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10 CFR Parts 429 and 430
Energy Conservation Program: Test Procedures for Central Air Conditioners and Heat Pumps; Proposed Rule

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: Supplemental notice of proposed rulemaking.

SUMMARY: The U.S. Department of Energy (DOE) proposes to revise its test procedures for central air conditioners and heat pumps (CAC/HP) established under the Energy Policy and Conservation Act. DOE published several proposals in a November 2015 supplemental notice of proposed rulemaking (SNOPR). DOE finalized some of the proposed test procedure amendments in a June 2016 final rule. This SNOPR proposes additional revisions to some of the amendments proposed in the past notices and proposes some additional amendments. Specifically, this SNOPR proposes 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 amendments as part of a new appendix M1 that would be the basis for making efficiency representations as of the compliance date for any amended energy conservation standards. Broadly speaking, the proposed amendments address the off-mode test procedures, clarifications on test set-up and fan delays, limits to gross indoor fin surface area for valid combinations, external static pressure conditions for testing, clarifications on represented values for CAC/HP that are distributed in commerce with multiple refrigerants, and the methodology for testing and calculating heating performance. DOE does not expect the proposed changes to appendix M to change measured efficiency. However, DOE has determined that the proposed procedures in new appendix M1 would change measured efficiency. DOE welcomes comments from the public on any subject within the scope of this test procedure rulemaking.

DATES: DOE will accept comments, data, and information regarding this supplemental notice of proposed rulemaking (SNOPR) no later than September 23, 2016. See section V, ''Public Participation,'' for details.

DOE will hold a public meeting on Friday, August 26, 2016, from 10 a.m. to 2 p.m., in Washington, DC. The meeting will also be broadcast as a webinar. See section V, Public Participation, for webinar registration information, participant instructions, and information about the capabilities available to webinar participants.

ADDRESSES: The public meeting will be held at the U.S. Department of Energy, Forrestal Building, Room 1E–245, 1000 Independence Avenue SW., Washington, DC 20585.

Any comments submitted must identify the Test Procedure SNOPR for central air conditioners and heat pumps, and provide docket number EERE–2016–BT–TP–0029 and/or regulatory information number (RIN) number 1904–AD 71. Comments may be submitted using any of the following methods:

(1) *Federal eRulemaking Portal: www.regulations.gov*. Follow the instructions for submitting comments.

(2) *Email: CACHeatPump2016TP0029@ee.doe.gov* Include the docket number and/or RIN in the subject line of the message.

(3) *Mail:* Appliance and Equipment Standards Program, U.S. Department of Energy, Building Technologies Office, Mailstop EE–5B, 1000 Independence Avenue SW., Washington, DC 20585–0121. If possible, please submit all items on a CD, in which case it is not necessary to include printed copies.

(4) *Hand Delivery/Courier:* Appliance and Equipment Standards Program, U.S. Department of Energy, Building Technologies Office, 950 L'Enfant Plaza, SW., 6th Floor, Washington, DC 20024. Telephone: (202) 586–6636. If possible, please submit all items on a CD, in which case it is not necessary to include printed copies.

For detailed instructions on submitting comments and additional information on the rulemaking process, see section V of this document (Public Participation).

Docket: The docket, which includes **Federal Register** notices, comments, and other supporting documents/materials, is available for review at *www.regulations.gov*. All documents in the docket are listed in the *www.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 instructions on how to access all documents, including public comments, in the docket. See section V for information on how to submit comments through *www.regulations.gov*.

FOR FURTHER INFORMATION CONTACT:

Ashley Armstrong, U.S. Department of Energy, Office of Energy Efficiency and Renewable Energy, Building Technologies Program, EE–5B, 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 submit a comment, review other public comments and the docket, or participate in the public meeting, contact the Appliance and Equipment Standards Program staff at (202) 586–6636 or by email: *CACHeatPump2016TP0029@ee.doe.gov*.

SUPPLEMENTARY INFORMATION: DOE is not proposing to incorporate any new standards by reference in this supplemental notice of proposed rulemaking.

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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 SNOPR. (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. Some of the amendments proposed in this SNOPR will alter the measured efficiency, as represented in the regulating metrics of seasonal energy efficiency ratio (SEER), energy efficiency ratio (EER), and heating seasonal performance factor (HSPF). These amendments are proposed as part of a new appendix M1. Use of the test procedure changes proposed in this notice as part of a new appendix M1, if adopted, would become mandatory to demonstrate compliance if the existing energy conservation standards are revised. (42 U.S.C. 6293(e)(2)) In revising the energy conservation standards in a separate rulemaking, DOE would create a cross-walk from the existing standards under the current test procedure to what the standards would be if tested using the revised test procedure.

On December 19, 2007, the President signed the Energy Independence and Security Act of 2007 (EISA 2007), Public Law 110–140, which contains numerous amendments to EPCA. 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 incorporated into the SEER and HSPF metrics, while off mode power consumption is separately regulated. This SNOPR includes proposals relevant to the determination of both SEER and HSPF (including standby mode) and off mode power consumption. DOE would then use the cross-walked equivalent of the existing standard as the baseline for its standards analysis to prevent backsliding as required under 42 U.S.C. 6295(o)(1).

B. Background

DOE initiated a round of test procedure revisions for CAC/HP by

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

publishing a notice of proposed rulemaking in the **Federal Register** on June 2, 2010 (June 2010 NOPR; 75 FR 31224). 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 69278) 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 36992.

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 and made several recommendations that are being addressed in the proposals of this SNOPR. DOE believes proposed changes are consistent with the intent of the Working Group.

This SNOPR 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.

II. Synopsis of the Supplemental Notice of Proposed Rulemaking

In this SNOPR, DOE proposes revising 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. In this SNOPR, DOE proposes two sets of changes: One set of proposed 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 proposed changes to create a new Appendix M1 that would be used for testing to demonstrate compliance with any amended energy conservation standards (agreed to be January 1, 2023 by the Working Group in the CAC rulemaking negotiations (CAC ECS: ASRAC Term Sheet, No. 76)). DOE requests comment on whether representations in accordance with Appendix M1 should be permitted prior to the compliance date of any amended energy conservation standards. DOE does not expect the proposed changes to Appendix M to change measured efficiency. However, DOE has determined that the proposed procedures in the new Appendix M1 would change measured efficiency.

In this SNOPR, DOE proposes the following changes to certification requirements:

(1) Certification of the indoor fan off delay used for coil-only tests.

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

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

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

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

(6) 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 without voiding the warranty; and

(7) Certification of details regarding the indoor units with which unmatched outdoor units are tested.

DOE proposes the following changes to Appendix M:

(1) Establishment of a 4-hour or 8-hour delay time before the power measurement for units that require the outdoor temperature setting to reach thermal equilibrium;

(2) A limit on the internal volume of lines and devices connected to measure pressure at refrigerant circuit locations where the refrigerant state can switch from liquid to vapor for different test operating conditions;

(3) Requiring bin-by-bin EER and coefficient of performance (COP) interpolations for all variable speed units, to calculate performance at intermediate compressor speeds;

(4) Requiring a 30-minute test without the outside-air apparatus connected (a ''non-ducted'' 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; and

(5) Imposing indoor coil size limits for split system ratings.

DOE proposes the following provisions for new Appendix M1:

(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 HSPF; and

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

If adopted, the test procedures proposed in this SNOPR to appendix M for subpart B to 10 CFR part 430 pertaining to the efficiency of CAC/HP would be effective 30 days after publication in the **Federal Register** (referred to as the ''effective date''). Pursuant to EPCA, manufacturers of covered products would be 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)) On or after 180 days after publication of a final rule, any representations made with respect to the energy use or efficiency of CAC/HPs would be required to be made in accordance with the results of testing pursuant to the amended test procedures. (42 U.S.C. 6293(c)(2))(42 U.S.C. 6293(c)(2))

If adopted, the test procedures proposed in this SNOPR for appendix M1 to subpart B of 10 CFR part 430 pertaining to the efficiency of CAC/HP would be effective 30 days after publication in the **Federal Register**. The appendix M1 procedures would be required to be used 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.

As noted in section I.A, 42 U.S.C. 6293(e) requires DOE to determine to what extent, if any, the proposed test procedure would alter the measured energy efficiency and measured energy use. DOE has determined that some of the proposed amendments in the new Appendix M1 would result in a change in measured energy efficiency and measured energy use for CAC/HP. DOE is conducting a separate rulemaking to amend the energy conservation standards for CAC/HP, which will take into account the test procedure revisions in Appendix M1. (CAC ECS: Docket No. EERE–2014–BT–STD–0048)

III. Discussion

This section discusses the revisions to the certification requirements and test procedure that DOE proposes in this SNOPR.

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

1. Representation Accommodation

The CAC/HP ECS Working Group made certain recommendations related to the Appendix M1 test procedure, with a recommended compliance date of January 1, 2023, for representations based on Appendix M1. (Docket No. EERE–2014–BT–STD–0048, No. 76, Recommendation #7) While the June 2016 Test Procedure Final Rule adopted mandatory testing requirements for representations of all basic models [81 FR at 37050–37051; 10 CFR 429.16(b)(2)(i)], the Working Group recommended several accommodations for representations for split systems:

○ 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.

(Docket No. EERE–2014–BT–STD–0048, No. 76, Recommendation #10)

DOE proposes to implement these recommendations, in their entirety, in 10 CFR 429.16 and 429.70.

2. Highest Sales Volume Requirement

The CAC/HP ECS Working Group recommended that DOE implement the following requirements for single-split-system air conditioners and suggested implementing regulatory text:

• 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.

(Docket No. EERE–2014–BT–STD–0048, No. 76, Recommendation #7)

DOE addressed the first and third bullets in a final rule published on June 8, 2016, (June 2016 final rule), but at that time declined to implement the second bullet, which recommends removing the requirement that the tested combination be the highest sales volume combination (HSVC). DOE also received comments from non-working group members regarding this requirement. JCI commented that the current language used in Appendix M denoting the HSVC match cannot be determined with exact statistics and that it actually inhibits the adoption of new and promising advancements in product design. (CAC TP: JCI, No. 66 at p. 4) In contrast, Unico commented that, as an indoor coil manufacturer, it believes it to be important that the outdoor unit manufacturer continue to test and rate the HSVC, as this is an integral requirement for their AEDM to maintain accuracy. (CAC TP: Unico, No. 63 at p. 2)

DOE believes the CAC/HP ECS Working Group recommendation adequately addresses JCI's concern about using the HSVC as a tested combination. In response to Unico, DOE notes that the requirements adopted in the June 2016 final rule require independent coil manufacturers (ICMs) to test their own equipment. It is the ICM's own responsibility to ensure the accuracy of its AEDMs. ICMs may conduct additional testing or work with outdoor unit manufacturers (OUMs) as needed to do so. For these reasons, DOE is proposing to remove the requirement that the tested combination be the HSVC. DOE proposes to apply the requirements as recommended by the CAC/HP ECS Working Group to all single-split-system air conditioners and heat pumps, including space-constrained and small-duct, high-velocity, distributed in commerce by an OUM.

3. Determination of Certified Rating for Multi-Split, Multi-Circuit, and Multi-Head Mini-Split Systems

In the June 2016 final rule, DOE modified the testing requirements for multi-head mini-split systems and multi-split systems, and added similar requirements for testing multi-circuit systems. DOE also clarified that these requirements apply to variable refrigerant flow (VRF) systems that are single-phase and less than 65,000 Btu/h.[5] For all multi-split, multi-circuit, and multi-head mini-split systems, DOE required that, at a minimum, each model of outdoor unit must be tested as part of a tested combination (as defined at 10 CFR 430.2) that includes only non-ducted indoor units. For any models of outdoor units also sold with ducted indoor units, a second ''tested combination'' including only ducted indoor units must be tested. DOE also allowed for manufacturers to rate a mixed non-ducted/ducted combination as the mean of the represented values for the tested non-ducted and ducted combinations, and allowed manufacturers to test and rate specific individual combinations as separate basic models, even if they share the same model of outdoor unit. 81 FR 37003–37005 (June 8, 2016)

DOE also added a requirement that for any models of outdoor units also sold with models of small-duct, high velocity (SDHV) indoor units, a ''tested combination'' composed entirely of SDHV indoor units must be used for testing and rating. However, such a system must be certified as a different basic model. Finally, DOE allowed mix-match ratings for SDHV and other non-ducted or ducted indoor units based on an average of the ratings of the two individual indoor unit types. 81 FR 37004 (June 8, 2016)

In the June 2010 NOPR, DOE had proposed lower minimum external static pressure (ESP) requirements for ducted multi-split systems (75 FR at 31232), and in the November 2015 SNOPR, DOE proposed to implement these requirements using the term ''short duct systems,'' which could refer to multi-split, multi-head mini-split, or multi-circuit systems with indoor units that produce a limited level of external static pressure. 80 FR at 69314 (Nov. 9, 2015). In response to the SNOPR, DOE received several comments regarding its terminology and testing requirements related to short-duct systems as well as requests for changing terminology and testing requirements to include low-static and mid-static systems, as recommended in the CAC/HP ECS Working Group Term Sheet. Therefore in the June 2016 final rule, DOE maintained the existing ducted system terminology and is addressing the earlier comments from stakeholders and recommendations from the Working Group in this SNOPR.

Unico supported DOE's definition of short-ducted systems which would create four indoor unit types for multi-split systems: Short-ducted (previously described as ''ducted''), conventional ducted, SDHV-ducted, and non-ducted. (CAC TP: Unico, No. 63 at p. 11) In the Term Sheet, the CAC/HP ECS Working Group recommended that DOE define ''low-static system'' and ''mid-static system'' as discussed in section III.C.1. (CAC ECS: Docket No. EERE–2014–BT–STD–0048, No. 76 at p. 1–2) These systems are essentially sub-categories of DOE's earlier proposal for short-ducted systems.

In addition, several stakeholders commented that multi-split systems may also be paired with models of conventional ducted indoor units. UTC/Carrier commented that some manufacturers also offer ducted units with external static pressure capabilities greater than 0.65 in w.c., the maximum external static pressure proposed by the Working Group for mid-static ducted units and recommended that DOE also include a requirement for separate multi-split system ratings with these ''standard'' ducted indoor units. (CAC TP: UTC/Carrier, No. 62 at p. 3–4)

Rheem commented that the definition of multi-split system is not limited to a specific duct configuration and that testing of all possible duct configurations should be considered. Rheem further commented that the testing requirements should be the same as single-split systems using conventional ducted indoor units because multi-split systems duct losses are the same as the standard single-split system. (CAC TP: Rheem, No. 69 at p. 5)

NEEA and NPCC commented that multi-split systems paired with more conventional blower coil indoor units should be testable with the external static pressure conditions specified for conventional blower coil units. (CAC TP: NEEA and NPCC, No. 64 at p. 3–4)

The California IOUs commented that additional testing is needed to ensure that the AEDM gives accurate ratings for all of the possible combinations when an outdoor unit of a multi-split system is paired with a conventional central forced air indoor unit. They said that, at present, a variable speed, mini-split outdoor unit is connected to an indoor unit(s) from the same manufacturer with complex software controls that produce the variable modes of operation needed to respond to indoor and outdoor conditions. They also asserted that the indoor units can be short ducted or ductless cassettes. Finally, they commented that, if the same outdoor section is installed with a central forced air unit, it will have indoor fan operation modes and significantly different power draw and may not be representative of the nuanced behavior of the ductless and short duct components. (CAC TP: California IOUs, No. 67 at p. 3)

Given the multiple types of indoor units with which these systems can be paired, several stakeholders also made recommendations related to the testing and rating requirements.

Unico commented that multi-split ratings should be listed with homogeneous type of indoor units, which should be based on tests or a valid AEDM. Unico commented that short-ducted, conventional-ducted, SDHV-ducted and non-ducted are different types and should all be tested and rated using the appropriate test procedure for the type, and that ratings with mixed types should be an average. (CAC TP: Unico, No. 63 at p. 2)

Mitsubishi proposed that given the potential additional testing requirements presented for systems with multiple families of ducted indoor unit (low-static, mid-static and standard-static ducted), a manufacturer be allowed to produce tested combinations of all low-static, all mid-static or all standard-static indoor units, and that, if they do not wish to have separate ratings, they must use the highest rating of external static pressure to establish the tested combination. (CAC TP: Mitsubishi, No. 68 at p. 3)

Goodman suggested that any combinations of non-ducted, low-static, mid-static and/or high-static indoor units be based on the highest static units in the combination if a single rating is to be used for all short-ducted indoor units. In addition, Goodman stated that it believes these combinations should have the capability of being rated and certified using either test data or an AEDM. Goodman suggested that, if multiple combinations of non-ducted, low-static, mid-static and/or high-static indoor units are matched with a particular outdoor unit, the testing should be performed using the appropriate test static for each indoor unit. (CAC TP: Goodman, No. 73 at p. 13–14)

DOE supports the Working Group recommendations to replace its proposal to use the terminology short-duct with low-static and mid-static. The proposed definitions for these terms are discussed in section III.C.1. In addition, DOE agrees that multi-split, multi-head mini-split, or multi-circuit systems can include conventional ducted indoor units. DOE notes that the proposed test procedure allows selection of an appropriate external static pressure for this case.

[5] A VRF system is a multi-split system with at least three compressor capacity stages, but most VRF systems have variable-speed compressors.

After reviewing the comments, DOE proposes that multi-split, multi-head mini-split, and multi-circuit systems can be tested and rated with five kinds of indoor units: Non-ducted, low-static ducted, mid-static ducted, conventional ducted, or SDHV. However, DOE agrees that if a manufacturer offers an outdoor model with all five kinds of indoor units, a requirement to determine a rating through testing of each could be burdensome. Therefore, DOE proposes 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. If a manufacturer also offers the model of outdoor unit with models of low-static, mid-static, and/or conventional ducted indoor units, the manufacturer must at a minimum also test and rate a second ''tested combination'' with the highest static variety of indoor unit offered. The manufacturer may also choose to test and rate additional ''tested combinations'' composed of the lower static varieties. In each case, the manufacturer must test with the appropriate external static pressure. DOE believes that this option reduces test burden sufficiently and is not proposing use of AEDMs for these systems.

DOE proposes 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 proposes 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. As noted in the June 2016 final rule, SDHV represented values must be a separate basic model. Any represented values for a mixed system including SDHV and another style of unit must be in the same basic model as the SDHV model. Tables III.1 and III.2 summarize example represented values.

Table III.1 Example Represented Values for Non-SDHV Multi-Split Systems

Basic Model	Indiv. Model # (Outdoor Unit)	Indiv. Model #(s) (Indoor Unit)	Sample Size	Non-ducted Rep. Value	Conv. Ducted Rep. Value	Mid-Static Rep. Value	Low-Static Rep. Value	Mix Rep. Value (ND/CD)	Mix Rep. Value (ND/MS)	Mix Rep. Value (ND/LS)	Mix Rep. Value (CD/MS)	Mix Rep. Value (CD/LS)
ABC	ABC	***	4	15.00	14.00	-	-	14.50	-	-	-	-
ABC-ND1	ABC	2-A123; 3-JH746	2	17.00	-	-	-	-	-	-	-	-
XYZ	XYZ	***	8	15.50	14.00	14.50	15.00	14.75	15.00	15.25	14.25	14.50

TABLE III.2—EXAMPLE REPRESENTED VALUES FOR SDHV MULTI-SPLIT SYSTEMS

Basic model	Individual model No. (outdoor unit)	Individual model No.(s) (indoor unit)	Sample size	SDHV rep. value	Mix rep. value (ND)	Mix rep. value (CD)	Mix rep. value (MS)	Mix rep. value (LS)
ABC–SDHV	ABC	* * *	6	11.50	13.25	12.75	....	....

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:

Indoor unit means part of a split-system air conditioner or heat pump that includes (a) an arrangement of refrigerant-to-air heat transfer coil(s) for transfer of heat between the refrigerant and the indoor air and (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.

Service coil means an arrangement of refrigerant-to-air heat transfer coil(s) and condensate drain pan that may or may not include sheet metal or plastic parts to direct/route airflow over the coil(s), external cabinetry, and/or a cooling mode expansion device, and is sold exclusively to replace an uncased coil or cased coil that has already been placed into service and is labeled accordingly.

In this SNOPR, DOE proposes to modify the adopted definition of service coil to more explicitly define what ''labeled accordingly'' means. Under 42 U.S.C. 6295(r), the Secretary may include any requirement which the Secretary determines is necessary to assure that each covered product to which such standard applies meets the required minimum level of energy efficiency or maximum quantity of energy use specified in such standard.

In this specific case, DOE believes service coils must be distinguished from indoor units to ensure compliance with the applicable energy conservation standards for central air conditioners and heat pumps. Specifically, DOE proposes 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, 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.

5. Efficiency Representations of Split-Systems for Multiple Refrigerants

Split-system CAC/HP are required to be tested as a system. Prior to the June 2016 final rule, the condensing unit was required to be tested with ''the evaporator coil that is likely to have the largest volume of retail sales with the particular model of condensing unit'' (commonly referred to as the highest sales volume combination or HSVC). 10 CFR 429.16(a)(2)(ii) as of January 1, 2016. The June 2016 final rule amended the definition of ''central air conditioner or central air conditioning heat pump'' to recognize instances in which there is no HSVC, *i.e.,* an outdoor unit is sold separately with no matching indoor unit, referred to as an ''outdoor unit with no match''. 81 FR at 36999 (June 8, 2016).

As discussed in the June 2016 final rule, outdoor units with no match are typically a result of the phase-out of HCFC-22 refrigerant. Effective January 1, 2010, the U.S. Environmental Protection Agency (EPA) banned the sale and distribution of those central air conditioning systems and heat pump systems that are designed to use HCFC-22 refrigerant. 74 FR 66450 (Dec. 15, 2009). EPA's rulemaking included an exception for the manufacture and importation of replacement components, as long as those components are not pre-charged with HCFC-22. Id. at 66459–60. Because complete HCFC-22 systems can no longer be distributed, DOE established test procedure requirements for outdoor units that have ''no match,'' or are not sold with a matching indoor unit, which includes those units designed to use HCFC-22.

The ''no match'' test procedure's goal is that the test should produce measurements of energy efficiency during a representative average use cycle (see 42 U.S.C. 6293(b)(3)) while also ensuring that any field-matched combination (including the new ''no-match'' outdoor unit and an existing indoor unit) meets the standard. Due to the nature of these no-match systems, however, neither the manufacturer nor DOE knows exactly what the paired system will be for an outdoor unit with no match. To ensure compliance, DOE established indoor unit specifications that are representative of a less efficient unit (representative of units on the market at the time of the change in EPA regulations) that could be paired with the given outdoor unit with no match. Specifically, DOE established a requirement that outdoor units without a matching indoor unit must be tested with an indoor unit with a normalized gross indoor fin surface (NGIFS) [6] no higher than 1.0 square inches per British thermal unit per hour (sq. in./Btu/hr). 81 FR at 37010 (June 8, 2016).

In response to the phase-out of HCFC-22, one course pursued by manufacturers has been to use the refrigerant R-407C, which can be used as a drop-in replacement for HCFC-22 if oil compatibility issues are addressed. (No. 1 at pp. 2–6) Because R-407C is a replacement for HCFC-22, it is possible for a central air conditioner to operate either with R-407C or with HCFC-22. Such a unit could be shipped charged with R-407C, or shipped without the refrigerant charge (*i.e.,* dry-shipped). A dry-shipped unit could then either be sold as part of an R-407C split-system, or sold as a replacement component and charged with HCFC-22. In any case, R-407C outdoor units are often marketed as replacements for HCFC-22 outdoor units, as indicated in marketing material. (Docket No. EERE–2016–BT–TP–0029–0007, –0008, –0009, –0010, –0011, –0012 and –0013) Some R-407C outdoor units are more explicitly marketed as HCFC-22 replacements than other units (*e.g.,* indicating that the outdoor unit is ''compatible with R-22 coils and linesets!''). ((Docket No. EERE–2016–BT–TP–0029–0010 at p. 1).

To address instances in which the manufacturer indicates that more than one refrigerant is acceptable for use in a unit (*i.e.,* the manufacturer specifications include use of multiple refrigerants or the warranty would not be voided by the use of more than one refrigerant), DOE is proposing that a split-system air conditioner or heat pump, including outdoor unit with no match, must be certified as a separate individual combination (including outdoor unit without match as applicable) for every acceptable refrigerant. Specifically, each individual combination (including outdoor unit without match corresponding to each acceptable refrigerant) 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 proposes that manufacturers must certify the refrigerant for every individual combination that is distributed in commerce (including every outdoor unit with no match). For models where the manufacturer only indicates one acceptable refrigerant (DOE expects this to be the majority of units), this proposal would simply entail certifying to DOE the refrigerant for which the model is designed. Finally, DOE proposes that 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 without a specific refrigerant specified or not charged with a specified refrigerant from the point of manufacture, a manufacturer must determine the represented value as an outdoor unit with no match.

Under this proposal, if an outdoor unit manufacturer (OUM) indicates as an acceptable refrigerant for a model of outdoor unit a refrigerant that is banned for inclusion in CAC/HP distributed as systems, such as HCFC-22, the OUM would have to determine represented values (*e.g.,* SEER) for the model of outdoor unit tested as an outdoor unit with no match. Within the same basic model, the manufacturer must determine a represented value for all individual split-system combinations using the same model of outdoor unit for any acceptable refrigerants with which the model of outdoor unit can legally be sold as a system. DOE has tentatively determined that specification by an OUM as to the acceptable refrigerant indicates the ultimate use or uses for which the unit was designed and manufactured.

Inclusion of HCFC-22 as an acceptable refrigerant by the manufacturer indicates that the model of outdoor unit was designed and manufactured to be sold separately as a replacement component (*i.e.,* as a model of outdoor unit with no match), because manufacturers are prohibited from selling and distributing central air conditioning systems and heat pump systems that use HCFC-22 refrigerant,

[6] NGIFS is equal to normalized gross indoor fin surface (for a conventional fin-tube heat exchanger, two times fin length times fin width times the number of fins) divided by the system cooling capacity.

except as replacement components (*i.e.,* outdoor units with no match).

As indicated previously in this discussion, it is DOE's understanding that the listing of acceptable refrigerants also impacts the unit's warranty. In order for a unit to remain under warranty, the unit generally must be operated and maintained as recommended by the manufacturer. If a manufacturer indicates that HCFC-22 is an acceptable refrigerant, its use in an outdoor unit would not be expected to void the warranty. Again, DOE understands conformance with the warranty to be an indication of the intended use for which a model is designed and manufactured. Additionally, DOE understands that manufacturer literature for some models may not explicitly state which refrigerants may be used without voiding the warranty and may instead generally refer to specific refrigerant characteristics for the warranty to remain valid. If for such a case, HCFC-22 meets the specified characteristics, DOE's proposal would require that the manufacturer certify, within the same basic model, an individual split-system combination or outdoor unit with no match for each refrigerant that meet these warranty criteria or characteristics.

Under the certification requirements proposed in this SNOPR, an outdoor unit for which both R-407C and HCFC-22 are acceptable refrigerants would need to be certified as a split-system combination and as an outdoor unit with no match, with representations for each. Per DOE's regulations established in the June 2016 final rule, outdoor units with no match cannot be certified using an AEDM, and the model of outdoor unit must be tested with an indoor unit meeting specified criteria. 81 FR at 37051 (June 8, 2016). Therefore, for a model of outdoor unit for which both R-407C and HCFC-22 are acceptable refrigerants, the outdoor unit with no match (with HCFC-22) must be tested and certified. In addition, DOE proposes to require that any split-system combination (with R-407C) must also be tested. The proposed certification requirements would represent the energy efficiency of an outdoor unit during a representative average use cycle for each intended sales scenario (*i.e.,* either sold as a split system and installed with a new matching indoor unit, or sold as a replacement component and installed with a legacy indoor unit).

In addition, DOE recognizes that concerns regarding warrantee coverage for a given refrigerant may not be a concern for all installers and consumers. Consequently, DOE is concerned that the lack of explicit indication that a unit is acceptable for use with HCFC-22 may not prevent installation of such units with the refrigerants, if the installers and consumers have reasonable confidence that the unit can operate with this refrigerant. Because of the similarity of HCFC-22 and R-407C and the history of CAC/HP being used interchangeably with both of these refrigerants, this issue could very well arise for any unit certified and warranted for use with R-407C. Hence, DOE proposes that any outdoor unit intended for use in a split system with R-407C, *i.e.* any unit shipped with a charge of any amount of R-407C, would also have to be rated as an outdoor unit with no match.

Nearly all outdoor units of split systems are shipped with a quantity of refrigerant charge that is close to the required charge for installation. This has been confirmed by observation of units tested by DOE. Line sets for connecting indoor units to outdoor units also are sold with an appropriate pre-charge to compensate for the different amount of charge that remains in the lines of different-length line sets. During set-up, the refrigerant charge of the assembled system is adjusted, and the pre-charging of the components limits the amount of refrigerant that is needed to be added or removed in order to match the charging conditions specified in the manufacturer's installation instructions. Because of this general practice to ship outdoor units with close to full charge, DOE considers use of a charge quantity that is much less than the charge specified by the instructions to be equivalent to shipping a unit without refrigerant. Hence, DOE proposes to require a no-match rating for outdoor units that are shipped with a charge amount such that adjustment of charge as specified in manufacturer's instructions requires addition of more than one pound of refrigerant.

As an example illustrating the certification requirement proposals discussed in this section, assume a manufacturer advertises a model of outdoor unit for use with either HCFC-22 or R-407C.

In 10 CFR 430.2 (as amended in the June 2016 final rule), DOE defines ''basic model'' for OUMs as ''all individual combinations having the same model of outdoor unit, which means comparably performing compressor(s) [a variation of no more than five percent in displacement rate (volume per time) as rated by the compressor manufacturer, and no more than five percent in capacity and power input for the same operating conditions as rated by the compressor manufacturer], outdoor coil(s) [no more than five percent variation in face area and total fin surface area; same fin material; same tube material], and outdoor fan(s) [no more than ten percent variation in air flow and no more than twenty percent variation in power input].'' According to this definition, the model of outdoor unit intended to be sold with both HCFC-22 and R-407C would represent multiple individual combinations within the same basic model. Therefore, a manufacturer has to determine a represented value for each single-split-system combination (sold for use with R-407C) as well as determine a represented value for the outdoor unit with no match (sold for use with HCFC-22). *See* 10 CFR 429.16(a)(1) (as amended in the June 2016 final rule), 81 FR 36001, 37056 (June 8, 2016).

Paragraph 10 CFR 429.16(b)(2)(i) (as amended in the June 2016 final rule) details the minimum testing requirements for each basic model, specified by equipment category. In this SNOPR, DOE is proposing to further specify in that same paragraph that when a basic model spans listed categories, as in this example, multiple testing requirements apply. Therefore, the manufacturer would have to test at least one single-split-system combination as well as the model of outdoor unit with 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 (*i.e.,* test as an outdoor unit with no match). Under 10 CFR 429.16(c)(1)(i) (as amended in the June 2016 final rule), any other single-split combinations within the basic model may be tested or rated using an AEDM according to the applicable requirements. 81 FR 36001, 37049 (June 8, 2016).

In the event that DOE determines a basic model is noncompliant with an applicable energy conservation standard, DOE may issue a notice of noncompliance determination that, among other things, informs the manufacturer of its obligation to cease distribution of the basic model immediately. (10 CFR 429.114(a)) Therefore, if any individual combination (including the outdoor unit with no match) fails to comply with the applicable standard, whether the combination has been tested or rated using an AEDM, the entire basic model must be removed from the market and the model of outdoor unit may not be sold at all.

DOE also notes that although the discussion in this section of the SNOPR is directly related to refrigerants, a basic model may span listed categories in other situations. For example, as mentioned in the June 2016 final rule, a model of outdoor unit may be sold both as part of a single-split system and as part of a multi-split system. 81 FR at 37005. In this case, the manufacturer would have to determine represented values within each of these categories as required by 429.16(a)(1) and would have to meet the testing requirements for each category in 429.16(b)(2)(i). Furthermore, if an individual combination that is either a single-split or multi-split system fails to comply with the standard, the model of outdoor unit may not be sold for use in either category.

DOE also proposes to add information to the items required to be provided in certification reports to address outdoor units with no match. The general certification requirements for air conditioners and heat pumps as amended in the June 2016 final rule already apply to outdoor units with no match. These requirements include reporting of SEER, the average off mode power consumption, the cooling capacity, the region(s) in which the basic model can be sold, HSPF (for heat pumps), and EER (for air conditioners), and non-public information including indoor air volume rate for the relevant operating modes (*e.g.,* full-load cooling, part-load cooling, full-load heating). 81 FR 36991, 37053 (June 8, 2016). In this SNOPR, DOE proposes 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.

Issue 1: DOE requests comment on its proposed certification requirements for outdoor units with no match. Also, DOE seeks comment on what fin style options should be considered as options for CCMS database data entry.

6. Representation Limitations for Independent Coil Manufacturers

In the June 2016 final rule, DOE discussed compliance with Federal (base national or regional) standards for CAC/HP. Specifically DOE cited a proposal in the November 2015 SNOPR to amend 10 CFR 430.32 to clarify that the least-efficient combination within each basic model must comply with the regional SEER and EER standards. 80 FR 69277, 69290 (Nov. 9, 2015). However, DOE declined to modify section 430.32 in the June 2016 final rule, instead stating that it would do so in the regional standards enforcement rulemaking. 81 FR 36991, 37012 (June 8, 2016). Instead, 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.

In response to the June 2016 final rule, Advanced Distributor Products (ADP) and Lennox International submitted separate but essentially identical letters and AHRI submitted a similar letter (Docket No. EERE–2016–BT–TP–0029–0006, –0005, and –0003) stating that this language, while intended to define that ICM ratings cannot provide a means for an outdoor unit to span regions, is inconsistent with the Regional Standards ASRAC Working Group agreement (Docket No. EERE–2011–BT–CE–0077–0070). ADP, Lennox, and AHRI suggested that language proposed in the regional standards enforcement NOPR (80 FR 72389–72390), but not finalized, captured the enforcement working group intent and avoids inadvertent limitations on independent coil manufacturers. Mortex also submitted a letter (Docket No. EERE–2016–BT–TP–0029–0004) commenting on the same language, also stating that it seems inconsistent with agreements made during the Regional Standards ASRAC Working Group. Mortex suggested that the requirement be removed from the test procedure.

DOE did not adopt the language proposed in the regional standards enforcement NOPR in response to comments submitted in that rulemaking. DOE agrees, however, that the language adopted at 429.16 inadvertently constrains ICMs beyond the bounds agreed to in the Regional Standards ASRAC Working Group. Accordingly, DOE proposes 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 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.

Issue 2: DOE requests comment on its proposed language in 429.16 related to allowable ICM ratings and compliance with regional standards.

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

The current SEER and HSPF equations (4.1–1 and 4.2–1) in the DOE test procedure for a CAC/HP having a two-capacity compressor require different calculations of quantities depending on whether the test unit would operate at low capacity, cycle between low and high capacity, or operate at high capacity in response to the building load (see sections 4.1.3 and 4.2.3). To determine which calculations to use for units that lock out low capacity operation at higher outdoor temperatures, the outdoor temperature at which the unit locks out low capacity operation must be known. Section 4.1.3 of Appendix M indicates that this information must be provided by the manufacturer. Similarly, a two-stage heat pump may lock out low capacity heating operation below a certain lock-out temperature, as indicated in section 4.2.3 of Appendix M. Therefore, DOE proposes to add language to require that the lock-out temperatures for such systems for both cooling and heating modes be provided in the certification report.

8. Represented Values of Cooling Capacity

In the November 2015 SNOPR, DOE proposed adding a requirement that the represented values of cooling capacity and heating capacity must be the mean of the values measured for the sample. In response, AHRI, Lennox, JCI, Ingersoll Rand, Goodman, UTC/Carrier, Nortek, and Rheem disagreed with the requirement that the represented capacity values must be the mean of the tested values, and recommended that DOE allow manufacturers to rate capacity conservatively. (CAC TP: AHRI, No. 70 at p. 10; Lennox, No. 61 at p. 8, 15; JCI, No. 66 at p. 15–16; Ingersoll Rand, No. 65 at p. 5; Goodman, No. 73 at p. 15; UTC/Carrier, No. 62 at p. 8; Nortek, No. 58 at p. 6; Rheem, No. 69 at p. 8) The commenters provided additional detail as summarized in the

June 2016 final rule. 81 FR 37014–15 (June 8, 2016).

After reviewing the comments, in the June 2016 final rule DOE required the represented value of cooling (or heating) capacity to be a self-declared value that is no less than 95 percent of the mean of the cooling (or heating) capacities measured for the units in the sample selected for testing or of the output simulated by the AEDM. DOE stated that this would allow manufacturers the flexibility to derate capacity with conservative values as requested by multiple commenters, while still providing consumers with information that is reasonably close to the performance they may expect when purchasing a system. *Id.;* 10 CFR 429.16(b)(3) and 429.16(d).

Upon review, DOE has determined that the regulatory text adopted allows for unlimited overrating of capacity but only underrating of 5 percent. Consequently, in this SNOPR, DOE is proposing 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 that final rule, DOE would still use the mean of the measured capacities in its enforcement provisions.

Issue 3: DOE requests comment on its proposal to allow a one-sided tolerance on represented values of cooling and heating capacity that allows underrating of any amount but only overrating up to 5 percent.

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

In this SNOPR, DOE proposes revisions to appendix M to subpart B of 10 CFR part 430. This section provides a discussion of those proposed changes. DOE proposes to make these changes to Appendix M effective 30 days after publication of a final rule in the **Federal Register.** Representations related to the efficiency of CAC/HP basic models must be based on testing in accordance with the final rule procedures not later than 180 days following publication of the final rule.

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

DOE finalized an off-mode test procedure in the June 2016 final rule. 81 FR, 36991, 37022–5 (June 8, 2016). However, DOE recognizes that the current regulations may not account for excessive variation in the test results for units with self-regulating crankcase heaters or for units where the crankcase heater power measurement could be affected by the ambient temperature. These potential variations could be due to the large thermal mass of the compressor and the resulting time required for the compressor temperature to reach equilibrium. Because the power input of a self-regulating heater would depend on the compressor temperature, the test result would depend on the temperature of the unit just prior to the test. If conducted shortly after the B test, which is one of the steady-state wet coil cooling-mode tests conducted in an 82 °F ambient temperature, the compressor would still be quite warm, and the measured power input would be significantly lower than if the test were conducted after the compressor equilibrates with the surrounding space temperature. DOE proposes further revision to the test procedure to resolve this issue. The proposal in this section would not impact the measured off-mode power input beyond potentially reducing variation in the measured result.

In the off-mode test procedure established in the June 2016 final rule, DOE established a test method for units with self-regulating crankcase heaters that called for start of the test in a room conditioned to 82 °F temperature, with the compressor at a temperature no lower than 81 °F. The room temperature is then adjusted at a rate of change of no more than 20 °F per hour to approach 72 °F for conducting a first heater power measurement, and then to approach a manufacturer-specified lower temperature, again at a rate of change no more than 20 °F per hour, before conducting the second power measurement. 81 FR at 37022 (June 8, 2016). A half-hour duration in the initial reduction in room temperature from 82 °F to 72 °F would be compliant with the prescribed 20 °F maximum temperature reduction rate. However, DOE testing shows that the time constant for compressor cooldown, or for approach to equilibrium of the power input a self-regulating crankcase heater attached to a compressor, is much longer than a half-hour. This issue would be exacerbated if the compressor has a sound blanket. Self-regulating crankcase heaters draw less power when they are warmer. Hence, if the temperature cooldown from 82 °F is initiated when the compressor is hot (*e.g.,* after running the B test), the compressor will still be very warm when the test is conducted, and the measured power input will be lower than for a test initiated with a compressor at the minimum 81 °F.

To determine the reasonable delay time for units to reach thermal equilibrium, DOE conducted tests using a 5-ton residential condensing unit. DOE connected a self-regulating crankcase heater to the compressor and measured heater power input, compressor shell temperature, and ambient temperature. DOE observed cooldown behavior and the corresponding increase in heater input power in a 60 °F environment both with and without a sound blanket covering the compressor after initially preheating the compressor to 120 °F to simulate warmup associated with refrigeration system operation. DOE used an exponential equation for the power input to the heater as a function of time to fit to the test data. The time constant for approach to equilibrium (time for the difference between the power input and the value it would attain after an infinite amount of time to drop by 63 percent) DOE observed in the tests was approximately 2 hours for tests without the sound blanket (bare shell) and 4 hours for tests with the sound blanket. DOE also observed that the crankcase heater power input generally approached to within 10 percent of its final value after passage of about two time constants (4 hours for bare-shell testing and 8 hours for sound blanket testing).

Based on the testing and analysis described in this preamble, DOE proposes adopting a time delay for testing units with self-regulating crankcase heaters or crankcase heating systems in which the heater control temperature sensor is affected by the heater. DOE proposes a 4-hour time delay for units where the compressors have no sound blanket, and an 8-hour time delay for units where the compressors do have sound blankets. The delay would take place after the room temperature reaches the lower target value and before making each of the power measurements (P_{1x} and P_{2x}). Also, the proposal would eliminate the 20 °F per hour room temperature reduction rate limit for any unit where ambient temperature can affect the measurement of crankcase heater power because the roughly half hour required for the temperature to transition at this rate from 82 °F to 72 °F would add unnecessarily to the compressor's equilibration time—equilibration would occur sooner if the ambient temperature more quickly drops to the final value rather than approaching it slowly.

Issue 4: DOE seeks comments from interested parties about its proposal to impose time delays to allow approach to equilibrium for measurements of off-mode power for units with self-regulating crankcase heaters. DOE requests comment regarding the 4-hour and 8-hour delay times proposed for units without and with compressor sound blankets, respectively.

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

In DOE's current test procedures at Appendix M, refrigerant pressure measurement is required when using the refrigerant enthalpy method as the secondary capacity measurement (see section 2.10.3 of 10 CFR part 430, subpart B, appendix M). Refrigerant pressure measurement is also required for some methods for setting or confirming refrigerant charge (see section 2.2.5 of 10 CFR part 430, subpart B, appendix M), unless otherwise instructed by the manufacturer's installation instructions.

DOE is aware that the pressure measurement devices may be installed at a location where the refrigerant state switches between liquid and vapor under different cooling and heating modes. In this case, the actual refrigerant charge in the unit could be different under different modes due to the transfer of refrigerant to and from the extra internal volumes in the refrigerant pressure lines, connections, and transducers or gauges.

DOE is also aware that the refrigerant charge in pressure measurement systems may affect cyclic testing. In a cooling test, the liquid refrigerant in the liquid refrigerant pressure measurement system is cooler than the refrigerant in the condenser. For a system with a fixed orifice expansion device, allowing the cooler refrigerant from the pressure measurement systems to flow into the evaporator before the fan delay ends could affect the cyclic performance.

These issues have the potential to impact test reproducibility and repeatability, in particular for small capacity mini-split heat pump systems with low system refrigerant charges, depending on the differences in internal volumes of the tubing, connections, and transducers, particularly from one laboratory to the next.

As part of the compressor calibration method, ASHRAE 37–2009 section 7.4.2 provides instructions for making refrigerant pressure measurements. For equipment not sensitive to refrigerant charge, the pressure measurement instruments may be connected via pressure measurement lines to the refrigerant lines without requiring that any preliminary tests be conducted to confirm that displacement of refrigerant into the pressure lines does not affect performance. The test standard sets a threshold for sensitivity to refrigerant charge, indicating that for equipment that is not sensitive to the charge, the refrigerant pressure lines must not affect the total charge by more than 0.5%.

To limit the amount of refrigerant charge that can transfer to and from the pressure measurement system, DOE proposes to require manufacturers to limit the total internal volume of pressure lines and pressure measurement devices connected at locations that can switch states from liquid to vapor for different operating modes or conditions. Based on the ASHRAE 37–2009 precedent, DOE selected a maximum internal volume connected at these locations that would represent at most 0.5 percent of the total system charge for the lowest-charge systems for which DOE collected information. The proposed maximum total internal volume of the pressure lines, connections and gauges would be 0.25 cubic inches per 12,000 Btu/hr certified cooling capacity. DOE selected this maximum volume based on a survey of refrigerant charge in mini-split heat pumps with capacities ranging from 9,000 to 33,000 Btu/hr.

DOE notes that the charge adjustment approach prescribed by ASHRAE 37–2009 for systems that are sensitive to refrigerant charge would not resolve the issue of displacement of refrigerant into the pressure lines because that approach is based on steady-state testing, for which the displaced refrigerant would remain in the lines. The required adjustment would add that same amount of refrigerant so that the charge actively circulating in the refrigerant circuit would be the same as if no pressure lines had been connected. In the present case, where refrigerant would be displaced between heating and cooling mode or between cycles of a cyclic test, simply adding the ''missing'' charge would not resolve the issue.

The internal volume of pressure measurement lines and connections can be determined using the tubing inner diameter or internal volume values found on pressure gauge or transducer manufacturer specification sheets. However, DOE is aware that the manufacturer specification sheets may not provide the internal volume of pressure gauges or pressure transducers, and they may not be easy to measure. Thus, DOE proposes to use 0.1 cubic inches as the default internal volume for each pressure transducer and 0.2 cubic inches for each pressure gauge, if internal volume is not provided in specification sheets. DOE proposes to include this requirement in section 2.2 of 10 CFR part 430, subpart B, appendix M.

Issue 5: DOE requests comment on its proposal to limit the internal volume of pressure measurement systems for cooling/heating heat pumps where the pressure measurement location may switch from liquid to vapor state when changing operating modes and for all systems undergoing cyclic tests. DOE also requests comment specifically on (a) the proposed 0.25 cubic inch per 12,000 Btu/h maximum internal volume for such systems, and (b) the proposals for default internal volumes to assign to pressure transducers and gauges of 0.1 and 0.2 cubic inches, respectively.

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

In the current DOE test procedure specified in section 3.2.4 and 3.6.4 of 10 CFR part 430, subpart B, appendix M, the building load is determined as a function of temperature, for both cooling and heating. Units equipped with variable speed compressors are tested at full, intermediate and minimum speeds. In calculating SEER and HSPF for variable speed units, there are three possible scenarios: (a) When the building load requires less than the minimum-speed capacity, the unit cycles at the minimum compressor speed to meet the load; (b) when the load requires more than the maximum-speed capacity, the unit operates constantly at full load; and (c) when the unit operates at an intermediate speed to meet a building load that is between the minimum-speed and maximum-speed capacities. Three outdoor temperatures are calculated for cooling and/or heating units equipped with variable speed compressors to bound the conditions in which scenario c would apply. These three outdoor temperatures are the balance points (temperatures at which the building load and delivered capacity are equal) for operation at the tested minimum, intermediate, and full compressor speeds. For all variable speed units operating in cooling mode and non-multi-split variable speed units operating in heating mode, the unit's EER and COP are calculated using quadratic functions. These quadratic functions are determined based on the EER or COP evaluated for the three calculated outdoor temperatures representing the minimum, intermediate, and full speed balance points.

In a final rule published October 22, 2007, DOE adopted a different approach for multi-split heat pumps. 72 FR 59906 (October 2007 Final Rule). DOE determined in that final rule that the quadratic fit would not be well-suited for multi-split units because the intermediate speed initially defined for variable-speed units is not likely the peak efficiency point for multi-split units. (see 71 FR 41320, 41325 (July 20, 2006)). In addition to allowing multi-split manufacturers some flexibility in selecting intermediate speeds for testing, DOE also adopted in the October 2007 final rule a two-piece linear relationship to represent EER and COP vs. temperature, rather than the quadratic fit used for other variable-speed units. 72 FR 59906 (Oct. 22, 2007).

As discussed in section III.C.3.d, AHRI provided variable speed and two stage heat data (under a Non-Disclosure Agreement to DOE's contractor) to allow evaluation of the impact on the HSPF differential associated with the new heating load line equation. In reviewing AHRI's variable speed heat pump heating test data, DOE's contractor discovered that the quadratic interpolation in some cases provides very poor estimation of COPs in the intermediate-speed operating range—in some cases predicting higher or lower COP values than all of the measured COP results. DOE has found similar issues with prediction of the cooling EER using the quadratic function, although DOE has less cooling mode data to review, and the most egregious errors in EER prediction for cooling mode are not as bad as the observed COP errors. Nevertheless, DOE believes such issues could very well cause significant errors in calculation of SEER for variable-speed units.

In this SNOPR, DOE evaluated two alternative interpolation methods for calculating SEER and HSPF for variable-speed CAC/HP in addition to the current quadratic function approach: (1) The linear interpolation method which currently applies only to multi-split units in heating mode (section 4.2.4.2 of 10 CFR part 430, subpart B, appendix M); and (2) a bin-by-bin interpolation method. The bin-by-bin method uses 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). Under the bin-by-bin method, an interpolation factor is first calculated, which represents the compressor operating speed needed to achieve balance between house load and delivered capacity. For example, if, for the specific temperature bin, the heating load is between the minimum-speed capacity and the intermediate-speed capacity, the interpolation factor is equal to the difference between the heating load and the minimum-speed capacity divided by the difference between the intermediate-speed capacity and the minimum-speed capacity. This factor is then applied to the COP values to determine COP when operating at the speed needed to deliver the desired heating load. The desired load is divided by this COP to determine power input. The interpolation is between the minimum speed and the intermediate speed performance values if the load is between the minimum and intermediate-speed capacities, or between the intermediate speed and the full speed performance values, if the load is between the intermediate and full speed capacities.

DOE found that HSPFs calculated with the current quadratic method deviated from HSPFs calculated using the bin-by-bin method up to 7.4 percent and the linear interpolation method deviated up to 2.9 percent from the bin-by-bin method. Calculations conducted for cooling mode SEER showed that SEER for the quadratic method deviated from the SEER calculated for the bin-by-bin method up to 2.5 percent. DOE believes that the bin-by-bin interpolation method is the most accurate of the three approaches (*i.e.,* DOE's current quadratic approach and the two alternative approaches considered for this SNOPR), because it is based on the best estimates of performance at the different compressor speeds for the specific ambient temperature considered for each bin. Hence, DOE proposes to require use of the 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. Because DOE believes that the bin-by-bin method is the most accurate, DOE does not propose for all variable-speed systems to adopt the linear approach currently used for multi-split systems. DOE would implement this change by revising the intermediate speed EER and COP equations in section of 4.1.4.2 and 4.2.4.2 of appendix M of 10 CFR part 430 subpart B.

Issue 6: DOE requests comment on the proposal to require the use of a bin-by-bin method to calculate EER and COP for intermediate-speed operation for SEER and HSPF calculations for variable-speed units.

4. Outdoor Air Enthalpy Method Test Requirements

In DOE's current test procedure in Section 2.10 of appendix M to subpart B of part 430, the outdoor air enthalpy method is an allowable secondary test method for split systems and single-package units. DOE currently requires that the outdoor air-side test apparatus be connected to the outdoor unit and used for measurements for the outdoor air enthalpy method during the ''official'' test. Additionally, DOE requires a preliminary test be conducted prior to conduct of the official test, in which the unit operates without the outdoor air-side test apparatus connected. After operating without the apparatus, the apparatus is connected, and the apparatus exhaust fan speed is adjusted until performance is verified as consistent with performance prior to attaching the apparatus. Specifically, the unit must operate for 30 minutes without the apparatus connected, followed by at least five consecutive readings with the apparatus connected (with measurements taken at one-minute intervals). The apparatus exhaust fan speed must be adjusted so that the averages for the evaporator and condenser temperatures, or the saturated temperatures corresponding to the measured pressures, agree within ± 0.5 °F between the tests with and without the apparatus connected. Additionally, a preliminary test is only required prior to the first steady-state cooling mode test and the first steady-state heating mode test, as long as the outdoor fan operates during all cooling mode steady-state tests at the same speed and during all heating mode steady-state tests at the same speed. However, the test procedure requires that a preliminary test be conducted prior to each cooling mode test where a different fan speed is used, and a similar requirement applies for heating mode tests.

The outdoor air enthalpy method includes two steps in order to verify the capacity determined from the indoor air enthalpy method during the official test. However, DOE is concerned that the tolerances on achieving the same condensing and evaporating conditions in the tests with and without the airflow measurement apparatus attached inherently introduces variability to the test results that could be eliminated by shifting to an official test with the apparatus not attached. DOE proposes to make such a change for the official test.

In this SNOPR, DOE proposes to require two-step measurements in the outdoor air enthalpy method only for cooling and heating mode tests that currently require preliminary tests (*i.e.,* the first cooling mode and heating mode tests, and any cooling mode and heating mode tests where a different outdoor fan speed is used). For example, if the unit uses a different outdoor fan speed for each test, the two-step approach would be required for each test condition. On the other hand, if the unit is a single-capacity unit and the outdoor fan uses the same fixed speed for all tests, the two-step approach would be required only for the A and H1 tests. DOE proposes that for all cooling and heating mode tests, a 30-minute test be conducted without the outside-air apparatus connected (''non-ducted'' test). For tests that do not require measurements for the outdoor air enthalpy method, this 30-minute test non-ducted test would constitute the official test. For tests that do require measurements using the outdoor air enthalpy method, DOE proposes to maintain the current approach, except for changing designation of what constitutes the official test. First, the current 30-minute preliminary test would be conducted without the outside-air apparatus attached (now the ''non-ducted'' test). Next, the outside-air apparatus would be attached. For this test, now termed the ''ducted'' test, the airflow would be adjusted so that condensing and evaporating conditions are matched within tolerances, and five consecutive readings would be required (as is required for the current test) to verify the primary capacity measurements. For the tests that require measurements using the outdoor air enthalpy method, DOE proposes that the following conditions must be met for the test to be considered valid:

(1) The energy balance specified in section 3.1.1 of appendix M to subpart B of part 430 is achieved for the ducted 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 and non-ducted tests cannot deviate more than 2.0 percent.

If the test is valid, the non-ducted test would be used as the official measurement for the specific test condition.

DOE believes that use of the outdoor air enthalpy method for only certain tests sufficiently measures and verifies the capacity determined from the indoor air enthalpy method, and that losing the benefit of two-step verification of the capacity determined during all of the official tests is outweighed by the three following benefits to DOE's proposal:

• *Better Representativeness of Field Use.* First, attachment of an apparatus for measurements for the outdoor air enthalpy method inherently affects the airflow pattern for the condenser (for example, by blocking any potential for partial recirculation of condenser discharge air to the inlet) and adds external static pressure for the outdoor fan to overcome. While DOE's procedure requires adjustment of apparatus exhaust fan speed to achieve similar performance to operation without the outdoor air-side apparatus, there is still a tolerance on this deviation in performance. Also, it may be impossible to exactly match no-discharge-duct performance—for example, if the discharge duct blocks partial air recirculation, total condenser fan airflow may have to be reduced to achieve the same condensing temperature, thus altering the condenser fan operating point. Therefore, DOE believes that removal of the requirement to connect the outdoor air-side test apparatus during the official test would allow for performance that better matches performance in the field.

• *Improved Test Reproducibility and Repeatability.* Second, to maintain similar performance to operation without the outdoor air-side apparatus, DOE currently requires that the apparatus exhaust fan speed be adjusted. Specifically, the averages for the evaporator and condenser temperatures, or the saturated temperatures corresponding to the measured pressures, must agree within ± 0.5 °F of the averages achieved when the apparatus was disconnected. However, if the outdoor air-side apparatus is connected during the official test, two different test labs could measure evaporate and condenser temperatures that differ by up to 1.0 °F when testing the same unit. This variation could, in turn, affect the measured cooling and/or heating capacity of the unit, and therefore would change the calculated SEER and/or HSPF. DOE believes that removing the ducted test requirement from the official test would reduce this variation in performance and therefore improve the reproducibility and repeatability of its test procedure.

• *Reduced Test Burden.* Third, for cooling mode and heating mode tests requiring a preliminary test, DOE's current test procedure requires a 30-minute non-ducted test and 5-minute ducted test be conducted as part of the preliminary test, in addition to the 30-minute official test. However, in DOE's proposal, separate 30-minute tests would not be required for the preliminary and official tests—only a single 30-minute non-ducted test would be performed as the official test, assuming the required tolerances and test conditions are met. DOE expects this removal of a required test to reduce the burden of testing units with the outdoor air enthalpy method as a secondary method.

Issue 7: DOE requests comment on its proposed modifications to requirements when using the outdoor air enthalpy method as the secondary test method, including its proposal that the official test be conducted without the outdoor air-side test apparatus connected.

5. Certification of Fan Delay for Coil-Only Units

In the cyclic dry-coil cooling-mode tests, the current regulatory text requires coil-only units to be tested with a time-delay relay. Section 3.5.1 of the current Appendix M states that the automatic controls that are normally installed with the test unit must govern the OFF/ON cycling of the air moving equipment on the indoor side. (10 CFR 430 Subpart B, App. M, 3.5.1) Under that section, the manufacturer is to control the indoor coil airflow for ducted coil-only units according to the rated ON and/or OFF delays provided by the relay. However, DOE understands that in typical installations, a time-delay relay, if it exists, would be part of the furnace function. DOE reviewed furnace product literature collected during the furnace fan rulemaking (see Docket Number EERE–2010–BT–STD–0011) representing a broad range of furnaces sold by major furnace manufacturers to determine whether they have time-delay relays available for cooling mode when installed with coil-only air conditioners. DOE found that in many furnace series, both old and new, from multiple manufacturers, cooling time delays are common, but they are exclusively used for the compressor off-cycle, and they have varying time-delay durations. Thus, DOE concludes that coil-only units are likely to be installed with time-delay relay control for cooling, but that the duration of the delay varies by furnace. DOE is proposing no change in the use of time delays for testing of coil-only units, but proposes to amend its certification report requirements to require coil-only ratings specify whether a time delay is included, and if so, the duration of the delay used. DOE would use the certified time delay for any testing to verify performance. Section 3.5.1 would indicate that the time delay used for testing of a coil-only system shall be as listed in the certification report.

Issue 8: DOE requests comments on its proposal to require certification reports for coil-only units to indicate whether testing was conducted using a time-delay relay to provide an off-cycle time delay, and the duration of the time delay.

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

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)) DOE is aware that many potential combinations of single-split-system condensing units and indoor coils could be tested even if they are not typically installed as a combination. Ratings of single-split-system coil-only combinations, for which the outdoor unit and indoor unit are not typically installed as a combination, would not be representative of an average use cycle. The CAC/HP ECS Working Group discussed this concept and the potentially undesirable impacts of rating combinations that are not distributed in commerce or installed for consumers. Specifically, the CAC/HP ECS Working Group addressed ratings based on a combination using a blower coil indoor unit consisting of a low-efficiency condensing unit paired with an indoor blower with unusually low input power, a concept the participants referred to as a ''golden blower.'' Such a combination would result in an inflated rating for a low-efficiency condensing unit that is not representative of its typical installed performance. (CAC ECS: ASRAC Public Meeting, No. 87 at p. 88) The concept of unrepresentative, high performance can apply to other design aspects of indoor units, such as units with an indoor coil size far larger than would be installed for the given system capacity. To help ensure that the test procedure results in ratings that are representative of average use, DOE proposes 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 proposes to limit the normalized gross indoor fin surface (NGIFS) for the indoor unit used for single-split-system coil-only tests be 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. An NGIFS greater than 2.0 sq.in./Btu/hr indicates that the system combines a low-capacity condensing unit with a high capacity indoor coil, *e.g.,* a 1.5-ton condensing unit paired with a 5-ton indoor coil. First, a house requiring a 1.5-ton air conditioner would be expected to have a commensurately-sized furnace, and a much larger indoor coil may not fit with the furnace or the existing available space. Second, such a combination might have good rated efficiency, but would provide poor dehumidification performance, due to the elevation of coil surface temperature (potentially above incoming air dew point temperature) associated with the large coil surface area. Because of the size compatibility and poor dehumidification performance, DOE understands that systems with an NGIFS greater than 2.0 sq.in/Btu/hr are not typically installed.

DOE evaluated the NGIFS for a representative data set of single-split-system coil-only combinations currently offered in the market to set this value. DOE's dataset included close to 100 two, three, and five-ton single-split-system coil-only combinations from multiple manufacturers that represent a majority of market share and span the available range of efficiency. Testing with a NGIFS no greater than 2.0 sq.in/Btu/hr would still reflect approximately 95 percent of the split-system coil-only combinations reviewed by DOE. DOE understands a single-split-system coil-only combination with an NGIFS that exceeds 2.0 sq.in/Btu/hr to be unrepresentative because it is unlikely to be distributed in commerce, which is supported by the review of NGIFS values for numerous rated combinations, as noted previously.

Issue 9: DOE requests comment on its proposal to limit the NGIFS of tested coil-only single-split systems to 2.0 sq.in/Btu/hr.

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

In the November 2015 SNOPR, DOE proposed several changes to the test procedure for variable-speed heat pumps. First, DOE proposed that the maximum compressor speed used for the test be fixed at the absolute maximum speed at which the compressor operates for the given operating mode (heating or cooling). In other words, the maximum compressor speed used in different cooling mode test conditions would be the same, equal to the absolute maximum speed used for cooling at any operating condition. DOE proposed a similar approach for heating, allowing for a different maximum speed than for cooling. 80 FR at 69307 (Nov. 9, 2015).

The June 2016 final rule discussed comments on this proposal, several of which indicated that the compressors of variable speed heat pumps very often operate at higher speeds at colder temperatures, which can enhance measured HSPF. 81 FR at 37029 (June 8, 2016). The comments indicated that for some of these heat pumps, the compressor cannot operate in a 47 °F ambient temperature at the same full speed that it uses in a 17 °F ambient temperature. Although DOE did not in that final rule modify the test procedure to allow different compressor speeds for the full-speed tests conducted at 17 °F, 35 °F, and 47 °F ambient temperatures, DOE did acknowledge that addressing this issue would improve the test method's representation of the improved performance of variable speed heat pumps that use higher speeds at lower temperatures, indicating that consideration would be given to such a test procedure revision in the future.[7] *Id.* In this SNOPR, DOE proposes such a test procedure revision.

The possible adoption of a 2 °F test for rating of variable speed heat pumps was proposed in the November 2015 SNOPR. 80 FR 69323 (Nov. 9, 2015) It was also discussed during the CAC/HP ECS Working Group meetings, ultimately leading to Recommendation #5 in the Term Sheet, that a 5 °F ambient temperature optional test be adopted for variable speed heat pumps under the new Appendix M1. (CAC ECS: ASRAC Term Sheet, No. 76 at p. 3) This proposed revision is discussed in greater detail in section III.C.4. Because the Appendix M1 test procedure changes would be required as the basis for efficiency representations on the effective date of any new energy conservation standards (January 1, 2023), the 5 °F test for variable speed heat pumps would not become an option for several years. Based on the stakeholder comments discussed in this preamble, some variable-speed heat pumps may be unable to operate as required by the appendix M procedure as finalized by the June 2016 final rule. In order to resolve this issue sooner than 2021, DOE proposes that the test procedure revisions to address it be adopted in appendix M rather than appendix M1. Hence, DOE proposes 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)

[7] The June 2016 final rule also changed the terminology for the highest compressor speed from ''maximum speed'' to ''full speed,'' as requested by several comments responding to the November 2015 SNOPR. 81 FR at 37030 (June 8, 2016).

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. In the current proposal, 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 is and 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 test procedure would not provide for this situation.
- If no 47 °F full-speed test is 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.

Development of these proposals and decisions regarding their details is explained further below.

As discussed in the June 2016 final rule, DOE believes that extrapolations of performance to lower temperatures should be based on tests conducted at the same speed and used to estimate performance where there is a good expectation that the speeds are also the same or at least not very different. Hence, DOE believes that calculation of performance below 17 °F must be based on a same-speed extrapolation (or on an interpolation using measurements for a lower-temperature test, such as for the proposed 5 °F test discussed in section III.C.4). For those heat pumps which cannot operate in the 47 °F ambient temperature at the same compressor speed used for the 17 °F full-speed test, DOE proposes use of average performance trends to represent the 47 °F test point so that a representative same-speed extrapolation can be done.

DOE evaluated the 17 °F-to-47 °F same-speed performance trends of heat pumps based on several sources including the AHRI database, data for two stage and variable speed heat pumps provided to DOE's contractor by AHRI during the CAC/HP ECS meetings, and product data sheets for 51 single-package heat pumps. The ratios for capacity and power input for the 17 °F test condition as compared to the 47 °F test condition are presented in Table III.4. The AHRI database provides capacity information for both 17 °F and 47 °F test conditions, but not power input for both. DOE did not consider variable speed models from the AHRI database in this analysis because of questions about whether the compressor speeds were the same for both test conditions for tests of these units. For the data provided by AHRI during the CAC/HP ECS meetings, DOE evaluated the two stage units and the variable speed units with a capacity ratio within a narrow range, to be sure that the results for these units were based on use of the same speed for both test conditions. Evaluation of the data for single-package units shows that they have a significantly lower capacity ratio, but roughly the same power input ratio, as compared with split systems. Consequently, DOE is proposing in this SNOPR a different standard capacity slope factor for single-package units.

TABLE III.3—AVERAGE HEAT PUMP CAPACITY AND POWER INPUT RATIOS FOR 17 °F AND 47 °F TESTS

Data source	Capacity ratio (17 °F vs. 47 °F)	Power input ratio (17 °F vs. 47 °F)
AHRI Database, Single-Stage and Two stage	0.618	Not available.
Split-System		
Single-Package	0.558	
Data Provided by AHRI During ASRAC Meetings:		
Two stage	0.623	0.886.
Variable speed*	0.637	0.875.
Data Sheets for Single-Package Units	0.557	0.874.

*Just for VS units with capacity ratio between 0.59 and 0.67, indicating high probability that compressor speed was the same for both 17 °F and 47 °F tests.

Based on the reviewed data, DOE selected capacity ratios equal to 0.62 for split systems and 0.56 for single-package units in order to calculate capacity slope factors. Also, DOE selected 0.88 as the power input ratio to use for calculating the power input slope factor. DOE proposes adopting slope factors that would be multiplied by the capacity or power input measured for the 17 °F ambient temperature in order to obtain the slope of the evaluated parameter per degree temperature rise. For example:

$$\text{Capacity Slope} = \dot{Q}_h^{k=2}(17) * \text{CSF}$$

Where:

Capacity Slope is the change in capacity per change in temperature in Btu/h-°F,

$\dot{Q}_h^{k=2}(17)$ is the capacity measured in the $H3_2$ Test in Btu/h, and

CSF is the Capacity Slope Factor in 1/°F.

The CSF is calculated from the selected capacity ratio as follows:

$$CSF = \frac{1 - CR}{(30°F) * CR}$$

Where CR is the capacity ratio.

The resulting values for the capacity slope factors are 0.0204/°F for split systems and 0.0262/°F for single-package systems. DOE adopted a similar approach for development of the Power Slope Factor (PSF), which is calculated to be 0.00455/°F for all systems.

DOE proposes use of these slope factors for any variable speed heat pumps for which the 47 °F full-speed test cannot be conducted at the same speed (represented by RPM or power input frequency) used in the 17 °F full-speed test. The slope factors would be used for calculation of representative capacity and power for operation at 47 °F ambient temperature for the purposes of calculating HSPF.

As mentioned in this preamble, DOE proposes that the 17 °F test be conducted using the maximum speed at which the system controls would operate the compressor during normal operation in this ambient temperature. This would help to ensure that the test procedure be representative of field operation, since, for cold temperatures close to 17 °F, the heat pump would be expected to be operating at full speed to satisfy the high heating loads expected for these temperatures. Further, DOE proposes that the 35 °F full-speed test, if conducted, use the same compressor speed as the 17 °F test, so that the impact of frosting and defrost for this test is not masked by an adjustment in compressor speed.

Issue 10: DOE requests comments on its proposal to require that full-speed tests conducted in 17 °F and 35 °F ambient temperatures use the maximum compressor speed at which the system controls would operate the compressor in normal operation in a 17 °F ambient temperatures. DOE requests comment on the proposed approach of using standardized slope factors for calculation of representative performance at 47 °F ambient temperature for heat pumps for which the 47 °F full-speed test cannot be conducted at the same speed as the 17 °F full-speed test. Further, DOE requests comment on the specific slope factors proposed, and/or data to show that different slope factors should be used.

In addition, DOE proposes 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. If the $H1_N$ test does 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. DOE proposes that the 47 °F full-speed test used to represent heat pump capacity would use the same maximum compressor speed that the control system would use during normal operation in 47 °F ambient temperatures in Appendix M1 (see section III.C.4) However, proposing flexibility in the selection of compressor speed for the test would be more consistent with the recent approach for measuring nominal heating capacity (prior to publication of the June 2016 final rule) because compressor speed requirements on the $H1_2$ test may not have been clearly defined at that time (see Appendix M to subpart B of part 430 as of January 1, 2016).

Issue 11: DOE requests comments on its proposal to allow the full speed test in 47 °F ambient temperature that is used to represent heat pump heating capacity, to use any speed that is no lower than used for the 95 °F full-speed cooling test for Appendix M.

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

In the June 2016 final rule, DOE maintained its proposal from the November 2015 SNOPR to allow manufacturers the option of specifying a break-in period to be conducted prior to testing under the DOE test procedure. DOE limited the optional break-in period to 20 hours, which is consistent with the test procedure final rule for commercial HVAC equipment (10 CFR 431.96). The duration of the compressor break-in period, if used, must be included in the certification report for CAC/HP (10 CFR 429.16). DOE also adopted the same provisions as the commercial HVAC rule regarding the requirement for manufacturers to record the use of a break-in period and its duration as part of the test data underlying their product certifications, the use for testing conducted by DOE of the same break-in period specified in product certifications, and use of the 20 hour break-in period for DOE testing of products certified using an AEDM. 81 FR at 37033 (Jun. 8, 2016).

Section 3.1.7 of Appendix M, ''Test Sequence'' indicates that manufacturers have the option to operate the equipment for a break-in period on to exceed 20 hours, and that this break-in period must be recorded in the test data underlying the certified rating if the manufacturer uses a break-in period. DOE has made reporting of the break-in period a certification report requirement. 81 FR at 37053 (June 8, 2016). Hence, the instructions to record the break-in period in the test report is not necessary in section 3.1.7. Also, DOE intends that tests conducted by third-party testing facilities should use the break-in period that is certified and proposes to modify the language to clarify that the certified break-in period is used for the test (whether conducted by a manufacturer or other party). DOE also proposes to clarify that each compressor should undergo the break-in according to the certified number of hours, for units with multiple compressors. Finally, DOE proposes to clarify that the break-in period should be conducted prior to the first 30 minutes test data collection period as required by the test methods in section 3 of Appendix M.

Issue 12: DOE requests comments on its clarifications regarding use of break-in, including use of the certified break-in period for each compressor of the unit, regardless of who conducts the test, prior to any test period used to measure performance.

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

In addition to the adopted portions of the AHRI Standard 1230–2010, DOE proposed additional provisions in the November 2015 SNOPR for testing of VRF Multi-Split Systems. This included a provision adopted as part of section 2.2.3.a of Appendix M in the June 2016 final rule requiring that for part load tests, the sum of the nominal heating or cooling capacities of the operational indoor units be within 5 percent of the intended system part load heating or cooling capacity. 81 FR at 37066 (June 8, 2016). DOE recognizes the intended system part load heating or cooling capacity is not clearly defined in the test procedure and that the sum of nominal capacities of the indoor units may very well be higher than the system part load capacity during the test (since the indoor units would be expected to be operating at part load, less than their nominal capacity, during a part load test). Therefore, DOE proposes to remove this 5 percent tolerance requirement.

Issue 13: DOE requests comments on removing from section 2.2.3.a of Appendix M the 5 percent tolerance for part load operation when comparing the sum of nominal capacities of the indoor units and the intended system part load capacity.

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

The June 2016 final rule provided instructions in 2.2.c of Appendix M for uncased coils, including instructions regarding the addition of internal insulation and/or sealing consistent with manufacturer's instructions. The section ends with a requirement that no extra insulating or sealing is allowed for cased coils. This statement was intended to indicate that no extra internal insulating or sealing is allowed. DOE believes that the statement as it stands may suggest that sealing is not allowed between a cased coil and its connections to inlet and outlet ducts. To prevent such confusion, DOE proposes to remove the statement about cased coils.

Issue 14: DOE requests comment on whether removing the statement about insulating or sealing cased coils in Appendix M, section 2.2.c would be sufficient to avoid confusion regarding whether sealing of duct connections is allowed.

C. Appendix M1 Proposal

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 69278, 69397 (Nov. 9, 2015) In this SNOPR, DOE also proposes to establish a new Appendix M1. The appendix would include all of the test procedure provisions in Appendix M as finalized in the June 2016 final rule, all of the proposed changes to Appendix M that are discussed in section III.B, and all of the additional proposals discussed in this section III.C, which would be included only in the new Appendix M1. DOE proposes 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 that phase-in of testing requirements for certain proposed new requirements for split systems would be as discussed in section III.A.1).

1. Minimum External Static Pressure Requirements

Most of the CAC/HP 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 [8] 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.[9] 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 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 69317–18 (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.[10] 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.[11] 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 this data. 80 FR 69318–19 (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, Lennox supported DOE's proposal to increase the minimum test static pressure to more accurately reflect field installation conditions. Lennox recommended that this level be set to 0.50 in. wc. for all capacities, commenting that the single set point simplifies the test procedure, is consistent with levels found in field studies, and avoids compliance issues related to minimum static pressure settings based upon capacity. (CAC TP:

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

[9] 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).

[10] 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.

[11] 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.

Lennox, No. 61 at p. 11) Lennox also commented that improvements in field practices to reduce installed static pressure in parallel with optimizing products for lower static pressures are a more effective measure to optimize field performance and reduce energy consumption. Lennox commented that products optimized for increased static pressures will likely result in increased energy consumption. (Lennox, No. 61 at p. 11) Unlike Lennox, Rheem did not agree in its comments that the assumption of poorly designed ductwork should be built into the test procedure. (CAC TP: Rheem, No. 69 at p. 16)

Many interested parties supported the proposal to increase the external static pressure requirement. NEEA and NPCC commented that the minor adjustments on either side of 0.50 in. wc. on the basis of system capacity would be a needless complication of the test procedure because NEEA and NPPC's field data does not suggest any correlation between the external static pressure a system faces and the system capacity. (CAC TP: NEEA and NPCC, No. 64 at p. 8) The California IOUs recommended that all capacities use 0.50 in. wc. to simplify testing. (CAC TP: California IOUs, No. 67 at p. 2) ACEEE, NRDC, and ASAP fully supported adopting 0.50 in. wc. for all units (in blower coil configuration), as 0.5 in. wc. would be closer to the levels found in thousands of residential duct systems tested. (CAC TP: ACEEE, NRDC, ASAP, No. 72 at p. 4)

Lennox and Rheem commented that DOE's assumption that a CAC system would be poorly maintained, such as containing fouled coils and filters, should not be built into the test procedure. (CAC TP: Lennox, No. 61 at p. 19; Rheem, No. 69 at p. 16) Lennox further commented that any accommodation for poor field conditions should be administered equitably across all product types. (CAC TP: Lennox, No. 61 at p. 19) Rheem also commented that although dirty filters and fouled coils can increase system static, Rheem considers undersized duct work as the leading cause of high pressure drop measured in field applications. (CAC TP: Rheem, No. 69 at p. 16) Rheem believed that requiring higher minimum external static pressure would reduce published ratings, which could confuse installers and consumers. Rheem commented that a new energy metric should be introduced that would distinguish ratings based on appendix M from ratings based on appendix M1. The California IOUs commented that, as shown in the ACCA Manual D,[12] the filter pressure drop value of 0.20 in. wc. is normal, and supported DOE's proposal. (CAC TP: California IOUs, No. 67 at p. 6)

After discussions that included the concerns from the comments summarized previously in this section, 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 comments in response to the November 2015 SNOPR, Unico supported the values discussed during the ASRAC meetings. (CAC TP: Unico, No. 63 at p. 12) JCI and Carrier commented that this topic has already been resolved through the ASRAC meetings.[13] (CAC TP: JCI, No. 66 at p. 21; Carrier, No. 62 at p. 20)

Based on DOE's analysis and consistent with the CAC/HP ECS Working Group Term Sheet, DOE proposes to adopt, for inclusion into 10 CFR part 430, subpart B, appendix M1, 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, a minimum external static pressure requirement of 0.50 in. wc. DOE is aware that such changes will impact the certification ratings for SEER, HSPF, and EER and is addressing such impact in the current energy conservation standards rulemaking.[14] For this reason, DOE is not proposing to make this change in appendix M.

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. In its comments, First Co. proposed to reduce the minimum static pressure for space-constrained and multi-family blower coils to 0.25 in. wc. or lower. (CAC TP: First Co., No. 56 at p. 2) The CAC/HP ECS Working Group included in its Final Term Sheet Recommendation #2, which is summarized in Table III.4 below. (CAC ECS: ASRAC Term Sheet, No. 76 at p. 2)

TABLE III.4—CAC/HP ECS WORKING GROUP RECOMMENDED MINIMUM EXTERNAL 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.
(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.4) to enable the proper administration of the CAC/HP ECS Working Group's recommended minimum external static pressure requirements. Recommendation #1 stated:

• Suggested definitions capture the intent of the Working Group and DOE should adopt them as is or modify them in a manner that captures the same intent.

• For those definitions that contain a maximum external static pressure requirement, the unit's maximum external static pressure would be determined using a dry coil test without electric heat installed and without an air filter installed at the unit's certified airflow, or, if the airflow is not certified, at an airflow of 400 cfm per ton of certified capacity.

• For those condensing units distributed in commerce with different indoor unit combinations, each specific combination would need to meet the applicable definition in order to be rated with the associated static.

The CAC/HP ECS Working Group's recommended definitions are as follows:

[12] *Manual D: Residential Duct Systems.* Arlington, VA: Air Conditioning Contractors of America (ACCA).

[13] The comment period for the November 2015 SNOPR was still open during the CAC/HP ECS Working Group negotiations.

[14] Docket No. EERE–2014–BT–STD–0048.

• A ceiling-mount blower coil system is a split-system central air conditioner or heat pump that contains a condensing unit and an indoor unit intended to be exclusively installed by being secured 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. The certified cooling capacity must be less than or equal to 36,000 Btu/h.

• A wall-mount blower coil system is a split-system central air conditioner or heat pump that contains a condensing unit and an indoor unit intended to be exclusively installed by having the back side of the unit secured to the wall within the conditioned space, with 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. The certified cooling capacity must be less than or equal to 36,000 Btu/h.

• Manufactured housing air conditioner coil system is a split-system air conditioner or heat pump that contains a condensing unit with an indoor unit that: (1) Is distributed in commerce for installation only in a manufactured home with the home and equipment complying with HUD Manufactured Home Construction Safety Standard 24 CFR part 3280; (2) has an external static pressure that must not exceed 0.4 inches of water; and (3) has an indoor unit that 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.'' Note, manufacturers must certify which combinations are manufactured housing air conditioner coil system.

• Low-static system means a ducted multi-split or multi-head mini-split system where all indoor sections produce greater than 0.01 and a maximum of 0.35 inches of water of external static pressure when operated at the full-load air volume rate not exceeding 400 cfm per rated ton of cooling.

• Mid-static system means a ducted multi-split or multi-head mini-split system where all indoor sections produce greater than 0.20 and a maximum of 0.65 inches of water of external static pressure when operated at the full-load air volume rate not exceeding 400 cfm per rated ton of cooling.

UTC/Carrier supported the low and medium static definitions as presented during the CAC/HP ECS Working Group meetings, in place of the short-duct unit definition DOE proposed in the November 2015 SNOPR. (CAC TP: UTC/Carrier, No. 62 at p. 3–4,19) AHRI and Mitsubishi recommended in their comments nearly identical definitions to those recommended in the CAC/HP ECS Working Group term sheet. (CAC TP: AHRI, No. 70 at p. 17; Mitsubishi, No. 68 at p. 2–3) Goodman generally supported the comments made by industry during the initial meetings of the CAC/HP ECS Working Group, in which additional sub-categories of ''short-ducted'' systems were proposed. Goodman recommended that DOE only include CAC/HP ECS Working Group's definitions and modifications to the test procedure in the ''M1'' test procedure and not part of ''M'' test procedure because the proposed modification to the test procedure would increase the measured energy consumption for those ''short-ducted'' systems being marketed under the current ''M'' test procedure. (CAC TP: Goodman, No. 73 at p. 6–7)

DOE agrees with the intent of Recommendation #1 and #2 of the CAC/HP ECS Working Group Term Sheet. 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. While conventional split systems are typically installed in attics or basements and require long ductwork to deliver conditioned air to the conditioned space, ceiling-mount systems, wall-mount systems, space-constrained systems, low-static systems and mid-static systems are installed in or in closer proximity to the spaces they condition, typically requiring shorter ductwork than conventional split systems. The field external static pressure for these non-conventional systems is lower than the external static pressure for conventional split systems as a result. In this SNOPR, DOE proposes to adopt the CAC/HP ECS Working Group recommended minimum external static pressure requirements for space-constrained systems, low-static systems, and mid-static systems to be more reflective of field conditions for these reasons, with one modification. DOE understands that when some 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 for these installations. Therefore, DOE also proposes to limit the CAC/HP ECS Working Group recommended minimum external static pressure requirement for space-constrained systems only to space-constrained indoor units and single-package space-constrained units.

The CAC/HP ECS Working Group tasked DOE with the determination of 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. DOE proposes to specify a minimum external static pressure requirement of 0.30 in. wc. for ceiling-mount and wall-mount systems, consistent with manufacturers' recommendations.

Mobile home [15] systems also have lower field external static pressure than conventional split systems. Mobile home systems are installed in homes that meet the HUD Manufactured Home Construction Safety Standard 24 CFR part 3280, which includes a maximum threshold of 0.30 in. wc. for the restrictiveness of ductwork. Consistent with these HUD requirements, the CAC/HP ECS Working Group recommendation, and the external static pressure requirements for mobile home systems in the DOE furnace fan test procedure, DOE proposes to adopt 0.30 in. wc. as the minimum external static pressure required for testing mobile home central air conditioning and heat pump systems.

In this SNOPR, DOE proposes to adopt the CAC/HP ECS Working Group recommendations for minimum external static pressure requirements for low-static and mid-static systems. By the definitions recommended by the Working Group, these systems are not capable of producing external static pressure significantly higher than the recommended minimum external static

[15] In previous rulemaking documents for the furnace fan test procedure and, DOE used the term ''manufactured home'' to be synonymous with ''mobile home,'' as used in some definitions in the **Federal Register**. 10 CFR 430.2. DOE will use the term ''mobile home'' in place of ''manufactured home'' hereinafter to be consistent with the **Federal Register** definitions that use ''mobile home'', such as for ''mobile home furnace.'' All provisions and statements regarding mobile homes and mobile home products are applicable to manufactured homes and manufactured home products.

pressure requirements. Consequently, DOE expects that any system that would meet these definitions would be incapable of properly conditioning a home that has ductwork with an external static pressure significantly higher than the proposed minimum.

The CAC/HP ECS Working Group did not recommend a change to 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. DOE proposes to adopt the Working Group recommendation for the minimum external static pressure requirement for SDHV systems.

Table III.5 summarizes DOE's proposed minimum external static pressure requirements.

TABLE III.5—PROPOSED 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
Low-Static	0.10
Mid-Static	0.30
Small Duct, High Velocity	1.15
Space-Constrained (indoor and single-package units only)	0.30

Issue 15: DOE requests comments on the proposed minimum external static pressure requirements.

DOE also agrees with the intent of the definitions recommended by the CAC/HP ECS Working Group. DOE proposes to adopt those definitions with minor modifications to make them consistent with other proposed regulatory language. For example, DOE is proposing to replace the term ''condensing unit'' in the CAC/HP ECS Working Group recommended definition for mobile home systems with the term ''outdoor unit'' to ensure that the definition applies to both mobile home air conditioners and heat pumps. DOE proposes 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:

- *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 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).

c. Certification Requirements

DOE proposes 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 may 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 are still 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 are also 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. DOE proposes 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.

Issue 16: DOE requests comment on the proposed definitions for kinds of CAC/HP associated with administering minimum external static pressure requirements.

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. ADP, Lennox, NEEA, NPCC, California IOUs, Rheem, ACEEE, NRDC, and ASAP did not support the proposal because it would make the ratings for units paired with condensing furnaces less reflective 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) JCI commented that this topic has already be resolved through the CAC/HP ECS Working Group meetings. (CAC TP: JCI, No. 66 at p. 21) Carrier commented to refer to the agreement on external static pressure from the CAC/HP ECS Working Group and expressed the view that this credit is contrary to better aligning the rating procedure with real world data. (CAC TP: Carrier, No. 62 at p. 21) As Carrier and JCI point out, Recommendation #2 of the CAC/HP ECS Working Group Term Sheet also states that the proposed reduction in minimum external static pressure required for units paired with condensing furnaces should not be used. (CAC ECS: CAC/HP ECS Working Group Term Sheet, No. 76 at p. 2)

In light of public comments and the consensus of the CAC/HP ECS Working Group, DOE is not proposing 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.

Issue 17: DOE requests comments on not including a reduced minimum external static pressure requirement for blower coil or single-package systems tested with a condensing furnace.

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 fans.[16] 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 response to the June 2010 NOPR, Earthjustice commented that the existing default values for coil-only units in the DOE test procedure were not supported by substantial evidence. Earthjustice stated that external static pressures measured from field data showed significantly higher values than DOE's default values in its existing test procedure. (CAC TP: Earthjustice, No. 15 at p. 2) 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.[17] (80 FR 69318)

DOE calculated the adjusted default fan power to be 441 W/1000 scfm. In the November 2015 SNOPR, DOE proposed to use this value in Appendix M1 of 10 CFR part 430 subpart B where Appendix M included a default fan power of 365 W/1000 scfm. DOE proposed not to make such replacements in Appendix M of 10 CFR part 430 subpart B.

In response to the November 2015 SNOPR, NEEA, NPCC, ACEEE, NRDC, ASAP, and the California IOUs 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, ASAP, No. 72 at p. 4; California IOUs, No. 67 at p. 2) ACEEE, NRDC, and ASAP also commented that they would be happy with 440 W/1000 scfm, as the implied precision of using 441W/1000 scfm is artificial. (CAC TP: ACEEE, NRDC, ASAP, No. 72 at p. 4)

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 shall 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 maintains its previous proposal to use a default value of 441 W/1000 scfm for split-system air conditioner, coil-only tests. DOE proposes to use this value in appendix M1 of 10 CFR part 430 subpart B in place of the default fan power of 365 W/1000 scfm that has 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.) that it 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.

DOE agrees with the CAC/HP ECS Working Group's recommendation to use a different default value for coil-only mobile home systems to reflect the difference in ductwork and, in turn, external static pressure of field installations of these systems. In this 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. However, in this case, DOE evaluated furnace fan power consumption at 0.54 in. wc., which is the 0.30 in. wc. recommended by the CAC/HP ECS Working Group plus 0.24 in. wc. to account for filter and indoor coil pressure drop. The resulting average indoor fan power consumption at the external static pressure representative of mobile home systems is 8% lower than the average indoor fan power consumption at the external static pressure representative of conventional systems. Applying the 8% reduction to the 441W/1000 scfm representing conventional indoor fan power consumption yields 406 W/1000 scfm. Thus, DOE proposes to use 406 W/1000 scfm as the default value for mobile home systems.

DOE notes that it used data from all of the furnaces in its database to calculate this value, instead of only mobile home furnaces, because its database includes a small number of mobile home furnaces that do not represent all capacities or motor technologies. DOE recognizes that including non-mobile home furnaces in this analysis may bias the result. Due to the space constraints typical of mobile home system installations, mobile home indoor units generally have more restrictive cabinets compared to conventional indoor units, which would be expected to increase the static pressure experienced by the indoor fan

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

[17] For a complete explanation of DOE's methodology, see 80 FR 69278, 69319–20 (Nov. 9, 2015).

and, in turn, increase indoor fan power consumption. Consequently, DOE expects that a default value calculated based on mobile home indoor fan performance data may result in a higher default value for these systems than the value proposed. In addition to the new default power values, DOE proposes to adjust measured capacity to account for the fan heat consistent with 441W/1000 scfm and 406 W/1000 scfm: 1,505 and 1,385 Btu/h per 1,000 scfm.

Issue 18: DOE requests comment on the proposed default fan power value for coil-only mobile home systems. DOE also requests mobile home indoor fan performance data for units of all capacities and that use all available motor technologies in order to allow confirmation that the proposed default value is a good representation for mobile home units.

The DOE test procedure needs a definition for a mobile home coil-only unit to appropriately apply the proposed default value for these kinds of CAC/HP. DOE proposes to define mobile home coil-only unit as:

• *Mobile home coil-only system* means a coil-only split system that includes an outdoor unit and coil-only indoor 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.''

Issue 19: DOE requests comments on its proposed definition for mobile home coil-only unit.

3. Revised Heating Load Line Equation

a. General Description of Heating Season Performance Factor (HSPF)

In the current test procedure, the HSPF determined for heat pumps in heating mode is calculated by evaluating the energy usage of both the heat pump unit (reverse refrigeration cycle) and the resistive heat component when matching the house heating load for the range of outdoor temperatures representing the heating season. The temperature range is split into 5-degree ''bins'', and an average temperature and total number of hours are assigned to each bin, based on weather data used to represent the heating season for each climate region. An HSPF value can be calculated for each climate region, but the HSPF rating is based on Region IV. In the HSPF calculation, the amount of heating delivered is set equal to the heating load, which increases as the bin temperature decreases. In the current test procedure, the heating load is proportional to the difference between 65 °F and the outdoor (bin) temperature. The heating load also is dependent on the size of the house that the unit heats. For the HSPF calculation the size of the house is set based on the capacity of the heat pump. For the current test procedure, the heating load is proportional to the heating capacity of the heat pump when operating at 47 °F outdoor temperature. The resulting relationship between heating load and outdoor temperature is called the heating load line equation—it slopes downward from low temperatures, dropping to zero at 65 °F. The slope of the heating load line equation affects HSPF both by dictating the heat pump capacity level used by two stage or variable speed heat pumps at a given outdoor temperature, and also by changing the amount of auxiliary electric resistance heat required when the unit's heat pumping capacity is lower than the heating load. The current test procedure defines two heating load levels, called the minimum heating load line and maximum heating load line. However, it is the minimum heating load line in Region IV that is used to determine HSPF for rating purposes.[18]

b. HSPF Issues

Studies have indicated that the current HSPF test and calculation procedure overestimates ratings because the current minimum heating load line equation is too low compared to real world situations.[19] In response to the November 2014 ECS RFI, NEEA and NPCC commented that the federal test procedure does a poor job representing balance point temperatures and electric heat energy use in the case of heat pump systems. They pointed out the inability of the test procedure to capture dynamic response to heating needs, such as the use of electric resistance (strip) heat during morning or afternoon temperature setup (*i.e.,* rewarming of the space after a thermostat setback period). They also expressed concerns about capturing the use of electric resistance heat during defrost cycles and at times when it shouldn't be needed, such as when outdoor temperatures are above 30 °F. (CAC ECS: NEEA & NPCC, No. 19 at p. 2)

DOE agreed with the NEEA and NPCC regarding balance point in the November 2015 SNOPR and noted that the heating balance point determined for a typical heat pump using the current minimum heating load line equation in Region IV is near 17 °F, while the typical balance point is in the range 26 to 32 °F, resulting from installing a proper-sized unit based on the design cooling load according to ACCA Manual S, 2014.[20] The low heating balance point means that the test procedure calculation adds in much less auxiliary heat than would actually be needed in cooler temperatures, thus inflating the calculated HSPF. Furthermore, the zero load point of 65 °F ambient, which is higher than the typical 50–60 °F zero load point,[21] causes the test procedure calculation to include more hours of operation at warmer outdoor temperatures, for which heat pump operation requires less energy input, again inflating the calculated HSPF. These effects result in overestimation of rated HSPF up to 30% compared to field performance, according to a paper by the Florida Solar Energy Center (FSEC).[22] For these reasons, DOE reviewed the choice of heating load line equation for HSPF ratings and proposed to modify it in the November 2015 SNOPR. 80 FR at 69320–2 (Nov. 9, 2015).

As part of its review for the November 2015 SNOPR, DOE considered a 2015

[18] See 10 CFR part 430, subpart B, appendix M, Section 1. Definitions.

[19] Erbs, D.G., C.E. Bullock, and R.J. Voorhis, 1986. ''New Testing and Rating Procedures for Seasonal Performance of Heat Pumps with Variable speed Compressors'', ASHRAE Transactions, Volume 92, Part 2B.

Francisco, Paul W., Larry Palmiter, and David Baylon, 2004. ''Understanding Heating Seasonal Performance Factors for Heat Pumps'', 2004 Proceedings of the ACEEE Summer Study on Energy Efficiency in Buildings.

Fairey, Philip, Danny S. Parker, Bruce Wilcox, and Matthew Lombardi, 2004. ''Climatic Impacts on Seasonal Heating Performance Factor (HSPF) and Seasonal Energy Efficiency Ratio (SEER) for Air-Source Heat Pumps'', ASHRAE Transactions, Volume 110, Part 2.

[20] *Manual S: Residential Equipment Selection* (2nd ed., Ver. 1.00). (2014). Arlington, VA: Air Conditioning Contractors of America (ACCA). pp. N7–N1.

[21] Francisco, Paul W., Larry Palmiter, and David Baylon, 2004. ''Understanding Heating Seasonal Performance Factors for Heat Pumps'', 2004 Proceedings of the ACEEE Summer Study on Energy Efficiency in Buildings.

[22] Fairey, Philip, Danny S. Parker, Bruce Wilcox, and Matthew Lombardi, 2004. ''Climatic Impacts on Seasonal Heating Performance Factor (HSPF) and Seasonal Energy Efficiency Ratio (SEER) for Air-Source Heat Pumps'', ASHRAE Transactions, Volume 110, Part 2.

Oak Ridge National Laboratory (ORNL) study [23] that examined the heating load line equation for cities representing the six climate regions of the HSPF test procedure in Appendix M. The study developed modified regional heating load line equations, including a heating load line equation for Region IV for calculation of a unit's HSPF. ORNL conducted building load analyses using the EnergyPlus simulation tool (see energyplus.net) using single-family Prototype Residential House models based on building characteristics specified by the 2006 International Energy Conservation Code (2006 IECC). The study concluded that a heating load line equation closer to the maximum load line equation of the current test procedure and with a lower zero-load ambient temperature would better represent field operation than the minimum load line equation presently used for HSPF rating values.

c. November 2015 SNOPR Heating Load Line Equation Proposal

In the November 2015 SNOPR, DOE proposed a new heating load line equation based on the findings of the ORNL study:

$$BL(T_j) = \frac{(T_{zl} - T_j)}{T_{zl} - T_{OD}} \cdot C \cdot \dot{Q}_c(95^\circ F) \text{ where,}$$

T_j = the outdoor bin temperature, °F
T_{zl} = the zero-load temperature, °F
T_{OD} = the outdoor design temperature, °F, which varies by climate region
C = the slope (adjustment) factor
$\dot{Q}_c(95\ ^\circ F)$ = the nominal cooling capacity at 95 °F, Btu/h

The proposed equation included the following changes from the current heating load line equation used for the HSPF calculation: [24]

- A zero-load temperature that varies by climate region, as shown in Table III.6, and is 55 °F for Region IV;
- The building load is proportional to the nominal cooling capacity at 95 °F, $\dot{Q}_c(95\ ^\circ F)$, as opposed to the heating capacity at 47 °F (except for heating-only heat pumps), to reflect typical selection of cooling/heating heat pumps based on cooling capacity; and
- The slope (adjustment) factor, C, is 1.3 rather than 0.77;

The November 2015 SNOPR also proposed revised heating load hours for each climate region, as shown in Table III.6. These hours are less than the current heating load hours by the number of hours in the temperature bins between the current and proposed zero-load temperatures.

TABLE III.6—CLIMATE REGION INFORMATION PROPOSED IN THE NOVEMBER 2015 SNOPR

Region No.	I	II	III	IV	V	VI
Heating Load Hours, HLH	562	909	1,363	1,701	2,202	* 1,974
Zero-Load Temperature, T_{zl}	60	58	57	55	55	58

* Pacific Coast Region.

The ORNL study developed heating load line equations consistent with the similar equations of the current test procedure, using the EnergyPlus heating and cooling loads calculated for the IECC 2006 building models developed for numerous cities of the climate regions of interest. The approach sized the house based on the heat pump cooling capacity rather than heating capacity, consistent with the sizing approach prescribed for heat pumps in ACCA Manual S, which is also based on cooling capacity. The study used the heat pump size recommendations based on the design cooling load calculated by EnergyPlus in its analysis. The design cooling load was determined for the 0.4% cooling design day dry-bulb temperature based on a 24-hour design day calculation using the heat balance method, which includes the effects of house thermal mass on the peak load. For Climate Region IV, used as the basis for the HSPF calculation, the study concluded that the appropriate slope factor (C in the equation defined above) is 1.3.

In the November 2015 SNOPR, DOE also proposed to eliminate maximum and minimum heating load line equations in an effort to focus on one load level that would best represent heating. As mentioned, the proposed heating load line equation is based on nominal cooling capacity rather than nominal heating capacity, which is intended to better reflect field installation practices than the basis on heating capacity of the current test procedure. This approach also justifiably benefits units with higher heating to cooling capacity ratios. Such units would have improved HSPF ratings, reflecting the shift of more heat from electric resistance to heat pumping. For the special case of heating-only heat pumps, DOE proposed to maintain a sizing approach based on heating capacity.

The ORNL study also evaluated the impact of the proposal on HSPF ratings. Based on the results, DOE estimated that HSPF would be reduced on average about 16 percent for single speed and two-stage heat pumps. Consistent with the requirements of 42 U.S.C. 6293(e), DOE will account for these changes in any proposed energy conservation standard, and this test procedure proposal would not be required as the basis for efficiency representations until the compliance date of any new energy conservation standard.

d. Comments on the November 2015 SNOPR

Comments expressed by stakeholders on the proposed heating load line equation, both in written form in response to the November 2015 SNOPR and verbally during the CAC/HP ECS Working Group meetings, are summarized in the following paragraphs, organized by common themes.

[23] 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).

[24] In the current test procedure, for all climate regions but Region V, the heating load based on minimum design heating requirement as a function of outdoor temperature T_j is $Q_h(47)$ * 0.77 * (65 − Tj)/60.

Field Representativeness of the Heating Load Line Equation

One common theme raised in the comments concerned the field representativeness of the data used to generate the proposed heating load line equation. Unico expressed concern regarding the data collected, requesting more time dedicated to research, particularly on the northward shift of heat pump use despite the majority still being sold in temperate climates. (CAC TP: Unico, No. 63 at p. 13) Lennox expressed concern that the building stock used to evaluate the change was outdated; the current load line should be aligned with the time period of the standard. (CAC TP: Lennox, No. 61 at p. 20) During the ASRAC meetings, Ingersoll-Rand expressed the same concern, adding that the housing stock would continue to improve over time, driving the slope down. (CAC ECS: ASRAC Public Meeting, No. 87 at p. 7) Ingersoll-Rand also expressed reservations that the ORNL report relied on data generated through simulations. (CAC ECS: ASRAC Public Meeting, No. 85 at p. 134).

Southern Company commented that basing the heating load line equation exclusively on the 2006 IECC standard unrealistically assumes flawless adoption and enforcement of building code standards and that even future housing stock would be much less tight (*i.e.,* would allow much more infiltration of outdoor air than allowed by the IECC 2006 building code). (CAC ECS: ASRAC Public Meeting, No. 85 at p. 130) ACEEE requested that simulation data generated in the ORNL report remain in the discussion as the report represents a substantial contribution. (CAC ECS: ASRAC Public Meeting, No. 85 at p. 134).

DOE understands the importance of developing the heating load line equation with data that accurately represents field conditions and operation. Regarding the relevancy of the 2006 IECC code, DOE maintains that it is an appropriate representation of the housing stock in 2021 for the purposes of developing the heating load line equation. A follow-up investigation by Lawrence Berkeley National Laboratory (LBNL) examining RECS data corroborated this claim, showing that vintage housing characteristics in 2021 would at best resemble new housing characteristics in 2005. (CAC ECS: ASRAC Public Meeting, No. 85 at p. 81) DOE also maintains that EnergyPlus simulation results provide the most accurate available picture of heating load requirements and their dependence on independent parameters, (*e.g.,* house design details, heat pump sizing, typical weather patterns). While the data from some direct field studies have been made available, none have included information on heat pump sizing, a vital parameter for fitting a heating load line curve to the data.

Impact on Model Differentiation

Another common theme expressed in the comments concerned the impact of the proposed heating load line equation on model differentiation. Mitsubishi suggested that the proposed changes would decrease performance differentiation between single stage, two stage, and variable speed systems and recommended DOE refrain from making any HSPF changes. (CAC TP: Mitsubishi, No. 68 at p. 5) Rheem, JCI, and Carrier/UTC concurred. (CAC TP: Rheem, No. 69 at p. 17; JCI, No. 66 at p. 13; Carrier/UTC, No. 62 at p. 21) ACEEE added that, in the short-term, accurately capturing relative performance of products should take precedence over better reflecting field energy use if the two are mutually exclusive. (CAC TP: ACEEE, No. 72 at p. 5) During the 2015–2016 CAC/HP ECS Working Group meetings, AHRI expressed concern over the lack of differentiation for variable speed products resulting from the proposed heating load line equation. (CAC ECS: ASRAC Public Meeting, No. 88 at p. 83) AHRI suggested a load line having a lower slope factor (equal to 1.02) and presented an initial assessment of the impact of both the DOE and AHRI proposals on product differentiation. Additionally, Southern Company stressed the importance of encouraging variable speed operation. (CAC ECS: ASRAC Public Meeting, No. 88 at p. 87).

To allow more detailed examination of this question, AHRI provided test data to DOE's contractor under a non-disclosure agreement. The data included performance measurements required to calculate HSPF using the current and the proposed test procedures, for a number of two stage and variable speed heat pumps. The calculations showed that the proposed heating load line equation (1.3 slope factor and 55 °F zero-load temperature, with sizing based on the nominal cooling capacity) would reduce the average HSPF difference between two stage and variable speed models as compared to the current heating load line equation (0.77 slope factor and 65 °F zero-load temperature, with sizing based on the nominal heating capacity) from 1 HSPF point currently to roughly 0.35. DOE presented the methodology, findings, conclusions, and implications of the analysis during the CAC/HP ECS Working Group meetings. (CAC ECS: ASRAC Public Meeting, No. 63 at pp. 1–7).

DOE acknowledges the impact on differentiation of variable speed heat pumps when calculating HSPF with a higher-slope factor heating load line equation. However, EPCA requires test procedures to be representative of the covered product's average use cycle—not that the test procedure should favor particular design options. (42 U.S.C. 6292(b)(3)) DOE evaluated the proposed amendment with a focus on accurately capturing field performance and believes that the performance of models that clearly perform better in the field will be captured and reflected in higher ratings when tested using a field-representative efficiency metric. Nevertheless, DOE agrees that all variable speed CAC/HP designs should be considered carefully in the analysis to assure that the resulting test procedure fairly represents their performance. As described below, ORNL has made some revisions in its analysis that DOE has incorporated into a revised proposal that improves the differentiation of variable speed heat pumps.

General Impact on Current HSPF Ratings

Comments on the overall impact of the proposed heating load line equation on current HSPF ratings were also received. Carrier/UTC reported a dramatic impact on all types of equipment, with reductions in HSPF ranging from 15 to 25 percent as a result of the proposed change in the November 2015 SNOPR. (CAC TP: Carrier/UTC, No. 62 at p. 21). Rheem commented that the proposal would reduce the HSPF of heat pumps designed for southern market installations but did not clarify why southern market heat pumps would be more affected. (CAC TP: Rheem, No. 69 at p. 17).

DOE notes that, as indicated in the ORNL report, field studies have shown that HSPF ratings based on the current test procedure may be higher than actual performance. Hence, a reduction in the rating with the revised test procedure would be consistent with observations of actual heat pump field performance.

Sizing Based on Cooling Capacity

Other comments addressed DOE's proposal in the November 2015 SNOPR to base the heating load line equation on cooling capacity rather than heating capacity. NEEA and NPCC recommended that each heat pump be assigned one of several heating load line equations based on heating capacity and balance point temperature. The appropriate heating load line equation would be the one where the load at 30 °F is most nearly equal to the heat pump capacity at that temperature. (CAC TP: NEEA and NPCC, No. 64 at p. 11) However, ACEEE expressed support for the cooling capacity basis during the ASRAC meetings. (CAC ECS: ASRAC Public Meeting, No. 88 at p. 92).

DOE understands that the balance point temperature for heat pumps operating in the field is closer to 30 °F than the 17 °F calculated for the current heating load line equation. For the heating load line equation proposed in the November 2015 SNOPR, the average balance point temperature is between 27 and 28 °F. However, DOE does not agree with NEEA and NPCC that heat pumps are typically sized in the field based on heating capacity or the balance point temperature. The sizing instructions outlined in ACCA Manual S specifically state that ''heat pump equipment shall not be sized for the design day heating load, or for an arbitrary thermal balance point.'' DOE further understands that most heat pump units in the field are sized based on cooling capacity as opposed to heat pump capacity, which is consistent with the Manual S provision that ''heat pumps shall be sized for cooling.'' [20] To ensure field representativeness, DOE proposes to maintain the approach that assumes heat pumps are sized based on cooling capacity. This approach also benefits heat pump units that have higher nominal heating to cooling capacity ratios by boosting their HSPF.

Overall Regulatory Approach

Other comments concerned the regulatory approach regarding the heating load line equation. Carrier/UTC encouraged DOE to go beyond adjusting the heating load line equation, suggesting that the current HSPF procedure does not adequately account for the benefits of variable speed designs and that DOE should fund research into a completely new procedure rather than applying corrections to the existing procedure by changing the slope (CAC TP: Carrier/UTC, No. 62 at p. 21). Unico suggested tabling the change until the next [CAC test procedure] rulemaking when and if there would be support for changing it (CAC TP: Unico, No. 63 at p. 13). JCI added that changing the temperature at which the heating cyclic test is performed would be acceptable for Appendix M1 but not for Appendix M. (CAC TP: JCI, No. 66 at p. 21).

ACEEE, NRDC, and ASAP proposed that AHRI, ASHRAE, DOE, and all other stakeholders begin work now on a new ''clean-sheet'' rating method for heat pumps, to be effective in the next rule after this current rulemaking, as was recently done for water heaters. ACEEE, NRDC, and ASAP stated that the current heat pump test method is obsolete. It was developed when essentially all air-source heat pumps were single-stage, and it appears that the present method is not technology-neutral. According to ACEEE, NRDC, and ASAP, the current test method should be revised to avoid penalizing advanced technologies with the potential for higher efficiency, lower heating bills, and reduced impact on winter grid peaks. ACEEE, NRDC, and ASAP recommended that the test procedure for variable speed heat pumps be revised in a future rulemaking to better reflect both the relative performance and field energy use of this equipment. (CAC TP: ACEEE, NRDC, and ASAP, No. 72 at p. 5–6).

CAC/HP ECS Working Group members ultimately did not agree on a resolution on the current heating load line equation regulatory approach and agreed (as reflected in the Final Term Sheet Recommendation #4) that DOE should make a final decision based on a review of available information. (CAC ECS: ASRAC Term Sheet, No. 76 at p. 3).

DOE acknowledges that another test method could be developed rather than the current heating load line equation approach, but DOE does not wish to propose a sweeping overhaul with this notice. DOE has taken the steps agreed to in the ASRAC Final Term Sheet: To evaluate past comments, improve the current analysis, and recommend an improved heating load line equation based on a modest departure from the existing approach. These steps taken leading up to the proposal in this notice do not preclude DOE from evaluating more fundamental changes in future rulemakings. DOE will continue to evaluate test methodologies and will work with AHRI and other interested parties to evaluate other approaches for testing heat pumps, determining the suitability of a more fundamental change in a future rulemaking.

In response to JCI's comment regarding changes to the cyclic test, DOE proposed in the November 2015 SNOPR to change the cyclic test temperature for variable speed heat pumps only in Appendix M1 of 10 CFR part 430 subpart B, and not to Appendix M of the same Part and Subpart—DOE has not changed this aspect of the proposal in this notice.

Heating Load Line Equation Slope Factor and Zero-Load Temperature

DOE also received specific recommendations on the heating load line equation slope factor and zero-load temperature. In its comments, Lennox opposed the heating load line equation slope factor change from 0.77 to 1.3 and recommended 1.02, citing better field representativeness and wider product differentiation. (CAC TP: Lennox, No. 61 at p. 12) During the ASRAC meetings, AHRI concurred, indicating that (a) differentiation of variable speed products from two stage or single stage products is better with the 1.02 slope factor, and (b) the 2012 IECC building requirements (for which the ORNL study showed a 1.02 slope) would better represent building stock in 2021 than the 2006 IECC requirements. (CAC ECS: ASRAC Public Meeting, No. 88 at p. 83).

Regarding the heating load line equation zero-load temperature, the California IOUs deferred to the CAC/HP ECS Working Group consensus, generally accepting the 55 °F zero-load temperature proposed in the November 2015 SNOPR. (CAC TP: CA IOUs, No. 67 at p.7) JCI suggested retaining the 65 °F intercept and 0.7 slope factor of the current test procedure. JCI argued for the 65 °F intercept, referring to evidence shared during the ASRAC meetings by Ingersoll-Rand, which JCI indicated shows that heat pump operation does occur at these mild conditions. JCI cited the negative impact on variable speed product differentiation in supporting the lower slope factor. (CAC TP: JCI, No. 66 at p. 13).

In response to JCI's concerns outlined in this preamble, model differentiation is not an EPCA requirement for test procedures.

Additional 35 °F Test for Variable-Speed Heat Pumps

In the November 2015 SNOPR, DOE requested comment regarding the appropriate approach for rating of variable-speed heat pumps if DOE were not to adopt the proposed general heating load line equation. More specifically, DOE was concerned about a potential inaccuracy associated with the use of extrapolation of the minimum-speed performance measured in 47 °F and 62 °F ambient temperatures for characterization of heat pump performance below 47 °F. In the November 2015 SNOPR, DOE described two options. In Option 1, DOE would base performance on minimum speed tests at 47 °F and intermediate speed tests at 35 °F, an approach which would involve no additional test burden. In Option 2, DOE would require an additional minimum speed test at 35 °F, which would likely be more accurate, at the cost of a higher test burden.

In its comments, UTC/Carrier supported Option 1, because it would not result in an increase in testing burden. (CAC TP: UTC/Carrier, No. 62 at p. 22) The California IOUs supported Option 2, and argued that the additional test burden would be justified by the accuracy improvements. (CAC TP: CA IOUs, No. 67 at p. 7) Johnson Controls asked for more time to study both options, requesting that the discussion be incorporated as part of the 2015–2016 ASRAC Negotiations. (CAC TP: JCI, No. 66 at p. 22).

DOE has responded to the comments received and addressed this issue in the context of the revised heating load line equation proposed in section III.C.3.i of this notice.

e. Modifications to the 2015 ORNL Analysis

Following the conclusion of the CAC/HP ECS Working Group meetings, ORNL reexamined key assumptions adopted in its 2015 report [23] and determined that three modifications would be beneficial in order to improve the field representativeness of the analysis. The analysis revisions and its results are described in an addendum to the 2015 report. (CAC TP: ORNL Report Addendum, No. 2) Ultimately the modifications to the analysis led DOE to propose lower heating load line equation slope factors (as discussed later in this section), which addresses the comments of several stakeholders.

First, ORNL removed continuous mechanical ventilation as a feature of the Prototype Residential Houses used in the analysis. While housing models used in the initial analysis included continuous mechanical ventilation, the 2006 IECC does not include that requirement, and DOE believes that a prototype design without continuous mechanical ventilation would be more representative of the average housing stock.

ORNL also modified the heat pump sizing approach used by the analysis. In the 2015 study, the auto-sizing feature of EnergyPlus was used. The auto-sizing feature uses a heat pump sized for the 0.4% cooling design dry-bulb temperature, based on a 24-hour design day calculation using the heat balance method, which includes the effects of house thermal mass on the peak load. However, this approach does not provide cooling capacity sufficient to meet the load for all hours of the year. For the revised analysis, ORNL increased the heat pump size so that cooling capacity would match or exceed the cooling load for all hours of the year. This increases heat pumps capacities from 6% to 12%, depending on the cities evaluated. This approach also better aligns the sizing approach of the analysis with the sizing assumptions used in the DOE test procedure, meaning that the heat pump's cooling capacity is very close to 1.1 times the cooling load for 95 °F ambient temperature, consistent with equation 4.1–2 of the current test procedure. ORNL also applied an additional 10% oversizing to heat pumps for Region V, based on the observation that this adjustment is required to achieve consistency with the 1.1 factor oversizing for cooling used in the DOE test procedure.

The changes in heating load and heat pumps sizing led to reduction in all of the regional heating load line equation slope factors. Removing continuous ventilation reduced both the zero-load temperatures and the heating load line equation slope factor across each region. This change reduced the heating load line equation slope factor an average (across all regions) of 5% while the zero-load temperatures dropped on average by about 1–2 °F. The adjustment in heat pump size led to an average additional reduction in the slope factor of roughly 9%, but did not change the zero-load temperatures. The calculated heating load line equation slope factors of the modified analysis vary sufficiently that DOE is proposing regional heating load line equation slope factors as opposed to a single slope factor, using Region IV as the basis for the HSPF rating. (CAC TP: ORNL Report Addendum, No. 2)

f. DOE Proposal Based on Revised Analysis

Based on ORNL's revised findings, DOE has revised its heating load line equation proposal from the November 2015 SNOPR. DOE introduced a final adjustment to the slope factors developed by ORNL to address variable speed systems. This aligns the analysis more closely with the range of capacity recommended in ACCA Manual S, which allows significantly more oversizing for variable-speed heat pumps than for single speed or two-stage heat pumps. The range of recommended capacity factor is 0.9 to 1.15 for single-stage heat pumps and 0.9 to 1.30 for variable-speed. DOE recognizes that such oversizing is much more tolerable with variable-speed heat pumps as compared to single-speed heat pumps, due to their ability to better match mild-weather loads in both heating and cooling seasons, and thus limit the inefficiencies associated with cycling losses. Based on the averages of these ranges, DOE calculated a size adjustment factor for variable-speed units equal to (0.9 + 1.30) divided by (0.9 + 1.15), which equals 1.07, essentially suggesting an additional 7 percent oversizing for variable-speed heat pumps. Applying this to the heating load line equation analysis leads to a corresponding reduction in the slope factors for variable-speed products. DOE notes that for consistency, this oversizing would be applied in seasonal performance calculations for cooling mode and for heating mode.

With the analysis changes and the adjustment for variable-speed models, DOE is proposing the following heating load line equation changes from the November 2015 SNOPR:

- The zero-load temperature would vary by climate region according to the values provided in Table III.10, but remain at 55 °F 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.7, 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.7, and be 1.07 for Region IV;

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.10.

TABLE III.7—CLIMATE REGION INFORMATION PROPOSED IN THIS NOTICE

Region No.	I	II	III	IV	V	VI*
Heating Load Hours	493	857	1280	1701	2202	1842
Zero-Load Temperature, T_{zl}	58	57	56	55	55	57
Heating Load Line Equation Slope Factor, C	1.10	1.06	1.29	1.15	1.16	1.11

TABLE III.7—CLIMATE REGION INFORMATION PROPOSED IN THIS NOTICE—Continued

Region No.	I	II	III	IV	V	VI *
Variable Speed Slope Factor, C_{VS}	1.03	0.99	1.20	1.07	1.08	1.03

* Pacific Coast Region.

Following from this proposed heating load line equation change, DOE also proposes in this 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.

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.

Issue 20: DOE requests comments on the adjustments to the proposals for calculating HSPF for heat pumps and SEER for variable-speed heat pumps.

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

DOE examined the impact of the present proposal on HSPF ratings based on test results for 2, 3, and 5-ton heat pumps provided by AHRI. Table III.8 presents the effect of different Region IV heating load line equation slope factors on the average HSPF of two-stage and variable speed units using these results. For two-stage units, the average HSPF reduction from measurements using the current test procedure to the current proposal would be 13.9%. For variable speed products, the average reduction resulting from the current proposal would be 15.3%. The purpose of the test procedure is to evaluate the performance during a representative average use cycle. Nevertheless, DOE believes that reasonable differentiation is still preserved with the current proposal in this SNOPR. Further, DOE believes that heat pumps with good heating mode performance will continue to stand out as compared to heat pumps without good heating mode performance. The test procedure changes proposed in this notice to allow higher speed operation at lower temperature and for a 5 °F optional test (see section III.C.4) should allow for even greater differentiation for variable-speed heat pumps with good heating performance.

TABLE III.8—EFFECT OF REGION IV SLOPE FACTORS ON HSPF OF TWO-STAGE (TS) AND VARIABLE SPEED (VS) MODELS

	Region IV slope factors				
	2010 Final rule *	1.02	1.15	1.30	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 all equipment: 0.77.
** Slope factor for two-stage equipment: 1.15. Slope factor for variable speed equipment: 1.07.

h. 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.9 includes the Working Group's recommended HSPF levels:

TABLE III.9—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

As mentioned, the Working Group ultimately left the decision of the appropriate heating load line equation factor up to DOE. The HSPF levels recommended by the Working Group are based on different heating load line equation factors than DOE is proposing in this SNOPR. Consequently, DOE determined HSPF levels that are consistent with those recommended by the Working Group but based on the 1.15 heating load line equation factor DOE proposes in this notice. DOE does not have access to all of the data or details of the methodology used by the Working Group to derive the HSPF levels it recommended. In the absence of this information, DOE used linear interpolation between the HSPF values recommended by the Working Group using 1.02 and 1.30 to derive the associated HSPF values using a heating load line equation factor of 1.15. DOE confirmed that linear interpolation provides good match to directly calculated results using available heat pump performance data. Specifically, the maximum deviation for an interpolated value is 0.04 HSPF points for a representative sample of heat pumps, and the average deviation is 0.005 HSPF points. Table III.10 includes the HSPF levels that are consistent with the Working Group recommended HSPF levels, but based on a 1.15 heating load line equation slope factor.

TABLE III.10—CAC/HP ECS WORKING GROUP RECOMMENDED HSPF LEVELS BASED ON CURRENTLY PROPOSED HEATING LOAD LINE EQUATION

Product class	HSPF
Split-System Heat Pumps	7.5
Single-Package Heat Pumps	6.8

Issue 21: DOE requests comments on the adjusted values of minimum HSPF based on the HSPF efficiency levels recommended by the CAC/HP ECS Working Group.

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

DOE discussed in the November 2015 SNOPR potential inaccuracy 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 69322–3 (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. Because the bins in this temperature range have many hours associated with them, the impact on HSPF of this inaccuracy can be significant, particularly with the current test procedure, which uses a 0.77 heating load line equation slope factor. 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 reduces the impact of the issue because the higher heating load reduces the weighting of the HSPF on minimum-speed performance. DOE indicated that, because the higher slope factor alleviated the minimum-speed inaccuracy, it did not propose any test procedure amendment to address this issue, but that it might reconsider this possibility if a lower heating load line equation slope factor were adopted. Id.

DOE proposed two potential approaches to resolve this minimum-speed issue. The first would have involved approximation of minimum-speed performance between 35 °F and 47 °F based on the intermediate-speed frosting-operation test at 35 °F and the minimum-speed test at 47 °F, and assuming that below 35 °F the nominal minimum speed is the same as the intermediate speed. This first approach would not have required any additional testing. The second approach discussed for resolving the issue was to require two additional tests, one intermediate-speed test at 17 °F and one minimum-speed frosting-operation test at 35 °F. DOE requested comment on which of these approaches would be preferable. 80 FR at 69323 (Nov. 9, 2015). A summary of the comments received is located in section III.C.3.d.

As discussed in this preamble, DOE is proposing in this SNOPR to reduce the heating load line equation slope factor to 1.07 for variable-speed heat pumps. At this level, the data currently available to DOE suggests that 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 is also proposing revision to the estimation of minimum-speed performance to reduce the impact of the error. Consistent with stakeholder comments, DOE is proposing to adopt the approach discussed in the November 2015 SNOPR that does not require additional testing. Further, DOE proposes that the approach be used only 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. For example, if the rotational compressor operating speed for a heat pump operating at its minimum speed remains constant down to 37 °F and the HSPF calculation considers minimum-speed operation only down to the 37 °F temperature bin (this would occur if the calculated heating load is equal to or greater than the intermediate-speed capacity for temperature bins below 37 °F), any rotational speed increase below 37 °F would not require use of the alternative calculation. DOE proposes adoption of a definition, ''minimum-speed-limiting variable-speed heat pump,'' to refer to such heat pumps.

For the variable-speed heat pumps for which DOE's contractor received data from AHRI during the 2015–2016 ASRAC Negotiations, use of this approach would reduce average HSPF from 9.26 to 9.13, reducing the VS/TS differential to 0.96, which is equivalent to the differential for a 1.02 slope factor without considering any different treatment of variable-speed heat pumps (see Table III–11). However, it is not clear that all the heat pumps of the AHRI dataset would have required use of the alternative calculation approach, so the actual reduction in the average HSPF could be less.

DOE notes that it described another option for reducing the minimum-speed inaccuracy in the November 2015 SNOPR, specifically requiring additional tests to more thoroughly explore the heat pump's performance for the range of different operating speeds and ambient conditions. DOE could consider additional tests to improve accuracy further. Potential additional tests would include an intermediate-speed test at 17 °F, and either minimum-speed frosting-condition tests near 35 °F or minimum-speed steady-state tests at 40 °F or above. The HSPF calculation could be adjusted to provide better estimates of variable-speed heat pump performance over the range of conditions considered in the calculation based on one or more of these tests.

DOE also proposes that certification reports indicate as part of non-public data whether the alternative calculation method was used to determine the heat pump's rating.

Issue 22: DOE requests comment on its proposal to require use of an alternative HSPF rating approach (for heat pumps that raise minimum compressor speed in ambient temperatures that impact the HSPF calculation) that estimates minimum-speed performance (a) between 35 °F and 47 °F using the intermediate-speed frosting-operation test at 35 °F and the minimum-speed test at 47 °F, and (b) below 35 °F assuming that minimum-speed and intermediate-speed performance are the same. In addition, DOE requests comment on including in certification reports for variable-speed heat pumps whether this alternative approach was used to determine the rating. Finally, DOE requests comment on whether any of the additional tests that could be used to further improve the accuracy of variable-speed heat pump performance estimates should be required in the test procedure.

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. 80 FR at 69322 (Nov. 9, 2015).

DOE proposed as part of Appendix M1 that for variable speed units that limit the maximum speed operation below 17 °F and have a low cutoff temperature (temperature below which the unit will not operate in heat pump mode) less than 12 °F, the manufacturer could choose to calculate the maximum heating capacity and the corresponding energy usage for ambient temperatures less than 17 °F based on two maximum speed tests at: (1) 17 °F outdoor temperature, and (2) 2 °F outdoor temperature or at the low cutoff temperature, whichever is higher.[25] 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 69323 (Nov. 9, 2015).

Testing done by ORNL found that the unit efficiency at maximum speed below 17 °F is slightly higher than the extrapolated values in the current test procedure, and this proposed option would provide a more accurate prediction of heat pump low-ambient performance, not only for those units that limit maximum speed operation below 17 °F, but also for those that do not.[26] DOE therefore proposed to revise Appendix M1 such that, for variable speed units that do not limit maximum speed operation below 17 °F, manufacturers would also have the option to use this revised method if it is more representative of low ambient performance. 80 FR at 69323 (Nov. 9, 2015).

DOE developed the proposal based on review of the results of a limited number of tests. DOE requested test results and other data to show whether the impact on HSPF of the proposal is similar for other variable speed heat pumps, and also requested comment on the additional test burden of the proposed modification. 80 FR at 69323 (Nov. 9, 2015).

Several stakeholders provided comments in response to these requests for data and comments.

JCI supported the proposal on the condition that the tests be made optional, but at a higher temperature (*e.g.,* 10 degrees) so that more test labs can perform the test. (CAC TP: JCI, No. 66 at p. 22)

Lennox and ADP expressed concerns over the difficulty of testing at 2 °F for many labs, commenting that the test would greatly increase the burden on manufacturers as it would greatly increase the test time to achieve the 2 °F test point, possibly require expensive hardware upgrades for labs, or force manufacturers to use outside labs. (CAC TP: Lennox, No. 61 at p. 20; ADP, No. 59 at p. 13)

Rheem commented that the proposed 2 °F outdoor temperature introduces testing variability, and that the very low test temperature introduces a significant test burden because it is rare for manufacturers or independent labs to have such facilities. Rheem commented that there is no justification that the resulting HSPF results will more closely match the resulting energy costs to consumers. Major capital investment by manufacturers and independent-labs would be required to add this capability. (CAC TP: Rheem, No. 69 at p. 17)

Unico commented that most heat pumps are not able to be tested below 17 °F and that most test laboratories cannot test below 17 °F. Nevertheless, they also mentioned (a) public interest in heat pumps that operate at significantly lower temperatures and (b) manufacturers that are publishing data and promoting such cold climate heat pumps. Unico expressed support for a separate heat pump test standard for cold-weather heat pumps, indicating that such a test standard would require testing at 2 °F. (CAC TP: Unico, No. 63 at p. 13)

UTC/Carrier commented that the test point at 2 °F outdoor temperature is challenging for most test facilities (if it is possible at all). (CAC TP: UTC/Carrier, No. 62 at p. 22)

The California IOUs and ACEEE, NRDC, and ASAP commented that in response to industry's concerns over testing at 2 °F, they recommend that variable speed heat pumps be tested at 5 °F, in addition to the 17 °F cold temperature point. ACEEE, NRDC, and ASAP commented that requiring the 5 °F test seems to be a reasonable way to differentiate excellent cold-temperature performance, which is critical for customer acceptance nationally, and for mitigating winter peaks for utilities. The California IOUs noted that the European standard requires testing at 5 °F and that manufacturers participate in the global market and Europe, so that they must test at 5 °F. (CAC TP: California IOUs, No. 67 at p. 7; ACEEE, NRDC, and ASAP, No. 72 at p. 5)

NEEA and NPCC commented that they do not believe that the current test procedure for variable speed systems in any way delivers annual energy use or efficiency ratings that are reasonably reflective of an average use cycle. (CAC TP: NEEA and NPCC, No. 64 at p. 9)

The possible adoption of a very-low-temperature test for rating of variable speed heat pumps was also discussed during the CAC/HP ECS Working Group meetings, ultimately leading to Recommendation #5 in the Term Sheet, that a 5 °F ambient temperature optional test be adopted for variable speed heat pumps. (CAC ECS: ASRAC Term Sheet, No. 76 at p. 3) Given the consensus among Working Group members regarding this recommendation, DOE believes that the concerns expressed by the initial comments about this optional test would be resolved by adopting a 5 °F ambient temperature for the test rather than the 2 °F initially proposed.

In addition, DOE discussed in the November 2015 SNOPR the possibility of making an adjustment to the test procedure to address potential accuracy issues associated with estimation of minimum-speed heat pump performance for temperatures below 47 °F based on extrapolation of the results of tests conducted in 47 °F and 62 °F ambient temperatures. Specifically, testing by ORNL indicated that the HSPF may be over-predicted for heat pumps that do not allow use of the same minimum speed for ambient temperatures below 47 °F. 80 FR 69322–3 (Nov. 9, 2015). However, DOE did not propose to make this change in the November 2015 SNOPR, explaining that the modification of the heating load line equation would sufficiently alleviate the potential inaccuracy, making adjustment to the test procedure unnecessary. However, DOE did request comment on preferences for approaches to modification to the test procedure in case the modified heating load line equation was not adopted, describing approaches that would involve an additional test and an approach that would not require additional testing. *Id. This issue and DOE's proposal to resolve it is discussed in greater detail in section III.C.3.i.*

The revised variable speed heat pump test procedure proposed in this notice would include 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

[25] In the November 2015 SNOPR, DOE proposed that in the case that the low cutoff temperature is higher than 12 °F, the manufacturer would not be allowed to utilize this option for calculation of the maximum heating load capacity.

[26] Rice et al. (2015) Review of Test Procedure for Determining HSPFs of Residential Variable-Speed Heat pumps. (Docket No. EERE–2009–BT–TP–0004–0047).

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 extrapolation below 5 °F 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.

Development of these proposals and decisions regarding their details is explained further below (except for the last proposal, which is discussed in section III.C.3.i).

For heat pumps using the 5 °F test, the CAC/HP ECS Working Group Term Sheet recommended use of interpolation to calculate heat pump performance in the temperature range from 5 °F to 17 °F based on the test results for the 5 °F and 17 °F tests (CAC ECS: ASRAC Term Sheet, No. 76 at p. 3, Recommendation #5) DOE considered what approach to use for calculation of heat pump performance below 5 °F, with the understanding that extrapolation of the 5 °F-to-17 °F trend below 5 °F is not likely to be accurate because full-speed operation could be very different at 5 °F than it is at 17 °F. Although the November 2015 SNOPR primarily addressed cases where the compressor speed could be lower at the lower temperature (see, *e.g.* 80 FR at 69323 (Nov. 9, 2015)), the comments focus more on the possibility of higher speed at lower temperature. In any case, as indicated in this preamble, DOE does not believe such extrapolation is appropriate when the compressor speeds may be very different. DOE considered different approaches to calculate the performance below 5 °F and evaluated some of them using data obtained from the NEEP cold climate heat pump database.[27] Many of the heat pumps in the database have performance data for both 5 °F and for a lower ambient temperature. DOE evaluated for each such heat pump of the database how closely the performance at the lower ambient temperature could be predicted using the other available performance data. DOE concluded that a good approach is to apply the 17 °F-to-47 °F slope below 5 °F, for both capacity and power input. Using this approach, the lower-temperature capacity and power input were predicted within 10 percent for at least two thirds of the evaluated heat pumps.[28] DOE considers this to be acceptable accuracy for HSPF calculations, considering that the annual hours with temperature lower than 5 °F are limited, representing roughly one percent of heating season hours in Region IV. Hence, DOE has proposed an approach for extrapolation of heat pump performance for temperatures below 5 °F based on the slopes of the capacity and power input levels between 17 °F and 47 °F.

Issue 23: DOE requests comment on the proposals for evaluation of heat pump capacity and power input as a function of ambient temperature based on test measurements, both for cases where a 5 °F test is conducted and where it isn't.

DOE chose a target wet bulb temperature for the 5 °F test equal to 3.5 °F, corresponding to roughly 60 percent relative humidity which is consistent with the range of relative humidity of the other low temperature heating mode tests.

Issue 24: DOE requests comment on the target wet bulb temperature for the 5 °F test.

Issue 25: DOE requests general comments regarding its proposal to adopt an optional 5 °F test and regarding any other details of the related amendments proposed for calculation of HSPF.

As discussed in this preamble, DOE has proposed changing the ambient temperature requirement for the very-low-temperature heating mode test for variable-speed heat pumps from 2 °F to 5 °F. DOE notes that it proposed a 2 °F test for triple-capacity northern heat pumps in the June 2010 NOPR which was established as part of the test procedure in the June 2016 final rule. 81 FR at 37020 (June 8, 2016).

Issue 26: DOE requests comments on whether the very-low-temperature heating mode test for triple-capacity northern heat pumps should be changed to a 5 °F test for consistency with the proposed 5 °F variable-speed test.

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 an initial regulatory flexibility analysis (IFRA) for any rule that by law must be proposed for public comment, unless the agency certifies that the rule, if promulgated, will not have a significant economic impact on a substantial number of small entities. As 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 proposed rule, which would amend the test procedure for CAC/HP, under the provisions of the Regulatory Flexibility Act and the procedures and policies published on February 19, 2003. DOE has estimated

[27] *http://www.neep.org/initiatives/high-efficiency-products/emerging-technologies/ashp/cold-climate-air-source-heat-pump.*

[28] In contrast, if extrapolation of performance based on the 5 °F and 17 °F tests was used below 5 °F, the capacity would be within the 10% tolerance for none of the heat pumps, and the power input would be within 10% for six percent of the analyzed heat pumps.

the impacts of the test procedure changes on small business manufacturers.

For the purpose of the regulatory flexibility analysis for this rule, the 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 small business size standards published by the SBA to determine whether any small entities would be required to comply with this 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 and are available at *https://www.sba.gov/sites/default/files/files/Size_Standards_Table.pdf*.

CAC/HP manufacturing is classified under NAICS 333415, ''Air Conditioning and Warm Air Heating Equipment and Commercial and Industrial Refrigeration Equipment Manufacturing.'' 70 FR 12395 (March 11, 2005). DOE reviewed publicly available data and contacted various companies on its complete list of manufacturers to determine whether they met the SBA's definition of a small business manufacturer. As a result of this review, DOE identified 22 manufacturers of CAC/HP that would be considered domestic small businesses with a total of less than 3 percent of the market sales.

Issue 27: DOE seeks comment on its estimate of the number of small entities that may be impacted by the proposed test procedure.

Potential impacts of the proposed test procedure on all manufacturers, including small businesses, come from impacts associated with the cost of proposed additional testing. DOE expects that many of the provisions proposed in this notice will result in no increase to test burden. DOE's proposals 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 proposal to require certification of the time delay used when testing coil-only units does not affect testing. DOE's proposal 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 proposal 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 proposal to include an optional test at 5 °F for variable speed heat pumps does not require manufacturers to do any additional testing. Other proposed provisions may increase test burden. DOE anticipates that its proposed 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 proposal 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 is proposing labeling requirements for the indoor and outdoor units of mobile home blower coil and coil-only systems and is also proposing that manufacturers include a specific designation in the installation instructions for these units. For further discussion of the proposed labeling requirements, see section III.C.1. As discussed in that section, DOE expects the additional cost to manufacturers associated with meeting the labeling requirement would be marginal as compared to the total production cost. The overall impact would be small.

As discussed in this preamble, DOE identified 22 domestic small business manufacturers of CAC/HP. 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 CAC/HP 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 CAC/HP 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 as revised by the proposal is $650.

DOE does not expect ICMs to incur any additional burden as a result of the proposed 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 proposed 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 proposed 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 proposed change.

Issue 28: DOE seeks comment on its estimate of the impact of the proposed test procedure amendments on 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 proposed rule, DOE proposes 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 proposed rule would amend the existing test procedures without affecting the amount, quality or distribution of energy usage, and, therefore, would 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 proposed 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 proposed 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 proposed 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, the proposed 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 proposed 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 proposed 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 (Pub. L. 105–277) requires Federal agencies to issue a Family Policymaking Assessment for any rule that may affect family well-being. This rule would 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 would 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 proposed 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 proposed 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 proposed 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 proposed 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 would 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 proposed 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. The Department has evaluated these standards and is unable to conclude whether they fully comply with the requirements of section 32(b) of the FEAA, (*i.e.,* that they were developed in a manner that fully provides for public participation, comment, and review). DOE will consult with the Attorney General and the Chairman of the FTC concerning the impact of these test procedures on competition, prior to prescribing a final rule.

M. Description of Materials Incorporated by Reference

In this SNOPR, DOE proposes to incorporate by reference (IBR) into the proposed 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 proposed in this SNOPR 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 proposed in this SNOPR 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 proposed in this SNOPR 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 proposed in this SNOPR 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 proposed in this SNOPR 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 proposed in this SNOPR 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 proposed in this SNOPR 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 proposed in this SNOPR 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 proposed in this SNOPR 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 SNOPR 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*.

V. Public Participation

A. Attendance at the Public Meeting

The time, date, and location of the public meeting are listed in the **DATES** and **ADDRESSES** sections at the beginning of this document. If you plan to attend the public meeting, please notify the Appliance and Equipment Standards staff at (202) 586–6636 or *Appliance_Standards_Public_Meetings@ee.doe.gov*.

Please note that foreign nationals participating in the public meeting are subject to advance security screening procedures which require advance notice prior to attendance at the public meeting. If a foreign national wishes to participate in the public meeting, please inform DOE as soon as possible by contacting Ms. Regina Washington at (202) 586–1214 or by email: *Regina.Washington@ee.doe.gov* so that the necessary procedures can be completed.

DOE requires visitors to have laptops and other devices, such as tablets, checked upon entry into the building. Any person wishing to bring these devices into the Forrestal Building will be required to obtain a property pass. Visitors should avoid bringing these devices, or allow an extra 45 minutes to check in. Please report to the visitor's desk to have devices checked before proceeding through security.

Due to the REAL ID Act implemented by the Department of Homeland Security (DHS), there have been recent changes regarding ID requirements for individuals wishing to enter Federal buildings from specific states and U.S. territories. Driver's licenses from the following states or territory will not be accepted for building entry and one of the alternate forms of ID listed below will be required. DHS has determined that regular driver's licenses (and ID cards) from the following jurisdictions are not acceptable for entry into DOE facilities: Alaska, American Samoa, Arizona, Louisiana, Maine, Massachusetts, Minnesota, New York, Oklahoma, and Washington. Acceptable alternate forms of Photo-ID include: U.S. Passport or Passport Card; an Enhanced

Driver's License or Enhanced ID-Card issued by the states of Minnesota, New York or Washington (Enhanced licenses issued by these states are clearly marked Enhanced or Enhanced Driver's License); a military ID or other Federal government issued Photo-ID card.

In addition, you can attend the public meeting via webinar. Webinar registration information, participant instructions, and information about the capabilities available to webinar participants will be published on DOE's Web site at: *https://www1.eere.energy.gov/buildings/appliance_standards/standards.aspx?productid=48&action=viewlive*. Participants are responsible for ensuring their systems are compatible with the webinar software.

B. Procedure for Submitting Prepared General Statements for Distribution

Any person who has plans to present a prepared general statement may request that copies of his or her statement be made available at the public meeting. Such persons may submit requests, along with an advance electronic copy of their statement in PDF (preferred), Microsoft Word or Excel, WordPerfect, or text (ASCII) file format, to the appropriate address shown in the **ADDRESSES** section at the beginning of this document. The request and advance copy of statements must be received at least one week before the public meeting and may be emailed, hand-delivered, or sent by mail. DOE prefers to receive requests and advance copies via email. Please include a telephone number to enable DOE staff to make follow-up contact, if needed.

C. Conduct of the Public Meeting

DOE will designate a DOE official to preside at the public meeting and may also use a professional facilitator to aid discussion. The meeting will not be a judicial or evidentiary-type public hearing, but DOE will conduct it in accordance with section 336 of EPCA (42 U.S.C. 6306). A court reporter will be present to record the proceedings and prepare a transcript. DOE reserves the right to schedule the order of presentations and to establish the procedures governing the conduct of the public meeting. After the public meeting, interested parties may submit further comments on the proceedings as well as on any aspect of the rulemaking until the end of the comment period.

The public meeting will be conducted in an informal, conference style. DOE will present summaries of comments received before the public meeting, allow time for prepared general statements by participants, and encourage all interested parties to share their views on issues affecting this rulemaking. Each participant will be allowed to make a general statement (within time limits determined by DOE), before the discussion of specific topics. DOE will allow, as time permits, other participants to comment briefly on any general statements.

At the end of all prepared statements on a topic, DOE will permit participants to clarify their statements briefly and comment on statements made by others. Participants should be prepared to answer questions by DOE and by other participants concerning these issues. DOE representatives may also ask questions of participants concerning other matters relevant to this rulemaking. The official conducting the public meeting will accept additional comments or questions from those attending, as time permits. The presiding official will announce any further procedural rules or modification of the above procedures that may be needed for the proper conduct of the public meeting.

A transcript of the public meeting will be included in the docket, which can be viewed as described in the Docket section at the beginning of this document. In addition, any person may buy a copy of the transcript from the transcribing reporter.

D. Submission of Comments

DOE will accept comments, data, and information regarding this proposed rule no later than the date provided in the **DATES** section at the beginning of this proposed rule. Interested parties may submit comments using any of the methods described in the **ADDRESSES** section at the beginning of this notice.

Under EPCA, DOE may not amass more than 270 days of public comment during a test procedure rulemaking. (42 U.S.C. 6293(b)(2)) Since the beginning of this test procedure rulemaking on June 2, 2010 (75 FR 31223), DOE has provided 216 days of public comment, in all.[29] Thus, DOE is providing 30 days of public comment for this SNOPR to ensure that parties have a chance to comment throughout the rest of this rulemaking.

Submitting comments via regulations.gov. The *regulations.gov* Web page will require you to provide your name and contact information. Your contact information will be viewable to DOE Building Technologies staff only. Your contact information will not be publicly viewable except for your first and last names, organization name (if any), and submitter representative name (if any). If your comment is not processed properly because of technical difficulties, DOE will use this information to contact you. If DOE cannot read your comment due to technical difficulties and cannot contact you for clarification, DOE may not be able to consider your comment.

However, your contact information will be publicly viewable if you include it in the comment or in any documents attached to your comment. Any information that you do not want to be publicly viewable should not be included in your comment, nor in any document attached to your comment. Persons viewing comments will see only first and last names, organization names, correspondence containing comments, and any documents submitted with the comments.

Do not submit to *regulations.gov* information for which disclosure is restricted by statute, such as trade secrets and commercial or financial information (hereinafter referred to as Confidential Business Information (CBI)). Comments submitted through *regulations.gov* cannot be claimed as CBI. Comments received through the Web site will waive any CBI claims for the information submitted. For information on submitting CBI, see the Confidential Business Information section.

DOE processes submissions made through *regulations.gov* before posting. Normally, comments will be posted within a few days of being submitted. However, if large volumes of comments are being processed simultaneously, your comment may not be viewable for up to several weeks. Please keep the comment tracking number that *regulations.gov* provides after you have successfully uploaded your comment.

Submitting comments via email, hand delivery, or mail. Comments and documents submitted via email, hand delivery, or mail also will be posted to *regulations.gov*. If you do not want your personal contact information to be publicly viewable, do not include it in your comment or any accompanying documents. Instead, provide your contact information on a cover letter. Include your first and last names, email address, telephone number, and optional mailing address. The cover letter will not be publicly viewable as long as it does not include any comments

[29] This includes the comment period from the April 2011 SNOPR and the comment period extension, the October 2011 SNOPR and its comment period extension, and the November 2015 SNOPR. *See* 76 FR 18105 (April 1, 2011); 76 FR 30555 (May 26, 2011); 76 FR 65616 (Oct. 24, 2011); 76 FR 79135 (Dec. 21, 2011); 80 FR 69277 (Nov. 9, 2015).

Include contact information each time you submit comments, data, documents, and other information to DOE. If you submit via mail or hand delivery, please provide all items on a CD, if feasible. It is not necessary to submit printed copies. No facsimiles (faxes) will be accepted.

Comments, data, and other information submitted to DOE electronically should be provided in PDF (preferred), Microsoft Word or Excel, WordPerfect, or text (ASCII) file format. Provide documents that are not secured, written in English and free of any defects or viruses. Documents should not contain special characters or any form of encryption and, if possible, they should carry the electronic signature of the author.

Campaign form letters. Please submit campaign form letters by the originating organization in batches of between 50 to 500 form letters per PDF or as one form letter with a list of supporters' names compiled into one or more PDFs. This reduces comment processing and posting time.

Confidential Business Information. According to 10 CFR 1004.11, any person submitting information that he or she believes to be confidential and exempt by law from public disclosure should submit via email, postal mail, or hand delivery two well-marked copies: One copy of the document marked confidential including all the information believed to be confidential, and one copy of the document marked non-confidential with the information believed to be confidential deleted. Submit these documents via email or on a CD, if feasible. DOE will make its own determination about the confidential status of the information and treat it according to its determination.

Factors of interest to DOE when evaluating requests to treat submitted information as confidential include: (1) A description of the items; (2) whether and why such items are customarily treated as confidential within the industry; (3) whether the information is generally known by or available from other sources; (4) whether the information has previously been made available to others without obligation concerning its confidentiality; (5) an explanation of the competitive injury to the submitting person which would result from public disclosure; (6) when such information might lose its confidential character due to the passage of time; and (7) why disclosure of the information would be contrary to the public interest.

It is DOE's policy that all comments may be included in the public docket, without change and as received, including any personal information provided in the comments (except information deemed to be exempt from public disclosure).

E. Issues on Which DOE Seeks Comment

Although DOE welcomes comments on any aspect of this proposal, DOE is particularly interested in receiving comments and views of interested parties concerning the following issues:

Issue 1: DOE requests comment on its proposed certification requirements for outdoor units with no match. Also, DOE seeks comment on what fin style options should be considered as options for CCMS database data entry.

Issue 2: DOE requests comment on its proposed language in 429.16 related to allowable ICM ratings and compliance with regional standards.

Issue 3: DOE requests comment on its proposal to allow a one-sided tolerance on represented values of cooling and heating capacity that allows underrating of any amount but only overrating up to 5 percent.

Issue 4: DOE seeks comments from interested parties about its proposal to impose time delays to allow approach to equilibrium for measurements of off-mode power for units with self-regulating crankcase heaters. DOE requests comment regarding the 4-hour and 8-hour delay times proposed for units without and with compressor sound blankets, respectively.

Issue 5: DOE requests comment on its proposal to limit the internal volume of pressure measurement systems for cooling/heating heat pumps where the pressure measurement location may switch from liquid to vapor state when changing operating modes and for all systems undergoing cyclic tests. DOE also requests comment specifically on (a) the proposed 0.25 cubic inch per 12,000 Btu/h maximum internal volume for such systems, and (b) the proposals for default internal volumes to assign to pressure transducers and gauges of 0.1 and 0.2 cubic inches, respectively.

Issue 6: DOE requests comment on the proposal to require the use of a bin-by-bin method to calculate EER and COP for intermediate-speed operation for SEER and HSPF calculations for variable-speed units.

Issue 7: DOE requests comment on its proposed modifications to requirements when using the outdoor air enthalpy method as the secondary test method, including its proposal that the official test be conducted without the outdoor air-side test apparatus connected.

Issue 8: DOE requests comments on its proposal to require certification reports for coil-only units to indicate whether testing was conducted using a time-delay relay to provide an off-cycle time delay, and the duration of the time delay.

Issue 9: DOE requests comment on its proposal to limit the NGIFS of tested coil-only single-split systems to 2.0 sq.in/Btu/hr.

Issue 10: DOE requests comments on its proposal to require that full-speed tests conducted in 17 °F and 35 °F ambient temperatures use the maximum compressor speed at which the system controls would operate the compressor in normal operation in a 17 °F ambient temperatures. DOE requests comment on the proposed approach of using standardized slope factors for calculation of representative performance at 47 °F ambient temperature for heat pumps for which the 47 °F full-speed test cannot be conducted at the same speed as the 17 °F full-speed test. Further, DOE requests comment on the specific slope factors proposed, and/or data to show that different slope factors should be used.

Issue 11: DOE requests comments on its proposal to allow the full speed test in 47 °F ambient temperature that is used to represent heat pump heating capacity, to use any speed that is no lower than used for the 95 °F full-speed cooling test for Appendix M.

Issue 12: DOE requests comments on its clarifications regarding use of break-in, including use of the certified break-in period for each compressor of the unit, regardless of who conducts the test, prior to any test period used to measure performance.

Issue 13: DOE requests comments on removing from section 2.2.3.a of Appendix M the 5 percent tolerance for part load operation when comparing the sum of nominal capacities of the indoor units and the intended system part load capacity.

Issue 14: DOE requests comment on whether removing the statement about insulating or sealing cased coils in Appendix M, section 2.2.c would be sufficient to avoid confusion regarding whether sealing of duct connections is allowed.

Issue 15: DOE requests comments on the proposed minimum external static pressure requirements.

DOE proposes 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 may 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 are still 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 are also 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. DOE proposes 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.

Issue 16: DOE requests comment on the proposed definitions for kinds of CAC/HP associated with administering minimum external static pressure requirements.

Issue 17: DOE requests comments on not including a reduced minimum external static pressure requirement for blower coil or single-package systems tested with a condensing furnace.

Issue 18: DOE requests comment on the proposed default fan power value for coil-only mobile home systems. DOE also requests mobile home indoor fan performance data for units of all capacities and that use all available motor technologies in order to allow confirmation that the proposed default value is a good representation for mobile home units.

Issue 19: DOE requests comments on its proposed definition for mobile home coil-only unit.

Issue 20: DOE requests comments on the adjustments to the proposals for calculating HSPF for heat pumps and SEER for variable-speed heat pumps.

Issue 21: DOE requests comments on the adjusted values of minimum HSPF based on the HSPF efficiency levels recommended by the CAC/HP ECS Working Group.

Issue 22: DOE requests comment on its proposal to require use of an alternative HSPF rating approach (for heat pumps that raise minimum compressor speed in ambient temperatures that impact the HSPF calculation) that estimates minimum-speed performance (a) between 35 °F and 47 °F using the intermediate-speed frosting-operation test at 35 °F and the minimum-speed test at 47 °F, and (b) below 35 °F assuming that minimum-speed and intermediate-speed performance are the same. In addition, DOE requests comment on including in certification reports for variable-speed heat pumps whether this alternative approach was used to determine the rating. Finally, DOE requests comment on whether any of the additional tests that could be used to further improve the accuracy of variable-speed heat pump performance estimates should be required in the test procedure.

Issue 23: DOE requests comment on the proposals for evaluation of heat pump capacity and power input as a function of ambient temperature based on test measurements, both for cases where a 5 °F test is conducted and where it isn't.

Issue 24: DOE requests comment on the target wet bulb temperature for the 5 °F test.

Issue 25: DOE requests general comments regarding its proposal to adopt an optional 5 °F test and regarding any other details of the related amendments proposed for calculation of HSPF.

Issue 26: DOE requests comments on whether the very-low-temperature heating mode test for triple-capacity northern heat pumps should be changed to a 5 °F test for consistency with the proposed 5 °F variable-speed test.

Issue 27: DOE seeks comment on its estimate of the number of small entities that may be impacted by the proposed test procedure.

Issue 28: DOE seeks comment on its estimate of the impact of the proposed test procedure amendments on small entities.

VI. Approval of the Office of the Secretary

The Secretary of Energy has approved publication of this proposed 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 August 1, 2016.

Kathleen B. Hogan,

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

For the reasons stated in the preamble, DOE is proposing to amend 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.

■ 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 sections 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. 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.

* * * * *

■ 3. Section 429.16 is amended by:

■ a. Revising paragraph (a)(1);

■ b. Redesignating paragraphs (a)(3) and (a)(4) as (a)(4) and (a)(5) and revising newly designated (a)(4)(i);

■ c. Adding new paragraph (a)(3);

■ d. Revising paragraph (b)(2)(i);

■ e. Revising the introductory text of paragraph (b)(3)(i), and revising paragraphs (b)(3)(iii) and (b)(3)(iv);

■ f. Revising paragraphs (c)(1)(i)(B), (c)(3), (d)(3) and (d)(4);

■ g. Revising paragraphs (e)(2), (e)(3) and (e)(4); and

■ h. Revising paragraphs (f) introductory text, (f)(1), (f)(2), (f)(4), and (f)(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, $P_{W,OFF}$, 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).	Every individual model distributed in commerce.
	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)).	Every individual combination distributed in commerce, including all coil-only and blower coil combinations. Every outdoor unit and indoor unit combination, must have a coil-only rating. 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 the particular model of outdoor unit.
	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.	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).	Every individual combination distributed in commerce.
	Single-Split-System Heat Pump (including Space-Constrained and SDHV).	
	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.* 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 with multiple refrigerants, a manufacturer must determine all represented values for each refrigerant that can be used in an individual combination of the basic model (including outdoor units with no match or ''tested combinations'') without voiding the manufacturer's warranty. This requirement may apply across the listed categories in the table in paragraph (a)(1) of this section. If the warranty information specifies acceptable refrigerant characteristics rather than specific refrigerants and HCFC–22 meets these characteristics, a manufacturer must determine represented values (including SEER, EER, HSPF, $P_{W,OFF}$, cooling capacity, and heating capacity, as applicable) for, at a minimum, an outdoor unit with no match. If a model of outdoor unit (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 without a specific refrigerant specified or not charged with a specified refrigerant from the point of manufacture, if the unit is shipped requiring addition of more than a pound of refrigerant to meet the charge recommended by the manufacturer's installation instructions (or section 2.2.5 of appendix M or appendix M1), or if the unit is shipped with any amount of charge of R–407C, a manufacturer must determine represented values (including SEER, EER, HSPF, $P_{W,OFF}$, cooling capacity, and heating capacity, as applicable) for, at a minimum, an outdoor unit with no match.

(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 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). Single-Package HP (including Space-Constrained).	The lowest SEER individual model.	N/A.
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 meeting the requirements of section 2.2h of appendix M or M1 to subpart B of part 430.
	Single-Split-System AC with Other Than Single-Stage or Two-Stage Compressor (including Space-Constrained and SDHV). Single-Split-System HP (including Space-Constrained and SDHV).	The model of outdoor unit.	A model of indoor unit. If the tested model of indoor unit is coil-only, it must meet the requirements of section 2.2h of appendix M or M1 to subpart B of part 430.
	Multi-Split, Multi-Circuit, or Multi-Head Mini-Split Split System—non-SDHV.	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 as certified by the OUM. If there are multiple models of outdoor unit with the same lowest SEER represented value, the ICM may select one for testing purposes.
	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 as certified by the OUM. If there are multiple models of outdoor unit with the same lowest SEER represented value, 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:

* * * * *

(iii) *Cooling Capacity.* The represented value of cooling capacity must be a self-declared value that is no more than 105 percent of the mean of the cooling capacities measured for the units in the sample, rounded:

(A) To the nearest 100 Btu/h if cooling capacity is less than 20,000 Btu/h,

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

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

(iv) *Heating Capacity.* The represented value of heating capacity must be a self-declared value that is no more than 105 percent of the mean of the heating capacities measured for the units in the sample, rounded:

(A) To the nearest 100 Btu/h if heating capacity is less than 20,000 Btu/h,

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

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

(c) * * *

(1) * * *

(i) * * *

(B) The representative 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 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.

* * * * *

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

(i) 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 in accordance with 10 CFR part 430, subpart B, appendix M1 and according to the requirements in paragraph (b)(3)(i) of this section.

(ii) For basic models composed of both non-ducted and ducted combinations, the represented value based on testing in accordance with 10 CFR part 430, subpart B, appendix M 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)(i) of this section. 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 based on testing in accordance with 10 CFR part 430, subpart B, appendix M1 for the mixed combination is the mean of the represented values for the individual component combinations as determined in accordance with paragraph (b)(3)(i) of this section.

(iii) For basic models composed of both SDHV and non-ducted or ducted combinations, the represented value based on testing in accordance with 10 CFR part 430, subpart B, appendix M 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)(i) of this section. 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 based on testing in accordance with 10 CFR part 430, subpart B, appendix M1 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)(i) 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) * * *

(3) Cooling capacity. The represented value of cooling capacity of an individual model/combination must be no greater than 105% of 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 105% of 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: The seasonal energy efficiency ratio (SEER 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 amended 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, the heating seasonal performance factor (HSPF in British thermal units per Watt-hour (Btu/W-h));

(ii) For central air conditioners (excluding space constrained products), the energy efficiency ratio (EER in British thermal units per Watt-hour (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 current 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 No.	Individual model Nos.		
		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	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 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).	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 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; the duration of the crankcase heater time delay for the shoulder season and heating season, if such time delay is employed; the maximum time between defrosts as allowed by the controls (in hours); 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^c value or whether the default value was used;

(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; and

(viii) For variable speed heat pumps, whether the optional $H4_2$ low temperature test was used to characterize performance at temperatures below 17 °F, whether the $H1_N$ or $H1_2$ test speed is the same as the $H3_2$ test speed, 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;

(x) For single-split-system coil-only ratings, NGIFS and the OFF-cycle time delay for the indoor fan, if used for certification testing; and

(xi) 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 (A) if using appendix M to subpart B of part 430 or the value determined in paragraph (B) 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)(i)(C) of this section, divided by the represented value of SEER, in Btu's per watt-hour, as determined in paragraph (b)(3)(i)(B) 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 SEER, 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)(i)(B) 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 (HSPF), 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 (A) if using appendix M to subpart B of part 430 or the value determined in paragraph (B) 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)(i)(C) of this section, divided by the represented value of SEER, in Btu's per watt-hour, as determined in paragraph (b)(3)(i)(B) 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 SEER, in Btu's per watt-hour, as determined in paragraph (b)(3)(i)(B) of this section;

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

(A) the estimated number of regional cooling load hours per year determined from Table 21 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 20 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 21 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)(i)(B);

(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 20 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 HSPF, 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 paragraph (e)(2)(i) and the introductory text of paragraph (e)(5)(iv) to read as follows:

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

* * * * *

(e) * * *

(2) * * *

(i) Conduct minimum testing and compare to AEDM output as described in paragraphs (A) and (B) respectively.

(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 greater than 105 percent of 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:

* * * * *

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■ 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.

§ 430.3 [Amended]

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

■ 7. 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 A2 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 or M1 to this subpart, and round off to the nearest 0.025 Btu/W-h.

(3) Determine EER as described in section 4.7 of appendix M or M1 to this subpart, and round off to the nearest 0.025 Btu/W-h.

(4) Determine heating seasonal performance factors (HSPF) as described in section 4.2 of appendix M or 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.

* * * * *

■ 8. Appendix M to subpart B of part 430 is amended by:

■ a. Revising the definition of ''service coil'' in Section 1.2., Definitions;

■ b. Revising paragraph c. and adding paragraphs g. and h. in Section 2.2, Test Unit Installation Requirements;

■ c. Revising paragraph a. in section 2.2.3;

■ d. Removing in, Section 2.10.1, paragraph (c) first sentence, the word ''preliminary'' and adding in its place the word ''non-ducted'';

■ e. Revising section 3.1.7;

■ f. Revising the introductory paragraph of section 3.5.1;

■ g. Revising section 3.6.4;

■ h. Revising section 3.11.1;

■ i. Revising section 3.11.1.1;

■ j. Revising section 3.11.1.2;

■ l. Revising paragraphs b., and d., in section 3.13.2;

■ m. Revising the last paragraph in section 4.1.3;

■ n. Revising section 4.1.4.2;

■ o. Revising paragraph b., in section 4.2;

■ p. Redesignating paragraph c. as paragraph d. in section 4.2 and adding paragraph c., respectively;

■ q. Revising the first paragraph in section 4.2.3;

■ r. Revising the second paragraph in section 4.2.4; and
■ s. Revising section 4.2.4.2.

The additions and revisions 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

* * * * *

1.2 Definitions

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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.

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2.2 Test Unit Installation Requirements.

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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 3, 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.

* * * * *

g. If pressure measurement devices are connected to refrigerant lines at locations where the refrigerant state changes from liquid to vapor for different parts of the test (*e.g.* heating mode vs. cooling mode, on-cycle vs. off-cycle during cyclic test), the total internal volume of the pressure measurement system (transducers, gauges, connections, and lines) must be no more than 0.25 cubic inches per 12,000 Btu/h certified cooling capacity. Calculate total system internal volume 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.

h. For single-split-system coil-only air conditioners, test using an indoor coil that has a normalized gross indoor fin surface (NGIFS) no greater than 2.0 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.
$\dot{Q}_c$ = 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.

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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.

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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.

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3.1 * * *

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3.1.7 Test Sequence

Before making test measurements used to calculate performance, operate the equipment for a ''break-in'' period, which may not exceed 20 hours. Each compressor of the unit must undergo this ''break-in'' period. Record the duration of the break-in period. When testing a ducted unit (except if a heating-only heat pump), conduct the A or A2 Test first to establish the cooling full-load air volume rate. * * *

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3.5.1 Procedures When Testing Ducted Systems

The automatic controls that are normally installed with the test unit must govern the OFF/ON cycling of the air moving equipment on the indoor side (exhaust fan of the airflow measuring apparatus and 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. * * *

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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 ($H2_2$). 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 13. The compressor shall operate at the same heating full speed, measured by RPM or power input frequency (Hz), equal to 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. 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. 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 13 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 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:

$\dot{Q}_{hcalc}^{k=2}(47) = \dot{Q}_h^{k=2}(17) * (1 + 30\ °F * CSF)$;

$\dot{E}_{hcalc}^{k=2}(47) = \dot{E}_h^{k=2}(17) * (1 + 30\ °F * PSF)$

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 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:

$\dot{Q}_h^{k=2}(35) = 0.90 * \{\dot{Q}_h^{k=2}(17) + 0.6 * [\dot{Q}_{hcalc}^{k=2}(47) - \dot{Q}_h^{k=2}(17)]\}$

$\dot{E}_h^{k=2}(35) = 0.985 * \{\dot{E}_h^{k=2}(17) + 0.6 * [\dot{E}_{hcalc}^{k=2}(47) - \dot{E}_h^{k=2}(17)]\}$

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 $\dot{E}_h^{k=2}(5)$ and $\dot{E}_h^{k=2}(5)$ from the $H4_2$ test, and evaluate all four according to section 3.10.

TABLE 13—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[4]	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] Defined in section 3.1.4.6 of this appendix.

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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 (''non-ducted'' 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'' test). No other cooling mode or heating mode tests require the ducted 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 test for each cooling mode test where a different fan speed is first used. This same requirement applies for the heating mode tests.

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3.11.1.1 Non-Ducted Test

a. For the non-ducted 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 8 or Table 15, whichever applies, test tolerances are satisfied.

b. For cases where a ducted test is not required per section 3.11.1.b of this appendix, the non-ducted test constitutes the ''official'' test for which validity is not based on comparison with a secondary test.

c. For cases where a ducted test is required per section 3.11.1.b of this appendix, the following conditions must be met for the non-ducted 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 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 and non-ducted tests must agree within 2.0 percent.

3.11.1.2 Ducted Test

a. The test conditions and tolerances for the ducted test are the same as specified for the official test.

b. After collecting 30 minutes of steady-state data during the non-ducted test, connect the outdoor air-side test apparatus to the unit for the ducted 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 non-ducted test. Calculate the averages for the ducted test using five or more consecutive readings taken at one minute intervals. Make these consecutive readings after re-establishing equilibrium conditions.

c. During the ducted test, at one minute intervals, measure the parameters required according to the indoor air enthalpy method and the outdoor air enthalpy method.

d. For cooling mode ducted tests, calculate capacity based on outdoor air-enthalpy measurements as specified in sections 7.3.3.2 and 7.3.3.3 of 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 ASHRAE Standard. Adjust the outdoor-side capacity according to section 7.3.3.4 of ASHRAE 37–2009 to account for line losses when testing split systems.

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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.

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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.

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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.

4.1 * * *

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4.1.3 SEER Calculations for an Air Conditioner or Heat Pump Having a Two-Capacity Compressor

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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.

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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 18. 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;

$EER^{k=2}(T_j)$ 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.

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4.2 * * *

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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), $Q^k_h(47) = \dot{Q}^{k=1}{}_h(47)$, the space heating capacity determined from the $H1_1$ test.

For all heat pumps, HSPF accounts for * * *

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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.

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4.2.4 * * *

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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. * * *

* * * * *

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.

■ 9. 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 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 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 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.

Ceiling-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 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.

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 must be determined 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).

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.

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 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.''

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 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 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.

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 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).

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 3 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**	Gen-eral	Cool-ing*	Heat-ing**
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 4	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	2.1; 2.2.a; 2.2.b; 2.2.c; 2.2.1;	3.1 (except	3.3-3.5	3.7–3.10	4.4;	4.1	4.2

		VRF SDHV†	2.2.2; 2.2.3.a; 2.2.3.c; 2.2.4; 2.2.5; 2.4-2.12	3.1.3, 3.1.4) 3.1.4.1.1c; 3.11-3.13			4.5		
	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 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. 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 3, 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 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.
$\dot{Q}_c$ = 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, 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 refrigerant lines at locations where the refrigerant state changes from liquid to vapor for different parts of the test (*e.g.* heating mode vs. cooling mode, on-cycle vs. off-cycle during cyclic test), the total internal volume of the pressure measurement system (transducers, gauges, connections, and lines) must be no more than 0.25 cubic inches per 12,000 Btu/h certified cooling capacity. Calculate total system internal volume 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.

h. For single-split-system coil-only air conditioners, test using an indoor coil that has a normalized gross indoor fin surface (NGIFS) no greater than 2.5 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.
$\dot{Q}_c$ = 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.

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 19 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 4 to 7. 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 4–7 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 11 to 14. 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:

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

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·ft^2· °F/Btu.

2.4.1 Outlet Plenum for the Indoor Unit

a. Attach a plenum to the outlet of the indoor coil. (Note: For some packaged systems, the indoor coil may be located in the outdoor test room.)

b. For systems having multiple indoor coils, 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. 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 2. 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 2—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 2.1.5.2 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 the two special cases noted in preamble. 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 ·ft² · °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·ft² · °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 · ft² · °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 of this appendix 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 non-ducted 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 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 · ft² · °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 all steady-state tests (*i.e.,* the A, A_2, A_1, B, B_2, B_1, C, C_1, EV, F_1, G_1, $H0_1$, H_1, $H1_2$, $H1_1$, HI_N, H_3, $H3_2$, and $H3_1$ Tests), in addition, use one of the acceptable secondary methods specified in section 2.10 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 3, 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 3, 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 3 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 3 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 3 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 3—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 3 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 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 3);
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 3 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 3 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 3 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 3 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 3 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 3 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:

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 0.2 (for all other systems). Table 4 specifies test conditions for these four tests.

TABLE 4—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 4. 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 5—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 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 4).

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 6—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 7 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 7 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 7 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 7 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 7—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]

TABLE 7—COOLING MODE TEST CONDITION FOR UNITS HAVING A VARIABLE-SPEED COMPRESSOR—Continued

Test description	Air entering indoor unit temperature (°F)		Air entering outdoor unit temperature (°F)		Compressor speed	Cooling air volume rate
	Dry bulb	Wet bulb	Dry bulb	Wet bulb		
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 8 are satisfied. For those continuously recorded parameters, use the entire data set from the 30-minute interval to evaluate Table 8 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 ducted coil-only system tests, decrease $\dot{Q}_c^k(T)$ by

$$\frac{1385\ Btu/h}{1000\ scfm} * \bar{\dot{V}}_s$$

and increase $\dot{E}_c^k(T)$ 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 ducted coil-only system tests, decrease $\dot{Q}_c^k(T)$ by

$$\frac{1505\ Btu/h}{1000\ scfm} * \bar{\dot{V}}_s$$

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

TABLE 8—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:		

TABLE 8—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—Continued

	Test operating tolerance [1]	Test condition tolerance [1]
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, $\bar{\dot{V}}$, 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 9 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 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 9 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 9—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 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 9 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,

$$q_{cyc,dry} = \frac{60 * \bar{\dot{V}} * C_{p,a} * \Gamma}{[v_n' * (1 + W_n)]} = \frac{60 * \bar{\dot{V}} * C_{p,a} * \Gamma}{v_n} \quad \text{and} \quad \Gamma = F_{CD}^* \int_{\tau_1}^{\tau_2} [T_{a1}(\tau) - T_{a2}(\tau)] \delta\tau, \ hr * {}^\circ F$$

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

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

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

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

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

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 normally installed with the test unit must govern the OFF/ON cycling of the air moving equipment on the indoor side (exhaust fan of the airflow measuring apparatus and 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 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 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,

Equation 3.5-3. $\frac{1505\, Btu/h}{1000\, scfm} * \bar{\dot{V}}_s * [\tau_2 - \tau_1]$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 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 0.2 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

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 of this section.

TABLE 10—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]

TABLE 10—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—Continued

Test description	Air entering indoor unit temperature (°F)		Air entering outdoor unit temperature (°F)		Heating air volume rate
	Dry bulb	Wet bulb	Dry bulb	Wet bulb	
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 11. 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:

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

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 11—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 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:

$$\dot{Q}_h^{k=1}(35) = 0.90 * \{\dot{Q}_h^{k=1}(17) + 0.6 * [\dot{Q}_h^{k=1}(47) - \dot{Q}_h^{k=1}(17)]\}$$

$$\dot{E}_h^{k=1}(35) = 0.985 * \{\dot{E}_h^{k=1}(17) + 0.6 * [\dot{E}_h^{k=1}(47) - \dot{E}_h^{k=1}(17)]\}$$

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 12 specifies test conditions for these nine tests.

TABLE 12—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 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 13. 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 13 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 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 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 HSPF as follows:

$\dot{Q}^{k=2}_{hcalc}(47) = \dot{Q}^{k=2}_h(17) * (1 + 30\,°F * CSF)$;

$\dot{E}^{k=2}_{hcalc}(47) = \dot{E}^{k=2}_h(17) * (1 + 30\,°F * PSF)$

Where:

$\dot{Q}^{k=2}_{hcalc}(47)$ and $\dot{E}^{k=2}_{hcalc}(47)$ are the capacity and power input representing full-speed operation at 47 °F for the HSPF calculations,

$\dot{Q}^{k=2}_h(17)$ is the capacity measured in the $H3_2$ test,

$\dot{E}^{k=2}_h(17)$ 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:

$\dot{Q}^{k=2}_h(35) = 0.90 * \{\dot{Q}^{k=2}_h(17) + 0.6 * [\dot{Q}^{k=2}_{hcalc}(47) - \dot{Q}^{k=2}_h(17)]\}$

$\dot{E}^{k=2}_h(35) = 0.985 * \{\dot{E}^{k=2}_h(17) + 0.6 * [\dot{E}^{k=2}_{hcalc}(47) - \dot{E}^{k=2}_h(17)]\}$

Where:

$\dot{Q}^{k=2}_{hcalc}(47)$ and $\dot{E}^{k=2}_{hcalc}(47)$ are the capacity and power input representing full-speed operation at 47 °F for the HSPF calculations, calculated as described in section b above.

$\dot{Q}^{k=2}_h(17)$ and $\dot{E}^{k=2}_h(17)$ 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 13—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	3.5	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 13 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 13 $H2_V$ 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 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:

$$\dot{Q}^{k=1}_h(35) = 0.90 * \{\dot{Q}^{k=1}_h(17) + 0.6 * [\dot{Q}^{k=1}_h(47) - \dot{Q}^{k=1}_h(17)]\}$$

$$\dot{E}^{k=1}_h(35) = 0.985 * \{\dot{E}^{k=1}_h(17) + 0.6 * [\dot{E}^{k=1}_h(47) - \dot{E}^{k=1}_h(17)]\}$$

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:

$$\dot{Q}_h^{k=3}(35) = \dot{Q}R_h^{k=2}(35) * \{\dot{Q}_h^{k=3}(17) + 1.20 * [\dot{Q}_h^{k=3}(17) - \dot{Q}_h^{k=3}(2)]\}$$

$$\dot{E}_h^{k=3}(35) = PR_h^{k=2}(35) * \{\dot{E}_h^{k=3}(17) + 1.20 * [\dot{E}_h^{k=3}(17) - \dot{E}_h^{k=3}(2)]\}$$

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 $Q_h^{k=2}(47)$ and $E_h^{k=2}(47)$ from the $H1_2$ test and evaluate them according to section 3.7 of this appendix. Determine the quantities $Q_h^{k=2}(35)$ and $E_h^{k=2}(35)$ from the $H2_2$ test and evaluate them according to section 3.9.1 of this appendix. Determine the quantities $Q_h^{k=2}(17)$ and $E_h^{k=2}(17)$ from the $H3_2$ test, determine the quantities $Q_h^{k=3}(17)$ and $E_h^{k=3}(17)$ from the $H3_3$ test, and determine the quantities $Q_h^{k=3}(2)$ and $E_h^{k=3}(2)$ from the $H4_3$ test. Evaluate all six quantities according to section 3.10 of this appendix. Use the paired values of $Q_h^{k=3}(35)$ and $E_h^{k=3}(35)$ 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 HSPF 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 14 specifies test conditions for all 13 tests.

TABLE 14—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]

TABLE 14—HEATING MODE TEST CONDITIONS FOR UNITS WITH A TRIPLE-CAPACITY COMPRESSOR—Continued

Test description	Air entering indoor unit temperature °F		Air entering outdoor unit temperature °F		Compressor capacity	Heating air volume rate
	Dry bulb	Wet bulb	Dry bulb	Wet bulb		
$H4_3$ Test (required, steady)	70	60 (max)	2	1	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 HSPF 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 15 are satisfied. For those continuously recorded parameters, use the entire data set for the 30-minute interval when evaluating Table 15 compliance. Determine the average electrical power consumption of the heat pump over the same 30-minute interval.

TABLE 15—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 mobile home 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 $\bar{\dot{V}}_s$ is the average measured indoor air volume rate expressed in units of cubic feet per minute of standard air (scfm).

For non-mobile home 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 $\bar{\dot{V}}_s$ 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 15 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 16 rather than Table 9. 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 Γ 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 16—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 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 17 during both the preliminary and official test periods. As noted in Table 17, 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 17) and (2) when defrosting, plus these same first 10 minutes after defrost termination (sub-interval D, as described in Table 17). Evaluate compliance with Table 17 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 17 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 17—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

TABLE 17—TEST OPERATING AND TEST CONDITION TOLERANCES FOR FROST ACCUMULATION HEATING MODE TESTS—Continued

	Test operating tolerance[1]		Test condition tolerance[1]
	Sub-interval H[2]	Sub-interval D[3]	Sub-interval H[2]
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:

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

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.

$\Gamma = \int^{tau;2}\tau_1[T_{a2}(\tau) - T_{a1}(\tau)]d\tau, hr * {}^\circ F$

$T_{a1}(\tau)$ = 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 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 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, $H3_2$, $H3_1$, $H3_3$, $H4_3$, and $H4_2$ 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 connected to the outdoor unit (''non-ducted'' 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'' test). No other cooling mode or heating mode tests require the ducted 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 test for each cooling mode test where a different fan speed is first used. This same requirement applies for the heating mode tests.

3.11.1.3 Non-Ducted Test

a. For the non-ducted 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 8 or Table 15, whichever applies, test tolerances are satisfied.

b. For cases where a ducted test is not required per section 3.11.1.b of this appendix, the non-ducted test constitutes the ''official'' test for which validity is not based on comparison with a secondary test.

c. For cases where a ducted test is required per section 3.11.1.b of this appendix, the following conditions must be met for the non-ducted test to constitute a valid ''official'' test:

(1) The energy balance specified in section 3.1.1 is achieved for the ducted 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 and non-ducted tests must agree within 2.0 percent.

3.11.1.4 Ducted Test

a. The test conditions and tolerances for the ducted test are the same as specified for the official test.

b. After collecting 30 minutes of steady-state data during the non-ducted test, connect the outdoor air-side test apparatus to the unit for the ducted 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 non-ducted test. Calculate the averages for the ducted test using five or more consecutive readings taken at one minute intervals. Make these consecutive readings after re-establishing equilibrium conditions.

c. During the ducted test, at one minute intervals, measure the parameters required according to the indoor air enthalpy method and the outdoor air enthalpy method.

d. For cooling mode ducted 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.

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 8 (cooling) or the Table 15 (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 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 ($P2_x$) 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 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 (SEER) Calculations. Calculate SEER 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,

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

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 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 18. 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 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,

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

$$\text{Equation 4.1.3-2} \quad \dot{E}_c^{k=1}(T_j) = \dot{E}_c^{k=1}(67) + \frac{\dot{E}_c^{k=1}(82) - \dot{E}_c^{k=1}(67)}{82-67} * (T_j - 67)$$

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 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,

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

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

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 18. 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 18—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 18. 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 18. 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 6 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 18. 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,

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 7) 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.

$PLF_j = 1 - C_D^c - [1 - X^{k=1}(T_j)]$, 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 18. 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 18 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 $\dot{Q}_c^{k=1}(T_j) < BL(T_j) < \dot{Q}_c^{k=v} \leq (T_j)$,

$$EER^{k=i}(T_j) = EER^{k=1}(T_j) + \frac{EER^{k=v}(T_j) - EER^{k=1}(T_j)}{Q^{k=v}(T_j) - Q^{k=1}(T_j)} * (BL(T_j) - Q^{k=1}(T_j))$$

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;

$EER^{k=2}(T_j)$ 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.

4.1.4.3 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. Refer to section 3.2.2.1 and Table 5 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 (HSPF) Calculations

Unless an approved alternative efficiency determination method is used, as set forth in 10 CFR 429.70(e). Calculate HSPF as follows: Six generalized climatic regions are depicted in Figure 1 and otherwise defined in Table 19. For each of these regions and for each applicable standardized design heating requirement, evaluate the heating seasonal performance factor using,

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

Where:

$e_h(T_j)/N$ = 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 19.

j = the bin number, dimensionless.

J = for each generalized climatic region, the total number of temperature bins, dimensionless. Referring to Table 19, 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 19—GENERALIZED CLIMATIC REGION INFORMATION

Region No.	I	II	III	IV	V	*VI
Heating Load Hours, HLH	493	857	1,280	1,701	2,202	1,842
Outdoor Design Temperature, T_{OD}	37	27	17	5	−10	30
Heating Load Line Equation Slope Factor, C	1.10	1.06	1.29	1.15	1.16	1.11
Variable Speed Slope Factor, C_{VS}	1.03	0.99	1.20	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	.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

$$\text{Equation 4.2-2} \qquad BL(T_j) = \frac{(T_{zl} - T_j)}{T_{zl} - T_{OD}} * C * \dot{Q}_c(95°F)$$

Where,

T_j = the outdoor bin temperature, °F

T_{zl} = the zero-load temperature, °F, which varies by climate region according to Table 19

T_{OD} = the outdoor design temperature, °F, which varies by climate region according to Table 19

C = the slope (adjustment) factor, which varies by climate region according to Table 19

$\dot{Q}_c(95\ °F)$ = 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, 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.

b. 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.

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.

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

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

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 19. 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

$$\text{Equation 4.2.1-3} \quad \delta(T_j) = \begin{Bmatrix} 0, if\ T_j \le T_{off}\ and\ \frac{\dot{Q}_h(T_j)}{3.413 * \dot{E}_h(T_j)} < 1 \\ 1/2, if\ T_{off} < T_j \le T_{on}\ and\ \frac{\dot{Q}_h(T_j)}{3.413 * \dot{E}_h(T_j)} \ge 1 \\ 1, if\ T_j > T_{on}\ and\ \frac{\dot{Q}_h(T_j)}{3.413 * \dot{E}_h(T_j)} \ge 1 \end{Bmatrix}$$

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.

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

$$\text{Equation 4.2.1-4} \quad \dot{Q}_h(T_j) = \begin{cases} \dot{Q}_h(17) + \frac{[\dot{Q}_h(47) - \dot{Q}_h(17)] * (T_j - 17)}{47 - 17}, & if\ T_j \ge 45\ °F\ or\ T_j \le 17\ °F \\ \dot{Q}_h(17) + \frac{[\dot{Q}_h(35) - \dot{Q}_h(17)] * (T_j - 17)}{35 - 17}, & if\ 17\ °F < T_j < 45\ °F \end{cases}$$

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.

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

$$\text{Equation 4.2.2-1} \quad \dot{Q}_h(T_j) = \dot{Q}_h^{k=1}(T_j) + \frac{\dot{Q}_h^{k=2}(T_j) - \dot{Q}_h^{k=1}(T_j)}{\mathrm{FP}_h^{k=2} - \mathrm{FP}_h^{k=1}} * \left[\mathrm{FP}_h(T_j) - \mathrm{FP}_h^{k=1}\right]$$

$$\text{Equation 4.2.2-2} \quad \dot{E}_h(T_j) = \dot{E}_h^{k=1}(T_j) + \frac{\dot{E}_h^{k=2}(T_j) - \dot{E}_h^{k=1}(T_j)}{\mathrm{FP}_h^{k=2} - \mathrm{FP}_h^{k=1}} * \left[\mathrm{FP}_h(T_j) - \mathrm{FP}_h^{k=1}\right]$$

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

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

Equation 4.2.2-4

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

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 11), $FP_h^{k=2}$ 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.

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

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

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

b. 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 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.

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) * $[1 - X^{k=2}(T_j)]$

If the $H1C_2$ test described in section 3.6.3 and Table 12 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:

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

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, $Q_h^{k=1}(47)$ 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

$$\dot{Q}_h^{k=1}(T_j) = \begin{cases} \dot{Q}_h^{k=1}(47) + \dfrac{\left[\dot{Q}_h^{k=1}(62) - \dot{Q}_h^{k=1}(47)\right] * (T_j - 47)}{62 - 47}, & if\ T_j \geq 47\ ^\circ\mathrm{F} \\ \dot{Q}_h^{k=v}(35) + \dfrac{\left[\dot{Q}_h^{k=1}(47) - \dot{Q}_h^{k=v}(35)\right] * (T_j - 35)}{47 - 35}, & if\ 35\ ^\circ\mathrm{F} \leq T_j < 47\ ^\circ\mathrm{F} \\ \dot{Q}_h^{k=v}(T_j), & if\ T_j < 35\ ^\circ\mathrm{F} \end{cases}$$

Equation 4.2.4-4

$$\dot{E}_h^{k=1}(T_j) = \begin{cases} \dot{E}_h^{k=1}(47) + \dfrac{\left[\dot{E}_h^{k=1}(62) - \dot{E}_h^{k=1}(47)\right] * (T_j - 47)}{62 - 47}, & if\ T_j \geq 47\ ^\circ\mathrm{F} \\ \dot{E}_h^{k=v}(35) + \dfrac{\left[\dot{E}_h^{k=1}(47) - \dot{E}_h^{k=v}(35)\right] * (T_j - 35)}{47 - 35}, & if\ 35\ ^\circ\mathrm{F} \leq T_j < 47\ ^\circ\mathrm{F} \\ \dot{E}_h^{k=v}(T_j), & if\ T_j < 35\ ^\circ\mathrm{F} \end{cases}$$

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 HSPF 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, 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

Equation 4.2.4–5 $\dot{Q}_h^{k=v}(T_j) = \dot{Q}_h^{k=v}(35) + M_Q * (T_j - 35)$

Equation 4.2.4–6 $\dot{E}_h^{k=v}(T_j) = \dot{E}_h^{k=v}(35) + M_E * (T_j - 35)$

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 $E_h^{k=1}(35)$, 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 $E_h^{k=2}(T_j)$ 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 19. 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:

$$\dot{m}_{da} = \bar{\dot{V}}_s * 0.075 \frac{lbm_{da}}{ft^3} * \frac{60_{min}}{hr} = \frac{\bar{\dot{V}}_{mx}}{v_n' * [1 + W_n]} * \frac{60_{min}}{hr} = \frac{\bar{\dot{V}}_{mx}}{v_n} * \frac{60_{min}}{hr}$$

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 19, calculate the nominal temperature of the air leaving the heat pump condenser coil using,

$$T_0(T_j) = 70°F + \frac{\dot{Q}_{hp}(T_j)}{\dot{m}_{da} * C_{p,da}}$$

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 19. Denote these capacities and electrical powers by using the subscript ''hp'' instead of ''h.'' Calculate the mass flow rate (expressed in pounds-mass of dry air per hour) and the specific heat of the indoor air (expressed in Btu/lbm$_{da}$ · °F) from the results of the H1$_2$ test using:

$$\dot{m}_{da} = \bar{\dot{V}}_s * 0.075\frac{lbm_{da}}{ft^3} * \frac{60_{min}}{hr} = \frac{\bar{\dot{V}}_{mx}}{v_n' * [1 + W_n]} * \frac{60_{min}}{hr} = \frac{\bar{\dot{V}}_{mx}}{v_n} * \frac{60_{min}}{hr}$$

$$C_{p,da} = 0.24 + 0.444 * W_n$$

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 19, 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,

$\dot{Q}_h(T_j) = \dot{Q}_h(T_j)$ $\dot{E}_h(T_j) = \dot{E}_{hp}(T_j)$ $\dot{E}_{CC}(T_j)$

Where,

$$\dot{Q}_{CC}(T_j) = \dot{m}_{da} * C_{p,da} * [T_{CC} - T_0(T_j)] \qquad \dot{E}_{CC}(T_j) = \frac{\dot{Q}_{CC}(T_j)}{3.413\frac{Btu/h}{W}}$$

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 19. 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 19, 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 19, 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 (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 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 $H2_2$ 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 Tj using

$$\dot{Q}_h^{k=3}(T_j) = \begin{cases} \dot{Q}_h^{k=3}(17) + \dfrac{[\dot{Q}_h^{k=3}(35) - \dot{Q}_h^{k=3}(17)] * (T_j - 17)}{35 - 17}, & if\ 17\ °F < T_j \leq 45\ °F \\ \dot{Q}_h^{k=3}(2) + \dfrac{[\dot{Q}_h^{k=3}(17) - \dot{Q}_h^{k=3}(2)] * (T_j - 2)}{17 - 2}, & if\ T_j \leq 17\ °F \end{cases}$$

$$\dot{E}_h^{k=3}(T_j) = \begin{cases} \dot{E}_h^{k=3}(17) + \dfrac{[\dot{E}_h^{k=3}(35) - \dot{E}_h^{k=3}(17)] * (T_j - 17)}{35 - 17}, & if\ 17\ °F < T_j \leq 45\ °F \\ \dot{E}_h^{k=3}(2) + \dfrac{[\dot{E}_h^{k=3}(17) - \dot{E}_h^{k=3}(2)] * (T_j - 2)}{17 - 2}, & if\ T_j \leq 17\ °F \end{cases}$$

Determine $\dot{Q}_h^{k=3}(17)$ and $\dot{E}_h^{k=3}(17)$ from the $H3_3$ test and determine $\dot{Q}_h^{k=2}(2)$ and $\dot{E}_h^{k=3}(2)$ 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:

$X^{k=3}(T_j) = BL(T_j)/\dot{Q}_h^{k=3}(T_j)$ and $PLF_j = 1 - C^h_D(k=3) * [1 - X^{k=3}(T_j)]$

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, $\delta'(Tj)$, 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.

$$\frac{e_h(T_j)}{N} = \dot{E}_h^{k=3}(T_j) * \delta'(T_j) * \frac{n_j}{N} \quad \text{and} \quad \frac{RH(T_j)}{N} = \frac{BL(T_j) - [\dot{Q}_h^{k=3}(T_j) * \delta\prime(T_j)]}{3.413 \frac{Btu/h}{W}} * \frac{n_j}{N}$$

where $\delta''(Tj)$ 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 11 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 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.

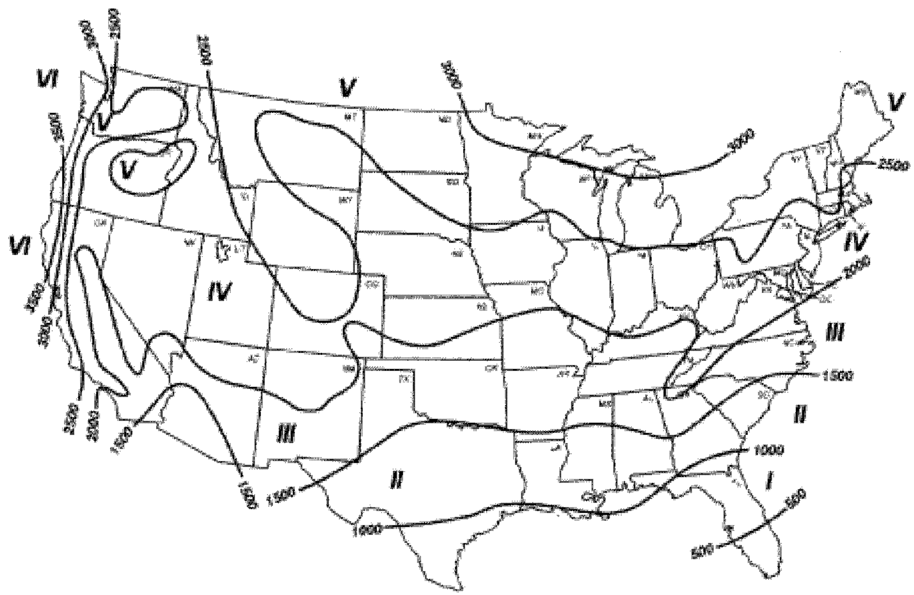

Figure 1—Heating Load Hours (HLH_A) for the United States

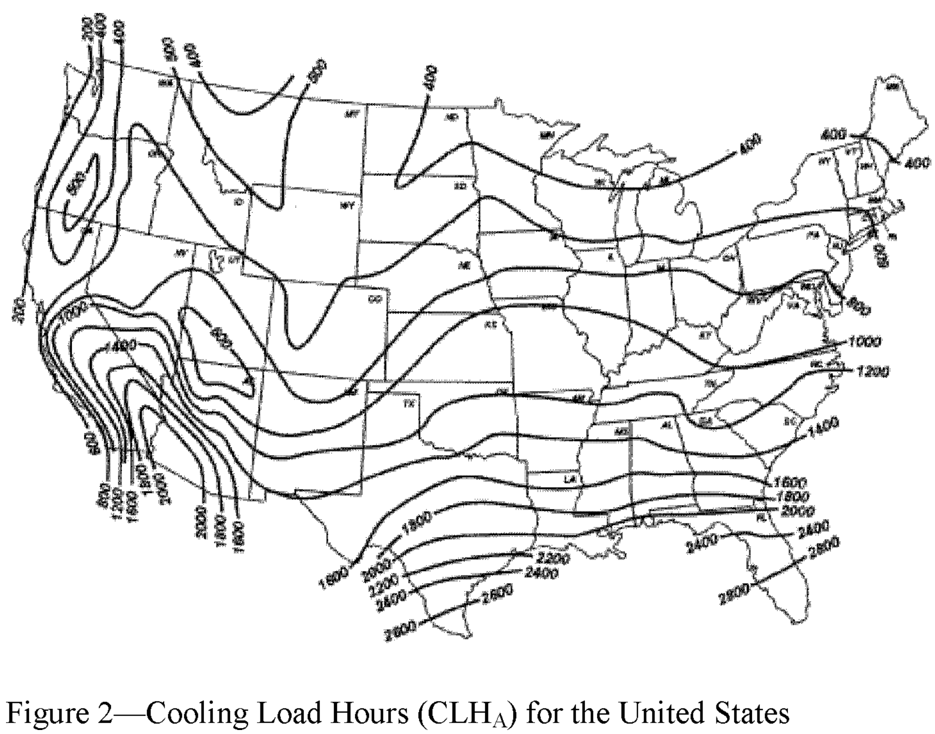

Figure 2—Cooling Load Hours (CLH_A) for the United States

TABLE 20—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 21

TABLE 21—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.

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