$$F_{CD} = \frac{1}{7}\sum_{i-6}^{i}\frac{\Delta T(Set\ SS)}{\Delta T(Set\ CYC)}$$

Each time a subsequent set of temperature differences is recorded (if sampling more frequently than every 5 minutes), calculate F_{CD} using the most recent seven sets of values. Continue these calculations until the 30-minute period is completed or until a value for F_{CD} is calculated that falls outside the allowable range of 0.94–1.06. If the latter occurs, immediately suspend the test and identify the cause for the disparity in the two temperature difference measurements. Recalibration of one or both sets of instrumentation may be required. If all the values for F_{CD} are within the allowable range, save the final value of the ratio from the 30-minute test as F_{CD}*. If the temperature sensors used to provide the primary measurement of the indoor-side dry bulb temperature difference during the steady-state dry-coil test and the subsequent cyclic dry-coil test are the same, set F_{CD}*= 1.

3.8 Test Procedures for the Cyclic Heating Mode Tests (the H0C₁, H1C, H1C₁ and H1C₂ Tests)

a. Except as noted below, conduct the cyclic heating mode test as specified in section 3.5 of this appendix. 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 17 rather than Table 10. Record the outdoor coil entering wet-bulb temperature according to the requirements given in section 3.5 of this appendix for the outdoor coil entering dry-bulb temperature. Drop the subscript ''dry'' used in variables cited in section 3.5 of this appendix 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 of this appendix except for making the following changes:

(1) When evaluating Equation 3.5–1, use the values of $\dot{V}$, $C_{p,a}$,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

$$\Gamma \text{ using, } \Gamma = F_{CD}^{*}\int_{\tau_1}^{\tau_2}[T_{a1}(\tau) - T_{a2}(\tau)]\delta\tau,\ \ hr \times {}^{\circ} F,$$

where F_{CD}* is the value recorded during the section 3.7 steady-state test conducted at the same test condition.

b. For ducted coil-only system heat pumps (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 ($\dot{V}_s$) 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 blower 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 blower 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 required 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 ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3) 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:

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

where:

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

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

$$COP_{ss}(T_{cyc}) = \frac{\dot{Q}_h^k(T_{cyc})}{3.413\frac{Btu/h}{W} * \dot{E}_h^k(T_{cyc})}$$

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.

$$HLF = \frac{q_{cyc}}{\dot{Q}_h^k(T_{cyc}) * \Delta\tau_{cyc}}$$

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 17—TEST OPERATING AND TEST CONDITION TOLERANCES FOR CYCLIC HEATING MODE TESTS

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

[1] See section 1.2 of this appendix, Definitions.

[2] 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 blower that ramps, the tolerances listed for the external resistance to airflow shall apply from 30 seconds after achieving full speed until ramp down begins.

[3] 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.

[4] Applies during the interval that at least one of the following—the compressor, the outdoor fan, or, if applicable, the indoor blower—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 $H2_I$ tests)

a. Confirm that the defrost controls of the heat pump are set as specified in section 2.2.1 of this appendix. 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 section 1.2 of this appendix, Definitions), 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 blower 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 blower. If it is installed, use the outlet damper box described in section 2.5.4.1 of this appendix 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 18 during both the preliminary and official test periods. As noted in Table 18, 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 18) and

(2) When defrosting, plus these same first 10 minutes after defrost termination (sub-interval D, as described in Table 18). Evaluate compliance with Table 18 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 18 at equal intervals that span 5 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 blower 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 coil-only system heat pumps, 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 ANSI/ASHRAE 37–2009) at equal intervals that span 10 minutes or less. (**Note:** In the first printing of ANSI/ASHRAE 37–2009, 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 18—TEST OPERATING AND TEST CONDITION TOLERANCES FOR FROST ACCUMULATION HEATING MODE TESTS

	Test operating tolerance [1]		Test condition tolerance [1]
	Sub-interval H [2]	Sub-interval D [3]	Sub-interval H [2]
Indoor entering dry-bulb temperature, °F	2.0	[4] 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	....	[5] 0.02
Electrical voltage, % of rdg	2.0	....	1.5

[1] See section 1.2 of this appendix, Definitions.
[2] Applies when the heat pump is in the heating mode, except for the first 10 minutes after termination of a defrost cycle.
[3] 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.
[4] For heat pumps that turn off the indoor blower during the defrost cycle, the noted tolerance only applies during the 10 minute interval that follows defrost termination.
[5] 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:

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

Where,

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

$C_{p,a} = 0.24 + 0.444 \cdot W_n$, 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, ft³/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.

$$\Gamma = \int_{\tau_1}^{\tau_2} [T_{a2}(\tau) - T_{a1}(\tau)] d\tau,\ hr * °F$$

$T_{a1}(\tau)$ = 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 blower 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 blower 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, ft³ 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 ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3).

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 coil-only system heat pumps, increase $\dot{Q}_h^k(35)$ by,

$$\frac{1250\ Btu/h}{1000\ scfm} * \bar{\dot{V}}_s * \frac{\Delta\tau_a}{\Delta\tau_{FR}}$$

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

$$\frac{365\ W}{1000\ scfm} * \bar{\dot{V}}_s * \frac{\Delta\tau_a}{\Delta\tau_{FR}}$$

where $\bar{\dot{V}}_s$ 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 blower, 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 blower 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 blower 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 blower motor had the frost accumulation heating mode test been conducted at ΔP_{min} using linear extrapolation:

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

(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 of this appendix to the value of 1 in all cases except for heat pumps having a demand-defrost control system (see section 1.2 of this appendix, Definitions). For such qualifying heat pumps, evaluate F_{def} using,

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

where:

$\Delta\tau_{def}$ = the time between defrost terminations (in hours) or 1.5, whichever is greater. A value of 6 must be assigned to $\Delta\tau_{def}$ if this limit is reached during a frost accumulation test and the heat pump has not completed a defrost cycle.

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

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-load air 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 of this appendix 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 of this appendix, 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 Methodst

3.11.1 If Using the Outdoor Air Enthalpy Method as the Secondary Test Method

a. For all cooling mode and heating mode tests, first conduct a test without the outdoor air-side test apparatus described in section 2.10.1 of this appendix connected to the outdoor unit (''free outdoor air'' test).

b. For the first section 3.2 steady-state cooling mode test and the first section 3.6 steady-state heating mode test, conduct a second test in which the outdoor-side apparatus is connected (''ducted outdoor air'' test). No other cooling mode or heating mode tests require the ducted outdoor air test 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 the ducted outdoor air test for 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 Free Outdoor Air Test

a. For the free outdoor air test, connect the indoor air-side test apparatus to the indoor coil; do not connect 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 5 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.,* seven consecutive 5-minute samples) is obtained where the Table 9 or Table 16, whichever applies, test tolerances are satisfied.

b. For cases where a ducted outdoor air test is not required per section 3.11.1.b of this appendix, the free outdoor air test constitutes the ''official'' test for which validity is not based on comparison with a secondary test.

c. For cases where a ducted outdoor air test is required per section 3.11.1.b of this appendix, the following conditions must be met for the free outdoor air test to constitute a valid ''official'' test:

(1) Achieve the energy balance specified in section 3.1.1 of this appendix for the ducted outdoor air test (*i.e.,* compare the capacities determined using the indoor air enthalpy method and the outdoor air enthalpy method).

(2) The capacities determined using the indoor air enthalpy method from the ducted outdoor air and free outdoor tests must agree within 2 percent.

3.11.1.2 Ducted Outdoor Air Test

a. The test conditions and tolerances for the ducted outdoor air test are the same as specified for the free outdoor air test described in Section 3.11.1.1 of this appendix.

b. After collecting 30 minutes of steady-state data during the free outdoor air test, connect the outdoor air-side test apparatus to the unit for the ducted outdoor air test. 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 during the free outdoor air test. Collect 30 minutes of steady-state data after re-establishing equilibrium conditions.

c. During the ducted outdoor air test, at intervals of 5 minutes or less, measure the parameters required according to the indoor air enthalpy method and the outdoor air enthalpy method for the prescribed 30 minutes.

d. For cooling mode ducted outdoor air tests, calculate capacity based on outdoor air-enthalpy measurements as specified in sections 7.3.3.2 and 7.3.3.3 of ANSI/ASHRAE 37–2009 (incorporated by reference, see

§ 430.3). For heating mode ducted tests, 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 ANSI/ASHRAE Standard. Adjust the outdoor-side capacity according to section 7.3.3.4 of ANSI/ASHRAE 37–2009 to account for line losses when testing split systems. As described in section 8.6.2 of ANSI/ASHRAE 37–2009, use the outdoor air volume rate as measured during the ducted outdoor air tests to calculate capacity for checking the agreement with the capacity calculated using the indoor air enthalpy method.

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 5 minutes or less. Determine average flow rate or average capacity from data sampled over a 30-minute period where the Table 9 (cooling) or the Table 16 (heating) tolerances are satisfied. Otherwise, conduct the calibration tests according to sections 5, 6, 7, and 8 of ASHRAE 23.1–2010 (incorporated by reference, see § 430.3); sections 5, 6, 7, 8, 9, and 11 of ASHRAE 41.9–2011 (incorporated by reference, see § 430.3); and section 7.4 of ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3).

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 ANSI/ASHRAE 37–2009.

3.11.3 If Using the Refrigerant-Enthalpy Method as the Secondary Test Method

Conduct this secondary method according to section 7.5 of ANSI/ASHRAE 37–2009. 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 specified in § 430.23 (for a single unit) and in 10 CFR 429.16 (for a sample).

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

3.13 Laboratory Testing to Determine Off Mode Average Power Ratings

Voltage tolerances: As a percentage of reading, test operating tolerance shall be 2.0 percent and test condition tolerance shall be 1.5 percent (see section 1.2 of this appendix for definitions of these tolerances).

Conduct one of the following tests: If the central air conditioner or heat pump lacks a compressor crankcase heater, perform the test in section 3.13.1 of this appendix; if the central air conditioner or heat pump has a compressor crankcase heater that lacks controls and is not self-regulating, perform the test in section 3.13.1 of this appendix; if the central air conditioner or heat pump has a crankcase heater with a fixed power input controlled with a thermostat that measures ambient temperature and whose sensing element temperature is not affected by the heater, perform the test in section 3.13.1 of this appendix; if the central air conditioner or heat pump has a compressor crankcase heater equipped with self-regulating control or with controls for which the sensing element temperature is affected by the heater, perform the test in section 3.13.2 of this appendix.

3.13.1 This Test Determines the Off Mode Average Power Rating for Central Air Conditioners and Heat Pumps That Lack a Compressor Crankcase Heater, or Have a Compressor Crankcase Heating System That Can Be Tested Without Control of Ambient Temperature During the Test. This Test Has No Ambient Condition Requirements

a. Test Sample Set-up and Power Measurement: For coil-only systems, provide a furnace or modular blower that is compatible with the system to serve as an interface with the thermostat (if used for the test) and to provide low-voltage control circuit power. Make all control circuit connections between the furnace (or modular blower) and the outdoor unit as specified by the manufacturer's installation instructions. Measure power supplied to both the furnace or modular blower and power supplied to the outdoor unit. Alternatively, provide a compatible transformer to supply low-voltage control circuit power, as described in section 2.2.d of this appendix. Measure transformer power, either supplied to the primary winding or supplied by the secondary winding of the transformer, and power supplied to the outdoor unit. For blower coil and single-package systems, make all control circuit connections between components as specified by the manufacturer's installation instructions, and provide power and measure power supplied to all system components.

b. Configure Controls: Configure the controls of the central air conditioner or heat pump so that it operates as if connected to a building thermostat that is set to the OFF position. Use a compatible building thermostat if necessary to achieve this configuration. For a thermostat-controlled crankcase heater with a fixed power input, bypass the crankcase heater thermostat if necessary to energize the heater.

c. Measure $P2_x$: If the unit has a crankcase heater time delay, make sure that time delay function is disabled or wait until delay time has passed. Determine the average power from non-zero value data measured over a 5-minute interval of the non-operating central air conditioner or heat pump and designate the average power as $P2_x$, the heating season total off mode power.

d. Measure P_x for coil-only split systems and for blower coil split systems for which a furnace or a modular blower is the designated air mover: Disconnect all low-voltage wiring for the *outdoor* components and *outdoor* controls from the low-voltage transformer. Determine the average power from non-zero value data measured over a 5-minute interval of the power supplied to the (remaining) low-voltage components of the central air conditioner or heat pump, or low-voltage power, P_x. This power measurement does not include line power supplied to the outdoor unit. It is the line power supplied to the air mover, or, if a compatible transformer is used instead of an air mover, it is the line power supplied to the transformer primary coil. If a compatible transformer is used instead of an air mover and power output of the low-voltage secondary circuit is measured, P_x is zero.

e. Calculate *P2*: Set the number of compressors equal to the unit's number of single-stage compressors plus 1.75 times the unit's number of compressors that are not single-stage.

For single-package systems and blower coil split systems for which the designated air mover is not a furnace or modular blower, divide the heating season total off mode power ($P2_x$) by the number of compressors to calculate *P2*, the heating season per-compressor off mode power. Round *P2* to the nearest watt. The expression for calculating *P2* is as follows:

$$P2 = \frac{P2_x}{number\ of\ compressors}.$$

For coil-only split systems and blower coil split systems for which a furnace or a modular blower is the designated air mover, subtract the low-voltage power (P_x) from the heating season total off mode power ($P2_x$) and divide by the number of compressors to calculate *P2*, the heating season per-compressor off mode power. Round *P2* to the nearest watt. The expression for calculating *P2* is as follows:

$$P2 = \frac{P2_x - P_x}{number\ of\ compressors}.$$

f. Shoulder-season per-compressor off mode power, P1: If the system does not have a crankcase heater, has a crankcase heater without controls that is not self-regulating, or has a value for the crankcase heater turn-on temperature (as certified in the DOE Compliance Certification Database) that is higher than 71 °F, P1 is equal to P2.

Otherwise, de-energize the crankcase heater (by removing the thermostat bypass or otherwise disconnecting only the power supply to the crankcase heater) and repeat the measurement as described in section 3.13.1.c of this appendix. Designate the measured average power as $P1_x$, the shoulder season total off mode power.

Determine the number of compressors as described in section 3.13.1.e of this appendix.

For single-package systems and blower coil systems for which the designated air mover is not a furnace or modular blower, divide the shoulder season total off mode power ($P1_x$) by the number of compressors to calculate *P1*, the shoulder season per-compressor off mode power. Round *P1* to the nearest watt. The expression for calculating *P1* is as follows:

$$P1 = \frac{P1_x}{number\ of\ compressors}.$$

For coil-only split systems and blower coil split systems for which a furnace or a modular blower is the designated air mover, subtract the low-voltage power (P_x) from the shoulder season total off mode power ($P1_x$) and divide by the number of compressors to calculate *P1*, the shoulder season per-compressor off mode power. Round *P1* to the nearest watt. The expression for calculating *P1* is as follows:

$$P1 = \frac{P1_x - P_x}{number\ of\ compressors}.$$

3.13.2 This Test Determines the Off Mode Average Power Rating for Central Air Conditioners and Heat Pumps for Which Ambient Temperature Can Affect the Measurement of Crankcase Heater Power

a. Test Sample Set-up and Power Measurement: Set up the test and measurement as described in section 3.13.1.a of this appendix.

b. Configure Controls: Position a temperature sensor to measure the outdoor dry-bulb temperature in the air between 2 and 6 inches from the crankcase heater control temperature sensor or, if no such temperature sensor exists, position it in the air between 2 and 6 inches from the crankcase heater. Utilize the temperature measurements from this sensor for this portion of the test procedure. Configure the controls of the central air conditioner or heat pump so that it operates as if connected to a building thermostat that is set to the OFF position. Use a compatible building thermostat if necessary to achieve this configuration.

Conduct the test after completion of the B, B_1, or B_2 test. Alternatively, start the test when the outdoor dry-bulb temperature is at 82 °F and the temperature of the compressor shell (or temperature of each compressor's shell if there is more than one compressor) is at least 81 °F. Then adjust the outdoor temperature at a rate of change of no more than 20 °F per hour and achieve an outdoor dry-bulb temperature of 72 °F. Maintain this temperature within ±2 °F while making the power measurement, as described in section 3.13.2.c of this appendix.

c. Measure $P1_x$: If the unit has a crankcase heater time delay, make sure that time delay function is disabled or wait until delay time has passed. Determine the average power from non-zero value data measured over a 5-minute interval of the non-operating central air conditioner or heat pump and designate the average power as $P1_x$, the shoulder season total off mode power. For units with crankcase heaters which operate during this part of the test and whose controls cycle or vary crankcase heater power over time, the test period shall consist of three complete crankcase heater cycles or 18 hours, whichever comes first. Designate the average power over the test period as $P1_x$, the shoulder season total off mode power.

d. Reduce outdoor temperature: Approach the target outdoor dry-bulb temperature by adjusting the outdoor temperature at a rate of change of no more than 20 °F per hour. This target temperature is five degrees Fahrenheit less than the temperature specified by the manufacturer in the DOE Compliance Certification Database at which the crankcase heater turns on. Maintain the target temperature within ±2 °F while making the power measurement, as described in section 3.13.2.e of this appendix.

e. Measure $P2_x$: If the unit has a crankcase heater time delay, make sure that time delay function is disabled or wait until delay time has passed. Determine the average non-zero power of the non-operating central air conditioner or heat pump over a 5-minute interval and designate it as $P2_x$, the heating season total off mode power. For units with crankcase heaters whose controls cycle or vary crankcase heater power over time, the test period shall consist of three complete crankcase heater cycles or 18 hours, whichever comes first. Designate the average power over the test period as $P2_x$, the heating season total off mode power.

f. Measure P_x for coil-only split systems and for blower coil split systems for which a furnace or modular blower is the designated air mover: Disconnect all low-voltage wiring for the *outdoor* components and *outdoor* controls from the low-voltage transformer. Determine the average power from non-zero value data measured over a 5-minute interval of the power supplied to the (remaining) low-voltage components of the central air conditioner or heat pump, or low-voltage power, P_x.. This power measurement does not include line power supplied to the outdoor unit. It is the line power supplied to the air mover, or, if a compatible transformer is used instead of an air mover, it is the line power supplied to the transformer primary coil. If a compatible transformer is used instead of an air mover and power output of the low-voltage secondary circuit is measured, P_x is zero.

g. Calculate *P1*:

Set the number of compressors equal to the unit's number of single-stage compressors plus 1.75 times the unit's number of compressors that are not single-stage.

For single-package systems and blower coil split systems for which the air mover is not a furnace or modular blower, divide the shoulder season total off mode power ($P1_x$) by the number of compressors to calculate *P1*, the shoulder season per-compressor off mode power. Round to the nearest watt. The expression for calculating *P1* is as follows:

$$P1 = \frac{P1_x}{number\ of\ compressors}.$$

For coil-only split systems and blower coil split systems for which a furnace or a modular blower is the designated air mover, subtract the low-voltage power (P_x) from the shoulder season total off mode power ($P1_x$) and divide by the number of compressors to calculate *P1*, the shoulder season per-compressor off mode power. Round to the nearest watt. The expression for calculating *P1* is as follows:

$$P1 = \frac{P1_x - P_x}{number\ of\ compressors}.$$

h. Calculate *P2*:

Determine the number of compressors as described in section 3.13.2.g of this appendix.

For single-package systems and blower coil split systems for which the air mover is not a furnace, divide the heating season total off mode power ($P2_x$) by the number of compressors to calculate *P2*, the heating season per-compressor off mode power. Round to the nearest watt. The expression for calculating *P2* is as follows:

$$P2 = \frac{P2_x}{number\ of\ compressors}.$$

For coil-only split systems and blower coil split systems for which a furnace or a modular blower is the designated air mover, subtract the low-voltage power (P_x) from the heating season total off mode power ($P2_x$) and divide by the number of compressors to calculate *P2*, the heating season per-compressor off mode power. Round to the nearest watt. The expression for calculating *P2* is as follows:

$$P2 = \frac{P2_x - P_x}{number\ of\ compressors}.$$

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 of this appendix, evaluate the seasonal energy efficiency ratio,

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

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 of this appendix, use a building cooling load, $BL(T_j)$. When referenced, evaluate $BL(T_j)$ for cooling using,

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

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 of this appendix, 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 a Blower Coil System Having a Single-Speed Compressor and Either a Fixed-Speed Indoor Blower or a Constant-Air-Volume-Rate Indoor Blower, or a Coil-Only System Air Conditioner or Heat Pump

a. Evaluate the seasonal energy efficiency ratio, expressed in units of Btu/watt-hour, using:

$SEER = PLF(0.5) * EER_B$

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 of this appendix, 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 of this appendix regarding the definition and calculation of $\dot{Q}_c(82)$ and $\dot{E}_c(82)$. Evaluate the cooling mode cyclic degradation factor C_D^c as specified in section 3.5.3 of this appendix.

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 Blower

4.1.2.1 Units Covered by Section 3.2.2.1 of This Appendix Where Indoor Blower Capacity Modulation Correlates With the Outdoor Dry Bulb Temperature

The manufacturer must provide information on how the indoor air volume rate or the indoor blower 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,

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

where:

$$X(T_j) = \begin{Bmatrix} BL(T_j)/\dot{Q}_c(T_j) \\ or \\ 1 \end{Bmatrix}$$ 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 19. Use Equation 4.1–2 to calculate the building load, $BL(T_j)$. Evaluate $\dot{Q}_c(T_j)$ using,

Equation 4.1.2-2 $$\dot{Q}_c(T_j) = \dot{Q}_c^{k=1}(T_j) + \frac{\dot{Q}_c^{k=2}(T_j) - \dot{Q}_c^{k=1}(T_j)}{\mathrm{FP}_c^{k=2} - \mathrm{FP}_c^{k=1}} * \left[\mathrm{FP}_c(T_j) - \mathrm{FP}_c^{k=1}\right]$$

where:

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

the space cooling capacity of the test unit at outdoor temperature T_j if operated at the cooling minimum air volume rate, Btu/h.

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

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 blower 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 of this appendix), $FP_c^{k=2}$ 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 of this appendix 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, Equation 4.1.2-3 $$\frac{e_c(T_j)}{N} = \frac{X(T_j) * \dot{E}_c(T_j)}{PLF_j} * \frac{n_j}{N}$$

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.

Evaluate the cooling mode cyclic degradation factor C_D^c as specified in section 3.5.3 of this appendix.

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

$$\dot{E}_c(T_j) = \dot{E}_c^{k=1}(T_j) + \frac{\dot{E}_c^{k=2}(T_j) - \dot{E}_c^{k=1}(T_j)}{\mathrm{FP}_c^{k=2} - \mathrm{FP}_c^{k=1}} * \left[\mathrm{FP}_c(T_j) - \mathrm{FP}_c^{k=1}\right]$$

where:

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

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

$\dot{E}_c^{k=2}(T_j) = \dot{E}_c^{k=2}(82) + \frac{\dot{E}_c^{k=2}(95) - \dot{E}_c^{k=2}(82)}{95-82} * (T_j - 82)$ 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 $\mathrm{FP}_c^{k=1}$, and $\mathrm{FP}_c^{k=2}$, and $\mathrm{FP}_c(T_j)$ 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 of this appendix 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 of This Appendix Where Indoor Blower Capacity Modulation Is Used To Adjust the Sensible to Total Cooling Capacity Ratio. Calculate SEER as specified in section 4.1.1 of this appendix.

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,

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

Equation 4.1.3-2 $$\dot{E}_c^{k=1}(T_j) = \dot{E}_c^{k=1}(67) + \frac{\dot{E}_c^{k=1}(82) - \dot{E}_c^{k=1}(67)}{82-67} * (T_j - 67)$$

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 of this appendix. 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,

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

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

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_2test, and all are calculated as specified in section 3.3 of this appendix.

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 of this appendix), cycle between low and high capacity (section 4.1.3.2 of this appendix), or operate at high capacity (sections 4.1.3.3 and 4.1.3.4 of this appendix) in responding to the building load. For units that lock out low capacity operation at higher outdoor temperatures, the outdoor temperature at which the unit locks out must be that specified by the manufacturer in the certification report 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, $\dot{Q}_c^{k=1}(T_j) \geq BL(T_j)$

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

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 19. Use Equations 4.1.3–1 and 4.1.3–2, respectively, to evaluate $\dot{Q}_c^{k=1}(T_j)$ and $\dot{E}_c^{k=1}(T_j)$. Evaluate the cooling mode cyclic degradation factor C_D^c as specified in section 3.5.3 of this appendix.

TABLE 19—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, $\dot{Q}_c^{k=1}(T_j)$ $BL(T_j)$ $\dot{Q}_c^{k=2}(T_j)$

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

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

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 19. 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.

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

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.

$PLF_j = 1 - C_D^c(k = 2) * [1 - X^{k=2}(T_j)$ the part load factor, dimensionless.

Obtain the fraction bin hours for the cooling season, $\frac{n_j}{N}$, from Table 19. 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 C_2 and D_2 tests described in section 3.2.3 and Table 7 of this appendix are not conducted, set C_D^c (k=2) equal to the default value specified in section 3.5.3 of this appendix.

4.1.3.4 Unit Must Operate Continuously at High (k=2) Compressor Capacity at Temperature T_j, $BL(T_j) \geq \dot{Q}_c^{k=2}(T_j)$

$$\frac{q_c(T_j)}{N} = \dot{Q}_c^{k=2}(T_j) * \frac{n_j}{N} \qquad \frac{e_c(T_j)}{N} = \dot{E}_c^{k=2}(T_j) * \frac{n_j}{N}$$

Obtain the fractional bin hours for the cooling season, n_j/N, from Table 19. 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,

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

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

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 of this appendix.

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 full 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 of this appendix. 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 8) E_V test of this appendix using,

$$\text{Equation 4.1.4-3} \quad \dot{Q}_c^{k=v}(T_j) = \dot{Q}_c^{k=v}(87) + M_Q * (T_j - 87)$$

$$\text{Equation 4.1.4-4} \quad \dot{E}_c^{k=v}(T_j) = \dot{E}_h^{k=v}(87) + M_E * (T_j - 87)$$

where $\dot{Q}_c^{k=v}(87)$ and $\dot{E}_c^{k=v}(87)$ are determined from the E_V test and calculated as specified in section 3.3 of this appendix.

Approximate the slopes of the k=v intermediate speed cooling capacity and electrical power input curves, M_Q and M_E, as follows:

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

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

where,

$$N_Q = \frac{\dot{Q}_c^{k=v}(87) - \dot{Q}_c^{k=1}(87)}{\dot{Q}_c^{k=2}(87) - \dot{Q}_c^{k=1}(87)} \qquad N_E = \frac{\dot{E}_c^{k=v}(87) - \dot{E}_c^{k=1}(87)}{\dot{E}_c^{k=2}(87) - \dot{E}_c^{k=1}(87)}$$

Use Equations 4.1.4–1 and 4.1.4–2, respectively, to calculate $\dot{Q}_c^{k=1}(87)$ and $\dot{E}_c^{k=1}(87)$.

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, $\dot{Q}_c^{k=1}(T_j) \geq BL(T_j)$

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

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 19. Use Equations 4.1.3–1 and 4.1.3–2, respectively, to evaluate $\dot{Q}_c^{k=1}(T_j)$ and $\dot{E}_c^{k=1}(T_j)$. Evaluate the cooling mode cyclic degradation factor C_D^c as specified in section 3.5.3 of this appendix.

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

$$\frac{q_c(T_j)}{N} = \dot{Q}_c^{k=i}(T_j) * \frac{n_j}{N} \qquad \frac{e_c(T_j)}{N} = \dot{E}_c^{k=i}(T_j) * \frac{n_j}{N}$$

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 19. 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:

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

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

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 ($\dot{Q}_c^{k=l}(T_l) = BL(T_1)$), °F. Determine T_1 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 of this appendix, 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 4.1.4–3 and 4.1–2 and solving for outdoor temperature.

T_2 = the outdoor temperature at which the unit, when operating at full 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.

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

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

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

4.1.4.3 Unit Must Operate Continuously at Full (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 of this appendix with the understanding that $\dot{Q}_c^{k=2}(T_j)$ and $\dot{E}_c^{k=2}(T_j)$ correspond to full compressor speed operation and are derived from the results of the tests specified in section 3.2.4 of this appendix.

4.1.5 SEER Calculations for an Air Conditioner or Heat Pump Having a Single Indoor Unit With Multiple Indoor Blowers

Calculate SEER using Eq. 4.1–1, where $q_c(Tj)/N$ and $e_c(Tj)/N$ are evaluated as specified in the applicable subsection.

4.1.5.1 For Multiple Indoor Blower Systems That Are Connected to a Single, Single-Speed Outdoor Unit

a. Calculate 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 the cooling minimum air volume rate and outdoor temperature T_j using the equations given in section 4.1.2.1 of this appendix. Calculate 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 the cooling full-load air volume rate and outdoor temperature T_j using the equations given in section 4.1.2.1 of this appendix. In evaluating the section 4.1.2.1 equations, determine the quantities $\dot{Q}_c^{k=1}(82)$ and $\dot{E}_c^{k=1}(82)$ from the B1 test, $\dot{Q}_c^{k=1}(95)$ and $\dot{E}_c^{k=1}(95)$ from the Al test, $\dot{Q}_c^{k=2}(82)$ and $\dot{E}_c^{k=2}(82)$ from the B2 test, and $\dot{Q}_c^{k=2}(95)$ and $\dot{E}_c^{k=2}(95)$ from the A2 test. Evaluate all eight quantities as specified in section 3.3 of this appendix. Refer to section 3.2.2.1 and Table 6 of this appendix for additional information on the four referenced laboratory tests.

b. Determine the cooling mode cyclic degradation coefficient, CD_c, as per sections 3.2.2.1 and 3.5 to 3.5.3 of this appendix. Assign this same value to $CD_c(K=2)$.

c. Except for using the above values of $\dot{Q}_c^{k=1}(T_j)$, $\dot{E}_c^{k=1}(T_j)$, $\dot{E}_c^{k=2}(T_j)$, $\dot{Q}_c^{k=2}(T_j)$, CD_c, and CD_c (K=2), calculate the quantities $q_c(T_j)/N$ and $e_c(T_j)/N$ as specified in section 4.1.3.1 of this appendix for cases where $\dot{Q}_c^{k=1}(T_j) \geq BL(T_j)$. For all other outdoor bin temperatures, T_j, calculate $q_c(Tj)/N$ and $e_c(Tj)/N$ as specified in section 4.1.3.3 of this appendix if $\dot{Q}_c^{k=2}(T_j) > BL(T_j)$ or as specified in section 4.1.3.4 of this appendix if $\dot{Q}_c^{k=2}(T_j) \leq BL(T_j)$.

4.1.5.2 Unit Operates at an Intermediate Compressor Speed (k=i) In Order To Match the Building Cooling Load at Temperature T_j, $\dot{Q}_c^{k=1}(T_j) < BL(T_j) < \dot{Q}_c^{k=2}(T_j)$

$$\frac{q_c(T_j)}{N} = \dot{Q}_c^{k=i}(T_j) * \frac{n_j}{N} \qquad \frac{e_c(T_j)}{N} = \dot{E}_c^{k=i}(T_j) * \frac{n_j}{N}$$

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 19. For each temperature bin where the unit operates at an intermediate compressor speed, determine the energy efficiency ratio $EER^{k=i}(T_j)$ using the following equations,

For each temperature bin where $\dot{Q}_c^{k=1}(T_j) < BL(T_j) < \dot{Q}_c^{k=v}(T_j)$,

$$EER^{k=i}(T_j) = EER^{k=1}(T_j) + \frac{EER^{k=v}(T_j) - EER^{k=1}(T_j)}{Q^{k=v}(T_j) - Q^{k=1}(T_j)} * \left(BL(T_j) - Q^{k=1}(T_j)\right)$$

For each temperature bin where $\dot{Q}_c^{k=v}(T_j) \leq BL(T_j) < \dot{Q}_c^{k=2}(T_j)$,

$$EER^{k=i}(T_j) = EER^{k=v}(T_j) + \frac{EER^{k=2}(T_j) - EER^{k=v}(T_j)}{Q^{k=2}(T_j) - Q^{k=v}(T_j)} * \left(BL(T_j) - Q^{k=v}(T_j)\right)$$

Where:

$EER^{k=1}(T_j)$ is the steady-state energy efficiency ratio of the test unit when operating at minimum compressor speed and temperature Tj, Btu/h per W, calculated using capacity $\dot{Q}_c^{k=1}(T_j)$ calculated using Equation 4.1.4–1 and electrical power consumption $\dot{E}_c^{k=1}(T_j)$ calculated using Equation 4.1.4–2;

$EER^{k=v}(T_j)$ is the steady-state energy efficiency ratio of the test unit when operating at intermediate compressor speed and temperature Tj, Btu/h per W, calculated using capacity $\dot{Q}_c^{k=v}(T_j)$ calculated using Equation 4.1.4–3 and electrical power consumption $\dot{E}_c^{k=v}(T_j)$ calculated using Equation 4.1.4–4;

$EER^{k=2}(T_j)$ is the steady-state energy efficiency ratio of the test unit when operating at full compressor speed and temperature Tj, Btu/h per W, calculated using capacity $\dot{Q}_c^{k=2}(T_j)$ and electrical power consumption $\dot{E}_c^{k=2}(T_j)$, both calculated as described in section 4.1.4; and

$BL(T_j)$ is the building cooling load at temperature T_j, Btu/h.

4.2 Heating Seasonal Performance Factor (HSPF) Calculations

Unless an approved alternative efficiency determination method is used, as set forth in 10 CFR 429.70(e), HSPF must be calculated as follows: Six generalized climatic regions are depicted in Figure 1 and otherwise defined in Table 20. For each of these regions and for each applicable standardized design heating requirement, evaluate the heating seasonal performance factor using,

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

where:

$e2(T_j)/N$ = 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 of this appendix, 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 section 4.2.5 of this appendix).

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 20.

j = the bin number, dimensionless.

J = for each generalized climatic region, the total number of temperature bins, dimensionless. Referring to Table 20, 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 of this appendix, 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 20—GENERALIZED CLIMATIC REGION INFORMATION

	Region No.					
	I	II	III	IV	V	VI
Heating Load Hours, HLH	750	1,250	1,750	2,250	2,750	*2,750
Outdoor Design Temperature, T_{OD}	37	27	17	5	−10	30

TABLE 20—GENERALIZED CLIMATIC REGION INFORMATION—Continued

	Region No.					
	I	II	III	IV	V	VI
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

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

Where:

T_{OD} = the outdoor design temperature, °F. An outdoor design temperature is specified for each generalized climatic region in Table 20.

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 section 1.2 of this appendix, Definitions), Btu/h.

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

$$DHR_{min} = \begin{cases} \dot{Q}_h^k(47) * \left[\dfrac{65 - T_{OD}}{60}\right], & for\ Regions\ I, II, III, IV, \&\ VI \\ \dot{Q}_h^k(47), & for\ Region\ V \end{cases}$$

and

$$DHR_{max} = \begin{cases} 2 * \dot{Q}_h^k(47) * \left[\dfrac{65 - T_{OD}}{60}\right], & for\ Regions\ I, II, III, IV, \&\ VI \\ 2.2 * \dot{Q}_h^k(47), & for\ Region\ V \end{cases}$$

Rounded to the nearest standardized DHR given in Table 20

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

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

b. For a section 3.6.2 single-speed heat pump or a two-capacity heat pump not covered by item d, $\dot{Q}_h^k(47) = \dot{Q}_h^{k=2}(47)$, the space heating capacity determined from the H1 or $H1_2$ test.

c. For a variable-speed heat pump, $\dot{Q}_h^k(47) = \dot{Q}_h^{k=N}(47)$, the space heating capacity determined from the $H1_N$ test.

d. For two-capacity, northern heat pumps (see section 1.2 of this appendix, Definitions), $\dot{Q}^k_h(47) = \dot{Q}^{k=1}_h(47)$, the space heating capacity determined from the $H1_1$ 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 of this appendix, whichever applies.

For heat pumps with heat comfort controllers (see section 1.2 of this appendix, Definitions), HSPF also accounts for resistive heating contributed when operating above for all heat pumps, see either section 4.2.1, 4.2.2, 4.2.3, or 4.2.4 of this appendix, whichever applies.

For heat pumps with heat comfort controllers (see section 1.2 of this appendix, Definitions), 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 of this appendix for the additional steps required for calculating the HSPF.

TABLE 21—STANDARDIZED DESIGN HEATING REQUIREMENTS

[Btu/h]
5,000
10,000
15,000
20,000
25,000
30,000
35,000
40,000
50,000
60,000
70,000
80,000
90,000
100,000
110,000
130,000

4.2.1 Additional Steps for Calculating the HSPF of a Blower Coil System Heat Pump Having a Single-Speed Compressor and Either a Fixed-Speed Indoor Blower or a Constant-Air-Volume-Rate Indoor Blower Installed, or a Coil-Only System Heat Pump

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

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

Where:

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

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.

$PLF_j = 1 - \dot{C}_D^h \cdot [1 - X(T_j)]$ 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 20. Evaluate the heating mode cyclic degradation factor $\dot{C}_D^h$ as specified in section 3.8.1 of this appendix.

Determine the low temperature cut-out factor using

Equation 4.2.1-3 $$\delta(T_j) = \begin{Bmatrix} 0, if\ T_j \leq T_{off}\ or\ \frac{\dot{Q}_h(T_j)}{3.413*\dot{E}_h(T_j)} < 1 \\ 1/2, if\ T_{off} < T_j \leq T_{on}\ and\ \frac{\dot{Q}_h(T_j)}{3.413*\dot{E}_h(T_j)} \geq 1 \\ 1, if\ T_j > T_{on}\ and\ \frac{\dot{Q}_h(T_j)}{3.413*\dot{E}_h(T_j)} \geq 1 \end{Bmatrix}$$

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,

Equation 4.2.1-4 $$\dot{Q}_h(T_j) = \begin{cases} \dot{Q}_h(17) + \frac{[\dot{Q}_h(47)-\dot{Q}_h(17)]*(T_j-17)}{47-17},\ if\ T_j \geq 45\ °F\ or\ T_j \leq 17\ °F \\ \dot{Q}_h(17) + \frac{[\dot{Q}_h(35)-\dot{Q}_h(17)]*(T_j-17)}{35-17},\ if\ 17\ °F < T_j < 45\ °F \end{cases}$$

Equation 4.2.1-5

$$\dot{E}_h(T_j) = \begin{cases} \dot{E}_h(17) + \dfrac{\left[\dot{E}_h(47) - \dot{E}_h(17)\right] * (T_j - 17)}{47 - 17}, & if\ T_j \geq 45\ °\text{F}\ or\ T_j \leq 17\ °\text{F} \\ \dot{E}_h(17) + \dfrac{\left[\dot{E}_h(35) - \dot{E}_h(17)\right] * (T_j - 17)}{35 - 17}, & if\ 17\ °\text{F} < T_j < 45\ °\text{F} \end{cases}$$

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

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 Blower

The manufacturer must provide information about how the indoor air volume rate or the indoor blower 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 of this appendix with the exception of replacing references to the H1C test and section 3.6.1 of this appendix with the $H1C_1$ test and section 3.6.2 of this appendix. 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

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

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

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

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

Equation 4.2.2-4

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

For units where indoor blower speed is the primary control variable, $\text{FP}_h^{k=1}$ denotes the fan speed used during the required $H1_1$ and $H3_1$ tests (see Table 12), $\text{FP}_h^{k=2}$ denotes the fan speed used during the required $H1_2$, $H2_2$, and $H3_2$ tests, and $\text{FP}_h(T_j)$ 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 of this appendix. Determine $\dot{Q}_h^{k=1}(35)$ and $\dot{E}_h^{k=1}(35)$ as specified in section 3.6.2 of this appendix; 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 of this appendix. 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 of this appendix.

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 differ depending upon whether the heat pump would operate at low capacity (section 4.2.3.1 of this appendix), cycle between low and high capacity (section 4.2.3.2 of this appendix), or operate at high capacity (sections 4.2.3.3 and 4.2.3.4 of this appendix) in responding to the building load. For heat pumps that lock out low capacity operation at low outdoor temperatures, the outdoor temperature at which the unit locks out must be that specified by the manufacturer in the certification report so that the appropriate equations can be selected.

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

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

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

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

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 of this appendix. 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 of this appendix, 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 of this appendix. 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 of this appendix, 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 of this appendix.

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, $\dot{Q}_h^{k=1}(T_j) \geq BL(T_j)$

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

$$\text{Equation 4.2.3-2} \quad \frac{RH(T_j)}{N} = \frac{BL(T_j) * [1 - \delta(T_j)]}{3.413 \frac{Btu/h}{W}} * \frac{n_j}{N}$$

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.

Evaluate the heating mode cyclic degradation factor C_D^h as specified in section 3.8.1 of this appendix.

Determine the low temperature cut-out factor using

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

where T_{off} and T_{on} are defined in section 4.2.1 of this appendix. Use the calculations given in section 4.2.3.3 of this appendix, 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

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

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

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

Where:

$X^{k=2}(T_j) = BL(T_j)/\dot{Q}_h^{k=2}(T_j)$. $PLF_j = 1 - C_D^h(k = 2) * [1 - X^{k=1}(T_j)]$

If the H1C2 test described in section 3.6.3 and Table 13 of this appendix is not conducted, set C_D^h (k=2) equal to the default value specified in section 3.8.1 of this appendix.

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, $BL(T_j) \geq \dot{Q}_h^{k=2}(T_j)$

$$\frac{e_h(T_j)}{N} = \dot{E}_h^{k=2}(T_j) * \delta'(T_j) * \frac{n_j}{N} \qquad \frac{RH(T_j)}{N} = \frac{BL(T_j) * [\dot{Q}_h^{k=2}(T_j) * \delta\prime(T_j)]}{3.413 \frac{Btu/h}{W}} * \frac{n_j}{N}$$

where:

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

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

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

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

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 of this appendix.

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 full compressor speed and outdoor temperature T_j by solving Equations 4.2.2–3 and 4.2.2–4, respectively, for k=2. For Equation 4.2.2–3, use $\dot{Q}_{hcalc}^{k=2}(47)$ to represent $\dot{Q}_h^{k=2}(47)$, and for Equation 4.2.2–4, use $\dot{E}_{hcalc}^{k=2}(47)$ to represent $\dot{E}_{hcalc}^{k=2}(47)$—evaluate $\dot{Q}_{hcalc}^{k=2}(47)$ and $\dot{E}_{hcalc}^{k=2}(47)$ as specified in section 3.6.4b of this appendix.

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

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

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 of this appendix. Approximate the slopes of the k=v intermediate speed heating capacity and electrical power input curves, M_Q and M_E, as follows:

$$M_Q = \left[\frac{\dot{Q}_h^{k=1}(62) - \dot{Q}_h^{k=1}(47)}{62-47} * (1 - N_Q)\right] + \left[N_Q * \frac{\dot{Q}_h^{k=2}(35) - \dot{Q}_h^{k=2}(17)}{35-17}\right]$$

$$M_E = \left[\frac{\dot{E}_h^{k=1}(62) - \dot{E}_h^{k=1}(47)}{62-47} * (1 - N_E)\right] + \left[N_E * \frac{\dot{E}_h^{k=2}(35) - \dot{E}_h^{k=2}(17)}{35-17}\right]$$

where,

$$N_Q = \frac{\dot{Q}_h^{k=v}(35) - \dot{Q}_h^{k=1}(35)}{\dot{Q}_h^{k=2}(35) - \dot{Q}_h^{k=1}(35)} \qquad N_E = \frac{\dot{E}_h^{k=v}(35) - \dot{E}_h^{k=1}(35)}{\dot{E}_h^{k=2}(35) - \dot{E}_h^{k=1}(35)}$$

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 $\frac{RH(T_j)}{N}$ and $\frac{e_h(T_j)}{N}$ differs depending upon whether the heat pump would operate at minimum speed (section 4.2.4.1 of this appendix), operate at an intermediate speed (section 4.2.4.2 of this appendix), or operate at full speed (section 4.2.4.3 of this appendix) 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

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

as specified in section 4.2.3.1 of this appendix. 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 of this appendix with "minimum speed" and section 3.6.4 of this appendix. Also, the last sentence of section 4.2.3.1 of this appendix 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,

$$\frac{e_h(T_j)}{N} = \dot{E}_h^{k=i}(T_j) * \delta(T_j) * \frac{n_j}{N}$$

where,

$$\dot{E}_h^{k=i}(T_j) = \frac{\dot{Q}_h^{k=i}(T_j)}{3.413\,\frac{Btu/h}{W} * COP^{k=i}(T_j)}$$

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 the following equations,

For each temperature bin where $\dot{Q}_h^{k=1}(T_j) < BL(T_j) < \dot{Q}_h^{k=v}(T_j)$,

$$COP_h^{k=i}(T_j) = COP_h^{k=1}(T_j) + \frac{COP_h^{k=v}(T_j) - COP_h^{k=1}(T_j)}{Q_h^{k=v}(T_j) - Q_h^{k=1}(T_j)} * (BL(T_j) - Q_h^{k=1}(T_j))$$

For each temperature bin where $\dot{Q}_h^{k=v}(T_j) \leq BL(T_j) < \dot{Q}_h^{k=2}(T_j)$,

$$COP_h^{k=i}(T_j) = COP_h^{k=v}(T_j) + \frac{COP_h^{k=2}(T_j) - COP_h^{k=v}(T_j)}{Q_h^{k=2}(T_j) - Q_h^{k=v}(T_j)} * (BL(T_j) - Q_h^{k=v}(T_j))$$

Where:

$COP_h^{k=1}(T_j)$ is the steady-state coefficient of performance of the heat pump when operating at minimum compressor speed and temperature Tj, dimensionless, calculated using capacity $\dot{Q}_h^{k=1}(T_j)$ calculated using Equation 4.2.4–1 and electrical power consumption $\dot{E}_h^{k=1}(T_j)$ calculated using Equation 4.2.4–2;

$COP_h^{k=v}(T_j)$ is the steady-state coefficient of performance of the heat pump when operating at intermediate compressor speed and temperature Tj, dimensionless, calculated using capacity $\dot{Q}_h^{k=v}(T_j)$ calculated using Equation 4.2.4–3 and electrical power consumption $\dot{E}_h^{k=v}(T_j)$ calculated using Equation 4.2.4–4;

$COP_h^{k=2}(T_j)$ is the steady-state coefficient of performance of the heat pump when operating at full compressor speed and temperature Tj, dimensionless, calculated using capacity $\dot{Q}_h^{k=2}(T_j)$ and electrical power consumption $\dot{E}_h^{k=2}(T_j)$, both calculated as described in section 4.2.4; and

$BL(T_j)$ is the building heating load at temperature T_j, Btu/h.

4.2.4.3 Heat Pump Must Operate Continuously at Full (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

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

as specified in section 4.2.3.4 of this appendix with the understanding that $\dot{Q}_h^{k=2}(T_j)$ and $\dot{E}_h^{k=2}(T_j)$ correspond to full compressor speed operation and are derived from the results of the specified section 3.6.4 tests of this appendix.

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 Blower Coil System Heat Pump Having a Heat Comfort Controller: Additional Steps for Calculating the HSPF of a Heat Pump Having a Single-Speed Compressor and Either a Fixed-Speed Indoor Blower or a Constant-Air-Volume-Rate Indoor Blower Installed, or a Coil-Only System Heat Pump

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 of this appendix (Equations 4.2.1–4 and 4.2.1–5) for each outdoor bin temperature, T_j, that is listed in Table 20. 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 test using:

$$\dot{m}_{da} = \overline{\dot{V}_s} * 0.075 \frac{lbm_{da}}{ft^3} * \frac{60_{min}}{hr} = \frac{\overline{\dot{V}}_{mx}}{v_n' * [1 + W_n]} * \frac{60_{min}}{hr} = \frac{\overline{\dot{V}}_{mx}}{v_n} * \frac{60_{min}}{hr}$$

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 20, calculate the nominal temperature of the air leaving the heat pump condenser coil using,

$$T_0(T_j) = 70°\text{F} + \frac{\dot{Q}_{hp}(T_j)}{\dot{m}_{da} * C_{p,da}}$$

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 of this appendix. 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 of this appendix), determine $\dot{Q}_h(T_j)$ and $\dot{E}_h(T_j)$ as specified in section 4.2.1 of this appendix (*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,

$$\dot{Q}_h(T_j) = \dot{Q}_{hp}(T_j) + \dot{Q}_{CC}(T_j) \qquad \dot{E}_h(T_j) = \dot{E}_{hp}(T_j) + \dot{E}_{CC}(T_j)$$

where,

$$\dot{Q}_{CC}(T_j) = \dot{m}_{da} * C_{p,da} * [T_{CC} - T_0(T_j)] \qquad \dot{E}_{CC}(T_j) = \frac{\dot{Q}_{CC}(T_j)}{3.413 \frac{Btu/h}{W}}$$

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 Blower

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 of this appendix (Equations 4.2.2–1 and 4.2.2–2) for each outdoor bin temperature, T_j, that is listed in Table 20. 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:

$$\dot{m}_{da} = \overline{\dot{V}_s} * 0.075 \frac{lbm_{da}}{ft^3} * \frac{60_{min}}{hr} = \frac{\overline{\dot{V}}_{mx}}{v_n' * [1 + W_n]} * \frac{60_{min}}{hr} = \frac{\overline{\dot{V}}_{mx}}{v_n} * \frac{60_{min}}{hr}$$

$$C_{p,da} = 0.24 + 0.444 * W_n$$

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 20, calculate the nominal temperature of the air leaving the heat pump condenser coil using,

$$T_0(T_j) = 70°\text{F} + \frac{\dot{Q}_{hp}(T_j)}{\dot{m}_{da} * C_{p,da}}$$

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 of this appendix with the exception of replacing references to the H1C test and section 3.6.1 of this appendix with the $H1C_1$ test and section 3.6.2 of this appendix. 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 of this appendix), determine $\dot{Q}_h(T_j)$ and $\dot{E}_h(T_j)$ as specified in section 4.2.2 of this appendix (*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,

$$\dot{Q}_h(T_j) = \dot{Q}_{hp}(T_j) + \dot{Q}_{CC}(T_j) \qquad \dot{E}_h(T_j) = \dot{E}_{hp}(T_j) + \dot{E}_{CC}(T_j)$$

where,

$$\dot{Q}_{CC}(T_j) = \dot{m}_{da} * C_{p,da} * [T_{CC} - T_0(T_j)] \qquad \dot{E}_{CC}(T_j) = \frac{\dot{Q}_{CC}(T_j)}{3.413\frac{Btu/h}{W}}$$

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 of this appendix for both high and low capacity and at each outdoor bin temperature, T_j, that is listed in Table 20. 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/$lbm_{da} \cdot$ °F) from the results of the $H1_1$ test using:

$$\dot{m}_{da}^{k=1} = \overline{\dot{V}_s} * 0.075\frac{lbm_{da}}{ft^3} * \frac{60_{min}}{hr} = \frac{\overline{\dot{V}}_{mx}}{v'_n * [1 + W_n]} * \frac{60_{min}}{hr} = \frac{\overline{\dot{V}}_{mx}}{v_n} * \frac{60_{min}}{hr}$$

$$C_{p,da}^{k=1} = 0.24 + 0.444 * W_n$$

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 20, calculate the nominal temperature of the air leaving the heat pump condenser coil when operating at low capacity using,

$$T_0^{k=1}(T_j) = 70°\text{F} + \frac{\dot{Q}_{hp}^{k=1}(T_j)}{\dot{m}_{da}^{k=1} * C_{p,da}^{k=1}}$$

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 20, calculate the nominal temperature of the air leaving the heat pump condenser coil when operating at high capacity using,

$$T_0^{k=2}(T_j) = 70°\text{F} + \frac{\dot{Q}_{hp}^{k=2}(T_j)}{\dot{m}_{da}^{k=2} * C_{p,da}^{k=2}}$$

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 of this appendix, 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 of this appendix), determine $\dot{Q}_h^{k=1}(T_j)$ and $\dot{E}_h^{k=1}(T_j)$ as specified in section 4.2.3 of this appendix (*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,

$$\dot{Q}_h^{k=1}(T_j) = \dot{Q}_{hp}^{k=1}(T_j) + \dot{Q}_{CC}^{k=1}(T_j) \qquad \dot{E}_h^{k=1}(T_j) = \dot{E}_{hp}^{k=1}(T_j) + \dot{E}_{CC}^{k=1}(T_j)$$

where,

$$\dot{Q}_{CC}^{k=1}(T_j) = \dot{m}_{da}^{k=1} * C_{p,da}^{k=1} * [T_{CC} - T_0^{k=1}(T_j)] \qquad \dot{E}_{CC}^{k=1}(T_j) = \frac{\dot{Q}_{CC}^{k=1}(T_j)}{3.413\frac{Btu/h}{W}}$$

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 of this appendix (*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,

$$\dot{Q}_h^{k=2}(T_j) = \dot{Q}_{hp}^{k=2}(T_j) + \dot{Q}_{CC}^{k=2}(T_j) \qquad \dot{E}_h^{k=2}(T_j) = \dot{E}_{hp}^{k=2}(T_j) + \dot{E}_{CC}^{k=2}(T_j)$$

where,

$$\dot{Q}_{CC}^{k=2}(T_j) = \dot{m}_{da}^{k=2} * C_{p,da}^{k=2} * [T_{CC} - T_0^{k=2}(T_j)] \qquad \dot{E}_{CC}^{k=2}(T_j) = \frac{\dot{Q}_{CC}^{k=2}(T_j)}{3.413\frac{Btu/h}{W}}$$

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]

4.2.6 Additional Steps for Calculating the HSPF of a Heat Pump Having a Triple-Capacity Compressor

The only triple-capacity heat pumps covered are triple-capacity, northern heat pumps. For such heat pumps, the calculation of the Eq. 4.2–1 quantities

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

differ depending on whether the heat pump would cycle on and off at low capacity (section 4.2.6.1 of this appendix), cycle on and off at high capacity (section 4.2.6.2 of this appendix), cycle on and off at booster capacity (section 4.2.6.3 of this appendix), cycle between low and high capacity (section 4.2.6.4 of this appendix), cycle between high and booster capacity (section 4.2.6.5 of this appendix), operate continuously at low capacity (4.2.6.6 of this appendix), operate continuously at high capacity (section 4.2.6.7 of this appendix), operate continuously at booster capacity (section 4.2.6.8 of this appendix), or heat solely using resistive heating (also section 4.2.6.8 of this appendix) in responding to the building load. As applicable, the manufacturer must supply information regarding the outdoor temperature range at which each stage of compressor capacity is active. As an informative example, data may be submitted in this manner: At the low (k=1) compressor capacity, the outdoor temperature range of operation is 40 °F ≤ T ≤ 65 °F; At the high (k=2) compressor capacity, the outdoor temperature range of operation is 20 °F ≤ T ≤ 50 °F; At the booster (k=3) compressor capacity, the outdoor temperature range of operation is −20 °F ≤ T ≤ 30 °F.

a. Evaluate the space heating capacity and electrical power consumption of the heat pump when operating at low compressor capacity and outdoor temperature Tj using the equations given in section 4.2.3 of this appendix for $\dot{Q}_h^{k=1}(T_j)$ and $\dot{E}_h^{k=1}(T_j)$) In evaluating the section 4.2.3 equations, 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 four quantities as specified in section 3.7 of this appendix. If, in accordance with section 3.6.6 of this appendix, the $H3_1$ test is conducted, calculate $\dot{Q}_h^{k=1}(17)$ and $\dot{E}_h^{k=1}(17)$ as specified in section 3.10 of this appendix and determine $\dot{Q}_h^{k=1}(35)$ and $\dot{E}_h^{k=1}(35)$ as specified in section 3.6.6 of this appendix.

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, evaluated as specified in section 3.7 of this appendix. Determine the equation input for $\dot{Q}_h^{k=2}(35)$ and $\dot{E}_h^{k=2}(35)$ from the $H2_2$, evaluated as specified in section 3.9.1 of this appendix. Also, determine $\dot{Q}_h^{k=2}(17)$ and $\dot{E}_h^{k=2}(17)$ from the $H3_2$ test, evaluated as specified in section 3.10 of this appendix.

c. Evaluate the space heating capacity and electrical power consumption of the heat pump when operating at booster compressor capacity and outdoor temperature Tj using

$$\dot{Q}_h^{k=3}(T_j) = \begin{cases} \dot{Q}_h^{k=3}(17) + \dfrac{[\dot{Q}_h^{k=3}(35) - \dot{Q}_h^{k=3}(17)] * (T_j - 17)}{35 - 17}, & if\ 17\ °F < T_j \leq 45\ °F \\ \dot{Q}_h^{k=3}(5) + \dfrac{[\dot{Q}_h^{k=3}(17) - \dot{Q}_h^{k=3}(5)] * (T_j - 5)}{17 - 5}, & if\ T_j \leq 17\ °F \end{cases}$$

$$\dot{E}_h^{k=3}(T_j) = \begin{cases} \dot{E}_h^{k=3}(17) + \dfrac{[\dot{E}_h^{k=3}(35) - \dot{E}_h^{k=3}(17)] * (T_j - 17)}{35 - 17}, & if\ 17\ °F < T_j \leq 45\ °F \\ \dot{E}_h^{k=3}(5) + \dfrac{[\dot{E}_h^{k=3}(17) - \dot{E}_h^{k=3}(5)] * (T_j - 5)}{17 - 5}, & if\ T_j \leq 17\ °F \end{cases}$$

Determine $\dot{Q}_h^{k=3}(17)$ and $\dot{E}_h^{k=3}(17)$ from the $H3_3$ test and determine $\dot{Q}_h^{k=2}(5)$ and $\dot{E}_h^{k=3}(5)$ from the $H4_3$ test. Calculate all four quantities as specified in section 3.10 of this appendix. Determine the equation input for $\dot{Q}_h^{k=3}(35)$ and $\dot{E}_h^{k=3}(35)$ as specified in section 3.6.6 of this appendix. 4.2.6.1 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, $\dot{Q}_h^{k=1}(T_j) \geq BL(T_j)$., and the heat pump permits low compressor capacity at T_j.

Evaluate the quantities

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

using Eqs. 4.2.3–1 and 4.2.3–2, respectively. Determine the equation inputs $X^{k=1}(T_j)$, PLF_j, and $\delta'(T_j)$ as specified in section 4.2.3.1 of this appendix. In calculating the part load factor, PLF_j, use the low-capacity cyclic-degradation coefficient C_D^h, [or equivalently, $C_D^h(k=1)$] determined in accordance with section 3.6.6 of this appendix.

4.2.6.2 Heat Pump Only Operates at High (k=2) Compressor Capacity at Temperature T_j and Its Capacity Is Greater Than or Equal to the Building Heating Load, $BL(T_j) < \dot{Q}_h^{k=2}(T_j)$

Evaluate the quantities

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

as specified in section 4.2.3.3 of this appendix. Determine the equation inputs $X^{k=2}(T_j)$, PLF_j, and $\delta'(T_j)$ as specified in section 4.2.3.3 of this appendix. In calculating the part load factor, PLF_j, use the high-capacity cyclic-degradation coefficient, $C_D^h(k=2)$ determined in accordance with section 3.6.6 of this appendix.

4.2.6.3 Heat Pump Only Operates at High (k=3) Compressor Capacity at Temperature T_j and Its Capacity Is Greater Than or Equal to the Building Heating Load, $BL(T_j) \leq \dot{Q}_h^{k=3}(T_j)$

Calculate $\frac{RH(T_j)}{N}$ and using Eq. 4.2.3-2. Evaluate $\frac{e_h(T_j)}{N}$ using

$$\frac{e_h(T_j)}{N} = \frac{X^{k=3}(T_j) * \dot{E}_h^{k=3}(T_j) * \delta'(T_j)}{PLF_j} * \frac{n_j}{N}$$

where:

$X^{k=3}(T_j) = BL(T_j)/\dot{Q}_h^{k=3}(T_j)$ and $PLF_j = 1 - C_D^h(k = 3) * [1 - X^{k=3}(T_j)]$

Determine the low temperature cut-out factor, $\delta'(T_j)$, using Eq. 4.2.3–3. Use the booster-capacity cyclic-degradation coefficient, $C_D^h(k=3)$ determined in accordance with section 3.6.6 of this appendix.

4.2.6.4 Heat Pump Alternates Between High (k=2) and Low (k=1) Compressor Capacity to Satisfy the Building Heating Load at a Temperature T_j, $\dot{Q}_h^{k=1}(T_j) < BL(T_j) < \dot{Q}_h^{k=2}(T_j)$

Evaluate the quantities

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

as specified in section 4.2.3.2 of this appendix. Determine the equation inputs $X^{k=1}(T_j)$, $X^{k=2}(T_j)$, and $\delta'(T_j)$ as specified in section 4.2.3.2 of this appendix.

4.2.6.5 Heat Pump Alternates Between High (k=2) and Booster (k=3) Compressor Capacity To Satisfy the Building Heating Load at a Temperature T_j, $\dot{Q}_h^{k=2}(T_j) < BL(T_j) < \dot{Q}_h^{k=3}(T_j)$

Calculate $\frac{RH(T_j)}{N}$ and using Eq. 4.2.3-2. Evaluate $\frac{e_h(T_j)}{N}$ using

$$\frac{e_h(T_j)}{N} = [X^{k=2}(T_j) * \dot{E}_h^{k=2}(T_j) + X^{k=3}(T_j) * \dot{E}_h^{k=3}(T_j)] * \delta'(T_j) * \frac{n_j}{N}$$

where:

$$X^{k=2}(T_j) = \frac{\dot{Q}_h^{k=3}(T_j) - BL(T_j)}{\dot{Q}_h^{k=3}(T_j) - \dot{Q}_h^{k=2}(T_j)}$$

and $X^{k=3}(T_j) = X^{k=2}(T_j)$ = the heating mode, booster capacity load factor for temperature bin j, dimensionless. Determine the low temperature cut-out factor, $\delta'(T_j)$, using Eq. 4.2.3–3.

4.2.6.6 Heat Pump Only Operates at Low (k=1) Capacity at Temperature T_j and Its Capacity Is Less Than the Building Heating Load, $BL(T_j) > \dot{Q}_h^{k=1}(T_j)$

$$\frac{e_h(T_j)}{N} = \dot{E}_h^{k=1}(T_j) * \delta'(T_j) * \frac{n_j}{N} \quad \text{and} \frac{RH(T_j)}{N} = \frac{BL(T_j) - [\dot{Q}_h^{k=1}(T_j) * \delta\prime(T_j)]}{3.413 \frac{Btu/h}{W}} * \frac{n_j}{N}$$

where the low temperature cut-out factor, $\delta'(Tj)$, is calculated using Eq. 4.2.3–3.

4.2.6.7 Heat Pump Only Operates at High (k=2) Capacity at Temperature Tj and Its Capacity Is Less Than the Building Heating Load, $BL(Tj) > \dot{Q}_h^{k=2}(T_j)$

Evaluate the quantities

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

as specified in section 4.2.3.4 of this appendix. Calculate $\delta''(Tj)$ using the equation given in section 4.2.3.4 of this appendix.

4.2.6.8 Heat Pump Only Operates at Booster (k=3) Capacity at Temperature Tj and Its Capacity Is Less Than the Building Heating Load, $BL(T_j) > \dot{Q}_h^{k=3}(T_j)$ or the System Converts to Using Only Resistive Heating

$$\frac{e_h(T_j)}{N} = \dot{E}_h^{k=3}(T_j) * \delta'(T_j) * \frac{n_j}{N} \quad \text{and} \frac{RH(T_j)}{N} = \frac{BL(T_j) - [\dot{Q}_h^{k=3}(T_j) * \delta\prime(T_j)]}{3.413 \frac{Btu/h}{W}} * \frac{n_j}{N}$$

where $\delta''(Tj)$ is calculated as specified in section 4.2.3.4 of this appendix if the heat pump is operating at its booster compressor capacity. If the heat pump system converts to using only resistive heating at outdoor temperature T_j, set $\delta'(T_j)$ equal to zero.

4.2.7 Additional Steps for Calculating the HSPF of a Heat Pump Having a Single Indoor Unit With Multiple Indoor Blowers

The calculation of the Eq. 4.2–1 quantities $e_h(T_j)/N$ and $RH(T_j)/N$ are evaluated as specified in the applicable subsection.

4.2.7.1 For Multiple Indoor Blower Heat Pumps That Are Connected to a Singular, Single-Speed Outdoor Unit

a. Calculate the space heating capacity, $\dot{Q}_h^{k=1}(Tj)$, and electrical power consumption, $\dot{E}_h^{k=1}(Tj)$, of the heat pump when operating at the heating minimum air volume rate and outdoor temperature T_j using Eqs. 4.2.2–3 and 4.2.2–4, respectively. Use these same equations to calculate the space heating capacity, $\dot{Q}_h^{k=2}(Tj)$ and electrical power consumption, $\dot{E}_h^{k=2}(Tj)$, of the test unit when operating at the heating full-load air volume rate and outdoor temperature T_j. In evaluating Eqs. 4.2.2–3 and 4.2.2– 4, determine the quantities $\dot{Q}_h^{k=1}(47)$ and $\dot{E}_h^{k=1}(47)$ from the $H1_1$ test; determine $\dot{Q}_h^{k=2}(47)$ and $\dot{E}_h^{k=2}(47)$ from the $H1_2$ test. Evaluate all four quantities according to section 3.7 of this appendix. Determine the quantities $\dot{Q}_h^{k=1}(35)$ and $\dot{E}_h^{k=1}(35)$ as specified in section 3.6.2 of this appendix. Determine $\dot{Q}_h^{k=2}(35)$ and $\dot{E}_h^{k=2}(35)$ from the $H2_2$ frost accumulation test as calculated according to section 3.9.1 of this appendix. Determine the quantities $\dot{Q}_h^{k=1}(17)$ 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. Evaluate all four quantities according to section 3.10 of this appendix. Refer to section 3.6.2 and Table 12 of this appendix for additional information on the referenced laboratory tests.

b. Determine the heating mode cyclic degradation coefficient, CD_h, as per sections

3.6.2 and 3.8 to 3.8.1 of this appendix. Assign this same value to $CD_h(k = 2)$.

c. Except for using the above values of $\dot{Q}_h^{k=1}(Tj)$, $\dot{E}_h^{k=1}(Tj)$, $\dot{Q}_h^{k=2}(Tj)$, $\dot{E}_h^{k=2}(Tj)$, CD_h, and $CD_h(k = 2)$, calculate the quantities $e_h(T_j)/N$ as specified in section 4.2.3.1 of this appendix for cases where $\dot{Q}_h^{k=1}(Tj) \geq BL(T_j)$. For all other outdoor bin temperatures, T_j, calculate $e_h(Tj)/N$ and $RH_h(Tj)/N$ as specified in section 4.2.3.3 of this appendix if $\dot{Q}_h^{k=2}(Tj) > BL(Tj)$ or as specified in section 4.2.3.4 of this appendix if $\dot{Q}_h^{k=2}(Tj) \leq BL(T_j)$.

4.2.7.2 For Multiple Indoor Blower Heat Pumps Connected to Either a Single Outdoor Unit With a Two-capacity Compressor or to Two Separate Single-Speed Outdoor Units of Identical Model, calculate the quantities $e_h(T_j)/N$ and $RH(T_j)/N$ as specified in section 4.2.3 of this appendix.

4.3 Calculations of Off-mode Power Consumption

For central air conditioners and heat pumps with a cooling capacity of:

Less than 36,000 Btu/h, determine the off mode represented value, $P_{W,OFF}$, with the following equation:

$$P_{W,OFF} = \frac{P1 + P2}{2};$$

greater than or equal to 36,000 Btu/h, calculate the capacity scaling factor according to:

$$F_{scale} = \frac{\dot{Q}_C(95)}{36,000},$$

where $\dot{Q}_C(95)$ is the total cooling capacity at the A or A_2 test condition, and determine the off mode represented value, $P_{W,OFF}$, with the following equation:

$$P_{W,OFF} = \frac{P1 + P2}{2 \times F_{scale}};$$

4.4 Rounding of SEER and HSPF for Reporting Purposes

After calculating SEER according to section 4.1 of this appendix and HSPF according to section 4.2 of this appendix round the values off as specified per § 430.23(m) of title 10 of the Code of Federal Regulations.

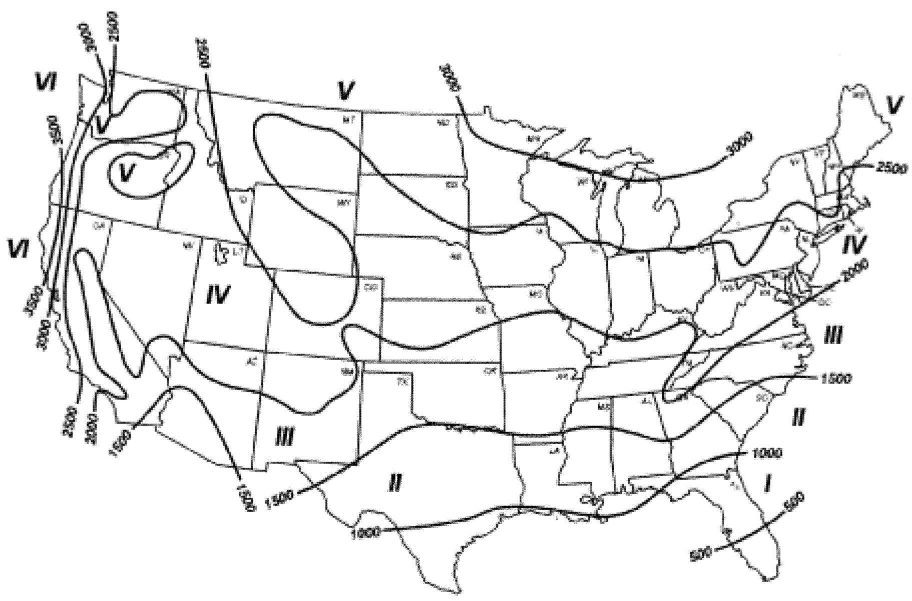

Figure 1—Heating Load Hours (HLH_A) for the United States

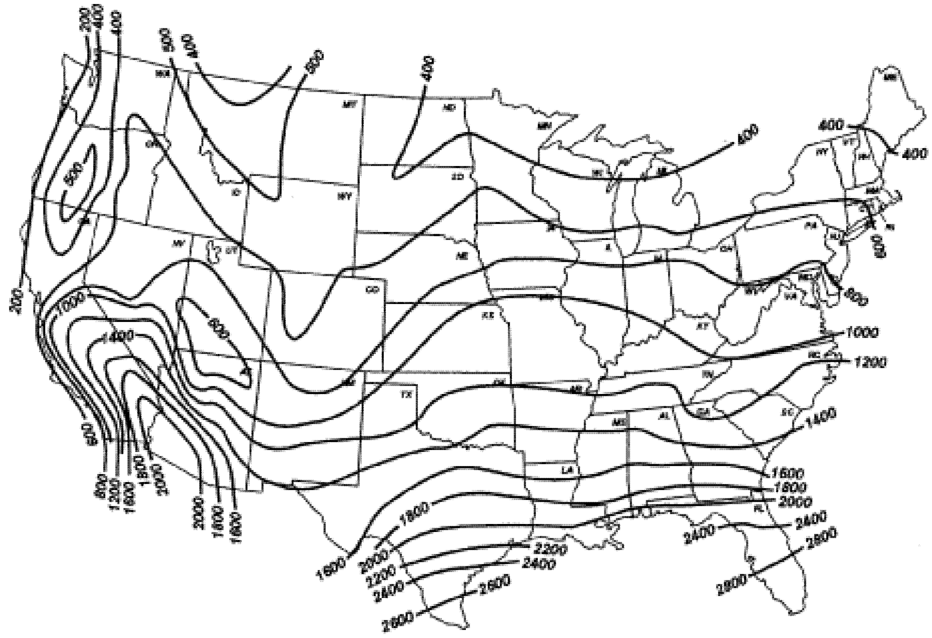

Figure 2—Cooling Load Hours (CLH_A) for the United States

TABLE 22—REPRESENTATIVE COOLING AND HEATING LOAD HOURS FOR EACH GENERALIZED CLIMATIC REGION

Climatic region	Cooling load hours CLH_R	Heating load hours HLH_R
I	2,400	750
II	1,800	1,250
III	1,200	1,750
IV	800	2,250
Rating Values	1,000	2,080
V	400	2,750
VI	200	2,750

4.5 Calculations of the SHR, Which Should Be Computed for Different Equipment Configurations and Test Conditions Specified in Table 23

TABLE 23—APPLICABLE TEST CONDITIONS FOR CALCULATION OF THE SENSIBLE HEAT RATIO

Equipment configuration	Reference table Number of appendix M	SHR computation with results from	Computed values
Units Having a Single-Speed Compressor and a Fixed-Speed Indoor blower, a Constant Air Volume Rate Indoor blower, or No Indoor blower.	4	B Test	SHR(B).
Units Having a Single-Speed Compressor That Meet the section 3.2.2.1 Indoor Unit Requirements.	5	B2 and B1 Tests	SHR(B1), SHR(B2).
Units Having a Two-Capacity Compressor	6	B2 and B1 Tests	SHR(B1), SHR(B2).
Units Having a Variable-Speed Compressor	7	B2 and B1 Tests	SHR(B1), SHR(B2).

The SHR is defined and calculated as follows:

$$SHR = \frac{Sensible\ Cooling\ Capacity}{Total\ Cooling\ Capacity}$$

$$= \frac{\dot{Q}_{sc}^{k}(T)}{\dot{Q}_{c}^{k}(T)}$$

Where both the total and sensible cooling capacities are determined from the same cooling mode test and calculated from data collected over the same 30-minute data collection interval.

4.6 Calculations of the Energy Efficiency Ratio (EER).

Calculate the energy efficiency ratio using.

$$EER = \frac{Total\ Cooling\ Capacity}{Total\ Electrical\ Power\ Consumption}$$

$$= \frac{\dot{Q}_{c}^{k}(T)}{\dot{E}_{c}^{k}(T)}$$

where $\dot{Q}_c{}^k(T)$ and $\dot{E}_c{}^k(T)$ are the space cooling capacity and electrical power consumption determined from the 30-minute data collection interval of the same steady-state wet coil cooling mode test and calculated as specified in section 3.3 of this appendix. Add the letter identification for each steady-state test as a subscript (*e.g., EER_{A_2}*) to differentiate among the resulting EER values.

■ 10. Add appendix M1 to subpart B of part 430 to read as follows:

Appendix M1 to Subpart B of Part 430—Uniform Test Method for Measuring the Energy Consumption of Central Air Conditioners and Heat Pumps

Prior to January 1, 2023, any representations, including compliance certifications, made with respect to the energy use, power, or efficiency of central air conditioners and central air conditioning heat pumps must be based on the results of testing pursuant to appendix M of this subpart.

On or after January 1, 2023, any representations, including compliance certifications, made with respect to the energy use, power, or efficiency of central air conditioners and central air conditioning heat pumps must be based on the results of testing pursuant to this appendix.

1 Scope and Definitions

1.1 Scope

This test procedure provides a method of determining SEER2, EER2, HSPF2 and $P_{W,OFF}$ for central air conditioners and central air conditioning heat pumps including the following categories:

(h) Split-system air conditioners, including single-split, multi-head mini-split, multi-split (including VRF), and multi-circuit systems

(i) Split-system heat pumps, including single-split, multi-head mini-split, multi-split (including VRF), and multi-circuit systems

(j) Single-package air conditioners

(k) Single-package heat pumps

(l) Small-duct, high-velocity systems (including VRF)

(m) Space-constrained products—air conditioners

(n) Space-constrained products—heat pumps

For the purposes of this appendix, the Department of Energy incorporates by reference specific sections of several industry standards, as listed in § 430.3. In cases where there is a conflict, the language of the test procedure in this appendix takes precedence over the incorporated standards.

All section references refer to sections within this appendix unless otherwise stated.

1.2 Definitions

Airflow-control settings are programmed or wired control system configurations that control a fan to achieve discrete, differing ranges of airflow—often designated for performing a specific function (*e.g.,* cooling, heating, or constant circulation)—without manual adjustment other than interaction with a user-operable control (*i.e.,* a thermostat) that meets the manufacturer specifications for installed-use. For the purposes of this appendix, manufacturer specifications for installed-use are those found in the product literature shipped with the unit.

Air sampling device is an assembly consisting of a manifold with several branch tubes with multiple sampling holes that draws an air sample from a critical location from the unit under test (*e.g.* indoor air inlet, indoor air outlet, outdoor air inlet, etc.).

Airflow prevention device denotes a device that prevents airflow via natural convection by mechanical means, such as an air damper box, or by means of changes in duct height, such as an upturned duct.

Aspirating psychrometer is a piece of equipment with a monitored airflow section that draws uniform airflow through the measurement section and has probes for measurement of air temperature and humidity.

Blower coil indoor unit means an indoor unit either with an indoor blower housed with the coil or with a separate designated air mover such as a furnace or a modular blower (as defined in appendix AA to this subpart).

Blower coil system refers to a split system that includes one or more blower coil indoor units.

Cased coil means a coil-only indoor unit with external cabinetry.

Ceiling-mount blower coil system means a split system for which a) the outdoor unit has a certified cooling capacity less than or equal to 36,000 Btu/h; b) the indoor unit(s) is/are shipped with manufacturer-supplied installation instructions that specify to secure the indoor unit only to the ceiling, within a furred-down space, or above a dropped ceiling of the conditioned space, with return air directly to the bottom of the unit without ductwork, or through the furred-down space, or optional insulated return air plenum that is shipped with the indoor unit; c) the installed height of the indoor unit is no more than 12 inches (not including condensate drain lines) and the installed depth (in the direction of airflow) of the indoor unit is no more than 30 inches; and d) supply air is discharged horizontally.

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. Determine these rate quantities from a single test or, if derived via interpolation, determine at a single set of operating conditions. COP is a dimensionless quantity. When determined for a ducted coil-only system, COP must be calculated using the default values for heat output and power input of a fan motor specified in sections 3.7 and 3.9.1 of this appendix.

Coil-only indoor unit means an indoor unit that is distributed in commerce without an indoor blower or separate designated air mover. A coil-only indoor unit installed in the field relies on a separately installed furnace or a modular blower for indoor air movement.

Coil-only system means a system that includes only (one or more) coil-only indoor units.

Condensing unit removes the heat absorbed by the refrigerant to transfer it to the outside environment and consists of an outdoor coil, compressor(s), and air moving device.

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

Continuously recorded, when referring to a dry bulb measurement, dry bulb temperature used for test room control, wet bulb temperature, dew point temperature, or relative humidity measurements, means that the specified value must be sampled at regular intervals that are equal to or less than 15 seconds.

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, and as its denominator 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.

Crankcase heater means any electrically powered device or mechanism for intentionally generating heat within and/or around the compressor sump volume. Crankcase heater control may be achieved using a timer or may be based on a change in temperature or some other measurable parameter, such that the crankcase heater is not required to operate continuously. A crankcase heater without controls operates continuously when the compressor is not operating.

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.

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

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.

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 either:

(1) 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) at least once for every ten minutes of compressor ON-time when space heating; or

(2) Operate as a feedback system that measures the length of the defrost period and adjusts defrost frequency accordingly. 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. (**Note:** Systems that vary defrost intervals according to outdoor dry-bulb temperature are not demand-defrost systems.) A demand-defrost control system, which otherwise meets the requirements, may allow time-initiated defrosts if, and only if, such defrosts occur after 6 hours of compressor operating time.

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 of this appendix.

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

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-package unit.

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. Determine these rate quantities from a single test or, if derived via interpolation, determine at a single set of operating conditions. EER is expressed in units of

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

When determined for a ducted coil-only system, EER must include, from this appendix, the section 3.3 and 3.5.1 default values for the heat output and power input of a fan motor. The represented value of EER determined in accordance with appendix M1 is EER2.

Evaporator coil means an assembly that absorbs heat from an enclosed space and transfers the heat to a refrigerant.

Heat pump means a kind of central air conditioner that utilizes an indoor conditioning coil, compressor, and refrigerant-to-outdoor air heat exchanger to provide air heating, and may also provide air cooling, air dehumidifying, air humidifying, air circulating, and air cleaning.

Heat pump having a heat comfort controller means a heat pump with controls that can regulate the operation of the electric resistance elements to assure that the air temperature leaving the indoor section does not fall below a specified temperature. 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.

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, and its denominator the heating capacity measured at the same test conditions used for the cyclic test, multiplied by the total time interval (ON plus OFF) of the cyclic-test.

Heating season means the months of the year that require heating, *e.g.,* typically, and roughly, October through April.

Heating seasonal performance factor 2 (HSPF2) means the total space heating required during the heating season, expressed in Btu, divided by the total electrical energy consumed by the heat pump system during the same season, expressed in watt-hours. The HSPF2 used to evaluate compliance with 10 CFR 430.32(c) is based on Region IV and the sampling plan stated in 10 CFR 429.16(a). HSPF2 is determined in accordance with appendix M1.

Independent coil manufacturer (ICM) means a manufacturer that manufactures indoor units but does not manufacture single-package units or outdoor units.

Indoor unit means a separate assembly of a split system that includes—

(a) An arrangement of refrigerant-to-air heat transfer coil(s) for transfer of heat between the refrigerant and the indoor air,

(b) A condensate drain pan, and may or may not include,

(c) Sheet metal or plastic parts not part of external cabinetry to direct/route airflow over the coil(s),

(d) A cooling mode expansion device,

(e) External cabinetry, and

(f) An integrated indoor blower (*i.e.* a device to move air including its associated motor). A separate designated air mover that may be a furnace or a modular blower (as defined in appendix AA to the subpart) may be considered to be part of the indoor unit. A service coil is not an indoor unit.

Low-static blower coil system means a ducted multi-split or multi-head mini-split system for which all indoor units produce greater than 0.01 in. wc. and a maximum of 0.35 in. wc. external static pressure when operated at the cooling full-load air volume rate not exceeding 400 cfm per rated ton of cooling.

Mid-static blower coil system means a ducted multi-split or multi-head mini-split system for which all indoor units produce greater than 0.20 in. wc. and a maximum of 0.65 in. wc. when operated at the cooling full-load air volume rate not exceeding 400 cfm per rated ton of cooling.

Minimum-speed-limiting variable-speed heat pump means a heat pump for which the compressor speed (represented by revolutions per minute or motor power input frequency) is higher than its value for operation in a 47 °F ambient temperature for any bin temperature T_j for which the calculated heating load is less than the calculated intermediate-speed capacity.

Mobile home blower coil system means a split system that contains an outdoor unit and an indoor unit that meet the following criteria:

(1) Both the indoor and outdoor unit are shipped with manufacturer-supplied installation instructions that specify installation only in a mobile home with the home and equipment complying with HUD Manufactured Home Construction Safety Standard 24 CFR part 3280;

(2) The indoor unit cannot exceed 0.40 in. wc. when operated at the cooling full-load air

volume rate not exceeding 400 cfm per rated ton of cooling; and

(3) The indoor and outdoor unit each must bear a label in at least ¼ inch font that reads ''For installation only in HUD manufactured home per Construction Safety Standard 24 CFR part 3280.''

Mobile home coil-only system means a coil-only split system that includes an outdoor unit and coil-only indoor unit that meet the following criteria:

(1) The outdoor unit is shipped with manufacturer-supplied installation instructions that specify installation only for mobile homes that comply with HUD Manufactured Home Construction Safety Standard 24 CFR part 3280,

(2) The coil-only indoor unit is shipped with manufacturer-supplied installation instructions that specify installation only in or with a mobile home furnace, modular blower, or designated air mover that complies with HUD Manufactured Home Construction Safety Standard 24 CFR part 3280, and has dimensions no greater than 20'' wide, 34'' high and 21'' deep, and

(3) The coil-only indoor unit and outdoor unit each has a label in at least ¼ inch font that reads ''For installation only in HUD manufactured home per Construction Safety Standard 24 CFR part 3280.''

Multi-head mini-split system means a split system that has one outdoor unit and that has two or more indoor units connected with a single refrigeration circuit. The indoor units operate in unison in response to a single indoor thermostat.

Multiple-circuit (or multi-circuit) system means a split system that has one outdoor unit and that has two or more indoor units installed on two or more refrigeration circuits such that each refrigeration circuit serves a compressor and one and only one indoor unit, and refrigerant is not shared from circuit to circuit.

Multiple-split (or multi-split) system means a split system that has one outdoor unit and two or more coil-only indoor units and/or blower coil indoor units connected with a single refrigerant circuit. The indoor units operate independently and can condition multiple zones in response to at least two indoor thermostats or temperature sensors. The outdoor unit operates in response to independent operation of the indoor units based on control input of multiple indoor thermostats or temperature sensors, and/or based on refrigeration circuit sensor input (*e.g.,* suction pressure).

*Nominal capacity means t*he capacity that is claimed by the manufacturer on the product name plate. Nominal cooling capacity is approximate to the air conditioner cooling capacity tested at A or A_2 condition. Nominal heating capacity is approximate to the heat pump heating capacity tested in the $H1_N$ test.

Non-ducted indoor unit means an indoor unit that is designed to be permanently installed, mounted on room walls and/or ceilings, and that directly heats or cools air within the conditioned space.

Normalized Gross Indoor Fin Surface (NGIFS) means the gross fin surface area of the indoor unit coil divided by the cooling capacity measured for the A or A_2 Test, whichever applies.

Off-mode power consumption means the power consumption when the unit is connected to its main power source but is neither providing cooling nor heating to the building it serves.

Off-mode season means, for central air conditioners other than heat pumps, the shoulder season and the entire heating season; and for heat pumps, the shoulder season only.

Outdoor unit means a separate assembly of a split system that transfers heat between the refrigerant and the outdoor air, and consists of an outdoor coil, compressor(s), an air moving device, and in addition for heat pumps, may include a heating mode expansion device, reversing valve, and/or defrost controls.

Outdoor unit manufacturer (OUM) means a manufacturer of single-package units, outdoor units, and/or both indoor units and outdoor units.

Part-load factor (PLF) means the ratio of the cyclic EER (or COP for heating) to the steady-state EER (or COP), where both EERs (or COPs) are determined based on operation at the same ambient conditions.

Seasonal energy efficiency ratio 2 (SEER2) 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 central air conditioner or heat pump during the same season, expressed in watt-hours. SEER2 is determined in accordance with appendix M1.

Service coil means an arrangement of refrigerant-to-air heat transfer coil(s), condensate drain pan, sheet metal or plastic parts to direct/route airflow over the coil(s), which may or may not include external cabinetry and/or a cooling mode expansion device, distributed in commerce solely for replacing an uncased coil or cased coil that has already been placed into service, and that has been labeled ''for indoor coil replacement only'' on the nameplate and in manufacturer technical and product literature. The model number for any service coil must include some mechanism (*e.g.,* an additional letter or number) for differentiating a service coil from a coil intended for an indoor unit.

Shoulder season means the months of the year in between those months that require cooling and those months that require heating, *e.g.,* typically, and roughly, April through May, and September through October.

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

Single-split system means a split system that has one outdoor unit and one indoor unit connected with a single refrigeration circuit.

Small-duct, high-velocity system means a split system for which all indoor units are blower coil indoor units that produce at least 1.2 inches (of water column) of external static pressure when operated at the full-load air volume rate certified by the manufacturer of at least 220 scfm per rated ton of cooling.

Split system means any central air conditioner or heat pump that has at least two separate assemblies that are connected with refrigerant piping when installed. One of these assemblies includes an indoor coil that exchanges heat with the indoor air to provide heating or cooling, while one of the others includes an outdoor coil that exchanges heat with the outdoor air. Split systems may be either blower coil systems or coil-only systems.

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

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.

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.

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

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.

Tested combination means a multi-head mini-split, multi-split, or multi-circuit system having the following features:

(1) The system consists of one outdoor unit with one or more compressors matched with between two and five indoor units;

(2) The indoor units must:

(i) Collectively, have a nominal cooling capacity greater than or equal to 95 percent and less than or equal to 105 percent of the nominal cooling capacity of the outdoor unit;

(ii) Each represent the highest sales volume model family, if this is possible while meeting all the requirements of this section. If this is not possible, one or more of the indoor units may represent another indoor model family in order that all the other requirements of this section are met.

(iii) Individually not have a nominal cooling capacity greater than 50 percent of the nominal cooling capacity of the outdoor unit, unless the nominal cooling capacity of the outdoor unit is 24,000 Btu/h or less;

(iv) Operate at fan speeds consistent with manufacturer's specifications; and

(v) All be subject to the same minimum external static pressure requirement while able to produce the same external static pressure at the exit of each outlet plenum when connected in a manifold configuration as required by the test procedure.

(3) Where referenced, ''nominal cooling capacity'' means, for indoor units, the highest cooling capacity listed in published product literature for 95 °F outdoor dry bulb temperature and 80 °F dry bulb, 67 °F wet bulb indoor conditions, and for outdoor units, the lowest cooling capacity listed in published product literature for these conditions. If incomplete or no operating conditions are published, use the highest (for indoor units) or lowest (for outdoor units) such cooling capacity available for sale.

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

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 or not 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).

Triple-capacity, northern heat pump means a heat pump that provides two stages of cooling and three stages of heating. The two common stages for both the cooling and heating modes are the low capacity stage and the high capacity stage. The additional heating mode stage is the booster capacity stage, which offers the highest heating capacity output for a given set of ambient operating conditions.

Triple-split system means a split system that is composed of three separate assemblies: An outdoor fan coil section, a blower coil indoor unit, and an indoor compressor section.

Two-capacity (or two-stage) compressor system means a central air conditioner or heat pump that has a compressor or a group of compressors operating with only two stages of capacity. For such systems, low capacity means the compressor(s) operating at low stage, or at low load test conditions. The low compressor stage that operates for heating mode tests may be the same or different from the low compressor stage that operates for cooling mode tests. For such systems, high capacity means the compressor(s) operating at high stage, or at full load test conditions.

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 heat pump is a two-capacity northern heat pump only when this feature is enabled at all times. The certified indoor coil model number must 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.

Uncased coil means a coil-only indoor unit without external cabinetry.

Variable refrigerant flow (VRF) system means a multi-split system with at least three compressor capacity stages, distributing refrigerant through a piping network to multiple indoor blower coil units each capable of individual zone temperature control, through proprietary zone temperature control devices and a common communications network. Note: Single-phase VRF systems less than 65,000 Btu/h are central air conditioners and central air conditioning heat pumps.

Variable-speed compressor system means a central air conditioner or heat pump that has a compressor that uses a variable-speed drive to vary the compressor speed to achieve variable capacities. *Wall-mount blower coil system* means a split system air conditioner or heat pump for which:

(a) The outdoor unit has a certified cooling capacity less than or equal to 36,000 Btu/h;

(b) The indoor unit(s) is/are shipped with manufacturer-supplied installation instructions that specify mounting only by:

(1) Securing the back side of the unit to a wall within the conditioned space, or

(2) Securing the unit to adjacent wall studs or in an enclosure, such as a closet, such that the indoor unit's front face is flush with a wall in the conditioned space;

(c) Has front air return without ductwork and is not capable of horizontal air discharge; and

(d) Has a height no more than 45 inches, a depth (perpendicular to the wall) no more than 22 inches (including tubing connections), and a width no more than 24 inches (parallel to the wall).

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 Overview and Conditions

(A) Test VRF systems using AHRI 1230–2010 (incorporated by reference, see § 430.3) and appendix M. Where AHRI 1230–2010 refers to the appendix C therein substitute the provisions of this appendix. In cases where there is a conflict, the language of the test procedure in this appendix takes precedence over AHRI 1230–2010.

For definitions use section 1 of appendix M and section 3 of AHRI 1230–2010. For rounding requirements, refer to § 430.23(m). For determination of certified ratings, refer to § 429.16 of this chapter.

For test room requirements, refer to section 2.1 of this appendix. For test unit installation requirements refer to sections 2.2.a, 2.2.b, 2.2.c, 2.2.1, 2.2.2, 2.2.3.a, 2.2.3.c, 2.2.4, 2.2.5, and 2.4 to 2.12 of this appendix, and sections 5.1.3 and 5.1.4 of AHRI 1230–2010. The ''manufacturer's published instructions,'' as stated in section 8.2 of ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3) and ''manufacturer's installation instructions'' discussed in this appendix mean the manufacturer's installation instructions that come packaged with or appear in the labels applied to the unit. This does not include online manuals. Installation instructions that appear in the labels applied to the unit take precedence over installation instructions that are shipped with the unit.

For general requirements for the test procedure, refer to section 3.1 of this appendix, except for sections 3.1.3 and 3.1.4, which are requirements for indoor air volume and outdoor air volume. For indoor air volume and outdoor air volume requirements, refer instead to section 6.1.5 (except where section 6.1.5 refers to Table 8, refer instead to Table 4 of this appendix) and 6.1.6 of AHRI 1230–2010.

For the test method, refer to sections 3.3 to 3.5 and 3.7 to 3.13 of this appendix. For cooling mode and heating mode test conditions, refer to section 6.2 of AHRI 1230–2010. For calculations of seasonal performance descriptors, refer to section 4 of this appendix.

(B) For systems other than VRF, only a subset of the sections listed in this test procedure apply when testing and determining represented values for a particular unit. Table 1 shows the sections of the test procedure that apply to each system. This table is meant to assist manufacturers in finding the appropriate sections of the test procedure; the appendix sections rather than the table provide the specific requirements for testing, and given the varied nature of available units, manufacturers are responsible for determining which sections apply to each unit tested based on the model characteristics. To use this table, first refer to the sections listed under ''all units''. Then refer to additional requirements based on:

(1) System configuration(s),

(2) The compressor staging or modulation capability, and

(3) Any special features.

Testing requirements for space-constrained products do not differ from similar equipment that is not space-constrained and thus are not listed separately in this table. Air conditioners and heat pumps are not listed separately in this table, but heating procedures and calculations apply only to heat pumps.

Table 1 Informative Guidance for Using Appendix M1

			Testing conditions	Testing procedures			Calculations		
			General	**General**	**Cooling***	**Heating****	**General**	**Cooling***	**Heating****
Requirements for all units (except VRF)			2.1; 2.2a-c; 2.2.1; 2.2.4; 2.2.4.1; 2.2.4.1 (1); 2.2.4.2; 2.2.5.1-5; 2.2.5.7-8; 2.3; 2.3.1; 2.3.2; 2.4; 2.4.1a,d; 2.5a-c; 2.5.1; 2.5.2 - 2.5.4.2; 2.5.5 – 2.13	3.1; 3.1.1-3; 3.1.5-9; 3.11; 3.12	3.3; 3.4; 3.5a-i	3.1.4.7; 3.1.9; 3.7a,b,d; 3.8a,d; 3.8.1; 3.9; 3.10	4.4; 4.5	4.1	4.2
Additional Requirements	**System Configurations (more than one may apply)**	Single-split system – blower coil	2.2a(1)		3.1.4.1.1; 3.1.4.1.1a,b; 3.1.4.2a-b; 3.1.4.3a-b	3.1.4.4.1; 3.1.4.4.2; 3.1.4.4.3a-b; 3.1.4.5.1; 3.1.4.5.2a-c; 3.1.4.6a-b			
		Single-split system - coil-only	2.2a(1); 2.2d,e; 2.4.2		3.1.4.1.1; 3.1.4.1.1c; 3.1.4.2c; 3.5.1	3.1.4.4.1; 3.1.4.4.2; 3.1.4.4.3c; 3.1.4.5.2d; 3.7c; 3.8b; 3.9f; 3.9.1b			
		Tri-split	2.2a(2)						
		Outdoor unit with no match	2.2e						
		Single-package	2.2.4.1(2); 2.2.5.6b; 2.4.2		3.1.4.1.1; 3.1.4.1.1a,b; 3.1.4.2a-b; 3.1.4.3a-b	3.1.4.4.1; 3.1.4.4.2; 3.1.4.4.3a-b; 3.1.4.5.1; 3.1.4.5.2a-c; 3.1.4.6a-b			
		Heat pump	2.2.5.6.a						
		Heating-only heat pump			3.1.4.1.1 Table 5	3.1.4.4.3			

		Two-capacity northern heat pump		3.1.4.4.2c; 3.1.4.5.2 c- d	3.2.3c	3.6.3			
		Triple-capacity northern heat pump			3.2.5	3.6.6			4.2.6
		SDHV (non-VRF)	2.2b; 2.4.1c; 2.5.4.3						
		Single- zone-multi-coil split and non-VRF multiple-split with duct	2.2a(1),(3); 2.2.3; 2.4.1b		3.1.4.1.1; 3.1.4.1.1a-b; 3.1.4.2a-b; 3.1.4.3a-b	3.1.4.4.1; 3.1.4.4.2; 3.1.4.4.3a-b; 3.1.4.5.1; 3.1.4.5.2a-c; 3.1.4.6a-b			
		Single-zone-multi-coil split and non-VRF multiple-split, ductless	2.2.a(1),(3); 2.2.3		3.1.4.1.2; 3.1.4.2d; 3.1.4.3c; 3.2.4c; 3.5c,g,h; 3.5.2; 3.8c	3.1.4.4.4; 3.1.4.5.2e; 3.1.4.6c; 3.6.4.c; 3.8c			
		VRF multiple-split† and VRF SDHV†	2.1; 2.2.a; 2.2.b; 2.2.c; 2.2.1; 2.2.2; 2.2.3.a; 2.2.3.c; 2.2.4; 2.2.5; 2.4-2.12	3.1 (except 3.1.3, 3.1.4) 3.1.4.1.1c; 3.11-3.13	3.3-3.5	3.7–3.10	4.4; 4.5	4.1	4.2
	Modulation Capability	Single speed compressor, fixed air volume rate			3.2.1	3.6.1		4.1.1	4.2.1
		Single speed compressor, VAV fan			3.2.2	3.6.2		4.1.2	4.2.2
		Two-capacity compressor		3.1.9	3.2.3	3.6.3		4.1.3	4.2.3
		Variable-speed compressor			3.2.4	3.6.4		4.1.4	4.2.4
	Special Features	Heat pump with heat comfort controller				3.6.5			4.2.5
		Units with a multi-speed outdoor fan	2.2.2						
		Single indoor unit having multiple indoor blowers			3.2.6	3.6.2; 3.6.7		4.1.5	4.2.7

*Does not apply to heating-only heat pumps.

**Applies only to heat pumps; not to air conditioners.

[†]Use AHRI 1230-2010 (incorporated by reference, see §430.3), with the sections referenced in section 2(A) of this appendix, in conjunction with the sections set forth in the table to perform test setup, testing, and calculations for determining represented values for VRF multiple-split and VRF SDHV systems.

NOTE: For all units, use section 3.13 of this appendix for off mode testing procedures and section 4.3 of this appendix for off mode calculations. For all units subject to an EER2 standard, use section 4.6 of this appendix to determine the energy efficiency ratio.

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, single-zone-multi-coil or multi-circuit air conditioners and heat pumps, however, use as many 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 ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3).

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. 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 ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3), subject to the following additional requirements:

(1) When testing split systems, follow the requirements given in section 6.1.3.5 of AHRI 210/240–2008 (incorporated by reference, see § 430.3). For the vapor refrigerant line(s), use the insulation included with the unit; if no insulation is provided, use insulation meeting the specifications for the insulation in the installation instructions included with the unit by the manufacturer; if no insulation is included with the unit and the installation instructions do not contain provisions for insulating the line(s), fully insulate the vapor refrigerant line(s) with vapor proof insulation having an inside diameter that matches the refrigerant tubing and a nominal thickness of at least 0.5 inches. For the liquid refrigerant line(s), use the insulation included with the unit; if no insulation is provided, use insulation meeting the specifications for the insulation in the installation instructions included with the unit by the manufacturer; if no insulation is included with the unit and the installation instructions do not contain provisions for insulating the line(s), leave the liquid refrigerant line(s) exposed to the air for air conditioners and heat pumps that heat and cool; or, for heating-only heat pumps, insulate the liquid refrigerant line(s) with insulation having an inside diameter that matches the refrigerant tubing and a nominal thickness of at least 0.5 inches. However, these requirements do not take priority over instructions for application of insulation for the purpose of improving refrigerant temperature measurement accuracy as required by sections 2.10.2 and 2.10.3 of this appendix. Insulation must be the same for the cooling and heating tests.

(2) When testing split systems, if the indoor unit does not ship with a cooling mode expansion device, test the system using the device as specified in the installation instructions provided with the indoor unit. If none is specified, test the system using a fixed orifice or piston type expansion device that is sized appropriately for the system.

(3) When testing triple-split systems (see section 1.2 of this appendix, Definitions), use the tubing length specified in section 6.1.3.5 of AHRI 210/240–2008 (incorporated by reference, see § 430.3) 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;

(4) When testing split systems having multiple indoor coils, connect each indoor blower coil unit to the outdoor unit using:

(a) 25 feet of tubing, or

(b) Tubing furnished by the manufacturer, whichever is longer.

(5) When testing split systems having multiple indoor coils, expose at least 10 feet of the system interconnection tubing to the outside conditions. If they are needed to make a secondary measurement of capacity or for verification of refrigerant charge, install refrigerant pressure measuring instruments as described in section 8.2.5 of ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3). Section 2.10 of this appendix specifies which secondary methods require refrigerant pressure measurements and section 2.2.5.5 of this appendix discusses use of pressure measurements to verify charge. 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, use the orientation for testing specified by the manufacturer in the certification report. 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 section 1 of this appendix, Definitions). 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 4, note 3 (see section 3.1.4 of this appendix). Except as noted in section 3.1.10 of this appendix, prevent the indoor air supplementary heating coils from operating during all tests. For uncased coils, create an enclosure using 1 inch fiberglass foil-faced ductboard having a nominal density of 6 pounds per cubic foot. Or alternatively, construct an enclosure using sheet metal or a similar material and insulating material having a thermal resistance (''R'' value) between 4 and 6 hr · ft² · °F/Btu. Size the enclosure and seal between the coil and/or drainage pan and the interior of the enclosure as specified in installation instructions shipped with the unit. Also seal between the plenum and inlet and outlet ducts.

d. When testing a coil-only system, install a toroidal-type transformer to power the system's low-voltage components, complying with any additional requirements for the transformer mentioned in the installation manuals included with the unit by the system manufacturer. If the installation manuals do not provide specifications for the transformer, use a transformer having the following features:

(1) A nominal volt-amp rating such that the transformer is loaded between 25 and 90 percent of this rating for the highest level of power measured during the off mode test (section 3.13 of this appendix);

(2) Designed to operate with a primary input of 230 V, single phase, 60 Hz; and

(3) That provides an output voltage that is within the specified range for each low-voltage component. Include the power consumption of the components connected to the transformer as part of the total system power consumption during the off mode tests; do not include the power consumed by the transformer when no load is connected to it.

e. Test an outdoor unit with no match (*i.e.,* that is not distributed in commerce with any indoor units) using a coil-only indoor unit with a single cooling air volume rate whose coil has:

(1) Round tubes of outer diameter no less than 0.375 inches, and

(2) A normalized gross indoor fin surface (NGIFS) no greater than 1.0 square inch per British thermal unit per hour (sq. in./Btu/hr). NGIFS is calculated as follows:

$$NGIFS = 2 \times L_f \times W_f \times N_f \div \dot{Q}_c(95)$$

where,

L_f = Indoor coil fin length in inches, also height of the coil transverse to the tubes.

W_f = Indoor coil fin width in inches, also depth of the coil.

N_f = Number of fins.

$\dot{Q}_c$ = the measured space cooling capacity of the tested outdoor unit/indoor unit combination as determined from the A_2 or A Test whichever applies, Btu/h.

f. If the outdoor unit or the outdoor portion of a single-package unit has a drain pan heater to prevent freezing of defrost water, energize the heater, subject to control to de-energize it when not needed by the heater's thermostat or the unit's control system, for all tests.

g. If pressure measurement devices are connected to a cooling/heating heat pump refrigerant circuit, the refrigerant charge M_t that could potentially transfer out of the connected pressure measurement systems (transducers, gauges, connections, and lines) between operating modes must be less than 2 percent of the factory refrigerant charge listed on the nameplate of the outdoor unit. If the outdoor unit nameplate has no listed refrigerant charge, or the heat pump is shipped without a refrigerant charge, use a factory refrigerant charge equal to 30 ounces per ton of certified cooling capacity. Use Equation 2.2–1 to calculate M_t for heat pumps that have a single expansion device located in the outdoor unit to serve each indoor unit, and use Equation 2.2–2 to calculate M_t for heat pumps that have two expansion devices per indoor unit.

$$\text{Equation 2.2-1} \qquad M_t = \rho * (V_5 * f_5 + V_6 * f_6 + V_3 + V_4 - V_2)$$

$$\text{Equation 2.2-2} \qquad M_t = \rho * (V_5 * f_5 + V_6 * f_6)$$

where:

V_i (i=2,3,4 . . .) = the internal volume of the pressure measurement system (pressure lines, fittings, and gauge and/or transducer) at the location i (as indicated in Table 2), (cubic inches)

f_i (i=5,6) = 0 if the pressure measurement system is pitched upwards from the pressure tap location to the gauge or transducer, 1 if it is not.

ρ = the density associated with liquid refrigerant at 100 °F bubble point conditions (ounces per cubic inch)

TABLE 2—PRESSURE MEASUREMENT LOCATIONS

Location	
Compressor Discharge	1
Between Outdoor Coil and Outdoor Expansion Valve(s)	2
Liquid Service Valve	3
Indoor Coil Inlet	4
Indoor Coil Outlet	5
Common Suction Port (*i.e.*, vapor service valve)	6
Compressor Suction	7

Calculate the internal volume of each pressure measurement system using internal volume reported for pressure transducers and gauges in product literature, if available. If such information is not available, use the value of 0.1 cubic inch internal volume for each pressure transducer, and 0.2 cubic inches for each pressure gauge.

In addition, for heat pumps that have a single expansion device located in the outdoor unit to serve each indoor unit, the internal volume of the pressure system at location 2 (as indicated in Table 2) must be no more than 1 cubic inches. Once the pressure measurement lines are set up, no change should be made until all tests are finished.

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 1 and Table 20 of section 4.2 of this appendix for information on region IV.) For heat pumps that use a time-adaptive defrost control system (see section 1.2 of this appendix, Definitions), the manufacturer must specify in the certification report the frosting interval to be used during frost accumulation tests and provide the procedure for manually initiating the defrost at the specified time.

2.2.2 Special Requirements for Units Having a Multiple-Speed Outdoor Fan

Configure the multiple-speed outdoor fan according to the installation manual included with the unit by the manufacturer, 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 Ducted Systems Using a Single Indoor Section Containing Multiple Indoor Blowers That Would Normally Operate Using Two or More Indoor Thermostats

Because these systems will have more than one indoor blower and possibly multiple outdoor fans and compressor systems, references in this test procedure to a singular indoor blower, outdoor fan, and/or compressor means all indoor blowers, all outdoor fans, and all compressor systems that are energized during the test.

a. Additional requirements for multi-split air conditioners and heat pumps. 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 must designate in the certification report the indoor coil(s) that are not providing heating or cooling during the test. For variable-speed systems, the manufacturer must designate in the certification report at least one indoor unit that is not providing heating or cooling for all tests conducted at minimum compressor speed. For all other part-load tests, the manufacturer must choose to turn off zero, one, two, or more indoor units. The chosen configuration must remain unchanged for all tests conducted at the same compressor speed/capacity. For any indoor coil that is not providing heating or cooling during a test, cease forced airflow through this indoor coil and block its outlet duct.

b. Additional requirements for ducted split systems with a single indoor unit containing multiple indoor blowers (or for single-package units with an indoor section containing multiple indoor blowers) where the indoor blowers are designed to cycle on and off independently of one another and are not controlled such that all indoor blowers are modulated to always operate at the same air volume rate or speed. For any test where the system is operated at its lowest capacity—*i.e.,* the lowest total air volume rate allowed when operating the single-speed compressor or when operating at low compressor capacity—turn off indoor blowers accounting for at least one-third of the full-load air volume rate unless prevented by the controls of the unit. In such cases, turn off as many indoor blowers as permitted by the unit's controls. Where more than one option exists for meeting this ‘‘off’’ requirement, the manufacturer must indicate in its certification report which indoor blower(s) are turned off. The chosen configuration shall remain unchanged for all tests conducted at the same lowest capacity configuration. For any indoor coil turned off during a test, cease forced airflow through any outlet duct connected to a switched-off indoor blower.

c. For test setups where the laboratory's physical limitations require use of more than the required line length of 25 feet as listed in section 2.2.a.(4) of this appendix, then the actual refrigerant line length used by the laboratory may exceed the required length and the refrigerant line length correction factors in Table 4 of AHRI 1230–2010 are applied to the cooling capacity measured for each cooling mode 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 so that the wet-bulb temperature is as listed in Tables 5 to 8. 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 5–8 list the applicable wet-bulb temperatures.

(2) Single-package 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 of this appendix. For dry coil tests on such units, it may be necessary to limit the moisture content of the air entering the outdoor coil of the unit to meet the requirements of section 3.4 of this appendix.

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 12 to 15. 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 (section 2.10.1 of this appendix) is used while testing a single-package 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

2.2.5.1 Instructions to Use for Charging

a. 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.

b. For systems consisting of an outdoor unit manufacturer's outdoor section and indoor section with differing charging procedures, adjust the refrigerant charge per the outdoor installation instructions.

c. For systems consisting of an outdoor unit manufacturer's outdoor unit and an independent coil manufacturer's indoor unit with differing charging procedures, adjust the refrigerant charge per the indoor unit's installation instructions. If instructions are provided only with the outdoor unit or are provided only with an independent coil manufacturer's indoor unit, then use the provided instructions.

2.2.5.2 Test(s) to Use for Charging

a. Use the tests or operating conditions specified in the manufacturer's installation instructions for charging. The manufacturer's installation instructions may specify use of tests other than the A or A_2 test for charging, but, unless the unit is a heating-only heat pump, determine the air volume rate by the A or A_2 test as specified in section 3.1 of this appendix.

b. If the manufacturer's installation instructions do not specify a test or operating conditions for charging or there are no manufacturer's instructions, use the following test(s):

(1) For air conditioners or cooling and heating heat pumps, use the A or A_2 test.

(2) For cooling and heating heat pumps that do not operate in the H1 or $H1_2$ test (*e.g.* due to shut down by the unit limiting devices) when tested using the charge determined at the A or A_2 test, and for heating-only heat pumps, use the H1 or $H1_2$ test.

2.2.5.3 Parameters to Set and Their Target Values

a. Consult the manufacturer's installation instructions regarding which parameters (*e.g.,* superheat) to set and their target values. If the instructions provide ranges of values, select target values equal to the midpoints of the provided ranges.

b. In the event of conflicting information between charging instructions (*i.e.,* multiple conditions given for charge adjustment where all conditions specified cannot be met), follow the following hierarchy.

(1) For fixed orifice systems:
(i) Superheat
(ii) High side pressure or corresponding saturation or dew-point temperature
(iii) Low side pressure or corresponding saturation or dew-point temperature
(iv) Low side temperature
(v) High side temperature
(vi) Charge weight

(2) For expansion valve systems:
(i) Subcooling
(ii) High side pressure or corresponding saturation or dew-point temperature
(iii) Low side pressure or corresponding saturation or dew-point temperature
(iv) Approach temperature (difference between temperature of liquid leaving condenser and condenser average inlet air temperature)
(v) Charge weight

c. If there are no installation instructions and/or they do not provide parameters and target values, set superheat to a target value of 12 °F for fixed orifice systems or set subcooling to a target value of 10 °F for expansion valve systems.

2.2.5.4 Charging Tolerances

a. If the manufacturer's installation instructions specify tolerances on target values for the charging parameters, set the values within these tolerances.

b. Otherwise, set parameter values within the following test condition tolerances for the different charging parameters:

11. Superheat: +/− 2.0 °F
12. Subcooling: +/− 2.0 °F
13. High side pressure or corresponding saturation or dew point temperature: +/− 4.0 psi or +/− 1.0 °F
14. Low side pressure or corresponding saturation or dew point temperature: +/− 2.0 psi or +/− 0.8 °F
15. High side temperature: +/− 2.0 °F
16. Low side temperature: +/− 2.0 °F
17. Approach temperature: +/− 1.0 °F
18. Charge weight: +/− 2.0 ounce

2.2.5.5 Special Charging Instructions

a. Cooling and Heating Heat Pumps

If, using the initial charge set in the A or A_2 test, the conditions are not within the range specified in manufacturer's installation instructions for the H1 or $H1_2$ test, make as small as possible an adjustment to obtain conditions for this test in the specified range. After this adjustment, recheck conditions in the A or A_2 test to confirm that they are still within the specified range for the A or A_2 test.

b. Single-Package Systems

i. Unless otherwise directed by the manufacturer's installation instructions, install one or more refrigerant line pressure gauges during the setup of the unit, located depending on the parameters used to verify or set charge, as described:

(1) Install a pressure gauge at the location of the service valve on the liquid line if charging is on the basis of subcooling, or high side pressure or corresponding saturation or dew point temperature;

(2) Install a pressure gauge at the location of the service valve on the suction line if charging is on the basis of superheat, or low side pressure or corresponding saturation or dew point temperature.

ii. Use methods for installing pressure gauge(s) at the required location(s) as indicated in manufacturer's instructions if specified.

2.2.5.6 Near-Azeotropic and Zeotropic Refrigerants

Perform charging of near-azeotropic and zeotropic refrigerants only with refrigerant in the liquid state.

2.2.5.7 Adjustment of Charge Between Tests

After charging the system as described in this test procedure, use the set refrigerant charge for all tests used to determine performance. Do not adjust the refrigerant charge at any point during testing. If measurements indicate that refrigerant charge has leaked during the test, repair the refrigerant leak, repeat any necessary set-up steps, and repeat all tests.

2.3 Indoor Air Volume Rates

If a unit's controls allow for overspeeding the indoor blower (usually on a temporary basis), take the necessary steps to prevent overspeeding during all tests.

2.3.1 Cooling Tests

a. Set indoor blower airflow-control settings (*e.g.,* fan motor pin settings, fan motor speed) according to the requirements that are specified in section 3.1.4 of this appendix.

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. Set indoor blower airflow-control settings (*e.g.,* fan motor pin settings, fan motor speed) according to the requirements that are specified in section 3.1.4 of this appendix.

b. Express the heating full-load 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 as described in section 2.4.1 of this appendix and, if installed, the inlet plenum described in section 2.4.2 of this appendix with thermal insulation having a nominal overall resistance (R-value) of at least 19 hr·ft² °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, or multiple indoor blowers within a single indoor section, attach a plenum to each indoor coil or indoor blower outlet. In order to reduce the number of required airflow measurement apparatuses (section 2.6 of this appendix), each such apparatus may serve multiple outlet plenums connected to a single common duct leading to the apparatus. More than one indoor test room may be used, which may use one or more common ducts leading to one or more airflow measurement apparatuses 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. The outlet air temperature grid(s) (section 2.5.4 of this appendix) and airflow measuring apparatus shall be located downstream of the inlet(s) to the common duct(s). For multiple-circuit (or multi-circuit) systems for which each indoor coil outlet is measured separately and its outlet plenum is not connected to a common duct connecting multiple outlet plenums, install the outlet air temperature grid and airflow measuring apparatus at each outlet plenum.

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 in Table 3. The limit depends only on the Cooling full-load air volume rate (see section 3.1.4.1.1 of this appendix) 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 9 of ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3) shows allowed options for the manifold configuration. The cross-sectional dimensions of plenum must be equal to the dimensions of the indoor unit outlet. See Figures 7a, 7b, and 7c of ANSI/ASHRAE 37–2009 for the minimum length of the (each) outlet plenum and the locations for adding the static pressure taps for ducted blower coil indoor units and single-package systems. See Figure 8 of ANSI/ASHRAE 37–2009 for coil-only indoor units.

TABLE 3—SIZE OF OUTLET PLENUM FOR SMALL-DUCT HIGH-VELOCITY INDOOR UNITS

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 ($4A/P$,) where A is the cross-sectional 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, a ducted blower coil indoor unit, or a single-package system. See Figures 7b and 7c of ANSI/ASHRAE 37–2009 for cross-sectional dimensions, the minimum length of the inlet plenum, and the locations of the static-pressure taps for ducted blower coil indoor units and single-package systems. See Figure 8 of ANSI/ASHRAE 37–2009 for coil-only indoor units. The inlet plenum duct size shall equal the size of the inlet opening of the air-handling (blower coil) unit or furnace. For a ducted blower coil indoor unit the set up may omit the inlet plenum if an inlet airflow prevention device is installed with a straight internally unobstructed duct on its outlet end with a minimum length equal to 1.5 times the square root of the cross-sectional area of the indoor unit inlet. See section 2.1.5.2 of this appendix for requirements for the locations of static pressure taps built into the inlet airflow prevention device. For all of these arrangements, make a manifold that connects the four static-pressure taps using one of the three configurations specified in section 2.4.1.d. of this appendix. Never use an inlet plenum when testing a non-ducted system.

2.5 Indoor Coil Air Property Measurements and Airflow Prevention Devices.

Follow instructions for indoor coil air property measurements as described in section 2.14 of this appendix, unless otherwise instructed in this section.

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 section 5.3 of ANSI/ASHRAE 41.1–2013 (incorporated by reference, see § 430.3) for guidance on constructing an air sampling device. No part of the air sampling device or the tubing transferring the sampled air to the sensor must be within two inches of the test chamber floor, and the transfer tubing must be insulated. The sampling device may also be used for measurement of dry bulb temperature by transferring the sampled air to a remotely located sensor(s). 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 sensor(s) may be used to determine the leaving air dry bulb temperature for all tests except:

(1) Cyclic tests; and

(2) Frost accumulation tests.

b. Install grids of temperature sensors to measure dry bulb temperatures of both the entering and leaving airstreams of the indoor unit. These grids of dry bulb temperature sensors may be used to measure average dry bulb temperature entering and leaving the indoor unit in all cases (as an alternative to the dry bulb sensor measuring the sampled air). The leaving airstream grid is required for measurement of average dry bulb temperature leaving the indoor unit for cyclic tests and frost accumulation tests. The grids are also required to measure the air temperature distribution of the entering and leaving airstreams as described in sections 3.1.8 of this appendix. Two such grids may be applied as a thermopile, to directly obtain the average temperature difference rather than directly measuring both entering and leaving average temperatures.

c. Use of airflow prevention devices. Use an inlet and outlet air damper box, or use an inlet upturned duct and an outlet air damper box when conducting one or both of the cyclic tests listed in sections 3.2 and 3.6 of this appendix on ducted systems. If not conducting any cyclic tests, an outlet air damper box is required when testing ducted and non-ducted heat pumps that cycle off the indoor blower during defrost cycles and there is no other means for preventing natural or forced convection through the indoor unit when the indoor blower is off. Never use an inlet damper box or an inlet upturned duct when testing non-ducted indoor units. An inlet upturned duct is a length of ductwork installed upstream from the inlet such that the indoor duct inlet opening, facing upwards, is sufficiently high to prevent natural convection transfer out of the duct. If an inlet upturned duct is used, install a dry bulb temperature sensor near the inlet opening of the indoor duct at a centerline location not higher than the lowest elevation of the duct edges at the inlet, and ensure that any pair of 5-minute averages of the dry bulb temperature at this location, measured at least every minute during the compressor OFF period of the cyclic test, do not differ by more than 1.0 °F.

2.5.1 Test Set-Up on the Inlet Side of the Indoor Coil: for Cases Where the Inlet Airflow Prevention Device is Installed

a. Install an airflow prevention device as specified in section 2.5.1.1 or 2.5.1.2 of this appendix, whichever applies.

b. For an inlet damper box, locate the grid of entering air dry-bulb temperature sensors, if used, and 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. For an inlet upturned duct, locate the grid of entering air dry-bulb temperature sensors, if used, and the air sampling device, or the sensor used to measure the water vapor content of the inlet air, at a location at least one foot downstream from the beginning of the insulated portion of the duct but before the static pressure measurement.

2.5.1.1 If the section 2.4.2 inlet plenum is installed, construct the airflow prevention device having a cross-sectional flow area equal to or greater than the flow area of the inlet plenum. Install the airflow prevention device upstream of the inlet plenum and construct ductwork connecting it to the inlet plenum. If needed, use an adaptor plate or a transition duct section to connect the airflow prevention device with the inlet plenum. Insulate the ductwork and inlet plenum with thermal insulation that has a nominal overall resistance (R-value) of at least 19 hr · ft^2 · °F/Btu.

2.5.1.2 If the section 2.4.2 inlet plenum is not installed, construct the airflow prevention device having a cross-sectional flow area equal to or greater than the flow area of the air inlet of the indoor unit. Install the airflow prevention device immediately upstream of the inlet of the indoor unit. If needed, use an adaptor plate or a short transition duct section to connect the airflow prevention device with the unit's air inlet. Add static pressure taps at the center of each face of a rectangular airflow prevention device, or at four evenly distributed locations along the circumference of an oval or round airflow prevention device. Locate the pressure taps at a distance from the indoor unit inlet equal to 0.5 times the square root of the cross sectional area of the indoor unit inlet. This location must be between the damper and the inlet of the indoor unit, if a damper is used. Make a manifold that connects the four static pressure taps using one of the configurations shown in Figure 9 of ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3). Insulate the ductwork with thermal insulation that has a nominal overall resistance (R-value) of at least 19 hr·ft^2·°F/Btu.

2.5.2 Test Set-Up on the Inlet Side of the Indoor Unit: for Cases Where No Airflow Prevention Device 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 of this appendix, preferably at the entrance plane of the inlet plenum. If the section 2.4.2 inlet plenum is not used (*i.e.* for non-ducted units) locate a grid approximately 6 inches upstream of the indoor unit inlet. In the case of a system having multiple non-ducted indoor units, do this for each indoor unit. 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

Fabricate pressure taps meeting all requirements described in section 6.5.2 of ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3) and illustrated in Figure 2A of AMCA 210–2007 (incorporated by reference, see § 430.3), however, if adhering strictly to the description in section 6.5.2 of ANSI/ASHRAE 37–2009, the minimum pressure tap length of 2.5 times the inner diameter of Figure 2A of AMCA 210–2007 is waived. 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 airflow prevention device. 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 of this appendix and the airflow measuring apparatus described below in section 2.6 of this appendix. 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·ft^2· °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). Turn off the sampler fan motor during the cyclic tests. Air leaving an indoor unit that is sampled by an air sampling device for remote water-vapor-content measurement must be returned to the interconnecting duct at a location:

(1) Downstream of the air sampling device;
(2) On the same side of the outlet air damper as the air sampling device; 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 of this appendix), 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 of this appendix. Mixing devices are described in sections 5.3.2 and 5.3.3 of ANSI/ASHRAE 41.1–2013 and section 5.2.2 of ASHRAE 41.2–1987 (RA 1992) (incorporated by reference, see § 430.3).

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 of this appendix. 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. If applicable, in lieu of installing a separate damper, use the outlet air damper box of sections 2.5 and 2.5.4.1 of this appendix 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, 5.3, 6, and 7 of ANSI/ASHRAE 41.1–2013 (incorporated by reference, see § 430.3).

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, 7.2, 7.3, and 7.4 of ASHRAE 41.6–2014 (incorporated by reference, see § 430.3). The temperature sensor (wick removed) must be accurate to within ±0.2 °F. If used, apply dew point hygrometers as specified in sections 4, 5, 6, 7.1, and 7.4 of ASHRAE 41.6–2014. 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 of this appendix), 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 airflow measuring apparatus as specified in section 6.2 and 6.3 of ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3). Place the static pressure taps and position the diffusion baffle (settling means) relative to the chamber inlet as indicated in Figure 12 of AMCA 210–07 and/or Figure 14 of ASHRAE 41.2–1987 (RA 1992) (incorporated by reference, see § 430.3). When measuring the static pressure difference across nozzles and/or velocity pressure at nozzle throats using electronic pressure transducers and a data acquisition system, if high frequency fluctuations cause measurement variations to exceed the test tolerance limits specified in section 9.2 and Table 2 of ANSI/ASHRAE 37–2009, dampen the measurement system such that the time constant associated with response to a step change in measurement (time for the response to change 63% of the way from the initial output to the final output) is no longer than five seconds.

b. Connect the airflow measuring apparatus to the interconnecting duct section described in section 2.5.4 of this appendix. See sections 6.1.1, 6.1.2, and 6.1.4, and Figures 1, 2, and 4 of ANSI/ASHRAE 37–2009; and Figures D1, D2, and D4 of AHRI 210/240–2008 (incorporated by reference, see § 430.3) with Addendum 1 and 2 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 ANSI/ASHRAE 37–2009 when testing triple-split units.)

2.7 Electrical Voltage Supply

Perform all tests at the voltage specified in section 6.1.3.2 of AHRI 210/240–2008 (incorporated by reference, see § 430.3) for ''Standard Rating Tests.'' If either the indoor or the outdoor unit has a 208V or 200V nameplate voltage and the other unit has a 230V nameplate rating, select the voltage supply on the outdoor unit for testing. Otherwise, supply each unit with its own nameplate voltage. 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 within 15 seconds prior to beginning an ON cycle. For ducted blower coil systems, the ON cycle lasts from compressor ON to indoor blower OFF. For ducted coil-only systems, the ON cycle lasts from compressor ON to compressor OFF. For non-ducted units, the ON cycle lasts from indoor blower ON to indoor blower 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 blower motor to within ±1.0 percent. If required according to sections 3.3, 3.4, 3.7, 3.9.1 of this appendix, and/or 3.10 of this appendix, this same instrumentation requirement (to determine the average electrical power consumption of the indoor blower motor to within ±1.0 percent) applies when testing air conditioners and heat pumps having a variable-speed constant-air-volume-rate indoor blower or a variable-speed, variable-air-volume-rate indoor blower.

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 of this appendix. In addition, for all steady-state tests, conduct a second, independent measurement of capacity as described in section 3.1.1 of this appendix. For split systems, use one of the following secondary measurement methods: outdoor air enthalpy method, compressor calibration method, or refrigerant enthalpy method. For single-package 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) On the outlet side, an outlet plenum containing static pressure taps (sections 2.4, 2.4.1, and 2.5.3 of this appendix),

(2) An airflow measuring apparatus (section 2.6 of this appendix),

(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 of this appendix), and

(4) On the inlet side, a sampling device and temperature grid (section 2.11.b of this appendix).

c. During the free outdoor air tests described in sections 3.11.1 and 3.11.1.1 of this appendix, 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, install pressure gages to the access valves or to ports created from tapping into the suction and discharge lines according to sections 7.4.2 and 8.2.5 of ANSI/ASHRAE 37–2009. 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 and measure refrigerant properties according to section 7.4.2 and 8.2.5 of ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3). If removing the refrigerant before applying refrigerant lines and subsequently recharging, use the steps in 7.4.2 of ANSI/ASHRAE 37–2009 in addition to the methods of section 2.2.5 of this appendix to confirm the refrigerant charge. Use refrigerant temperature and pressure measuring instruments that meet the specifications given in sections 5.1.1 and 5.2 of ANSI/ASHRAE 37–2009.

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 ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3) 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 ANSI/ASHRAE 37–2009. Refrigerant flow measurement device(s), if used, must be either elevated at least two feet from the test chamber floor or placed upon insulating material having a total thermal resistance of at least R–12 and extending at least one foot laterally beyond each side of the device(s)' exposed surfaces.

2.11 Measurement of Test Room Ambient Conditions

Follow instructions for setting up air sampling device and aspirating psychrometer as described in section 2.14 of this appendix, unless otherwise instructed in this section.

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 ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3)), add instrumentation to permit measurement of the indoor test room dry-bulb temperature.

b. On the outdoor side, use one of the following two approaches, except that approach (1) is required for all evaporatively cooled units and units that transfer condensate to the outdoor unit for evaporation using condenser heat.

(1) Use sampling tree air collection on all air-inlet surfaces of the outdoor unit.

(2) Use sampling tree air collection on one or more faces of the outdoor unit and demonstrate air temperature uniformity as follows. Install a grid of evenly distributed thermocouples on each air-permitting face on the inlet of the outdoor unit. Install the thermocouples on the air sampling device, locate them individually or attach them to a wire structure. If not installed on the air sampling device, install the thermocouple grid 6 to 24 inches from the unit. Evenly space the thermocouples across the coil inlet surface and install them to avoid sampling of discharge air or blockage of air recirculation. The grid of thermocouples must provide at least 16 measuring points per face or one measurement per square foot of inlet face

area, whichever is less. Construct this grid and use as per section 5.3 of ANSI/ASHRAE 41.1–2013 (incorporated by reference, see § 430.3). The maximum difference between the average temperatures measured during the test period of any two pairs of these individual thermocouples located at any of the faces of the inlet of the outdoor unit, must not exceed 2.0 °F, otherwise use approach (1).

Locate the air sampling devices at the geometric center of each side; the branches may be oriented either parallel or perpendicular to the longer edges of the air inlet area. Size the air sampling devices in the outdoor air inlet location such that they cover at least 75% of the face area of the side of the coil that they are measuring.

Review air distribution at the test facility point of supply to the unit and remediate as necessary prior to the beginning of testing. Mixing fans can be used to ensure adequate air distribution in the test room. If used, orient mixing fans such that they are pointed away from the air intake so that the mixing fan exhaust does not affect the outdoor coil air volume rate. Particular attention should be given to prevent the mixing fans from affecting (enhancing or limiting) recirculation of condenser fan exhaust air back through the unit. Any fan used to enhance test room air mixing shall not cause air velocities in the vicinity of the test unit to exceed 500 feet per minute.

The air sampling device may be larger than the face area of the side being measured. Take care, however, to prevent discharge air from being sampled. If an air sampling device dimension extends beyond the inlet area of the unit, block holes in the air sampling device to prevent sampling of discharge air. Holes can be blocked to reduce the region of coverage of the intake holes both in the direction of the trunk axis or perpendicular to the trunk axis. For intake hole region reduction in the direction of the trunk axis, block holes of one or more adjacent pairs of branches (the branches of a pair connect opposite each other at the same trunk location) at either the outlet end or the closed end of the trunk. For intake hole region reduction perpendicular to the trunk axis, block off the same number of holes on each branch on both sides of the trunk.

Connect a maximum of four (4) air sampling devices to each aspirating psychrometer. In order to proportionately divide the flow stream for multiple air sampling devices for a given aspirating psychrometer, the tubing or conduit conveying sampled air to the psychrometer must be of equivalent lengths for each air sampling device. Preferentially, the air sampling device should be hard connected to the aspirating psychrometer, but if space constraints do not allow this, the assembly shall have a means of allowing a flexible tube to connect the air sampling device to the aspirating psychrometer. Insulate and route the tubing or conduit to prevent heat transfer to the air stream. Insulate any surface of the air conveying tubing in contact with surrounding air at a different temperature than the sampled air with thermal insulation with a nominal thermal resistance (R-value) of at least 19 hr • ft^2 • °F/Btu. Alternatively the conduit may have lower thermal resistance if additional sensor(s) are used to measure dry bulb temperature at the outlet of each air sampling device. No part of the air sampling device or the tubing conducting the sampled air to the sensors may be within two inches of the test chamber floor.

Take pairs of measurements (*e.g.* dry bulb temperature and wet bulb temperature) used to determine water vapor content of sampled air in the same location.

2.12 Measurement of Indoor Blower 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 ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3).

2.14 Air Sampling Device and Aspirating Psychrometer Requirements

Make air temperature measurements in accordance with ANSI/ASHRAE 41.1–2013 (incorporated by reference, see § 430.3), unless otherwise instructed in this section.

2.14.1 Air Sampling Device Requirements

The air sampling device is intended to draw in a sample of the air at the critical locations of a unit under test. Construct the device from stainless steel, plastic or other suitable, durable materials. It shall have a main flow trunk tube with a series of branch tubes connected to the trunk tube. Holes must be on the side of the sampler facing the upstream direction of the air source. Use other sizes and rectangular shapes, and scale them accordingly with the following guidelines:

1. Minimum hole density of 6 holes per square foot of area to be sampled.
2. Sampler branch tube pitch (spacing) of 6 ± 3 in.
3. Manifold trunk to branch diameter ratio having a minimum of 3:1 ratio.
4. Distribute hole pitch (spacing) equally over the branch (½ pitch from the closed end to the nearest hole).
5. Maximum individual hole to branch diameter ratio of 1:2 (1:3 preferred).

The minimum average velocity through the air sampling device holes must be 2.5 ft/s as determined by evaluating the sum of the open area of the holes as compared to the flow area in the aspirating psychrometer.

2.14.2 Aspirating Psychrometer

The psychrometer consists of a flow section and a fan to draw air through the flow section and measures an average value of the sampled air stream. At a minimum, the flow section shall have a means for measuring the dry bulb temperature (typically, a resistance temperature device (RTD) and a means for measuring the humidity (RTD with wetted sock, chilled mirror hygrometer, or relative humidity sensor). The aspirating psychrometer shall include a fan that either can be adjusted manually or automatically to maintain required velocity across the sensors.

Construct the psychrometer using suitable material which may be plastic (such as polycarbonate), aluminum or other metallic materials. Construct all psychrometers for a given system being tested, using the same material. Design the psychrometers such that radiant heat from the motor (for driving the fan that draws sampled air through the psychrometer) does not affect sensor measurements. For aspirating psychrometers, velocity across the wet bulb sensor must be 1000 ± 200 ft/min. For all other psychrometers, velocity must be as specified by the sensor manufacturer.

3 Testing Procedures

3.1 General Requirements

If, during the testing process, an equipment set-up adjustment is made that would have altered the performance of the unit during any 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.

Use the testing procedures in this section to collect the data used for calculating

(1) Performance metrics for central air conditioners and heat pumps during the cooling season;

(2) Performance metrics for heat pumps during the heating season; and

(3) Power consumption metric(s) for central air conditioners and heat pumps during the off mode season(s).

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 full-capacity cooling-mode test and (for a heat pump) the full-capacity heating-mode test, use one of the acceptable secondary methods specified in section 2.10 of this appendix to determine indoor space conditioning capacity. Calculate this secondary check of capacity according to section 3.11 of this appendix. 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 ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3) (and, if testing a coil-only system, compare capacities before making 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 of this appendix.

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 blower 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 AHRI 210/240–2008 (incorporated by reference, see § 430.3) when obtaining the airflow through the outdoor coil.

3.1.3.1 Double-Ducted

For products intended to be installed with the outdoor airflow ducted, install the unit with outdoor coil ductwork installed per manufacturer installation instructions. The unit must operate between 0.10 and 0.15 in H_2O external static pressure. Make external static pressure measurements in accordance with ANSI/ASHRAE 37–2009 section 6.4 and 6.5.

3.1.4 Airflow Through the Indoor Coil

Determine airflow setting(s) before testing begins. Unless otherwise specified within this or its subsections, make no changes to the airflow setting(s) after initiation of testing.

3.1.4.1 Cooling Full-Load Air Volume Rate

3.1.4.1.1 Cooling Full-Load Air Volume Rate for Ducted Units

Identify the certified Cooling full-load air volume rate and certified instructions for setting fan speed or controls. If there is no certified Cooling full-load air volume rate, use a value equal to the certified cooling capacity of the unit times 400 scfm per 12,000 Btu/h. If there are no instructions for setting fan speed or controls, use the as-shipped settings. Use the following procedure to confirm and, if necessary, adjust the Cooling full-load air volume rate and the fan speed or control settings to meet each test procedure requirement:

a. For all ducted blower coil systems, except those having a constant-air-volume-rate indoor blower:

Step (1) Operate the unit under conditions specified for the A (for single-stage units) or A_2 test using the certified fan speed or controls settings, and adjust the exhaust fan of the airflow measuring apparatus to achieve the certified Cooling full-load air volume rate;

Step (2) Measure the external static pressure;

Step (3) If this external static pressure is equal to or greater than the applicable minimum external static pressure cited in Table 4, the pressure requirement is satisfied; proceed to step 7 of this section. If this external static pressure is not equal to or greater than the applicable minimum external static pressure cited in Table 4, proceed to step 4 of this section;

Step (4) Increase the external static pressure by adjusting the exhaust fan of the airflow measuring apparatus until either

(i) The applicable Table 4 minimum is equaled or

(ii) The measured air volume rate equals 90 percent or less of the Cooling full-load air volume rate, whichever occurs first;

Step (5) If the conditions of step 4 (i) of this section occur first, the pressure requirement is satisfied; proceed to step 7 of this section. If the conditions of step 4 (ii) of this section occur first, proceed to step 6 of this section;

Step (6) Make an incremental change to the setup of the indoor blower (*e.g.,* next highest fan motor pin setting, next highest fan motor speed) and repeat the evaluation process beginning above, at step 1 of this section. If the indoor blower setup cannot be further changed, increase the external static pressure by adjusting the exhaust fan of the airflow measuring apparatus until the applicable Table 4 minimum is equaled; proceed to step 7 of this section;

Step (7) The airflow constraints have been satisfied. Use the measured air volume rate as the Cooling full-load air volume rate. Use the final fan speed or control settings for all tests that use the Cooling full-load air volume rate.

b. For ducted blower coil systems with a constant-air-volume-rate indoor blower. 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 4 value that does not cause automatic shutdown of the indoor blower or air volume rate variation Q_{Var}, defined as follows, greater than 10 percent.

$$Q_{Var} = \left[\frac{Q_{max} - Q_{min}}{\left(\frac{Q_{max} + Q_{min}}{2}\right)}\right] * 100$$

Where:

Q_{max} = maximum measured airflow value

Q_{min} = minimum measured airflow value

Q_{Var} = airflow variance, percent

Additional test steps as described in section 3.3.e of this appendix are required if the measured external static pressure exceeds the target value by more than 0.03 inches of water.

c. For coil-only indoor units. 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-load air volume rate.

TABLE 4—MINIMUM EXTERNAL STATIC PRESSURE FOR DUCTED BLOWER COIL SYSTEMS

Product variety	Minimum external static pressure (in. wc.)
Conventional (*i.e.,* all central air conditioners and heat pumps not otherwise listed in this table)	0.50
Ceiling-mount and Wall-mount	0.30
Mobile Home	0.30
Low Static	0.10
Mid Static	0.30
Small Duct, High Velocity	1.15
Space-constrained	0.30

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

[2] See section 1.2, Definitions, to determine for which Table 4 product variety and associated minimum external static pressure requirement equipment qualifies.

[3] 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.

d. For ducted systems having multiple indoor blowers within a single indoor section, obtain the full-load air volume rate with all indoor blowers operating unless prevented by the controls of the unit. In such cases, turn on the maximum number of indoor blowers permitted by the unit's controls. Where more than one option exists for meeting this ''on'' indoor blower requirement, which indoor blower(s) are turned on must match that specified in the certification report. Conduct section 3.1.4.1.1 setup steps for each indoor blower separately. If two or more indoor blowers are connected to a common duct as per section 2.4.1 of this appendix, temporarily divert their air volume to the test room when confirming or adjusting the setup configuration of individual indoor blowers. The allocation of the system's full-load air volume rate assigned to each ''on'' indoor blower must match that specified by the manufacturer in the certification report.

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

Identify the certified cooling minimum air volume rate and certified instructions for setting fan speed or controls. If there is no certified cooling minimum air volume rate, use the final indoor blower control settings as determined when setting the cooling full-load air volume rate, and readjust the exhaust fan of the airflow measuring apparatus if necessary to reset to the cooling full load air volume obtained in section 3.1.4.1 of this appendix. Otherwise, calculate the target external static pressure and follow instructions a, b, c, d, or e of this section. The target external static pressure, ΔP_{st_i}, for any test ''i'' with a specified air volume rate not equal to the Cooling full-load air volume rate is determined as follows:

$$\Delta P_{st\_i} = \Delta P_{st\_full} \left[\frac{Q_i}{Q_{full}}\right]^2$$

Where:

ΔP_{st_i} = target minimum external static pressure for test i;

ΔP_{st_full} = minimum external static pressure for test A or A_2 (Table 4);

Q_i = air volume rate for test i; and

Q_{full} = Cooling full-load air volume rate as measured after setting and/or adjustment as described in section 3.1.4.1.1 of this appendix.

a. For a ducted blower coil system without a constant-air-volume indoor blower, adjust for external static pressure as follows:

Step (1) Operate the unit under conditions specified for the B_1 test using the certified fan speed or controls settings, and adjust the exhaust fan of the airflow measuring apparatus to achieve the certified cooling minimum air volume rate;

Step (2) Measure the external static pressure;

Step (3) If this pressure is equal to or greater than the minimum external static pressure computed above, the pressure requirement is satisfied; proceed to step 7 of this section. If this pressure is not equal to or greater than the minimum external static pressure computed above, proceed to step 4 of this section;

Step (4) Increase the external static pressure by adjusting the exhaust fan of the airflow measuring apparatus until either

(i) The pressure is equal to the minimum external static pressure computed above or

(ii) The measured air volume rate equals 90 percent or less of the cooling minimum air volume rate, whichever occurs first;

Step (5) If the conditions of step 4 (i) of this section occur first, the pressure requirement is satisfied; proceed to step 7 of this section. If the conditions of step 4 (ii) of this section occur first, proceed to step 6 of this section;

Step (6) Make an incremental change to the setup of the indoor blower (*e.g.,* next highest fan motor pin setting, next highest fan motor speed) and repeat the evaluation process beginning above, at step 1 of this section. If the indoor blower setup cannot be further changed, increase the external static pressure by adjusting the exhaust fan of the airflow measuring apparatus until it equals the minimum external static pressure computed above; proceed to step 7 of this section;

Step (7) The airflow constraints have been satisfied. Use the measured air volume rate as the cooling minimum air volume rate. Use the final fan speed or control settings for all tests that use the cooling minimum air volume rate.

b. For ducted units with constant-air-volume indoor blowers, 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 an automatic shutdown of the indoor blower or air volume rate variation Q_{Var}, defined in section 3.1.4.1.1.b of this appendix, greater than 10 percent, while being as close to, but not less than the target minimum external static pressure. Additional test steps as described in section 3.3.e of this appendix are required if the measured external static pressure exceeds the target value by more than 0.03 inches of water.

c. For ducted two-capacity coil-only systems, the cooling minimum air volume rate is the higher of—

(1) The rate specified by the installation instructions included with the unit by the manufacturer; or

(2) 75 percent of the cooling full-load air volume rate. During the laboratory tests on a coil-only (fanless) system, 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 blower 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 blower, use the lowest fan setting allowed for cooling.

e. For ducted systems having multiple indoor blowers within a single indoor section, operate the indoor blowers such that the lowest air volume rate allowed by the unit's controls is obtained when operating the lone single-speed compressor or when operating at low compressor capacity while meeting the requirements of section 2.2.3.2 of this appendix for the minimum number of blowers that must be turned off. Using the target external static pressure and the certified air volume rates, follow the procedures described in section 3.1.4.2.a of this appendix if the indoor blowers are not constant-air-volume indoor blowers or as described in section 3.1.4.2.b of this appendix if the indoor blowers are not constant-air-volume indoor blowers. The sum of the individual ''on'' indoor blowers' air volume rates is the cooling minimum air volume rate for the system.

3.1.4.3 Cooling Intermediate Air Volume Rate

Identify the certified cooling intermediate air volume rate and certified instructions for setting fan speed or controls. If there is no certified cooling intermediate air volume rate, use the final indoor blower control settings as determined when setting the cooling full load air volume rate, and readjust the exhaust fan of the airflow measuring apparatus if necessary to reset to the cooling full load air volume obtained in section 3.1.4.1 of this appendix. Otherwise, calculate target minimum external static pressure as described in section 3.1.4.2 of this appendix, and set the air volume rate as follows.

a. For a ducted blower coil system without a constant-air-volume indoor blower, adjust for external static pressure as described in section 3.1.4.2.a of this appendix for cooling minimum air volume rate.

b. For a ducted blower coil system with a constant-air-volume indoor blower, conduct the E_V Test at an external static pressure that does not cause an automatic shutdown of the indoor blower or air volume rate variation Q_{Var}, defined in section 3.1.4.1.1.b of this appendix, greater than 10 percent, while being as close to, but not less than the target minimum external static pressure. Additional test steps as described in section 3.3.e of this appendix are required if the measured external static pressure exceeds the target value by more than 0.03 inches of water.

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-load air volume rate for:

(1) Ducted blower coil system heat pumps that do not have a constant-air-volume indoor blower, and that operate at the same airflow-control setting during both the A (or A_2) and the H1 (or $H1_2$) Tests;

(2) Ducted blower coil system heat pumps with constant-air-flow indoor blowers that provide the same airflow for the A (or A_2) and the H1 (or $H1_2$) Tests; and

(3) Ducted heat pumps that are tested with a coil-only indoor unit (except two-capacity northern heat pumps that are tested only at low capacity cooling—see section 3.1.4.4.2 of this appendix).

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. Use the final indoor blower control settings as determined when setting the Cooling full-load air volume rate, and readjust the exhaust fan of the airflow measuring apparatus if necessary to reset to the cooling full-load air volume obtained in section 3.1.4.1 of this appendix. For heat pumps that meet the above criterion ''2,'' test at an external static pressure that does not cause an automatic shutdown of the indoor blower or air volume rate variation Q_{Var}, defined in section 3.1.4.1.1.b of this appendix, greater than 10 percent, while being as close to, but not less than, the same Table 4 minimum external static pressure as was specified for the A (or A_2) cooling mode test. Additional test steps as described in section 3.9.1.c of this appendix are required if the measured external static pressure exceeds the target value by more than 0.03 inches of water.

3.1.4.4.2 Ducted Heat Pumps Where the Heating and Cooling Full-Load Air Volume Rates Are Different Due to Changes in Indoor Blower Operation, i.e. Speed Adjustment by the System Controls

Identify the certified heating full-load air volume rate and certified instructions for setting fan speed or controls. If there is no certified heating full-load air volume rate, use the final indoor blower control settings as determined when setting the cooling full-load air volume rate, and readjust the exhaust fan of the airflow measuring apparatus if necessary to reset to the cooling full-load air volume obtained in section 3.1.4.1 of this appendix. Otherwise, calculate the target minimum external static pressure as described in section 3.1.4.2 of this appendix and set the air volume rate as follows.

a. For ducted blower coil system heat pumps that do not have a constant-air-volume indoor blower, adjust for external static pressure as described in section 3.1.4.2.a of this appendix for cooling minimum air volume rate.

b. For ducted heat pumps tested with constant-air-volume indoor blowers installed, conduct all tests that specify the heating full-load air volume rate at an external static pressure that does not cause an automatic shutdown of the indoor blower or air volume rate variation Q_{Var}, defined in section 3.1.4.1.1.b of this appendix, greater than 10 percent, while being as close to, but not less than the target minimum external static pressure. Additional test steps as described in section 3.9.1.c of this appendix are required if the measured external static pressure exceeds the target value by more than 0.03 inches of water.

c. When testing ducted, two-capacity blower coil system northern heat pumps (see section 1.2 of this appendix, Definitions), use the appropriate approach of the above two cases. For coil-only system northern heat pumps, the heating full-load air volume rate is the lesser of the rate specified by the manufacturer in the installation instructions included with the unit 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.

d. For ducted systems having multiple indoor blowers within a single indoor section, obtain the heating full-load air volume rate using the same ''on'' indoor blowers as used for the Cooling full-load air volume rate. Using the target external static pressure and the certified air volume rates, follow the procedures as described in section 3.1.4.4.2.a of this appendix if the indoor blowers are not constant-air-volume indoor blowers or as described in section 3.1.4.4.2.b of this appendix if the indoor blowers are constant-air-volume indoor blowers. The sum of the individual ''on'' indoor blowers' air volume rates is the heating full-load air volume rate for the system.

3.1.4.4.3 Ducted Heating-Only Heat Pumps

Identify the certified heating full-load air volume rate and certified instructions for setting fan speed or controls. If there is no certified heating full-load air volume rate, use a value equal to the certified heating capacity of the unit times 400 scfm per 12,000 Btu/h. If there are no instructions for setting fan speed or controls, use the as-shipped settings.

a. For all ducted heating-only blower coil system heat pumps, except those having a constant-air-volume-rate indoor blower. Conduct the following steps only during the first test, the H1 or $H1_2$ test:

Step (1) Adjust the exhaust fan of the airflow measuring apparatus to achieve the certified heating full-load air volume rate.

Step (2) Measure the external static pressure.

Step (3) If this pressure is equal to or greater than the Table 4 minimum external static pressure that applies given the heating-only heat pump's rated heating capacity, the pressure requirement is satisfied; proceed to step 7 of this section. If this pressure is not equal to or greater than the applicable Table 4 minimum external static pressure, proceed to step 4 of this section;

Step (4) Increase the external static pressure by adjusting the exhaust fan of the airflow measuring apparatus until either—

(i) The pressure is equal to the applicable Table 4 minimum external static pressure; or

(ii) The measured air volume rate equals 90 percent or less of the heating full-load air volume rate, whichever occurs first;

Step (5) If the conditions of step 4 (i) of this section occur first, the pressure requirement is satisfied; proceed to step 7 of this section. If the conditions of step 4 (ii) of this section occur first, proceed to step 6 of this section;

Step (6) Make an incremental change to the setup of the indoor blower (*e.g.,* next highest fan motor pin setting, next highest fan motor speed) and repeat the evaluation process beginning above, at step 1 of this section. If the indoor blower setup cannot be further changed, increase the external static pressure by adjusting the exhaust fan of the airflow measuring apparatus until it equals the applicable Table 4 minimum external static pressure; proceed to step 7 of this section;

Step (7) The airflow constraints have been satisfied. Use the measured air volume rate as the heating full-load air volume rate. Use the final fan speed or control settings for all tests that use the heating full-load air volume rate.

b. For ducted heating-only blower coil system heat pumps having a constant-air-volume-rate indoor blower. For all tests that specify the heating full-load air volume rate, obtain an external static pressure that does not cause an automatic shutdown of the indoor blower or air volume rate variation Q_{Var}, defined in section 3.1.4.1.1.b of this section, greater than 10 percent, while being as close to, but not less than, the applicable Table 4 minimum. Additional test steps as described in section 3.9.1.c of this appendix are required if the measured external static pressure exceeds the target value by more than 0.03 inches of water.

c. For ducted heating-only coil-only system heat pumps in 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-load 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.

3.1.4.5 Heating Minimum Air Volume Rate

3.1.4.5.1 Ducted Heat Pumps Where the Heating and Cooling Minimum Air Volume Rates are the Same

a. Use the cooling minimum air volume rate as the heating minimum air volume rate for:

(1) Ducted blower coil system heat pumps that do not have a constant-air-volume indoor blower, and that operates at the same airflow-control setting during both the A_1 and the $H1_1$ tests;

(2) Ducted blower coil system heat pumps with constant-air-flow indoor blowers installed that provide the same airflow for the A_1 and the $H1_1$ Tests; and

(3) Ducted coil-only system heat pumps.

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. Use the final indoor blower control settings as determined when setting the cooling minimum air volume rate, and readjust the exhaust fan of the airflow measuring apparatus if necessary to reset to the cooling minimum air volume rate obtained in section 3.1.4.2 of this appendix. For heat pumps that meet the above criterion ''2,'' test at an external static pressure that does not cause an automatic shutdown of the indoor blower or air volume rate variation Q_{Var}, defined in section 3.1.4.1.1.b, greater than 10 percent, while being as close to, but not less than, the same target minimum external static pressure as was specified for the A_1 cooling mode test. Additional test steps as described in section 3.9.1.c of this appendix are required if the measured external static pressure exceeds the target value by more than 0.03 inches of water.

3.1.4.5.2 Ducted Heat Pumps Where the Heating and Cooling Minimum Air Volume Rates Are Different Due to Indoor Blower Operation, *i.e.* Speed Adjustment by the System Controls

Identify the certified heating minimum air volume rate and certified instructions for setting fan speed or controls. If there is no certified heating minimum air volume rate, use the final indoor blower control settings as determined when setting the cooling minimum air volume rate, and readjust the exhaust fan of the airflow measuring apparatus if necessary to reset to the cooling minimum air volume obtained in section 3.1.4.2 of this appendix. Otherwise, calculate the target minimum external static pressure as described in section 3.1.4.2 of this appendix.

a. For ducted blower coil system heat pumps that do not have a constant-air-volume indoor blower, adjust for external static pressure as described in section 3.1.4.2.a of this appendix for cooling minimum air volume rate.

b. For ducted heat pumps tested with constant-air-volume indoor blowers installed, 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 an automatic shutdown of the indoor blower while being as close to, but not less than the air volume rate variation Q_{Var}, defined in section 3.1.4.1.1.b of this appendix, greater than 10 percent, while being as close to, but not less than the target minimum external static pressure. Additional test steps as described in section 3.9.1.c of this appendix are required if the measured external static pressure exceeds the target value by more than 0.03 inches of water.

c. For ducted two-capacity blower coil system northern heat pumps, use the appropriate approach of the above two cases.

d. For ducted two-capacity coil-only system heat pumps, use the cooling minimum air volume rate as the heating minimum air volume rate. For ducted two-capacity coil-only system northern heat pumps, use the cooling full-load air volume rate as the heating minimum air volume rate. For ducted two-capacity heating-only coil-only system heat pumps, the heating minimum air volume rate is the higher of the rate specified by the manufacturer in the test setup instructions included with the unit or 75 percent of the heating full-load air volume rate. During the laboratory tests on a coil-only system, 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 blower 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 blower, use the lowest fan setting allowed for heating.

f. For ducted systems with multiple indoor blowers within a single indoor section, obtain the heating minimum air volume rate using the same ''on'' indoor blowers as used for the cooling minimum air volume rate. Using the target external static pressure and the certified air volume rates, follow the procedures as described in section 3.1.4.5.2.a of this appendix if the indoor blowers are not constant-air-volume indoor blowers or as described in section 3.1.4.5.2.b of this appendix if the indoor blowers are constant-air-volume indoor blowers. The sum of the individual ''on'' indoor blowers' air volume rates is the heating full-load air volume rate for the system.

3.1.4.6 Heating Intermediate Air Volume Rate

Identify the certified heating intermediate air volume rate and certified instructions for setting fan speed or controls. If there is no certified heating intermediate air volume rate, use the final indoor blower control settings as determined when setting the heating full-load air volume rate, and readjust the exhaust fan of the airflow measuring apparatus if necessary to reset to the cooling full-load air volume obtained in section 3.1.4.2 of this appendix. Calculate the target minimum external static pressure as described in section 3.1.4.2 of this appendix.

a. For ducted blower coil system heat pumps that do not have a constant-air-volume indoor blower, adjust for external static pressure as described in section 3.1.4.2.a of this appendix for cooling minimum air volume rate.

b. For ducted heat pumps tested with constant-air-volume indoor blowers installed, conduct the $H2_V$ Test at an external static pressure that does not cause an automatic shutdown of the indoor blower or air volume rate variation Q_{Var}, defined in section 3.1.4.1.1.b of this appendix, greater than 10 percent, while being as close to, but not less than the target minimum external static pressure. Additional test steps as described in section 3.9.1.c of this appendix are required if the measured external static pressure exceeds the target value by more than 0.03 inches of water.

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

The manufacturer must specify the heating nominal air volume rate and the instructions for setting fan speed or controls. Calculate target minimum external static pressure as described in section 3.1.4.2 of this appendix. Make adjustments as described in section 3.14.6 of this appendix for heating intermediate air volume rate so that the target minimum external static pressure is met or exceeded.

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 ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3)), 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. Dew point must be within 2 °F of the required inlet conditions.

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 ANSI/ASHRAE 37–2009. When using the outdoor air enthalpy method, follow sections 7.7.2.1 and 7.7.2.2 of ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3) to calculate the air volume rate through the outdoor coil. To express air volume rates in terms of standard air, use:

$$\text{Equation 3-1} \qquad \overline{\dot{V}_s} = \frac{\overline{\dot{V}_{mx}}}{0.075\frac{lbm_{da}}{ft^3}*v_n'*[1+W_n]} = \frac{\overline{\dot{V}_{mx}}}{0.075\frac{lbm_{da}}{ft^3}*v_n}$$

Where:

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

$\overline{V}_{mx}$ = 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 ANSI/ASHRAE 37–2009, the second IP equation for Q_{mi} should read,

$$Q_{mi} = 1097CA_n\sqrt{P_V v_n'})$$

3.1.7 Test Sequence

Before making test measurements used to calculate performance, operate the equipment for the ''break-in'' period specified in the certification report, which may not exceed 20 hours. Each compressor of the unit must undergo this ''break-in'' period. 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-load air 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 a 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, and the first test using the heating minimum air volume rate must 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 of this appendix. 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 of this appendix to minimize the temperature spread.

3.1.9 Requirement for the Air Temperature Distribution Entering the Outdoor Coil

Monitor the Temperatures of the Air Entering the Outdoor Coil Using Air Sampling Devices and/or Temperature Sensor Grids, Maintaining the Required Tolerances, if Applicable, as Described in section 2.11 of this appendix

3.1.10 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 of this appendix, 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 of this appendix, 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-load air 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 System Having a Single-Speed Compressor and Fixed Cooling Air Volume Rate

This set of tests is for single-speed-compressor units that do not have a cooling minimum air volume rate or a cooling intermediate air volume rate that is different than the cooling full load air volume rate. 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 (for outdoor units with no match) or 0.2 (for all other systems). Table 5 specifies test conditions for these four tests.

TABLE 5—COOLING MODE TEST CONDITIONS FOR UNITS HAVING A SINGLE-SPEED COMPRESSOR AND A FIXED COOLING AIR VOLUME RATE

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-load [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 of this appendix.
[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 Where the Indoor Section Uses a Single Variable-Speed Variable-Air-Volume Rate Indoor Blower or Multiple Indoor Blowers

3.2.2.1 Indoor Blower Capacity Modulation That Correlates With the Outdoor Dry Bulb Temperature or Systems With a Single Indoor Coil but Multiple Indoor Blowers

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 0.2.

3.2.2.2 Indoor Blower 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 of this appendix and Table 5. Use a cooling full-load air volume rate that represents a normal 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 6—COOLING MODE TEST CONDITIONS FOR UNITS WITH A SINGLE-SPEED COMPRESSOR THAT MEET THE SECTION 3.2.2.1 INDOOR UNIT REQUIREMENTS

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 of this appendix.
[3] Defined in section 3.1.4.2 of this appendix.

[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 Section 1.2 of This Appendix, Definitions)

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 0.2. Table 7 specifies test conditions for these six tests.

b. For units having a variable-speed indoor blower 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 installation. Additionally, if conducting the 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 section 1.2 of this appendix, Definitions) in the same way as a single speed heat pump with the unit operating exclusively at low compressor capacity (see section 3.2.1 of this appendix and Table 5).

d. If a two-capacity air conditioner or heat pump locks out low-capacity operation at higher outdoor temperatures, then use the two 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 7—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]).
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 of this appendix.
[3] Defined in section 3.1.4.2 of this appendix.
[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 8 specifies test conditions for these seven tests. The compressor shall operate at the same cooling full speed, measured by RPM or power input frequency (Hz), for both the A_2 and B_2 tests. The compressor shall operate at the same cooling minimum speed, measured by RPM or power input frequency (Hz), for the B_1, F_1, G_1, and I_1 tests. Determine the cooling intermediate compressor speed cited in Table 8 using:

$$\text{Cooling intermediate speed} = \text{Cooling minimum speed} + \frac{\text{Cooling full speed} - \text{Cooling minimum speed}}{3}$$

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 blower 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 installation. Additionally, if conducting the 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 8 tests specified for a minimum compressor speed, turn off at least one indoor unit. 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 8 E_V Test, a cooling-mode intermediate compressor speed that falls within ¼ and ¾ of the difference between the full 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 8—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	Cooling Full	Cooling Full-Load.[2]
B_2 Test—required (steady, wet coil).	80	67	82	[1] 65	Cooling Full	Cooling Full-Load.[2]
E_V Test—required (steady, wet coil).	80	67	87	[1] 69	Cooling Intermediate	Cooling Intermediate.[3]
B_1 Test—required (steady, wet coil).	80	67	82	[1] 65	Cooling Minimum	Cooling Minimum.[4]
F_1 Test—required (steady, wet coil).	80	67	67	[1] 53.5	Cooling Minimum	Cooling Minimum.[4]
G_1 Test [5]—optional (steady, dry-coil).	80	([6])	67		Cooling Minimum	Cooling Minimum.[4]
I_1 Test[5]—optional (cyclic, dry-coil).	80	([6])	67		Cooling 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 of this appendix.
[3] Defined in section 3.1.4.3 of this appendix.
[4] Defined in section 3.1.4.2 of this appendix.
[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.

3.2.5 Cooling Mode Tests for Northern Heat Pumps With Triple-Capacity Compressors

Test triple-capacity, northern heat pumps for the cooling mode in the same way as specified in section 3.2.3 of this appendix for units having a two-capacity compressor.

3.2.6 Tests for an Air Conditioner or Heat Pump Having a Single Indoor Unit Having Multiple Indoor Blowers and Offering Two Stages of Compressor Modulation

Conduct the cooling mode tests specified in section 3.2.3 of this appendix.

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 blower 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 section 1.2 of this appendix, Definitions):

(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 of this appendix cases where its control is required, the water vapor content of the air entering the outdoor coil.

Refer to section 3.11 of this appendix 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 ANSI/ASHRAE 37–2009 for the indoor air enthalpy method and the user-selected secondary method. Make said Table 3 measurements at equal intervals that span 5 minutes or less. Continue data sampling until reaching a 30-minute period (*e.g.,* seven consecutive 5-minute samples) where the test tolerances specified in Table 9 are satisfied. For those continuously recorded parameters, use the entire data set from the 30-minute interval to evaluate Table 9 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 and sensible cooling capacity as specified in sections 7.3.3.1 and 7.3.3.3 of ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3). To calculate capacity, use the averages of the measurements (*e.g.* inlet and outlet dry bulb and wet bulb temperatures measured at the psychrometers) that are continuously recorded for the same 30-minute interval used as described above to evaluate compliance with test tolerances. 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. Use the values of the specific heat of air given in section 7.3.3.1 of ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3) for calculation of the sensible cooling capacities. Assign the average total space cooling capacity, average sensible cooling capacity, and electrical power consumption over the 30-minute data collection interval to the variables $\dot{Q}_c^k(T)$, $\dot{Q}_{sc}^k(T)$ and $\dot{E}_c^k(T)$, respectively. For these three 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 full speed, k=1 to denote low capacity or minimum speed, and k=v to denote the intermediate speed.

d. For mobile home and space-constrained ducted coil-only system tests, decrease $\dot{Q}_c^k(T)$ by

$$\frac{1385\ Btu/h}{1000\ scfm} * \overline{\dot{V}_s}$$

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

$$\frac{406\ W}{1000\ scfm} * \overline{\dot{V}_s}$$

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).

For non-mobile, non-space-constrained home ducted coil-only system tests, decrease $\dot{Q}_c^k(T)$ by

$$\frac{1505\ Btu/h}{1000\ scfm} * \overline{\dot{V}_s}$$

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

$$\frac{441\ W}{1000\ scfm} * \overline{\dot{V}_s}$$

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 9—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 [1]
Indoor dry-bulb, °F		
Entering temperature	2.0	0.5
Leaving temperature	2.0	
Indoor wet-bulb, °F		
Entering temperature	1.0	[2] 0.3
Leaving temperature	[2] 1.0	
Outdoor dry-bulb, °F		
Entering temperature	2.0	0.5
Leaving temperature	[3] 2.0	
Outdoor wet-bulb, °F		
Entering temperature	1.0	[4] 0.3
Leaving temperature	[3] 1.0	
External resistance to airflow, inches of water	0.05	[5] 0.02
Electrical voltage, % of reading.	2.0	1.5
Nozzle pressure drop, % of reading.	2.0	

[1] See section 1.2 of this appendix, Definitions.
[2] Only applies during wet coil tests; does not apply during steady-state, dry coil cooling mode tests.
[3] Only applies when using the outdoor air enthalpy method.
[4] Only applies during wet coil cooling mode tests where the unit rejects condensate to the outdoor coil.
[5] Only applies when testing non-ducted units.

e. For air conditioners and heat pumps having a constant-air-volume-rate indoor blower, 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 blower 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 blower 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 blower motor at ΔP_{min} using linear extrapolation:

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

(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 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 of this appendix 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 ANSI/ASHRAE 37–2009). In preparing for the section 3.5 cyclic tests of this appendix, record the average indoor-side air volume rate, $\dot{V}$, 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 blower (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 blower turned off (see section 3.5 of this appendix), include the electrical power used by the indoor blower motor among the recorded parameters from the 30-minute test.

c. If the temperature sensors used to provide the primary measurement of the indoor-side dry bulb temperature difference during the steady-state dry-coil test and the subsequent cyclic dry-coil test are different, include measurements of the latter sensors among the regularly sampled data. Beginning at the start of the 30-minute data collection period, measure and compute the indoor-side air dry-bulb temperature difference using both sets of instrumentation, ΔT (Set SS) and ΔT (Set CYC), for each equally spaced data sample. If using a consistent data sampling rate that is less than 1 minute, calculate and record minutely averages for the two temperature differences. If using a consistent sampling rate of one minute or more, calculate and record the two temperature differences from each data sample. After having recorded the seventh (i=7) set of temperature differences, calculate the following ratio using the first seven sets of values:

$$F_{CD} = \frac{1}{7}\sum_{i-6}^{i} \frac{\Delta T(Set\ SS)}{\Delta T(Set\ CYC)}$$

Each time a subsequent set of temperature differences is recorded (if sampling more frequently than every 5 minutes), calculate F_{CD} using the most recent seven sets of values. Continue these calculations until the 30-minute period is completed or until a value for F_{CD} is calculated that falls outside the allowable range of 0.94–1.06. If the latter occurs, immediately suspend the test and identify the cause for the disparity in the two temperature difference measurements. Recalibration of one or both sets of instrumentation may be required. If all the values for F_{CD} are within the allowable range, save the final value of the ratio from the 30-minute test as F_{CD}*. If the temperature sensors used to provide the primary measurement of the indoor-side dry bulb temperature difference during the steady-state dry-coil test and the subsequent cyclic dry-coil test are the same, set F_{CD}*= 1.

3.5 Test Procedures for the Cyclic Dry-Coil Cooling-Mode Tests (the D, D_1, D_2, and I_1 Tests)

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 blower, the manufacturer has the option of electing at the outset whether to conduct the cyclic test with the indoor blower enabled or disabled. Always revert to testing with the indoor blower disabled if cyclic testing with the fan enabled is unsuccessful.

a. For all cyclic tests, the measured capacity must be adjusted for the thermal mass stored in devices and connections located between measured points. Follow the procedure outlined in section 7.4.3.4.5 of ASHRAE 116–2010 (incorporated by reference, see § 430.3) to ensure any required measurements are taken.

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. If an upturned duct is used, measure the dry-bulb temperature at the inlet of the device at least once every minute and ensure that its test operating tolerance is within 1.0 °F for each compressor OFF period.

c. Sections 3.5.1 and 3.5.2 of this appendix specify airflow requirements through the indoor coil of ducted and non-ducted indoor units, respectively. In all cases, use the exhaust fan of the airflow measuring apparatus (covered under section 2.6 of this appendix) along with the indoor blower 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 blower 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 blower, 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 blower 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 indoor blower, temporarily remove the indoor blower.

e. Conduct three complete compressor OFF/ON cycles with the test tolerances given in Table 10 satisfied. Calculate the degradation coefficient C_D for each complete cycle. If all three C_D values are within 0.02 of the average C_D then stability has been achieved, use the highest C_D value of these three. If stability has not been achieved, conduct additional cycles, up to a maximum of eight cycles, until stability has been achieved between three consecutive cycles. Once stability has been achieved, use the highest C_D value of the three consecutive cycles that establish stability. If stability has not been achieved after eight cycles, use the highest C_D from cycle one through cycle eight, or the default C_D, whichever is lower.

f. With regard to the Table 10 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 blower (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 blower coil systems tested with the unit's indoor blower operating for the cycling test, integrate electrical power from indoor blower OFF to indoor blower 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 10—TEST OPERATING AND TEST CONDITION TOLERANCES FOR CYCLIC DRY COIL COOLING MODE TESTS

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

[1] See section 1.2 of this appendix, Definitions.
[2] 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 blower that ramps, the tolerances listed for the external resistance to airflow apply from 30 seconds after achieving full speed until ramp down begins.
[3] Shall at no time exceed a wet-bulb temperature that results in condensate forming on the indoor coil.
[4] The test condition must be the average nozzle pressure difference or velocity pressure measured during the steady-state dry coil test.
[5] Applies during the interval when at least one of the following—the compressor, the outdoor fan, or, if applicable, the indoor blower—are operating except for the first 30 seconds after compressor start-up.

If the Table 10 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,

$$q_{cyc,dry} = \frac{60*\bar{\dot{V}}*C_{p,a}*\Gamma}{[v_n'*(1+W_n)]} = \frac{60*\bar{\dot{V}}*C_{p,a}*\Gamma}{v_n} \quad \text{and} \quad \Gamma = F_{CD}^* \int_{\tau_1}^{\tau_2} [T_{a1}(\tau) - T_{a2}(\tau)]\delta\tau,\ hr * {}^\circ\text{F}$$

Where,

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

$T_{a1}(\tau)$ = 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.

Adjust the total space cooling delivered, $q_{cyc,dry}$, according to calculation method outlined in section 7.4.3.4.5 of ASHRAE 116–2010 (incorporated by reference, see § 430.3).

3.5.1 Procedures When Testing Ducted Systems

The automatic controls that are installed in 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 the indoor blower of the test unit). For ducted coil-only systems rated based on using a fan time-delay relay, control the indoor coil airflow according to the OFF delay listed by the manufacturer in the certification report. For ducted units having a variable-speed indoor blower 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 coil-only systems, cycle the indoor coil airflow in unison with the cycling of the compressor. If air damper boxes are used, close them on the inlet and outlet side 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 blower. For mobile home and space-constrained ducted coil-only systems increase $e_{cyc,dry}$ by the quantity,

$$\text{Equation 3.5-2.} \quad \frac{406\, W}{1000\, scfm} * \bar{\dot{V}}_s * [\tau_2 - \tau_1]$$

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

$$\text{Equation 3.5-3.} \quad \frac{1385\, Btu/h}{1000\, scfm} * \bar{\dot{V}}_s * [\tau_2 - \tau_1]$$

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 ducted non-mobile, non-space-constrained home coil-only units increase $e_{cyc,dry}$ by the quantity,

$$\text{Equation 3.5-2.} \quad \frac{441\, W}{1000\, scfm} * \bar{\dot{V}}_s * [\tau_2 - \tau_1]$$

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

$$\text{Equation 3.5-3.} \quad \frac{1505\, Btu/h}{1000\, scfm} * \bar{\dot{V}}_s * [\tau_2 - \tau_1]$$

where $\overline{V}_s$ 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 blower 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 blower power measured during or following the dry coil steady-state test; or,

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

(1) Measure the electrical power consumed by the variable-speed indoor blower 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 midpoint 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 blower 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 the test setup instructions included with the unit by the manufacturer specifies a ramp interval that exceeds 45 seconds, use a 45-second ramp interval nonetheless when estimating the fan energy.

3.5.2 Procedures When Testing Non-Ducted Indoor Units

Do not use airflow prevention devices when conducting cyclic tests on non-ducted indoor 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 blower 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 blower 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 blower during the 3 minutes after compressor cutoff to the integrated cooling capacity, $q_{cyc,dry}$. For the case where the non-ducted indoor unit uses a variable-speed indoor blower 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 of this appendix for ducted units having a disabled variable-speed indoor blower.

3.5.3 Cooling-Mode Cyclic-Degradation Coefficient Calculation

Use the two 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 for variable-speed compressor systems and outdoor units with no match, and 0.20 for all other systems. 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

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

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 and Fixed Heating Air Volume Rate

This set of tests is for single-speed-compressor heat pumps that do not have a heating minimum air volume rate or a heating intermediate air volume rate that is different than the heating full load air volume rate. Conducting a very low temperature test (H4) is optional. 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 five tests are specified in Table 11 of this section.

TABLE 11—HEATING MODE TEST CONDITIONS FOR UNITS HAVING A SINGLE-SPEED COMPRESSOR AND A FIXED-SPEED INDOOR BLOWER, A CONSTANT AIR VOLUME RATE INDOOR BLOWER, OR COIL-ONLY

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]
H3 Test (required, steady)	70	60(max)	17	15	Heating Full-load.[1]
H4 Test (optional, steady)	70	60(max)	5	3(max)	Heating Full-load.[1]

[1] Defined in section 3.1.4.4 of this appendix.

[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 Single Indoor Unit Having Either (1) a Variable-Speed, Variable-Air-Rate Indoor Blower Whose Capacity Modulation Correlates With Outdoor Dry Bulb Temperature or (2) Multiple Indoor Blowers

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 ($H2_1$) and a very low temperature test ($H4_2$) 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 12. 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:

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

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

where,

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

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

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 of this appendix; 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 of this appendix; 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 of this appendix.

TABLE 12—HEATING MODE TEST CONDITIONS FOR UNITS WITH A SINGLE-SPEED COMPRESSOR THAT MEET THE SECTION 3.6.2 INDOOR UNIT REQUIREMENTS

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]
$H2_1$ 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]
$H4_2$ Test (optional, steady)	70	60(max)	5	3(max)	Heating Full-load.[1]

[1] Defined in section 3.1.4.4 of this appendix.
[2] Defined in section 3.1.4.5 of this appendix.
[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 Section 1.2 of This Appendix, Definitions), Including Two-Capacity, Northern Heat Pumps (see Section 1.2 of This Appendix, Definitions)

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 ($H3_2$). Conducting a very low temperature test ($H4_2$) is optional. 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 of this appendix seasonal performance calculations; and

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

If the two conditions in a.(1) and a.(2) of this section are met, an alternative to conducting the $H2_1$ frost accumulation is to use the following equations to approximate the capacity and electrical power:

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

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

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 of this appendix. 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 of this appendix.

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 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 13 specifies test conditions for these nine tests.

TABLE 13—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[7], cyclic)	70	60 (max)	47	43	High	([3])
$H1_1$ Test (required)	70	60 (max)	47	43	Low	Heating Minimum.[1]
$H1C_1$ Test (optional, cyclic)	70	60 (max)	47	43	Low	([4])
$H2_2$ Test (required)	70	60 (max)	35	33	High	Heating Full-Load.[2]
$H2_1$ Test[5] [6] (required)	70	60 (max)	35	33	Low	Heating Minimum.[1]
$H3_2$ Test (required, steady)	70	60 (max)	17	15	High	Heating Full-Load.[2]
$H3_1$ Test[5] (required, steady)	70	60 (max)	17	15	Low	Heating Minimum.[1]
$H4_2$ Test (Optional, steady)	70	60 (max)	5	3 (max)	High	Heating Full-Load.[2]

[1] Defined in section 3.1.4.5 of this appendix.
[2] Defined in section 3.1.4.4 of this appendix.
[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 *HSPF2* 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.
[7] Required only if the heat pump locks out low capacity operation at lower outdoor temperatures.

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 ($H1_N$ and $H1_1$), one frost accumulation test ($H2_V$), and one low temperature test ($H3_2$). Conducting one or more of the following tests is optional: An additional high temperature test ($H1_2$), an additional frost accumulation test ($H2_2$), and a very low temperature test ($H4_2$). 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 nine tests are specified in Table 14. The compressor shall operate at the same heating full speed, measured by RPM or power input frequency (Hz), as the maximum speed at which the system controls would operate the compressor in normal operation in 17 °F ambient temperature, for the $H1_2$, $H2_2$ and $H3_2$ Tests. The compressor shall operate for the $H1_N$ test at the maximum speed at which the system controls would operate the compressor in normal operation in 47 °F ambient temperature. The compressor shall operate at the same heating minimum speed, measured by RPM or power input frequency (Hz), for the $H0_1$, $H1C_1$, and $H1_1$ Tests. Determine the heating intermediate compressor speed cited in Table 14 using the heating mode full and minimum compressors speeds and:

$$\text{Heating intermediate speed} = \text{Heating minimum speed} + \frac{\text{Heating full speed} - \text{Heating minimum speed}}{3}$$

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

b. If one of the high temperature tests ($H1_2$ or $H1_N$) is conducted using the same compressor speed (RPM or power input frequency) as the $H3_2$ test, set the 47 °F capacity and power input values used for calculation of HSPF2 equal to the measured values for that test:

$$\dot{Q}_{hcalc}^{k=2}(47) = \dot{Q}_h^{k=2}(47);\ \dot{E}_{hcalc}^{k=2}(47) = \dot{E}_h^{k=2}(47)$$

Where:

$\dot{Q}_{hcalc}{}^{k=2}(47)$ and $\dot{E}_{hcalc}{}^{k=2}(47)$ are the capacity and power input representing full-speed operation at 47 °F for the HSPF2 calculations,

$\dot{Q}_h{}^{k=2}(47)$ is the capacity measured in the high temperature test ($H1_2$ or $H1_N$) which used the same compressor speed as the $H3_2$ test, and

$\dot{E}_h{}^{k=2}(47)$ is the power input measured in the high temperature test ($H1_2$ or $H1_N$) which used the same compressor speed as the $H3_2$ test.

Evaluate the quantities $\dot{Q}_h{}^{k=2}(47)$ and from $\dot{E}_h{}^{k=2}(47)$ according to section 3.7.

Otherwise (if no high temperature test is conducted using the same speed (RPM or power input frequency) as the $H3_2$ test), calculate the 47 °F capacity and power input values used for calculation of HSPF2 as follows:

$$\dot{Q}_{hcalc}^{k=2}(47) = \dot{Q}_h^{k=2}(17) * (1 + 30°F * CSF);$$

$$\dot{E}_{hcalc}^{k=2}(47) = \dot{E}_h^{k=2}(17) * (1 + 30°F * PSF)$$

Where:

$\dot{Q}_{hcalc}{}^{k=2}(47)$ and $\dot{E}_{hcalc}{}^{k=2}(47)$ are the capacity and power input representing full-speed operation at 47 °F for the HSPF2 calculations,

$\dot{Q}_h{}^{k=2}(17)$ is the capacity measured in the $H3_2$ test,

$\dot{E}_h{}^{k=2}(17)$ is the power input measured in the $H3_2$ test,

CSF is the capacity slope factor, equal to 0.0204/°F for split systems and 0.0262/°F for single-package systems, and

PSF is the Power Slope Factor, equal to 0.00455/°F.

c. 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:

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

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

Where:

$\dot{Q}_{hcalc}{}^{k=2}(47)$ and $\dot{E}_{hcalc}{}^{k=2}(47)$ are the capacity and power input representing full-speed operation at 47 °F for the HSPF2 calculations, calculated as described in section b above.

$\dot{Q}_h{}^{k=2}(17)$ and $\dot{E}_h{}^{k=2}(17)$ are the capacity and power input measured in the $H3_2$ test.

d. Determine the quantities $\dot{Q}_h{}^{k=2}(17)$ and $\dot{E}_h{}^{k=2}(17)$ from the $H3_2$ test, determine the quantities $Q_h{}^{k=2}(5)$ and $E_h{}^{k=2}(5)$ from the $H4_2$ test, and evaluate all four according to section 3.10.

TABLE 14—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	Heating Minimum	Heating Minimum.[1]
$H1_2$ test (optional, steady).	70	60 (max)	47	43	Heating Full[4]	Heating Full-Load.[3]
$H1_1$ test (required, steady).	70	60(max)	47	43	Heating Minimum	Heating Minimum.[1]
$H1_N$ test (required, steady).	70	60(max)	47	43	Heating Full[5]	Heating Full-Load.[3]
$H1C_1$ test (optional, cyclic).	70	60(max)	47	43	Heating Minimum	([2])
$H2_2$ test (optional).	70	60 (max)	35	33	Heating Full[4]	Heating Full-Load.[3]
$H2_V$ test (required).	70	60 (max)	35	33	Heating Intermediate	Heating Intermediate.[6]
$H3_2$ test (required, steady).	70	60 (max)	17	15	Heating Full[4]	Heating Full-Load.[3]
$H4_2$ test (optional, steady).	70	60 (max)	5	3 (max)	Heating Full	Heating Full-Load.[3]

[1] Defined in section 3.1.4.5 of this appendix.
[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 $H1_1$ test.
[3] Defined in section 3.1.4.4 of this appendix.
[4] Maximum speed that the system controls would operate the compressor in normal operation in 17 °F ambient temperature. The $H1_2$ test is not needed if the $H1_N$ test uses this same compressor speed.
[5] Maximum speed that the system controls would operate the compressor in normal operation in 47 °F ambient temperature.
[6] Defined in section 3.1.4.6 of this appendix.

e. For multiple-split heat pumps (only), the following procedures supersede the above requirements. For all Table 14 tests specified for a minimum compressor speed, turn off at least one indoor unit. 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 14 $H2_V$ test, a heating mode intermediate compressor speed that falls within ¼ and ¾ of the difference between the full 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 section 1.2 of this appendix, Definitions) 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 of this appendix 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.)

3.6.6 Heating Mode Tests for Northern Heat Pumps with Triple-Capacity Compressors

Test triple-capacity, northern heat pumps for the heating mode as follows:

a. Conduct one maximum temperature test ($H0_1$), two high temperature tests ($H1_2$ and $H1_1$), one frost accumulation test ($H2_2$), two low temperature tests ($H3_2$, $H3_3$), and one very low temperature test ($H4_3$). 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 4.2.6 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 test to determine $\dot{Q}_h^{k=1}(35)$ and $E_h^{k=1}(35)$ is to use the following equations to approximate this capacity and electrical power:

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

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

In evaluating the above equations, determine the quantities $Q_h^{k=1}(47)$ from the $H1_1$ test and evaluate them according to section 3.7 of this appendix. 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 of this appendix. Use the paired values of $\dot{Q}_h^{k=1}(35)$ and $\dot{E}_h^{k=1}(35)$ derived from conducting the $H2_1$ frost accumulation test and evaluated as specified in section 3.9.1 of this appendix or use the paired values calculated using the above default equations, whichever contribute to a higher Region IV HSPF2 based on the DHRmin.

b. Conducting a frost accumulation test ($H2_3$) with the heat pump operating at its booster capacity is optional. If this optional test is not conducted, determine $\dot{Q}_h^{k=3}(35)$ and $\dot{E}_h^{k=3}(35)$ using the following equations to approximate this capacity and electrical power:

$$\dot{Q}_h^{k=3}(35) = QR_h^{k=2}(35) * \left\{\dot{Q}_h^{k=3}(17) + 1.20 * \left[\dot{Q}_h^{k=3}(17) - \dot{Q}_h^{k=3}(5)\right]\right\}$$

$$\dot{E}_h^{k=3}(35) = PR_h^{k=2}(35) * \left\{\dot{E}_h^{k=3}(17) + 1.20 * \left[\dot{E}_h^{k=3}(17) - \dot{E}_h^{k=3}(5)\right]\right\}$$

Where:

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

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

Determine the quantities $\dot{Q}_h^{k=2}(47)$ and $\dot{E}_h^{k=2}(47)$ from the $H1_2$ test and evaluate them according to section 3.7 of this appendix. Determine the quantities $\dot{Q}_h^{k=2}(35)$ and $\dot{E}_h^{k=2}(35)$ from the $H2_2$ test and evaluate them according to section 3.9.1 of this appendix. Determine the quantities $\dot{Q}_h^{k=2}(17)$ and $\dot{E}_h^{k=2}(17)$ from the $H3_2$ test, determine the quantities $\dot{Q}_h^{k=3}(17)$ and $\dot{E}_h^{k=3}(17)$ from the $H3_3$ test, and determine the quantities $\dot{Q}_h^{k=3}(5)$ and $\dot{E}_h^{k=3}(5)$ from the $H4_3$ test. Evaluate all six quantities according to section 3.10 of this appendix. Use the paired values of $\dot{Q}_h^{k=3}(35)$ and $\dot{E}_h^{k=3}(35)$ derived from conducting the $H2_3$ frost accumulation test and calculated as specified in section 3.9.1 of this appendix or use the paired values calculated using the above default equations, whichever contribute to a higher Region IV HSPF2 based on the DHRmin.

c. Conduct the optional high temperature cyclic test ($H1C_1$) to determine the heating mode cyclic-degradation coefficient, C_D^h. A default value for C_D^h of 0.25 may be used in lieu of conducting the cyclic. If a triple-capacity heat pump locks out low capacity operation at lower outdoor temperatures, conduct the high temperature cyclic test ($H1C_2$) to determine the high capacity heating mode cyclic-degradation coefficient, C_D^h (k=2). 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)]. Finally, if a triple-capacity heat pump locks out both low and high capacity operation at the lowest outdoor temperatures, conduct the low temperature cyclic test ($H3C_3$) to determine the booster-capacity heating mode cyclic-degradation coefficient, C_D^h (k=3). The default C_D^h (k=3) is the same value as determined or assigned for the high capacity cyclic-degradation coefficient, C_D^h [or equivalently, C_D^h (k=2)]. Table 15 specifies test conditions for all 13 tests.

TABLE 15—HEATING MODE TEST CONDITIONS FOR UNITS WITH A TRIPLE-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,[8] cyclic)	70	60(max)	47	43	High	(3)
$H1_1$ Test (required)	70	60(max)	47	43	Low	Heating Minimum [1]
$H1C_1$ Test (optional, cyclic)	70	60(max)	47	43	Low	(4)
$H2_3$ Test (optional, steady)	70	60(max)	35	33	Booster	Heating Full-Load [2]
$H2_2$ Test (required)	70	60(max)	35	33	High	Heating Full-Load [2]
$H2_1$ Test (required)	70	60(max)	35	33	Low	Heating Minimum [1]
$H3_3$ Test (required, steady)	70	60(max)	17	15	Booster	Heating Full-Load [2]
$H3C_3$ Test[5 6] (optional, cyclic)	70	60(max)	17	15	Booster	(7)
$H3_2$ Test (required, steady)	70	60(max)	17	15	High	Heating Full-Load [2]
$H3_1$ Test[5] (required, steady)	70	60(max)	17	15	Low	Heating Minimum [1]
$H4_3$ Test (required, steady)	70	60(max)	5	3(max)	Booster	Heating Full-Load [2]

[1] Defined in section 3.1.4.5 of this appendix.
[2] Defined in section 3.1.4.4 of this appendix.
[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 4.2.6 HSPF2 calculations.
[6] If table note [5] applies, the section 3.6.6 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.
[7] 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 $H3_3$ test.
[8] Required only if the heat pump locks out low capacity operation at lower outdoor temperatures

3.6.7 Tests for a Heat Pump Having a Single Indoor Unit Having Multiple Indoor Blowers and Offering Two Stages of Compressor Modulation. Conduct the Heating Mode Tests Specified in Section 3.6.3 of this Appendix

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 blower 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 of this appendix 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 ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3) for the indoor air enthalpy method and the user-selected secondary method. Make said Table 3 measurements at equal intervals that span 5 minutes or less. Continue data sampling until a 30-minute period (*e.g.,* seven consecutive 5-minute samples) is reached where the test tolerances specified in Table 16 are satisfied. For those continuously recorded parameters, use the entire data set for the 30-minute interval when evaluating Table 16 compliance. Determine the average electrical power consumption of the heat pump over the same 30-minute interval.

TABLE 16—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 [1]
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	[2] 1.0	
External resistance to airflow, inches of water	0.05	[3] 0.02
Electrical voltage, % of reading	2.0	1.5
Nozzle pressure drop, % of reading	2.0	

[1] See section 1.2 of this appendix, Definitions.
[2] Only applies when the Outdoor Air Enthalpy Method is used.
[3] 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 ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3). To calculate capacity, use the averages of the measurements (*e.g.* inlet and outlet dry bulb temperatures measured at the psychrometers) that are continuously recorded for the same 30-minute interval used as described above to evaluate compliance with test tolerances. 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 of this appendix. Additionally, for the heating mode, use the superscript to denote results from the optional $H1_N$ test, if conducted.

c. For mobile home and space-constrained coil-only system heat pumps, increase $\dot{Q}_h^k(T)$ by

$$\frac{1385\,BTU/h}{1000\,scfm} * \bar{\dot{V}}_s$$

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

$$\frac{406\,W}{1000\,scfm} * \bar{\dot{V}}_s$$

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

For non-mobile home, non-space-constrained coil-only system heat pumps, increase $\dot{Q}_h^k(T)$ by

$$\frac{1505\,BTU/h}{1000\,scfm} * \bar{\dot{V}}_s$$

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

$$\frac{441\,W}{1000\,scfm} * \bar{\dot{V}}_s$$

where $\bar{V}_s$ 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 cycle 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 16 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 cyclic heating mode test, which is described in section 3.8 of this appendix, record the average indoor-side air volume rate, $\dot{V}$, 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 blower motor ($\dot{E}_{fan,1}$):

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

(2) The heat pump has a (variable-speed) constant-air volume-rate indoor blower 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 blower 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 blower motor if the 30-minute test had been conducted at ΔP_{min} using linear extrapolation:

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

(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.

e. If the temperature sensors used to provide the primary measurement of the indoor-side dry bulb temperature difference during the steady-state dry-coil test and the subsequent cyclic dry-coil test are different, include measurements of the latter sensors among the regularly sampled data. Beginning at the start of the 30-minute data collection period, measure and compute the indoor-side air dry-bulb temperature difference using both sets of instrumentation, ΔT (Set SS) and ΔT (Set CYC), for each equally spaced data sample. If using a consistent data sampling rate that is less than 1 minute, calculate and record minutely averages for the two temperature differences. If using a consistent sampling rate of one minute or more, calculate and record the two temperature differences from each data sample. After having recorded the seventh (i=7) set of temperature differences, calculate the following ratio using the first seven sets of values:

$$F_{CD} = \frac{1}{7}\sum_{i-6}^{i} \frac{\Delta T(Set\ SS)}{\Delta T(Set\ CYC)}$$

Each time a subsequent set of temperature differences is recorded (if sampling more frequently than every 5 minutes), calculate F_{CD} using the most recent seven sets of values. Continue these calculations until the 30-minute period is completed or until a value for F_{CD} is calculated that falls outside the allowable range of 0.94–1.06. If the latter occurs, immediately suspend the test and identify the cause for the disparity in the two temperature difference measurements. Recalibration of one or both sets of instrumentation may be required. If all the values for F_{CD} are within the allowable range, save the final value of the ratio from the 30-minute test as $F_{CD}*$. If the temperature sensors used to provide the primary measurement of the indoor-side dry bulb temperature difference during the steady-state dry-coil test and the subsequent cyclic dry-coil test are the same, set $F_{CD}*= 1$.

3.8 Test Procedures for the 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 of this appendix. 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 17 rather than Table 10. Record the outdoor coil entering wet-bulb temperature according to the requirements given in section 3.5 of this appendix for the outdoor coil entering dry-bulb temperature. Drop the subscript ''dry'' used in variables cited in section 3.5 of this appendix when referring to quantities from the cyclic heating mode test. If available, use electric resistance heaters (see section 2.1 of this appendix) to minimize the variation in the inlet air temperature. Determine the total space heating delivered during the cyclic heating test, q_{cyc}, as specified in section 3.5 of this appendix 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

$$\Gamma \text{ using, } \Gamma = F_{CD}^{*}\int_{\tau_1}^{\tau_2}[T_{a1}(\tau) - T_{a2}(\tau)]\delta\tau,\ hr \times {}^\circ F,$$

where F_{CD}^* is the value recorded during the section 3.7 steady-state test conducted at the same test condition.

b. For ducted coil-only system heat pumps (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 blower 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 blower 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 required 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 ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3) 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:

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

Where:

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

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

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

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.

$$HLF = \frac{q_{cyc}}{\dot{Q}_h^k(T_{cyc}) * \Delta\tau_{cyc}}$$

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 17—TEST OPERATING AND TEST CONDITION TOLERANCES FOR CYCLIC HEATING MODE TESTS

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

[1] See section 1.2 of this appendix, Definitions.
[2] 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 blower that ramps, the tolerances listed for the external resistance to airflow shall apply from 30 seconds after achieving full speed until ramp down begins.
[3] The test condition must be the average nozzle pressure difference or velocity pressure measured during the steady-state test conducted at the same test conditions.
[4] Applies during the interval that at least one of the following—the compressor, the outdoor fan, or, if applicable, the indoor blower—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 of this appendix. 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 section 1.2 of this appendix, Definitions), 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 blower 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 blower. If it is installed, use the outlet damper box described in section 2.5.4.1 of this appendix 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 18 during both the preliminary and official test periods. As noted in Table 18, 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 18) and

(2) When defrosting, plus these same first 10 minutes after defrost termination (sub-interval D, as described in Table 18). Evaluate compliance with Table 18 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 18 at equal intervals that span 5 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 blower 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 coil-only system heat pumps, 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 ANSI/ASHRAE 37–2009) at equal intervals that span 10 minutes or less. (**Note:** In the first printing of ANSI/ASHRAE 37–2009, 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 18—TEST OPERATING AND TEST CONDITION TOLERANCES FOR FROST ACCUMULATION HEATING MODE TESTS

	Test operating tolerance [1]		Test condition tolerance [1]
	Sub-interval H [2]	Sub-interval D [3]	Sub-interval H [2]
Indoor entering dry-bulb temperature, °F	2.0	[4] 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		[5] 0.02
Electrical voltage, % of reading	2.0		1.5

[1] See section 1.2 of this appendix, Definitions.

[2] Applies when the heat pump is in the heating mode, except for the first 10 minutes after termination of a defrost cycle.

[3] 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.

[4] For heat pumps that turn off the indoor blower during the defrost cycle, the noted tolerance only applies during the 10 minute interval that follows defrost termination.

[5] 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:

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

where,

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

$C_{p,a} = 0.24 + 0.444 \cdot W_n$, 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.

$$\Gamma = \int_{\tau_1}^{\tau_2} [T_{a2}(\tau) - T_{a1}(\tau)] d\tau, \; hr * {}^\circ F$$

$T_{a1}(\tau)$ = 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 blower 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 blower 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 ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3).

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 mobile home and space-constrained coil-only system heat pumps, increase $\dot{Q}_h^k(35)$ by

$$\frac{1385\,BTU/h}{1000\,scfm} * \bar{\dot{V}}_s$$

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

$$\frac{406\,W}{1000\,scfm} * \bar{\dot{V}}_s$$

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

For non-mobile home, non-space-constrained coil-only system heat pumps, increase $\dot{Q}_h^k(35)$ by

$$\frac{1505\,BTU/h}{1000\,scfm} * \bar{\dot{V}}_s$$

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

$$\frac{441\,W}{1000\,scfm} * \bar{\dot{V}}_s$$

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

c. For heat pumps having a constant-air-volume-rate indoor blower, 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 blower 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 blower 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 blower motor had the frost accumulation heating mode test been conducted at ΔP_{min} using linear extrapolation:

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

(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 of this appendix to the value of 1 in all cases except for heat pumps having a demand-defrost control system (see section 1.2 of this appendix, Definitions). For such qualifying heat pumps, evaluate F_{def} using,

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

where:

$\Delta\tau_{def}$ = the time between defrost terminations (in hours) or 1.5, whichever is greater. Assign a value of 6 to $\Delta\tau_{def}$ if this limit is reached during a frost accumulation test and the heat pump has not completed a defrost cycle.

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

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-load air 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 and Very Low Temperature Heating Mode Tests (the H3, H3$_2$, H3$_1$, H3$_3$, H4, H4$_2$, and H4$_3$ Tests)

Except for the modifications noted in this section, conduct the low temperature and very low temperature heating mode tests using the same approach as specified in section 3.7 of this appendix 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 the capacity and power input, conduct a defrost cycle. This defrost cycle may be manually or automatically initiated. Terminate the defrost sequence using the heat pump's defrost controls. Begin the 30-minute data collection interval described in section 3.7 of this appendix, from which the capacity and power input 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.

a. For all cooling mode and heating mode tests, first conduct a test without the outdoor air-side test apparatus described in section 2.10.1 of this appendix connected to the outdoor unit (''free outdoor air'' test).

b. For the first section 3.2 steady-state cooling mode test and the first section 3.6 steady-state heating mode test, conduct a second test in which the outdoor-side apparatus is connected (''ducted outdoor air'' test). No other cooling mode or heating mode tests require the ducted outdoor air test 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 the ducted outdoor air test for 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 Free Outdoor Air Test

a. For the free outdoor air test, connect the indoor air-side test apparatus to the indoor coil; do not connect 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 5 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.,* seven consecutive 5-minute samples) is obtained where the Table 9 or Table 16, whichever applies, test tolerances are satisfied.

b. For cases where a ducted outdoor air test is not required per section 3.11.1.b of this appendix, the free outdoor air test constitutes the ''official'' test for which validity is not based on comparison with a secondary test.

c. For cases where a ducted outdoor air test is required per section 3.11.1.b of this appendix, the following conditions must be met for the free outdoor air test to constitute a valid ''official'' test:

(1) The energy balance specified in section 3.1.1 of this appendix is achieved for the ducted outdoor air test (*i.e.,* compare the capacities determined using the indoor air enthalpy method and the outdoor air enthalpy method).

(2) The capacities determined using the indoor air enthalpy method from the ducted outdoor air and free outdoor air tests must agree within 2 percent.

3.11.1.2 Ducted Outdoor Air Test

a. The test conditions and tolerances for the ducted outdoor air test are the same as specified for the official test, where the official test is the free outdoor air test described in section 3.11.1.1 of this appendix.

b. After collecting 30 minutes of steady-state data during the free outdoor air test, connect the outdoor air-side test apparatus to the unit for the ducted outdoor air test. 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 during the free outdoor air test. Collect 30 minutes of steady-state data after re-establishing equilibrium conditions.

c. During the ducted outdoor air test, at intervals of 5 minutes or less, measure the parameters required according to the indoor air enthalpy method and the outdoor air enthalpy method for the prescribed 30 minutes.

d. For cooling mode ducted outdoor air tests, calculate capacity based on outdoor air-enthalpy measurements as specified in sections 7.3.3.2 and 7.3.3.3 of ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3). For heating mode ducted tests, 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 ANSI/ASHRAE Standard. Adjust the outdoor-side capacity according to section 7.3.3.4 of ANSI/ASHRAE 37–2009 to account for line losses when testing split systems. As described in section 8.6.2 of ANSI/ASHRAE 37–2009, use the outdoor air volume rate as measured during the ducted outdoor air tests to calculate capacity for checking the agreement with the capacity calculated using the indoor air enthalpy method.

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 5 minutes or less. Determine average flow rate or average capacity from data sampled over a 30-minute period where the Table 9 (cooling) or the Table 16 (heating) tolerances are satisfied. Otherwise, conduct the calibration tests according to sections 5, 6, 7, and 8 of ASHRAE 23.1–2010 (incorporated by reference, see § 430.3); sections 5, 6, 7, 8, 9, and 11 of ASHRAE 41.9–2011 (incorporated by reference, see § 430.3); and section 7.4 of ANSI/ASHRAE 37–2009 (incorporated by reference, see § 430.3).

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 ANSI/ASHRAE 37–2009.

3.11.3 If Using the Refrigerant-Enthalpy Method as the Secondary Test Method

Conduct this secondary method according to section 7.5 of ANSI/ASHRAE 37–2009. 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 ANSI/ASHRAE Standard.

3.12 Rounding of Space Conditioning Capacities for Reporting Purposes

a. When reporting rated capacities, round them off as specified in § 430.23 (for a single unit) and in 10 CFR 429.16 (for a sample).

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

3.13 Laboratory Testing To Determine Off Mode Average Power Ratings

Voltage tolerances: As a percentage of reading, test operating tolerance must be 2.0 percent and test condition tolerance must be 1.5 percent (see section 1.2 of this appendix for definitions of these tolerances).

Conduct one of the following tests: If the central air conditioner or heat pump lacks a compressor crankcase heater, perform the test in section 3.13.1 of this appendix; if the central air conditioner or heat pump has a compressor crankcase heater that lacks controls and is not self-regulating, perform the test in section 3.13.1 of this appendix; if the central air conditioner or heat pump has a crankcase heater with a fixed power input controlled with a thermostat that measures ambient temperature and whose sensing element temperature is not affected by the heater, perform the test in section 3.13.1 of this appendix; if the central air conditioner or heat pump has a compressor crankcase heater equipped with self-regulating control or with controls for which the sensing element temperature is affected by the heater, perform the test in section 3.13.2 of this appendix.

3.13.1 This Test Determines the Off Mode Average Power Rating for Central Air Conditioners and Heat Pumps That Lack a Compressor Crankcase Heater, or Have a Compressor Crankcase Heating System That Can Be Tested Without Control of Ambient Temperature During the Test. This Test Has No Ambient Condition Requirements

a. Test Sample Set-up and Power Measurement: For coil-only systems, provide a furnace or modular blower that is compatible with the system to serve as an interface with the thermostat (if used for the test) and to provide low-voltage control circuit power. Make all control circuit connections between the furnace (or modular blower) and the outdoor unit as specified by the manufacturer's installation instructions. Measure power supplied to both the furnace (or modular blower) and power supplied to the outdoor unit. Alternatively, provide a compatible transformer to supply low-voltage control circuit power, as described in section 2.2.d of this appendix. Measure transformer power, either supplied to the primary winding or supplied by the secondary winding of the transformer, and power supplied to the outdoor unit. For blower coil and single-package systems, make all control circuit connections between components as specified by the manufacturer's installation instructions, and provide power and measure power supplied to all system components.

b. Configure Controls: Configure the controls of the central air conditioner or heat pump so that it operates as if connected to a building thermostat that is set to the OFF position. Use a compatible building thermostat if necessary to achieve this configuration. For a thermostat-controlled crankcase heater with a fixed power input, bypass the crankcase heater thermostat if necessary to energize the heater.

c. Measure $P2_x$: If the unit has a crankcase heater time delay, make sure that time-delay function is disabled or wait until delay time has passed. Determine the average power from non-zero value data measured over a 5-minute interval of the non-operating central air conditioner or heat pump and designate the average power as $P2_x$, the heating season total off mode power.

d. Measure P_x for coil-only split systems and for blower coil split systems for which a furnace or a modular blower is the designated air mover: Disconnect all low-voltage wiring for the *outdoor* components and *outdoor* controls from the low-voltage transformer. Determine the average power from non-zero value data measured over a 5-minute interval of the power supplied to the (remaining) low-voltage components of the central air conditioner or heat pump, or low-voltage power, P_x. This power measurement does not include line power supplied to the outdoor unit. It is the line power supplied to the air mover, or, if a compatible transformer is used instead of an air mover, it is the line power supplied to the transformer primary coil. If a compatible transformer is used instead of an air mover and power output of the low-voltage secondary circuit is measured, P_x is zero.

e. Calculate *P2:* Set the number of compressors equal to the unit's number of single-stage compressors plus 1.75 times the unit's number of compressors that are not single-stage.

For single-package systems and blower coil split systems for which the designated air mover is not a furnace or modular blower, divide the heating season total off mode power ($P2_x$) by the number of compressors to calculate *P2,* the heating season per-compressor off mode power. Round *P2* to the nearest watt. The expression for calculating *P2* is as follows:

$$P2 = \frac{P2_x}{number\ of\ compressors}.$$

For coil-only split systems and blower coil split systems for which a furnace or a modular blower is the designated air mover, subtract the low-voltage power (P_x) from the heating season total off mode power (P_x) and divide by the number of compressors to calculate *P2,* the heating season per-compressor off mode power. Round *P2* to the nearest watt. The expression for calculating *P2* is as follows:

$$P2 = \frac{P2_x - P_x}{number\ of\ compressors}.$$

f. Shoulder-season per-compressor off mode power, P1: If the system does not have a crankcase heater, has a crankcase heater without controls that is not self-regulating, or has a value for the crankcase heater turn-on temperature (as certified to DOE) that is higher than 71 °F, P1 is equal to P2.

Otherwise, de-energize the crankcase heater (by removing the thermostat bypass or otherwise disconnecting only the power supply to the crankcase heater) and repeat the measurement as described in section 3.13.1.c of this appendix. Designate the measured average power as $P1_x$, the shoulder season total off mode power.

Determine the number of compressors as described in section 3.13.1.e of this appendix.

For single-package systems and blower coil systems for which the designated air mover is not a furnace or modular blower, divide the shoulder season total off mode power ($P1_x$) by the number of compressors to calculate *P1,* the shoulder season per-compressor off mode power. Round *P1* to the nearest watt. The expression for calculating *P1* is as follows:

$$P1 = \frac{P1_x}{number\ of\ compressors}.$$

For coil-only split systems and blower coil split systems for which a furnace or a modular blower is the designated air mover, subtract the low-voltage power (P_x) from the shoulder season total off mode power ($P1_x$) and divide by the number of compressors to calculate *P1,* the shoulder season per-compressor off mode power. Round *P1* to the nearest watt. The expression for calculating *P1* is as follows:

$$P1 = \frac{P1_x - P_x}{number\ of\ compressors}.$$

3.13.2 This Test Determines the Off Mode Average Power Rating for Central Air Conditioners and Heat Pumps for Which Ambient Temperature Can Affect the Measurement of Crankcase Heater Power

a. Test Sample Set-up and Power Measurement: set up the test and measurement as described in section 3.13.1.a of this appendix.

b. Configure Controls: Position a temperature sensor to measure the outdoor dry-bulb temperature in the air between 2 and 6 inches from the crankcase heater control temperature sensor or, if no such temperature sensor exists, position it in the air between 2 and 6 inches from the crankcase heater. Utilize the temperature measurements from this sensor for this portion of the test procedure. Configure the controls of the central air conditioner or heat pump so that it operates as if connected to a building thermostat that is set to the OFF position. Use a compatible building thermostat if necessary to achieve this configuration.

Conduct the test after completion of the B, B_1, or B_2 test. Alternatively, start the test when the outdoor dry-bulb temperature is at 82 °F and the temperature of the compressor shell (or temperature of each compressor's shell if there is more than one compressor) is at least 81 °F. Then adjust the outdoor temperature and achieve an outdoor dry-bulb temperature of 72 °F. If the unit's compressor has no sound blanket, wait at least 4 hours after the outdoor temperature reaches 72 °F. Otherwise, wait at least 8 hours after the outdoor temperature reaches 72 °F. Maintain this temperature within ±2 °F while the compressor temperature equilibrates and while making the power measurement, as described in section 3.13.2.c of this appendix.

c. Measure $P1_x$: If the unit has a crankcase heater time delay, make sure that time-delay function is disabled or wait until delay time has passed. Determine the average power from non-zero value data measured over a 5-minute interval of the non-operating central air conditioner or heat pump and designate the average power as $P1_x$, the shoulder season total off mode power. For units with crankcase heaters which operate during this part of the test and whose controls cycle or vary crankcase heater power over time, the test period shall consist of three complete crankcase heater cycles or 18 hours, whichever comes first. Designate the average power over the test period as $P1_x$, the shoulder season total off mode power.

d. Reduce outdoor temperature: Approach the target outdoor dry-bulb temperature by adjusting the outdoor temperature. This target temperature is five degrees Fahrenheit less than the temperature certified by the manufacturer as the temperature at which the crankcase heater turns on. If the unit's compressor has no sound blanket, wait at least 4 hours after the outdoor temperature reaches the target temperature. Otherwise, wait at least 8 hours after the outdoor temperature reaches the target temperature. Maintain the target temperature within ±2 °F while the compressor temperature equilibrates and while making the power measurement, as described in section 3.13.2.e of this appendix.

e. Measure $P2_x$: If the unit has a crankcase heater time delay, make sure that time-delay function is disabled or wait until delay time has passed. Determine the average non-zero power of the non-operating central air conditioner or heat pump over a 5-minute interval and designate it as $P2_x$, the heating season total off mode power. For units with crankcase heaters whose controls cycle or vary crankcase heater power over time, the test period shall consist of three complete crankcase heater cycles or 18 hours, whichever comes first. Designate the average power over the test period as $P2_x$, the heating season total off mode power.

f. Measure P_x for coil-only split systems and for blower coil split systems for which a furnace or modular blower is the designated air mover: Disconnect all low-voltage wiring for the *outdoor* components and *outdoor* controls from the low-voltage transformer. Determine the average power from non-zero value data measured over a 5-minute interval of the power supplied to the (remaining) low-voltage components of the central air conditioner or heat pump, or low-voltage power, P_x. This power measurement does not include line power supplied to the outdoor unit. It is the line power supplied to the air mover, or, if a compatible transformer is used instead of an air mover, it is the line power supplied to the transformer primary coil. If a compatible transformer is used instead of an air mover and power output of the low-voltage secondary circuit is measured, P_x is zero.

g. Calculate *P1:*

Set the number of compressors equal to the unit's number of single-stage compressors plus 1.75 times the unit's number of compressors that are not single-stage.

For single-package systems and blower coil split systems for which the air mover is not a furnace or modular blower, divide the shoulder season total off mode power ($P1_x$) by the number of compressors to calculate *P1,* the shoulder season per-compressor off mode power. Round to the nearest watt. The expression for calculating *P1* is as follows:

$$P1 = \frac{P1_x}{number\ of\ compressors}.$$

For coil-only split systems and blower coil split systems for which a furnace or a modular blower is the designated air mover, subtract the low-voltage power (P_x) from the shoulder season total off mode power ($P1_x$) and divide by the number of compressors to calculate *P1,* the shoulder season per-compressor off mode power. Round to the nearest watt. The expression for calculating *P1* is as follows:

$$P1 = \frac{P1_x - P_x}{number\ of\ compressors}.$$

h. Calculate *P2:*

Determine the number of compressors as described in section 3.13.2.g of this appendix.

For, single-package systems and blower coil split systems for which the air mover is not a furnace, divide the heating season total off mode power ($P2_x$) by the number of compressors to calculate *P2,* the heating season per-compressor off mode power. Round to the nearest watt. The expression for calculating *P2* is as follows:

$$P2 = \frac{P2_x}{number\ of\ compressors}.$$

For coil-only split systems and blower coil split systems for which a furnace or a modular blower is the designated air mover, subtract the low-voltage power (P_x) from the heating season total off mode power ($P2_x$) and divide by the number of compressors to calculate *P2,* the heating season per-compressor off mode power. Round to the nearest watt. The expression for calculating *P2* is as follows:

$$P2 = \frac{P2_x - P_x}{number\ of\ compressors}.$$

4 Calculations of Seasonal Performance Descriptors

4.1 Seasonal Energy Efficiency Ratio (SEER2) Calculations

Calculate SEER2 as follows: For equipment covered under sections 4.1.2, 4.1.3, and 4.1.4 of this appendix, evaluate the seasonal energy efficiency ratio,

$$\text{Equation 4.1-1 } SEER2 = \frac{\sum_{j=1}^{8} q_c(T_j)}{\sum_{j=1}^{8} e_c(T_j)} = \frac{\sum_{j=1}^{8} \frac{q_c(T_j)}{N}}{\sum_{j=1}^{8} \frac{e_c(T_j)}{N}}$$

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 of this appendix, use a building cooling load, $BL(T_j)$. When referenced, evaluate $BL(T_j)$ for cooling using,

$$\text{Equation 4.1-2} \quad BL(T_j) = \frac{(T_j - 65)}{95 - 65} * \frac{\dot{Q}_c^{k=2}(95)}{1.1} * V$$

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 of this appendix, 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.

V is a factor equal to 0.93 for variable-speed heat pumps and otherwise equal to 1.0.

4.1.1 SEER2 Calculations for a Blower Coil System Having a Single-Speed Compressor and Either a Fixed-Speed Indoor Blower or a Constant-Air-Volume-Rate Indoor Blower, or a Single-Speed Coil-Only System Air Conditioner or Heat Pump

a. Evaluate the seasonal energy efficiency ratio, expressed in units of Btu/watt-hour, using:

$SEER2 = PLF(0.5) * EER_B$

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 of this appendix, 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 of this appendix regarding the definition and calculation of $\dot{Q}_c(82)$ and $\dot{E}_c(82)$. Evaluate the cooling mode cyclic degradation factor C_D^c as specified in section 3.5.3 of this appendix.

4.1.2 SEER2 Calculations for an Air Conditioner or Heat Pump Having a Single-Speed Compressor and a Variable-Speed Variable-Air-Volume-Rate Indoor Blower

4.1.2.1 Units Covered by Section 3.2.2.1 of This Appendix Where Indoor Blower Capacity Modulation Correlates With the Outdoor Dry Bulb Temperature

The manufacturer must provide information on how the indoor air volume rate or the indoor blower speed varies over the outdoor temperature range of 67 °F to 102 °F. Calculate SEER2 using Equation 4.1–1. Evaluate the quantity $q_c(T_j)/N$ in Equation 4.1–1 using,

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

where:

$$X(T_j) = \begin{Bmatrix} BL(T_j)/\dot{Q}_c(T_j) \\ or \\ 1 \end{Bmatrix}$$ 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 19. Use Equation 4.1–2 to calculate the building load, $BL(T_j)$. Evaluate $\dot{Q}_c(T_j)$ using,

Equation 4.1.2-2 $$\dot{Q}_c(T_j) = \dot{Q}_c^{k=1}(T_j) + \frac{\dot{Q}_c^{k=2}(T_j) - \dot{Q}_c^{k=1}(T_j)}{\mathrm{FP}_c^{k=2} - \mathrm{FP}_c^{k=1}} * [\mathrm{FP}_c(T_j) - \mathrm{FP}_c^{k=1}]$$

where:

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

the space cooling capacity of the test unit at outdoor temperature T_j if operated at the cooling minimum air volume rate, Btu/h.

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

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 blower 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 of this appendix), $FP_c^{k=2}$ 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 of this appendix 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, Equation 4.1.2–3

$$\frac{e_c(T_j)}{N} = \frac{X(T_j) * \dot{E}_c(T_j)}{PLF_j} * \frac{n_j}{N}$$

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. Evaluate the cooling mode cyclic degradation factor C_D^c as specified in section 3.5.3 of this appendix.

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

$$\dot{E}_c(T_j) = \dot{E}_c^{k=1}(T_j) + \frac{\dot{E}_c^{k=2}(T_j) - \dot{E}_c^{k=1}(T_j)}{\mathrm{FP}_c^{k=2} - \mathrm{FP}_c^{k=1}} * [\mathrm{FP}_c(T_j) - \mathrm{FP}_c^{k=1}]$$

where:

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

the electrical power consumption of the test unit at outdoor temperature T_j if operated at the cooling minimum air volume rate, W.

$\dot{E}_c^{k=2}(T_j) = \dot{E}_c^{k=2}(82) + \frac{\dot{E}_c^{k=2}(95) - \dot{E}_c^{k=2}(82)}{95-82} * (T_j - 82)$ 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 of this appendix 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 of This Appendix Where Indoor Blower Capacity Modulation Is Used to Adjust the Sensible to Total Cooling Capacity Ratio

Calculate SEER2 as specified in section 4.1.1 of this appendix.

4.1.3 SEER2 Calculations for an Air Conditioner or Heat Pump Having a Two-Capacity Compressor

Calculate SEER2 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,

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

Equation 4.1.3-2 $$\dot{E}_c^{k=1}(T_j) = \dot{E}_c^{k=1}(67) + \frac{\dot{E}_c^{k=1}(82) - \dot{E}_c^{k=1}(67)}{82-67} * (T_j - 67)$$

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 of this appendix. 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,

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

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

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 of this appendix.

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 of this appendix), cycle between low and high capacity (section 4.1.3.2 of this appendix), or operate at high capacity (sections 4.1.3.3 and 4.1.3.4 of this appendix) in responding to the building load. For units that lock out low capacity operation at higher outdoor temperatures, the outdoor temperature at which the unit locks out must be that specified by the manufacturer in the certification report 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, $\dot{Q}_c^{k=1}(T_j) \geq BL(T_j)$

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

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.

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 19. Use Equations 4.1.3–1 and 4.1.3–2, respectively, to evaluate $\dot{Q}_c^{k=1}(T_j)$ and $\dot{E}_c^{k=1}(T_j)$. Evaluate the cooling mode cyclic degradation factor C_D^c as specified in section 3.5.3 of this appendix.

TABLE 19—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

TABLE 19—DISTRIBUTION OF FRACTIONAL HOURS WITHIN COOLING SEASON TEMPERATURE BINS—Continued

Bin number, j	Bin temperature range °F	Representative temperature for bin °F	Fraction of of total temperature bin hours, n_j/N
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) <($\dot{Q}_c^{k=2}(T_j)$

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

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

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 19. 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.

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

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.

$PLF_j = 1 - C_D^c(k=2) * [1 - X^{k=2}(T_j)]$, the part load factor, dimensionless.

Obtain the fractional bin hours for the cooling season, n_j/N, from Table 19. 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 C_2 and D_2 tests described in section 3.2.3 and Table 7 of this appendix are not conducted, set C_D^c (k=2) equal to the default value specified in section 3.5.3 of this appendix.

4.1.3.4 Unit Must Operate Continuously at High (k=2) Compressor Capacity at Temperature T_j, BL(T_j) ≥$\dot{Q}_c^{k=2}(T_j)$

$$\frac{q_c(T_j)}{N} = \dot{Q}_c^{k=2}(T_j) * \frac{n_j}{N} \qquad \frac{e_c(T_j)}{N} = \dot{E}_c^{k=2}(T_j) * \frac{n_j}{N}$$

Obtain the fractional bin hours for the cooling season, n_j/N, from Table 19. 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 SEER2 Calculations for an Air Conditioner or Heat Pump Having a Variable-Speed Compressor

Calculate SEER2 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, Equation 4.1.4-1 $$\dot{Q}_c^{k=1}(T_j) = \dot{Q}_c^{k=1}(67) + \frac{\dot{Q}_c^{k=1}(82) - \dot{Q}_c^{k=1}(67)}{82-67} * (T_j - 67)$$

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

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 of this appendix. 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 full 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 of this appendix. 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 8) E_V test of this appendix using,

Equation 4.1.4-3 $$\dot{Q}_c^{k=v}(T_j) = \dot{Q}_c^{k=v}(87) + M_Q * (T_j - 87)$$

Equation 4.1.4-4 $$\dot{E}_c^{k=v}(T_j) = \dot{E}_h^{k=v}(87) + M_E * (T_j - 87)$$

where $\dot{Q}_c^{k=v}(87)$ and $\dot{E}_c^{k=v}(87)$ are determined from the E_V test and calculated as specified in section 3.3 of this appendix. Approximate the slopes of the k=v intermediate speed cooling capacity and electrical power input curves, M_Q and M_E, as follows:

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

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

where,

$$N_Q = \frac{\dot{Q}_c^{k=v}(87) - \dot{Q}_c^{k=1}(87)}{\dot{Q}_c^{k=2}(87) - \dot{Q}_c^{k=1}(87)} \qquad N_E = \frac{\dot{E}_c^{k=v}(87) - \dot{E}_c^{k=1}(87)}{\dot{E}_c^{k=2}(87) - \dot{E}_c^{k=1}(87)}$$

Use Equations 4.1.4–1 and 4.1.4–2, respectively, to calculate $\dot{Q}_c^{k=1}(87)$ and $\dot{E}_c^{k=1}(87)$.

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, $\dot{Q}_c^{k=1}(T_j) \geq BL(T_j)$.

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

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 19. Use Equations 4.1.3–1 and 4.1.3–2, respectively, to evaluate $\dot{Q}_c^{k=1}(T_j)$ and $\dot{E}_c^{k=1}(T_j)$. Evaluate the cooling mode cyclic degradation factor C_D^c as specified in section 3.5.3 of this appendix.

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, $\dot{Q}_c^{k=1}(T_j) < BL(T_j) < \dot{Q}_c^{k=2}(T_j)$.

$$\frac{q_c(T_j)}{N} = \dot{Q}_c^{k=i}(T_j) * \frac{n_j}{N} \qquad \frac{e_c(T_j)}{N} = \dot{E}_c^{k=i}(T_j) * \frac{n_j}{N}$$

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 19 of this section. For each temperature bin where the unit operates at an intermediate compressor speed, determine the energy efficiency ratio $EER^{k=i}(T_j)$ using the following equations,

For each temperature bin where $\dot{Q}_c^{k=1}(T_j) < BL(T_j) < \dot{Q}_c^{k=v}(T_j)$,

$$EER^{k=i}(T_j) = EER^{k=1}(T_j) + \frac{EER^{k=v}(T_j) - EER^{k=1}(T_j)}{Q^{k=v}(T_j) - Q^{k=1}(T_j)} * (BL(T_j) - Q^{k=1}(T_j))$$

For each temperature bin where $\dot{Q}_c^{k=v}(T_j) \leq BL(T_j) < \dot{Q}_c^{k=2}(T_j)$,

$$EER^{k=i}(T_j) = EER^{k=v}(T_j) + \frac{EER^{k=2}(T_j) - EER^{k=v}(T_j)}{Q^{k=2}(T_j) - Q^{k=v}(T_j)} * (BL(T_j) - Q^{k=v}(T_j))$$

Where:

$EER^{k=1}(T_j)$ is the steady-state energy efficiency ratio of the test unit when operating at minimum compressor speed and temperature Tj, Btu/h per W, calculated using capacity $\dot{Q}_c^{k=1}(T_j)$ calculated using Equation 4.1.4–1 and electrical power consumption $\dot{E}_c^{k=1}(T_j)$ calculated using Equation 4.1.4–2;

$EER^{k=v}(T_j)$ is the steady-state energy efficiency ratio of the test unit when operating at intermediate compressor speed and temperature Tj, Btu/h per W, calculated using capacity $\dot{Q}_c^{k=v}(T_j)$ calculated using Equation 4.1.4–3 and electrical power consumption $\dot{E}_c^{k=v}(T_j)$ calculated using Equation 4.1.4–4;

$EER2^{k=2}(T_j)$ is the steady-state energy efficiency ratio of the test unit when operating at full compressor speed and temperature Tj, Btu/h per W, calculated using capacity $\dot{Q}_c^{k=2}(T_j)$ and electrical power consumption $\dot{E}_c^{k=2}(T_j)$, both calculated as described in section 4.1.4; and

$BL(T_j)$ is the building cooling load at temperature T_j, Btu/h.

4.1.4.3 Unit must operate continuously at full (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 of this appendix with the understanding that $\dot{Q}_c^{k=2}(T_j)$ and $\dot{E}_c^{k=2}(T_j)$ correspond to full compressor speed operation and are derived from the results of the tests specified in section 3.2.4 of this appendix.

4.1.5 SEER2 Calculations for an Air Conditioner or Heat Pump Having a Single Indoor Unit With Multiple Indoor Blowers

Calculate SEER2 using Eq. 4.1–1, where $q_c(Tj)/N$ and $e_c(Tj)/N$ are evaluated as specified in the applicable subsection.

4.1.5.1 For Multiple Indoor Blower Systems That Are Connected to a Single, Single-Speed Outdoor Unit

a. Calculate 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 the cooling minimum air volume rate and outdoor temperature T_j using the equations given in section 4.1.2.1 of this appendix. Calculate 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 the cooling full-load air volume rate and outdoor temperature T_j using the equations given in section 4.1.2.1 of this appendix. In evaluating the section 4.1.2.1 equations, determine the quantities $\dot{Q}_c^{k=1}(82)$ and $\dot{E}_c^{k=1}(82)$ from the B1 test, $\dot{Q}_c^{k=1}(95)$ and $\dot{E}_c^{k=1}(95)$ from the Al test, $\dot{Q}_c^{k=2}(82)$ and $\dot{E}_c^{k=2}(82)$ from the B2 test, and $\dot{Q}_c^{k=2}(95)$ and $\dot{E}_c^{k=2}(95)$ from the A_2 test. Evaluate all eight quantities as specified in section 3.3. Refer to section 3.2.2.1 and Table 6 for additional information on the four referenced laboratory tests.

b. Determine the cooling mode cyclic degradation coefficient, C_D^c, as per sections 3.2.2.1 and 3.5 to 3.5.3 of this appendix. Assign this same value to $C_D^c(K=2)$.

c. Except for using the above values of $\dot{Q}_c^{k=1}(T_j)$, $\dot{E}_c^{k=1}(T_j)$, $\dot{E}_c^{k=2}(T_j)$, $\dot{Q}_c^{k=2}(T_j)$, C_D^c, and C_D^c (K=2), calculate the quantities $q_c(T_j)/N$ and $e_c(T_j)/N$ as specified in section 4.1.3.1 of this appendix for cases where $\dot{Q}_c^{k=1}(T_j) \geq BL(T_j)$. For all other outdoor bin temperatures, T_j, calculate $q_c(Tj)/N$ and $e_c(Tj)/N$ as specified in section 4.1.3.3 of this appendix if $\dot{Q}_c^{k=2}(T_j) > BL(T_j)$ or as specified in section 4.1.3.4 of this appendix if $\dot{Q}_c^{k=2}(T_j) \leq BL(T_j)$.

4.1.5.2 For Multiple Indoor Blower Systems That Are Connected to Either a Lone Outdoor Unit Having a Two-Capacity Compressor or Two Separate But Identical Model Single-Speed Outdoor Units. Calculate the Quantities $q_c(Tj)/N$ and $e_c(Tj)/N$ as Specified in Section 4.1.3 of This Appendix

4.2 Heating Seasonal Performance Factor 2 (HSPF2) Calculations

Unless an approved alternative efficiency determination method is used, as set forth in 10 CFR 429.70(e). Calculate HSPF2 as follows: Six generalized climatic regions are depicted in Figure 1 and otherwise defined in Table 20. For each of these regions and for each applicable standardized design heating requirement, evaluate the heating seasonal performance factor using,

Equation 4.2-1
$$HSPF2 = \frac{\sum_j^J n_j * BL(T_j)}{\sum_j^J e_h(T_j) + \sum_j^J RH(T_j)} * F_{def} = \frac{\sum_j^J \left[\frac{n_j}{N} * BL(T_j)\right]}{\sum_j^J \frac{e_h(T_j)}{N} + \sum_j^J \frac{RH(T_j)}{N}} * F_{def}$$

Where:

$e_h(T_j)/N$ = The ratio of the electrical energy consumed by the heat pump during periods of 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 (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 of this appendix, 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 section 4.2.5 of this appendix).

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 20.

j = the bin number, dimensionless.

J = for each generalized climatic region, the total number of temperature bins, dimensionless. Referring to Table 20, 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 of this appendix, 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 20—GENERALIZED CLIMATIC REGION INFORMATION

Region Number	I	II	III	IV	V	* VI
Heating Load Hours, HLH	493	857	1247	1701	2202	1842
Outdoor Design Temperature, T_{OD}	37	27	17	5	−10	30
Heating Load Line Equation Slope Factor, C	1.10	1.06	1.30	1.15	1.16	1.11
Variable-speed Slope Factor, C_{VS}	1.03	0.99	1.21	1.07	1.08	1.03
Zero-Load Temperature, T_{zl}	58	57	56	55	55	57
j T_j (°F)	Fractional Bin Hours, n_j/N					
1 62	0	0	0	0	0	0
2 57	.239	0	0	0	0	0
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

$$\text{Equation 4.2-2} \qquad BL(T_j) = \frac{(T_{zl}-T_j)}{T_{zl}-5°F} * C * \dot{Q}_c(95°F)$$

where,

T_j = the outdoor bin temperature, °F

T_{zl} = the zero-load temperature, °F, which varies by climate region according to Table 20

C = the slope (adjustment) factor, which varies by climate region according to Table 20

$\dot{Q}_c(95°F)$ = the cooling capacity at 95 °F determined from the A or A_2 test, Btu/h

For heating-only heat pump units, replace $\dot{Q}_c(95°F)$ in Equation 4.2–2 with $\dot{Q}_h(47°F)$

$\dot{Q}_h(47°F)$= the heating capacity at 47 °F determined from the H, $H1_2$ or $H1_N$ test, Btu/h.

a. For all heat pumps, HSPF2 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 HSPF2 calculations for all heat pumps, see either section 4.2.1, 4.2.2, 4.2.3, or 4.2.4 of this appendix, whichever applies.

b. For heat pumps with heat comfort controllers (see section 1.2 of this appendix, Definitions), HSPF2 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 of this appendix for the additional steps required for calculating the HSPF2.

4.2.1 Additional Steps for Calculating the HSPF2 of a Blower Coil System Heat Pump Having a Single-Speed Compressor and Either a Fixed-Speed Indoor Blower or a Constant-Air-Volume-Rate Indoor Blower, or a Single-Speed Coil-Only System Heat Pump

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

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

Where:

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

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 20. Evaluate the heating mode cyclic degradation factor C_D^h as specified in section 3.8.1 of this appendix.

Determine the low temperature cut-out factor using

Equation 4.2.1-3 $$\delta(T_j) = \begin{Bmatrix} 0, if\ T_j \le T_{off}\ or\ \frac{\dot{Q}_h(T_j)}{3.413*\dot{E}_h(T_j)} < 1 \\ 1/2, if\ T_{off} < T_j \le T_{on}\ and\ \frac{\dot{Q}_h(T_j)}{3.413*\dot{E}_h(T_j)} \ge 1 \\ 1, if\ T_j > T_{on}\ and\ \frac{\dot{Q}_h(T_j)}{3.413*\dot{E}_h(T_j)} \ge 1 \end{Bmatrix}$$

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.

If the H4 test is not conducted, calculate $Q_h(T_j)$ and $E_h(T_j)$ using

Equation 4.2.1-4 $$\dot{Q}_h(T_j) = \begin{cases} \dot{Q}_h(17) + \frac{[\dot{Q}_h(47)-\dot{Q}_h(17)]*(T_j-17)}{47-17}, & if\ T_j \ge 45\ °F\ or\ T_j \le 17\ °F \\ \dot{Q}_h(17) + \frac{[\dot{Q}_h(35)-\dot{Q}_h(17)]*(T_j-17)}{35-17}, & if\ 17\ °F < T_j < 45\ °F \end{cases}$$

Equation 4.2.1-5

$$\dot{E}_h(T_j) = \begin{cases} \dot{E}_h(17) + \frac{[\dot{E}_h(47)-\dot{E}_h(17)]*(T_j-17)}{47-17}, & if\ T_j \ge 45\ °F\ or\ T_j \le 17\ °F \\ \dot{E}_h(17) + \frac{[\dot{E}_h(35)-\dot{E}_h(17)]*(T_j-17)}{35-17}, & if\ 17\ °F < T_j < 45\ °F \end{cases}$$

where $\dot{Q}_h(47)$ and $\dot{E}_h(47)$ are determined from the H1 test and calculated as specified in section 3.7 of this appendix; $\dot{Q}_h(35)$ and $\dot{E}_h(35)$ are determined from the H2 test and calculated as specified in section 3.9.1 of this appendix; and $\dot{Q}_h(17)$ and

$\dot{E}_h(17)$ are determined from the H3 test and calculated as specified in section 3.10 of this appendix.

If the H4 test is conducted, calculate $\dot{Q}_h(T_j)$ and $\dot{E}_h(T_j)$ using

Equation 4.2.1-6

$$\dot{Q}_h\left(T_j\right) = \begin{cases} \dot{Q}_h(17) + \dfrac{\left[\dot{Q}_h(47) - \dot{Q}_h(17)\right] * (T_j - 17)}{47 - 17}, & if\ T_j \geq 45\ °\mathrm{F} \\ \dot{Q}_h(17) + \dfrac{\left[\dot{Q}_h(35) - \dot{Q}_h(17)\right] * (T_j - 17)}{35 - 17}, & if\ 17\ °\mathrm{F} \leq T_j < 45\ °\mathrm{F} \\ \dot{Q}_h(5) + \dfrac{\left[\dot{Q}_h(17) - \dot{Q}_h(5)\right] * (T_j - 5)}{17 - 5}, & if\ T_j < 17\ °\mathrm{F} \end{cases}$$

Equation 4.2.1-7

$$\dot{E}_h\left(T_j\right) = \begin{cases} \dot{E}_h(17) + \dfrac{\left[\dot{E}_h(47) - \dot{E}_h(17)\right] * (T_j - 17)}{47 - 17}, & if\ T_j \geq 45\ °\mathrm{F} \\ \dot{E}_h(17) + \dfrac{\left[\dot{E}_h(35) - \dot{E}_h(17)\right] * (T_j - 17)}{35 - 17}, & if\ 17\ °\mathrm{F} \leq T_j < 45\ °\mathrm{F} \\ \dot{E}_h(5) + \dfrac{\left[\dot{E}_h(17) - \dot{E}_h(5)\right] * (T_j - 5)}{17 - 5}, & if\ T_j < 17\ °\mathrm{F} \end{cases}$$

where $\dot{Q}_h(47)$ and $\dot{E}_h(47)$ are determined from the H1 test and calculated as specified in section 3.7 of this appendix; $\dot{Q}_h(35)$ and $\dot{E}_h(35)$ are determined from the H2 test and calculated as specified in section 3.9.1 of this appendix; $\dot{Q}_h(17)$ and $\dot{E}_h(17)$ are determined from the H3 test and calculated as specified in section 3.10 of this appendix; $\dot{Q}_h(5)$ and $\dot{E}_h(5)$ are determined from the H4 test and calculated as specified in section 3.10 of this appendix.

4.2.2 Additional Steps for Calculating the HSPF2 of a Heat Pump Having a Single-Speed Compressor and a Variable-Speed, Variable-Air-Volume-Rate Indoor Blower

The manufacturer must provide information about how the indoor air volume rate or the indoor blower 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 of this appendix with the exception of replacing references to the H1C test and section 3.6.1 of this appendix with the H1C$_1$ test and section 3.6.2 of this appendix. 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

Equation 4.2.2-1 $$\dot{Q}_h\left(T_j\right) = \dot{Q}_h^{k=1}\left(T_j\right) + \frac{\dot{Q}_h^{k=2}(T_j) - \dot{Q}_h^{k=1}(T_j)}{\mathrm{FP}_h^{k=2} - \mathrm{FP}_h^{k=1}} * \left[\mathrm{FP}_h\left(T_j\right) - \mathrm{FP}_h^{k=1}\right]$$

Equation 4.2.2-2 $$\dot{E}_h\left(T_j\right) = \dot{E}_h^{k=1}\left(T_j\right) + \frac{\dot{E}_h^{k=2}(T_j) - \dot{E}_h^{k=1}(T_j)}{\mathrm{FP}_h^{k=2} - \mathrm{FP}_h^{k=1}} * \left[\mathrm{FP}_h\left(T_j\right) - \mathrm{FP}_h^{k=1}\right]$$

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

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

Equation 4.2.2-4

$$\dot{E}_h^k(T_j) = \begin{cases} \dot{E}_h^k(17) + \frac{\left[\dot{E}_h^k(47) - \dot{E}_h^k(17)\right] * (T_j - 17)}{47-17}, & if\ T_j \geq 45\ °\text{F}\ or\ T_j \leq 17\ °\text{F} \\ \dot{E}_h^k(17) + \frac{\left[\dot{E}_h^k(35) - \dot{E}_h^k(17)\right] * (T_j - 17)}{35-17}, & if\ 17\ °\text{F} < T_j < 45\ °\text{F} \end{cases}$$

If the $H4_2$ test is not conducted, calculate the space heating capacity and electrical power consumption at high capacity (k=2) at outdoor temperature Tj using Equations 4.2.2–3 and 4.2.2–4 for k=2.

If the $H4_2$ test is conducted, calculate the space heating capacity and electrical power consumption at high capacity (k=2) at outdoor temperature Tj using Equations 4.2.2–5 and 4.2.2–6.

Equation 4.2.2-5

$$\dot{Q}_h^{k=2}(T_j) = \begin{cases} \dot{Q}_h^{k=2}(17) + \frac{\left[\dot{Q}_h^{k=2}(47) - \dot{Q}_h^{k=2}(17)\right] * (T_j - 17)}{47-17}, & if\ T_j \geq 45\ °\text{F} \\ \dot{Q}_h^{k=2}(17) + \frac{\left[\dot{Q}_h^{k=2}(35) - \dot{Q}_h^{k=2}(17)\right] * (T_j - 17)}{35-17}, & if\ 17\ °\text{F} \leq T_j < 45\ °\text{F} \\ \dot{Q}_h^{k=2}(5) + \frac{\left[\dot{Q}_h^{k=2}(17) - \dot{Q}_h^{k=2}(5)\right] * (T_j - 5)}{17-5}, & if\ T_j < 17\ °\text{F} \end{cases}$$

Equation 4.2.2-6

$$\dot{E}_h^{k=2}(T_j) = \begin{cases} \dot{E}_h^{k=2}(17) + \frac{\left[\dot{E}_h^{k=2}(47) - \dot{E}_h^{k=2}(17)\right] * (T_j - 17)}{47-17}, & if\ T_j \geq 45\ °\text{F} \\ \dot{E}_h^{k=2}(17) + \frac{\left[\dot{E}_h^{k=2}(35) - \dot{E}_h^{k=2}(17)\right] * (T_j - 17)}{35-17}, & if\ 17\ °\text{F} \leq T_j < 45\ °\text{F} \\ \dot{E}_h^{k=2}(5) + \frac{\left[\dot{E}_h^{k=2}(17) - \dot{E}_h^{k=2}(5)\right] * (T_j - 5)}{17-5}, & if\ T_j < 17\ °\text{F} \end{cases}$$

For units where indoor blower 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 12), $FP_h^{k=2}$ 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 of this appendix. Determine $\dot{Q}_h^{k=1}(35)$ and $\dot{E}_h^{k=1}(35)$ as specified in section 3.6.2 of this appendix; 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 of this appendix. 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 of this appendix. Determine $\dot{Q}_h^{k=2}(5)$ and $\dot{E}_h^{k=2}(5)$ from the $H4_2$ test and the calculation specified in section 3.10 of this appendix.

4.2.3 Additional Steps for Calculating the HSPF2 of a Heat Pump Having a Two-Capacity Compressor

The calculation of the Equation 4.2–1 quantities differ depending upon whether the heat pump would operate at low capacity (section 4.2.3.1 of this appendix), cycle between low and high capacity (section 4.2.3.2 of this appendix), or operate at high capacity (sections 4.2.3.3 and 4.2.3.4 of this appendix) in responding to the building load. For heat pumps that lock out low capacity operation at low outdoor temperatures, the outdoor temperature at which the unit locks out must be that specified by the manufacturer in the certification report so that the appropriate equations can be selected.

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

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

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

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

b. If the $H4_2$ test is not conducted, 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. If the $H4_2$ test is conducted, 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 using Equations 4.2.2–5 and 4.2.2–6, respectively.

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 of this appendix. 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 of this appendix, 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 in this appendix. 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 of this appendix, 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 of this appendix. Determine $\dot{Q}_h^{k=2}(5)$ and $\dot{E}_h^{k=2}(5)$ from the $H4_2$ test and the calculation specified in section 3.10 of this appendix.

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, $\dot{Q}_h^{k=1}(T_j) \geq BL(T_j)$

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

$$\text{Equation 4.2.3-2} \quad \frac{RH(T_j)}{N} = \frac{BL(T_j) * [1 - \delta(T_j)]}{3.413 \frac{Btu/h}{W}} * \frac{n_j}{N}$$

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.

Evaluate the heating mode cyclic degradation factor C_D^h as specified in section 3.8.1 of this appendix.

Determine the low temperature cut-out factor using

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

where T_{off} and T_{on} are defined in section 4.2.1 of this appendix. Use the calculations given in section 4.2.3.3 of this appendix, 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

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

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

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

where:

$X^{k=2}(T_j) = BL(T_j)/\dot{Q}_h^{k=2}(T_j)$. $PLF_j = 1 - C^h_D(k = 2) * [1 - X^{k=2}(T_j)]$

If the $H1C_2$ test described in section 3.6.3 and Table 13 of this appendix is not conducted, set $C_D{}^h$ (k=2) equal to the default value specified in section 3.8.1 of this appendix.

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, $BL(T_j) \geq \dot{Q}_h^{k=2}(T_j)$

$$\frac{e_h(T_j)}{N} = \dot{E}_h^{k=2}(T_j) * \delta'(T_j) * \frac{n_j}{N} \qquad \frac{RH(T_j)}{N} = \frac{BL(T_j) * [\dot{Q}_h^{k=2}(T_j) * \delta'(T_j)]}{3.413 \frac{Btu/h}{W}} * \frac{n_j}{N}$$

where:

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

4.2.4 Additional Steps for Calculating the HSPF2 of a Heat Pump Having a Variable-Speed Compressor. Calculate HSPF2 Using Equation 4.2–1

The calculation of Equation 4.2-1 quantities $\frac{RH(T_j)}{N}$ and $\frac{e_h(T_j)}{N}$ differs depending upon whether the heat pump would operate at minimum speed (section 4.2.4.1 of this appendix), operate at an intermediate speed (section 4.2.4.2 of this appendix), or operate at full speed (section 4.2.4.3 of this appendix) in responding to the building load.

a. Minimum Compressor Speed. 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

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

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

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 of this appendix.

b. Minimum Compressor Speed for Minimum-speed-limiting Variable-speed Heat Pumps: 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 Equation 4.2.4–3

Equation 4.2.4-3

$$\dot{Q}_h^{k=1}(T_j) = \begin{cases} \dot{Q}_h^{k=1}(47) + \frac{[\dot{Q}_h^{k=1}(62) - \dot{Q}_h^{k=1}(47)] * (T_j - 47)}{62 - 47}, & if\ T_j \geq 47\ °\text{F} \\ \dot{Q}_h^{k=v}(35) + \frac{[\dot{Q}_h^{k=1}(47) - \dot{Q}_h^{k=v}(35)] * (T_j - 35)}{47 - 35}, & if\ 35\ °\text{F} \leq T_j < 47\ °\text{F} \\ \dot{Q}_h^{k=v}(T_j), & if\ T_j < 35\ °\text{F} \end{cases}$$

Equation 4.2.4-4

$$\dot{E}_h^{k=1}(T_j) = \begin{cases} \dot{E}_h^{k=1}(47) + \frac{[\dot{E}_h^{k=1}(62) - \dot{E}_h^{k=1}(47)] * (T_j - 47)}{62 - 47}, & if\ T_j \geq 47\ °\text{F} \\ \dot{E}_h^{k=v}(35) + \frac{[\dot{E}_h^{k=1}(47) - \dot{E}_h^{k=v}(35)] * (T_j - 35)}{47 - 35}, & if\ 35\ °\text{F} \leq T_j < 47\ °\text{F} \\ \dot{E}_h^{k=v}(T_j), & if\ T_j < 35\ °\text{F} \end{cases}$$

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 of this appendix; $\dot{Q}_h^{k=v}(35)$ and $\dot{E}_h^{k=v}(35)$ are determined from the $H2_v$ test and are calculated as specified in section 3.9 of this appendix; and $\dot{Q}_h^{k=v}(T_j)$ and $\dot{E}_h^{k=v}(T_j)$ are calculated using equations 4.2.4–5 and 4.2.4–6, respectively.

c. Full Compressor Speed for Heat Pumps for which the $H4_2$ test is not Conducted. 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 full compressor speed and outdoor temperature T_j by solving Equations 4.2.2–3 and 4.2.2–4, respectively, for k=2, using $\dot{Q}_{hcalc}^{k=2}(47)$ to represent $\dot{Q}_h^{k=2}(47)$ and $\dot{E}_{hcalc}^{k=2}(47)$ to represent $\dot{E}_h^{k=2}(47)$ (see section 3.6.4.b of this appendix regarding determination of the capacity and power input used in the HSPF2 calculations to represent the $H1_2$ Test). 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 methods specified in section 3.10 of this appendix.

d. Full Compressor Speed for Heat Pumps for which the $H4_2$ test is Conducted. For T_j above 17 °F, 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 full compressor speed as described above for heat pumps for which the $H4_2$ is not conducted. For T_j between 5 °F and 17 °F, 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 full compressor speed using the following equations:

$$\dot{Q}_h^{k=2}(T_j) = \dot{Q}_h^{k=2}(5) + \frac{\dot{Q}_h^{k=2}(17) - \dot{Q}_h^{k=2}(5)}{17-5} * (T_j - 5)$$

$$\dot{E}_h^{k=2}(T_j) = \dot{E}_h^{k=2}(5) + \frac{\dot{E}_h^{k=2}(17) - \dot{E}_h^{k=2}(5)}{17-5} * (T_j - 5)$$

Determine $\dot{Q}_h^{k=2}(17)$ and $\dot{E}_h^{k=2}(17)$ from the $H3_2$ test, and $\dot{Q}_h^{k=2}(5)$ and $\dot{E}_h^{k=2}(5)$ from the $H4_2$ test, using the methods specified in section 3.10 of this appendix for all four values. For T_j below 5 °F, 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 full compressor speed using the following equations:

$$\dot{Q}_h^{k=2}(T_j) = \dot{Q}_h^{k=2}(5) - \frac{\dot{Q}_{hcalc}^{k=2}(47) - \dot{Q}_h^{k=2}(17)}{47-17} * (5 - T_j)$$

$$\dot{E}_h^{k=2}(T_j) = \dot{E}_h^{k=2}(5) - \frac{\dot{E}_{hcalc}^{k=2}(47) - \dot{E}_h^{k=2}(17)}{47-17} * (5 - T_j)$$

Determine $\dot{Q}_{hcalc}^{k=2}(47)$ and $\dot{E}_{hcalc}^{k=2}(47)$ as described in section 3.6.4.b of this appendix. Determine $\dot{Q}_h^{k=2}(17)$ and $\dot{E}_h^{k=2}(17)$ from the $H3_2$ test, using the methods specified in section 3.10 of this appendix.

e. Intermediate Compressor Speed. 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

$$\text{Equation 4.2.4-5} \quad \dot{Q}_h^{k=v}(T_j) = \dot{Q}_h^{k=v}(35) + M_Q * (T_j - 35)$$

$$\text{Equation 4.2.4-6} \quad \dot{E}_h^{k=v}(T_j) = \dot{E}_h^{k=v}(35) + M_E * (T_j - 35)$$

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 of this appendix. Approximate the slopes of the k=v intermediate speed heating capacity and electrical power input curves, M_Q and M_E, as follows:

$$M_Q = \left[\frac{\dot{Q}_h^{k=1}(62) - \dot{Q}_h^{k=1}(47)}{62-47} * (1 - N_Q)\right] + \left[N_Q * \frac{\dot{Q}_h^{k=2}(35) - \dot{Q}_h^{k=2}(17)}{35-17}\right]$$

$$M_E = \left[\frac{\dot{E}_h^{k=1}(62) - \dot{E}_h^{k=1}(47)}{62-47} * (1 - N_E)\right] + \left[N_E * \frac{\dot{E}_h^{k=2}(35) - \dot{E}_h^{k=2}(17)}{35-17}\right]$$

where,

$$N_Q = \frac{\dot{Q}_h^{k=v}(35) - \dot{Q}_h^{k=1}(35)}{\dot{Q}_h^{k=2}(35) - \dot{Q}_h^{k=1}(35)} \qquad N_E = \frac{\dot{E}_h^{k=v}(35) - \dot{E}_h^{k=1}(35)}{\dot{E}_h^{k=2}(35) - \dot{E}_h^{k=1}(35)}$$

Use Equations 4.2.4–1 and 4.2.4–2, respectively, to calculate $\dot{Q}_h^{k=1}(35)$ and $\dot{E}_h^{k=1}(35)$, whether or not the heat pump is a minimum-speed-limiting variable-speed heat pump.

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

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

as specified in section 4.2.3.1 of this appendix. Except now use Equations 4.2.4–1 and 4.2.4–2 (for heat pumps that are not minimum-speed-limiting) or Equations 4.3.4–3 and 4.2.4–4 (for minimum-speed-limiting variable-speed heat pumps) 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 of this appendix with ''minimum speed'' and section 3.6.4 of this appendix. Also, the last sentence of section 4.2.3.1 of this appendix 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,

$$\frac{e_h(T_j)}{N} = \dot{E}_h^{k=i}(T_j) * \delta(T_j) * \frac{n_j}{N}$$

where:

$$\dot{E}_h^{k=i}(T_j) = \frac{\dot{Q}_h^{k=i}(T_j)}{3.413\frac{Btu/h}{W} * COP^{k=i}(T_j)}$$

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 the following equations,

For each temperature bin where $\dot{Q}_h^{k=1}(T_j) < BL(T_j) < \dot{Q}_h^{k=v}(T_j)$,

$$COP_h^{k=i}(T_j) = COP_h^{k=1}(T_j) + \frac{COP_h^{k=v}(T_j) - COP_h^{k=1}(T_j)}{Q_h^{k=v}(T_j) - Q_h^{k=1}(T_j)} * (BL(T_j) - Q_h^{k=1}(T_j))$$

For each temperature bin where $\dot{Q}_h^{k=v}(T_j) \leq BL(T_j) < \dot{Q}_h^{k=2}(T_j)$,

$$COP_h^{k=i}(T_j) = COP_h^{k=v}(T_j) + \frac{COP_h^{k=2}(T_j) - COP_h^{k=v}(T_j)}{Q_h^{k=2}(T_j) - Q_h^{k=v}(T_j)} * (BL(T_j) - Q_h^{k=v}(T_j))$$

Where:

$COP_h^{k=1}(T_j)$ is the steady-state coefficient of performance of the heat pump when operating at minimum compressor speed and temperature Tj, dimensionless, calculated using capacity $\dot{Q}_h^{k=1}(T_j)$ calculated using Equation 4.2.4–1 or 4.2.4–3 and electrical power consumption $\dot{E}_h^{k=1}(T_j)$ calculated using Equation 4.2.4–2 or 4.2.4–4;

$COP_h^{k=v}(T_j)$ is the steady-state coefficient of performance of the heat pump when operating at intermediate compressor speed and temperature Tj, dimensionless, calculated using capacity $\dot{Q}_h^{k=v}(T_j)$ calculated using Equation 4.2.4–5 and electrical power consumption $\dot{E}_h^{k=v}(T_j)$ calculated using Equation 4.2.4–6;

$COP_h^{k=2}(T_j)$ is the steady-state coefficient of performance of the heat pump when operating at full compressor speed and temperature Tj, dimensionless, calculated using capacity $\dot{Q}_h^{k=2}(T_j)$ and electrical power consumption $\dot{E}_h^{k=2}(T_j)$, both calculated as described in section 4.2.4; and

$BL(T_j)$ is the building heating load at temperature T_j, Btu/h.

4.2.4.3 Heat Pump Must Operate Continuously at Full (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

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

as specified in section 4.2.3.4 of this appendix with the understanding that $\dot{Q}_h^{k=2}(T_j)$ and $\dot{E}_h^{k=2}(T_j)$ correspond to full compressor speed operation and are derived from the results of the specified section 3.6.4 tests of this appendix.

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 Blower Coil System Heat Pump Having a Heat Comfort Controller: Additional Steps for Calculating the HSPF2 of a Heat Pump Having a Single-Speed Compressor and Either a Fixed-Speed Indoor Blower or a Constant-Air-Volume-Rate Indoor Blower Installed, or a Single-Speed Coil-Only System Heat Pump

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 of this appendix (Equations 4.2.1–4 and 4.2.1–5) for each outdoor bin temperature, T_j, that is listed in Table 20. 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:

$$\dot{m}_{da} = \overline{\dot{V}_s} * 0.075 \frac{lbm_{da}}{ft^3} * \frac{60_{min}}{hr} = \frac{\overline{\dot{V}}_{mx}}{v_n' * [1 + W_n]} * \frac{60_{min}}{hr} = \frac{\overline{\dot{V}}_{mx}}{v_n} * \frac{60_{min}}{hr}$$

$$C_{p,da} = 0.24 + 0.444 * W_n$$

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 20, calculate the nominal temperature of the air leaving the heat pump condenser coil using,

$$T_0(T_j) = 70°\text{F} + \frac{\dot{Q}_{hp}(T_j)}{\dot{m}_{da} * C_{p,da}}$$

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 of this appendix. 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 of this appendix), determine $\dot{Q}_h(T_j)$ and $\dot{E}_h(T_j)$ as specified in section 4.2.1 of this appendix (*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,

$$\dot{Q}_h(T_j) = \dot{Q}_{hp}(T_j) + \dot{Q}_{CC}(T_j) \qquad \dot{E}_h(T_j) = \dot{E}_{hp}(T_j) + \dot{E}_{CC}(T_j)$$

where,

$$\dot{Q}_{CC}(T_j) = \dot{m}_{da} * C_{p,da} * [T_{CC} - T_0(T_j)] \qquad \dot{E}_{CC}(T_j) = \frac{\dot{Q}_{CC}(T_j)}{3.413\frac{Btu/h}{W}}$$

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 HSPF2 of a Heat Pump Having a Single-Speed Compressor and a Variable-Speed, Variable-Air-Volume-Rate Indoor Blower

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 of this appendix (Equations 4.2.2–1 and 4.2.2–2) for each outdoor bin temperature, T_j, that is listed in Table 20. 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 $H1_2$ test using:

$$\dot{m}_{da} = \overline{\dot{V}_s} * 0.075 \frac{lbm_{da}}{ft^3} * \frac{60_{min}}{hr} = \frac{\overline{\dot{V}}_{mx}}{v_n' * [1 + W_n]} * \frac{60_{min}}{hr} = \frac{\overline{\dot{V}}_{mx}}{v_n} * \frac{60_{min}}{hr}$$

$$C_{p,da} = 0.24 + 0.444 * W_n$$

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 20, calculate the nominal temperature of the air leaving the heat pump condenser coil using,

$$T_0(T_j) = 70°\text{F} + \frac{\dot{Q}_{hp}(T_j)}{\dot{m}_{da} * C_{p,da}}$$

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 of this appendix with the exception of replacing references to the H1C test and section 3.6.1 of this appendix with the $H1C_1$ test and section 3.6.2 of this appendix. 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 of this appendix), determine $\dot{Q}_h(T_j)$ and $\dot{E}_h(T_j)$ as specified in section 4.2.2 of this appendix (*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,

$$\dot{Q}_h(T_j) = \dot{Q}_{hp}(T_j) + \dot{Q}_{CC}(T_j) \qquad \dot{E}_h(T_j) = \dot{E}_{hp}(T_j) + \dot{E}_{CC}(T_j)$$

where,

$$\dot{Q}_{CC}(T_j) = \dot{m}_{da} * C_{p,da} * [T_{CC} - T_0(T_j)] \qquad \dot{E}_{CC}(T_j) = \frac{\dot{Q}_{CC}(T_j)}{3.413\frac{Btu/h}{W}}$$

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 HSPF2 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 of this appendix for both high and low capacity and at each outdoor bin temperature, T_j, that is listed in Table 20. 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:

$$\dot{m}_{da}^{k=1} = \bar{\dot{V}}_s * 0.075\frac{lbm_{da}}{ft^3} * \frac{60_{min}}{hr} = \frac{\bar{\dot{V}}_{mx}}{v_n' * [1 + W_n]} * \frac{60_{min}}{hr} = \frac{\bar{\dot{V}}_{mx}}{v_n} * \frac{60_{min}}{hr}$$

$$C_{p,da}^{k=1} = 0.24 + 0.444 * W_n$$

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 20, calculate the nominal temperature of the air leaving the heat pump condenser coil when operating at low capacity using,

$$T_0^{k=1}(T_j) = 70°\text{F} + \frac{\dot{Q}_{hp}^{k=1}(T_j)}{\dot{m}_{da}^{k=1} * C_{p,da}^{k=1}}$$

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 20, calculate the nominal temperature of the air leaving the heat pump condenser coil when operating at high capacity using,

$$T_0^{k=2}(T_j) = 70°\text{F} + \frac{\dot{Q}_{hp}^{k=2}(T_j)}{\dot{m}_{da}^{k=2} * C_{p,da}^{k=2}}$$

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 of this appendix, 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 of this appendix), determine $\dot{Q}_h^{k=1}(T_j)$ and $\dot{E}_h^{k=1}(T_j)$ as specified in section 4.2.3 of this appendix (*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,

$$\dot{Q}_{CC}^{k=1}(T_j) = \dot{m}_{da}^{k=1} * C_{p,da}^{k=1} * [T_{CC} - T_0^{k=1}(T_j)] \qquad \dot{E}_{CC}^{k=1}(T_j) = \frac{\dot{Q}_{CC}^{k=1}(T_j)}{3.413\frac{Btu/h}{W}}$$

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 of this appendix (*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,

$\dot{Q}_h^{k=2}(T_j) = \dot{Q}_{hp}^{k=2}(T_j) + \dot{Q}_{CC}^{k=2}(T_j)$ $\dot{E}_h^{k=2}(T_j) = \dot{E}_{hp}^{k=2}(T_j) + \dot{E}_{CC}^{k=2}(T_j)$

where,

$$\dot{Q}_{CC}^{k=2}(T_j) = \dot{m}_{da}^{k=2} * C_{p,da}^{k=2} * [T_{CC} - T_0^{k=2}(T_j)] \qquad \dot{E}_{CC}^{k=2}(T_j) = \frac{\dot{Q}_{CC}^{k=2}(T_j)}{3.413\frac{Btu/h}{W}}$$

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 HSPF2 of a Heat Pump Having a Variable-Speed Compressor [Reserved]

4.2.6 Additional Steps for Calculating the HSPF2 of a Heat Pump Having a Triple-Capacity Compressor

The only triple-capacity heat pumps covered are triple-capacity, northern heat pumps. For such heat pumps, the calculation of the Eq. 4.2–1 quantities

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

differ depending on whether the heat pump would cycle on and off at low capacity (section 4.2.6.1 of this appendix), cycle on and off at high capacity (section 4.2.6.2 of this appendix), cycle on and off at booster capacity (section 4.2.6.3 of this appendix), cycle between low and high capacity (section 4.2.6.4 of this appendix), cycle between high and booster capacity (section 4.2.6.5 of this appendix), operate continuously at low capacity (section 4.2.6.6 of this appendix), operate continuously at high capacity (section 4.2.6.7 of this appendix), operate continuously at booster capacity (section 4.2.6.8 of this appendix), or heat solely using resistive heating (also section 4.2.6.8 of this appendix) in responding to the building load. As applicable, the manufacturer must supply information regarding the outdoor temperature range at which each stage of compressor capacity is active. As an informative example, data may be submitted in this manner: At the low (k=1) compressor capacity, the outdoor temperature range of operation is 40 °F ≤ T ≤ 65 °F; At the high (k=2) compressor capacity, the outdoor temperature range of operation is 20 °F ≤ T ≤ 50 °F; At the booster (k=3) compressor capacity, the outdoor temperature range of operation is −20 °F ≤ T ≤ 30 °F.

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 the equations given in section 4.2.3 of this appendix for $\dot{Q}_h^{k=1}(T_j)$ and $\dot{E}_h^{k=1}(T_j)$) In evaluating the section 4.2.3 equations, 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 four quantities as specified in section 3.7 of this appendix. If, in accordance with section 3.6.6 of this appendix, the $H3_1$ test is conducted, calculate $\dot{Q}_h^{k=1}(17)$ and $\dot{E}_h^{k=1}(17)$ as specified in section 3.10 of this appendix and determine $\dot{Q}_h^{k=1}(35)$ and $\dot{E}_h^{k=1}(35)$ as specified in section 3.6.6 of this appendix.

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, evaluated as specified in section 3.7 of this appendix. Determine the equation input for $\dot{Q}_h^{k=2}(35)$ and $\dot{E}_h^{k=2}(35)$ from the $H2_2$ test evaluated as specified in section 3.9.1 of this appendix. Also, determine $\dot{Q}_h^{k=2}(17)$ and $\dot{E}_h^{k=2}(17)$ from the $H3_2$ test, evaluated as specified in section 3.10 of this appendix.

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

$$\dot{Q}_h^{k=3}(T_j) = \begin{cases} \dot{Q}_h^{k=3}(17) + \dfrac{\left[\dot{Q}_h^{k=3}(35) - \dot{Q}_h^{k=3}(17)\right] * (T_j - 17)}{35 - 17}, & if\ 17\ °\mathrm{F} < T_j \leq 45\ °\mathrm{F} \\ \dot{Q}_h^{k=3}(5) + \dfrac{\left[\dot{Q}_h^{k=3}(17) - \dot{Q}_h^{k=3}(5)\right] * (T_j - 5)}{17 - 5}, & if\ T_j \leq 17\ °\mathrm{F} \end{cases}$$

$$\dot{E}_h^{k=3}(T_j) = \begin{cases} \dot{E}_h^{k=3}(17) + \dfrac{\left[\dot{E}_h^{k=3}(35) - \dot{E}_h^{k=3}(17)\right] * (T_j - 17)}{35 - 17}, & if\ 17\ °\mathrm{F} < T_j \leq 45\ °\mathrm{F} \\ \dot{E}_h^{k=3}(5) + \dfrac{\left[\dot{E}_h^{k=3}(17) - \dot{E}_h^{k=3}(5)\right] * (T_j - 5)}{17 - 5}, & if\ T_j \leq 17\ °\mathrm{F} \end{cases}$$

Determine $\dot{Q}_h^{k=3}(17)$ and $\dot{E}_h^{k=3}(17)$ from the $H3_3$ test and determine $\dot{Q}_h^{k=2}(5)$ and $\dot{E}_h^{k=3}(5)$ from the $H4_3$ test. Calculate all four quantities as specified in section 3.10 of this appendix. Determine the equation input for $\dot{Q}_h^{k=3}(35)$ and $\dot{E}_h^{k=3}(35)$ as specified in section 3.6.6 of this appendix.

4.2.6.1 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, $\dot{Q}_h^{k=1}(T_j) \geq BL(T_j)$., and the Heat Pump Permits Low Compressor Capacity at T_j. Evaluate the Quantities

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

using Eqs. 4.2.3–1 and 4.2.3–2, respectively. Determine the equation inputs $X^{k=1}(T_j)$, PLF_j, and $\delta'(T_j)$ as specified in section 4.2.3.1. In calculating the part load factor, PLF_j, use the low-capacity cyclic-degradation coefficient C_D^h, [or equivalently, $C_D^h(k=1)$] determined in accordance with section 3.6.6 of this appendix.

4.2.6.2 Heat Pump Only Operates at High (k=2) Compressor Capacity at Temperature T_j and Its Capacity Is Greater Than or Equal to the Building Heating Load, $BL(T_j) < \dot{Q}_h^{k=2}(T_j)$

Evaluate the quantities

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

as specified in section 4.2.3.3 of this appendix. Determine the equation inputs $X^{k=2}(T_j)$, PLF_j, and $\delta'(T_j)$ as specified in section 4.2.3.3 of this appendix. In calculating the part load factor, PLF_j, use the high-capacity cyclic-degradation coefficient, $C_D^h(k=2)$ determined in accordance with section 3.6.6 of this appendix.

4.2.6.3 Heat Pump Only Operates at High (k=3) Compressor Capacity at Temperature T_j and its Capacity Is Greater Than or Equal to the Building Heating Load, $BL(T_j) \leq \dot{Q}_h^{k=3}(T_j)$

Calculate $\frac{RH(T_j)}{N}$ and using Eq. 4.2.3-2. Evaluate $\frac{e_h(T_j)}{N}$ using

$$\frac{e_h(T_j)}{N} = \frac{X^{k=3}(T_j) * \dot{E}_h^{k=3}(T_j) * \delta'(T_j)}{PLF_j} * \frac{n_j}{N}$$

where

$$X^{k=3}(T_j) = BL(T_j)/\dot{Q}_h^{k=3}(T_j) \text{ and } PLF_j = 1 - C_D^h(k=3) * [1 - X^{k=3}(T_j)$$

Determine the low temperature cut-out factor, $\delta'(T_j)$, using Eq. 4.2.3–3. Use the booster-capacity cyclic-degradation coefficient, $C_D^h(k=3)$ determined in accordance with section 3.6.6 of this appendix.

4.2.6.4 Heat Pump Alternates Between High (k=2) and Low (k=1) Compressor Capacity To Satisfy the Building Heating Load at a Temperature T_j, $\dot{Q}_h^{k=1}(T_j) < BL(T_j) < \dot{Q}_h^{k=2}(T_j)$

Evaluate the quantities

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

as specified in section 4.2.3.2 of this appendix. Determine the equation inputs $X^{k=1}(T_j)$, $X^{k=2}(T_j)$, and $\delta'(T_j)$ as specified in section 4.2.3.2 of this appendix.

4.2.6.5 Heat Pump Alternates Between High (k=2) and Booster (k=3) Compressor Capacity To Satisfy the Building Heating Load at a Temperature T_j, $\dot{Q}_h^{k=2}(T_j) < BL(T_j) < \dot{Q}_h^{k=3}(T_j)$

Calculate $\frac{RH(T_j)}{N}$ and using Eq. 4.2.3-2. Evaluate $\frac{e_h(T_j)}{N}$ using

$$\frac{e_h(T_j)}{N} = [X^{k=2}(T_j) * \dot{E}_h^{k=2}(T_j) + X^{k=3}(T_j) * \dot{E}_h^{k=3}(T_j)] * \delta'(T_j) * \frac{n_j}{N}$$

where:

$$X^{k=2}(T_j) = \frac{\dot{Q}_h^{k=3}(T_j) - BL(T_j)}{\dot{Q}_h^{k=3}(T_j) - \dot{Q}_h^{k=2}(T_j)}$$

and $X^{k=3}(T_j) = X^{k=2}(T_j)$ = the heating mode, booster capacity load factor for temperature bin j, dimensionless. Determine the low temperature cut-out factor, $\delta'(T_j)$, using Eq. 4.2.3–3.

4.2.6.6 Heat Pump Only Operates at Low (k=1) Capacity at Temperature T_j and Its Capacity Is Less Than the Building Heating Load, $BL(T_j) > \dot{Q}_h^{k=1}(T_j)$

$$\frac{e_h(T_j)}{N} = \dot{E}_h^{k=1}(T_j) * \delta'(T_j) * \frac{n_j}{N} \quad \text{and} \quad \frac{RH(T_j)}{N} = \frac{BL(T_j) - [\dot{Q}_h^{k=1}(T_j) * \delta\prime(T_j)]}{3.413\frac{Btu/h}{W}} * \frac{n_j}{N}$$

where the low temperature cut-out factor, $\delta'(T_j)$, is calculated using Eq. 4.2.3–3.

4.2.6.7 Heat Pump Only Operates at High (k=2) Capacity at Temperature T_j and Its Capacity Is Less Than the Building Heating Load, $BL(T_j) > \dot{Q}_h^{k=2}(T_j)$

Evaluate the quantities

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

as specified in section 4.2.3.4 of this appendix. Calculate $\delta''(T_j)$ using the equation given in section 4.2.3.4 of this appendix.

4.2.6.8 Heat Pump Only Operates at Booster (k=3) Capacity at Temperature T_j and Its Capacity Is Less Than the Building Heating Load, $BL(T_j) > \dot{Q}_h^{k=3}(T_j)$ or the System Converts To Using Only Resistive Heating

$$\frac{e_h(T_j)}{N} = \dot{E}_h^{k=3}(T_j) * \delta'(T_j) * \frac{n_j}{N} \quad \text{and} \quad \frac{RH(T_j)}{N} = \frac{BL(T_j) - [\dot{Q}_h^{k=3}(T_j) * \delta\prime(T_j)]}{3.413\frac{Btu/h}{W}} * \frac{n_j}{N}$$

where $\delta''(T_j)$ is calculated as specified in section 4.2.3.4 of this appendix if the heat pump is operating at its booster compressor capacity. If the heat pump system converts to using only resistive heating at outdoor temperature T_j, set $\delta'(T_j)$ equal to zero.

4.2.7 Additional Steps for Calculating the HSPF2 of a Heat Pump Having a Single Indoor Unit With Multiple Indoor Blowers. The Calculation of the Eq. 4.2–1 Quantities $e_h(T_j)/N$ and $RH(T_j)/N$ Are Evaluated as Specified in the Applicable Subsection

4.2.7.1 For Multiple Indoor Blower Heat Pumps That Are Connected to a Singular, Single-Speed Outdoor Unit

a. Calculate the space heating capacity, $\dot{Q}_h^{k=1}$ (Tj), and electrical power consumption, $\dot{E}_h^{k=1}$ (Tj), of the heat pump when operating at the heating minimum air volume rate and outdoor temperature T_j using Eqs. 4.2.2–3 and 4.2.2–4, respectively. Use these same equations to calculate the space heating capacity, $\dot{Q}_h^{k=2}$ (Tj) and electrical power consumption, $\dot{E}_h^{k=2}$ (Tj), of the test unit when operating at the heating full-load air volume rate and outdoor temperature T_j. In evaluating Eqs. 4.2.2–3 and 4.2.2– 4, determine the quantities $\dot{Q}_h^{k=1}(47)$ and $\dot{E}_h^{k=1}(47)$ from the $H1_1$ test; determine

$\dot{Q}_h^{k=2}(47)$ and $\dot{E}_h^{k=2}(47)$ from the $H1_2$ test. Evaluate all four quantities according to section 3.7 of this appendix. Determine the quantities $\dot{Q}_h^{k=1}(35)$ and $\dot{E}_h^{k=1}(35)$ as specified in section 3.6.2 of this appendix. Determine $\dot{Q}_h^{k=2}(35)$ and $\dot{E}_h^{k=2}(35)$ from the $H2_2$ frost accumulation test as calculated according to section 3.9.1 of this appendix. Determine the quantities $\dot{Q}_h^{k=1}(17)$ 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. Evaluate all four quantities according to section 3.10 of this appendix. Refer to section 3.6.2 and Table 12 of this appendix for additional information on the referenced laboratory tests.

b. Determine the heating mode cyclic degradation coefficient, C_D^h, as per sections 3.6.2 and 3.8 to 3.8.1 of this appendix. Assign this same value to $C_D^h(k = 2)$.

c. Except for using the above values of $\dot{Q}_h^{k=1}(Tj)$, $\dot{E}_h^{k=1}(Tj)$, $\dot{Q}_h^{k=2}(Tj)$, $\dot{E}_h^{k=2}(Tj)$, C_D^h, and $C_D^h(k = 2)$, calculate the quantities $e_h(T_j)/N$ as specified in section 4.2.3.1 of this appendix for cases where $\dot{Q}_h^{k=1}(Tj) \geq BL(T_j)$. For all other outdoor bin temperatures, T_j, calculate $e_h(Tj)/N$ and $RH_h(Tj)/N$ as specified in section 4.2.3.3 of this appendix if $\dot{Q}_h^{k=2}(Tj) > BL(Tj)$ or as specified in section 4.2.3.4 of this appendix if $\dot{Q}_h^{k=2}(Tj) \leq BL(T_j)$.

4.2.7.2 For Multiple Indoor Blower Heat Pumps Connected to Either a Single Outdoor Unit With a Two-Capacity Compressor or to Two Separate but Identical Model Single-Speed Outdoor Units. Calculate the Quantities $e_h(T_j)/N$ and $RH(T_j)/N$ as Specified in Section 4.2.3 of This Appendix

4.3 Calculations of Off-Mode Power Consumption

For central air conditioners and heat pumps with a cooling capacity of: Less than 36,000 Btu/h, determine the off mode represented value, $P_{W,OFF}$, with the following equation:

$$P_{W,OFF} = \frac{P1 + P2}{2};$$

greater than or equal to 36,000 Btu/h, calculate the capacity scaling factor according to:

$$F_{scale} = \frac{\dot{Q}_C(95)}{36{,}000},$$

where, $\dot{Q}_C(95)$ is the total cooling capacity at the A or A_2 test condition, and determine the off mode represented value, $P_{W,OFF}$, with the following equation:

$$P_{W,OFF} = \frac{P1 + P2}{2 \times F_{scale}};$$

4.4 Rounding of SEER2 and HSPF2 for Reporting Purposes

After calculating SEER2 according to section 4.1 of this appendix and HSPF2 according to section 4.2 of this appendix round the values off as specified per § 430.23(m) of title 10 of the Code of Federal Regulations.

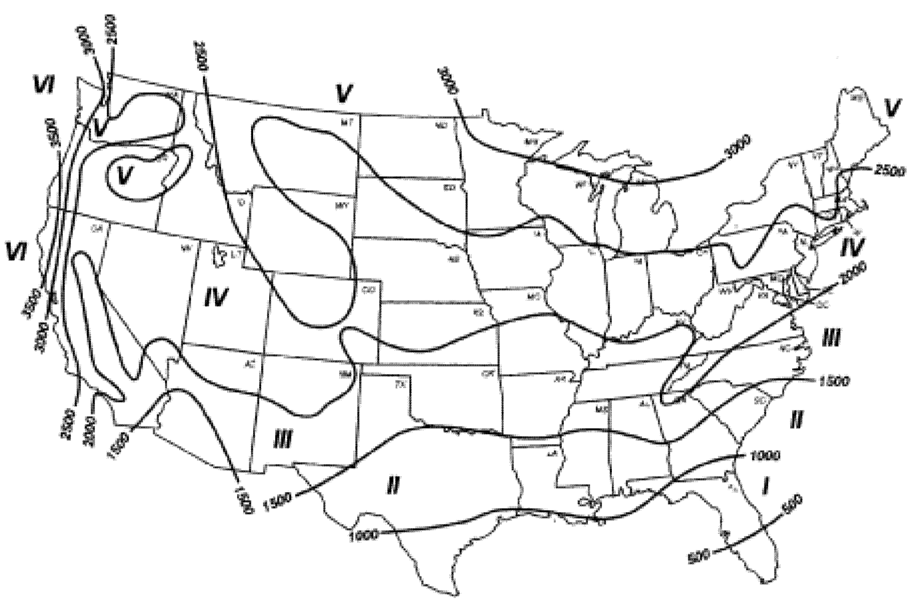

Figure 1—Climatic Regions I through VI for the United States

TABLE 21—REPRESENTATIVE COOLING AND HEATING LOAD HOURS FOR EACH GENERALIZED CLIMATIC REGION

Climatic region	Cooling load hours CLH_R	Heating load hours HLH_R
I	2,400	493
II	1,800	857
III	1,200	1,247
IV	800	1,701
Rating Values	1,000	1,572
V	400	2,202
VI	200	1,842

4.5 Calculations of the SHR, Which Should Be Computed for Different Equipment Configurations and Test Conditions Specified in Table 22.

TABLE 22—APPLICABLE TEST CONDITIONS FOR CALCULATION OF THE SENSIBLE HEAT RATIO

Equipment configuration	Reference table number of Appendix M	SHR computation with results from	Computed values
Units Having a Single-Speed Compressor and a Fixed-Speed Indoor Blower, a Constant Air Volume Rate Indoor Blower, or Single-Speed Coil-Only.	4	B Test	SHR(B).
Units Having a Single-Speed Compressor That Meet the section 3.2.2.1 Indoor Unit Requirements.	5	B2 and B1 Tests	SHR(B1), SHR(B2).
Units Having a Two-Capacity Compressor	6	B2 and B1 Tests	SHR(B1), SHR(B2).
Units Having a Variable-Speed Compressor	7	B2 and B1 Tests	SHR(B1), SHR(B2).

The SHR is defined and calculated as follows:

$$SHR = \frac{Sensible\ Cooling\ Capacity}{Total\ Cooling\ Capacity}$$

$$= \frac{\dot{Q}_{sc}^{k}(T)}{\dot{Q}_{c}^{k}(T)}$$

Where both the total and sensible cooling capacities are determined from the same cooling mode test and calculated from data collected over the same 30-minute data collection interval.

4.6 Calculations of the Energy Efficiency Ratio (EER)

Calculate the energy efficiency ratio using,

$$EER = \frac{Total\ Cooling\ Capacity}{Total\ Electrical\ Power\ Consumption}$$

$$= \frac{\dot{Q}_{c}^{k}(T)}{\dot{E}_{c}^{k}(T)}$$

where $\dot{Q}_c^k(T)$ and $\dot{E}_c^k(T)$ are the space cooling capacity and electrical power consumption determined from the 30-minute data collection interval of the same steady-state wet coil cooling mode test and calculated as specified in section 3.3 of this appendix. Add the letter identification for each steady-state test as a subscript (*e.g.,* EER_{A_2}) to differentiate among the resulting EER values. The represented value of EER is determined from the A or A_2 test, whichever is applicable. The represented value of EER determined in accordance with this appendix is called EER2.

[FR Doc. 2016–30004 Filed 1–4–17; 8:45 am]

BILLING CODE 6450–01–P