DEPARTMENT OF ENERGY

10 CFR Parts 429 and 430

[Docket No. EERE–2009–BT–TP–0004]

RIN 1904–AB94

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 established under the Energy Policy and Conservation Act. DOE proposed amendments to the test procedure in a June 2010 notice of proposed rulemaking (NOPR), an April 2011 supplemental notice of proposed rulemaking (SNOPR), and an October 2011 SNOPR. DOE provided additional time for stakeholder comment in a December 2011 extension of the comment period for the October 2011 SNOPR. DOE received further public comment for revising the test procedure in a November 2014 Request for Information for energy conservation standards for central air conditioners and heat pumps. DOE proposes in this SNOPR: A new basic model definition as it pertains to central air conditioners and heat pumps and revised rating requirements; revised alternative efficiency determination methods; termination of active waivers and interim waivers; revised procedures to determine off mode power consumption; changes to the test procedure that would improve test repeatability and reduce test burden; clarifications to ambiguous sections of the test procedure intended also to improve test repeatability; inclusion of, amendments to, and withdrawals of test procedure revisions proposed in published test procedure notices in the rulemaking effort leading to this supplemental notice of proposed rulemaking; and changes to the test procedure that would improve field representativeness. Some of these proposals also include incorporation by reference of updated industry standards. 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 December 9, 2015. See section V, ''Public Participation,'' for details.

ADDRESSES: Any comments submitted must identify the SNOPR for test procedures for central air conditioners and heat pumps, and provide docket number EE–2009–BT–TP–0004 and/or regulatory information number (RIN) number 1904–AB94. 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: RCAC-HP-2009-TP-0004@ee.doe.gov*. Include the docket number EE–2009–BT–TP–0004 and/or 1904–AB94 RIN in the subject line of the message.
3. *Mail:* Ms. Brenda Edwards, U.S. Department of Energy, Building Technologies Office, Mailstop EE–2J, 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:* Ms. Brenda Edwards, U.S. Department of Energy, Building Technologies Office, 950 L'Enfant Plaza SW., Suite 600, Washington, DC 20024.

Telephone: (202) 586–2945. 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, public meeting attendee lists and transcripts, comments, and other supporting documents/materials, is available for review at *www.regulations.gov*. All documents in the docket are listed in the regulations.gov index. However, some documents listed in the index, such as those containing information that is exempt from public disclosure, may not be publicly available.

A link to the docket Web page can be found at: *www1.eere.energy.gov/buildings/appliance_standards/rulemaking.aspx/ruleid/72*. This Web page will contain a link to the docket for this notice on the *www.regulations.gov* site. The *www.regulations.gov* 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 regulations.gov.

FOR FURTHER INFORMATION CONTACT:

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Johanna Hariharan, 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.Hariharan@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 Ms. Brenda Edwards at (202) 586–2945 or by email: *Brenda.Edwards@ee.doe.gov*.

SUPPLEMENTARY INFORMATION: DOE intends to incorporate by reference the following industry standards into Part 430:

(1) ANSI/AHRI 210/240–2008 with Addenda 1 and 2: Performance Rating of Unitary Air-Conditioning & Air-Source Heat Pump Equipment, 2012;

(2) AHRI 210/240-Draft: Performance Rating of Unitary Air-Conditioning & Air-Source Heat Pump Equipment;

(3) ANSI/AHRI 1230–2010 with Addendum 2: Performance Rating of Variable Refrigerant Flow (VRF) Multi-Split Air-Conditioning and Heat Pump Equipment, 2010;

(4) ASHRAE 23.1–2010: Methods of Testing for Rating the Performance of Positive Displacement Refrigerant Compressors and Condensing Units that Operate at Subcritical Temperatures of the Refrigerant;

(5) ASHRAE Standard 37–2009, Methods of Testing for Rating Electrically Driven Unitary Air-Conditioning and Heat Pump Equipment;

(6) ASHRAE 41.1–2013: Standard Method for Temperature Measurement; ASHRAE 41.6–2014: Standard Method for Humidity Measurement;

(7) ASHRAE 41.9–2011: Standard Methods for Volatile-Refrigerant Mass Flow Measurements Using Calorimeters;

(8) ASHRAE/AMCA 51–07/210–07, Laboratory Methods of Testing Fans for Certified Aerodynamic Performance Rating.

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

Draft is available on the rulemaking Web page (Docket EERE–2009–BT–TP–0004–0045).

Copies of ASHRAE 23.1–2010, ASHRAE Standard 37–2009, ASHRAE 41.1–2013, and ASHRAE 41.9–2011 can be purchased from ASHRAE's Web site at *https://www.ashrae.org/resources-publications*.

Copies of ASHRAE/AMCA 51–07/210–07 can be purchases from AMCA's Web site at *http://www.amca.org/store/index.php*.

Table of Contents

I. Authority and Background

A. Authority

Title III, Part B of the Energy Policy and Conservation Act of 1975 (EPCA or the Act), Pub. L. 94–163 (42 U.S.C. 6291 – 6309, as codified), established the Energy Conservation Program for Consumer Products Other Than Automobiles, a program covering most major household appliances, including the single phase central air conditioners and heat pumps [1] with rated cooling capacities less than 65,000 British thermal units per hour (Btu/h) that are the focus of this notice.[2] (42 U.S.C. 6291(1)–(2), (21) and 6292(a)(3))

Under EPCA, the program consists of four activities: (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 for certifying to DOE that their products comply with applicable energy conservation standards adopted pursuant to EPCA and for representing the efficiency of those products. (42 U.S.C. 6293(c); 42 U.S.C. 6295(s)) Similarly, DOE must use these test procedures in any enforcement action to determine whether covered products comply with these energy conservation standards. (42 U.S.C. 6295(s)) Under 42 U.S.C. 6293, EPCA sets forth criteria and procedures for DOE's adoption and amendment of such test procedures. Specifically, EPCA provides that an amended test procedure shall produce results which measure the energy

[1] Where this notice uses the terms ''HVAC'' or ''CAC/CHP'', they are in reference specifically to central air conditioners and heat pumps as covered by EPCA.

[2] For editorial reasons, upon codification in the U.S. Code, Part B was re-designated Part A.

efficiency, energy use, or estimated annual operating cost of a covered product over an average or representative period of use, and shall not be unduly burdensome to conduct. (42 U.S.C. 6293(b)(3)) 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)) Furthermore, DOE must review test procedures at least once every 7 years. (42 U.S.C 6293(b)(1)(A)) DOE last published a test procedure final rule for central air conditioner and heat pumps on October 22, 2007. 72 FR 59906. Finally, in any rulemaking to amend a test procedure, DOE must determine whether and the extent to which the proposed test procedure would change the measured efficiency of a system that was tested under the existing test procedure. (42 U.S.C. 6293(e)(1)) If DOE determines that the amended test procedure would alter the measured efficiency of a covered product, DOE must amend the applicable energy conservation standard accordingly. (42 U.S.C. 6293(e)(2))

DOE's existing test procedures for central air conditioners and heat pumps 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 of these products. Some amendments proposed in this SNOPR will not alter the measured efficiency of central air conditioners and heat pumps, and thus are being proposed as revisions to the current Appendix M. Other amendments proposed in this SNOPR will alter the measured efficiency, as represented in the regulating metrics of energy efficiency ratio (EER), seasonal energy efficiency ratio (SEER), and heating seasonal performance factor (HSPF). These amendments are proposed as part of a new Appendix M1. The test procedure changes proposed in this notice as part of a new Appendix M1, if adopted, would not become mandatory until the existing energy conservation standards are revised. (42 U.S.C. 6293(e)(2)) In revising the energy conservation standards, 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. DOE would then use the cross-walked equivalent of the existing standard as the baseline for its standards analysis to prevent back-sliding as required under 42 U.S.C. 6295(o)(1).

On December 19, 2007, the President signed the Energy Independence and Security Act of 2007 (EISA 2007), Pub. L. 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 central air conditioners and heat pumps, standby mode is incorporated into the SEER metric, while off mode power consumption is separately regulated. This SNOPR includes proposals relevant to the determination of both SEER (including standby mode) and off mode power consumption.

10 CFR 430.27 allows manufacturers to submit an application for an interim waiver and/r a petition for a waiver granting relief from adhering to the test procedure requirements found under 10 CFR part 430, subpart B, Appendix M. For those waivers that are active, however, 10 CFR 430.27(l) requires DOE to amend its regulations so as to eliminate any need for the continuation of such waivers. To this end, this notice proposes relevant amendments to its test procedure concerning such waivers.

B. Background

This SNOPR addresses proposals and comments from three separate rulemakings, two guidance documents, and a working group: (1) Proposals for off mode test procedures made in earlier notices as part of this rulemaking (Docket No. EERE–2009–BT–TP–0004); (2) proposals regarding alternative efficiency determination methods (Docket No. EERE–2011–BT–TP–0024); (3) stakeholder comments from a request for information regarding energy conservation standards (Docket No. EERE–2014–BT–STD–0048); (4) a draft guidance document related to testing and rating split systems with blower coil units (Docket No. EERE–2014–BT–GUID–0033); (5) a draft guidance document that deals with selecting units for testing, rating, and certifying split-system combinations, including discussion of basic models and of condensing units and evaporator coils sold separately for replacement installation (Docket No. EERE–2014–BT–GUID–0032); and (6) the recommendations of the regional standards enforcement Working Group (Docket No. EERE–2011–BT–CE–0077).

DOE's initial proposals for estimating off mode power consumption in the test procedure for central air conditioners and heat pumps were shared with the public in a notice of proposed rulemaking published in the **Federal Register** on June 2, 2010 (June 2010 NOPR; 75 FR 31224) and at a public meeting at DOE headquarters in Washington, DC on June 11, 2010. Subsequently, DOE published a supplemental notice of proposed rulemaking (SNOPR) on April 1, 2011, in response to comments received on the June 2010 NOPR and due to the results of additional laboratory testing conducted by DOE. (April 2011 SNOPR) 76 FR 18105, 18127. DOE received additional comments in response to the April 2011 SNOPR and proposed an amended version of the off mode procedure that addressed those comments in a second SNOPR on October 24, 2011 (October 2011 SNOPR). 76 FR 65616. DOE received additional comments during the comment period of the October 24, 2011 SNOPR and the subsequent extended comment period. 76 FR 79135.

Between the April 2011 and October 2011 SNOPRs, DOE published a direct final rule (DFR) in the **Federal Register** on June 27, 2011 that set forth amended energy conservation standards for central air conditioners and central air conditioning heat pumps, including a new standard for off mode electrical power consumption. (June 2011 DFR) 76 FR 37408. Units manufactured on or after January 1, 2015, are subject to that standard for off mode electrical power consumption. 10 CFR 430.32(c)(6). However, on July 8, 2014, DOE published an enforcement policy statement regarding off mode standards for central air conditioners and central air conditioning heat pumps [3] (July 2014 Enforcement Policy Statement) specifying that DOE will not assert civil penalty authority for violation of the off mode standard until 180 days following publication of a final rule establishing a test method for measuring off mode electrical power consumption.

DOE also pursued, in a request for information (RFI) published on April 18, 2011 (AEDM RFI) (76 FR 21673), and a NOPR published on May 31, 2012 (AEDM NOPR) (77 FR 32038), revisions to its existing alternative efficiency determination methods (AEDM) and alternative rating methods (ARM) requirements to improve the approach by which manufacturers may use

[3] Available at: *http://energy.gov/sites/prod/files/2014/07/f17/Enforcement%20Policy%20Statement%20-%20cac%20off%20mode.pdf* (Last accessed March 30, 2015.)

modeling techniques as the basis to certify consumer products and commercial and industrial equipment covered under EPCA. DOE also published a final rule regarding AEDM requirements for commercial and industrial equipment only (Commercial Equipment AEDM FR). 78 FR 79579. This SNOPR addresses the proposals made and comments received in the AEDM NOPR applicable to central air conditioners and heat pumps and makes additional proposals.

On June 13, 2014, DOE published a notice of intent to form a working group to negotiate enforcement of regional standards for central air conditioners and requested nominations from parties interested in serving as members of the Working Group. 79 FR 33870. On July 16, 2014, the Department published a notice of membership announcing the eighteen nominations that were selected to serve as members of the Working Group, in addition to two members from Appliance Standards and Rulemaking Federal Advisory Committee (ASRAC), and one DOE representative. 79 FR 41456. The Working Group identified a number of issues related to testing and certification that are being addressed in this rule. In addition, all nongovernmental participants of the Working Group approved the final report contingent on upon the issuance of the final guidance on Docket No. EERE–2014–BT–GUID–0032 0032 and Docket No. EERE–2014–BT–GUID–0033 consistent with the understanding of the Working Group as set forth in its recommendations. (Docket No. EERE–2011–BT–CE–0077–0070, Attachment) This SNOPR responds to comments on the August 19 and 20, 2014, guidance documents related to testing and rating split systems, which are discussed in more detail in section III.A. The proposed changes supplant these two draft guidance documents; DOE will not finalize the draft guidance documents and instead will provide any necessary clarity through this notice and the final rule. DOE believes the proposed changes are consistent with the intent of the Working Group.

On November 5, 2014, DOE published a request for information for energy conservation standards (ECS) for central air conditioners and heat pumps (November 2014 ECS RFI). 79 FR 65603. In response, several stakeholders provided comments suggesting that DOE amend the current test procedure. This SNOPR responds to those test procedure-related comments.

II. Summary of the Supplementary Notice of Proposed Rulemaking

This supplementary notice of proposed rulemaking (SNOPR) proposes revising the certification requirements and test procedure for central air conditioners and heat pumps based on various published material as discussed in section I.B.

DOE proposes to revise the basic model definition, add additional definitions for clarity, make certain revisions to the testing requirements for determination of certified ratings, add certain certification reporting requirements, revise requirements for determination of represented values, and add product-specific enforcement provisions. Some of the proposed revisions to the certification requirements would impact the energy conservation standard and thus would not be effective until the compliance date of any amended energy conservation standards.

DOE proposes to update requirements for Alternative Rating Methods (ARMs) used to determine performance metrics for central air conditioners and heat pumps based on the regulations for Alternative Efficiency Determination Methods (AEDMs) that are used to estimate performance for commercial HVAC equipment. Specifically, for central air conditioners and heat pumps, DOE proposes: (1) Revisions to nomenclature regarding ARMs; (2) rescinding DOE pre-approval of an ARM prior to use; (3) AEDM validation requirements; (4) a verification testing process; (5) actions a manufacturer could take following a verification test failure; and (6) consequences for invalid ratings. These proposed changes do not impact the energy conservation standard.

DOE proposes to revise the test procedure such that tests of multi-circuit products, triple-capacity northern heat pump products, and multi-blower products can be performed without the need of an interim waiver or a waiver. Existing interim waivers and waivers, as applicable, regarding these products would terminate on the effective date of a final rule promulgating the proposals in this SNOPR. DOE also reaffirms that the waivers associated with multi-split products have already terminated and that these products can also be tested using the current and proposed test procedure. These proposed changes do not impact the energy conservation standard and thus are proposed as part of revisions to Appendix M.

DOE also proposes to clarify that air-to-water heat pump products integrated with domestic water heating are not subject to central air conditioner and heat pump energy conservation standards. Accordingly, the waiver regarding these products would terminate effective 180 days after publication of a final rule that incorporates the proposals in this SNOPR.

DOE proposes revisions to the test methods and calculations for off mode power consumption that were proposed or modified in the June 2010 NOPR, April 2011 SNOPR, and October 2011 SNOPR. These revisions address comments received in response to the October 2011 SNOPR suggesting that test methods and calculations more accurately represent off-mode power consumption in field applications. These proposed changes do not impact the energy conservation standard. Specifically, DOE proposes the following:

(1) Establishment of separate testing and calculations that would depend on whether the tested unit is equipped with a crankcase heater and whether the crankcase heater is controlled during the test;

(2) Alteration of the testing temperatures such that the crankcase heater is tested in outdoor air conditions that are representative of the shoulder and heating seasons;

(3) Changing of the testing methodology for determining the power consumption of the low-voltage components (P_X);

(4) Changing of the calculation of the off mode power rating ($P_{W,OFF}$) such that the off mode power for the shoulder and heating seasons are equally weighted;

(5) Implementation of a time delay credit for energy consumption, including credits in the form of scaling factors and multipliers for energy-efficient products that require larger crankcase heaters to maintain product reliability;

(6) Addition of an alternative energy determination method for determining off mode power for coil-only split-systems; and

(7) Inclusion of a means for calculating a basic model's annual off mode energy use, from which manufacturers could make representations about their products' off mode energy use.

DOE also proposes changes to improve the repeatability and reduce the test burden of the test procedure. These proposed changes do not impact the energy conservation standard. Specifically, DOE proposes the following:

(1) Clarification of fan speed settings;

(2) Clarification of insulation requirements for refrigerant lines and addition of a requirement for insulating mass flow meters;

(3) Addition of a requirement to demonstrate inlet air temperature uniformity for the outdoor unit using thermocouples;

(4) Addition of a requirement that outdoor air conditions be measured using sensors measuring the air captured by the air sampling device(s) rather than the temperature sensors located in the air stream approaching the inlets;

(5) Addition of a requirement that the air sampling device and the tubing that transfers the collected air to the dry bulb temperature sensor be at least two inches from the test chamber floor, and a requirement that humidity measurements be based on dry bulb temperature measurements made at the same location as the corresponding wet bulb temperature measurements used to determine humidity;

(6) Clarification of maximum speed for variable-speed compressors;

(7) Addition of requirements that improve consistency of refrigerant charging procedures;

(8) Allowance of an alternative arrangement for cyclic tests to replace the currently-required damper in the inlet portion of the indoor air ductwork for single-package ducted units;

(9) Clarification of the proper supply voltage for testing;

(10) Revision of the determination of the coefficient of cyclic degradation (C_D);

(11) Option for a break-in period of up to 20 hours;

(12) Update of references to industry standards where appropriate;

(13) Withdrawal of all references to ASHRAE Standard 116–1995;

(14) Inclusion of information from the draft AHRI 210/240; and

(15) Provisions regarding damping of pressure transducer signals to avoid exceeding test operating tolerances due to high frequency fluctuations.

Lastly, DOE proposes clarifications of any sections of the test procedure that may be ambiguous. Specifically, DOE proposes to add reference to an industry standard for testing variable refrigerant flow multi-split systems; replace the informative guidance table for using the test procedure; and clarify definitions of multi-split systems and mini-split systems, which DOE now proposes to call single-zone-multiple-unit systems. These proposed changes do not impact the energy conservation standard.

DOE notes that all the above-listed proposed changes to the test procedure would not impact the energy conservation standard and as such are proposed as part of a revised Appendix M. Given the extensive changes proposed for Appendix M, DOE has provided a full re-print of Appendix M in the regulatory text of this SNOPR that includes the changes proposed in this SNOPR as well as those proposed in the June 2010 NOPR and the April 2011 and October 2011 SNOPRs that have not been withdrawn.

DOE also proposes various changes to the test procedure that would affect the energy conservation standard and proposes incorporating these changes in a new appendix, Appendix M1 to Subpart B of 10 CFR part 430, which includes the text of Appendix M to Subpart B of 10 CFR part 430 with amendments as proposed in this SNOPR. Specifically, DOE proposes the following:

(1) Increase the minimum external static pressure requirements for conventional central air conditioners and heat pumps to better represent the external static pressure conditions in field installations; [4]

(2) Add a minimum external static pressure adjustment to correct for potentially unrepresentative external static pressure conditions for blower coil systems tested with condensing furnaces;

(3) Raise the default fan power for coil-only systems;

(4) Adjust the heating load line equation such that the zero load point occurs at 55 °F for Region IV, the adjustment factor is 1.3, and the heating load is tied with the heat pump's cooling capacity; and

(5) Revise the heating mode test procedure to allow more options for products equipped with variable-speed compressors.

DOE proposes to make the test procedure revisions in this SNOPR as reflected in the revised Appendix M to Subpart B of 10 CFR part 430 effective on a date 180 days after publication of the test procedure final rule in the **Federal Register** and mandatory for testing to determine compliance with the existing energy conservation standards for central air conditioners and heat pumps as of that date. DOE proposes to make the test procedure revisions in this SNOPR as reflected in the proposed new Appendix M1 to Subpart B of 10 CFR part 430 effective on the compliance date of the revised energy conservation standards for central air conditioners and heat pumps and mandatory for testing to determine compliance with said revised standards as of that date. DOE will address any comments received in response to this SNOPR in the test procedure final rule.

As noted in section I.A, 42 U.S.C. 6293(e) requires that DOE shall 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 these proposed amendments would result in a change in measured energy efficiency and measured energy use for central air conditioners and heat pumps. Therefore, DOE is conducting a separate rulemaking to amend the energy conservation standards for central air conditioners and heat pumps with respect to the revised test procedure, once its proposals become final. (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. Definitions, Testing, Rating, and Compliance of Basic Models of Central Air Conditioners and Heat Pumps

On August 19 and 20, 2014, DOE issued two draft guidance documents regarding the test procedure for central air conditioners and heat pumps. One guidance document dealt with testing and rating split systems with blower coil indoor units (Docket No. EERE–2014–BT–GUID–0033); and the other dealt more generally with selecting units for testing, rating, and certifying split-system combinations, including discussion of basic models and of condensing units and evaporator coils sold separately for replacement installation (Docket No. EERE–2014–BT–GUID–0032). The comments in response to these draft guidance documents are discussed in this section of the notice. DOE has proposed changes to the substance of the draft guidance that reflects the comments received as well as the recommendations of the regional standards enforcement Working Group (Docket No. EERE–2011–BT–CE–0077–0070, Attachment). The proposed changes supplant the two draft guidance documents; DOE will not finalize the draft guidance documents and instead will provide any necessary clarity through this notice and the final rule.

1. Basic Model Definition

In the August 20, 2014 draft guidance document (Docket No. EERE–2014–BT–GUID–0032), DOE clarified that a basic

[4] Conventional central air conditioners and heat pumps are those products that are not short duct systems (see section III.F.2) or small-duct, high-velocity systems.

model means all units of a given type (or class thereof) having the same primary energy source, and which have essentially identical electrical, physical, and functional characteristics that affect energy efficiency. 10 CFR 430.2. DOE noted that for split-system units, this includes a condensing (outdoor) unit and a coil-only or blower coil indoor unit.[5]

In the guidance document, DOE also stated that if a company intended to claim ratings for each combination of outdoor unit and indoor unit, it must certify all possible model combinations as separate basic models. Only the basic model combinations that include a highest sales volume combination (HSVC) indoor unit for a given outdoor unit must be tested, while the other basic models may be rated with an ARM. Alternatively, the manufacturer could make all combinations of a given model of outdoor unit part of the same basic model and not rate all individual combinations. However, all combinations within the basic model would have to have the same represented efficiency, based on the least efficient combination. This association would be included in the certification report.

In response to the draft guidance document, AHRI and Johnson Controls (JCI) stated that there was a difference between DOE's definition of Basic Model and the industry's use of Basic Model Groups (Docket No. EERE–2014–BT–GUID–0032, AHRI, No. 8 at p. 1; JCI, No. 5 at p. 3) Johnson Controls specified that most manufacturers consider a specific outdoor model with all combinations of indoor units to be a basic model and notes that DOE's definition appeared to allow outdoor units to be combined into a basic model if they share the same ratings. (*Id.*)

DOE reviewed AHRI's Operations Manual for Unitary Small Air-Conditioners and Air-Source Heat Pumps (Includes Mixed-Match Coils) (Rated Below 65,000 Btu/h) Certification Program (AHRI OM 210/240—January 2014).[6] This document specifies the following definitions:

A Split System BMG [Basic Model Group [7]] consists of products with the same Outdoor Unit used with several Indoor Unit combinations (*i.e.* horizontal, vertical, A-coil, etc.). Same Outdoor Unit refers to models with the same or comparable compressor, used with the same outdoor coil surface area and the same outdoor air quantity.

An ICM [Independent Coil Manufacturer] BMG consists of coils (Indoor Units) with matching capacity ranges of 6,000 Btu/h and the following identical geometry parameters: Air-handler, evaporator fan type, evaporator number of rows, type of equipment (air-cooled, water-cooled or evaporatively-cooled), evaporator tube centers, evaporator fin types, evaporator fins/inch, evaporator tube OD, evaporator expansion device, fin length per slab, fin height per slab, number of slabs in the coil, fin material type, tube material type, and total number of active tubes (refer to Table H1).

In order to create consistency within the industry, DOE proposes to modify its basic model definition for central air conditioners and heat pumps. Specifically, DOE proposes that manufacturers would have a choice in how to assign individual models (for single-package units) or combinations (for split systems) to basic models. Specifically, manufacturers may consider each individual model/combination its own basic model, or manufacturers may assign all individual models of the same single-package system or all individual combinations using the same model of outdoor unit (for outdoor unit manufacturers (OUM)) or model of indoor unit (for independent coil manufacturers (ICM)) to the same basic model.

DOE believes that this proposal is consistent with the existing general definition of basic model which refers to all units having the same primary energy source and having essentially identical electrical, physical, and functional characteristics that affect energy consumption or energy efficiency. However, DOE proposes to further define the physical characteristics necessary to assign individual models or combinations to the same basic model:

(i) For split-systems manufactured by independent coil manufacturers (ICMs) and for small-duct, high velocity systems: All individual combinations having the same model of indoor unit, which means the same or comparably performing indoor coil(s) [same face area; fin material, depth, style (*e.g.* wavy, louvered), and density (fins per inch); tube pattern, material, diameter, wall thickness, and internal enhancement], indoor fan(s) [same air flow with the same indoor coil and external static pressure, same power input], auxiliary refrigeration system components if present (*e.g.* expansion valve), and controls.

(ii) for split-systems manufactured by outdoor unit manufacturers (OUMs): All individual combinations having the same model of outdoor unit, which means the same or comparably performing compressor(s) [same displacement rate (volume per time) and same capacity and power input when tested under the same operating conditions], outdoor coil(s) [same face area; fin material, depth, style (*e.g.* wavy, louvered), and density (fins per inch); tube pattern, material, diameter, wall thickness, and internal enhancement], outdoor fan(s) [same air flow with the same outdoor coil, same power input], auxiliary refrigeration system components if present (*e.g.* suction accumulator, reversing valve, expansion valve), and controls.

The proposed requirements for single-package models combine the requirements listed describing the characteristics of the same models of indoor units and same models of outdoor units. DOE requests comment on its proposal to modify the definition of ''basic model'', as well as the proposed physical characteristics required for assigning individual models or combinations to the same basic model, as described above.

If manufacturers assign each individual model or combination to its own basic model, DOE proposes that each individual model/combination must be tested and that an AEDM cannot be applied. This option would limit a manufacturer's risk in terms of noncompliance but would represent increased testing burden compared to the other option.

If manufacturers assign all individual combinations of a model of outdoor unit (for OUMs) or model of indoor unit (for ICMs) to a single basic model, DOE further proposes that, in contrast to the draft guidance document and DOE's current regulations, each individual combination within a basic model (*i.e.,* having the same model of outdoor unit for OUMs, or having the same model of indoor unit for ICMs) must be certified with a rating determined for that individual combination. In other words, individual combinations within the same basic model that have different SEER ratings, for example, would be certified with their individual ratings, rather than with the lowest SEER of the basic model. However, only one individual combination in each basic

[5] DOE notes that a blower coil indoor unit may consist of separate units, one that includes the indoor coil and another that is an air mover, either a modular blower or a furnace. Alternatively, a blower coil indoor unit may be a single unit that includes both the indoor coil and the indoor fan. Hence, in further discussion, ''blower coil indoor unit'' may be any one of these three options.

[6] Available at: *www.ahrinet.org/App_Content/ahri/files/Certification/OM%20pdfs/USE_OM.pdf* (Last accessed March 20, 2015.)

[7] According to the AHRI General Operations Manual, a basic model is a product possessing a discrete performance rating, whereas a basic model group is a set of models that share characteristics that allow the performance of one model to be representative of the group, although the group does not have to share discrete performance. (General OM—October 2013). Available at: *www.ahrinet.org/App_Content/ahri/files/Certification/OM%20pdfs/General_OM.pdf.* (Last accessed March 24, 2015.)

model would have to be tested (see section III.A.3.a), while the others may be rated using an AEDM. This option reduces testing burden but increases risk. Specifically, if any one of the combinations within a basic model fails to meet the applicable standard, then all of the combinations within the basic model fail, and the entire basic model must be taken off the market *(i.e.,* the model of outdoor unit for OUMs and the model of indoor unit for ICMs). All combinations offered for sale (*e.g.,* for OUMs, based on a given model of outdoor unit which is the basis of the basic model) must be certified, and all of these combinations within the basic model must meet applicable standards. DOE notes that under this proposed rule, ICMs and OUMs will continue to have an independent obligation to test, provide certified ratings, and ensure compliance with applicable standards.

By way of example, a manufacturer has two models of outdoor units, models A and B. Each of models A and B can be paired with any of three models of indoor units—models 1, 2, and 3. Per the guidance document, the manufacturer could either: (1) Make each combination a separate basic model (*i.e.,* A–1, A–2, A–3, B–1, B–2, and B–3), test the HSVC for each model of outdoor unit (A and B), and rate the other basic models with an ARM; (2) make each combination a separate basic model and test each of them; or (3) make combinations A–2 and A–3 part of basic model A–1 (and similarly B–2 and B–3 part of B–1) and represent the efficiency of all three with the same certified rating at the least efficient combination in the basic model. In this proposal, the manufacturer could either: (1) Make each combination a separate basic model and test and rate each combination; or (2) make combinations A–2 and A–3 part of basic model A–1 (and similarly B–2 and B–3 part of B–1), test the HSVC combination for the model of outdoor unit, and test or use an AEDM to rate the efficiency of all other combinations in the basic model.

DOE notes that unlike in the current ''basic model'' definition that contains less detail on what constitutes essentially identical characteristics, under DOE's new proposal, manufacturers would not be able to assign different models of outdoor units (for OUMs) or models of indoor units (for ICMs) to a single basic model Based on a review of certification data, it appears that most manufacturers are not currently doing this, so DOE expects this proposal to have limited impact on current practices.

Additional rating and certification requirements for single-package models and multi-split, multi-circuit, and single-zone-multiple-coil models are described in section III.A.3.c.

Revisions to the test procedure as proposed in section III.D of this SNOPR enable the determination of off mode power consumption, which reflects the operation of the contributing components: Crankcase heater and low-voltage controls. Varying designs of these components produce different off mode power consumption. DOE proposes that if individual combinations that are otherwise identical are offered with multiple options for off mode related components, manufacturers at a minimum must rate the individual combination with the crankcase heater and controls which are the most consumptive (*i.e.,* would result in the largest value of $P_{W,OFF}$). If a manufacturer wishes to also make representations for less consumptive off mode options for the same individual combination, the manufacturer may provide separate ratings, but the manufacturer must differentiate the individual model numbers for these ratings. These individual combinations would be within the same basic model. DOE discusses this in relation to single-package units in section III.A.3.e.

DOE also proposes to clarify that a central air conditioner or central air conditioning heat pump may consist of: A single-package unit; an outdoor unit and one or more indoor units (*e.g.,* a single-split or multi-split system); an indoor unit only (rated as a combination by an ICM with an OUM's outdoor unit); or an outdoor unit only (with no match, rated by an OUM with the coil specified in this test procedure). DOE has proposed adding these specifications to the definition of central air conditioner or central air conditioning heat pump in 10 CFR 430.2. In the certification reports submitted by OUMs for split systems, DOE proposes that manufacturers must report the basic model number as well as the individual model numbers of the indoor unit(s) and the air mover where applicable.

2. Additional Definitions

In order to specify differences in the proposed basic model definition for ICMs and OUMs, DOE also proposes the following definitions:

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

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

With respect to any given basic model, a manufacturer could be an ICM or an OUM. DOE notes that the use of the term ''manufacturer'' in these definitions refers to any person who manufactures, produces, assembles, or imports a consumer product. *See* 42 U.S.C. 6291(10, 12).

DOE also proposes to define variable refrigerant flow (VRF) systems as a kind of multi-split system. DOE notes that not all VRF systems are commercial equipment. Therefore, the proposed definition also clarifies that VRF systems that are single-phase and less than 65,000 btu/h are a kind of central air conditioners and central air conditioning heat pumps.

DOE also proposes to modify the definition of indoor unit. DOE noted in market research that ICMs may not always provide cooling mode expansion devices with indoor units. Therefore to provide clarity in the testing and rating requirements, DOE proposes to change the definition of ''indoor unit'' to clarify that it may not include the cooling mode expansion device. Also, for reasons discussed in section III.A.3.f, DOE proposes to include the casing in the definition so that uncased coils will not be considered indoor units:

Indoor unit transfers heat between the refrigerant and the indoor air, and consists of an indoor coil and casing and may include a cooling mode expansion device and/or an air moving device.

DOE proposes to specify in Appendix M that if the indoor unit does not ship with a cooling mode expansion device, the system should be tested using the device as specified in the installation instructions provided with the indoor unit, or if no device is specified, using a TXV. DOE notes that the AHRI program does not appear to assume that the expansion device is necessarily provided with the coil, *i.e.,* AHRI's operations manual specifies that for testing for the AHRI certification program, the ICM must provide an indoor coil and expansion device.

Finally, DOE is proposing to clarify several other definitions currently in 10 CFR 430.2 with minor wording changes and move them to 10 CFR 430, Subpart B, Appendix M. The proposed definition of central air conditioner or central air conditioning heat pump in 10 CFR 430.2 refers the reader to the additional central air conditioner-related definitions in Appendix M. Locating all of the relevant definitions in the appendix will make it easier to find and reference them. DOE also proposes to remove entirely the definitions for ''condenser-evaporator coil combination'' and ''coil family'' as those terms no longer appear in the proposed regulations.

3. Determination of Certified Rating

During the regional standards Working Group meetings, participants invested a great deal of time and energy discussing the relationship between system ratings and an effective enforcement plan. As part of the negotiations, the Working Group requested that DOE issue guidance regarding the applicability of regional standards to indoor units and outdoor units distributed separately and the applicability of regional standards to different combinations of indoor and outdoor units. DOE developed two draft guidance documents to address these issues. After consideration of the Working Group's discussions and the comments received on the two draft guidance documents, DOE determined that regulatory changes would be necessary to implement the approach agreed to by the Working Group. DOE is proposing several of those regulatory changes as part of this rulemaking. The remainder of the necessary regulatory changes will be addressed in a forthcoming regional standards enforcement notice of proposed rulemaking.

During the pendency of the rulemakings (CAC TP and Regional Standards), DOE reaffirms its commitment to the approach advocated by the Working Group, subject to consideration of comments received in the rulemakings to effectuate the necessary changes to the regulations. The following sections describe the two guidance documents and DOE's proposals to address them as part of this rulemaking.

a. Single-Split-System Air Conditioners Rated by OUMs

In the August 20, 2014 draft guidance document (Aug 20 Guidance) (EERE–2014–BT–GUID–0032), DOE proposed to clarify that when selecting which split-system air conditioner and heat pump units to test (in accordance with the DOE test procedure), a unit of each outdoor model must be paired with a unit of one selected indoor model. 10 CFR 429.16(a)(2)(i). Specifically, the manufacturer must test the condenser-evaporator coil combination that includes the model of evaporator coil that is likely to have the largest volume of retail sales with the particular model of condensing unit. 10 CFR 429.16(a)(2)(ii) (This combination is also known as the highest sales volume combination or HSVC.) That is, the HSVC for each condensing unit may not be rated using an ARM. (See section III.B regarding DOE's proposal to switch from ARMs to AEDMs for this product.)

The guidance further stated that for any other split-system combination that includes the same outdoor unit model but a different indoor unit model than the HSVC, manufacturers may determine represented values of energy efficiency (including those values that, for each combination, must be reported in certifications to DOE) of a split-system central air conditioner or heat pump basic model combination either by testing the combination in accordance with the DOE test procedure or by applying an ARM that has been approved by DOE in accordance with the provisions of 10 CFR 429.70(e)(1) and (2). 10 CFR 429.16(a)(2)(ii)(A) and (B)(1).

In the August 19, 2014 draft guidance document (August 19 Guidance) (EERE–2014–BT–GUID–0033), DOE proposed to clarify that split-system central air conditioners other than those with single-speed compressors may be tested and rated using a blower coil only if the condensing unit is sold exclusively for use with a blower coil indoor unit. 10 CFR 429.16(a)(2)(ii). The guidance stated that there is no provision in the Code of Federal Regulations (CFR) permitting use of a blower coil for testing and rating a split-system central air conditioner where the condensing unit is also offered for sale with a coil-only indoor unit, and that, furthermore, there is no provision in the CFR permitting the use of a blower coil for testing and rating a condensing unit with a single-speed compressor.

Commenters generally agreed with the information in the August 20 Guidance regarding selecting units for testing, rating, and certifying split-system combinations. In addition, in response to the August 19 Guidance, DOE received nearly identical comments from several stakeholders generally agreeing with the intent of the guidance to emphasize that single-speed compressor products must be tested and rated with a coil-only system as HSVC. (Docket No. EERE–2014–BT–GUID–0033, AHRI No. 8 at p. 2; Nordyne, No. 9 at p. 1; Lennox, No. 4 at p. 2; Ingersoll Rand, No. 3 at p. 1; Goodman, No. 10 at p. 1; Rheem, No. 2 at p. 2; JCI, No. 5 at p. 2–3) These stakeholders, as well as Mortex, clarified that other combinations besides the HSVC, including blower coil combinations, can be rated through testing or using an ARM. (*Id.;* Mortex, No. 6 at p. 1) Stakeholders recommended language identical to or similar to the following:

Split-system central air conditioners with single-speed compressors must be tested and rated using a coil-only for the HSVC. 10 CFR 429.16(a)(2)(ii). Such single-speed systems may be rated with other coil-only and blower coil indoor units through the use of a DOE approved ARM or by testing. 10 CFR 429.16(a)(2)(ii)(A) and 10 CFR 429.16(a)(2)(ii)(B). Furthermore, there is no provision in the CFR permitting the use of a blowercoil for testing and rating a condensing unit with a single-speed compressor for the HSVC, unless:

- [Version 1] the unit is a mini-split, multi-split or through-the-wall, OR
- [Version 2] the unit is sold and installed only with blower-coil indoor units.

(Version 1: Docket No. EERE–2014–BT–GUID–0033, Lennox, No. 4 at p. 2; Ingersoll Rand, No. 3 at p. 2; Goodman, No. 10 at p. 3; Rheem, No. 2 at p. 3; JCI, No. 5 at p. 4; Version 2: AHRI No. 8 at p. 3; Nordyne, No. 9 at p. 2)

AHRI and several manufacturers disputed that when using a compressor other than single speed, the HSVC can never be a blower coil unless it is exclusively used with a blower coil. AHRI and the manufacturers reported that many multi-stage capacity products are tested and rated with high efficiency blower coil or furnace products as the HSVC even though those systems are also rated for coil-only use. (Docket No. EERE–2014–BT–GUID–0033, AHRI No. 8 at p. 2; Nordyne, No. 9 at p. 2; Lennox, No. 4 at p. 2; Ingersoll Rand, No. 3 at p. 2; Goodman, No. 10 at p. 2; Rheem, No. 2 at p. 2; Carrier, No. 7 at p. 1) Johnson Controls responded that they test and rate multi-speed compressor units with blower coils or furnace/coils as the HSVC. (JCI, No. 5 at p. 3). AHRI and the manufacturers reported that not allowing this could limit the application of high performing products, and that it is important for units designed for blower coil to also be rated as coil-only to offer certain consumers a compromise of cost and performance. AHRI and the manufacturers proposed the following modified language:

Split-system central air conditioners other than those with single-speed compressors (two-stage or multi-stage) may be tested and rated using a blower-coil only as HSVC only if the condensing unit design intent is for use with a blower-coil indoor unit (*e.g.* the evaporator coil that is likely to have the largest volume of retails sales with the particular model of condensing unit is a blower-coil).

(Docket No. EERE–2014–BT–GUID–0033, AHRI No. 8 at p. 3; Nordyne, No. 9 at p. 2; Lennox, No. 4 at p. 3; Ingersoll Rand, No. 3 at p. 2; Goodman, No. 10 at p. 3; Rheem, No. 2 at p. 3; JCI, No. 5 at p. 4; Carrier, No. 7 at p. 2 with slightly different language)

After reviewing the comments, DOE proposes to make changes to 10 CFR 429.16 to revise the testing and rating requirements for single-split-system air conditioners. (See section III.F.4

regarding discussion of new definitions including ''single-split-system.'') These changes will occur in two phases. In the first phase, prior to the compliance date of any amended energy conservation standards, DOE proposes only a slight change to the current requirements. Specifically, DOE proposes that for single-split-system air conditioners with single capacity condensing units, each model of outdoor unit must be tested with the model of coil-only indoor unit that is likely to have the largest volume of retail sales with the particular model of outdoor unit. For split-system air conditioners with other than single capacity condensing units each model of outdoor unit must also be tested with the model of coil-only indoor unit likely to have the largest sales volume unless the model of outdoor unit is sold only with model(s) of blower coil indoor units, in which case it must be tested and rated with the model of blower coil indoor unit likely to have the highest sales volume. However, any other combination may be rated through testing or use of an AEDM. (See section III.B regarding proposed changes from ARM to AEDM.) Therefore, both single capacity and other than single capacity systems may be rated with models of both coil-only or blower coil indoor units, but if the system is sold with a model of coil-only indoor unit, it must, at a minimum, be tested in that combination.

In the second phase, DOE anticipates that any amended energy conservation standards will be based on blower coil ratings. Therefore, DOE proposes that all single-split-system air conditioner basic models be tested and rated with the model of blower coil indoor unit likely to have the largest volume of retail sales with that model of outdoor unit. Manufacturers would be required to also rate all other blower coil and coil-only combinations within the basic model but would be permitted do so through testing or an AEDM. DOE believes that this proposal will offer the benefits of design for high performance through the use of blower coils as well as providing appropriate representations for coil-only combinations. In addition, given that most basic models are currently submitted as blower coil ratings, this change will align DOE requirements with industry practice. This proposed change would also be accounted for in the parallel energy conservation standards rulemaking, and is contingent upon any proposed amended standards being based on blower coil ratings.

Table III.1 summarizes these proposed changes.

TABLE III.1—TEST REQUIREMENTS FOR SINGLE-SPLIT-SYSTEM NON-SPACE-CONSTRAINED AIR CONDITIONERS RATED BY OUMS

Date	Equipment type	Must test each:	With:
Before the compliance date for any amended energy conservation standards.	Split-System AC with single capacity condensing unit.	Model of Outdoor Unit	The model of coil-only indoor unit that is likely to have the largest volume of retail sales with the particular model of outdoor unit.
	Split-System AC with other than single capacity condensing unit.	Model of Outdoor Unit	The model of coil-only indoor unit that is likely to have the largest volume of retail sales with the particular model of outdoor unit, unless the model of outdoor unit is only sold with model(s) of blower coil indoor units in which case, the model of blower coil indoor unit that is likely to have the largest volume of retail sales with the particular model of outdoor unit.
After the compliance date for any amended energy conservation standards.	Split-system AC	Model of Outdoor Unit	The model of blower coil indoor unit that is likely to have the largest volume of retail sales with the particular model of outdoor unit.

In order to facilitate these changes, DOE also proposes definitions of blower coil indoor unit and coil-only indoor unit:

- *Blower coil indoor unit* means the indoor unit of a split-system central air conditioner or heat pump that includes a refrigerant-to-air heat exchanger coil, may include a cooling-mode expansion device, and includes either an indoor blower housed with the coil or a separate designated air mover such as a furnace or a modular blower (as defined in Appendix AA).
- *Blower coil system* refers to a split-system that includes one or more blower coil indoor units.
- *Coil-only indoor unit* means the indoor unit of a split-system central air conditioner or heat pump that includes a refrigerant-to-air heat exchanger coil and may include a cooling-mode expansion device, but does not include an indoor blower housed with the coil, and does not include a separate designated air mover such as a furnace or a modular blower (as defined in Appendix AA). A coil-only indoor unit is designed to use a separately-installed furnace or a modular blower for indoor air movement.
- *Coil-only system* refers to a system that includes one or more coil-only indoor units.

DOE notes that these proposed testing requirements, when combined with the proposed definition for basic model, require that each basic model have at least one rating determined through testing; no basic model can be rated solely using an AEDM.

DOE also proposes that in the certification report, manufacturers state whether each rating is for a coil-only or blower coil combination.

DOE seeks comment on its proposed changes to the determination of certified ratings for single-split-system air conditioners when rated by an OUM, as well as on the proposed definitions for blower coil and coil-only indoor units.

b. Split-System Heat Pumps and Space-Constrained Split Systems

The current requirements for split-system heat pumps in 10 CFR 429.16 require testing a condenser-evaporator coil combination with the evaporator coil likely to have the largest volume of retail sales with the particular model of condensing unit. The coil-only requirement does not apply to split-system heat pumps, because central heat pump indoor units nearly always include both a coil and a fan.

In this notice, DOE proposes to slightly modify the wording explaining this requirement; specifically, the requirement would use the more general terms ''indoor unit'' and ''outdoor unit,'' rather than ''evaporator coil'' and ''condensing unit,'' since the requirement addresses heat pumps. DOE also proposes to apply this same test requirement to space-constrained split-system air conditioners and heat pumps. The current requirements in 10 CFR

429.16 do not specifically call out space-constrained systems, and as such, the current coil-only requirements for split-system air conditioners apply to space-constrained split-system air conditioners. Therefore, this proposal will change test procedures for space-constrained split-system air conditioners but will not change, other than in nomenclature, the test procedures for space-constrained split-system heat pumps.

c. Multi-Split, Multi-Circuit, and Single-Zone-Multiple-Coil Units

The current requirements in 10 CFR 429.16(a)(2)(ii) specify that multi-split systems and mini-split systems designed to always be installed with more than one indoor unit (now proposed to be called single-zone-multiple-coil units, see section III.F.4) be tested using a ''tested combination'' as defined in 10 CFR 430.2. For multi-split systems, each model of condensing unit currently must be tested with a non-ducted tested combination and a ducted tested combination. Furthermore, current requirements for testing with a coil-only indoor unit do not apply to mini-splits or multi-splits, as the general use of these terms in the industry refers to specific types of systems with blower coil indoor units. *Id.*

The current requirements also state that for other multi-split systems that include the same model of condensing unit but a different set of evaporator coils, whether the evaporator coil(s) are manufactured by the same manufacturer or by a component manufacturer (*i.e.,* ICM), the rating must be: (1) Set equal to the rating for the non-ducted indoor unit system tested (for systems composed entirely of non-ducted units), (2) set equal to the rating for the ducted indoor unit system tested (for systems composed entirely of ducted units), or (3) set equal to the mean of the values for the two systems (for systems having a mix of non-ducted and ducted indoor units). (10 CFR 429.16(a)(2)(ii))

In this notice, DOE proposes a slight modification to the testing requirements for single-zone-multiple-coil and multi-split systems, and adds similar requirements for testing multi-circuit systems (see section III.C.2 for more information about these systems). DOE also clarifies that these requirements apply to VRF systems that are single-phase and less than 65,000 Btu/h (see section III.A.3.c for more details). For all multi-split, multi-circuit, and single-zone-multiple-coil split systems, DOE proposes that at a minimum, each model of outdoor unit must be tested as part of a tested combination (as defined in the CFR) composed entirely of non-ducted indoor units. For any models of outdoor units also sold with short-ducted indoor units, a second ''tested combination'' composed entirely of short-ducted indoor units would be required to be tested. DOE also proposes the manufacturers may rate a mixed non-ducted/short-ducted combination as the mean of the represented values for the tested non-ducted and short-ducted combinations.

Under the proposed definition of basic model, these three combinations (non-ducted, short-ducted, and mixed) would represent a single basic model. When certifying the basic model, manufacturers should report ''* * *'' for the indoor unit model number, and report the test sample size as the total of all the units tested for the basic model, not just the units tested for each combination. For example, if the manufacturer tests 2 units of a non-ducted combination and 2 units of a short-ducted combination, and also rates a mix-match combination, the manufacturer should specify ''4'' as the test sample size for the basic model, while providing the rating for each combination. DOE also proposes that manufacturers be allowed to test and rate specific individual combinations as separate basic models, even if they share the same model of outdoor unit. In this case, the manufacturer must provide the individual model numbers for the indoor units rather than stating ''* * *''. Table III.2 provides an example of both situations.

TABLE III.2—EXAMPLE RATINGS FOR MULTI-SPLIT SYSTEMS

Basic model	Individual model (outdoor unit)	Individual model (indoor unit)	Sample size	Ducted rating	Non-ducted rating	Mix rating
ABC	ABC	* * *	4	14	15	14.5
ABC1	ABC	2–A123; 3–JH746	2		17	

DOE requests comment on whether additional requirements are necessary for multi-split systems paired with models of conventional ducted indoor units rather than short-duct indoor units.

DOE also notes that the test procedure currently allows testing of only non-ducted or short-ducted systems, and not combinations of the two. Therefore to rate individual mix-match combinations, manufacturers would have to test 4 units—2 ducted and 2 short-ducted. DOE requests comment on whether manufacturers should have the ability to test mix-match systems using the test procedure rather than rating them using an average of the other tested systems. DOE also requests comment on whether manufacturers should be able to rate mix-match systems using other than a straight average, such as a weighting by the number of non-ducted or short-ducted units. Finally, DOE requests comment on whether the definition of ''tested combination'' is appropriate for rating specific individual combinations, or whether manufacturers should be given more flexibility, such as testing with more than 5 indoor units.

In reviewing the market for multi-split systems, DOE determined that some are sold by OUMs with only models of small-duct, high velocity (SDHV) indoor units, or with a mix of models of short-duct and SDHV units. (See section III.F.2 regarding the proposed definition of short ducted systems.) These kinds of units are not currently explicitly addressed in DOE's test requirements. Therefore, DOE proposes to add a requirement that for any models of outdoor units also sold with models of 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.

DOE notes that multi-split systems consisting of a model of outdoor unit paired with models of non-ducted or short-ducted units must meet the energy conservation standards for split-system air conditioners or heat pumps, while systems consisting of a model of outdoor unit paired with models of small-duct, high-velocity indoor units must meet SDHV standards. DOE proposes to add a limitations section in 429.16 that would require models of outdoor units that are rated and distributed in combinations that span multiple product classes to be tested and certified as compliant with the applicable standard for each product class. Even if a manufacturer sells a combination including models of both SDHV and other non-ducted or short-ducted indoor units, DOE proposes that the manufacturer may not provide a mix-match rating for such combinations. DOE requests comment on whether manufacturers would want to rate such combinations, and if so, how they would prefer to rate them (*i.e.,* by by taking the mean of a sample of tested non-ducted units and a sample of tested SDHV units or by testing a combination on non-ducted and SDHV units), and whether the SDHV or split-system standard would be most appropriate.

DOE understands that manufacturers of multi-split systems commonly only test one sample rather than complying with the sampling plan requirements in 429.16(a)(2)(i), which require a sample of two. DOE may consider moving toward a single unit sample for single-zone multiple-coil and multi-split system models, but in order to do so, DOE requires information on manufacturing and testing variability associated with these systems. In particular, DOE requires data to allow it to understand how a single unit sample may be representative of the population. DOE also requests information on what tolerances would need to be applied to the ratings of these units based on a single unit sample in order to account for the variability.

d. Basic Models Rated by ICMs

The current requirements in 10 CFR 429.16(a) require that each condensing unit of a split system must be tested using the HSVC associated with that condensing unit. There are no current requirements for testing each model of indoor unit of a split system. Non-HSVC combinations can be rated using an ARM, assuming the condensing unit of the combination has a separate HSVC rating based on testing. DOE understands that ICMs typically do not test all of their models of indoor units, but rather use OUM test data for outdoor units to generate ratings for their models. (See section III.B on AEDMs for further information.) In this notice, DOE proposes that ICMs must test and provide certified ratings for each model of indoor unit (*i.e.,* basic model) with the least-efficient model of outdoor unit with which it will be paired, where the least- efficient model of outdoor unit is the outdoor unit in the lowest-SEER combination as certified by the OUM. If more than one model of outdoor unit (with which the ICM wishes to rate the model of indoor unit) has the same lowest-SEER rating, the ICM may select one for testing purposes. This applies to both conventional (*i.e.,* non-short-duct, non-SDHV) split-systems and SDHV systems. ICMs must rate all other individual combinations of the same model of indoor unit, but may determine those ratings through testing or use of an AEDM.

DOE understands that this proposal would increase test burden for ICMs beyond the testing they currently conduct to meet ARM validation requirements. However, DOE believes this burden is outweighed by the benefit of providing more accurate ratings for models of indoor units sold by ICMs. Additional discussion regarding potential test requirements for ICMs can be found in the stakeholder comments regarding AEDMs in section III.B.5.

DOE understands that the proposed definition of basic model for an ICM, including what constitutes the ''same'' model of indoor unit and thus would be required to be tested, is important for accurately assessing the test burden for manufacturers as a result of this test proposal. DOE seeks comment on the basic model definition in section III.A.1. DOE also seeks comment on the proposed testing requirements for ICMs.

e. Single-Package Systems

In the current regulations, 10 CFR 429.16(a)(2)(i) states that each single-package system a must have a sample of sufficient size tested in accordance with the applicable provisions of Subpart B. In this notice, DOE proposes that the lowest SEER individual model within each basic model must be tested. DOE expects that in most cases, each single-package system will represent its own basic model. However, based on the proposal for the definition of basic model in section III.A.1, this may not always be the case. DOE notes that regardless, AEDMs do not apply to single-package models—manufacturers may either test and rate each individual single-package model, or if multiple individual models are assigned to the same basic model per the proposed requirements in the basic model definition, the manufacturer would be required to test only the lowest SEER individual model within the basic model and use that to determine the rating for the basic model.

DOE requests comment on the likelihood of multiple individual models of single-package units meeting the requirements proposed in the basic model definition to be assigned to the same basic model. DOE also requests comment on whether, if manufacturers are able to assign multiple individual models to a single basic model, manufacturers would want to use an AEDM to rate other individual models within the same basic model other than the lowest SEER individual model. Finally, DOE requests comment on whether manufacturers would want to employ an AEDM to rate the off-mode power consumption for other variations of off-mode associated with the basic model other than the variation tested.

DOE also proposes to specify this same requirement for space-constrained single-package air conditioners and heat pumps, which are currently not explicitly identified in the test requirement section.

f. Replacement Coils

DOE stated in the August 20 Guidance that an individual condensing unit or coil must meet the current Federal standard (National or regional) when paired with the appropriate other new part to make a system when tested in accordance with the DOE test procedure and sampling plan.

In response, AHRI and manufacturers commented that they believed the intent of the guidance was to clarify how the outdoor section of a split system used in a replacement situation can be tested and rated to meet the appropriate efficiency requirements. However, they felt this language should not apply to the indoor coil. AHRI stated that indoor coil is rarely changed and when it is, such as for an irreparable leak, it requires an exact replacement. In addition, they note that warranties can extend up to 10 years. Commenters also expressed the view that the guidance would not result in an improvement to installed product efficiency. (Docket No. EERE–2014–BT–GUID–0032, AHRI, No. 8 at pp. 2–3; Rheem, No. 2 at p. 3; Goodman, No. 10 at pp. 2–3; Ingersoll Rand, No. 3 at p. 2; Lennox, No. 4 at p. 2; Nordyne, No. 9 at p. 2) AHRI and the manufacturers recommended removing indoor coils from the draft guidance language on replacement. (*Id.;* JCI, No. 5 at p. 6)

Johnson Controls added further detail that using the term coil does not differentiate between service parts (listed with part numbers) and finished component assemblies (listed as a coil model) or between evaporator coils and condenser coils. Johnson Controls added that replacement parts cannot be rated as a finished coil assembly because the replacement parts do not contain sheet metal parts required to complete the installation. They also added that where the physical characteristics of an evaporator coil are significantly different when compared to a new system, replacing the old evaporator coil with a new coil model rather than a replacement part could result in increased cost and reduced performance, reliability, and comfort. (Docket No. EERE–2014–BT–GUID–0032, JCI, No. 5 at pp. 4–6)

Mortex also commented that replacement with a different evaporator coil design and size could lead to issues of fitting or size constraint problems and refrigerant metering and charging differences. The end result (if design air volume rate is hampered and refrigerant circuit performance is modified) could lead to less efficiency than the pre-failure situation. (Docket No. EERE–2014–BT–GUID–0032, Mortex, No. 6 at p. 1)

DOE also notes that the ASRAC regional standards enforcement Working Group agreed that manufacturers do not need to keep track of components including uncased coils. (Docket No. EERE–2011–BT–CE–0077–0070, Attachment)

In consideration of the comments and the Working Group proposals, DOE notes that its proposed definition of ''indoor unit'' refers to the box rather than just a coil. Accordingly, legacy indoor coil replacements and uncased coils would not meet the definition of indoor unit. Furthermore, by defining air conditioners and heat pumps as consisting of a single-package unit, an outdoor unit and one or more indoor units, an indoor unit only, or an outdoor unit only, legacy indoor coil replacements and uncased coils would not meet the definition of a central air conditioner or heat pump. Hence, they would not need to be tested or certified as meeting the standard.

g. Outdoor Units With No Match

For split-system central air conditioners and heat pumps, current DOE regulations require that manufacturers test the condensing unit and ''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). 10 CFR 4429.16(a)(2)(ii). 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, manufacturers inquired how to test and rate individual components—because these components are sold separately, there are no highest sales volume combinations. Because the EPA prohibits distribution of new HCFC–22 condensing unit and coil combinations (*i.e.,* complete systems), there is no such thing as a HSVC, and hence, testing and rating of new HCFC–22 combinations cannot be conducted using the existing test procedure.

DOE expects that the HCFC–22 indoor and outdoor units remaining on the market are part of legacy offerings that were initially sold five or more years ago. These components of HCFC–22 systems were in production for sale as part of matched systems before the EPA regulations became effective on January 1, 2010. While EPA's rulemaking bans the sale of HCFC–22 systems that are charged with refrigerant while allowing sale of uncharged components of such systems, EPA's rule has no effect on the efficiency rating of these systems or on requirements for DOE efficiency standards that they must meet. The DOE test procedure used prior to January 15, 2010 that would have been used to rate these systems is no longer valid, thus these ratings can no longer be used as the basis for representing their efficiency. The individual indoor coils and outdoor units of such systems that could potentially meet the current standard may continue to be manufactured only if the manufacturer uses a valid test procedure to ensure compliance (*i.e.,* to certify compliance) and for representations.

Generally, when a model cannot be tested in accordance with the DOE test procedure, manufacturers must submit a petition for a test procedure waiver for DOE to assign an alternative test method. 10 CFR 430.27(a)(1) Instead, DOE proposes in this notice a test procedure that may be used for rating and certifying the compliance of these outdoor units. DOE proposes in this notice to specify coil characteristics that should be used when testing models of outdoor units that do not have a HSVC. Specifically, these requirements include limitations on coil tube geometries and dimensions and coil fin surface area. These outdoor unit models, when tested with the specified indoor units, must meet applicable Federal standards. (See section III.A.4 for more information on compliance.) This proposal is consistent with the regional standards enforcement Working Group recommendation that a person cannot install a replacement outdoor unit unless it is certified as part of a combination that meets the applicable standard. (Docket No. EERE–2011–BT–CE–0077–0070, Attachment) The new test procedure would be effective (*i.e.,* allowed for use for such certifications) 30 days after it is finalized and would be required for use for such systems (*i.e.,* rather than any granted waiver test procedure) beginning 180 days after it is finalized.

In response to the August 20, 2014 draft guidance document, Carrier requested clarification that the finalized guidance would replace DOE's draft guidance document issued on January 1, 2012, regarding central air conditioning systems and air conditioning heat pump systems that are designed to use dry R–22 condensing units. (Docket No. EERE–2014–BT–GUID–0032, Carrier, No. 7 at p. 2) If finalized, this proposed test procedure would replace both the 2012 guidance document for dry R–22 units as well as the 2014 draft guidance document on unit selection regarding condensing units for replacement applications.

4. Compliance With Federal (National or Regional) Standards

In the August 20, 2014 draft guidance document (EERE–2014–BT–GUID–0032), DOE discussed whether each basic model of split-system air conditioner or heat pump has to meet the applicable standard. DOE stated that compliance with standards is based on the statistical concept that an entire population of units (where ''unit'' refers to a complete system) of a basic model must meet the standard, recognizing that efficiency measurements for some units may be better or worse than the standard due to manufacturing or testing variation. Manufacturers apply the statistical formulae in 10 CFR 429.16 to demonstrate compliance, and DOE applies the statistical formulae in 10 CFR part 429, subpart C, Appendix A to determine compliance.

Further, DOE stated that the only condensing units and coils that may be installed in the region are those that can meet the regional standard when tested and rated as a new system in accordance with the test procedure and sampling plan as described above.

In response, AHRI and several manufacturers recommended the following additions to DOE's statements regarding compliance:

> ''Compliance with *national or regional standards* is based on the statistical concept that an entire population of units (where ''unit'' refers to a complete system) of a basic model *including Highest Sales Volume Tested Combination and all other combinations* must meet the standard, recognizing that some individual units may perform slightly better or worse than the design due to manufacturing or testing variation.''

(Docket No. EERE–2014–BT–GUID–0032, AHRI, No. 8 at p. 2; Rheem, No. 2 at p. 2; Goodman, No. 10 at p. 2; Ingersoll Rand, No. 3 at p. 1; Lennox, No. 4 at p. 2; Nordyne, No.

9 at pp. 1–2; JCI, No. 5 at p. 3; Carrier, No. 7 at p. 6)

In addition, Carrier commented that with respect to the discussion about selection of units for testing, the HSVC should be determined for the applicable region. (Docket No. EERE–2014–BT–GUID–0032, Carrier, No. 7 at p. 4)

AHRI and several manufacturers recommended the following addition to the paragraph on condensing units sold as replacements:

''In summary, DOE interprets for the regional standard to require that the least efficient rating combination for a specified model of condensing unit must be 14 SEER with a coil only rating where 14 SEER is the regional standard. Any model that has a certified combination below the regional standard cannot be installed in the region. This interpretation of the regional standard also applies to units shipped without refrigerant charge.''

(Docket No. EERE–2014–BT–GUID–0032, AHRI, No. 8 at p. 2; Rheem, No. 2 at p. 3; Goodman, No. 10 at p. 3; Ingersoll Rand, No. 3 at p. 3; Lennox, No. 4 at p. 3; Nordyne, No. 9 at pp. 2–3; JCI, No. 5 at p. 6)

Carrier provided slightly different recommended language:

''Given the different Federal standards, National and regional, the least efficient rating combination for a specified model of condensing unit must: (i) in the regions where the regional standard applies, be rated and certified on as performing at or above the current regional standard with a coil only rating; and (ii) where the National standard applies, be rated and certified as performing at or above the current National standard with a coil only rating. For purposes of clarity, any basic model that has a certified combination below the current regional standard cannot be installed in the region. This interpretation also applies to dry condensing units.'' (Docket No. EERE–2014–BT–GUID–0032, Carrier, No. 7 at pp. 1–2)

In contrast, Carrier also suggested that the guidance document discussion of unit selection and basic models should replace references to ''Federal standard'' with ''Federal (national or regional) standard''. (Carrier, No. 7 at pp. 4–5)

The regional standards enforcement Working Group suggested the regional standards required clarification because a particular condensing unit may have a range of efficiency ratings when paired with various indoor evaporator coils and/or blowers. The Working Group provided the following four recommendations to clarify the regional standards: That (1) the least-efficient rated combination for a specified model of condensing unit must be 14 SEER for models installed in the Southeast and Southwest regions; (2) the least-efficient rated combination for a specified model of condensing unit must meet the minimum EER for models installed in the Southwest region; (3) any condensing unit model that has a certified combination that is below the regional standard(s) cannot be installed in that region; and (4) a condensing unit model certified below a regional standard by the original equipment manufacturer cannot be installed in a region subject to a regional standard(s) even with an independent coil manufacturer's indoor coil or air handler combination that may have a certified rating meeting the applicable regional standard(s). (Docket No. EERE–2011–BT–CE–0077–0070, Attachment)

After reviewing stakeholder comments and the Working Group report, DOE agrees that all individual models or combinations within a basic model must meet the applicable national or regional standard. DOE proposes to add requirements to the relevant provisions of section 430.32 that the least-efficient combination of each basic model must comply with the regional SEER and EER standards.

In addition, as noted in section III.A.1, DOE proposes that if any individual combination within a basic model fails to meet the standard, the entire basic model (*i.e.,* model of outdoor unit) must be removed from the market. In order to clarify the limitations on sales of models of outdoor units across regions with different standards, DOE proposes to add a limitation in section 429.16 that any model of outdoor unit that is certified in a combination that does not meet all regional standards cannot also be certified in a combination that meets the regional standard(s). Outdoor unit model numbers cannot span regions unless the model of outdoor unit is compliant with all standards in all possible combinations. If a model of outdoor unit is certified below a regional standard, then it must have a unique individual model number for distribution in each region. For example:

Basic model	Individual model # (outdoor unit)	Individual model # (indoor unit)	Certified rating (SEER/EER)	Permitted?
AB12	ABC**#**-***	SO123	14.5/12.0	NO.
AB12	ABC**#**-***	SW123	15.0/12.8	
AB12	ABC**#**-***	N123	13.9/11.7	
CD13	CDESO**-*#*	SO123	14.5/12.0	YES.
CD13	CDESW**-*#*	SW123	15.0/12.8	
CD13	CDEN***-*#*	N123	13.9/11.7	
EF12	EFCS**#**-***	SO123	14.5/12.2	YES.
EF12	EFCS**#**-***	SW123	14.6/12.4	
EF12	EFCN**#**-***	N123	13.9/11.7	

5. Certification Reports

To maximize test repeatability and reproducibility for assessment and enforcement testing, DOE proposes to amend the certification reporting requirements.

DOE proposes to clarify what basic model number and individual model numbers must be reported for central air conditioners and heat pumps:

Equipment type	Basic model number	Individual model number(s)		
		1	2	3
Single Package	Number unique to the basic model.	Package	N/A	N/A.
Split System (rated by OUM)	Number unique to the basic model.	Outdoor Unit	Indoor Unit(s)	Air Mover (or N/A if rating coil-only system or fan is part of indoor unit model number).

Equipment type	Basic model number	Individual model number(s)		
		1	2	3
Outdoor Unit Only	Number unique to the basic model.	Outdoor Unit	N/A	N/A.
Split-System or SDHV (rated by ICM).	Number unique to the basic model.	Outdoor Unit	Indoor Unit(s)	N/A.

Each basic model number must be unique in some way so that all individual models or combinations within the same basic model can be identified.

DOE also proposes to require product-specific information at 10 CFR 429.16(c)(4) that is not public and will not be displayed in DOE's database. Several proposed requirements are addressed in the remainder of this notice in response to comments on specific issues or in relation to test procedure changes. In addition, several other requirements are discussed in this section.

In order for DOE to replicate the test setup for its assessment tests, DOE proposes that manufacturers that wish to certify multi-split, multiple-circuit, and single-zone-multiple-coil systems report the number of indoor units tested with the outdoor unit, the nominal cooling capacity of each indoor unit and outdoor unit, and the indoor units that are not providing heating or cooling for part-load tests. Manufacturers that wish to certify systems that operate with multiple indoor fans within a single indoor unit shall report the number of indoor fans; the nominal cooling capacity of the indoor unit and outdoor unit; which fan(s) are operating 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.

Similarly, DOE proposes that for those models of indoor units designed for both horizontal and vertical installation or for both up-flow and down-flow vertical installations, the orientation used during certification testing shall be included on the certification test reports.

DOE also proposes that the maximum time between defrosts as allowed by the controls be included on the certification test reports. For units with time-adaptive defrost control, the frosting interval used during the Frost Accumulation tests and the associated procedure for manually initiating defrost at the specified time, if applicable, should also be included on the certification test reports.

DOE also proposes that for variable-speed units, the compressor frequency set points and the required dip switch/control settings for step or variable components should be included. For variable-speed heat pumps, DOE proposes that manufacturers report whether the unit controls restrict use of minimum compressor speed operation for some range of operating ambient conditions, whether the unit controls restrict use of maximum compressor speed operation for any ambient temperatures below 17 °F, and whether the optional $H4_2$ low temperature test was used to characterize performance at temperatures below 17 °F.

Finally, DOE proposes that manufactures report air volume rates and airflow-control settings.

DOE recognizes that additional reporting requirements in certification test reports increases reporting burden because manufacturers must spend additional time to add such content to the report. However, DOE believes that a knowledgeable person in the field would not find the additional information difficult to provide and could do so in a reasonable amount of time. Thus, DOE does not believe that the added reporting requirements are significantly burdensome to warrant excluding them. DOE requests comment on this issue.

6. Represented Values

DOE proposes to make several additions to the represented value requirements in 10 CFR 429.16. First, DOE proposes to add a requirement that the represented value of cooling capacity, heating capacity, and sensible heat ratio (SHR) shall be the mean of the values measured for the sample. Second, DOE proposes to move the provisions currently in 10 CFR 430.23 regarding calculations of various measures of energy efficiency and consumption for central air conditioners to 10 CFR 429.16. Specifically, while Part 430 would refer to the test procedure appendix and section therein to use for each metric and the rounding requirements for test results of individual units, Part 429 would refer to how to calculate annual operating cost for the sample based on represented values of cooling capacity and SEER, and how to round the represented values based on the sample for other measures of energy efficiency and consumption. DOE proposes minor changes to the calculations of annual operating cost to address changes proposed in Appendix M and M1. Table III.3 shows the proposed rounding requirements for each section. DOE requests comment on these values.

TABLE III.3—ROUNDING PROPOSALS

Measure	10 CFR 430.23 (one unit)	10 CFR 429.16 (sample)
Cooling capacity/heating capacity:		
<20,000 Btu/h	nearest 50 Btu/h	nearest 100 Btu/h.
≥20,000 Btu/h and <38,000 Btu/h	nearest 100 Btu/h	nearest 200 Btu/h.
≥38,000 Btu/h and <65,000 Btu/h	nearest 250 Btu/h	nearest 500 Btu/h
Annual operating cost	N/A	nearest dollar per year.
EER/SEER/HSPF/APF	nearest 0.025	nearest 0.05.
Off-mode power consumption	nearest 0.5 watt	nearest watt.
Sensible heat ratio	nearest 0.5%	nearest percent (%).

7. Product-Specific Enforcement Provisions

DOE proposes to verify during assessment or enforcement testing the cooling capacity certified for each basic model or individual combination. DOE proposes to measure the cooling capacity of each tested unit pursuant to the test requirements of 10 CFR part 430. The results of the measurement(s) will be compared to the value of cooling capacity certified by the manufacturer. If the measurement is within five percent of the certified cooling capacity, DOE will use the certified cooling capacity as the basis for determining SEER. Otherwise, DOE will use the measured cooling capacity as the basis for determining SEER.

DOE also proposes to require manufacturers to report the cyclic degradation coefficient (C_D) value used to determine efficiency ratings. In this proposal, DOE would run C_D testing as part of any assessment or verification testing, except when testing an outdoor unit with no match. If the measurement is 0.02 or more greater than the certified value, DOE would use the measurement as the basis for calculation of SEER or HSPF. Otherwise, DOE would use the certified value. For models of outdoor units with no match, DOE would always use the default value.

B. Alternative Efficiency Determination Methods

1. General Background

For certain consumer products and commercial equipment, DOE's existing regulations allow the use of an alternative efficiency determination method (AEDM) or alternative rating method (ARM), in lieu of actual testing, to estimate the ratings of energy consumption or efficiency of basic models by simulating their energy consumption or efficiency at the test conditions required by the applicable DOE test procedure. The simulation method permitted by DOE for use in rating split-system central air conditioners and heat pumps, in accordance with 10 CFR 429.70(e), is referred to as an ARM. In contrast to an AEDM, an ARM must be approved by DOE prior to its use.

The simulation methods represented by AEDMs or ARMs are computer modeling or mathematical tools that predict the performance of non-tested individual or basic models. They are derived from mathematical models and engineering principles that govern the energy efficiency and energy consumption of a particular basic model of covered product based on its design characteristics. (In the context of this discussion, the term ''covered product'' applies both to consumer products and commercial and industrial equipment that are covered under EPCA.) These computer modeling and mathematical tools can provide a relatively straightforward means to predict the energy usage or efficiency characteristics of an individual or basic model of a given covered product and reduce the burden and cost associated with testing certain covered products that are inherently difficult or expensive to test. When properly developed, they can predict the performance of a product accurately enough to be statistically representative under DOE's sampling requirements.

On April 18, 2011, DOE published a Request for Information (AEDM RFI) in the **Federal Register**. 76 FR 21673. Through the AEDM RFI, DOE requested suggestions, comments, and information relating to the Department's intent to expand and revise its existing AEDM and ARM requirements for consumer products and commercial and industrial equipment covered under EPCA. In response to comments it received on the AEDM RFI, DOE published a Notice of Proposed Rulemaking (AEDM NOPR) in the **Federal Register** on May 31, 2012. 77 FR 32038. DOE also held a public meeting on June 5, 2012, to present proposals in the AEDM NOPR and to receive comments from stakeholders. In the AEDM NOPR, DOE proposed the elimination of ARMs, and the expansion of AEDM applicability to those products for which DOE allowed the use of an ARM (*i.e.,* split-system central air conditioners and heat pumps). 77 FR at 32055. Furthermore, DOE proposed a number of requirements that manufacturers must meet in order to use an AEDM as well as a method that DOE would employ to determine if an AEDM was used appropriately along with specific consequences for misuse of an AEDM. 77 FR at 32055–56.

The purpose of the AEDM rulemaking was to establish a uniform, systematic, and fair approach to the use of modeling techniques that would enable DOE to ensure that products in the marketplace are correctly rated—irrespective of whether they are rated based on physical testing or modeling—without unnecessarily burdening regulated entities. DOE solicited suggestions, comments, and information related to its proposal and accepted written comments on the AEDM NOPR through July 2, 2012. DOE subsequently formed a working group through the Appliance Standards and Rulemaking Federal Advisory Committee (ASRAC) (see the Notice of Intent To Form the Commercial HVAC, WH, and Refrigeration Certification Working Group and Solicit Nominations To Negotiate Commercial Certification Requirements for Commercial HVAC, WH, and Refrigeration Equipment, published on March, 12, 2013, 78 FR 15653), which addressed revisions to the AEDM requirements for commercial and industrial equipment covered by EPCA and resulted in the subsequent publishing of a SNOPR on October 22, 2013 (78 FR 62472) and a final rule on December 31, 2013 (78 FR 79579). In the final rule, DOE made, among others changes, revisions to pre-approval requirements, validation requirements, and DOE verification testing requirements for the AEDM process for commercial HVAC equipment.

In this notice, DOE proposes modifications to the central air conditioners and heat pump AEDM requirements proposed in the AEDM NOPR with consideration of the comments received on the AEDM NOPR specific to these products, as well as the requirements implemented for commercial HVAC equipment in the December 2013 AEDM final rule.

2. Terminology

In the AEDM NOPR, DOE proposed to eliminate the term ''alternate rating method'' (ARM) and instead use the term ''alternative efficiency determination method'' (AEDM) to refer to any modeling technique used to rate and certify covered products. 77 FR 32038, 32040 (May 31, 2012). DOE proposed to refer to any technique used to model product performance as an AEDM, but recognized that there are product-specific considerations that should be accounted for in the development of an AEDM and thus, in the proposed methodology for validating product-specific AEDMs. *Id.*

DOE received a number of comments in response to its proposal to solely apply the term AEDM to any modeling technique used to rate and certify covered products. Bradford White Corporation (Bradford White), United Technologies Climate, Controls & Security and ITS Carrier (UTC/Carrier), and Nordyne, LLC (Nordyne) agreed with DOE that one term should be used. (Docket No. EERE–2011–BT–TP–0024, Bradford White, No. 38 at p. 1; UTC/Carrier, No. 56 at p. 1; Nordyne, No. 55 at p. 1) [8] AAON, Inc. (AAON) supported

[8] Unless otherwise specified, further references in this section (section III.B) to comments received by DOE are to those associated with the AEDM rulemaking (Docket No. EERE–2011–BT–TP–0024). References to the public meeting are to the June 5, 2012 public meeting on the AEDM NOPR, the transcript of which is in the AEDM rulemaking docket.

DOE's proposal to combine requirements for ARMs and AEDMs, but did not differentiate between the terminology and the methodological changes proposed. (AAON, No. 40 at p. 2) DOE also received a number of comments, both written and at the public meeting, regarding the differences in ARM and AEDM methodology. Those comments are discussed in section III.B.3 of this document. In addition, DOE received numerous comments regarding the validation of AEDMs for different product types, which are discussed in section III.B.4 of this document.

In response to comments received, DOE is continuing to propose the use of one term, AEDM, to refer to all modeling techniques used to develop certified ratings of covered products. DOE believes that since the two methods are conceptually similar, the use of one term is appropriate. DOE would like to clarify that the use of one term to refer to all modeling techniques used to develop certified ratings of covered products and equipment does not indicate a uniform process or requirements for their use across all covered products, nor does it imply that DOE will not include any of the current ARM provisions as part of the proposed AEDM provisions. Further, similar to the differences between AEDMs for distribution transformers and commercial HVAC products, DOE proposes validation requirements that will account for the differences between HVAC products and other covered equipment.

3. Elimination of the Pre-Approval Requirement

Under current regulations, ARMs used by manufacturers of split-system central air conditioners and central heat pumps must be approved by the Department before use. (10 CFR 429.70(e)(2)) Manufacturers who elect to use an ARM to rate untested basic models pursuant to 10 CFR 429.16(a)(2)(ii)(B)(1) must, among other requirements, submit to the Department full documentation of the rating method including a description of the methodology, complete test data on four mixed systems per each ARM, and product information on each indoor and outdoor unit of those systems. Furthermore, manufacturers are not permitted to use the ARM as a rating tool prior to receiving Departmental approval.

In the AEDM RFI, DOE requested comment on the necessity of a pre-approval requirement for AEDMs and/or ARMs. 76 FR 21673, 21674 (April 18, 2011). Based on the comments received in response to the AEDM RFI, DOE perceived no benefit in the additional burden imposed by a pre-approval requirement and that a pre-approval process could cause time-to-market delays. Pursuant to those comments, DOE proposed in the AEDM NOPR to eliminate the pre-approval process currently in place for central air conditioner and heat pump ARMs. 77 FR 32038, 32040–41 (May 31, 2012). DOE believed that this would reduce the burden currently placed on manufacturers by eliminating the time-to-market delays caused by completing the necessary request for approval before bringing products to market. Furthermore, DOE believed that elimination of the pre-approval requirement would promote innovation because an ARM would not need to be approved or re-approved to account for any changes in technology. *Id.*

In the AEDM NOPR, DOE sought comment regarding its proposal to eliminate the pre-approval requirement for ARMs for central air conditioners and heat pumps and received mixed responses. Modine Manufacturing Corporation (Modine) supported DOE's proposal to eliminate the pre-approval requirement. (Modine, No. 42 at p. 1) Lennox International, Inc. (Lennox) and Unico, Inc. (Unico), however, suggested that removal of the pre-approval requirement could lead to incorrect ratings and unfair competition in the marketplace, which could negatively impact consumers. (Lennox, No. 46 at p. 2; Unico, No. 54 at p. 2) Furthermore, Johnson Controls, Inc. (JCI) commented that it was particularly important that manufacturers continue to be allowed to use pre-approved ARMs because the new AEDM provisions, by eliminating pre-approval, introduce regulatory risk that is not present under current ARM requirements. (JCI, No. 66 at pp. 2)

Other interested parties specifically recommended that participation in a voluntary industry certification program (VICP),[9] or review of an AEDM or ARM by a qualified engineer, could reduce or eliminate the need for pre-approval. AHRI, Rheem Manufacturing Company (Rheem), Goodman Global, Inc. (Goodman), and Unico suggested that DOE should consider pre-approval for manufacturers not participating in a VICP, and that at a minimum, review by a professional engineer should be required. (AHRI, No. 61 at p. 2; Rheem, No. 59 at p. 2; Goodman, No. 53 at p. 1; Unico, No. 54 at p. 5) Likewise, Lennox agreed that if DOE does not maintain pre-approval in general, it could still require pre-approval for those who do not participate in a VICP . (Lennox, No. 46 at pp. 2 and 4) Lennox and Rheem commented that a pre-approval requirement for manufacturers who do not participate in a VICP could protect consumers from unsubstantiated ratings. (Rheem, No. 59 at p. 2; Lennox, No. 46 at p. 2)

DOE does not agree with JCI's suggestion that the elimination of pre-approval could create additional burden for manufacturers in cases where they fail to meet certified ratings and are subsequently required to re-substantiate their AEDM. DOE also does not agree with Rheem Lennox, and Unico who claim that the elimination of pre-approval will lead to incorrect ratings in the marketplace or create unfair competition. Pre-approval of an ARM that is used to certify a basic model rating does not mean that the basic model is correctly rated. Products that are certified using an approved ARM are subject to the same assessment testing and enforcement actions as those products certified through testing and/or use of an AEDM. Further, DOE currently has the authority to review approved ARMs at any time, including review of documentation of tests used to support the ARM. DOE may also test products that were certified using an ARM to determine compliance with the applicable sampling provisions, as well as with federal standards. Should DOE determine that products were incorrectly rated, DOE may require that the ARM is no longer used. Similarly, AEDMs used to certify ratings are subject to review at any time, as well as the potential for suspension should DOE determine that products were incorrectly rated. Additionally, as discussed in section III.A.3.a, each basic model must have at least one rating determined through testing; no basic model can be rated solely using an AEDM, which reduces the likelihood of significant error. Finally, use of a pre-approved ARM does not insulate a manufacturer from responsibility for the accuracy of their ratings, and the misconception that it does presents another reason to eliminate DOE review. Most manufacturers have not updated their ARMs and submitted the revised ARM for DOE review as required by regulation since prior to the last standards update and, thus, are effectively using unapproved or outdated ARMs currently. For these reasons, it is DOE's view that the elimination of the pre-approval process would not have a substantive

[9] A Voluntary Industry Certification Program, or VICP, is an independent, third-party program that conducts ongoing verification testing of members' products.

detrimental effect on the accuracy of a manufacturer's ratings, will improve manufacturers' ability to introduce new products into the marketplace, and will not represent a significant change from the status quo.

For the forgoing reasons, in this SNOPR, DOE proposes to eliminate the pre-approval process for ARMs for split-system central air conditioners and heat pumps. As stated in the AEDM NOPR, DOE believes that this will reduce time-to-market delays, facilitate innovation, and eliminate the time required to complete the approval process. Furthermore, DOE emphasizes that the Department's treatment of products that are currently rated and certified with the use of an ARM does not differ from its treatment of products currently rated and certified using an AEDM, except for the pre-approval requirement. (See for example 10 CFR 429.70(c).)

In addition, DOE proposes that manufacturers may only apply an AEDM if it (1) is derived from a mathematical model that estimates performance as measured by the applicable DOE test procedure; and (2) has been validated with individual combinations that meet current Federal energy conservation standards (as discussed in the next section). Furthermore, DOE proposes records retention requirements and additional manufacturer requirements to permit DOE to audit AEDMs through simulations, review of data and analyses, and/or certification testing.

4. AEDM Validation

In the AEDM NOPR, DOE proposed product-specific AEDM validation requirements meant to reduce confusion and allow for easier development and utilization of AEDMs by manufacturers. 77 FR 32044–32045. The proposed validation requirements applicable to central air conditioner and heat pump products would have required manufacturers to:

a. Test a minimum of five basic models, including at least one basic model from each product class to which the AEDM would be applied.

b. Test the smallest and largest capacity basic models from the product class with the highest sales volume.

c. Test the basic model with the highest sales volume from the previous year, or the basic model which is expected to have the highest sales volume for newly introduced basic models.

d. Validate only with test data that meets applicable Federal energy conservation standards and was derived using applicable DOE testing procedures.

In response to these proposed validation requirements, DOE received a number of comments from stakeholders addressing specific products covered by the AEDM rule. Comments applicable to the proposed requirements for central air conditioner and heat pump products are discussed in the following sections.

a. Number of Basic Models From a Product Class Necessary To Validate an AEDM

Commenter responses with regard to the minimum sample size of one unit each of five different basic models were mixed, with some commenters agreeing with DOE's proposal and some offering alternative sample sizes. Both AAON and Goodman agreed with DOE's proposal that a minimum of one unit each of five basic models be tested to validate the AEDM. (AAON, No. 40 at p. 6; Goodman, No. 53 at p. 2) AHRI, however, commented that it was not realistic for a manufacturer who produces two basic models, for example, to be required to validate an AEDM based on a minimum sample of five units of the same two basic models. (AHRI, Public Meeting Transcript, No. 69 at p. 154) Furthermore, AHRI stated that it is disproportionately burdensome to require testing of at least five basic models for small manufacturers who manufacture or wish to use an AEDM for only a few basic models compared to manufacturers who offer many basic models and many product classes. AHRI recommended that DOE require testing of only 3 basic models if the AEDM is to be applied to 15 or fewer basic models. (AHRI No. 61 at p. 3) United Cool Air agreed with AHRI's concerns and stated that to obtain data that are statistically robust enough to meet the validation requirements, testing of at least two to five units of many basic models would be necessary, which may be too burdensome for built-to-order and small manufacturers. This would be particularly burdensome in cases where models used for testing cannot be sold. (United Cool Air, No. 51 at pp. 7, 10, and 11) Acknowledging the amount of work and complex testing required for validation of an AEDM, Zero Zone, Inc. (Zero Zone) noted that it would be difficult for small manufacturers to comply. Zero Zone recommended that small manufacturers could be exempt or have a different sample size requirement. (Zero Zone, Public Meeting Transcript, No. 69 at p. 65)

Other stakeholders commented on the validation requirements for specific products. JCI stated that testing of five units is unnecessarily burdensome and suggested that testing a minimum of three units would be sufficient to validate HVAC AEDMs. (JCI, No. 66 at p. 6) First Co. stated that DOE's proposed requirements would unreasonably burden small manufacturers, especially independent coil manufacturers because they would not have knowledge of which condensing unit model is expected to have the highest sales volume in the coming year. First Co. stated that this proposed requirement is unnecessary and should be eliminated given that the proposed validation requirements already include testing of the smallest and largest capacity basic model from the product class with the highest sales volume, and that the current minimum number of tests required for obtaining ARM approval is four. (First Co., No. 45 at p. 2) JCI agreed with First Co., stating that the proposal would create an overrepresentation of the highest sales volume product class because the highest sales volume basic model is most likely from that product class, and along with the requirement to test the smallest and largest capacity basic model from that product class, would require testing of three basic models from the highest sales volume product class. (JCI, No. 66 at p. 7) Goodman, on the other hand, stated that an additional test beyond the currently required four tests would not cause significant burden. (Goodman, No. 53 at p. 2)

DOE notes that in its proposed revisions to the determination of certified ratings for central air conditioners and heat pumps (discussed in section III.A.3), manufacturers must test each basic model; specifically for split-system air conditioners and heat pumps, OUMs must test each model of outdoor unit with at least one model of indoor unit (highest sales volume), and ICMs must test each model of indoor unit with at least one model of outdoor unit (lowest SEER). Manufacturers would only be able to use AEDMs for other individual combinations within the same basic model—in other words, other combinations of models of indoor units with the same model of outdoor unit. DOE does not seek to require additional testing to validate an AEDM beyond what is proposed under 10 CFR 429.16(a)(1)(ii). Therefore, the testing burden required to validate an AEDM would depend on the number of basic models each manufacturer must rate. Furthermore, because ICMs must test each model of indoor unit with the lowest-SEER model of outdoor unit with which it is paired, First Co.'s concerns related to predicting the highest sales volume model would no longer be relevant. DOE requests comment on its proposal related to the testing requirements for validation of an AEDM.

Regarding the proposed requirement to test a basic model from each applicable product class for HVAC products, Goodman believes that the current definition of ''product class'' does not address the specific issues raised by split-system central air conditioners and heat pumps, which consist of separate indoor and outdoor coils that only function as intended when paired with one another to form a unitary split-system central air conditioner or heat pump. Hence, Goodman suggested that DOE consider the following product types to constitute individual validation classes: Split-system air conditioners, split-system heat pumps, single-package air conditioners, and single-package heat pumps. (Goodman, No. 53 at p. 4) UTC/Carrier proposed separate validation classes for the categories mentioned by Goodman, but also proposed that central air conditioners and heat pumps should include distinct validation classes for space-constrained air conditioners and space-constrained heat pumps. (UTC/Carrier, No. 56 at p. 2) United Cool Air stated that DOE did not properly address classification of space-constrained HVAC systems. (United Cool Air, No. 51 at p. 4, 13) United Cool Air's comments align with comments from Carrier that DOE should create a separate product class for space-constrained equipment.

In response, DOE notes that the proposed testing requirements in 429.16 require testing at least one individual model/combination within each basic model. Therefore, by default manufacturers would be testing all basic models from each product class in which they manufacture units.

b. Selection of Capacity Variations of a Basic Model for Validating an AEDM

Regarding selection of basic models for validating an AEDM, both Nordyne and Goodman agreed with DOE's proposal that the basic models selected for validating an AEDM must include the smallest capacity basic model as well as the largest capacity basic model (or a basic model within 25 percent of the largest capacity). (Nordyne, No. 55 at p. 2; Goodman, No. 53 at p. 2) Rheem, however, disagreed and stated that the requirement to test the smallest and largest capacity basic model was too restrictive and does not account for outliers or differences in technology across product classes. (Rheem, No. 59 at p. 4) Furthermore, Lennox noted that the manufacturer is most suited to determine which models should be used for validation and that requirements for particular capacities do not account for variation in product design and construction. (Lennox, No. 46 at p. 4)

DOE's intention when proposing to require that a manufacturer test both the smallest and largest capacity basic models within the product class with the highest sales volume was to ensure that the AEDM could accurately predict the efficiency of those products at the extremes of a manufacturer's product line. As variations in product design and construction across all capacities should be accounted for when testing all basic models, DOE withdraws the proposal regarding selecting the smallest and largest capacity basic models from the product class with the highest sales volume for testing for validation of the AEDM. DOE notes that in the proposed revisions to the determination of certified ratings, each basic model must be tested and an AEDM can only be used to certify other individual combinations that are part of the same basic model.

c. Use of the Highest Sales Volume Basic Model for Validating an AEDM

Many interested parties recommended that DOE continue to require that split-system manufacturers test each condensing unit they manufacture with the evaporator coil that is likely to have the largest volume of retail sales (*i.e.,* the highest sales volume combination, or HSVC) because the data resulting from these test combinations are critical to independent coil manufacturers (ICMs) in determining accurate ratings for their products since they must determine their ratings based on pairings with condensing units offered by other manufacturers. AHRI stated that DOE should retain requirements for testing based on the HSVC for central air conditioners and heat pumps. (AHRI, No. 61 at p. 2) UTC/Carrier agreed that DOE should allow split-systems to retain the HSVC process, as is required by current ARM regulations. (UTC/Carrier, No. 56 at p. 1) Lennox disagreed with removing the requirement for testing based on HSVC because the current AHRI certification program and independent coil manufacturing industry depend on this requirement, and the data from HSVC test results are used by independent coil manufacturers (ICMs) as the input to their ARM. (Lennox, No. 46 at p. 4)

Unico stated that DOE should maintain the current ARM requirements for central air conditioners and heat pumps because as an indoor coil manufacturer, Unico relies on the accuracy of the ratings published by the manufacturer of the outdoor unit and decreasing the accuracy of those ratings would increase their own risk of failure. Unico stressed that it was particularly important for DOE to allow manufacturers' rating methodology to rely on curve fit data, and specifically proposed that for validating an AEDM, matched system manufacturers should test at least the highest sales volume combination for each outdoor unit. (Unico, No. 54 at pp. 2, 4, and 6) Mortex Products, Inc. (Mortex) stated that in order for ICMs to rate indoor coils accurately using the ARM, the system manufacturer's HSVC data is necessary, and if HSVC data were no longer obtained from tests, but generated using an AEDM, the accuracy of the indoor coil ratings would be affected. (Mortex, No. 58 at p. 1)

DOE recognizes the concerns of stakeholders who commented that eliminating the requirement to test the HSVC for split-system products could increase the burden on ICMs. DOE does not intend to eliminate that requirement and notes that such requirement is proposed to be retained in this notice, as discussed in section III.A.3.a. However, DOE also proposes additional requirements for ICMs that are discussed in section III.B.5. DOE also notes that the ARM provisions in the current regulations do not clearly apply to ICMs, and most ICMs do not have DOE-approved ARMs.

DOE's proposal in the AEDM NOPR required re-validation when the HSVC changes. In response, Goodman stated that for split-system CACs and HPs, testing the highest or expected highest sales volume combination basic model would be appropriate as long as DOE does not require re-validation of the AEDM if another basic model subsequently becomes the highest sales volume combination. Determination of the highest volume basic model should be based on sales data of the prior year, or sales data or forecasts of the year of the AEDM's validation. (Goodman, No. 53 at p. 3) United Cool Air was also concerned that additional testing would be required if the highest selling basic model changed. (United Cool Air, No. 51 at p. 9)

In response to the concerns of Goodman and United Cool Air regarding re-validation if the HSVC changed, DOE agrees that re-validation should not be required if test data used to validate the AEDM was based on an expected HSVC that subsequently becomes a lower sales volume model and is not proposing such a requirement in this notice. DOE agrees with Goodman that determination of the highest volume basic model should be based on sales data of the prior year, sales data or forecasts of the year of the AEDM's validation, or other similar information. Selection of the highest volume basic model should reflect a good faith effort by the manufacturer to predict the combination most likely to result in the highest volume of sales. DOE notes that it may verify compliance with this HSVC testing requirement.

d. Requirements for Test Data Used for Validation

In AEDM NOPR, DOE did not propose requirements on the test data used for validation of an AEDM because any non-testing approaches to certifying central air conditioners and heat pumps via an ARM were to be approved by DOE prior to use. 77 FR 32043. However, if DOE adopts the current proposal to remove the pre-approval requirement, certified ratings generated using an AEDM would be unreliable without other requirements to validate the AEDM against actual test data. Therefore, DOE proposes in this notice to adopt requirements on test data similar to those used for validation for commercial HVAC and water heating equipment, as published in the AEDM final rule 78 FR 79579, 79584 (Dec. 31, 2013). Specifically, (1) for energy-efficiency metrics, the predicted efficiency using the AEDM may not be more than 3 percent greater than that determined through testing; (2) for energy consumption metrics, the predicted efficiency using the AEDM may not be more than 3 percent less than that determined through testing; and (3) the predicted efficiency or consumption for each individual combination calculated using the AEDM must comply with the applicable Federal energy conservation standard. Furthermore, the test results used to validate the AEDM must meet or exceed the applicable Federal standards, and the test must have been performed in accordance with the applicable DOE test procedure. If DOE has ordered the use of an alternative test method for a particular basic model through the issuance of a waiver, that is the applicable test procedure.

DOE proposes a validation tolerance of 3 percent because the variability in a manufacturer's lab and within a basic model should be more limited than lab-to-lab variability. DOE proposes tolerances for verification testing of 5 percent to account for added lab-to-lab variability.

5. Requirements for Independent Coil Manufacturers

In the AEDM NOPR, DOE did not propose a statistical sampling requirement for independent coil manufacturers (ICMs) that would be distinct from the sampling required to validate an AEDM for HVAC products. 77 FR at 32043. In response, Unico commented that ICMs should test coils of each fin-pattern, varying the number of rows, fin density, tube type, circuiting, and frontal area. (Unico, No. 54 at p. 4) Mortex stated that their ARMs are based on data from a ''matched system'' tested by an OUM. Mortex uses an ARM to simulate the performance of their own coil in a matched system by substituting the geometry of the indoor evaporator coil used by the manufacturer of the condensing unit with the geometry of their own coil. (Mortex, No. 58 at p. 1)

While DOE understands that ICMs currently use ratings from OUMs to predict the efficiency of their coil models, as discussed in section III.A.3.d, DOE is now proposing to require that ICMs test each of model of indoor units (*i.e.,* basic models) with the least efficient model of outdoor unit with which it will be paired. In order to validate an AEDM for split-systems rated by ICMs for other individual combinations within each basic model, DOE also proposes that ICMs must use the individual combinations the ICMs would be required to test under the proposed text in 10 CFR 429.16. DOE seeks comment on this proposal.

In regard to Unico's suggestion to test indoor units with coils of varying fin-patterns, DOE refers stakeholders to the definition of a basic model in section III.A.1, and particularly what constitutes the same model of indoor unit. DOE notes that the manner in which manufacturers apply the basic model provisions would impact what models of indoor units are required for testing.

6. AEDM Verification Testing

DOE may randomly select and test a single unit of a basic model pursuant to 10 CFR 429.104. This authority extends to all DOE covered products, including those certified using an AEDM. In the AEDM NOPR, DOE clarified that a selected unit would be tested using the applicable DOE test procedure at an independent, third-party laboratory accredited to the International Organization for Standardization (ISO)/International Electrotechnical Commission (IEC), ''General requirements for the competence of testing and calibration laboratories,'' ISO/IEC 17025:2005E. 77 FR 32038, 32057 (May 31, 2012).

In this notice, DOE proposes further verification testing methods. Specifically, DOE proposes that verification testing conducted by the DOE will be (1) on a retail unit or a unit provided by the manufacturer if a retail unit is not available, (2) at an independent, third-party testing facility or a manufacturer's facility upon DOE's request if the former is not capable of testing such a unit, and (3) conducted with no communication between the lab and the manufacturer without DOE authorization.

DOE also proposes clarification of requirements for determining that a model does not meet its certified rating, as proposed in the AEDM NOPR. Specifically, DOE proposes that an individual combination would be considered as having not met its certified rating if, even after applying the five percent tolerance between the test results and the rating as specified in the proposed 10 CFR 429.70(e)(5)(vi), the test results indicate the individual combination being tested is less efficient or consumes more energy than indicated by its certified rating. DOE notes that this approach will not penalize manufacturers for applying conservative ratings to their products. That is, if the test results indicate that the individual combination being tested is *more* efficient or consumes *less* energy than indicated by its certified rating, DOE would consider that individual combination to meet its certified rating. DOE seeks comment on whether this is a reasonable approach to identify an individual combination's failure to meet its certified rating.

In the AEDM NOPR, DOE also proposed the actions DOE would take in response to individual models that fail to meet their certified ratings. 77 FR at 32056. Many stakeholders submitted comments suggesting that DOE should determine the cause of the test failure prior to taking any additional action. UTC/Carrier commented that failure of a single unit test result could be a result of a defective unit and further urged DOE to define a process to contest test results from a third party lab. (UTC/Carrier, No. 56 at p. 2) JCI had a similar concern regarding potential errors in test set-up and proposed that DOE should work with the manufacturer to determine the root cause of the failure, performing additional testing if necessary. (JCI, No. 66 at p. 8) Rheem agreed with JCI that DOE should work with the manufacturer to determine whether the root cause is associated with test variability, AEDM model inaccuracy, or manufacturing variability. Rheem added that DOE should clarify what constitutes a ''failure'' as well as develop a detailed plan for selection, testing, evaluation, manufacturer notification, and resolution. (Rheem, No. 59 at p. 4) Lennox also agreed that DOE should not immediately require modification of an

AEDM without first finding the cause of the failure. (Lennox, No. 46 at pp. 4–5) Additionally, Ingersoll Rand requested that DOE allow for a dialogue with the manufacturer to ensure that the sample unit was not defective and that the test was set up correctly. (Ingersoll Rand, Public Meeting Transcript, No. 69 at p. 187) AHRI agreed that it would be valuable to specify particular steps manufacturers and DOE must take in the case of a test failure and incorporate a defective sample provision, and recommended that DOE provide data, a failure report, and other necessary information to the manufacturer for proper analysis of the test failure. (AHRI, No. 61 at pp. 6–7)

Unico and manufacturers of products other than HVAC suggested that DOE should not only share the data with the manufacturer, but also allow the manufacturer to review or witness testing done by a lab. This would allow for better understanding of potential discrepancies in test results and ensure that failure was not merely a result of variation in test set-up. (Unico, No. 54 at p. 4) AHRI and UTC/Carrier suggested that manufacturers should be allowed to participate in commissioning of their equipment prior to the assessment test since proper set-up is critical. AHRI added that manufacturers should have an opportunity to repair a unit, if defective, while it is in the assessment lab. (AHRI, No. 61 at pp. 6–7; Carrier, Public Meeting Transcript, No. 69 at p. 218) Further, UTC/Carrier urged DOE to specify an appeals process for tests that a manufacturer believes were tested with improper test set-up. (UTC/Carrier, Public Meeting Transcript, No. 69 at p. 195; UTC/Carrier, No. 56 at p. 3)

DOE agrees that determining the root cause of the failure to meet certified ratings is important; however, DOE stresses that this would be the manufacturer's responsibility. DOE is aware that in order to determine the cause of the failure, the manufacturer will need to review the data from DOE's testing. DOE therefore proposes that when an individual combination fails to meet certified ratings, DOE will provide to the manufacturer a test report that includes a description of test set-up, test conditions, and test results. DOE will provide the manufacturer with an opportunity to respond to the lab report by presenting all claims regarding testing validity, and if the manufacturer was not on-site for initial set-up, to purchase an additional unit from retail to test following the requirements in 429.110(a)(3). This process is designed to provide manufacturers the opportunity to raise concerns about the test set-up, taking into account various comments from stakeholders. DOE will consider any response offered by the manufacturer within a designated time frame before deciding upon the validity of the test results. Only after following these steps will the Department make a determination that the rating for the basic model is invalid and require the manufacturer to take subsequent action, as described in section III.B.7.

7. Failure To Meet Certified Ratings

In the AEDM NOPR, DOE proposed a method of determining whether a model meets its certified rating whereby the assessment test result would be compared to the certified rating for that model. If the test result was not within the tolerance in the proposed section 429.70(c), the model would be considered as having not met its certified rating. In this case DOE proposed to require that manufacturers re-validate the AEDM that was used to certify the product within 30 days of receiving the test report from the Department. DOE also proposed to require that manufacturers incorporate DOE's test data into the re-validation of the AEDM. If after inclusion of DOE's test data and re-validation, the AEDM-certified ratings change for any models, then the manufacturer would be required to re-rate and re-certify those models. The manufacturer would not be required to perform additional testing in this re-validation process unless the manufacturer finds it necessary in order to meet the requirements enumerated in the proposed section 429.70. 77 FR 32028, 32056.

A few stakeholders provided comments on the aforementioned proposals. Zero Zone commented that the failure of a single test unit to meet its certified rating should not automatically necessitate re-validation, but suggested that the manufacturer should decide on the appropriate course of action. (Zero Zone, No. 64 at p. 3) UTC/Carrier commented that DOE should not require re-validation based on a single unit's test result because the failure could be a result of a defective unit. (UTC/Carrier, No. 56 at p. 2) Lennox opposed DOE's proposal to require manufacturers to incorporate DOE test data into their AEDM if a model is determined not to meet its certified rating because they believe that DOE data may be erroneous and only the best available data should be used to validate an AEDM. (Lennox, No. 46 at p. 5) JCI stated that without additional information as to why a particular product failed a test, it is not reasonable to assume that all models rated with the AEDM must be re-rated. (JCI, No. 66 at pp. 9–10).

In consideration of the above mentioned comments, DOE proposed to allay concerns via the proposal in section III.B.6, which provides manufacturers an opportunity to review the data from DOE's testing and present claims regarding testing validity. Based on these comments, DOE also proposes an exception to re-validation of the AEDM in cases where the determination of an invalid rating for that basic model is the first for models certified with an AEDM. In such cases, the manufacturer must conduct additional testing and re-rate and re-certify the individual combinations within the basic model that were improperly rated using the AEDM.

DOE also proposes that if DOE has determined that a manufacturer made invalid ratings on individual combinations within two or more basic models rated using the manufacturer's AEDM within a 24 month period, the manufacturer must test the least efficient and most efficient combination within each basic model in addition to the combination specified in 429.16(a)(1)(ii). The twenty-four month period begins with a DOE determination that a rating is invalid through the process outlined above. If DOE has determined that a manufacturer made invalid ratings on more than four basic models rated using the manufacturer's AEDM within a 24-month period, the manufacturer may no longer use an AEDM.

Finally, DOE proposes additional requirements for manufacturers to regain the privilege of using an AEDM, including identifying the cause(s) for failure, taking corrective action, performing six new tests per basic model, and obtaining DOE authorization.

DOE created this proposal under the expectation that each manufacturer will use only a single AEDM for all central air conditioner and central air conditioning heat pumps. DOE requests comment on whether manufacturers would typically apply more than one AEDM and if they would, the differences between such AEDMs.

8. Action Following a Determination of Noncompliance

In the AEDM NOPR, DOE explained that if a model failed to meet the applicable Federal energy conservation standard during assessment testing, DOE may pursue enforcement testing pursuant to 10 CFR 429.110. DOE also stated that if an individual model was determined to be noncompliant, then all other individual models within that basic model would be considered noncompliant. This is consistent with

DOE's approach for all covered products. All other basic models rated with the AEDM would be unaffected pending additional investigation. Furthermore, DOE proposed that if a noncompliant model was used for validation of an AEDM, the AEDM must be re-validated within 30 days of notification, pursuant to requirements enumerated in 10 CFR 429.70. Notably, DOE did not propose that manufacturers must re-test basic models used to validate an AEDM when there is no determination of noncompliance. 77 FR 32056.

In response, JCI agreed that all AEDM-rated models should not be disqualified if one model is found out of compliance. (JCI, No. 66 at p. 9)

DOE reiterates that for central air conditioners and central air conditioning heat pumps, if an individual combination was determined to be noncompliant, then all other individual combinations within that basic model would be considered noncompliant. DOE is not proposing in this SNOPR that other basic models rated with the AEDM be considered non-compliant. However, DOE notes that an AEDM must be validated using test data for individual combinations that meet the current Federal energy conservation standards. Therefore, if a noncompliant model was used for validation of an AEDM, manufacturers would be expected to re-validate the AEDM in order to continue using it. The requirements for additional testing based on invalid ratings, as discussed in the previous section, may also apply.

C. Waiver Procedures

10 CFR 430.27(l) requires DOE to publish in the **Federal Register** a notice of proposed rulemaking to amend its regulations so as to eliminate any need for the continuation of waivers and as soon thereafter as practicable, DOE will publish a final rule in the **Federal Register**. As of the issuance date of this notice, a total of four waivers (and one interim waiver) for central air conditioner and heat pump products are active. They are detailed in the Table III.4, with the section reference to this notice included for discussion regarding DOE's proposed amended regulations and intention for subsequent waiver termination.

TABLE III.4—ACTIVE WAIVERS AND ACTIVE INTERIM WAIVERS

Air Conditioners and Heat Pumps, Consumer		
Scope	Decision & order	Termination
ECR International, Inc., Multi-zone Unitary Small Air Conditioners and Heat Pumps	(Petition & Interim Waiver, 78 FR 47681, 8/6/2013).	III.C.2
Daikin AC (Americas), Inc., Heat Pump & Water Heater Combination	76 FR 11438, 3/2/2011	III.C.1
Daikin AC (Americas), Inc., Heat Pump & Water Heater Combination	75 FR 34731, 6/18/2010	III.C.1
Hallowell International, Triple-Capacity Northern Heat Pumps	75 FR 6013, 2/5/2010	III.C.4
Cascade Group, LLC, Multi-blower Air-Conditioning and Heating Equipment	73 FR 50787, 8/28/2008	III.C.3

DOE notes that four waivers previously associated with both commercial equipment and consumer products, as listed in Table III.3, were terminated for consumer products as of the October 22, 2007 Final Rule (72 FR 59906, 59911) and for commercial equipment as of the May 16, 2012 Final Rule (77 FR 28928, 28936). In this SNOPR, DOE reaffirms that these waivers have been terminated for consumer products and that the products in question can be tested using the current and proposed test procedure for central air conditioners and heat pumps.

TABLE III.5—TERMINATED WAIVERS

Scope	Decision & order
Daikin U.S. Corporation, Multi-split Heat Pumps and Heat Recovery Systems	73 FR 39680, 7/10/2008.
Mitsubishi Electric and Electronics USA, Inc., Variable Refrigerant Flow Zoning Air Conditioners and Heat Pumps.	72 FR 17528, 4/9/2007.
Fujitsu General Limited, Multi-split Products	72 FR 71383, 12/17/2007.
Samsung Air Conditioning, Multi-split Products	72 FR 71387, 12/17/2007.

1. Termination of Waivers Pertaining to Air-to-Water Heat Pump Products With Integrated Domestic Water Heating

DOE has granted two waivers to Daikin Altherma for the air-to-water heat pump with integrated domestic water heating; one on June 18, 2010 and a second on March 2, 2011. 75 FR 34731 and 76 FR 11438. As described in Daikin's petitions, the Daikin Altherma system consists of an air-to-water heat pump that provides hydronic space heating and cooling as well as domestic hot water functions. It operates either as a split system with the compressor unit outdoors and the hydronic components in an indoor unit, or as a single-package configuration in which all system components are combined in a single outdoor unit. In both the single-package and the split-system configurations, the system can include a domestic hot water supply tank that is located indoors. These waivers were granted on the grounds that the existing DOE test procedure contained in Appendix M to Subpart B of 10 CFR part 430 addresses only air-to-air heat pumps and does not include any provisions to account for the operational characteristics of an air-to-water heat pump, or any central air-conditioning heat pump with an integrated domestic hot water component.

According to the definition set forth in EPCA and 10 CFR 430.2, a central air conditioner is a product, other than a packaged terminal air conditioner, which is powered by single phase electric current, air cooled, rated below 65,000 Btu per hour, not contained within the same cabinet as a furnace, the rated capacity of which is above 225,000 Btu per hour, and is a heat pump or a cooling unit only. (42 U.S.C. 6291(21)) The heat pump definition in EPCA and 10 CFR 430.2 requires that a heat pump utilize a refrigerant-to-outdoor air heat exchanger, effectively excluding heat pump products classified as air-to-water. (42 U.S.C. 6291(24)) In addition, because the definition of a central air conditioner, which also applies to heat pumps, requires products to be ''air cooled,'' products that rely exclusively on refrigerant-to-water heat exchange on the indoor side are effectively excluded from the definition of, and the existing efficiency standards for, central air conditioners and heat pumps.

Based upon the description in the waiver petitions for the Daikin Altherma air-to-water heat pumps with integrated domestic water heater, DOE has determined that these products rely exclusively on refrigerant-to-water heat exchange on the indoor side, and thus would not be subject to the central air conditioner or heat pump standards and would not be required to be tested and rated for the purpose of compliance with DOE standards for central air conditioners or heat pumps. Thus, if this interpretation is adopted, these waivers would terminate on the effective date of a notice finalizing the proposals in this notice.

2. Termination of Waivers Pertaining to Multi-Circuit Products

DOE granted ECR International (ECR) an interim waiver on August 6, 2013, for its line of Enviromaster International (EMI) products. 78 FR 47681. ECR describes in its petitions that its multi-zone air conditioners and heat pumps each comprise a single outdoor unit combined with two or more indoor units, which each comprise a refrigeration circuit, a single air handler, a single control circuit, and an expansion valve, intended for independent zone-conditioning. The outdoor unit contains one fixed-speed compressor for each refrigeration circuit; all zones utilize the same condenser fan and defrost procedures but refrigerant is not mixed among the zones. 78 FR at 47686. These products are similar to multiple-split (or multi-split) air conditioners or heat pumps, which are defined and covered by current test procedure (Appendix M to Subpart B of 10 CFR part 430). However, they are distinct from, and therefore not classified as, multi-split products due to differences in refrigerant circuitry. The separate refrigeration circuits of the ECR product line are not amenable to the test procedures for multi-split systems, specifically the procedures calling for operation at different levels of compressor speed or staging, because the individual compressors are not necessarily variable-speed. Hence, alternative procedures have been developed, as described in the interim waiver. DOE proposes to address products such as the ECR product line in the DOE test procedure. DOE also proposes to define such a product as a ''multi-circuit air conditioner or heat pump'' and provide testing requirements for such for such products at 10 CFR 429.16(a)(1)(ii)(A).

For the duration of the interim waiver period, either until 180 days after the publication of the interim waiver (the interim waiver period) or until DOE issued its determination on the petition for waiver, whichever occurred earlier, DOE granted ECR permission to use the proposed alternative test procedure to test and rate its multi-circuit products. 78 FR 47681, 47682 (Aug. 6, 2013). The requirements in the alternative test procedure comprise methods to establish air volume rate, procedures for testing, and adjustments to equations used to calculate SEER and HSPF. Following publication of the Notice of Grant of Interim Waiver, DOE received no comments regarding this alternative test procedure. After the interim waiver period, DOE did not issue a final decision and order on ECR's petition for waiver, therefore, the interim waiver will terminate upon the publication of a test procedure final rule for central air conditioners and heat pumps, and the alternative test procedure included therein shall cease from being applicable to testing and rating ECR's multi-circuit products and multi-circuit products in general, absent amendments regarding provisions for testing such products. Therefore, DOE proposes in this notice testing requirements for manufacturers who wish to certify multi-circuit products.

According to Appendix M to Subpart B of 10 CFR part 430, Section 2.4.1b, systems with multiple indoor coils are tested in a manner where each indoor unit is outfitted with an outlet plenum connecting to a common duct so that each indoor coil ultimately connects to an airflow measuring apparatus.[10] In testing a multi-circuit system in this manner, the data collection, performance measurement, and reporting is done only on the system level. ECR took issue with this, citing inadequate data accountability, and thus argued in its petition for waiver to individually test each indoor unit. *Id.* Current test procedures for systems with multiple indoor coils, however, produce ratings that are repeatable and accurate even though monitoring of all indoor units are not required by regulation, or common industry practice. DOE also notes that the common duct testing approach has been adopted by industry standards and is an accepted method for testing systems having multiple indoor units. ECR's petition did not identify specific differences between the indoor units of its new product line and the indoor units of multi-splits that would make the common-duct approach unsuitable for its products. Further, the interim waiver approach of using multiple airflow measuring devices, one for each indoor unit, represents unnecessary test burden. Therefore, DOE proposes to adopt for multi-circuit products the same common duct testing approach used for testing multi-split products.

The alternative test procedure in the interim waiver calls for separate measurement of performance for each indoor unit for each required test condition, and requires that all indoor units be operating during each of these separate measurements. The overall system performance for the given test condition is calculated by summing the capacities and power inputs measured for all of the indoor units and adding to the power input sum the average of the power measurements made for outdoor unit for the set of tests. *Id.* In contrast, DOE's current proposal involves use of the common duct to measure the full system capacity, thus allowing use of a single test for each operating condition. DOE requests comment on whether this method will yield accurate results that are representative of the true performance of these systems.

3. Termination of Waiver and Clarification of the Test Procedure Pertaining to Multi-Blower Products

On August 28, 2008, DOE published a decision and order granting Cascade Group, LLC a waiver from the Central Air Conditioner and Heat Pump Test Procedure for its line of multi-blower indoor units that may be combined with one single-speed heat pump outdoor unit, one two-capacity heat pump outdoor unit, or two separate single-speed heat pump outdoor units. 73 FR 50787, 50787–97. DOE proposed revisions to the test procedure in the June 2010 NOPR to accommodate the certification testing of such products. 75 FR 31237. NEEA responded in the subsequent public comment period, recommending DOE defer action on test procedure changes until such a product is actually being tested, certified and sold. (NEEA, No. 7 at pp. 4–5). Mitsubishi recommended DOE either use AHRI Standard 1230–2010 to rate such a product or does not amend the

[10] When the indoor units are installed in separate indoor chambers for the test, the test procedure allows common ducting to a separate airflow measuring apparatus for each indoor chamber.

test procedure to allow coverage of such a product. (Mitsubishi, No. 12 at p. 2).

DOE notes that AHRI Standard 1230–2010, which provides testing procedures for products with variable speed or multi-capacity compressors, may not be suitable for testing the subject products, which are equipped with single-speed compressors; however, the test procedure, as proposed in the June 2010 NOPR enables testing of such products. DOE therefore retains its proposal in the June 2010 NOPR to adopt that test procedure, except for the following revisions.

The proposal in the June 2010 NOPR amended Appendix M to Subpart B of 10 CFR part 430 with language in sections 3.1.4.1.1e and 3.1.4.2e that suggested that test setup information may be obtained directly from manufacturers. DOE is revising that proposal to eliminate the need for communication between third-party test laboratories and manufacturers, such that the test setup is conducted based on information found in the installation manuals included with the unit by the manufacturer. DOE is proposing that much of that information be provided to DOE as part of certification reporting. These proposed modifications regarding test setup can be found in section 3.1.4.1.1d and 3.1.4.2e of the proposed Appendix M in this notice. DOE requests comment on its proposals for multi-blower products, including whether individual adjustments of each blower are appropriate and whether external static pressures measured for individual tests may be different.

Because the proposed test procedure amendments would allow testing of Cascade Group, LLC's line of multi-blower products, DOE proposes to terminate the waiver currently in effect for those multi-blower products effective 180 days after publication of the test procedure final rule.

4. Termination of Waiver Pertaining to Triple-Capacity, Northern Heat Pump Products

On February 5, 2010, DOE granted Hallowell International a waiver from the DOE Central Air Conditioner and Heat Pump Test Procedure for its line of boosted compression heat pumps. 75 FR 6014, 6014–18. DOE proposed revisions to its test procedures in the June 2010 NOPR to accommodate the certification testing of such products. 75 FR 31223, 31238 (June 2, 2010). NEEA expressed support for DOE's proposal in the subsequent public comment period but urged DOE to ensure that the northern climate test procedure can be used by variable speed systems that can meet the appropriate test conditions, and that the procedures can accurately assess the performance of these systems relative to more conventional ones. (NEEA, No. 7 at p. 5). NEEA also urged DOE to require publishing of Region V ratings for heat pumps. Mitsubishi supported DOE's proposed changes to cover triple-capacity, northern heat pumps but requested that DOE reevaluate the testing of inverter-driven compressor systems to permit better demonstration of the system's capabilities at heating at low ambient conditions. (Mitsubishi, No. 12 at p. 3).

DOE believes that the test procedure as proposed in the June 2010 NOPR, along with the proposed revisions to the test procedure for heating tests conducted on units equipped with variable-speed compressors, as discussed in section III.H.5, would produce performance that represents an average period of use of such products. Because the proposed test procedure amendments would allow testing of Hallowell International's line of triple-capacity, northern heat pump products, DOE proposes to terminate the waiver currently in effect for those products effective 180 days after publication of the test procedure final rule.

D. Measurement of Off Mode Power Consumption

In the June 2010 NOPR, DOE proposed a first draft of testing procedures and calculations for off mode power consumption. 75 FR 31223, 31238 (June 2, 2010). In the following April 2011 SNOPR, DOE proposed a second draft, revising said testing procedures and calculations based on stakeholder-identified issues and changes to the test procedure proposals in the 2010 June NOPR and on DOE-conducted laboratory testing. 76 FR 18105, 18111 (April 1, 2011). In the October 2011 SNOPR, DOE proposed a third draft, further revising the testing procedures and calculations for off mode power consumption based primarily on stakeholder comments regarding burden of test as received during the April 2011 SNOPR comment period. 76 FR 65616, 65618–22 (Oct. 24, 2011). From the original and extended comment period of the October 2011 SNOPR DOE received stakeholder comments, which are the basis of DOE's proposed fourth draft in this notice, further revising testing procedures and calculations for off mode power consumption. None of the proposals listed in this section impact the energy conservation standard.

1. Test Temperatures

In the October 2011 SNOPR, DOE proposed to base the off mode power consumption rating ($P_{W,OFF}$) on an average of wattages *P1* and *P2,* which would be recorded at the different outdoor ambient temperatures of 82 °F and 57 °F, respectively. DOE intended that, for systems with crankcase heater controls, the measurement at the higher ambient temperature would measure the off mode contribution that was more representative of the shoulder seasons. The lower measurement was intended to represent off mode power use for an air conditioner during the heating season. 76 FR at 65621.

In response to the October 2011 SNOPR, a joint comment from Pacific Gas and Electric and Southern California Edison, hereafter referred to as the California State Investor Owned Utilities (CA IOUs), and a joint comment from the American Council for an Energy-Efficient Economy (ACEEE) and Appliance Standards Awareness Program (ASAP) expressed concern that the 57 °F test point could create a loophole wherein a crankcase heater could be designed to turn on just below 57 °F and result in an underestimation of the system's energy consumption. The off mode power consumption would be underestimated because the energy consumption of the crankcase heater would not be included in either *P1* or *P2.* (CA IOUs, No. 33 at p. 2; ACEEE and ASAP, No. 34 at p. 2) A joint comment from the Northwest Energy Efficiency Alliance (NEEA) and the Northwest Power and Conservation Council (NPCC), hereafter referred to as the Joint Efficiency Advocates, also disputed DOE's proposal to test units at two fixed temperatures and disagreed with DOE's contention that the proposed *P2* test temperature (57 °F) is sufficiently low that the crankcase heater would be energized. (Joint Efficiency Advocates, No. 35 at p. 3)

Both the CA IOUs and the Joint Efficiency Advocates proposed that DOE require manufacturers to specify the temperature at which the crankcase heater turns on and off, and then to run one off mode test 3–5 °F below the point at which the crankcase heater turns on (''on'' set point temperature) and the other off mode test 3–5 °F above the temperature at which the crankcase heater turns off (''off'' set point temperature). (CA IOUs, No. 33 at p. 2; Joint Efficiency Advocates, No. 35 at p. 3) However, the Joint Efficiency Advocates only proposed this rating method for constant wattage crankcase heaters. (Joint Efficiency Advocates, No. 35 at p. 3) The Joint Efficiency Advocates stated that two measurements are insufficient for systems that have a heater with wattage that varies according to temperature and suggested that the crankcase heater power for systems with variable wattage be tested at three temperatures. Specifically, the Joint Efficiency Advocates recommended testing at 3–5 °F below the ''on'' set point temperature, at 47 °F, and at 17 °F. (Joint Efficiency Advocates, No. 35 at p. 4) The Joint Efficiency Advocates additionally recommended that systems with temperature-controlled crankcase heaters should be tested for off mode power use when cold (*i.e.,* before the system is run). (Joint Efficiency Advocates, No. 35 at p. 4)

In the December 2011 extension notice for comments on the October 2011 SNOPR, DOE requested comment on the CA IOUs' suggestion that the test procedure should measure *P1* at a temperature that is 3–5 °F above the manufacturer's reported ''off'' set point and measure *P2* at a temperature that is 3–5 °F lower than the ''on'' set point. 76 FR 79135 (Dec. 21, 2011). The Joint Efficiency Advocates commented in support of the CA IOU proposal. (Joint Efficiency Advocates, No. 43 at p. 2) However, they also reiterated that crankcase heater power for systems with variable wattage should be tested at three temperatures, namely, 3–5 °F below the ''on'' set point temperature, 47 °F, and 17 °F. (Joint Efficiency Advocates, No. 43 at p. 2)

AHRI commented that DOE should modify the test procedure by having up to three rating temperatures, depending on the manufacturer control protocol. The first test would be conducted at 72 °F immediately after the B, C, or D test to verify whether the crankcase heater is on. The second test would be conducted at 5 °F below the temperature at which the manufacturer specifies the crankcase heater turns on. The third test would be conducted at 5 °F below the temperature at which the crankcase heater turns off and would only apply to air conditioners with crankcase heater controls that turn off the crankcase heater during winter. AHRI commented that it could accept the CA IOUs proposal to test at 3–5 °F below the heater turn-on temperature and at 3–5 °F above the heater turn-off temperature if DOE did not accept AHRI's proposal. (AHRI, No. 41 at p. 2) Goodman commented in support of AHRI's recommendation. (Goodman, No. 42 at p. 1)

Many of the commenters' recommended changes are reflected in this proposed rule. DOE proposes to require manufacturers to include in certification reports the temperatures at which the crankcase heater is designed to turn on and turn off for the heating season, if applicable. These temperatures are used in the proposed tests described in the following paragraphs.

DOE proposes to replace the off mode test at 82 °F with a test at 72±2 °F and replace the off mode test at 57 °F with a test at a temperature which is 5±2 °F below a manufacturer-specified turn-on temperature. This approach maintains the intent of the off mode power consumption rating ($P_{W,OFF}$) as a representation of the off mode power consumption for the shoulder and heating seasons, addresses AHRI's proposed modification of the test procedure, and addresses ACEEE and ASAP's concerns regarding the potential for a loophole at the 57 °F test point.

DOE does not propose to adopt an additional test point at a temperature of 17 °F, as recommended by the stakeholders; (Efficiency Advocates, No. 35 at p. 4; AHRI, No. 41 at p. 2) at a temperature 5 °F below the temperature at which the crankcase heater turns off, as recommended by AHRI; (AHRI, No. 41 at p. 2) or at a temperature 3–5 °F above the heater turn-off temperature, as recommended by the CA IOUs and the Joint Efficiency Advocates. (CA IOUs, No. 33 at p. 2; Joint Efficiency Advocates, No. 35 at p. 3) Manufacturer literature provides data on variable wattage crankcase heaters, otherwise known as self-regulating crankcase heaters, which show that power input for such heaters is a linear function of outdoor ambient temperature (*i.e.,* the input power can be represented with insignificant error as a constant times the outdoor ambient temperature plus another constant). As such, DOE maintains that two test points are adequate for characterizing the off mode power consumption for self-regulating crankcase heaters by establishing a linear fit from the two test outputs. DOE also believes that one of the two test points is adequate for characterizing the off mode power consumption for constant wattage crankcase. DOE does not believe that the additional accuracy gained from additional test points merits the additional test burden. The modifications in this proposal should help to minimize the test burden while maintaining the accuracy of off mode power ratings. DOE requests comments on these proposals.

2. Calculation and Weighting of P1 and P2

Stakeholders submitted comments discussing the most appropriate way to weight *P1* and *P2* in order to measure the total off mode power draw. In the October 2011 SNOPR, DOE proposed to require calculation of the total off mode power consumption based upon an arithmetic mean of the power readings *P1* and *P2*. 76 FR 65616, 65621 (Oct. 24, 2011).

The Joint Efficiency Advocates opposed the DOE's proposal in the October 2011 SNOPR. (Joint Efficiency Advocates, No. 35 at p. 4) The CA IOUs proposed to weight *P1* by 25% and *P2* by 75%, because this weighting would be more representative of actual heater operation than equally weighting *P1* and *P2*. (Joint Utilities, No. 33 at p. 2) Conversely, Goodman and AHRI opposed the CA IOUs' proposal because there was inadequate data available to support weighting *P1* by 25% and *P2* by 75%. Further, Goodman and AHRI stated that the CA IOUs' proposal would not fairly differentiate between products with different crankcase heater turn-on and turn-off temperatures. A unit with a lower turn-on and a higher turn-off temperature would consume less overall energy, but a manufacturer would have no incentive to use the lowest possible temperatures because the rating would not change. (Goodman, No. 42 at p. 2; AHRI, No. 41 at p. 3)

AHRI, Goodman, and the Joint Efficiency Advocates suggested that average power should be calculated by weighting the off mode hours using a bin method, in a manner consistent with the calculations of seasonal active-mode. (AHRI, No. 41 at p.3; Goodman, No. 42 at p. 1; Joint Efficiency Advocates, No. 35 at p. 5; Joint Efficiency Advocates, No. 43 at p. 3) AHRI provided a detailed methodology for calculating the off mode power rating in an excel spreadsheet submitted with its written comments. (AHRI, No. 41 at p. 2) AHRI introduced bin calculations to calculate seasonal *P1* and *P2* values, including recommending a different set of fractional bin-hours for the shoulder season. Goodman supported AHRI's proposal. (Goodman, No. 42 at p. 1) However, AHRI and Goodman commented that if DOE did not accept AHRI's proposed calculation, DOE should implement a 50% weighting of *P1* and *P2* as proposed in the October 2011 SNOPR. (AHRI, No. 41 at p. 3; Goodman, No. 42 at p. 2)

After reviewing the Off-Mode Power excel spreadsheet from AHRI and the comments received from stakeholders, DOE retains its proposal from the October 2011 SNOPR, which gives equal weighting to *P1* and *P2* for the calculation of the off mode power rating ($P_{W,OFF}$). 76 FR 65616, 65620 (Oct. 24, 2011). Comments from the stakeholders did not provide any data that support selection of specific weights for *P1* and *P2*. Therefore DOE cannot confirm that AHRI's suggested temperature bin-hour calculation method is representative of the off mode power for the shoulder and heating seasons.

3. Products With Large, Multiple or Modulated Compressors

In the October 2011 SNOPR, DOE proposed to adjust the measured off mode power draw for systems with multiple compressors and apply a scaling factor to systems larger than 3 tons. 76 FR at 65621–22. The CA IOUs and the Joint Efficiency Advocates disagreed with DOE's approach. (Joint Efficiency Advocates, No. 35 at p. 5; CA IOUs, No. 33 at p. 2; CA IOUs, No. 40 at p. 1) The CA IOUs commented that adjusting the off mode power draw for systems with multiple compressors and applying a scaling factor to extra-large systems would not represent actual off mode power consumption and recommended that DOE not reduce the calculated off mode power based on the number of compressors. (CA IOUs, No. 33 at p. 2)

AHRI and Goodman disagreed with CA IOUs' suggestion to eliminate the adjustment based on the number of compressors as it may potentially discourage the development and use of higher efficiency products. (AHRI, No. 36 at p. 2; AHRI, No. 41 at p. 3; Goodman, No. 42 at p. 2) Moreover, AHRI requested that a similar credit be given to products using modulating compressors due to the typical application where a higher charge is a requirement of the high efficiency systems. (AHRI, No. 36 at p. 2) AHRI also disagreed with the idea of eliminating the scaling factor proposed for rating larger compressors. (AHRI, No. 41 at p. 3) Lastly, AHRI recommended that the measurement of the off mode power consumption and of the low-voltage power from the controls for the shoulder season be divided by the number of compressors or number of discrete controls, as is currently done for the measurements in the heating season. (AHRI, No. 36 at p. 2)

DOE is aware that some systems may require higher wattage heaters to protect system reliability. Specifically, larger-capacity units may have larger-capacity compressors, which (at a high level) have larger shells with more surface area that can cool them off, thus requiring more heater wattage. They may also have more lubricant, thus it takes more heater wattage to heat up the lubricant to acceptable level (for example after a power outage) before restart. To avoid situations that force manufacturers to potentially compromise the reliability of their systems by downsizing crankcase heater wattages to meet off mode power requirements, DOE proposes to retain the recommended scaling factor for large capacity systems.

Additionally, DOE does not want to penalize manufacturers of multiple compressor systems, which are highly efficient but also need to employ larger crankcase heaters for safe and reliable operation given the additional shell surface area and lubricant. Therefore, DOE agrees with AHRI's recommendation and proposes that the off mode power consumption for the shoulder season and heating season, as well as the low-voltage power from the controls, be divided by the number of compressors to determine off mode power consumption on a per-compressor basis.

The direct final rule also did not consider the possible applicability of the new off mode standards to high-efficiency air conditioners and heat pumps that achieve high SEER and HSPF ratings using both large heat exchangers and compressor modulation. The correlation of the use of modulating compressors with high refrigerant charge, which is indicative of larger heat exchangers, was mentioned in the AHRI comment. (AHRI, No. 41 at p. 3) DOE does not want to penalize manufacturers for selling high efficiency units. Therefore, DOE agrees with AHRI's recommendation to apply a multiplier to the calculation of the per-compressor off mode power for the shoulder season and heating season for modulated compressors, but proposes a multiplier of 1.5, as modulating technology is not a multiple-compressor technology (with a multiplier of 2+). DOE requests comment on the multiplier of 1.5 for calculating the shoulder season and heating season per-compressor off mode power for modulated compressors.

4. Procedure for Measuring Low-Voltage Component Power

In the October 2011 SNOPR, DOE proposed to measure the power from low-voltage components, P_x, after each of the two tests conducted at *T1* and *T2*. 76 FR 65628–30. Although this would ensure that the low-voltage power consumption at each temperature test point would be removed from the respective off mode power consumption, AHRI expressed concern about excessive manufacturer test burden. AHRI recommended that P_x not be re-measured, as it does not change with temperature and not re-measuring it avoids automatic and unwanted operation of the crankcase heater. (AHRI, No. 36 at p. 3)

DOE agrees with AHRI that the low voltage power consumption does not change with temperature, although slight and insignificant fluctuations in the low-voltage power may occur due to the relationship of resistivity and conductivity to temperature. Moreover, DOE does not believe that these fluctuations outweigh the test burden added from reconfiguring the system for measuring the low-voltage power a second time. As such, the test procedure has been revised so that the measurement of P_x is not repeated. DOE proposes to require that the measurement of P_x occur after the measurement of the heating season total off mode power, $P2_x$, which reduces test burden by requiring a single disconnection of the low-voltage wires.

Additionally, DOE is aware that many control types exist for crankcase heaters, and certain control methodologies cycle the crankcase heater on and off during the 5-minute interval during which P_x is being measured. Since P_x measures the power of functioning components, only non-zero values of measured power should be used in the calculations. DOE has therefore included in the proposed test procedure a requirement to record only non-zero data for the determination of P_x.

5. Revision of Off-Mode Power Consumption Equations

As a result of the proposed revisions to the test procedure discussed in section III.D.3 and section III.D.4, the equations from the October 2011 SNOPR for determining *P1* for crankcase heaters without controls and for determining *P2* for crankcase heaters with controls are simplified in this proposal. The revised equations are:

$$P1 = \frac{P1_x - P1_D}{no\ of\ compressors} + P1_D,$$

and

$$P2 = \frac{P2_x - P1_x}{no\ of\ compressors} + P1,$$

respectively. 76 FR 65616, 65629–30 (Oct. 24, 2011). $P1_D$ is the off mode power with the crankcase heater disconnected, which is equal to the low-voltage power, P_x. $P1_x$ is the shoulder-season total off mode power, $P2_x$ is the heating-season total off mode power, *P1* is the per-compressor shoulder-season total off mode power, and *P2* is the per-compressor heating-season total off mode power.

The proposed revisions to section III.D.3 (per-compressor representation of *P1*) and section III.D.4 (temperature-independence of P_x) of this notice allow for the simplification of the equations that would be used to calculate power for crankcase heaters with or without controls. The two proposed revisions are based on the following three premises: (1) The representations of *P1* and *P2* would both be calculated on a per-compressor basis (as discussed in section III.D.3); (2) The value of P_x would not vary with temperature and would thus be the same at *T1* as it is at *T2* (as discussed in section III.D.4); (3) The following would apply under the proposed method: $P2 = P2_x - P_{xi}$ $P1 = P1_x - P_x$. (As discussed in the October 2011 SNOPR at 76 FR 65629). Applying the three premises to the equations for *P1* and *P2* from the October 2011 SNOPR results in the following simplification:

$$P1 = \frac{P1_x}{no\ of\ compressors}$$

and

$$P2 = \frac{P2_x}{no\ of\ compressors}$$

6. Off-Mode Power Consumption for Split Systems

AHRI commented that language in the October 2011 SNOPR may have caused stakeholders to infer that every blower coil indoor unit combination and every coil-only indoor unit combination must be tested to determine off mode power consumption. (AHRI, No. 36 at p. 2) AHRI recommended that DOE only require testing of the outdoor condensing unit for the highest sale-volume combination of each basic model to determine the off mode power consumption and allow use of an alternative rating method (ARM) to reduce test burden. (AHRI, No. 36 at p. 2)

In this SNOPR, DOE proposes generally that each basic model would be required to have all applicable represented values (SEER, EER, HSPF, or $P_{W,OFF}$) of a specified individual combination determined through testing. The other individual combinations within each basic model may be tested or rated using AEDMs. As such, only one individual combination within each basic model would be required to be tested to determine off mode power consumption.

Additionally, upon reviewing the test procedures of furnace products, DOE found that the indoor off mode power in coil-only split-systems (that would be installed in the field with a furnace) was accounted for in the furnace test methodology. The indoor power for coil-only systems consists of the controls for the electronic expansion valve drawing power from control boards either indoor in the furnace assembly or outdoor in the condensing unit. To avoid double-counting indoor off mode power between two products, DOE proposes to exclude measurement of the low-voltage power if the controls for the indoor components receive power from a control board dedicated to a furnace assembly. For blower coil indoor units in which the air mover is a furnace, the same proposal applies. For blower coil indoor units in which the designated air mover is not a furnace, since the off mode power of the indoor components is not accounted for in any other product's test methodology, DOE proposes to adopt language to include the low-voltage power from the indoor unit when measuring off mode power consumption for blower coil systems.

7. DOE requests comment on its proposal to exclude low-voltage power from the indoor unit when measuring off mode power consumption for coil-only split-system air conditioners and for blower coil split system air conditioners for which the air mover is a furnace. DOE also requests comment on its proposal to include the low-voltage power from the indoor unit when measuring off mode power consumption for blower coil split-system air conditioners with an indoor blower housed with the coil and for heat pumps.

Time Delay Credit

To provide an additional incentive for manufacturers to reduce energy consumption, AHRI and Goodman suggested adding a credit for crankcase heaters that incorporate a time delay before turning on during the shoulder season. (AHRI, No. 41 at p. 2; Goodman, No. 42 at p. 1) The off mode period in the calculation methodology designates extended periods during which the unit is idle. DOE proposes to adopt an energy consumption credit that would be proportional to the duration of the delay, as implemented in the calculation of the off mode energy consumption for the shoulder season, *E1,* in the proposed off mode test procedure. DOE is also proposing, for products in which a time delay relay is installed but the duration of the delay is not specified in the manufacturer's installation instructions shipped with the product or in the certification report, a default period of non-operation of 15 minutes out of every hour, resulting in a 25% savings in shoulder-season off mode energy consumption. To reduce potential instances of the misuse of this incentive, DOE also proposes requiring manufacturers to report the duration of the crankcase heater time delay for the shoulder season and heating season that was used during certification testing. DOE is also considering adding a verification method to 429.134. DOE requests comment on the proposed method for accounting for the use of a time delay, the default period of non-operation, and the possibility of a verification test for length of time delay.

8. Test Metric for Off-Mode Power Consumption

The June 2010 NOPR proposed a test procedure that would measure the average off mode power consumption, $P_{W,OFF}$, of a central air conditioner or heat pump. 75 FR 31238–39. Additionally, the amended energy conservation standards for central air conditioners and heat pumps in the June 2011 DFR included standards for off mode power consumption that were defined in terms of $P_{W,OFF}$. 76 FR 37408, 37411. The Joint Efficiency Advocates and the CA IOUs commented that the test procedure should calculate energy use and not average power draw. (Joint Efficiency Advocates, No. 43 at p. 3; CA IOUs, No. 33 at p. 1) The CA IOUs stated that DOE should measure energy use because control systems on the crankcase heater can save power by reducing run time, which is not captured by a power-draw metric. (CA IOUs, No. 33 at p. 1) The Joint Efficiency Advocates also requested that any standards promulgated should be based on energy use. (Joint Efficiency Advocates, No. 43 at p. 2) To maintain consistency with the off mode standards, the test procedure must measure off mode power consumption rather than energy use. However, DOE recognizes that adopting a bin-based approach to calculate $P_{W,OFF}$ does not provide a final off mode value that is indicative of actual power consumption. DOE is aware of alternative methods to determine a power rating. However, in consideration of testing burden, DOE proposes to implement a method of calculation that would closely approximate the actual off mode power consumption via a simple average of the shoulder and heating season measured values. Although this metric will not directly translate into instantaneous off mode power consumption, annual energy costs, or national energy consumption, it does provide a standardized method of calculation that is representative of average off mode power consumption. The average off mode power calculation can be used for ranking models based on their performance when idle, as well as for comparing a model's performance to the DOE standards.

DOE is aware that measurement of energy use for a specified test period would enable calculation of annual energy consumption and operating costs and, on a larger scale, national energy savings and national energy consumption solely due to equipment idling. Therefore, DOE has proposed optional equations that a manufacturer could use to determine the actual off mode energy consumption, based on the hours of off mode operation and off mode power for the shoulder and heating seasons, to provide additional information to consumers. Energy consumption would be specific to a single location and its unique set of cooling, heating, and shoulder season hours. DOE requests comment on such equations.

9. Impacts on Product Reliability

AHRI and Bristol Compressors submitted comments expressing concern that regulating crankcase heater energy consumption could have a negative impact on product reliability (AHRI, No. 41 at pp. 1–2; Bristol, No. 39 at p. 1) Bristol Compressors remarked that simply turning the crankcase heater off at specific outdoor ambient temperatures would expose many compressors to conditions that would reduce the effective life of the product or, at worst, cause immediate failure. Bristol requested that DOE allow additional time for research on technological options that could save energy in a manner similar to controls based on outdoor ambient temperature, but that do not impact the reliability of the product. (Bristol, No. 39 at p. 1) AHRI asked DOE to conduct further research to determine if regulating crankcase heater energy consumption has a negative impact on product reliability and to consider additional amendments to the test procedure, if deemed necessary, to limit impacts on product reliability. (AHRI, No. 41 at p. 2)

DOE expects that this proposed off mode test method will allow manufacturers to meet the June 2011 off mode standards without causing a shift in the reliability of the overall market of central air conditioners and heat pumps. DOE requests comments on the issue of compressor reliability as it relates to crankcase heater operation in light of the test method proposed in this rule.

10. Representative Measurement of Energy Use

In the April 2011 SNOPR DOE proposed modifications to the laboratory tests and algorithms for determining the off mode power of central air conditioners and heat pumps. 76 FR 18105, 18107–09 (April 1, 2011). DOE received comments indicating that the April 2011 SNOPR was overly burdensome, and the October 2011 SNOPR proposed a revised method that was intended to reduce this burden. 76 FR 65616 (Oct. 24, 2011).

Following the October 2011 SNOPR, the Joint Efficiency Advocates stated that, while minimizing test burden is important, DOE is also obligated by statute to prescribe a test procedure that measures the energy use of a covered product during a representative average use cycle or period of use. (42 U.S.C. 629(b)(3)) The Joint Efficiency Advocates stated that the Department's proposal was far from accomplishing that statutory requirement. (Joint Efficiency Advocates, No. 35 at p. 2) The CA IOUs noted that the test procedure revisions presented in the October 2011 SNOPR would not encourage innovative designs of heating systems in off mode, and that the results produced by the test procedure would be misleading to consumers, because the reported values would not be indicative of actual power draw if DOE were to require measurements based on fixed outdoor temperatures and use a simple average of *P1* and *P2.* (CA IOUs, No. 33 at p. 1)

However, in the December 2011 extension notice, DOE proposed to consider the suggestion by the CA IOUs to use the actual outdoor temperatures at which the crankcase heater turns on or off to measure *P1* and *P2,* as discussed in section III.D.2. The CA IOUs subsequently submitted comments that reaffirmed this proposal, and recommended that DOE consider its proposals to use a weighted average of P1 and P2 and to not adjust power draw for systems with multiple compressors or large-capacity systems. (CA IOUs, No. 40 at p.1) The Joint Efficiency Advocates conveyed strong support for the CA IOUs' proposal and remarked that the test procedure would not be indicative of actual energy use if DOE did not adopt the CA IOUs' proposal. (Joint Efficiency Advocates, No. 43 at p. 1; Joint Efficiency Advocates, No. 43 at p. 3)

As previously discussed, DOE must develop test procedures to measure energy use that balance test burden with measurement accuracy. The off mode test procedures published in the original NOPR and the first SNOPR were judged by stakeholders to be too complex and burdensome. As a result, DOE proposed a test method in the second SNOPR that was simplified and designed to result in comparatively less test burden. The simplified test procedure, however, may have impacted the ability to provide a measurement that is representative of an average use cycle or period of use. In this third SNOPR, DOE has made additional revisions and believes that this new proposed off mode test procedure limits test burden to a reasonable extent and will provide a means for measuring off mode power use in a representative manner.

E. Test Repeatability Improvement and Test Burden Reduction

42 U.S.C. 6293(b)(3) states that any test procedure prescribed or amended shall be reasonably designed to produce test results which measure energy efficiency and energy use of a covered product during a representative average period of use and shall not be unduly burdensome to conduct. This section discusses proposals to improve test procedure clarity and to reduce test burden. None of the proposals listed in this section would alter the average measured energy consumption of a representative set of models.

1. Indoor Fan Speed Settings

Indoor unit fan speed is typically adjustable during test set-up to assure that the provided air volume rate is appropriate for the field-installed ductwork system serving the building in which the unit is actually installed. The DOE test procedure accounts for these variable settings by establishing specific requirements for external static pressure and air volume rate during the test. For an indoor coil tested with an indoor fan installed, DOE's test procedure requires that (a) external static pressure be not less than a minimum value that depends on cooling capacity [11] and product class, ranging from 1.10 to 1.20 inches of water column (in. wc.) for small-duct, high-velocity systems and from 0.10 to 0.20 in. wc. for all other systems except non-ducted units (see 10 CFR part 430, subpart B, Appendix M, Table 2); and (b) the air volume rate divided by the total cooling capacity not exceed a maximum value of 37.5 cubic feet per minute of standard air (scfm) per 1000 Btu/h of cooling capacity [12] (see 10 CFR part 430, subpart B, Appendix M, Section 3.1.4.1.1).

Requirement (a) is more easily met using higher fan speeds, while requirement (b) is more easily met by lower fan speeds. DOE realizes that more than one speed setting may meet both the minimum static pressure and the maximum air volume rate requirements. Section 3.1.4.1.1(a)(6) of the current DOE test procedure for air conditioners and heat pumps allows adjustment of the fan speed to a higher setting if the first selected setting does not meet the minimum static pressure requirement at 95 percent of the cooling full-load air volume rate.[13] This step suggests that common test practice would be to initially select lower fan speeds to meet the requirements before attempting higher speeds. However, the test procedure does not, for cases in which two different settings could both meet the air volume rate and static pressure requirements, explicitly specify that the lower of the two settings should be used for the test. The fan power consumption would generally be less at lower speeds, but compressor power consumption may be reduced at conditions of higher air volume rate—hence it is not known prior to testing whether a higher or lower air volume rate will maximize the SEER or HSPF for a given individual model. However, DOE is aware that efficiency ratings are generally better when products are tested at the lowest airflow-control settings intended for cooling (or heating) operation that will satisfy both the minimum static pressure and maximum air volume rate requirements. DOE therefore proposes that blower coil products tested with an indoor fan installed be tested using the lowest speed setting that satisfies the minimum static pressure and the maximum air volume rate requirements, if applicable, if more than one of these settings satisfies both requirements. This is addressed in section 2.3.1.a of Appendix M.

For a coil-only system, *i.e.,* a system that is tested without an indoor fan installed, the pressure drop across the indoor unit must not exceed 0.3 inches of water for the A test (or A_2 test for two-capacity or variable-capacity systems), and the maximum air volume rate per capacity must not exceed 37.5 cubic feet per minute of standard air (scfm) per 1000 Btu/h. (10 CFR part 430, subpart B, Appendix M, Section 3.1.4.1.1) For such systems, higher air volume rates enhance the heat transfer rate of the indoor coil, and therefore may maximize the measured system capacity and efficiency. In addition, the energy use and heat input attributed to the fan energy for such products is a fixed default value in the test procedure, and is set at 365 W per 1,000 scfm (10 CFR part 430, subpart B, Appendix M, Section 3.3(d)). Thus, the impact from fan power on the efficiency measurement if air volume rate is increased may be more modest than for a unit tested with the indoor fan installed. However, a maximum external static pressure of 0.3 in. wc. is specified for the indoor coil assembly in order to represent the field-installed conditions. To minimize potential testing variability due to the use of different air volume rates, DOE proposes to require for coil-only systems for which the maximum air flow (37.5 scfm/1000 Btuh) or maximum pressure drop (0.3 in wc) are exceeded when using the specified air flow rate, the highest air flow rate that satisfies both the maximum static pressure and the maximum air volume rate requirements should be used. This is specified in section 3.1.4.1.1.c of Appendix M.

Improper fan speed implemented during testing may have a marked impact on product performance, and inconsistent implementation of speed adjustments may be detrimental to test repeatability. DOE therefore proposes to require that manufacturers include in their certification report the speed setting and/or alternative instructions for setting fan speed to the speed upon which the rating is based.

For consistency with the furnace fan test procedure, DOE proposes to add to Appendix M (and also Appendix M1) the definition for ''airflow-control setting'' that has been adopted in Appendix AA to refer to control settings used to obtain fan motor operation for specific functions.

DOE requests comment on its proposals regarding requirements on fan speed settings during test setup.

[11] Or heating capacity for heating-only heat pumps.

[12] Such a requirement does not exist for heating-only heat pumps.

[13] For heating-only heat pumps, Section 3.1.4.4.3(a)(6) allows adjustment of the fan speed to a higher setting if the first selected setting does not meet the requirements minimum static pressure requirement at 95 percent of the heating full-load air volume rate.

2. Requirements for the Refrigerant Lines and Mass Flow Meter

Section 2.2(a) of 10 CFR part 430, subpart B, Appendix M provides instructions for insulating the ''low-pressure'' line(s) of a split-system. In the cooling mode, the vapor refrigerant line connecting the indoor and outdoor units is operating at low refrigerant pressure. However, in the heating mode, the vapor refrigerant line connecting the indoor and outdoor units operates at high pressure, providing high pressure vapor to the indoor unit. To improve clarity and ensure that the language of the test procedure refers specifically to the actual functions of the refrigerant lines, DOE proposes to refer to the lines as ''vapor refrigerant line'' and ''liquid refrigerant line''.

Section 2.2(a) of 10 CFR part 430, subpart B, Appendix M and AHRI 210/240–2008 Section 6.1.3.5 both require insulation on the vapor refrigerant line and do not state what insulation, if any, is required on the liquid refrigerant line. Differences in product design and in the parts manufacturers decide to ship with the unit may lead to varying interpretations regarding the need to insulate the liquid refrigerant line during the test and may therefore introduce test variability. Furthermore, there may be unnecessary burden on test laboratories if they choose to add insulation when manufacturers do not to ship liquid refrigerant line insulation with the unit. While DOE wishes to clarify requirements for insulation of refrigerant lines, there are two factors that make such a determination difficult: (1) There may be reasons both for insulating and for not insulating the liquid refrigerant tubing—if not insulated, additional subcooling of the refrigerant liquid as it passes through the line prior to its expansion in the indoor unit may increase cooling capacity and thus increase the measured SEER. However, the increased subcooling of the liquid would increase the load on the outdoor coil during the heating mode of a heat pump, which may slightly reduce evaporating temperature and thus both reduce heat pump capacity and increase compressor power input. On the other hand, insulating the liquid line would result in higher measurements of HSPF for a heat pump when compared with measurements with the liquid line not insulated, but would result in lower measurements of the SEER; (2) DOE has observed that installation manuals for air conditioners and heat pumps generally indicate that liquid lines should be insulated in special circumstances (*e.g.,* running the line through a warm space or extra-long refrigerant line runs), but do not provide guidance on the use of insulation in the absence of such conditions.

Because DOE seeks to minimize test variability associated with the use of insulation, this notice includes a proposal for determining the insulation requirement for the test based on the materials and information included by the manufacturer with the test unit. Under this proposal, test laboratories would install the insulation shipped with the unit. If the unit is not shipped with insulation, the test laboratory would install the insulation specified in the installation manuals included with the unit by the manufacturer. Should the installation instructions not provide sufficient guidance on the means of insulating, liquid line insulation would be used only if the product is a heating-only heat pump. These proposed requirements are intended to reduce test burden and improve test repeatability for cooling and heating products, as well as heating-only products. DOE requests comment on its proposal to require that test laboratories install the insulation included with the unit or, if insulation was not furnished with the unit, follow the insulation specifications in the manufacturer's installation instructions. DOE also requests comment on its proposal to require liquid line insulation of heating-only heat pumps.

In cases where the refrigerant enthalpy method is used as a secondary measurement of indoor space conditioning capacity, uninsulated surfaces of the refrigerant lines and the mass flow meter may also contribute to thermal losses. DOE does not believe that preventing the incremental thermal losses associated with the mass flow meter components and its support structure would make a measurable impact on efficiency measurements. However, DOE does recognize the possibility that thermal loss might reduce the efficiency measurement, particularly during heating mode tests if the mass flow meter is placed on the test chamber floor, which might be cooler than the air within the room. To enhance test repeatability among various laboratories that may use different mass flow meters with varying materials for support structures, DOE proposes to require use of a thermal barrier to prevent such thermal transfers between the flow meter and the test chamber floor if the meter is not mounted on a pedestal or other support elevating it at least two feet from the floor. DOE proposes to add these requirements to Appendix M, section 2.10.3. DOE requests comment on this means to prevent meter-to-floor thermal transfer.

3. Outdoor Room Temperature Variation

Depending on the operating characteristics of the test laboratory's outdoor room conditioning equipment, temperature or humidity levels in the room may vary during testing. For this reason, a portion of the air approaching the outdoor unit's coil is sampled using an air sampling device (see Appendix M, section 2.5). The air sampling device, described in ASHRAE Standard 41.1–2013, consists of multiple manifolded tubes with a number of inlet holes, and is often called an air sampling tree. If, during testing, the air entering the outdoor unit of a product is monitored only on one of its faces and there is significant spatial variation of the room's air conditions, the measured conditions for the monitored face may not be indicative of the average conditions for the inlet air across all faces.

To ensure that the measurements account for variation in the conditions in the outdoor room of the test chamber, DOE proposes to require demonstration of air temperature uniformity over all of the air-inlet surfaces of the outdoor unit using thermocouples, if sampling tree air collection is performed only on one face of the outdoor unit. Specifically, DOE would require that the thermocouples be evenly distributed over the inlet air surfaces such that there is one thermocouple measurement representing each square foot of air-inlet area. The maximum temperature spread to demonstrate uniformity, *i.e.,* the maximum allowable difference in temperature between the measurements at the warmest location and at the coolest location, would be 1.5 °F (DOE proposes to add these requirements to Appendix M, section 2.11.b). This is the same maximum spread allowable for measurement of indoor unit capacity using thermocouple grids, as described in 10 CFR part 430, subpart B, Appendix M, Section 3.1.8, in which the maximum spread among the measured temperatures on the thermocouple grid in the outlet plenum of the indoor coil must not exceed 1.5 °F dry bulb. If this specified measurement of temperature uniformity cannot be demonstrated, DOE would require sampling tree collection of air from all air-inlet surfaces of the outdoor unit. DOE seeks comment for the proposed 1.5 °F maximum spread for demonstration of outdoor air temperature uniformity, the proposed one square foot per thermocouple basis for thermocouple distribution, and the proposed requirement that an air sampling device be used on all outdoor unit air-inlet surfaces if temperature uniformity is not demonstrated. DOE proposes to add these requirements to Appendix M, section 2.11.b.

4. Method of Measuring Inlet Air Temperature on the Outdoor Side

To ensure test repeatability, DOE seeks to ensure that temperature measurements taken during the test are as accurate as possible. DOE is aware that measurement of outdoor inlet temperatures is commonly based on measurements of the air collected by sampling devices that use high-accuracy dry bulb temperature and humidity measurement devices, and that the accuracy of these devices may be better than that of thermocouples. DOE proposes to require that the dry bulb temperature and humidity measurements, that are used to verify that the required outdoor air conditions have been maintained, be measured for the air collected by the air sampling devices (*e.g.,* rather than being measured by temperature sensors located in the air stream approaching the air inlets). DOE requests comment on this proposal.

5. Requirements for the Air Sampling Device

In evaluating various test setups and laboratory conditions, DOE has observed that certain setup conditions of the air sampling equipment could lead to measurement error or variability between laboratories. Specifically, the temperature of air collected by indoor and outdoor room air sampling devices could potentially change as it passes through the air collection system, leading to inaccurate temperature measurement if the air collection devices or the conduits conducting the air to the measurement location are in contact with the chamber floor or with ambient air at temperatures different from the indoor or outdoor room. To prevent this potential cause of error or uncertainty, DOE proposes to require that no part of the room air sampling device or the means of air conveyance to the dry bulb temperature sensor be within two inches of the test chamber floor. DOE also proposes to require those surfaces of the air sampling device and the means of air conveyance that are not in contact with the indoor and outdoor room air be insulated.

A potential contributor to error or uncertainty in the measurement of humidity is the taking of dry bulb and wet bulb measurements in different locations, if there is significant cool down of air between the two locations. While ASHRAE Standard 41.1–2013 provides an example of an air sampling device with a dry bulb and wet bulb thermometer placed close together, the figure is merely illustrative. To minimize measurement error or uncertainty, DOE proposes to require that humidity measurements and dry bulb temperature measurements used to determine the moisture content of air be made at the same location in the air sampling device.

As discussed in section III.E.14, DOE has also proposed several amendments to air sampling procedures that are included in a draft revision of AHRI 210/240–2008. DOE requests comments on all of these related proposals, including its proposal to require that the air sampling device and its components be prevented from touching the test chamber floor, to require insulation of those surfaces of the air sampling device and components that are not in contact with the chamber room air, and that dry bulb temperature and humidity measurements used to determine the moisture content of air be made at the same location in the air sampling device.

6. Variation in Maximum Compressor Speed With Outdoor Temperature

When testing an air conditioner or heat pump with a variable-speed compressor, the compressor must be tested at three different speeds: Maximum, intermediate, and minimum. Some air conditioners and heat pumps with a variable-speed compressor operate such that their maximum allowed compressor speed varies with the outdoor temperature. However, the test procedure does not explicitly state whether the maximum compressor speed refers to a fixed value or a temperature-dependent value. As such, DOE proposes that the maximum compressor speed be fixed during testing through modification of the control algorithm used for the particular product such that the speed does not change with the outdoor temperature. DOE requests comment on this proposal.

7. Refrigerant Charging Requirements

Near-azeotropic and zeotropic refrigerant blends are composed of multiple refrigerants with a range of boiling points. Gaseous charging of refrigerant blends is inappropriate because it can result in higher concentrations of the higher-vapor pressure constituents being charged into the unit, which can alter refrigerant performance characteristics and thus, unit performance. DOE recognizes that technicians certified to handle refrigerants via the Environment Protection Agency's (EPA) Section 608 Technician Certification Program, as mandated by 40 CFR 82.161, are required to be knowledgeable of charging methods for refrigerant blends. However, to ensure consistent practices within the context of the DOE test procedure, DOE proposes to require that near-azeotropic and zeotropic refrigerant blends be charged in the liquid state rather than the vapor state. This is found in section 2.2.5.8 of Appendix M. DOE requests comments on this proposal.

Current language in Appendix M to Subpart B of 10 CFR part 430 does not prohibit testers from changing the amount of refrigerant charge in a system during the course of air conditioner and heat pump performance tests. Changing the amount of refrigerant may result in a higher SEER and/or a higher HSPF that does not reflect the actual performance of a unit in the field. In the June 2010 NOPR, DOE proposed to adopt into the test procedure select parts of the 2008 AHRI General Operations Manual that contains language disallowing changing the refrigerant charge after system setup. (75 FR 31234–5) AHRI and NEEA supported this proposal. (AHRI, No. 6 at p. 3; NEEA, No. 7 at p. 4) To ensure that performance tests reflect operation in the field, and to improve consistency in results between test facilities, DOE intends to retain the proposal made in the June 2010 NOPR. Specifically, DOE retains the proposed requirement that once the system has been charged with refrigerant consistent with the installation instructions shipped with the unit (or with other provisions of the test procedure, if the installation instructions are not provided or not clear), all tests must be conducted with this charge.

DOE is aware that refrigerant charging instructions are different for different products, but that in some cases, such instructions may not be provided. More specifically, the appropriate charging method may vary among products based upon their refrigerant metering devices. The thermostatic expansion valve (TXV) type metering device is designed to maintain a specific degree of superheat.[14] Electronic expansion valve (EXV) type metering devices function similarly to TXV type metering devices, but use sensors, a control system, and an actuator to set the valve position to allow more sophisticated control of the degree of superheat. Fixed orifice is

[14] The degree of superheat is the extent to which a fluid is warmer than its bubble point temperature at the measured pressure, *i.e.,* the difference between a fluid's measured temperature and the saturation temperature at its measured pressure.

another type of expansion device commonly used for air conditioners. In contrast to a TXV or EXV, a fixed orifice does not actively respond to system pressures or temperatures to maintain a fixed degree of superheat. The refrigerant charge can affect the measured system efficiency. Systems with different expansion devices react differently to variation in the charge, and they also generally require different procedures for ensuring that the system is properly charged. As the charging operation may differ among these types of metering devices, and misidentification may lead to inconsistent charging and unrepeatable testing, DOE proposes to require manufacturers to report the type of metering device used during certification testing.

If charging instructions are not provided in the manufacturer's installation instructions shipped with the unit, DOE proposes standardized charging procedures to ensure consistent testing in a manner that reflects field practices. For a unit equipped with a fixed orifice type metering device for which the manufacturer's installation instructions shipped with the unit do not provide refrigerant charging procedures, DOE proposes that the unit be charged at the A or A_2 test condition, requiring addition of charge until the superheat temperature measured at the suction line upstream of the compressor is 12 °F with tolerance discussed in section III E.14.[15] For a unit equipped with a TXV or EXV type metering device for which the manufacturer's installation instructions shipped with the unit do not provide refrigerant charging procedures, DOE proposes that the unit be charged at the A or A_2 condition, requiring addition of charge until the subcooling[16] temperature measured at the condenser outlet is 10 °F with tolerance discussed in section III E.14.[17]

For heating-only heat pumps for which refrigerant charging instructions are not provided in the manufacturer's installation instructions shipped with the unit, the proposed standardized charging procedure would be followed while performing refrigerant charging at the H1 or $H1_2$ condition. DOE also proposes that charging be done for the H1 or $H1_2$ test condition for cooling/heating heat pumps which fail to operate properly in heating mode when charged using the standardized charging procedure for the A or A_2 test condition. In such cases, some of the tests conducted using the initial charge may have to be repeated to ensure that all tests (cooling and heating) are conducted using the same refrigerant charge. DOE proposes to add these requirements to Appendix M in a new section 2.2.5.8.

DOE requests comments on the proposed standardized charging procedures to be applied to units for which the installation instructions shipped with the unit do not provide charging instructions.

DOE understands that manufacturers may provide installation instructions with different charging procedures for the indoor and outdoor units. In such cases, DOE proposes to require charging based on the installation instructions shipped with the outdoor unit for outdoor unit manufacturer products and based on the installation instructions shipped with the indoor unit for independent coil manufacturer products, unless otherwise specified by either installation instructions. DOE requests comments on this proposal.

Single-package central air conditioners and heat pumps may be shipped with refrigerant already charged into the unit. Verifying the proper amount of refrigerant charge is valuable for increasing test repeatability. To this end, DOE believes that the benefits of installing pressure gauges on a single-package unit to help verify charge and to monitor refrigerant conditions generally outweigh the potential drawbacks associated with connecting the gauges (*e.g.,* refrigerant transfer from the product into the gauges and hoses or refrigerant leakage); calculating the superheat or subcooling quantities used to determine whether the unit is charged properly requires knowledge of the refrigerant pressure, and the quantity of charge transferred from the unit when connecting a pressure gauge set is generally a very small percentage of the unit's charge. Further, assessing the refrigerant charge may improve repeatability of the tests and measured efficiency. DOE therefore proposes that refrigerant line pressure gauges be installed during the setup of single-package and split-system central air conditioner and heat pump products, unless otherwise specified by the instructions. DOE also proposes that the refrigerant charge be verified per the charging instructions and, if charging instructions are not provided in the installation instructions shipped with the unit, the refrigerant charge would be verified based on the standardized charging procedure described above. DOE requests comments on these proposals.

As discussed in section III.E.14, DOE has also proposed several amendments to charging procedures that are included in a draft revision of AHRI 210/240–2008. DOE requests comment on all aspects of its proposals to amend the refrigerant charging procedures.

8. Alternative Arrangement for Thermal Loss Prevention for Cyclic Tests

10 CFR part 430, subpart B, Appendix M, Section 2.5(c) requires use of damper boxes in the inlet and outlet ducts of ducted units to prevent thermal losses during the OFF period of the compressor OFF/ON cycle for the cooling or heating cyclic tests. However, DOE is aware that installation of such dampers for single-package ducted units can be burdensome because the unit must be located in the outdoor chamber and there may be limited space in the chamber and in between the inlet and outlet ducts to install the required transition ducts, insulation, and dampers. To preserve the intent of the air damper boxes, reduce testing burden, and accommodate variations in chamber size, DOE proposes an alternative testing arrangement to prevent thermal losses during the compressor OFF period that would eliminate the need to install a damper in the inlet duct that conveys indoor chamber air to the indoor coil.

The proposed alternative testing arrangement would allow the use of a duct configuration that relies on changes in duct height, rather than a damper, to eliminate natural convection thermal transfer out of the indoor duct during OFF periods of the ''cold'' or heat generated by the system during the ON periods. An example of such an arrangement would be an upturned duct installed at the inlet of the indoor duct, such that the indoor duct inlet opening, facing upwards, is sufficiently high to prevent natural convection transfer out of the duct. DOE also proposes to require installation of 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. Measurement and recording of dry bulb temperature at this location would be required at least every minute during the compressor OFF period to confirm that no thermal loss occurs. DOE

[15] The range of superheating temperatures was generalized from industry-accepted practice and state-level authority regulations on refrigerant charging for non-TXV systems.

[16] The degree of subcooling or subcooling temperature is the extent to which a fluid is cooler than its refrigerant bubble point temperature at the measured pressure, *i.e.,* the bubble point temperature at a fluid's measured pressure minus its measured temperature. Bubble point temperature is the temperate at a given pressure at which vapor bubbles just begin to form in the refrigerant liquid.

[17] The range of subcooling temperatures was generalized from manufacturer-published and technician-provided service instructions and are typical of industry practice.

proposes a maximum permissible variation in temperature measured at this location during the OFF period of ±1.0 °F.

DOE seeks comment on its proposal in section 2.5(c) of Appendix M to allow, for cyclic tests, alternative arrangements to replace the currently-required damper in the inlet portion of the indoor air ductwork for single-package ducted units. DOE also requests comment on the proposed requirements for ensuring that there are no thermal losses during the OFF portion of the test, including the location of the proposed dry bulb temperature sensor, the requirements for recorded temperatures, and the ±1.0 °F allowable variation in temperature measured by this sensor.

9. Test Unit Voltage Supply

The current DOE test procedure references ARI Standard 210/240–2006 Section 6.1.3.2 for selecting the proper electrical voltage supply, which generally requires that, for tests performed at standard rating conditions (referred to as ''Standard Rating tests'' in Standard 210/240), the tests be conducted at the product's nameplate rated voltage and frequency. This section also requires that Standard Rating tests be performed at 230 V for air-cooled equipment rated with 208–230 V dual nameplate voltages, and that all other dual nameplate voltage equipment be tested at both voltages or at the lower of the two voltages if only a single Standard Rating is to be published. DOE recognizes that nameplate voltages may differ for indoor and outdoor units. This may result in a difference of voltage supplied to the indoor and outdoor units in accordance with the current test requirement. DOE realizes that, in most cases, this voltage difference that may occur during testing is not representative of field operation where indoor and outdoor units are typically supplied with the same voltage. As such, DOE proposes to clarify that the outdoor voltage supply requirement supersedes the indoor requirement if the provisions result in a difference for the indoor and outdoor voltage supply. That is, both the indoor and outdoor units shall be tested at the same voltage supplied to the outdoor unit.

10. Coefficient of Cyclic Degradation

The cooling coefficient of degradation, C_D^c, is the ratio of the EER measured for cycling (or intermittent) operation to the EER that would be measured for steady operation. The heating coefficient of degradation, C_D^h, is a similar factor that characterizes efficiency reduction for cycling operation during heat pump operation. The test procedures to determine these two coefficients are the same except for the testing conditions and unit operation mode, and the changes discussed in this section are applied to both metrics. Therefore, for the sake of simplicity and clarity, only the cooling coefficient of degradation is discussed here.

The current test procedure gives manufacturers the option to use a default cyclic degradation coefficient (C_D) value of 0.25 instead of running the optional cyclic test. In response to the June 2010 NOPR, which proposed some modifications related to the optional tests but not the default value, NEEA commented that its laboratory testing demonstrated that the default value 0.25 is not representative of system performance, especially for TXV-equipped systems, and instead supported using the actual tested values in determining ratings. (NEEA, No. 7 at pp. 6–7) DOE reviewed results from its own testing of 19 split-system and single-package air conditioners and heat pumps from 1.5 to 5 tons and found that the tested C_D values range from 0.02 to 0.18, with an average of 0.09. It also found no correlation between C_D and SEER, EER, or cooling capacity. DOE also reviewed the AHRI 210/240-Draft (see section III.E.14), which updates the cooling C_D^c value to 0.2. DOE believes this default value may be more in-line with actual tested values, and DOE proposes to update the default cooling C_D^c value in Appendix M to 0.2. At this time, DOE is not proposing to update the default heating C_D^h value. In evaluating appropriate default values, DOE also reviewed its testing requirements to measure C_D.

DOE is aware of various issues that occur when conducting the test procedure to measure the degradation coefficient, such as the inability to attain stable capacity measurements from cycle to cycle and burdensome testing time to attain stability, and believes that these are symptoms of cyclic instability. DOE believes that the variation in cooling capacity during the test to determine C_D^c is exacerbated by the short compressor on-time specified for each cycle and by the effect of response time, sensitivity, and repeatability errors. DOE understands the importance of having a minimally burdensome test procedure. However, DOE recognizes that the current test method for measuring C_D^c, although clear in description and intent, does not provide requirements for cyclic stability of measured capacity over successive on-cycles during the test. Therefore, DOE proposes the following procedure based on cyclic testing data to clarify the test procedure, address cyclic stability, and offer default procedures to allow for test burden relief.

DOE has obtained cyclic test data that show that as cycles are tested, either capacity reaches steady-state or capacity fluctuates constantly and consistently. Therefore, DOE proposes that before determining C_D^c, three ''warm up'' cycles for a unit with a single-speed compressor or two-speed compressor or two ''warm up'' cycles for a unit with a variable speed compressor must be conducted. Then, conduct a minimum of three complete cycles after the warm-up period, taking a running average of C_D^c after each additional cycle. If after three cycles, the average of three cycles does not differ from the average of two cycles by more than 0.02, the three-cycle average should be used. If it differs by more than 0.02, up to two more valid cycles will be conducted. If the average C_D^c of the last three cycles are within 0.02 of or lower than the previous three cycles, use the average C_D^c of all valid cycles. After the fifth valid cycle, if the average C_D^c of the last three cycles is more than 0.02 higher than the previous three cycles, the default value will be used. The same changes are proposed for the test method to determine the heating coefficient of degradation.

Given these changes to address, DOE proposes that unlike the current test procedure, manufacturers must conduct the specified testing required to measure C_D for each tested unit. The default value may only be used if stability or the test tolerance is not achieved or when testing outdoor units with no match.

DOE requests comment regarding the proposed revisions to the cyclic test procedure for the determination of both the cooling and heating coefficient of degradation. DOE also requests additional test data that would support the proposed specifications, or changes to, the number of warm-up cycles, the cycle time for variable speed units, the number of cycles averaged to obtain the value, and the stability criteria.

11. Break-In Periods Prior to Testing

On June 1, 2012, AHRI submitted a supplement to the comments it submitted on January 20, 2012, as part of the extended comment period on the October 2011 SNOPR. In these supplementary comments, AHRI requested that DOE implement an optional 75-hour break-in period for testing central air conditioners and heat pumps. It stated that scroll compressors, which are the type of compressors most commonly used in central air conditioners and heat pumps, achieve their design efficiency after 75 hours of operation, so the allowance for a break-in period of this length would ensure that the product being tested is operating as intended by the manufacturer and would provide a result that is more representative of average use. AHRI also cited a study of compressor break-in periods to justify this period of time,[18] and added that, while AHRI's certification program for central air conditioners and heat pumps does not specify a minimum break-in period, it does allow manufacturers to specify a break-in period for their products. According to AHRI's comments, some manufacturers request a break-in period in excess of 100 hours, while others request 50 hours or less.

Furthermore, AHRI commented that implementation of an optional break-in period for central air conditioners and heat pumps would be consistent with a similar provision in the DOE test procedures for commercial heating and air-conditioning equipment, which DOE adopted in a final rule published May 16, 2012. 77 FR 28928. As stated in the final rule, the purpose of including this option for testing commercial HVAC equipment was to ensure that the equipment being tested would have time to achieve its optimal performance prior to conducting the test. DOE placed a maximum limit of 20 hours on the allowed period of break-in, regardless of the break-in period recommended by the manufacturer, explaining that such a limit was necessary to minimize the burden imposed by this provision. In addition, DOE required that manufacturers who use the optional break-in period report the duration of their break-in as part of the test data underlying the certification that is required to be maintained under 10 CFR 429.71. DOE stated that it would use the same break-in period for any DOE-initiated testing as the manufacturer used in its certified ratings or, in the case of ratings based upon use of an alternate efficiency determination method (AEDM), the maximum 20-hour break-in period. 77 FR 28928, 28944.

After consideration of the potential improvement in performance and increased test burden that may result from implementation of an optional 75-hour break-in period, DOE believes that the lengthy break-in period is not appropriate or justified. In reviewing the paper that AHRI cited in its comments, DOE noted that, while the data indicate that products with scroll compressors do appear to converge upon a more consistent result after compressor break-in periods exceeding 75 hours, the most significant improvement in compressor performance and reduction in variation among compressor models both appear to occur during roughly the first 20 hours of run time.[19] Moreover, scroll compressors in use at the time of this paper's publication in 1996 may have required longer break-in periods to address the surface quality of the internal components resulting from the manufacturing processes of that time, whereas compressors in use today have benefitted from improvements in the manufacturing technology for scroll compressors over the past 20 years. In addition, while the paper also supports AHRI's comment that smaller compressors require more time to reach their optimal performance than larger compressors, it does not show the absolute size of the compressors that were studied and makes comparisons based only on their relative sizes. Therefore, it is difficult to precisely determine how this data would apply to a central air conditioner or heat pump compressor versus a commercial air conditioner or heat pump. Finally, since DOE determined in the May 16, 2012 commercial HVAC equipment final rule that a 20 hour maximum break-in time would be sufficient for small commercial air-conditioning products, which are of a capacity similar to central air-conditioning products, DOE does not see justification for a break-in period longer than 20 hours for products. 77 FR 28928.

In consideration of AHRI's comments on the merits of conducting a break-in period prior to testing of central air conditioners and heat pumps, DOE proposes in this SNOPR to allow manufacturers the option of specifying a break-in period to be conducted prior to testing of these products under the DOE test procedure. However, due to the excessive test burden that could be imposed by allowing lengthy break-in times, DOE proposes to limit the optional break-in period to 20 hours, which is consistent with the test procedure final rule for commercial HVAC equipment. DOE also proposes to adopt the same provisions as the commercial HVAC rule regarding the requirement for manufacturers to report the use of a break-in period and its duration as part of the test data underlying their product certifications, the use of the same break-in period specified in product certifications for testing conducted by DOE, and use of the 20 hour break-in period for products certified using an AEDM.

DOE requests comments on its proposal to allow an optional break-in period of up to 20 hours prior to testing as part of the DOE test procedure for central air conditioners and heat pumps.

12. Industry Standards That Are Incorporated by Reference

In the June 2010 NOPR, DOE proposed two ''housekeeping'' updates throughout Appendix M regarding test procedure references. 75 FR 31243. The first is an update of the incorporation by reference (IBR) from ARI Standard 210/240–2006 to ANSI/AHRI 210/240–2008, which provides additional test unit installation requirements and requirements on apparatus used during testing. The second update involves changes to references from 10 CFR 430.22 to 10 CFR 430.3, as the listing of those materials incorporated by reference was relocated. In the public comment period following the NOPR, AHRI expressed support for updating the test procedure to reference current AHRI and ASHRAE standards. (AHRI, No. 6 at p. 6). DOE is maintaining its position in the June 2010 NOPR for both proposals and therefore implemented the reference updates in the reprint of Appendix M of this notice. However, DOE proposes in this SNOPR to incorporate by reference the 210/240 standard having the most recent amendments at the time of this notice, *i.e.,* ANSI/AHRI 210/240–2008 with Addendum 2.[20] The changes incorporated by these amendments relate to replacing the Integrated Part Load Value (IPLV) efficiency metric with the Integrated Energy Efficiency Ratio (IEER) metric, as well as adding the methodology for determining IEER for water- and evaporatively-cooled products. These changes are relevant only to commercial equipment and are not relevant to the DOE test procedure for central air conditioners and heat pumps. Therefore updating references to the latest version of ANSI/AHRI 210/240 will not impact the ratings or energy conservation standards for central air conditioners and heat pumps.

In addition, in this SNOPR, DOE proposes to update the IBR from ASHRAE Standard 37–2005, Methods of Testing for Rating Unitary Air-Conditioning and Heat Pump Equipment to ASHRAE Standard 37–2009, Methods of Testing for Rating Electrically Driven Unitary Air-Conditioning and Heat Pump

[18] Khalifa, H.E. ''Break-in Behavior of Scroll Compressors'' (1996). *International Compressor Engineering Conference.* Paper 1145.

[19] Ibid. pp. 442–443.

[20] ANSI/AHRI 210/240–2008 with Addendum 2 is named as such but includes changes per an Addendum 1 on the same standard.

Equipment; ASHRAE 41.9–2000, Calorimeter Test Standard Methods for Mass Flow Measurements of Volatile Refrigerants to ASHRAE 41.9–2011, Standard Methods for Volatile-Refrigerant Mass Flow Measurements Using Calorimeters; and ASHRAE/AMCA 51–1999/210–1999, Laboratory Methods of Testing Fans for Aerodynamic Performance Rating to ASHRAE/AMCA 51–07/210–07, Laboratory Methods of Testing Fans for Certified Aerodynamic Performance Rating. None of these updates includes significant changes to the sections referenced in the DOE test procedure and thus will not impact the ratings or energy conservation standards for central air conditioners and heat pumps.[21]

Additionally, DOE proposes to update the IBR from ASHRAE 41.1–1986 (Reaffirmed 2006), Standard Method for Temperature Measurement to ASHRAE 41.1–2013, Standard Method for Temperature Measurement, as well as the IBR to ASHRAE 41.6–1994, Standard Method for Measurement of Moist Air Properties to ASHRAE 41.6–2014, Standard Method for Humidity Measurement. In the updated versions of these standards, specifications for measuring wet-bulb temperature were moved from ASHRAE 41.1 to ASHRAE 41.6. None of these updates includes significant changes to the sections referenced in the DOE test procedure and thus will not impact the ratings or energy conservation standards for central air conditioners and heat pumps.

Also, DOE proposes to update the IBR from ASHRAE 23–2005, Methods of Testing for Rating Positive Displacement Refrigerant Compressors and Condensing Units to ASHRAE 23.1–2010 Methods of Testing for Rating the Performance of Positive Displacement Refrigerant Compressors and Condensing Units That Operate at Subcritical Temperatures of the Refrigerant. ASHRAE 23 has been withdrawn and has been replaced by ASHRAE 23.1 and ASHRAE 23.2. ASHRAE 23.2 deals with supercritical pressure conditions, which are not relevant to the DOE test procedure, so will not be referenced. None of these updates includes significant changes to the sections referenced in the DOE test procedure and thus will not impact the ratings or energy conservation standards for central air conditioners and heat pumps.

[21] ASHRAE 37–2009 only updates to more recent versions of other standards it references. ASHRAE/AMCA 51–07/210–07 made slight changes to the figure referenced by DOE, which DOE has determined to be insignificant.

DOE also proposes to revise its existing IBRs to AHRI 210/240–2008 with Addendums 1 and 2, ANSI/AHRI 1230–2010 with Addendum 2, ASHRAE 23.1–2010 (updated from ASHRAE 23–2005), ASHRAE 37–2009 (updated from 2005), ASHRAE 41.1–2013 (updated from 1986 version), ASHRAE 41.2–1987, ASHRAE 41.6–2014 (updated from 1994 reaffirmed in 2001 version), ASHRAE 41.9–2011 (updated from 2000 version), and ASHRAE/AMCA 51–07/210–07 (updated from 1999 version) to incorporate only the sections currently referenced or proposed to be referenced in the DOE test procedure. DOE requests comment on its proposed sections for incorporation and specifically on whether any additional sections may be necessary to conduct a test of a unit.

DOE also proposes to revise the definition of ''continuously recorded'' based on changes to ASHRAE 41.1. ASHRAE 41.1–86 specified the maximum time intervals for sampling dry-bulb temperature. The updated version, ASHRAE 41.1–2013 does not contain specifications for sampling intervals. DOE proposes to require that dry-bulb temperature, wet bulb temperature, dew point temperature, and relative humidity data be ''continuously recorded,'' that is, sampled and recorded at 5 second intervals or less. DOE is proposing this requirement as a means of verifying that temperature condition requirements are met for the duration of the test. DOE requests comment on its revised sampling interval for dry-bulb temperature, wet bulb temperature, dew point temperature, and relative humidity.

13. Withdrawing References to ASHRAE Standard 116–1995 (RA 2005)

In the June 2010 NOPR, DOE proposed referencing ASHRAE Standard 116–1995 (RA 2005) within the DOE test procedure to provide additional informative guidance for the equations used to calculate SEER and HSPF for variable-speed systems. 75 FR 31223, 31243 (June 2, 2010). In the subsequent public comment period, AHRI expressed support for DOE's proposal to reference ASHRAE 116. (AHRI, No. 6 at p. 6). However, in section III.H.4 of this notice, DOE proposes to change the heating load line, and as such the equations for HSPF in ASHRAE Standard 116 are no longer applicable. In order to prevent confusion, DOE proposes in this notice to withdraw the proposal made in the June 2010 NOPR to reference ASHRAE 116 for both HSPF and SEER and is removing those instances of references to said standard from the test procedure.

Appendix M only references ASHRAE 116 in one other location, regarding the requirements for the air flow measuring apparatus. Upon review, DOE has determined that referencing ASHRAE Standard 37 instead provides sufficient information. As a result, in this NOPR, DOE also proposes to revise its reference for the requirements of the air flow measuring apparatus to ASHRAE Standard 37–2009 rather than ASHRAE 116, and proposes to remove the incorporation by reference to ASHRAE 116 from the code of federal regulations related to central air conditioners and heat pumps.

14. Additional Changes Based on AHRI 210/240-Draft

In August 2015, AHRI provided a draft version of AHRI 210/240 for the docket that will supersede the 2008 version once it is published. (AHRI Standard 210/240-Draft, No. 45, See EERE–2009–BT–TP–0004–0045) The draft version includes a number of revisions from the 2008 version, some of which already exist in DOE's test procedure, and some of which do not.

Regarding test installation requirements, the AHRI 210/240-Draft added new size requirements for the inlet duct to the indoor unit. If used, the inlet duct size to the indoor unit is required to equal the size of the inlet opening of the air-handling (blower-coil) unit or furnace, with a minimum length of 6 inches. Regarding the testing procedure, the AHRI 210/240-Draft added new external static pressure requirements for units intended to be installed with the airflow to the outdoor coil ducted. These new requirements provide for testing of these products more consistently with the way that they are intended to be used in the field. Also regarding the testing procedure, the AHRI 210/240-Draft specified a new requirement for the dew point temperature of the indoor test room when the air surrounding the indoor unit is not supplied from the same source as the air entering the indoor unit. DOE proposes to adopt these three revisions in this SNOPR.

The AHRI 210/240-Draft includes several differences as compared to the current DOE test procedure for setting air volume rates during testing. Specifically:

(a) Air volume rates would be specified by the manufacturer;

(b) For systems tested with indoor fans installed in which the fans have permanent-split-capacitor (PSC) or constant-torque motors, there would be minimum external static pressure requirements for operating modes other than full-load cooling; and

(c) A criterion is defined for acceptable air flow stability for systems tested with constant-air-volume indoor fans (these are fans with controls that vary fan speed to maintain a constant air volume rate).

DOE proposes to adopt these changes because they will improve repeatability and the consistency of testing among different laboratories.

The AHRI 210/240-Draft also includes a more thorough procedure for setting of refrigerant charge than exists in the DOE test procedure. The new approach addresses potential issues associated with conflicting guidelines that might be provided by manufacturer's installation instructions and indicates how to address ranges of target values provided in instructions. DOE is proposing these changes because they improve test repeatability. The AHRI 210/240-Draft also specifies both a target value tolerance and a maximum tolerance but does not specify in what circumstances each of these apply. DOE proposes to adopt the maximum tolerance only. However, DOE may consider adopting only the target value tolerance or both the target value and maximum tolerance. DOE requests comment on the appropriate use of the target value and maximum tolerances, as well as data to support the appropriate selection of tolerance. DOE notes that the tolerances adopted in the DOE test procedure should be achievable by test lab personnel without the presence or direct input of the manufacturer.

Finally, the AHRI 210/240-Draft includes specifications for air sampling that provide more detail than provided in existing standards. DOE proposes to incorporate these specifications by reference in order to improve test procedure repeatability and consistency. The proposal currently cites the AHRI 210/240-Draft, which is not possible for the final rule associated with this rulemaking. However, DOE expects that the AHRI standard will be finalized in time to allow the final rule to amend the CFR to incorporate this material.

DOE notes that the final published version of what is currently the AHRI 210/240-Draft may not be identical to the current draft. If AHRI makes other than minor editorial changes to the sections DOE references in this SNOPR after publication of this SNOPR, DOE proposes to adopt the current draft content into its regulations and not incorporate by reference the modified test procedure.

15. Damping Pressure Transducer Signals

ASHRAE 37–2009, which DOE proposes in this SNOPR to be incorporated by reference into the DOE test procedure, includes requirements for maximum allowable variation of specific measurements for a valid test. Specifically, Table 2 of the standard indicates that the test operating tolerance (total observed range) of the nozzle pressure drop may be no more than 2 percent of the average value of reading. Section 5.3.1 of the standard indicates that the nozzle pressure drop (or the nozzle throat velocity pressure) may be measured with manometers or electronic pressure transducers. These measurements are made to determine air flow. Section 8.7.2 of the standard requires that measurements shall be recorded at equal intervals that span five minutes or less when evaluating cooling capacity.

DOE is aware that when nozzle pressure drop measurements are made with pressure transducers and recorded using a computer-based data acquisition system, high frequency pressure fluctuations can cause observed pressure variations in excess of the 2 percent test operating tolerance, even when air flows are steady and non-varying. DOE proposes to add clarifying language in the test procedure that would allow for damping of the measurement system to prevent such high-frequency fluctuations from affecting recorded pressure measurements. The proposal would allow for damping of the measurement system so that the time constant for response to a step change in pressure (*i.e.* the time required for the indicated measurement to change 63% of the way from its initial value to its final value) is no more than five seconds. This damping could be achieved in any portion of the measurement system. Examples of damping approaches include adding flow resistance to the pressure signal tubing between the pressure tap and the transducer, using a transducer with internal averaging of its output, or filtering the transducer output signal, digital averaging of the measured pressure signals. DOE requests comment on this proposal, including on whether the proposed maximum time constant is appropriate.

F. Clarification of Test Procedure Provisions

Ensuring repeatability of test results requires that all parties that test a unit use the same set of instructions to set up the unit, conduct the test, and calculate test results. A test laboratory may be tempted to contact the product's manufacturer or other sources of information not referenced or allowed by the test procedure if there is a lack of clarity in the installation instructions shipped with the unit or ambiguities within the test procedure itself. Currently, certain sections of the DOE test procedure for central air conditioners and heat pumps in Appendix M to Subpart B of 10 CFR part 430 permit such consultation with the manufacturer. In the June 2010 NOPR, DOE proposed to allow lab-manufacturer communication as long as test unit installation and laboratory testing are conducted in complete compliance with all requirements in the DOE test procedure and the unit is installed according to the manufacturer's installation instructions. 75 FR 31223, 31235 (June 2, 2010). In the subsequent public comment period, AHRI expressed support regarding DOE's proposal. (AHRI, No. 6 at p. 3). Mitsubishi also supported adding test procedure to clarify that interaction with the manufacturer is allowed. (Mitsubishi, No. 12 at p. 2). NEEA did not object to DOE's proposal. (NEEA, No. 7 at p. 4). Because the reliance upon such consultation could lead to variability in test results among laboratories by manufacturers providing different testing instructions, DOE seeks to limit such occurrences to the maximum extent possible by ensuring that all required testing conditions and product setup information is either specified in the test procedure, certified to DOE, or stated in installation manuals shipped with the unit by the manufacturer. DOE believes that the proposed revisions in this rule provide such clarity and allow for models to be tested and rated in an equitable manner across manufacturers. Upon implementing such clarifications, laboratories will no longer need to contact the manufacturer for advice on implementation of the test procedure. If questions arise about a specific test procedure provision, the test lab and/or the manufacturer should seek guidance from DOE. DOE believes that this change will eliminate inconsistent testing due to different test laboratories seeking and receiving different information regarding unclear instructions. Thus, DOE proposes the following changes to the test procedure to address test procedure provisions that may be ambiguous or unclear in their intent and also withdraws the proposal it made in the June 2010 NOPR that placed no restrictions on interactions between manufacturers and third-party test laboratories 75 FR at 31235.

1. Manufacturer Consultation

DOE proposes to clarify the test procedure provisions regarding the specifications for refrigerant charging prior to testing, with input on certain details from the AHRI 210/240-Draft, as discussed in section III.*E*.14. Section 2.2.5 of the test procedure provides refrigerant charging instructions but also states, ''For third-party testing, the test laboratory may consult with the manufacturer about the refrigerant charging procedure and make any needed corrections so long as they do not contradict the published installation instructions.'' The more thorough refrigerant charging requirements proposed in this notice should preclude the need for any manufacturer consultation, since they include steps to take in cases where manufacturer's installation instructions fail to provide information regarding refrigerant charging or provide conflicting requirements. Consultation with the manufacturer should thus become unnecessary, and DOE proposes to remove the current test procedure's allowance for contacting the manufacturer to receive charging instructions. In instances where multiple sets of instructions are specified or are included with the unit and the instructions are unclear on which set to test with, DOE proposed in the June 2010 NOPR to use the instructions ''most appropriate for a normal field installation.'' 75 FR 31235, 31250. (June 2, 2010) NEEA supported this proposal. (NEEA, No. 7 at p. 4). DOE proposes to maintain this position in this rulemaking, proposing the use of field installation criteria if instructions are provided for both field and lab testing applications.

In the June 2010 NOPR, DOE proposed requirements for the low-voltage transformer used when testing coil-only air conditioners and heat pumps, and required metering of such low-voltage component energy consumption during all tests. 75 FR 31238. In the April 2011 SNOPR, in response to the June 2010 NOPR public meeting comments, DOE proposed revised requirements such that metering of low-voltage component energy consumption is required during only the proposed off mode testing, citing that such changes would require adjustments to the standard levels currently being considered. 76 FR 18109. The proposal therein consisted of language that suggested that test setup information may be obtained directly from manufacturers. In the effort to remain objective during testing, DOE is hereby revising certain language in the proposal such that communication between third party test laboratories and manufacturers are eliminated, and such information when needed for test setup can be found in the installation manuals included with the unit by the manufacturer.

Regarding the use of an inlet plenum, section 2.4.2 of the test procedure states, ''When testing a ducted unit having an indoor fan (and the indoor coil is in the indoor test room), the manufacturer has the option to test with or without an inlet plenum installed. Space limitations within the test room may dictate that the manufacturer choose the latter option.'' To eliminate the need for the test laboratory to confirm with the manufacturer whether the inlet plenum was installed during the manufacturer's test, DOE proposes to require manufacturers to report on their certification report whether the test was conducted with or without an inlet plenum installed.

Further, it is unclear in certain sections of the test procedure which ''test setup instructions'' are to be referenced for preparing the unit for testing. Ambiguous references to ''test setup instructions'' and/or ''manufacturer specifications'' may lead to the use of instructions or specifications provided by the manufacturer that are possibly out-of-date or otherwise not applicable to the products being tested. DOE therefore proposes to amend references in the test procedure to test setup instructions or manufacturer specifications by specifying that these refer to the test setup instructions included with the unit. DOE proposes to implement this change in the following sections: 2.2.2, 3.1.4.2(c), 3.1.4.4.2(c), 3.1.4.5(d), and 3.5.1(b)(3).

2. Incorporation by Reference of ANSI/AHRI Standard 1230–2010

ANSI/AHRI Standard 1230–2010 ''Performance Rating of Variable Refrigerant Flow (VFR) Multi-Split Air-Conditioning and Heat Pump Equipment'' with Addendum 2 (AHRI Standard 1230–2010) prescribes test requirements for both consumer and commercial variable refrigerant flow multi-split systems. On May 16, 2012, DOE incorporated this standard by reference into test procedures for testing commercial variable refrigerant flow multi-split systems at 10 CFR 431.96. 77 FR 28928. DOE recognizes that consumer variable refrigerant flow multi-split systems have similarities to their commercial counterparts. Therefore, to maintain consistency of testing consumer and commercial variable refrigerant flow multi-split systems, DOE proposes to incorporate by reference the sections of AHRI Standard 1230–2010 that are relevant to consumer variable refrigerant flow multi-split systems (namely, sections 3 (except 3.8, 3.9, 3.13, 3.14, 3.15, 3.16, 3.23, 3.24, 3.26, 3.27, 3.28, 3.29, 3.30, and 3.31), 5.1.3, 5.1.4, 6.1.5 (except Table 8), 6.1.6, and 6.2) into the existing test procedure for central air conditioners and heat pumps at Appendix M to Subpart B of 10 CFR part 430. To ensure that there is no confusion with future definition changes in industry test procedures, DOE is including the terms ''Multiple-split (or multi-split) system'', ''Small-duct, high-velocity system'', ''Tested combination'', ''Variable refrigerant flow system'' and ''Variable-speed compressor system'' into its list of definitions in Appendix M to Subpart B of 10 CFR part 430.

10 CFR 429.16 requires the use of a ''tested combination,'' as defined in 10 CFR 430, subpart B, Appendix M, section 1.B, when rating multi-split systems. In response to a May 27, 2008 letter from AHRI to DOE, DOE proposed changes in the ''tested combination'' definition in the June 2010 NOPR. 75 FR 31223, 31231 (June 2, 2010). In comments responding to the NOPR, AHRI urged DOE to adopt AHRI Standard 1230–2010 for all requirements pertaining to multi-split systems. (AHRI, No. 6 at pp. 1–2) Mitsubishi recommended likewise. (Mitsubishi, No. 12 at p. 1) AHRI Standard 1230–2010, published after the June 2010 NOPR, duplicates most of the requirements for tested combinations that DOE proposed in the June 2010 NOPR except for the following requirements, which DOE proposes in this notice to adopt to reduce manufacturer test burden: lower the maximum number of indoor units matched to an outdoor unit; and the option to use another indoor model family if units from the highest sales volume model family cannot be combined so that the sum of their nominal capacities is in the required range of the outdoor unit's nominal capacity (between 95 and 105 percent). The proposal in June 2010 NOPR also used the term ''nominal cooling capacity,'' which may be ambiguous; DOE also intends to clarify that such a term should be interpreted as 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 as the lowest cooling capacity listed in published product literature for these

conditions. If incomplete or no operating conditions are reported, the highest (for indoor units) or lowest (for outdoor units) such cooing capacity shall be used. Finally, AHRI 1230 uses the term ''model family'' but does not define the term. DOE requests comment on an appropriate definition of ''model family'' for DOE to adopt in the final rule. In summary, DOE proposes to omit AHRI's definition of tested combination, found in section 3.26, from the IBR of AHRI Standard 1230–2010 into Appendix M to Subpart B of 10 CFR part 430, and make amendments to the proposal from the June 2010 NOPR.

During testing for ducted systems with indoor fans installed, the rise in static pressure between the air inlet and the outlet (called external static pressure (ESP)) must be adjusted to a prescribed minimum that varies with system cooling capacity. The minimum ESPs are 0.10 in. wc. for units with cooling capacity less than 28,800 Btu/h; 0.15 in. wc. for units with cooling capacity from 29,000 Btu/h to 42,500 Btu/h; and 0.20 in. wc. for units with cooling capacity greater than 43,000 Btu/h. Multi-split systems are composed of multiple indoor units, which may be designed for installation with short-run ducts. Such indoor units generally cannot deliver the minimum ESPs prescribed by the current test procedure. Hence, lower minimum ESP requirements may be necessary for testing of ducted multi-split systems.

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. In its comments, AHRI urged DOE to adopt the minimum ESP requirements from AHRI Standard 1230–2010 as DOE was aware that the standard was being developed at that time. AHRI expressed concern over the potential abuse of lower multi-split minimum ESPs requirements by manufacturers of ducted single-indoor-unit split-system products. Specifically, they were concerned that the lower ESP were allowed for very specific installation applications which could not be assured by the manufacturer, and thus might be used more widely than intended. AHRI therefore argued against changing ESP requirements. (AHRI, No. 6 at p. 2). Mitsubishi recommended likewise. (Mitsubishi, No. 12 at p. 2). NEEA recommended establishing minimum ESP requirements that are the same as those of conventional systems. (NEEA, No. 7 at p. 2) AHRI Standard 1230–2010 does not include minimum ESP requirements for multi-split systems with short-run ducted indoor units. In order to accommodate the design differences of these indoor units, DOE proposes to omit Table 8 of AHRI Standard 1230–2010 from the IBR into Appendix M and to set minimum ESP requirements for systems with short-run ducted indoor units at the levels and cooling capacity thresholds as proposed in the June 2010 NOPR. Furthermore, DOE proposes to implement these requirements by (a) defining the term ''Short duct systems,'' to refer to ducted systems whose indoor units can deliver no more than 0.07 in. wc. ESP when delivering the full load air volume rate for cooling operation, and (b) adding the NOPR-proposed minimum ESP levels to Table 3 of Appendix M (this is the table that specifies minimum ESP), indicating that these minimum ESPs are for short duct systems. DOE proposes using the new term ''Short duct system'' rather than ''Multi-split system'' for these minimum ESPs because multi-circuit or mini-split systems could potentially also include similar short-ducted indoor units. DOE proposes a limitation in the level of ESP that eligible indoor units can deliver in order to prevent the potential abuse of the reduced ESP requirement mentioned by AHRI. DOE requests comment on these proposals, including the value of maximum ESP attainable by eligible systems.

DOE notes that in conjunction with the adopted portions of the AHRI Standard 1230–2010 , the following sections of the proposed test procedure found in Appendix M may apply to testing VRF multi-split systems: section 1 (definitions); section 3.12 (rounding of space conditioning capacities for reporting purposes); 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 (test unit installation requirements); Table 3 in section 3.1.4.1.1c (external static pressure requirements); section 3.1 except section 3.1.3 and 3.1.4 (general requirements of the testing procedure); sections 3.3, 3.4, and 3.5 (procedures for cooling-mode tests); sections 3.7, 3.8, 3.9, and 3.10 (procedures for heating-mode tests); section 3.13 (procedure for off mode average power rating); and section 4 (calculations of seasonal performance descriptors).

DOE requests comment on the incorporation by reference of AHRI 1230–2010, and in particular the specific sections of Appendix M and AHRI 1230–2010 that DOE proposes to apply to testing VRF systems.

3. Replacement of the Informative Guidance Table for Using the Federal Test Procedure

The intent of the set of four tables at the beginning of ''Section 2, Testing Conditions'' of the current test procedure (10 CFR part 430, subpart B, Appendix M) is to provide guidance to manufacturers regarding testing conditions, testing procedures, and calculations appropriate to a product class, system configuration, modulation capability, and special features of products. DOE recognizes that the current table format may be difficult to follow. Therefore, DOE has developed a more concise table and proposes using it in place of the current table. DOE requests comment on this proposed change and/or whether additional modifications to the new table could be implemented to further improve clarity.

4. Clarifying the Definition of a Mini-Split System

Current definitions in 10 CFR part 430, subpart B, Appendix M define a mini-split air conditioner and heat pump as ''a system that has a single outdoor section and one or more indoor sections, which cycle on and off in unison in response to a single indoor thermostat.'' When DOE introduced this definition, mini-split systems solely employed one or more non-ducted or short-duct wall-, ceiling-, or floor-mounted indoor units (*i.e.,* non-conventional units), and the market for mini-split products reflected such type and quantity of indoor units. It was common understanding that when testing or purchasing a mini-split system, the system would have a non-conventional indoor unit.

Nevertheless, DOE recognizes that further clarification and specificity in terminology would alleviate ambiguity in how to categorize mini-split products. To differentiate the two types of products, DOE proposes deleting the definition of mini-split air conditioners and heat pumps, and adding two definitions for: (1) Single-zone-multiple-coil split-system, representing a split-system that has one outdoor unit and that has two or more coil-only or blower coil indoor units connected with a single refrigeration circuit, where the indoor units operate in unison in response to a single indoor thermostat; and (2) single-split-system, representing a split-system that has one outdoor unit and that has one coil-only or blower coil indoor unit connected to its other component(s) with a single refrigeration circuit. DOE seeks comment on this proposal.

5. Clarifying the Definition of a Multi-Split System

A multiple-split (or multi-split) system is currently defined in 10 CFR part 430, subpart B, Appendix M as ''a split-system having two or more indoor units, which respond to multiple thermostats.'' Technologies exist on the market that operate like multi-split systems but incorporate multiple outdoor units into the same package. To clearly define what arrangement qualifies as a multi-split system, DOE proposes to clarify the definition of multi-split system to specify that multi-split systems are to have only one outdoor unit. (DOE notes that it proposes to separately define multi-circuit units as units that incorporate multiple outdoor units into the same package. This is discussed in section III.C.2.) Finally, DOE proposes to clarify that if a model of outdoor unit could be used both for single-zone-multiple-coil split-systems and for multi-split-systems, it should be tested as a multi-split system.

G. Test Procedure Reprint

The test procedure changes proposed in this SNOPR as well as in the June 2010 NOPR, April 2011 SNOPR, and October 2011 SNOPR occur throughout large portions of Appendix M to 10 CFR part 430 Subpart B. In order to improve clarity regarding the proposed test procedure, in the regulatory text for this SNOPR, DOE has reprinted the entirety of Appendix M, including all changes proposed in this SNOPR as well as those in the previous NOPR and SNOPRs that are still applicable. Table III.6 lists those proposals from the previous notices that appear without modification in this regulatory text reprint, and provides reference to the respective revised section(s) in the regulatory text. Table III.7 lists those proposals from the previous notices that either are proposed to be withdrawn or amended in this SNOPR or propose no amendments to the test procedure, and provides reference to the respective preamble section for the discussion of the revision, including stakeholder comments from the original proposal, and the revised section(s) in the regulatory text, if any. The proposed amendments to Appendix M would not change the rated values.

Because Appendix M1, as discussed in I.A, is substantially similar to Appendix M, DOE is only printing the proposed regulatory text for Appendix M1 where it differs from the proposed regulatory text for Appendix M. Proposed changes relevant to Appendix M1 are discussed in section III.H.

TABLE III.6—PROPOSALS FROM PRIOR NOTICES ADOPTED WITHOUT MODIFICATION IN THIS SNOPR

Section	Proposal to . . .	Reference	Action	Preamble discussion	Regulatory text location *
June 2010 NOPR					
A.7	Add Calculations for Sensible Heat Ratio.	75 FR 31229	Upheld	III.I.5	3.3c, 4.6.
A.10	Add Definitions Terms Regarding Standby Power.	75 FR 31231	Upheld	None	Definitions.
B.4	Allow a Wider Tolerance on Air Volume Rate To Yield More Repeatable Laboratory Setups.	75 FR 31233	Upheld	None	3.1.4.1.1a.4b.
B.5	Change the Magnitude of the Test Operating Tolerance Specified for the External Resistance to Airflow.	75 FR 31234	Upheld	None	3.3d Table, 3.5h Table, 3.7a Table, 3.8.1 Table, 3.9f Table.
	Change the Magnitude of the Test Operating Tolerance Specified for the Nozzle Pressure Drop.	75 FR 31234	Upheld	None	3.3d Table, 3.5h Table, 3.7a Table, 3.8.1 Table.
B.6	Modify Refrigerant Charging Procedures: Disallow Charge Manipulation after the Initial Charge.	75 FR 31234	Upheld	III.E.7	2.2.5.
B.7	Require All Tests be Performed with the Same Refrigerant Charge Amount.	75 FR 31235, 31250.	Upheld	III.F.1	2.2.5.8.
B.8	When Determining the Cyclic Degradation Coefficient CD, Correct the Indoor-Side Temperature Sensors Used During the Cyclic Test To Align With the Temperature Sensors Used During the Companion Steady-State Test, If Applicable: Equation.	75 FR 31235	Upheld	None	3.4c, 3.5i, 3.7e, 3.8.
	When Determining the Cyclic Degradation Coefficient CD, Correct the Indoor-Side Temperature Sensors Used During the Cyclic Test To Align With the Temperature Sensors Used During the Companion Steady-State Test, If Applicable: Sampling Rate.	75 FR 31236	Upheld	None	3.3b, 3.7a, 3.9e, 3.11.1.1, 3.11.1.3, 3.11.2a.
B.9	Clarify Inputs for the Demand Defrost Credit Equation.	75 FR 31236	Upheld	None	3.9.2a.
B.10	Add Calculations for Sensible Heat Ratio.	75 FR 31237	Upheld	III.I.5	3.3c, 4.6.
B.11	Incorporate Changes To Cover Testing and Rating of Ducted Systems Having More Than One Indoor Blower.	75 FR 31237	Upheld	III.C.3	2.2.3, 2.2.3b, 2.4.1b, 3.1.4.1.1d, 3.1.4.2e, 3.1.4.4.2d, 3.1.4.5.2f, 3.2.2, 3.2.2.1, 3.6.2, 3.2.6, 3.6.7, 4.1.5, 4.1.5.1, 4.1.5.2, 4.2.7, 4.2.7.1, 4.2.7.2, 3.2.2.2 Table, 3.6.2 Table.
B.12	Add Changes To Cover Triple-Capacity, Northern Heat Pumps.	75 FR 31238	Upheld	III.C.4	3.6.6, 4.2.6.
B.13	Specify Requirements for the Low-Voltage Transformer Used When Testing for Off-Mode Power Consumption.	75 FR 31238	Upheld	III.F.1	2.2d.

TABLE III.6—PROPOSALS FROM PRIOR NOTICES ADOPTED WITHOUT MODIFICATION IN THIS SNOPR—Continued

Section	Proposal to . . .	Reference	Action	Preamble discussion	Regulatory text location *
B.14	Add Testing Procedures and Calculations for Off Mode Power Consumption.	75 FR 31238	Upheld	III.D	Definitions, 3.13, 4.3, 4.4.
		April 2011 SNOPR			
III.A	Revise Test Methods and Calculations for Off-Mode Power and Energy Consumption.	76 FR 18107	Upheld	III.D	Definitions, 3.13, 4.3, 4.4.
III.B	Revise Requirements for Selecting the Low-Voltage Transformer Used During Off-Mode Test(s).	76 FR 18109	Upheld	III.F.1	2.2d.
III.D	Add Calculation of the Energy Efficiency Ratio for Cooling Mode Steady-State Tests.	76 FR 18111	Upheld	None	4.7.
III.E	Revise Off-Mode Performance Ratings	75 FR 31238	Upheld	III.D	Definitions, 3.13, 4.3, 4.4.
		October 2011 SNOPR			
III.A	Reduce Testing Burden and Complexity.	76 FR 65618	Upheld	III.D	Definitions, 3.13, 4.3, 4.4.
III.B	Add Provisions for Individual Component Testing.	76 FR 65619	Upheld	III.D	Definitions, 3.13, 4.3, 4.4.
III.C	Add Provisions for Length of Shoulder and Heating Seasons.	76 FR 65620	Upheld	III.D	Definitions, 3.13, 4.3, 4.4.
III.D	Revise Test Methods and Calculations for Off-Mode Power and Energy Consumption.	76 FR 65620	Upheld	III.D	Definitions, 3.13, 4.3, 4.4.
III.D.1	Add Provisions for Large Tonnage Systems.	76 FR 65621	Upheld	III.D	Definitions, 3.13, 4.3, 4.4.
III.D.2	Add Requirements for Multi-Compressor Systems.	76 FR 65622	Upheld	III.D	Definitions, 3.13, 4.3, 4.4.

* Section numbers in this column refer to the proposed Appendix M test procedure in this notice.

TABLE III.7—PROPOSALS FROM PRIOR NOTICES WITHDRAWN OR AMENDED IN THIS SNOPR OR PROPOSED NO CHANGE TO THE TEST PROCEDURE

Section	Proposal to . . .	Reference	Action	Preamble discussion	Regulatory text location *
		June 2010 NOPR			
A.1	Set a Schedule for Coordinating the Publication of the Test Procedure and Energy Conservation Standards.	75 FR 31227	No Change **	None	None.
A.2	Bench Testing of Third-Party Coils	75 FR 31227	No Change **	None	None.
A.3	No Change to Default Values for Fan Power.	75 FR 31227	Amended	III.H.3	*10 CFR Part 430, Subpart B, Appendix M1* 3.3d, 3.5.1, 3.7c, 3.9.1b.
A.4	No Change to External Static Pressure Values.	75 FR 31228	Amended	III.H.1	*10 CFR Part 430, Subpart B, Appendix M1* 3.1.4.1.1c. Table.
A.5	No Conversion to Wet-Coil Cyclic Testing.	75 FR 31228	No Change **	III.I.4	None.
A.6	No Change to Test Procedure for Testing Systems with "Inverter-Driven Compressor Technology".	75 FR 31229	No Change **	None	None.
A.8	Regional Rating Procedure	75 FR 31229	Withdrawn †	None	None.
A.9	Modify Definition of Tested Combination.	75 FR 31230	Amended	III.F.2	*10 CFR 430.2* Definitions.
	Add Minimum ESP for Short Duct Systems.	75 FR 31230	Amended	III.F.2	3.1.4.1.1c. Table.
	Clarify That Optional Tests May Be Conducted without Forfeiting Use of the Default Value(s).	75 FR 31230	Withdrawn †	None	None.
B.1	Modify the Definition of "Tested Combination".	75 FR 31231	Amended	III.F.2	*10 CFR 430.2* Definitions.
B.2	Add Minimum ESP for Short Duct Systems.	75 FR 31232	Amended	III.F.2	3.1.4.1.1c. Table.
	Add Indoor Unit Design Characteristics for Limiting Application of Minimum ESP for Short Duct Systems.	75 FR 31232	Amended	III.F.2	3.1.4.1.1c. Table header.
B.3	Clarify That Optional Tests May Be Conducted Without Forfeiting Use of the Default Value(s).	75 FR 31233	Withdrawn †	None	None.
B.6	No Adoption of Requirement of Manufacturer Sign-Off after Charging Refrigerant.	75 FR 31234	No Change **	None	None.
B.7	Allow Interactions between Manufacturers and Third-Party Testing Laboratory.	75 FR 31235	Withdrawn	III.F	None.

TABLE III.7—PROPOSALS FROM PRIOR NOTICES WITHDRAWN OR AMENDED IN THIS SNOPR OR PROPOSED NO CHANGE TO THE TEST PROCEDURE—Continued

Section	Proposal to . . .	Reference	Action	Preamble discussion	Regulatory text location *
B.15	Add Parameters for Establishing Regional Standards.	75 FR 31239	Withdrawn †	None	None.
B.15a	Use a Bin Method for Single-Speed SEER Calculations for the Hot-Dry Region and National Rating.	75 FR 31240	Withdrawn †	None	None.
B.15b	Add New Hot-Dry Region Bin Data	75 FR 31240	Withdrawn †	None	None.
B.15c	Add Optional Testing at the A and B Test Conditions With the Unit in a Hot-Dry Region Setup.	75 FR 31241	Withdrawn †	None	None.
B.15d	Add a New Equation for Building Load Line in the Hot-Dry Region.	75 FR 31242	Withdrawn †	None	None.
B.16	Add References to ASHRAE 116–1995 for Equations That Calculate SEER and HSPF for Variable Speed Systems.	75 FR 31243	Withdrawn	III.*E*.13	None.
B.17	Update Test Procedure References	75 FR 31243	Amended	III.*E*.12	*10 CFR 430.3* Definitions.
April 2011 SNOPR					
III.C	Withdraw of the Proposal To Add the New Regional Performance Metric SEER Hot-Dry.	76 FR 18110	No Change **	None	None.
October 2011 SNOPR					
Proposals are Upheld					

* Section numbers in this column refer to the proposed Appendix M test procedure in this Notice, unless otherwise specified.
** These items were discussed in the NOPR or SNOPR but did not propose changes to the test procedure.
† Associated proposals regarding the SEER Hot-Dry metric, as indicated, are withdrawn because DOE withdrew the SEER Hot-Dry metric in the April 2011 SNOPR. 76 FR 18110.

H. Improving Field Representativeness of the Test Procedure

DOE received comments from stakeholders during the public comment period following the November 2014 ECS RFI requesting changes to the test procedure that would improve field representativeness. Such changes would impact the rated efficiency of central air conditioners and heat pumps. As discussed in section I.A, any amendments proposed in this SNOPR that would alter the measured efficiency, as represented in the regulating metrics of EER, SEER, and HSPF, are proposed as part of a new Appendix M1 to Subpart B of 10 CFR part 430. The test procedure changes proposed as part of a new Appendix M1, if adopted, would not become mandatory until the existing energy conservation standards are revised to account for the changes to rated values. (42 U.S.C. 6293(e)(2)) These changes, including the relevant stakeholder comments, are discussed in the following subsections.

1. Minimum External Static Pressure Requirements for Conventional Central Air Conditioners and Heat Pumps

Most of the central air conditioners and heat pumps used 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 central air conditioner or heat pump 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 ESP imposed by the ductwork affects the power consumed by the indoor blower, and therefore also affects the SEER and/or HSPF of a central air conditioner or heat pump.

The current DOE test procedure [22] stipulates that certification tests for central air conditioners and heat pumps which are not short duct systems (see section III.F.2) or small-duct, high-velocity systems [23] (*i.e.,* conventional central air conditioners and heat pumps) must be performed with an ESP 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.

[22] Table 3 of 10 CFR 430 Subpart B Appendix M

[23] 10 CFR 430 Subpart B Appendix M Section 1. Definitions defines a small-duct, high-velocity system as a system that contains a blower and indoor coil combination that is designed for, and produces, at least 1.2 inches (of water) of external static pressure when operated at the full-load air volume rate of 220–350 cfm per rated ton of cooling. When applied in the field, small-duct products use high-velocity room outlets (*i.e.,* generally greater than 1000 fpm) having less than 6.0 square inches of free area.

DOE decided in the June 2010 NOPR not to propose revisions to minimum external static pressure requirements, stating that new values and a consensus standard were not readily available. 75 FR 13223, 31228 (June 2, 2010). NEEA responded during the subsequent public comment period that current ESP minimums were too low and recommended DOE adopt an ESP test requirement of 0.5 in. wc. (NEEA, No. 7 at p. 3). Earthjustice commented that retention of the existing ESP values is not supported by evidence. (Earthjustice, No. 15 at pp. 1–2). Southern California Edison (SCE), the Southern California Gas Company (SCGC), and San Diego Gas and Electric (SDGE) (together, the Joint California Utilities) included with its comments two studies showing field measurements of ESP with an average of 0.5–0.8 in. w.c and urged the Department to adopt an external static pressure test point of 0.5 in. wc. (Joint California Utilities, No. 9 at p. 3). ACEEE suggested that field data is available for DOE to consider new values of ESP. (ACEEE, No. 8 at pp. 2–3).

Stakeholders also commented in response to the November 2014 ECS RFI that the current requirements for minimum ESP are unrepresentative of field practice. PG&E commented that the ESP for central air conditioners and heat pumps needs to be set at 0.5 in. wc. or higher for ducted systems. (Docket No. EERE–2014–BT–STD–0048, PG&E, No. 15 at p. 3) ACEEE advocated similarly: Default ESP used in the current federal test procedure should be raised from the current 0.1 to 0.2 in. wc. to at least 0.5 in. wc. to represent field practice. (*Id.;* ACEEE, No. 21 at p. 2) ASAP & ASE & NRDC commented that the ESP in the current test procedure is unrealistically low, adding that DOE should reference to the ESP values adopted by the recently finalized furnace fan rulemaking which has an ESP value of 0.5 in. wc.[24] (*Id.;* ASAP & ASE & NRDC, No. 20 at p. 1).

Central air conditioners and heat pumps are generally equipped with air filters when used in the field. Section 3.1.4.1.1c of 10 CFR part 430, subpart B, Appendix M requires that any unit tested without an air filter installed be tested with ESP increased by 0.08 in. wc. to represent the filter pressure drop. University of Alabama commented during the public comment period of the November 2014 ECS RFI that the actual combined ESP requirements in the field are typically 3 to 5 times greater with more effective filters and typical duct designs. The unrealistically low rating conditions result in little incentive for manufacturers to incorporate improved fan wheel designs. Improvements in SEER gained by replacing inexpensive forward-curve fan wheels will be negligible but demand and energy savings in actual installations will be significant. (Docket No. EERE–2014–BT–STD–0048, University of Alabama, No. 6 at p. 1).

Furnaces use the same ductwork as central air conditioners and heat pumps to distribute air in a residence. NEEA & NPCC commented that the ESP selected for testing of furnace fans is substantially higher than the 0.1 to 0.2 in. wc. prescribed by the federal CAC/HP test procedure. They also mentioned that field data from Pacific Northwest shows that the minimum required ESP is 0.5 in. wc. regardless of system capacity. NEEA & NPCC recommended that the ESP requirement for measurement of cooling efficiency be close to 0.6 in. wc. because air volume rates for cooling (and heating for heat pumps) are greater than typical furnace heating air volume rates. However, they suggested DOE adopt the ESP level required for testing of furnace fans as a simple approach. (Docket No. EERE–2014–BT–STD–0048, NEEA & NPCC, No. 19 at p. 2).

In response to stakeholder comment over multiple public meetings that the minimum ESP values intended for testing are indeed unrepresentative of the ESPs in field installations, and field studies indeed demonstrating the same, DOE proposes in this SNOPR revising the ESP requirements for most central air conditioners and heat pumps, *e.g.,* those that do not meet the proposed requirements for short duct systems or the established requirements for small-duct, high-velocity (SDHV) systems.

DOE is not considering revising the minimum ESP requirement for SDHV systems. DOE is, however, proposing to establish a new category of ducted systems, short duct systems, which would have lower ESP requirements for testing—this is discussed in section III.F.2.

To meet the requirement set forth in 42 U.S.C. 6293(b)(3) providing that test procedures be reasonably designed to produce test results which measure energy efficiency of a covered product during a representative average period of use, DOE reviewed available field data to determine appropriate ESP values. DOE gathered field studies and research reports, where publically available, to estimate field ESPs. 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 central air conditioner or heat pump systems in residences, with the data collected varying by location, representation of system static pressure measurements, and equipment's age and ductwork arrangement, vintage, and air-tightness. 79 FR 500 (Jan. 3, 2014). DOE observed measured ESPs to range from 0.20 to 0.70 in. wc. DOE used three statistical approaches to determine an average representation of ESP from the range of ESPs: a simple-average approach, a sample-size-exclusion approach, and a most-samples approach. DOE then performed reconciliation, through equal weighting of the results from the three approaches, to obtain a ''middle ground'' value of 0.32 in. wc. as the ESP representing a typical residence with a new space conditioning system.

DOE is aware that units used in certification laboratory testing have not aged and are thus not representative of seasoned systems in the field. Namely, dust, dander, and other airborne particulates, commonly deposited as foulant onto in-duct components in field installations, are unaccounted for in controlled testing environments. Foulant fills air gaps of the air filter and evaporator coil and restricts air volume rate, thus increasing ESP. This occurrence is not accounted for in certification testing environments. Therefore, DOE included an ESP adder for component foulant build-up to the test procedure to better reflect a representative average period of use. To determine the value of this adder, DOE examined the aforementioned field studies that captured the ESP contribution from vintage, and certainly fouled, air filters and evaporator coils. From the contributing studies, DOE estimates an average pressure drop due to the filter's foulant of 0.13 in. wc. based on the difference in static pressure contributions between fouled filters and clean filters. DOE also examined publicly available reference material and research to determine the pressure drop from the build-up of foulant on evaporator coils. Three resources in the public domain were identified that documented the impact of evaporator coil fouling on ESP in applications.[25] From this literature, DOE estimates an average pressure drop resulting from evaporator coil fouling of 0.07 in. wc. These additional pressure drops result in a total of 0.20 in. wc. being added to the revised ESP value, as mentioned. DOE seeks comment on its proposal to include in the ESP requirement a pressure drop contribution associated with average typical filter and indoor coil fouling levels and its use of residential-based indoor coil and filter fouling pressure drop data to estimate the appropriate ESP contribution. DOE also requests any data that would validate the proposed ESP contributions or suggestions of adjustments that should be made to improve representativeness of the values in this proposal. DOE notes that addition of these pressure drop contributions is consistent with the approach adopted for testing of furnace fans, which are tested without the filter and air conditioning coil, and for which the ESP selected for testing reflects the field fouling associated with these components.

Consistent with the current motivation in current certification procedures to promulgate policy that represents the majority of products in the field (10 CFR 429.16(a)(2)(ii)), DOE selected the capacity with the largest volume of retail sales, 3 tons, as the rated cooling capacity category to adopt

[24] Docket No. EERE–2010–BT–TP–0010–0043.

[25] 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.

the minimum ESP requirement based on the field data and the adjustments. For the other cooling capacity categories, NEEA commented that ESP should not vary with capacity. (NEEA, No. 7 at p. 3). DOE considered the stakeholder comment and the higher ESPs indicative of larger homes, and proposes a compromise approach to use the current 0.05 in. wc. step variation among capacities.

In conclusion, DOE proposes 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 ESP requirements of 0.45 in. wc. for units with rated cooling capacity of 28,800 Btu/h or less; 0.50 in. wc. for units with rated cooling capacity of 29,000 Btu/h or more and 42,500 Btu/h or less; and 0.55 in. wc. for units with rated cooling capacity of 43,000 Btu/h or more. (DOE is not making such a revision in 10 CFR part 430, subpart B, Appendix M.) The proposed minimum ESP requirements are shown in Table III.8. DOE is aware that such changes will impact the certification ratings SEER, HSPF, and EER and is addressing such impact in the current energy conservation standards rulemaking.[26] DOE requests comment on these proposals.

TABLE III.8—PROPOSED MINIMUM ESP REQUIREMENTS FOR CENTRAL AIR CONDITIONERS AND HEAT PUMPS OTHER THAN MULTI-SPLIT SYSTEMS AND SMALL-DUCT, HIGH-VELOCITY SYSTEMS[27]

Rated cooling or heating capacity (Btu/h)	Minimum ESP (in. wc.)
Up Thru 28,800	0.45
29,000 to 42,500	0.50
43,000 and Above	0.55

2. Minimum External Static Pressure Adjustment for Blower Coil Systems Tested With Condensing Furnaces

As discussed in section III.H.1, DOE proposes to increase the minimum ESP required for testing blower coil central air conditioners and heat pumps. DOE notes that there are three different blower coil configurations: (1) An air handling unit which is a single piece of equipment containing a blower and a coil; (2) a coil paired with a separately-housed modular blower; (3) a coil paired with a separate furnace. The existing federal test procedure for central air conditioners and heat pumps does not require different minimum ESPs for these different blower coil configurations, even though the heat exchanger of a furnace may impose additional pressure drop on the air stream. The additional pressure drop can contribute to higher blower power, which may negatively affect the performance rating for a central air conditioner. Further, condensing furnaces, which have more heat transfer surface exposed to the flowing air than non-condensing furnaces, may impose even more pressure drop.

Given the potential disadvantage associated with the rating of an air conditioner with a condensing furnace as the designated air mover, DOE proposes an adjustment to the minimum external static pressure requirement for a rated blower coil combination using a condensing furnace as the air mover in order to mitigate the impact on air-conditioner ratings of furnace efficiency improvements. To aid the selection of representative ESP adjustments, DOE conducted laboratory testing for two condensing and three non-condensing furnaces to determine typical furnace heat exchanger pressure drop levels. DOE measured the pressure rise provided by each furnace when operating in the maximum airflow-control setting at a representative air volume rate, first as delivered and then with the furnace heat exchanger(s) removed. DOE measured average furnace heat exchanger pressure drop equal to 0.47 in. wc. for the condensing furnaces and 0.27 in. wc. for the non-condensing furnaces. The data suggest that condensing furnace pressure drop is roughly 0.2 in. wc. higher than non-condensing furnace pressure drop. However, DOE notes that cooling operation may be at lower air volume rates than the maximum cooling air volume rate used in the tests, since furnaces can be paired with air-conditioners having a range of capacities. Based on these results, DOE proposes to include in Appendix M1 of 10 CFR part 430 Subpart B a requirement of a downward adjustment of the required ESP equal to 0.1 in. wc. when testing an air conditioner in a blower-coil configuration (or single-package configuration) in which a condensing furnace is in the air flow path. DOE is not making such a revision in 10 CFR part 430, subpart B, Appendix M. DOE requests comments on this proposal.

3. Default Fan Power for Coil-Only Systems

The default fan power is used to represent fan power input when testing coil-only air conditioners, which do not include their own fans.[28] The default was discussed in the June 2010 NOPR, in which DOE did not propose to revise it due to uncertainty on whether higher default values better represent field installations. 75 FR 31227 (June 2, 2010). In response to the June 2010 NOPR, Earthjustice commented that the existing default fan power for coil-only units in the DOE test procedure is not supported by substantial evidence. ESPs measured from field data show significant higher values than the requirements in the existing test procedure. (Earthjustice, No. 15 at p. 2) However, to be consistent with the increase in ESP used for testing blower coil products, as discussed in section III.H.1, this notice proposes updating the default fan power (hereinafter referred to as ''the default value'') used for testing coil-only products. DOE used circulation blower electrical power data collected for the furnace fan rulemaking (79 FR 38129, July 3, 2014) to determine an appropriate default value for coil-only products.

DOE collected circulation blower consumption data from product literature, testing, and exchanges with manufacturers as part of the furnace fan rulemaking. These data are often provided in product literature in the form of tables listing air volume rate and circulation blower electrical power input across a range of ESP for each of the blower's airflow-control settings. DOE collected such data for over 100 furnace fans of non-weatherized gas furnace products for the furnace fan rulemaking. DOE used this database to calculate an appropriate default value to represent circulation blower electrical power for typical field operating conditions for air conditioning, consistent with the required ESP values proposed for blower coil split-systems. From the perspective of the furnace providing the air movement, the ESP is higher than that required for testing blower coil systems to account for the cooling coil and the air filter that would be installed for a coil-only test, since furnace airflow performance is determined without the coil and filter installed. DOE used pressure drop associated with the filter equal to 0.08 in. wc., consistent with the required ESP addition when testing without an air filter installed. In addition, DOE

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

[27] DOE did not increase the ESP requirement for small-duct, high-velocity units because the existing values in the test procedure represent field operations.

[28] See 10 CFR 430 Subpart B Appendix M section 3.3.d.

estimates that the typical pressure drop associated with an indoor coil is 0.16 in. wc. DOE added the resulting sum, 0.24 in. wc., to the required ESP levels for testing a blower coil system to obtain the ESP levels it used to calculate the power input for furnaces in the furnace fan database.

The air volume rate at which central air conditioner and heat pumps are required to operate according to the DOE test procedure varies with capacity. Typically, units are tested and operated in the field while providing between 350 and 450 cfm per ton of cooling capacity. For the purpose of determining the appropriate default value, DOE investigated furnace fan performance at the ESP values discussed above while providing 400 cfm per ton of cooling capacity.

A product that incorporates a furnace fan can often be paired with one of multiple air conditioners of varying cooling capacities, depending on the installation. For example, a non-weatherized gas furnace model may be designed to be paired with either a 2, 3, or 4 ton coil-only indoor unit. These combinations are possible because the circulation blower in the furnace has multiple airflow-control settings. Multiple airflow-control settings allow the furnace to be configured to provide the target air volume rate for either 2, 3, or 4 ton coil-only indoor units by designating a different airflow-control setting for cooling. For furnaces with multiple such airflow-control settings that are suitable for air conditioning units, DOE calculated fan power for each of these settings since they all represent valid field operating conditions.

DOE then organized the results of the calculations by blower motor technology used and manufacturer, averaging over both to calculate an overall average default value. The distribution of motor technology follows projected distribution of motors used in furnaces in the field in the year 2021. By this time, there will be some small impact on this distribution associated with the furnace fan rule. DOE averaged by manufacturer based on market share.

The default fan power in the existing DOE test procedure does not vary among different capacities. DOE maintains the same approach for the adjusted default fan power. Using the aforementioned methodology, DOE calculated the adjusted default fan power to be 441 W/1000 cfm and proposes 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 cfm. DOE is not making such replacements in Appendix M of 10 CFR part 430 Subpart B.

4. Revised Heating Load Line

In the current test procedure, the heating seasonal performance factor (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 for each climate region used to represent the heating season—for the HSPF rating, this is Region IV. The amount of heating delivered at each temperature increases as the temperature decreases. This amount is dependent on the size of the house that the unit is heating. In addition, there is a relationship between the size of the house and the capacity of the heat pump selected to heat it. 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 heating load is also proportional to the difference between 65 °F and the outdoor temperature. The resulting relationship between heating load and outdoor temperature is called the heating load line—it slopes downward from low temperatures, dropping to zero at 65 °F. The slope of the heating load line affects HSPF both by dictating the heat pump capacity level used by two-capacity or variable-capacity 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 line. The current test procedure defines two 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.[29]

Studies have indicated that the current HSPF test and calculation procedure overestimates ratings because the current minimum heating load line is too low compared to real world situations.[30] 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 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. (NEEA & NPCC, No. 19 at p. 2)

DOE agrees with NEEA and NPCC and notes that the heating balance point determined for a typical heat pump using the current minimum heating load line 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. 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,[31] 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).[32] For these reasons, DOE reviewed the choice of heating load line for HSPF ratings and proposes to modify it.

[29] See 10 CFR 430 Subpart B Appendix M Section 1. Definitions.

[30] 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.

[31] 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.

[32] 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.

As part of this review, ORNL conducted building load analysis using the EnergyPlus simulation tool on a prototype residential house based on the 2006 IECC code and summarized the study in a report to DOE.[33] In general, the studies indicate that a heating load level closer to the maximum load line and with a lower zero load ambient temperature is more representative than the minimum load line presently used for HSPF rating values.

Based on the results from the ORNL studies, DOE proposes the new heating load line equation to be used for calculation of HSPF as:

$$BL(T_j) = \frac{(T_{zl} - T_j)}{T_{zl} - T_{OD}} \cdot DHR$$

Where

T_j = the outdoor bin temperature, °F
T_{OD} = the outdoor design temperature, °F
DHR = the design heating requirement, Btu/h
T_{zl} = the zero load temperature, °F

The proposed equation includes the following changes from the current heating load line used for calculation of HSPF:[34]

• The equation form does not differ by region;
• The zero load temperature varies by climate region, as shown in Table III.6, and for Region IV is at 55 °F, which is closer to what occurs in the field;
• The design heating requirement is a function of the adjustment factor, or the slope of the heating load line, and is 1.3 rather than 0.77; and
• The heating load is tied with the nominal heat pump cooling capacity used for unit sizing rather than the heating capacity (except for heating-only heat pumps).

Revised heating load hours were determined for the new zero load temperatures of each climate region. The revised heating load hours are given below in Table III.9.

TABLE III.9—GENERALIZED CLIMATIC REGIONAL INFORMATION

Region No.	I	II	III	IV	V	VI
Heating Load Hours	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 proposed heating load line simulates the actual building load in different climate regions, so the maximum and minimum heating load lines of the current test procedure are not needed. The ORNL building simulation results show that the same equation matching the building load applies well to all regions. DOE therefore proposes eliminating maximum and minimum DHR definitions.

DOE believes that it is more appropriate to base the heating load line on nominal cooling capacity rather than nominal heating capacity, because heat pumps are generally sized based on a residence's cooling load. For the special case of heating-only heat pumps, which clearly would be sized based on heating capacity rather than cooling capacity, DOE proposes that the nominal heating capacity at 47 °F would replace the cooling capacity in the proposed load line equation. This is consistent with the building heating load analysis.

The proposed altered heating load line would alter the measurement of HSPF. DOE estimates that HSPF would be reduced on average about 16 percent for single speed heat pumps and two capacity heat pumps. The impact on the measurement for variable-speed heat pumps is discussed in section III.H.5. 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 become effective until the compliance date of any new energy conservation standard.

In response to the November 2014 ECS RFI, University of Alabama commented that the current test procedure for central air conditioners and heat pumps include cooling bin data at 67 °F and heating bin data at 62 °F. This results in a dead band of 5 °F. Because the current test procedure prescribes the indoor temperature set point to be 70 °F for heating, and 80 °F for cooling, the temperature difference of 10 °F is inconsistent with the dead band of 5 °F from the temperature bin. University of Alabama also suggested adopting 62 °F and 52 °F as the zero load points for cooling and heating modes, respectively. (University of Alabama, No. 6 at p. 1–2)

The indoor dry bulb set temperature of 70 °F for heating and 80 °F for cooling represent field set temperature for central air conditioners and heat pumps in a typical residential household. These two temperatures are also used in other product or equipment classes such as the commercial unitary air conditioners and heat pumps.[35]

In this notice, DOE proposes to revise the heating load line which shifts the heating balance point and zero load point to lower ambient temperatures. These amendments reflect more representative unit field operations and energy use characteristics. The revised heating load line lowers the zero load point for heating in region IV to 55 °F. Given the cooling-mode zero load point of 65 °F, the proposed change would increase the temperature difference between the heating and cooling zero load points to 10 °F, which equals the temperature difference between cooling and heating modes thermostat set points. The proposal would hence make these values more consistent with each other, whether or not this consistency is necessary for accuracy of the test procedure.

As a result of this proposed heating load line change, DOE also proposes that cyclic testing for variable speed heat pumps be run at 47 °F instead of 62 °F, as required by the current test procedure (see Appendix M, section 3.6.4 Table 11). The test would still be

[33] 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).

[34] Most commonly used heating load equation based on minimum design heating requirement and region IV: $Q_h(47) * 0.77*(65–Tj)/60$.

[35] See ANSI/AHRI Standard 340/360–2007 with Addenda 1 and 2, Performance rating of commercial and industry unitary air-conditioning and heat pump equipment.

conducted using minimum compressor speed. With the modified heating load line there would be no heat pump operation at 62 °F, so cyclic testing at 47 °F would be more appropriate. DOE seeks comment on this proposal.

DOE proposes to make the changes to the test procedure as mentioned in this subsection only in Appendix M1 of 10 CFR part 430 Subpart B, and is not making such changes to Appendix M of the same Part and Subpart.

5. Revised Heating Mode Test Procedure for Products Equipped With Variable-Speed Compressors

A recent Bonneville Power Administration (BPA) commissioned study done by Ecotope, Inc., and an Oak Ridge National Lab (ORNL)/Tennessee Valley Authority (TVA) field test found the heating performance of a variable speed heat pump, based on field data, is much lower than the rated HSPF.[36] Therefore, DOE revisited the heating season ratings procedure for variable speed heat pumps, is are found in section 4.2.4 of Appendix M of 10 CFR part 430 Subpart B.

The HSPF is calculated by evaluating the energy usage of both the heat pump unit (reverse refrigeration cycle) and the resistive heat component when matching the dwelling heating load at each outdoor bin temperature. Currently, both the minimum and the maximum capacities are calculated at each outdoor bin temperature to determine whether the variable speed heat pump capacity can or cannot meet the building heating load. At an outdoor bin temperature where the heat pump minimum capacity is higher than the building heating load, the heat pump cycles at minimum speed. The energy usage at such outdoor bin temperature is determined by the energy usage of the heat pump at minimum speed and the unit cyclic loss. At an outdoor bin temperature where the heat pump maximum capacity is lower than the building heating load, the heat pump operates at maximum speed. The energy usage at such outdoor bin temperature is determined by the energy usage of the heat pump at maximum speed and of the additional resistive heat required to meet the building load.

In the current test procedure, the capacity and the corresponding energy usage at minimum speeds are determined by the two minimum speed tests at 47 °F and 62 °F (outdoor temperature[37]), assuming the capacity and energy usage is linear to the outdoor temperature and the compressor speed does not change with the outdoor temperature. The capacity and the corresponding energy usage at maximum speeds are determined by the two maximum speed tests at 47 °F and at 17 °F, assuming the compressor speed does not change with the outdoor temperature. Both the minimum and the maximum capacities and energy usages are also used to estimate the heat pump operating capacity and energy usage when the heat pump operates at an intermediate speed to match the building heating load.

In reviewing these calculations, DOE compared the efficiencies (capacity divided by energy usage; at maximum speed, intermediate speed, and minimum speed at ambient temperatures representing the heating season) calculated using the method in current test procedure to the efficiencies tested in the lab at each of the 5 °F bin temperatures representing the heating season, and found two discrepancies where the efficiencies are not predicted accurately by the test procedure.

The first discrepancy occurs only for the variable speed heat pump that prevents minimum speed operation at outdoor temperatures below 47 °F. In the mid-range outdoor temperature range (17–47 °F), the efficiencies are over-predicted. The cause of this over-prediction is that the unit's actual minimum capacity is higher than the calculated minimum capacity in the range of outdoor temperature 17–47 °F. The calculated minimum capacity is based on the assumption that the unit can operate at the minimum speed in this range, which is not true with such units.

DOE considered two alternative methods to provide more accurate efficiency predictions for mid-range outdoor temperatures. In the first method, the minimum capacity and the corresponding energy usage for outdoor temperatures lower than 47 °F would be determined by the minimum speed tests at 47 °F and the intermediate speed test at 35 °F, which are both required test points in current test procedure. The new calculation method results in the capacity and energy usage more representative of the unit operation performance in the temperature region 35–47 °F. The HSPF calculated with this option agrees with the tested HSPF within 6%. This option does not require additional testing beyond what is required in the current test procedure.

In the second method, the minimum capacity and the corresponding energy usage for outdoor temperature lower than 47 °F would be determined by minimum speed tests at 47 °F and at 35 °F, where the test point of minimum speed at 35 °F is an additional test point that is not required in the current test procedure. In addition, the intermediate capacity and the corresponding energy usage would be modified for more accurate efficiency prediction at the outdoor temperature range 17–35 °F. This is done by defining the medium speed test as the average of the maximum and minimum speed and using the medium speed test at 17 °F and the intermediate speed test at 35 °F to determine the intermediate capacity and the corresponding energy usage, where the test at the medium speed at 17 °F is a test point not required in the current test procedure. With this method, the unit's calculated performance is well matched with the unit's actual operation in the outdoor temperature region 17–35 °F. The HSPF calculated with this option aligns with the tested HSPF within 2%. However, this option requires two additional test points, medium speed at 17 °F and minimum speed at 35 °F, which adds test burden for manufacturers.

After considering these two alternative methods with regard to the current test procedure, DOE further evaluated the impact of the proposed heating load line change (see section III.H.4) on the variable speed HSPF rating. DOE found that efficiencies calculated with the modified heating load line and with the current variable speed heat pump rating method match rather closely with those calculated from a more detailed set of test data at each outdoor bin temperature. The calculated HSPFs agree within 1 percent. Use of the proposed load line greatly reduces the error in the test procedure calculation from the speed limiting controls at ambient temperatures below 47 °F. The net effect is that the ratings calculation approach using the proposed load line with the current test points gives results close to those with more detailed data sets. However, because this also removes an artificial HSPF benefit that such units were obtaining, the net reduction in rated HSPF for such units could be as much as 26%.[38] DOE believes that this indicates that the modified heating load

[36] Larson, Ben, Bob Davis, Jeffrey Uslan, and Lucinda Gilman, 2013. *Variable Capacity Heat Pump Field Study, Final Report,* Ecotope, Inc., Bonneville Power Administration, August.

Munk, J.D., Halford, C., and Jackson, R.K., 2013. *Component and System Level Research of Variable Capacity Heat Pumps,* ORNL/TM–2013/36, August.

[37] All temperatures in section III.H.5, if not noted otherwise, mean outdoor temperature.

[38] 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).

line is sufficient to address the HSPF over-prediction issue for the variable speed heat pumps. Therefore, at this time, DOE does not propose changes specifically to the variable speed test points or heating calculations in the proposed Appendix M1. However, DOE notes that should stakeholder comments on this notice provide sufficient justification to retract the proposal to adopt the proposed modified heating load line, DOE would instead adopt, as part of Appendix M1, modifications to the variable speed heating calculations for units that prevent minimum speed operation. DOE requests comment on whether, in the case that the proposed heating load line is not adopted, DOE should modify the HSPF rating procedure for variable speed heat pumps using option 1, which is less accurate but has no additional test burden, or option 2, which is more accurate but with higher test burden.

The second potential discrepancy between the efficiencies (capacity divided by energy usage) calculated using the method in the current test procedure with the efficiencies tested in the lab at each outdoor bin temperature occurs at temperatures lower than 17 °F, where the test procedure assumes the heat pump operates at the maximum speed. The capacity and the corresponding energy usage at maximum speed at different outdoor bin temperatures are determined by the two maximum speed tests at 47 °F and at 17 °F, assuming the compressor speed does not change with the outdoor temperature. However, DOE found that some variable speed heat pumps do not allow maximum speed operation when the outdoor temperature is below 17 °F. For such units, the assumption in the current test procedure is not appropriate. The impact of this discrepancy on the HSPF is not significantly changed by the proposed heating load line revision.

DOE proposes as part of Appendix M1 that for the variable speed units that limit the maximum speed operation below 17 °F and have a low cutoff temperature less than 12 °F, the manufacturer could choose to calculate the maximum heating capacity and the corresponding energy usage through two maximum speed tests at: (1) 17 °F outdoor temperature, and (2) 2 °F outdoor temperature or at a low cutoff temperature, whichever is higher.[39] With this proposed change, manufacturers could 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.

The 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.[40] DOE therefore proposes 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.

DOE believes that the proposed revision reflects field energy use more accurately. However, DOE acknowledges that the limited test results available show very small improvements in the accuracy of the rating method. Because the proposed revision adds an additional test burden (one new test), DOE has proposed to make it optional rather than mandatory. However, DOE would consider making this proposal mandatory for some or all variable speed units, given additional information. Specifically, DOE requests test results and other data that demonstrate whether HSPF results for other variable speed heat pumps would be more significantly impacted by this proposed option, as well as whether the additional test burden would offset the advantages of the proposed modification.

DOE notes that the proposed revision also adds additional complexity to the test procedure in terms of which combinations of tests need to be conducted. In the current test procedure, to calculate the maximum speed performance in the temperature range 17–45 °F, the maximum speed performance at 35 °F is required. However, the maximum speed 35 °F test is not required and performance at 35 °F may instead be calculated from the two maximum speed tests at 17 °F and 47 °F. Therefore, even though manufacturers who choose to rate with the optional low ambient point would no longer need the maximum speed 47 °F point to calculate energy use at maximum speed below 17 °F, they would need either the maximum speed 47 °F test point or 35 °F test point to calculate the capacity and energy use at maximum speed at 35 °F. They may also wish to conduct the maximum speed 47 °F test point to rate heating capacity, although in the proposed Appendix M1, this is only required for heating-only heat pumps.

In summary, with the proposed option for calculating maximum speed performance below 17 °F, manufacturers would test at both maximum speed at 2 °F (or low cutoff temperature) and maximum speed at 17 °F. For rating at 35 °F, they would also test at either maximum speed at 35 °F or maximum speed at 47 °F. Finally, to rate heating capacity or nominal heating capacity (for units whose controls do not allow maximum speed operation at 47 °F), they may also choose to test at either maximum speed at 47 °F allowed by their standard controls or cooling capacity maximum speed at 47 °F, respectively. Table III.10 lists the maximum speed test combination options for the variable speed heat pumps. The test combination option 1 is the default in current test procedure.

TABLE III.10—PROPOSED MAXIMUM SPEED HEATING TEST COMBINATION OPTIONS FOR UNITS HAVING A VARIABLE-SPEED COMPRESSOR

Test description (outdoor dry bulb temperatures)	Current test procedure (Option 1)	Option 2	Option 3	Option 4	Option 5
$H1_N$ (2 °F)	optional if using nominal heating capacity.			for nominal heating capacity.	for nominal heating capacity.
$H1_2$ (47 °F)	X	X	for heating capacity only.		X.
$H2_2$ (35 °F)			X	X	
$H3_2$ (17 °F)	X	X	X	X	X.

[39] 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.

[40] EERE–2009–BT–TP–0004–0047.

TABLE III.10—PROPOSED MAXIMUM SPEED HEATING TEST COMBINATION OPTIONS FOR UNITS HAVING A VARIABLE-SPEED COMPRESSOR—Continued

Test description (outdoor dry bulb temperatures)	Current test procedure (Option 1)	Option 2	Option 3	Option 4	Option 5
$H4_2$ (2 °F) *		X	X	X	X.

* Or low cutoff temperature, whichever is higher.
Note: For units with a low cutoff temperature higher than 12 °F, options 2 through 5 are not available.

DOE additionally notes that all proposed changes in this subsection would change the efficiency ratings of units and are therefore proposed as part of Appendix M1 of 10 CFR 430 Subpart B. Such proposed changes would not appear in Appendix M of the same Part and Subpart.

I. Identified Test Procedure Issues DOE may Consider in Future Rulemakings

Various comments from stakeholders during the public comment period following the publication of the November 2014 ECS RFI raised additional test procedure issues. The stakeholders requested that DOE consider these issues when amending its test procedures. After careful consideration of these issues, DOE believes that either they cannot be resolved or that they require additional action at this time, and therefore declines to address them in this SNOPR. Discussion of these test procedure issues follows in the subsequent subsections.

1. Controlling Variable Capacity Units to Field Conditions

Central air conditioners and heat pumps can be divided into single-speed, two-capacity, or variable capacity (or speed) units based on capacity modulation. System controls are typically more complex with the increasing modulating capability. The DOE test procedure prescribes different testing requirements for units depending on whether they are single-speed, two-capacity, or variable capacity (or speed) in order to characterize the efficiency ratings accurately.

In response to the RFI, stakeholders submitted several comments that address the more complex operation of variable capacity central air conditioners and heat pumps. Stakeholders also submitted comments highlighting the need for improvement in the test procedure's ability to accurately predict energy use in the field, even for units that do not have variable capacity capability. PG&E urged DOE to revise the current test procedure to reflect the more nuanced operation of modern variable speed central air conditioners and heat pumps over the full range of outdoor conditions, given that variable speed units operate differently from the traditional single-speed or two-capacity units. (PG&E, No. 15 at p. 2)

Edison Electric Institute commented that the current test procedure for central air conditions and heat pumps need to be updated to avoid ''gaming'' of system controls to maximize rated SEER and EER, as there is an increase in using variable speed controls for motors, compressors, and variable refrigerant flow. (EEI, No. 18 at p. 3)

NEEA & NPCC commented that the current test procedure does not appropriately test the operation of variable capacity systems. These systems operate much differently in the field than the forced operating conditions with which they are currently tested under waivers and artificially created laboratory conditions. As a result, the efficiency ratings and estimated energy use of these systems cannot be reliably determined. NEEA & NPCC also claimed that the field data shows that systems from different manufacturers with identical HSPF and SEER ratings and identical rated capacity will use significantly different amounts of energy under identical environmental conditions. (NEEA & NPCC, No. 19 at p. 2) NEEA & NPCC also showed the field energy use profiles for six units. They further commented that variable capacity systems behave in a nearly infinite variety of ways under similar outdoor and indoor temperature conditions, and much of this behavior occurs outside the bounds of the test procedure conditions. (NEEA & NPCC, No. 19 at p. 4) NEEA and NPCC commented that test procedure updates to variable capacity equipment will have an impact on the energy savings of these systems. They also commented that the test procedure more accurately representing the field energy use for heat pump systems could vary significantly by climate zone. (NEEA & NPCC, No. 19 at p. 10)

ASAP, ASE, and NRDC commented that the current method for testing variable-capacity units used by manufacturers who have obtained test procedure waivers may not provide good representation of energy use in the field or reasonable relative rankings of product. Representative ratings of variable-capacity products will become more important in the future as variable-capacity units become more widely adopted. (ASAP & ASE & NRDC, No. 20 at p. 1)

PG&E commented that central air conditioners and heat pumps should be tested at part load and cyclic testing under conditions that represent field operations. (PG&E, No. 15 at p. 3) However, PG&E did not provide further detail on what part load and cyclic conditions would be field representative.

ACEEE commented that the current federal test procedure has been awkward for rating new technologies, notably ductless equipment, and probably some types of modulating equipment. (ACEEE, No. 21 at p. 2)

As discussed in section III.H.5, DOE proposes to amend the testing requirements for units equipped with a variable speed compressor during heating mode operation. These proposed amendments would improve the field representativeness of variable speed units and better characterize the field energy use. However, DOE acknowledges that further improvements as suggested by the stakeholders could be possible if more detailed field testing data is available. DOE may consider in a future rulemaking additional amendments to improve the test procedure's representation of field energy use. In regards to ductless and modulating equipment, DOE's existing test procedure already covers testing and rating of these technologies.

2. Revised Ambient Test Conditions

Central air conditioners and heat pumps operate in a wide range of weather conditions throughout the year. Further, both the range of temperature and humidity conditions associated with most of these products' energy use also varies from one climate region to another. The test procedure prescribes calculation of seasonal energy efficiency metrics for cooling and heating based on a finite set of test conditions intended to represent the range of operating conditions while avoiding excess test burden.

DOE decided in the June 2010 NOPR not to propose modifications to convert to wet-coil cyclic testing as data and information were not available to quantify subsequent impacts. 75 FR 31223, 31228 (June 2, 2010). In response to the June 2010 NOPR, SCE, SCGC and SDGE submitted a joint comment recommending DOE require that manufacturers disclose performance data at a range of test conditions, as specified in the Consensus Agreement. The joint comment further explained that program designers need to know how equipment performs in a range of conditions in order for rebate and incentive programs to be effective. This could also make it possible for consumers to select products with performance characteristics that meet their needs. (Docket EERE–2009–BT–TP–0004, SCE, SCGC, and SDGE, No. 9, at p.3)

In the current AHRI certified directory,[41] manufacturers report the full load capacity and EER in addition to SEER for central air conditioners. Manufacturers also report heating capacities and EERs at both 47 °F and 17 °F ambient test conditions in addition to the seasonal efficiency metric HSPF for heat pumps. Cooling capacity and EER at full load are also reported in addition to SEER for heat pumps. DOE believes that this rating data provides sufficient information for determining rebate and incentive programs for program designers.

NREL commented that the existing DOE testing and certification requirements for central air conditioners and heat pumps do not provide sufficient data to compare different units. NREL also urged DOE to adopt different testing conditions for the hot dry and hot humid region. NREL further commented that measurement of water condensation must be reported with higher fidelity than the sensible heat ratio. Latent loads and moisture removal should be reported in each test condition. (EERE–2009–BT–TP–0004, NREL, No. 14 at p. 1)

DOE does not intend to establish different test conditions for various regions of this country. DOE believes that it would add significant burden to manufacturers to report the latent loads and moisture removal in each test condition. In this SNOPR, DOE revises the certification requirement to include reporting the sensible heat ratio. See section III.I.5 for more details. DOE believes that the sensible heat ratio provides a good indication of the moisture removal capability for central air conditioners and heat pumps.

Stakeholders submitted a number of comments on the revised ambient test condition in response to the RFI published on November 5, 2014. 79 FR 65603. University of Alabama commented that the testing conditions prescribed in the federal test procedure for central air conditioners and heat pumps are not representative of actual operation in the field. The outdoor temperatures used for rating should be expanded from 2 to 3 for constant speed units and from 5 to 6 for multi-capacity and variable speed units. The rating points can be used to determine more appropriate SEER and HSPF for climates outside of the current DOE zone 4 conditions. Specifically, University of Alabama proposed the cooling indoor dry bulb and wet bulb temperatures to be 77 °F and 64.4 °F, instead of the current requirement of 80 °F and 67 °F, respectively. Heating indoor dry bulb temperature should use 68 °F instead of the current requirement of 70 °F. For the outdoor conditions, testing at 113 °F, 95 °F, and 77 °F have been proposed for the cooling mode, and 41 °F, 23 °F, and 5 °F have been proposed for the heating mode, respectively. (University of Alabama, No. 6 at p. 1–2)

PG&E commented that DOE should amend the test procedure to require testing at 76 °F dry bulb with 50% relative humidity indoor conditions to represent the comfort desired in dwellings. (PG&E, No. 15 at p. 3) However, PG&E did not provide further detail on why the revised test condition is more representative than the requirements in the current federal test procedure.

PG&E also commented that the current cooling condition at 95 °F does not fully capture the peak load experienced by consumers in the hottest summer weather. PG&E further urged DOE to revise the test procedure to account for ambient dry bulb conditions of 105 °F or 115 °F experienced by consumers in the desert climates. (PG&E, No. 15 at p. 3)

Moreover, PG&E commented that DOE should adopt the testing at outdoor ambient temperatures that generate a performance map of the system for use in annual energy use simulation. (PG&E, No. 15 at p. 3) However, there is no further detail provided regarding this comment.

EEI suggested that DOE revise the indoor air inlet dry bulb/wet bulb temperatures to be lowered from 80 °F/67 °F to 78 °F/61 °F, respectively. Such a change would create more realistic indoor conditions that would require dehumidification to ensure properly managed indoor air quality. (EEI, No. 18 at p. 4) However, EEI did not provide further detailed justifications why such a change would create more realistic indoor conditions than the current federal testing requirements.

NEEA and NPCC commented that the current federal test procedure does not capture performance under the full range of operating conditions for which many of these systems are designed. Some air conditioners perform significantly better at temperatures above 100 °F than others, but based on the current test procedure, there is no testing requirement for temperatures above 95 °F. For heat pumps, systems may perform differently above 47 °F and below 17 °F conditions. NEEA and NPCC commented that the test procedure and the resulting ratings should expose these differences and allow the market to properly select the systems that are most appropriate and most efficient for individual climate conditions. (NEEA & NPCC, No. 19 at p. 2)

ASAP, ASE, and NRDC commented that the test conditions defined in the current test procedure do not reflect field conditions. Adding a test point for SEER ratings at an outdoor temperature above 95 °F and adding a test point for HSPF ratings at an outdoor temperature below 17 °F would incentivize manufacturers to provide good efficiency performance at these temperatures. In addition, requiring reporting of performance at each of the outdoor temperature test points would allow efficiency program administrators to incentivize equipment that will perform well in their region. (ASAP & ASE & NRDC, No. 20 at p. 2)

DOE appreciates that there may be value in providing more performance data, and that the range of operating conditions in the field may be more extensive than that represented by the current test. However, the extensive study and test work that would have to be conducted to properly assess and choose a better range of test conditions has not been completed. Hence, although DOE has proposed some changes to the test conditions required for testing of variable-speed heat pumps in heating mode, DOE has not proposed changes as extensive as the comments suggest. DOE may consider additional changes addressing these issues in future test procedure rulemakings.

[41] *http://www.ahridirectory.org/ahridirectory/pages/home.aspx.*

3. Performance Reporting at Certain Air Volume Flow Rates

Central air conditioners and heat pumps condition the indoor air to satisfy cooling and heating requirements of a house. For ducted central air conditioners and heat pumps, indoor air is driven by the blower of the air handling unit or the furnace. Air volume rate affects the heat transferred between the air conditioning device and indoor air, and also affects the performance ratings of an air conditioner or heat pump.

University of Alabama recommended that all performance results for central air conditioners and heat pumps be reported within the air volume rate range of 375 to 425 cfm per ton, and that the air volume rates be included in the reporting requirements. Higher air volume rates will result in reduced dehumidification capability and cause thermal comfort issue. (University of Alabama, No. 9 at p. 1)

The current DOE test procedure requires that full load air volume rate be no more than 37.5 standard cfm (scfm) per 1,000 Btu/h of cooling capacity (see 10 CFR part 430, subpart B, Appendix M, Section 3.1.4.1.1), but the test procedure does not have a minimum air volume rate requirement. DOE has proposed in this notice to require reporting of the cooling full load air volume rate as part of certification reporting. See section III.I.5 for more details. The air volume rate is also reported in the AHRI certification database.[42] DOE believes that these requirements will ensure that air volume rates used for rating central air conditioners and heat pumps are in an appropriate range.

4. Cyclic Test With a Wet Coil

The DOE test procedure for central air conditioners and heat pumps prescribe specific test conditions under which units are to be tested. These test conditions include both steady-state and cyclic tests. A dry coil test refers to the test conditions that do not result in moisture condensing on the indoor coil, and a wet coil test refers to the test conditions that result in moisture condensing on the indoor coil. DOE proposed in the June 2010 NOPR not to amend the existing cyclic testing requirement from dry coil test to wet coil test. DOE concluded that there was no sufficient data to show a greater benefit to using wet coil cyclic test versus the dry coil cyclic test. 75 FR 31223, 31227 (June 2, 2010).

In response to the RFI regarding central air conditioners and heat pumps (79 FR 65603, November 5, 2014), ASAP & ASE & NRDC commented that the cyclic test in the current test procedure is conducted using a dry coil, which is not representative of field conditions. Using the same indoor conditions (*i.e.,* 80 °F dry bulb and 67 °F wet bulb) for the cyclic tests as used for the steady-state test would better reflect the cyclic performance of central air conditioners and heat pumps. (ASAP & ASE & NRDC, No. 20 at p. 2) DOE believes this approach may have merit, but has not sufficiently studied it to have proposed its inclusion in the test procedure at this time. DOE may consider adopting the approach in a future rulemaking.

5. Inclusion of the Calculation for Sensible Heating Ratio

Air conditioning reduces air temperature and also reduces humidity. Cooling associated with air temperature reduction is called sensible capacity, while cooling associated with dehumidification is called latent capacity. The balance of these capacities for a given air conditioner operating in a given set of operating conditions is represented as sensible heat ratio (SHR), which is equal to sensible cooling divided by total cooling. Air conditioners can be designed to operate with high or low SHR depending on the air conditioning needs. Similarly, an air conditioner can be optimized to maximize efficiency depending on the indoor humidity level.

In the June 2010 NOPR, DOE proposed including the calculation for (SHR at the B, B_1, or B_2 test condition (82 °F dry bulb, 65 °F wet bulb, outside air) in the test procedure. 75 FR 31223, 31229 (June 2, 2010). DOE received comments regarding the inclusion of calculations for SHR in the subsequent public comment period. AHRI supported adoption of the SHR, provided that it is based off the total net capacity and is a reported value only. (AHRI, No. 6 at p. 4) Ingersoll Rand agreed with AHRI. (Ingersoll Rand, No. 10 at pp. 2–3) Lennox likewise agreed with AHRI regarding adding calculations for SHR and further requested that DOE provide calculations for SHR at outdoor ambient conditions of 82 °F. (Lennox, No. 11 at p. 1) Building Science Corporation stated that the calculation of the SHR was a favorable step towards inclusion of a dehumidification performance rating, but requested determining SHR at multiple outdoor and indoor conditions and reporting a metric for moisture removal efficacy. (Building Science Corporation, No. 16 at p. 1) NEEA concurred with DOE's proposal in the NOPR to add calculations of sensible heat ratio (SHR) to the test procedure requirements. (NEEA, No. 7 at p. 6) The People's Republic of China World Trade Organization Technical Barriers to Trade National Notification and Enquiry Center (China WTO) suggested that SHR be calculated at the same SEER test conditions. (China WTO, No. 18 at p. 4).

DOE does not believe that measurements at multiple indoor or outdoor conditions are necessary to obtain a SHR value that represents unit operation during an average use cycle or period. (42 U.S.C. 6293(b)(3)) Therefore, DOE is maintaining its position in the NOPR to include calculation for sensible heat ratio at only the condition at which products are rated (82 °F dry bulb, 65 °F wet bulb, outside air), and proposes to include this change to the revised Appendix M test procedure in this notice. DOE notes that the addition of these calculations does not add significant test burden because the existing measurement instruments, used for determining the inputs for SEER, can also determine the inputs for SHR.

The June 2010 NOPR highlighted a Joint Utilities recommendation that DOE should require all units be certified and rated for sensible heat ratio (SHR) at 82 °F ambient dry bulb temperature. 75 FR 31223, 31229 (June 2, 2010). DOE believes that the existing certification test procedures and ratings are sufficient to determine product efficiency; efforts to establish dehumidification performance for central air conditioner and heat pumps are not currently necessary given that the primary function of the subject products is not dehumidification, nor would doing so be helpful in improving the accuracy of product efficiency.

In response to the RFI regarding central air conditioners and heat pumps (79 FR 65603, November 5, 2014), stakeholders submitted several comments on the reporting requirements related to the SHR. PG&E commented that the test procedure should adopt testing that characterizes the sensible heat ratios for high (western dry climates, approximately 500 cfm/ton) and low (eastern humid climates, approximately 350 cfm/ton) evaporator coil air volume rate. (Docket No. EERE–2014–BT–STD–0048, PG&E, No. 15 at p. 3) Edison Electric Institute commented that the test procedure should take into account a dehumidification requirement as homes are getting tighter with fewer air changes. (*Id.;* EEI, No. 18 at p. 3) ASAP & ASE & NRDC requested DOE require reporting sensible heat ratio for central air conditioners and heat pumps. Sensible heat ratio would provide more

[42] AHRI Directory of Certified Product Performance: *https://www.ahridirectory.org/ahridirectory/pages/home.aspx*.

information to consumers and contractors about appropriate units for their region and also allow efficiency program administrators to better target efficiency programs for central air conditioners and heat pumps. (*Id.;* ASAP & ASE & NRDC, No. 20 at p. 2)

In response to the stakeholder comments, DOE understands that air volume rate can be controlled properly to suit the dehumidification purposes. However, manufacturers can design their products to meet the needs of consumers in different climate regions. Therefore, DOE does not intend at this time to develop a test procedure that requires different air volume rates based on the climate region. DOE does, however, realize the merit of reporting SHR for consumer choices. As such, DOE proposes to simply require the reporting of the SHR value calculated based on full-load cooling test conditions at the outdoor ambient conditions proposed earlier in this section: 82 °F dry bulb and 65 °F wet bulb.

J. Compliance With Other Energy Policy and Conservation Act Requirements

1. Test Burden

EPCA requires that any test procedures prescribed or amended shall be reasonably designed to produce test results which measure 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. (42 U.S.C. 6293(b)(3)) For the reasons that follow, DOE has tentatively concluded that revising the DOE test procedure, per the proposals in this SNOPR, to measure the energy consumption of central air conditioners and heat pumps in active mode and off mode would produce the required test results and would not result in any undue burdens.

As discussed in section IV.B of this SNOPR, the proposed test procedures to determine the active-mode and standby-mode energy use would require use of the same testing equipment and facilities that manufacturers are currently using for testing to determine CAC and CHP ratings for certifying performance to DOE. While this notice proposes clarifications to the test procedures, and proposes adopting into regulation the test procedures associated with a number of test procedure waivers, most of the proposals would not affect test time or the equipment and facilities required to conduct testing. Possible changes in test burden associated with the proposals of this notice apply to off mode testing and requirements for testing of basic models by ICMs.

The proposals include additional testing to determine off mode energy use, as required by EPCA. (42 U.S.C. 6295(gg)(2)(A)) This additional testing may require investment in additional temperature-controlled facilities. However, DOE's proposal does not require that every individual combination be tested for off mode, allowing sufficient use of AEDMs in order to reduce test burden.

The proposals also call for testing to determine performance for ICMs. Specifically, the proposals call for testing of one split system combination for each model of indoor unit sold by an ICM. While this change would increase test burden for these manufacturers, DOE believes it is the appropriate minimum test burden to validate ratings for these systems, as it is consistent with current requirements for OUMs, for which testing is required for every model of outdoor unit. DOE requests comment on this issue.

DOE allows manufacturers to pursue an alternative efficiency determination method process to certify products without the need of testing. In this notice, DOE revises and clarifies such requirements, as detailed in section III.B, to continue to enable manufacturers who wish to reduce testing burden to utilize this method.

As detailed in section III.C, manufacturers of certain products covered by test procedures waivers, have already utilized the alternative test procedures provided to them for certification testing. Thus, the inclusion of said alternative test procedures into the test procedure, as revised in this notice, does not add additional test burden.

In addition, DOE carefully considered the testing burden on manufacturers in proposing a modified off mode test procedure that is less burdensome than the proposals it made in the April 2011 SNOPR and October 2011 SNOPR and that addresses stakeholder comment regarding the test burden of such prior proposals. Further discussion regarding test burden associated with the proposals set forth in this notice for determining off mode power consumption can be found in section III.D.

DOE set forth proposals to improve test repeatability, improve the readability and clarity of the test procedure, and utilize industry procedures that manufacturers may be aware of in an effort to reduce the test burden. Sections III.E, III.F, and III.G presents additional detail regarding such proposals.

Although DOE proposes to change the current test procedure in a manner that would impact measured energy efficiency, amend existing requirements, and increase the testing time for such tests, DOE carefully considered the impact on testing burden and made efforts to balance accuracy, repeatability, and test burden during the course of the development of such proposals. Further discussion is found in section III.H.

Therefore, DOE determined that the proposed revisions to the central air conditioner and heat pump test procedure would produce test results that measure energy consumption during a period of representative use, and that the test procedure would not be unduly burdensome to conduct.

2. Potential Incorporation of International Electrotechnical Commission Standard 62301 and International Electrotechnical Commission Standard 62087

Under 42 U.S.C. 6295(gg)(2)(B), EPCA directs DOE to consider IEC Standard 62301 and IEC Standard 62087 when amending test procedures for covered products to include standby mode and off mode power measurements.

DOE reviewed IEC Standard 62301, ''Household electrical appliances— Measurement of standby power'' (Edition 2.0 2011–01),[43] and determined that the procedures contained therein for preparation of the unit under test and for conducting the test are already set forth in the amended test procedure, as proposed in this notice, for determining off mode power consumption and for determining the components (cyclic degradation coefficient) that make up standby power for central air conditioners and heat pumps. Therefore, DOE determined that referencing IEC Standard 62301 is not necessary for the proposed test procedure that is the subject of this rulemaking.

DOE reviewed IEC Standard 62087, ''Methods of measurement for the power consumption of audio, video, and related equipment'' (Edition 3.0 2011–04), and determined that it would not be applicable to measuring power consumption of HVAC products such as central air conditioners and heat pumps. Therefore, DOE determined that referencing IEC Standard 62087 is not necessary for the proposed test procedure that is the subject of this rulemaking.

[43] IEC Standard 62301 covers measurement of power consumption for standby mode and low power modes, as defined therein.

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 central air conditioners and heat pumps, under the provisions of the Regulatory Flexibility Act and the procedures and policies published on February 19, 2003. DOE tentatively concludes and certifies that the proposed rule, if adopted, would not result in a significant impact on a substantial number of small entities. The factual basis for this certification is set forth below.

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 750 employees or fewer. DOE used the small business size standards published on January 31, 1996, as amended, by the SBA to determine whether any small entities would be required to comply with the rule. 61 FR 3280, 3286, as amended at 67 FR 3041, 3045 (Jan. 23, 2002) and at 69 FR 29192, 29203 (May 21, 2004); see also 65 FR 30836, 30850 (May 15, 2000), as amended at 65 FR 53533, 53545 (Sept. 5, 2000). 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 *www.sba.gov/idc/groups/public/documents/sba_homepage/serv_sstd_tablepdf.pdf*.

Central air conditioner and heat pump 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 AHRI's listing of central air conditioner and heat pump product manufacturer members and surveyed the industry to develop a list of domestic manufacturers. As a result of this review, DOE identified 22 manufacturers of central air conditioners and heat pumps, of which 15 would be considered small manufacturers with a total of approximately 3 percent of the market sales. 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. In the June 2010 NOPR, DOE estimated the incremental cost of the proposed additional tests described in 10 CFR part 430, subpart B, Appendix M (proposed section 3.13) to be an increase of $1,000 to $1,500 per unit tested, indicating that the largest additional cost would be associated with conducting steady-state cooling mode tests and the dry climate tests for the SEER–HD rating). 75 FR at 31243 (June 2, 2010). DOE has eliminated tests associated with the SEER–HD rating from the proposals in this notice. DOE conservatively estimates that off mode testing might cost $1,000 (roughly one-fifth of the $5000 cost of active mode testing—see 75 FR at 31243 (June 2, 2010)). Assuming two off mode tests per tested model, this is an average test cost of $2,000 per model.

The proposals of this notice also require that ICMs test one combination of every basic model (*i.e.,* model of indoor unit). Based on a test cost estimate of $5000 and two tests per model, the costs for this proposal are $10,000 for each basic model.

Because the incremental cost of running the extra off mode tests is the same for all manufacturers, DOE believes that all manufacturers would incur comparable costs for testing to certify off mode power use for basic models as a result of the proposed test procedure. DOE expects that small manufacturers will incur less testing expense compared with larger manufacturers as a result of the proposed testing requirements because they have fewer basic models and thus require proportionally less testing when compared with large manufacturers that have many basic models. DOE recognizes, however, that smaller manufacturers may have less capital available over which to spread the increased costs of testing.

With respect to the provisions addressing AEDMs, the proposals contained herein would not increase the testing or reporting burden of outdoor unit manufacturers who currently use, or are eligible to use, an AEDM to certify their products. The proposal would eliminate the ARM nomenclature and treat these methods as AEDMs, eliminate the pre-approval requirement for product AEDMs, revise the requirements for validation of an AEDM in a way that would not require more testing than that required by the AEDM provisions included in the March 7, 2011 Certification, Compliance and Enforcement Final Rule (76 FR 12422) (''March 2011 Final Rule''), and amend the process that DOE promulgated in the March 2011 Final Rule for validating AEDMs and verifying certifications based on the use of AEDMs. Because these AEDM-related proposals would either have no effect on test burden or decrease burden related to testing (*e.g.,* elimination of ARM pre-approval), DOE has determined these proposals would result in no significant change in testing or reporting burden. The proposals contained herein would not increase the testing or reporting burden of outdoor unit or independent coil manufacturers besides the revision to the requirements for validation of an AEDM, of which burden is outweighed by the benefit of providing more accurate ratings for models of indoor units sold by ICMs, as discussed in section III.A.3.d.

To evaluate the potential cost impact of the other test-related proposals, DOE compared the cost of the testing to the total value added by the manufacturers to determine whether the impact of the proposed test procedure amendments is significant. The value added represents the net economic value that a business creates when it takes manufacturing inputs (*e.g.,* materials) and turns them into manufacturing outputs (*e.g.,* manufactured goods). Specifically, as defined by the U.S. Census, the value added statistic is calculated as the total value of shipments (products manufactured plus receipts for services rendered) minus the cost of materials, supplies, containers, fuel, purchased electricity, and contract work expenses.

DOE analyzed the impact on the smallest manufacturers of central air conditioners and heat pumps because these manufacturers would likely be the most vulnerable to cost increases. DOE calculated the additional testing expense as a percentage of the average value added statistic for the five individual firms in the 25 to 49 employee size category in NAICS 333415 as reported by the U.S. Census (U.S. Bureau of the Census, American Factfinder, 2002 Economic Census, Manufacturing, Industry Series, Industry Statistics by Employment Size, *http://factfinder.census.gov/servlet/EconSectorServlet?_lang=en&ds_name=EC0200A1&_SectorId=31&_ts=288639767147*). The average annual value for manufacturers in this size range from the census data was $1.26 million in 2001$, per the 2002 Economic Census, or approximately $1.52 million per year in 2009$ after adjusting for inflation using the implicit price deflator for gross domestic product (U.S. Department of Commerce Bureau of Economic Analysis, *www.bea.gov/national/nipaweb/SelectTable.asp*).

DOE also examined the average value added statistic provided by census for all manufacturers with fewer than 500 employees in this NAICS classification as the most representative value from the 2002 Economic Census data of the central air conditioner manufacturers with fewer than 750 employees that are considered small businesses by the SBA (15 manufacturers). The average annual value added statistic for all small manufacturers with fewer than 500 employees was $7.88 million (2009$).

Given this data, and assuming the range of estimates of additional costs, $2,000 for OUMs and $10,000 for ICMs for the additional testing costs, DOE concluded that the additional costs for testing of a single basic model product under the proposed requirements would be up to approximately 0.7 percent of annual value added for the 5 smallest firms, and approximately 0.13 percent of the average annual value added for all small central air conditioner or heat pump manufacturers (15 firms). DOE estimates that testing of basic models may not have to be updated more than once every 5 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 requires that only the highest sales volume split-system combinations be laboratory tested. 10 CFR 430.24(m). The majority of central air conditioners and heat pumps offered by a manufacturer are typically split-systems that are not required to be laboratory tested but can be certified using an alternative rating method that does not require DOE testing of these units. DOE reviewed the available data for five of the smallest manufacturers to estimate the incremental testing cost burden for those small firms that might experience the greatest relative burden from the revised test procedure. These manufacturers had an average of 10 models requiring testing (AHRI Directory of Certified Product Performance, *www.ahridirectory.org/ahridirectory/pages/home.aspx*), while large manufacturers will have well over 100 such models. The additional testing cost for final certification for 10 models was estimated at $4,000 to $100,000. Meanwhile, these certifications would be expected to last the product life, estimated to be at least 5 years based on the time frame established in EPCA for DOE review of central air conditioner efficiency standards. This test burden is therefore estimated to be approximately 1.3 percent of the estimated 5-year value added for the smallest five manufacturers. DOE believes that these costs are not significant given other, much more significant costs that the small manufacturers of central air conditioners and heat pumps incur in the course of doing business. DOE seeks comment on its estimate of the impact of the proposed test procedure amendments on small entities and its conclusion that this impact is not significant.

Accordingly, as stated above, DOE tentatively concludes and certifies that this proposed rule would not have a significant economic impact on a substantial number of small entities. Accordingly, DOE has not prepared an initial regulatory flexibility analysis (IRFA) for this rulemaking. DOE will provide its certification and supporting statement of factual basis to the Chief Counsel for Advocacy of the SBA for review under 5 U.S.C. 605(b).

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 20 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 supplemental 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. Pub. L. 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 the 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) the following two test standards published by AHRI: ANSI/AHRI 210/240–2008 with Addenda 1 and 2, titled ''Performance Rating of Unitary Air-Conditioning & Air-Source Heat Pump Equipment;'' and ANSI/AHRI 1230–2010 with Addendum 2, titled ''Performance Rating of Variable Refrigerant Flow (VRF) Multi-Split Air-Conditioning and Heat Pump Equipment.'' DOE also proposes to IBR a draft version of ASHRAE 210/240 which has not yet been published. DOE also proposes to update its IBR to the most recent version of the following standards published by ASHRAE: 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, 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 41.6–2014 titled ''Standard Method for Humidity Measurement'', and ASHRAE 41.9–2011titled ''Standard Methods for Volatile-Refrigerant Mass Flow Measurements Using Calorimeters''. Finally, DOE proposes to updates its IBR to the most recent version of the following test procedure from ASHRAE and AMCA: ASHRAE/AMCA 51–07/210–07, 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*. AHRI Standard 210/240-Draft is a draft version of AHRI 210/240 that AHRI provided to DOE in 2015. AHRI Standard 210/240-Draft will supersede the 2008 version once it is published. The draft version is available on the rulemaking Web page (Docket EERE–2009–BT–TP–0004–0045).

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. The current DOE test procedure references a previous version of this standard, ASHRAE 37–2005. 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 41.6–2014 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*.

ASHRAE/AMCA 51–07/210–07 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 air flow 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 ASHRAE/AMCA 51–07/210–07 that address test conditions. The current DOE test procedure references a previous version of this standard, ASHRAE/AMCA 51–99/210–99. ASHRAE/AMCA 51–07/210–07 can be purchased from AMCA's Web site at *http://www.amca.org/store/index.php*.

V. Public Participation

A. 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.

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.

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).

B. 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:

1. The details characterizing the same model of indoor unit, same model of outdoor unit, and same single-package model;

2. Its proposed changes to the determination of certified ratings for single-split-system air conditioners, specifically in its proposed phased approach where in the first phase

manufacturers must certify all models of outdoor units with the model of coil-only indoor unit that is likely to have the largest volume of retail sales with the particular model of outdoor unit but may use the model of blower coil indoor unit likely to have the highest sales if the model of outdoor unit is sold only with models of blower coil indoor units, and may use testing or AEDMs to rate other combinations; and in the second phase manufacturers must certify all models of outdoor units with the model of blower coil indoor unit that is likely to have the largest volume of retail sales with that model of outdoor unit but must rate other blower coil or coil-only combinations through testing or AEDMs;

3. Its proposed definitions for blower coil and coil-only indoor units;

4. Whether additional testing and rating requirements are necessary for multi-split systems paired with models of conventional ducted indoor units rather than short-duct units;

5. Whether manufacturers or other stakeholders support ratings for mix-match multi-split systems including models of both SDHV and non-ducted or short-ducted indoor units, and if so, how they should be rated (*i.e.,* by by taking the mean of a sample of tested non-ducted units and a sample of tested SDHV units or by testing a combination on non-ducted and SDHV units), and whether the SDHV or split-system standard would be most appropriate;

6. Whether manufacturers support having the ability to test mix-match systems using the test procedure rather than rating them using an average of the other tested systems;

7. Whether manufacturers support the rating of mix-match systems using other than a straight mean, such as a weighting by the number of non-ducted or short-ducted units;

8. Whether the definition of ''tested combination'' is appropriate for rating specific individual combinations, or whether manufacturers want more flexibility such as testing with more than 5 indoor units;

9. Information and data on manufacturing and testing variability associated with multi-split systems that would allow it to understand how a single unit may be representative of the population and what tolerances would need to be applied to ratings based on a single unit sample in order to account for variability;

10. The basic model definition in section III.A.1;

11. Its proposal for ICMs to test each model of indoor unit with the lowest-SEER model of outdoor unit that is certified as a part of a basic model by an OUM as well as any test burden associated with this proposal;

12. The likelihood of multiple individual models of single-package units meeting the requirements proposed in the basic model definition to be assigned to the same basic model;

13. Whether, if manufacturers are able to assign multiple individual single-package models to a single basic model, whether manufacturers would want to use an AEDM to rate other individual models within the same basic model other than the lowest SEER individual model;

14. Whether manufacturers would want to employ an AEDM to rate the off-mode power consumption for other variations of off-mode associated with the single-package basic model other than the variation tested;

15. The reporting burden associated with the proposed certification reporting requirements proposed in this notice;

16. The additions to the represented value requirements for cooling capacity, heating capacity, and SHR, as well as the proposed rounding requirements;

17. The proposal to not require additional testing to validate an AEDM beyond the testing required under 429.16(a)(2)(ii) for split-system air conditioners and heat pumps where manufacturers must test each basic model, being each model of outdoor unit, with at least one model of indoor unit;

18. The proposal that ICMs must use the combinations they would be required to test, under 429.16, to validate an AEDM that is intended to be used for other individual combinations within each basic model;

19. Whether the approach to not penalize manufacturers for applying conservative ratings to their products is reasonable to identify an individual combination's failure to meet its certified rating;

20. Whether manufacturers would typically apply more than one AEDM, and if they would, the differences between such AEDMs;

21. Its proposal for multi-circuit products to adopt the same common duct testing approach used for testing multi-split products; and whether this method will yield accurate results that are representative of the true performance of these systems;

22. Its proposals for multi-blower products, including whether individual adjustments of each blower are appropriate and whether external static pressures measured for individual tests may be different;

23. Its proposal to require a test for off mode power consumption at 72±2 °F, a second test at the temperature below a turn-on temperature specified by the manufacturer, a tolerance on the temperature, and the proposal that manufacturers include in certification reports the temperatures at which the crankcase heater is designed to turn on and turn off for the heating season, if applicable;

24. The proposal to replace the off mode test at 57 °F with a test at a temperature which is 5±2 °F below a manufacturer-specified turn-on temperature to maintain the intent of the off mode power consumption rating as a rating that measures the off mode power consumption for the heating season, and allay the stakeholders' concerns of a loophole at the 57 °F test point;

25. The proposal to use a per-compressor off mode power consumption metric so as to not penalize manufacturers of products with multiple compressor systems, which are highly efficient and require larger crankcase heaters for safe and reliable operation;

26. The proposal on the multiplier of 1.5 for determining the shoulder season and heating season per-compressor off mode power so as to not penalize manufacturers of products with modulated compressors, which require a larger crankcase heater to ensure safe and reliable operation;

27. The proposal to more accurately reflect the off mode power consumption for coil-only and blower coil split-system units by excluding the low-voltage power from the indoor unit when measuring off mode power consumption for coil-only split-system air conditioners and including the low-voltage power from the indoor unit when measuring off mode power consumption for blower coil split-system air conditioning and heat pumps;

28. The proposal to incent manufacturers of products with time delays by adopting a credit to shoulder season energy consumption that is proportional to the duration of the delay or a default of 25% savings in shoulder season off mode energy consumption and the possibility of a verification test for length of time delay;

29. The proposal to add optional informational equations to determine the actual off mode energy consumption, based on the hours of off mode operation and off mode power for the shoulder and heating seasons;

30. Whether regulating crankcase heater energy consumption has a negative impact on product reliability in light of the test method proposed in this rule;

31. The proposal to improve repeatability of testing central air conditioner and heat pump products by requiring the lowest fan speed setting that meets minimum static pressure and maximum air volume rate requirements for blower coil systems and requiring the lowest fan speed settings that meets the maximum static pressure and maximum air volume rate requirements for coil-only indoor units;

32. The proposal to mirror how insulation is installed in the field by requiring test laboratories either install the insulation shipped with the unit or use insulation as specified in the manufacturer's installation manuals included with the unit;

33. The proposal to clarify liquid refrigerant line insulation requirements by requiring such insulation only if the product is a heating-only heat pump;

34. The proposal to prevent thermal losses from the refrigerant mass flow meter to the floor by requiring a thermal barrier if the meter is not mounted on a pedestal or is not elevated;

35. The proposal to require either an air sampling device used on all outdoor unit air-inlet surfaces or demonstration of air temperature uniformity for the outdoor unit vis-a-vis 1.5 °F maximum spread of temperatures measured by thermocouples distributed one thermocouple per square feet of air-inlet surface of the outdoor unit;

36. The proposal to require that the dry bulb temperature and humidity measurements used to verify that the required outdoor air conditions have been maintained be measured for the air collected by the air sampling device (*e.g.,* rather than being measured by temperature sensors located in the air stream approaching the air inlets);

37. The proposal to limit thermal losses by preventing the air sampling device from nearing the test chamber floor, insulating air sampling device surfaces, and requiring dry bulb and humidity measurements be made at the same location in the air sampling device;

38. The proposal to fix maximum compressor speed when testing at each of the outdoor temperature for those control systems that vary maximum compressor speed with outdoor temperature;

39. The proposal to prevent improper refrigerant charging techniques by requiring charging of near-azeotropic and zeotropic refrigerant blends in the liquid state only;

40. The proposal to require, for air conditioners and cooling-and-heating heat pumps refrigerant charging at the A or A_2 test condition, and for heating-only heat pumps refrigerant charging at the H_1 or $H1_2$ test condition, to meet a 12 ± 2 °F superheat temperature requirement for units equipped with fixed orifice type metering devices and a 10 ± 2 °F subcooling temperature requirement for units equipped with thermostatic expansion valve or electronic expansion valve type metering devices, if no manufacturer installation instructions provide guidance on charging procedures;

41. The proposal to verify functionality of heat pumps at the H_1 or $H1_2$ test condition after charging at the A or A_2 test condition, and if non-functional, the proposal to adjust refrigerant charge to the requirements of the proposed standardized charging procedure at the H_1 or $H1_2$ test condition;

42. The proposal to require refrigerant charging based on the outdoor installation instructions for outdoor unit manufacturer products and refrigerant charging based on the indoor installation instructions for independent coil manufacturer products, where both the indoor and outdoor installation instructions are provided and advise differently, unless otherwise specified by either installation instructions;

43. The proposal to require installation of pressure gauges and verification of refrigerant charge amount and, if charging instructions are not available adjust charge based on the proposed refrigerant charging procedure;

44. All aspects of its proposals to amend the refrigerant charging procedures;

45. The proposal to allow for cyclic tests of single-package ducted units an upturned duct as an alternative arrangement to replace the currently-required damper in the inlet portion of the indoor air ductwork;

46. The proposal to further justify adequacy of the alternative arrangement in preventing thermal losses during the OFF portion of the cyclic test by proposing installing a dry bulb temperature sensor near the indoor inlet and requiring the maximum permissible range of the recorded temperatures during the OFF period be no greater than 1.0 °F;

47. The proposed revisions to the cyclic test procedure for the determination of both the cooling and heating coefficient of degradation, including additional test data that would support the proposed specifications, or changes to, the number of warm-up cycles, the cycle time for variable speed units, the number of cycles averaged to obtain the value, and the stability criteria;

48. The proposal to allay stakeholder concerns regarding compressor break-in period by allowing an optional break-in period of up to 20 hours prior to testing;

49. Its proposed limitation of incorporation by reference to industry standards to specific sections necessary for the test procedure, including any specific sections stakeholders feel should be referenced that are not;

50. The proposed sampling interval for dry-bulb temperatures, wet bulb temperature, dew point temperature, and relative humidity;

51. The appropriate use of the target value and maximum tolerances for refrigerant charging, as well as data to support the appropriate selection of tolerance;

52. The proposal for damping pressure transducer signals including whether the proposed maximum time constant is appropriate;

53. Setting a definition for short duct systems to mean ducted systems whose indoor units can deliver no more than 0.07 in. wc. ESP when delivering the full load air volume rate for cooling operation, and requiring such systems meet the minimum ESP levels as proposed in the NOPR: 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;

54. The incorporation by reference of AHRI 1230–2010, and in particular the specific sections of Appendix M and AHRI 1230–2010 that DOE proposes to apply to testing VRF systems;

55. The proposed change to the informative tables at the beginning of Section 2. Testing Conditions and/or whether additional modifications to the new table could be implemented to further improve clarity;

56. Its proposal to delete the definition of mini-split air conditioners and heat pumps, and define (1) single-zone-multiple-coil split-system to represent a split-system that has one outdoor unit and that has two or more coil-only or blower coil indoor units connected with a single refrigeration circuit, where the indoor units operate in unison in response to a single indoor thermostat; and (2) single-split-system to represent a split-system that has one outdoor unit and that has one coil-only or blower coil indoor unit connected to its other component(s) with a single refrigeration circuit;

57. Its proposal to include in the ESP requirement a pressure drop contribution associated with average typical filter and indoor coil fouling levels and its use of residential-based indoor coil and filter fouling pressure drop data to estimate the appropriate

ESP contribution; DOE also requests data that would validate the proposed ESP contributions or suggest adjustments that should be made to improve representativeness of the values in this proposal;

58. Its proposals to set higher minimum ESP requirements for systems other than multi-split systems and small-duct, high-velocity systems and report the external static pressure used during their certification tests;

59. Its proposal to implement an allowance in ESP for air-conditioning units tested in blower-coil (or single-package) configuration in which a condensing furnace is in the air flow path during the test. DOE seeks comment regarding the proposed 0.1 in. wc. ESP reduction for such tests, including test data to support suggestions regarding different reductions.

60. Its proposal to revise the heating load line that shifts the heating balance point and zero load point to lower ambient temperatures that better reflect field operations and energy use characteristics, as well as its proposal to perform cyclic testing for variable speed heat pumps at 47 °F instead of at 62 °F;

61. Whether, in the case that the proposed heating load line is not adopted, DOE should modify the HSPF rating procedure for variable speed heat pumps at mid-range outdoor temperatures using option 1: Which entails basing performance on minimum speed tests at 47 °F and intermediate speed test at 35 °F and is the less accurate option but has no additional test burden; or option 2: Which entails basing performance on minimum speed tests at 47 °F and at 35 °F and is more accurate but with higher test burden;

62. Test results and other data regarding whether HSPF results for other variable speed heat pumps would be more significantly impacted by this change to the test procedure to test at maximum speed at 2 °F outdoor temperature or at low cutoff temperature, whichever is higher (in conjunction with the test at maximum speed at 17 °F outdoor temperature) as well as whether the additional test burden would offset the advantages of the proposed modification;

63. The estimate of the number of small entities that may be impacted by the proposed test procedure and its conclusion that the impact is not significant.

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 21, 2015.

Kathleen B. Hogan,

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

For the reasons set forth in the preamble, DOE proposes to amend parts 429 and 430 of chapter II of Title 10, Subpart B, Code of Federal Regulations, to read as follows:

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. Amend § 429.12 by revising paragraphs (b)(8) and (12) to read as follows:

§ 429.12 General requirements applicable to certification reports.

* * * * *

(b) * * *

(8) The test sample size (*i.e.,* number of units tested for the basic model, or in the case of single- split-system central air conditioners and central air conditioning heat pumps, for each individual combination). Enter ''0'' if an AEDM was used in lieu of testing;

* * * * *

(12) If the test sample size is listed as ''0'' to indicate the certification is based upon the use of an alternate way of determining measures of energy conservation, identify the method used for determining measures of energy conservation (such as ''AEDM,'' or linear interpolation). Manufacturers of commercial packaged boilers, commercial water heating equipment, commercial refrigeration equipment, and commercial HVAC equipment must provide the manufacturer's designation (name or other identifier) of the AEDM used; and

* * * * *

■ 3. Section 429.16 is revised to read as follows:

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

(a) *Determination of Certified Rating.* Determine the certified rating for each basic model through testing pursuant to paragraph (a)(1)(ii) of this section. For single-split-systems, manufacturers must certify additional ratings for each individual combination within the same basic model either based on testing or by using an AEDM subject to the limitations of paragraph (a)(2) of this section. This includes blower coil and coil-only systems both before and after the compliance date of any amended energy conservation standards. For multi-split, multi-circuit, and single-zone-multiple-coil systems, each basic model must include a rating for a non-ducted combination and may also include ratings for a ducted combination and a mixed non-ducted/short-ducted combination per the requirements specified in this section. If individual models of single-package systems or individual combinations of split-systems that are otherwise identical are offered with multiple options for off mode-related components, rate the individual model/combination with the crankcase heater and controls that are the most consumptive. A manufacturer may also certify less consumptive off mode options; however, the manufacturer must differentiate the individual model numbers in its certification report.

(1) *Units to be tested.*

(i) *General.* The general requirements of § 429.11 apply to central air conditioners and heat pumps; and

(ii) *Model selection for testing.* (A) Except for single-split-system non-space-constrained air conditioners, determine represented values for each basic model through testing of the following, specific, individual model or combination pursuant to the table below.

Category	Equipment type	Must test each:	With:
Single-Package Unit	Single-Package AC	Basic Model	Lowest SEER individual model.

Category	Equipment type	Must test each:	With:
	Single-Package HP.		
	Space-Constrained Single-Package AC.		
	Space-Constrained Single-Package HP.		
Outdoor Unit and Indoor Unit (Rated by OUM).	Single-Split-System HP	Model of Outdoor Unit	The model of indoor unit that is likely to have the largest volume of retail sales with the particular model of outdoor unit.
	Space-Constrained Split-System AC.		
	Space-Constrained Split-System HP.		
	Multi-Split, Multi-Circuit, or Single-Zone-Multiple-Coil Split System.	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 short-ducted indoor units, a second "tested combination" composed entirely of short-ducted indoor units must be tested (in addition to the non-ducted combination). For any models of outdoor units also sold with models of SDHV indoor units, a second (or third) "tested combination" composed entirely of SDHV units must be tested (in addition to the non-ducted combination and, if tested, the short-ducted combination).
Indoor Unit Only (Rated by ICM).	Single-Split-System	Model of Indoor Unit	Least efficient model of outdoor unit with which it will be paired, where the least efficient model of outdoor unit is the outdoor unit in the lowest SEER combination as certified by the OUM). If there are multiple models of outdoor units with the same lowest-SEER rating, the ICM may select one for testing purposes.
	Small-Duct, High Velocity Systems.		
Outdoor Unit Only	Outdoor Unit Only	Model of Outdoor Unit	Model of indoor unit meeting the requirements of section 2.2e of Appendix M (or M1) to Subpart B of 10 CFR Part 430.

(B) For single-split-system, non-space-constrained air conditioners rated by OUMs, determine represented values for each basic model through testing of the following, specific, individual combination, with requirements depending on date and pursuant to the table below.

Date	Equipment type	Must test each:	With:
Before the compliance date of any amended energy conservation standards (with a compliance date after January 1, 2017).	Split-System AC with single capacity condensing unit.	Model of Outdoor Unit	The model of coil-only indoor unit that is likely to have the largest volume of retail sales with the particular model of outdoor unit.
	Split-System AC with other than single capacity condensing unit.	Model of Outdoor Unit	The model of coil-only indoor unit that is likely to have the largest volume of retail sales with the particular model of outdoor unit, unless the model of outdoor unit is only sold with model(s) of blower coil indoor units, in which case the model of blower coil indoor unit (with designated air mover as applicable) that is likely to have the largest volume of retail sales with the particular model of outdoor unit.
On or after the compliance date of any amended energy conservation standards with which compliance is required on or after January 1, 2017.	Split-system AC	Model of Outdoor Unit	The model of blower coil indoor unit that is likely to have the largest volume of retail sales with the particular model of outdoor unit.

(iii) *Sampling plans and representative values.* (A) Each basic model (for single-package systems) or individual combination (for split–systems) tested must have a sample of sufficient size tested in accordance with the applicable provisions of this subpart. The represented values for any basic model or individual combination must be assigned such that:

(*1*) Any represented value of power consumption or other measure of energy consumption for which consumers would favor lower values must be greater than or equal to the higher of:

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

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

and $\bar{x}$ is the sample mean; n is the number of samples; and x_i is the i^{th} sample; Or, (*ii*) The upper 90 percent confidence limit (UCL) of the true mean divided by 1.05, where:

$$UCL = \bar{x} + t_{.90}(\frac{s}{\sqrt{n}})$$

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

(*2*) Any represented value of the energy efficiency or other measure of energy consumption for which consumers would favor higher values shall be less than or equal to the lower of:

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

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

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

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

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

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

(*3*) The represented value of cooling capacity is the mean of the capacities measured for the sample, rounded:

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

(*ii*) 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

(*iii*) 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.

(*4*) The represented value of heating capacity is the mean of the capacities measured for the sample, rounded:

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

(*ii*) 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

(*iii*) 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.

(*5*) The represented value of sensible heat ratio (SHR) is the mean of the SHR measured for the sample, rounded to the nearest percent (%).

(B) 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.

(C) 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:

(*1*) The quotient of the represented value of cooling capacity, in Btu's per hour as determined in paragraph (a)(1)(iii)(A)(*3*) of this section, divided by the represented value of SEER, in Btu's per watt-hour, as determined in paragraph (a)(1)(iii)(A)(*2*) of this section;

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

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

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

(D) 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:

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

(*i*) 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 (a)(1)(iii)(A)(*2*) of this section;

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

(*iii*) 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;

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

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

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

(*i*) 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 (a)(1)(iii)(A)(*3*) of this section, or the represented value of heating capacity (for air-source heat pumps that provide only heating), as determined in paragraph (a)(1)(iii)(A)(*4*) 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 (a)(1)(iii)(A)(*2*) of this section;

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

(*iii*) The adjustment factor of 1.30, which serves to adjust the calculated design heating requirement and heating load hours to the actual load experienced by a heating system;

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

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

(E) Determine the represented value of estimated annual operating cost for air-source heat pumps that provide both heating and cooling by calculating the sum of the quantity determined in paragraph (a)(1)(iii)(C) of this section added to the quantity determined in paragraph (a)(1)(iii)(D) of this section.

(F) 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:

(*1*) The quotient of the represented value of cooling capacity, in Btu's per hour, determined in paragraph (a)(1)(iii)(A)(*3*) of this section divided by the represented value of SEER, in Btu's per watt-hour, determined in paragraph (a)(1)(iii)(A)(*2*) of this section;

(*2*) The estimated number of regional cooling load hours per year determined from Table 21 in section 4.3.2 of appendix M or Table 20 in section 4.3.2 of appendix M1, as applicable, to subpart B of part 430;

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

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

(G) 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:

(*1*) When using Appendix M to subpart B of Part 430, the product of:

(*i*) The estimated number of regional heating load hours per year determined from Table 21 in section 4.3.2 of appendix M to subpart B of part 430;

(*ii*) 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 (a)(1)(iii)(A)(*2*);

(*iii*) 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;

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

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

(*2*) When using Appendix M1 to subpart B of Part 430, the product of:

(*i*) The estimated number of regional heating load hours per year determined from Table 20 in section 4.2 of appendix M1 to subpart B of Part 430;

(*ii*) 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 section (a)(1)(iii)(A)(*3*), or the represented value of heating capacity (for air-source heat pumps that provide only heating), as determined in section (a)(1)(iii)(A)(*4*), 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 (a)(1)(iii)(A)(*2);*

(*iii*) The adjustment factor of 1.30, which serves to adjust the calculated design heating requirement and heating load hours to the actual load experienced by a heating system;

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

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

(H) For air-source heat pumps that provide both heating and cooling, the estimated regional annual operating cost is the sum of the quantity determined in paragraph (a)(1)(iii)(F) of this section added to the quantity determined in paragraph (a)(1)(iii)(G) of this section.

(I) The cooling mode efficiency measure for cooling-only units and for air-source heat pumps that provide cooling is the represented value of the SEER, in Btu's per watt-hour, pursuant to paragraph (a)(1)(iii)(A)(*2*) of this section.

(J) The heating mode efficiency measure for air-source heat pumps is the represented value of the HSPF, in Btu's per watt-hour for each applicable standardized design heating requirement within each climatic region, pursuant to paragraph (a)(1)(iii)(A)(*2*) of this section.

(K) Round represented values of estimated annual operating cost to the nearest dollar per year. Round represented values of EER, SEER, HSPF, and APF to the nearest 0.05. Round represented values of off-mode power consumption, pursuant to paragraph (a)(1)(iii)(A)(*1)* to the nearest watt.

(2) *Units not required to be tested.*

(i) *For basic models rated by ICMs and single-split-system air conditioners, split-system heat pumps, space-constrained split-system heat pumps, and space-constrained split-system air conditioners.* For every individual combination within a basic model other than the individual combination required to be tested pursuant to paragraph (a)(1)(ii) of this section, either:

(A) A sample of sufficient size, comprised of production units or representing production units, must be tested as complete systems with the resulting ratings for the combination obtained in accordance with paragraphs (a)(1)(i) and (iii) of this section; or

(B) The representative values of the measures of energy efficiency must be assigned through the application of an AEDM in accordance with paragraph (a)(3) of this section and § 429.70. An AEDM may only be used to rate individual combinations in a basic model other than the combination required for mandatory testing under paragraph (a)(1)(ii) of this section. No basic model may be rated with an AEDM.

(ii) *For multi-split systems, multi-circuit systems, and single-zone-multiple-coil systems.* The following applies:

(A) For basic models composed of both non-ducted and short-ducted units, the represented value for the mixed non-ducted/short-ducted combination is the mean of the represented values for the non-ducted and short-ducted combinations as determined in accordance with paragraph (a)(1)(iii)(A) of this section.

(B) 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 a different basic model, and the provisions of (a)(1)(i) through (a)(1)(iii) and (a)(2)(ii)(A) of this section apply.

(3) *Alternative efficiency determination methods.* In lieu of testing, represented values of efficiency or consumption may be determined through the application of an AEDM pursuant to the requirements of § 429.70 and the provisions of this section.

(i) *Power or energy consumption.* Any represented value of the average off mode power consumption or other measure of energy consumption of an individual combination for which consumers would favor lower values must be greater than or equal to the output of the AEDM.

(ii) *Energy efficiency.* Any represented value of the SEER, EER, HSPF or other measure of energy efficiency of an individual combination for which consumers would favor higher values must be less than or equal to the output of the AEDM.

(b) *Limitations.* The following section explains the limitations for certification of models.

(1) *Regional.* Any model of outdoor unit that is certified in a combination that does not meet all regional standards cannot also be certified in a combination that meets the regional standard(s). Outdoor unit model numbers cannot span regions unless the model of outdoor unit is compliant with all standards in all possible combinations. If a model of outdoor unit is certified below a regional standard, then it must have a unique individual model number for distribution in each region.

(2) *Multiple product classes.* Models of outdoor units that are rated and distributed in combinations that span multiple product classes must be tested and certified pursuant to paragraph (a) as compliant with the applicable standard for each product class.

(c) *Certification reports.* This paragraph specifies the information that must be included in a certification report.

(1) *General.* The requirements of § 429.12 apply to central air conditioners and heat pumps.

(2) *Public product-specific information.* Pursuant to § 429.12(b)(13), for each basic model (for single-package systems) or individual combination (for 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 sensible heat ratio calculated based on full-load cooling conditions at the outdoor ambient conditions of 82 °F dry bulb and 65 °F wet bulb; and

(i) For heat pumps, the heating seasonal performance factor (HSPF in British thermal units per Watt-hour (Btu/W-h));

(ii) For air conditioners (excluding space constrained), the energy efficiency ratio (EER in British thermal units per Watt-hour (Btu/W-h));

(iii) For single-split-system equipment, whether the rating is for a coil-only or blower coil system; and

(iv) For multi-split, multiple-circuit, and single-zone-multiple-coil systems (including VRF), whether the rating is for a non-ducted, short-ducted, SDHV, or mixed non-ducted and short-ducted system.

(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 No(s).		
		1	2	3
Single Package	Number unique to the basic model.	Package	N/A	N/A.
Split System (rated by OUM).	Number unique to the basic model.	Outdoor Unit	Indoor Unit(s)	Air Mover (or N/A if rating coil-only system or fan is part of indoor unit model number).
Outdoor Unit Only	Number unique to the basic model.	Outdoor Unit	N/A	N/A.
Split-System or SDHV (rated by ICM).	Number unique to the basic model.	Outdoor Unit	Indoor Unit(s)	N/A.

(4) *Additional product-specific information.* Pursuant to § 429.12(b)(13), for each individual model/combination, 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 (cfm)); 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 (cfm) 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; the C_D^c value used to represent cooling mode cycling losses; the temperatures at which the crankcase heater with controls is designed to turn on and designed to turn off for the heating season, 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; and

(i) For heat pumps, the C_D^h value used;

(ii) For multi-split, multiple-circuit, and single-zone-multiple-coil 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) are operating 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 models tested with an indoor blower installed, 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 units, 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 unit controls restrict use of minimum compressor speed operation for some range of operating ambient conditions, whether the unit controls restrict use of maximum compressor speed operation for any ambient temperatures below 17 °F, and whether the optional $H4_2$ low temperature test was used to characterize performance at temperatures below 17 °F.

(d) *Alternative efficiency determination methods.* Alternative methods for determining efficiency or energy use for central air conditioners and heat pumps can be found in § 429.70(e) of this subpart.

■ 4. Amend § 429.70 by revising paragraph (e) to read as follows:

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

* * * * *

(e) *Alternate Efficiency Determination Method (AEDM) for central air conditioners and heat pumps.* This paragraph sets forth the requirements for a manufacturer to use an AEDM to rate central air conditioners and heat pumps

(1) *Criteria an AEDM must satisfy.* A manufacturer may not apply an AEDM to an individual combination to determine its certified ratings (SEER, EER, HSPF, and/or $P_{W,OFF}$) pursuant to this section unless authorized pursuant to § 429.16(a)(2) and:

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

(ii) The manufacturer has validated the AEDM in accordance with paragraph (e)(2) of this section and using individual combinations that meet the current Federal energy conservation standards.

(2) *Validation of an AEDM.* Before using an AEDM, the manufacturer must validate the AEDM's accuracy and reliability as follows:

(i) The manufacturer must complete testing of each basic model as required under § 429.16(a)(1)(ii). Using the AEDM, calculate the energy use or efficiency for each of the tested individual combinations within each basic model. Compare the rating 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.

(ii) *Individual combination tolerances.* This paragraph provides the tolerances applicable to individual combinations rated using an AEDM.

(A) For an energy-efficiency metric, the predicted efficiency for each individual combination calculated by applying the AEDM may not be more than three percent greater than the efficiency determined from the corresponding test of the combination.

(B) For an energy-consumption metric, the predicted energy consumption for each individual combination, calculated by applying the AEDM, may not be more than three percent less than the energy consumption determined from the corresponding test of the combination.

(C) The predicted energy efficiency or consumption for each individual combination calculated by applying the AEDM must meet or exceed the applicable federal energy conservation standard.

(iii) *Additional test unit requirements.* Each test must have been performed in accordance with the DOE test procedure applicable at the time the individual combination being rated with the AEDM is distributed in commerce.

(3) *AEDM records retention requirements.* If a manufacturer has used an AEDM to determine representative values pursuant to this section, the manufacturer must have available upon request for inspection by the Department records showing:

(i) The AEDM, including the mathematical model, the engineering or statistical analysis, and/or computer simulation or modeling that is the basis of the AEDM;

(ii) Product information, complete test data, AEDM calculations, and the statistical comparisons from the units tested that were used to validate the AEDM pursuant to paragraph (e)(2) of this section; and

(iii) Product information and AEDM calculations for each individual combination certified using the AEDM.

(4) *Additional AEDM requirements.* If requested by the Department and at DOE's discretion, the manufacturer must perform at least one of the following:

(i) Conduct simulations before representatives of the Department to predict the performance of particular individual combinations; or

(ii) Provide analyses of previous simulations conducted by the manufacturer; or

(iii) Conduct certification testing of individual combinations selected by the Department.

(5) *AEDM verification testing.* DOE may use the test data for a given individual combination generated pursuant to § 429.104 to verify the certified rating determined by an AEDM as long as the following process is followed:

(i) *Selection of units.* DOE will obtain one or more units for test from retail, if available. If units cannot be obtained from retail, DOE will request that a unit be provided by the manufacturer;

(ii) *Lab requirements.* DOE will conduct testing at an independent, third-party testing facility of its choosing. In cases where no third-party laboratory is capable of testing the equipment, testing may be conducted at a manufacturer's facility upon DOE's request.

(iii) *Testing.* At no time during verification testing may the lab and the manufacturer communicate without DOE authorization. If during test set-up or testing, the lab indicates to DOE that it needs additional information regarding a given individual combination in order to test in accordance with the applicable DOE test procedure, DOE may organize a meeting between DOE, the manufacturer and the lab to provide such information.

(iv) *Failure to meet certified rating.* If an individual combination tests worse than its certified rating (*i.e.,* lower than the certified efficiency rating or higher than the certified consumption rating) by more than 5%, or the test results in a different cooling capacity than its certified cooling capacity by more than 5%, 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:

(A) May present any and all claims regarding testing validity; and

(B) If not on site for the initial test set-up, must test at least one additional unit of the same combination obtained from a retail source at its own expense, following the test requirements in § 429.110(a)(3). When testing at an independent lab, the manufacturer may choose to have DOE and the manufacturer present.

(v) *Tolerances.* This subparagraph specifies the tolerances DOE will permit when conducting verification testing.

(A) For consumption metrics, the result from a DOE verification test must be less than or equal to 1.05 multiplied by the certified rating.

(B) For efficiency metrics, the result from a DOE verification test must be greater than or equal to 1.05 multiplied by the certified rating.

(vi) *Invalid rating.* If, following discussions with the manufacturer and a retest where applicable, DOE determines that the verification testing was conducted appropriately in accordance with the DOE test procedure, DOE will issue a determination that the ratings for the basic model are invalid. The manufacturer must conduct additional testing and re-rate and re-certify the individual combinations within the basic model that were rated using the AEDM based on all test data collected, including DOE's test data.

(vii) *AEDM use.* This subparagraph specifies when a manufacturer's use of an AEDM may be restricted due to prior invalid ratings.

(A) If DOE has determined that a manufacturer made invalid ratings on individual combinations within two or more basic models rated using the manufacturer's AEDM within a 24 month period, the manufacturer must test the least efficient and most efficient combination within each basic model in addition to the combination specified in § 429.16(a)(1)(ii). The twenty-four month period begins with a DOE determination that a rating is invalid through the process outlined above.

(B) If DOE has determined that a manufacturer made invalid ratings on more than four basic models rated using the manufacturer's AEDM within a 24-month period, the manufacturer may no longer use an AEDM.

(C) If a manufacturer has lost the privilege of using an AEDM, the manufacturer may regain the ability to use an AEDM by:

(*1*) Investigating and identifying cause(s) for failures;

(*2*) Taking corrective action to address cause(s);

(*3*) Performing six new tests per basic model, a minimum of two of which must be performed by an independent, third-party laboratory from units obtained from retail to validate the AEDM; and

(*4*) Obtaining DOE authorization to resume use of an AEDM.

* * * * *

■ 5. Amend § 429.134 by adding paragraph (g) to read as follows:

§ 429.134 Product-specific enforcement provisions.

* * * * *

(g) *Central air conditioners and heat pumps.*—(1) *Verification of cooling capacity.* The cooling capacity of each tested unit of the basic model (for single package systems) or individual combination (for split-systems) will be measured pursuant to the test requirements of § 430.23(m). The results of the measurement(s) will be compared to the value of cooling capacity certified by the manufacturer.

(i) If the measurement(s) (either the measured cooling capacity for a single unit sample or the average of the measured cooling capacities for a multiple unit sample) is less than or equal to 1.05 multiplied by the certified cooling capacity and greater than or equal to 0.95 multiplied by the certified cooling capacity, the certified cooling capacity will be used as the basis for determining SEER.

(ii) Otherwise, the measurement(s) (either the measured cooling capacity for a single unit sample or the average of the measured cooling capacities for a multiple unit sample, as applicable) will be used as the basis for determining SEER.

(2) *Verification of* C_D *value*—(i) For central air conditioners and heat pumps other than models of outdoor units with no match, the C_D^c and/or C_D^h value of the basic model (for single package systems) or individual combination (for split-systems), as applicable, will be measured pursuant to the test requirements of § 430.23(m) for each unit tested. The results of the measurement(s) for each C_D^c or C_D^h value will be compared to the C_D^c or C_D^h value certified by the manufacturer.

(A) If the results of the measurement(s) (either the measured value for a single unit sample or the average of the measured values for a multiple unit sample) is 0.02 or more greater than the certified C_D^c or C_D^h value, the average measured C_D^c or C_D^h value will serve as the basis for calculation of SEER or HSPF for the basic model/individual combination.

(B) For all other cases, the certified C_D^c or C_D^h value will be used as the basis for calculation of SEER or HSPF for the basic model/individual combination.

(ii) For models of outdoor units with no match, or for tests in which the criteria for the cyclic test in 10 CFR part 430, subpart B, Appendix M or M1, as applicable, section 3.5e, cannot be achieved, DOE will use the default C_D^c and/or C_D^h value pursuant to 10 CFR part 430.

PART 430—ENERGY CONSERVATION PROGRAM FOR CONSUMER PRODUCTS

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

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

■ 7. Section 430.2 is amended by:

■ a. Removing the definitions of ''ARM/simulation adjustment factor,'' ''coil family,'' ''condenser-evaporator coil combination'', ''condensing unit,'' ''evaporator coil'', ''heat pump,'' ''indoor unit,'' ''outdoor unit,'' ''small duct, high velocity system,'' and ''tested combination;'' and

■ b. Revising the definitions of ''basic model;'' and ''central air conditioner'' to read as follows:

§ 430.2 Definitions.

* * * * *

Basic model means all units of a given type of covered product (or class thereof) manufactured by one manufacturer; having the same primary energy source; and, which have essentially identical electrical, physical, and functional (or hydraulic) characteristics that affect energy consumption, energy efficiency, water consumption, or water efficiency; and

(1) With respect to general service fluorescent lamps, general service incandescent lamps, and incandescent reflector lamps: Lamps that have essentially identical light output and electrical characteristics—including lumens per watt (lm/W) and color rendering index (CRI).

(2) With respect to faucets and showerheads: Have the identical flow control mechanism attached to or installed within the fixture fittings, or the identical water-passage design features that use the same path of water in the highest flow mode.

(3) With respect to furnace fans: Are marketed and/or designed to be installed in the same type of installation; and

(4) With respect to central air conditioners and central air conditioning heat pumps:

(i) Essentially identical electrical, physical, and functional (or hydraulic) characteristics means:

(A) For split-systems manufactured by independent coil manufacturers (ICMs) and for small-duct, high velocity systems: All individual combinations having the same model of indoor unit, which means the same or comparably performing indoor coil(s) [same face area; fin material, depth, style (*e.g.,* wavy, louvered), and density (fins per inch); tube pattern, material, diameter, wall thickness, and internal enhancement], indoor blower(s) [same air flow with the same indoor coil and external static pressure, same power input], auxiliary refrigeration system components if present (*e.g.,* expansion valve), and controls.

(B) for split-systems manufactured by outdoor unit manufacturers (OUMs): All individual combinations having the same model of outdoor unit, which means the same or comparably performing compressor(s) [same displacement rate (volume per time) and same capacity and power input when tested under the same operating conditions], outdoor coil(s) [same face area; fin material, depth, style (*e.g.,* wavy, louvered), and density (fins per inch); tube pattern, material, diameter, wall thickness, and internal enhancement], outdoor fan(s) [same air flow with the same outdoor coil, same power input], auxiliary refrigeration system components if present (*e.g.,* suction accumulator, reversing valve, expansion valve), and controls.

(C) for single-package models: All individual models having the same or comparably performing compressor(s) [same displacement rate (volume per time) and same capacity and power input when tested under the same operating conditions], outdoor coil(s) and indoor coil(s) [same face area; fin material, depth, style (*e.g.,* wavy, louvered), and density (fins per inch); tube pattern, material, diameter, wall thickness, and internal enhancement], outdoor fan(s) [same air flow with the same outdoor coil, same power input], indoor blower(s) [same air flow with the same indoor coil and external static pressure, same power input], auxiliary refrigeration system components if present (*e.g.* suction accumulator, reversing valve, expansion valve), and controls.

(ii) For single-split-system and single-package models, manufacturers may instead choose to make each individual combination or model its own basic model provided the testing and rating requirements in 10 CFR 429.16 are met.

(iii) For multi-split, multi-circuit, and single-zone-multiple-coil models, a basic model may not include both individual SDHV combinations and non-SDHV combinations even when they include the same model of outdoor unit. The manufacturer may choose to identify specific individual combinations as additional basic models.

* * * * *

Central air conditioner or central air conditioning heat pump means a product, other than a packaged terminal air conditioner or packaged terminal heat pump, which is powered by single phase electric current, air cooled, rated below 65,000 Btu per hour, not contained within the same cabinet as a furnace, the rated capacity of which is above 225,000 Btu per hour, and is a heat pump or a cooling unit only. A central air conditioner or central air conditioning heat pump may consist of: A single-package unit; an outdoor unit and one or more indoor units; an indoor unit only; or an outdoor unit only. In the case of an indoor unit only or an outdoor unity only, the unit must be tested and rated as a system (combination of both an indoor and an outdoor unit). For all central air conditioner and central air conditioning heat pump-related definitions, see appendices M or M1 of subpart B of this part.

■ 8. Section 430.3 is amended by:

■ a. Revising paragraphs (c)(1) and (g)(2);

■ b. Adding paragraphs (c)(3) and (c)(4);

■ c. Removing paragraphs (g)(3);

■ d. Redesignating paragraphs (g)(4) through (g)(14) as (g)(3) through (g)(13); and

■ e. Revising newly redesignated (g)(3) through (g)(9).

The revisions and additions read as follows:

§ 430.3 Materials incorporated by reference.

* * * * *

(c) * * *

(1) AHRI 210/240–2008 with Addendums 1 and 2 (formerly ARI Standard 210/240), Performance Rating of Unitary Air-Conditioning & Air-Source Heat Pump Equipment, sections 6.1.3.2, 6.1.3.4, 6.1.3.5 and figures D1, D2, D4, approved by ANSI December, 2012, IBR approved for appendix M and M1 to subpart B.

* * * * *

(3) ANSI/AHRI 1230–2010 with Addendum 2, Performance Rating of Variable Refrigerant Flow Multi-Split Air-Conditioning and Heat Pump Equipment, sections 3 (except 3.8, 3.9, 3.13, 3.14, 3.15, 3.16, 3.23, 3.24, 3.26, 3.27, 3.28, 3.29, 3.30, and 3.31), 5.1.3, 5.1.4, 6.1.5 (except Table 8), 6.1.6, and 6.2, approved August 2, 2010, Addendum 2 dated June 2014, IBR approved for appendices M and M1 to subpart B.

(4) AHRI 210/240-Draft, Performance Rating of Unitary Air-Conditioning & Air-Source Heat Pump Equipment, appendix E, section E4, Docket No. EERE–2009–BT–TP–0004 No. 45.

* * * * *

(g) * * *

(2) ASHRAE 23.1–2010, Methods of Testing for Rating the Performance of Positive Displacement Refrigerant Compressors and Condensing Units that Operate at Subcritical Temperatures of the Refrigerant, sections 5, 6, 7, and 8 only, approved January 28, 2010, IBR approved for appendices M and M1 to subpart B.

(3) ASHRAE 37–2009, Methods of Testing for Rating Electrically Driven Unitary Air-Conditioning and Heat Pump Equipment, approved June 25, 2009, IBR approved for appendix AA subpart to B. Sections 5.1.1, 5.2, 5.5.1, 6.1.1, 6.1.2, 6.1.4, 6.4, 6.5, 7.3, 7.4, 7.5, 7.7.2.1, 7.7.2.2, 8.1.2, 8.1.3, 8.2, 8.6.2; figures 1, 2, 4, 7a, 7b, 7c, 8; and table 3 only IBR approved for appendices M and M1 to subpart B.

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(4) ASHRAE 41.1–1986 (Reaffirmed 2006), Standard Method for Temperature Measurement, approved February 18, 1987, IBR approved for appendices E and AA to subpart B.

(5) ASHRAE 41.1–2013, Standard Method for Temperature Measurement, approved January 30, 2013, IBR approved for appendix X1 to subpart B. Sections 4, 5, 6, 7.2, and 7.3 only, IBR approved for appendices M and M1 to subpart B.

(6) ASHRAE 41.2–1987 (Reaffirmed 1992), Standard Methods for Laboratory Airflow Measurement, section 5.2.2 and figure 14, approved October 1, 1987, IBR approved for appendices M and M1 to subpart B.

(7) ASHRAE 41.6–2014, Standard Method for Humidity Measurement, sections 4, 5, 6, and 7.1, approved July 3, 2014, sections 4, 5, 6, and 7 only IBR approved for appendices M and M1 to subpart B.

(8) ASHRAE 41.9–2011, Standard Methods for Volatile-Refrigerant Mass Flow Measurements Using Calorimeters, approved February 3, 2011, sections 5, 6, 7, 8, 9, and 11 only IBR approved for appendices M and M1 to subpart B.

(9) ASHRAE/AMCA 51–07/210–07, Laboratory Methods of Testing Fans for Certified Aerodynamic Performance Rating, figures 2A and 12, approved August 17, 2007, IBR approved for appendices M and M1 to subpart B.

* * * * *

■ 9. 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. All values discussed in this section must be determined using a single appendix.

(1) Cooling capacity must be determined 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) Seasonal energy efficiency ratio (SEER) must be determined from section 4.1 of appendix M or M1 to this subpart, and rounded off to the nearest 0.025 Btu/W-h.

(3) When representations are made of energy efficiency ratio (EER), EER must be determined in section 4.7 of appendix M or M1 to this subpart, and rounded off to the nearest 0.025 Btu/W-h.

(4) Heating seasonal performance factors (HSPF) must be determined in section 4.2 of appendix M or M1 to this subpart, and rounded off to the nearest 0.025 Btu/W-h.

(5) Average off mode power consumption must be determined according to section 4.3 of appendix M or M1 to this subpart, and rounded off to the nearest 0.5 W.

(6) Sensible heat ratio (SHR) must be determined according to section 4.6 of appendix M or M1 to this subpart, and rounded off to the nearest 0.5 percent (%).

(7) All other measures of energy efficiency or consumption or other useful measures of performance must be determined using appendix M or M1 of this subpart.

* * * * *

■ 10. Revise appendix M to subpart B of part 430 to read as follows:

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

Note: Prior to May 9, 2016, any representations, including compliance certifications, made with respect to the energy use, power, or efficiency of central air conditioners and central air conditioning heat pumps must be based on the results of testing pursuant to either this appendix or the procedures in Appendix M as it appeared at 10 CFR part 430, subpart B, Appendix M, in the 10 CFR parts 200 to 499 edition revised as of January 1, 2015. Any representations made with respect to the energy use or efficiency of such central air conditioners and central air conditioning heat pumps must be in accordance with whichever version is selected.

On or after May 9, 2016 and prior to the compliance date for any amended energy conservation standards, any representations, including compliance certifications, made with respect to the energy use, power, or efficiency of central air conditioners and central air conditioning heat pumps must be based on the results of testing pursuant to this appendix.

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

1. Scope and Definitions

1.1 Scope.

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

(a) Split-system air conditioners; and single-zone-multiple-coil, multi-split (including VRF), and multi-circuit systems

(b) Split-system heat pumps and single-zone-multiple-coil, multi-split (including VRF), and multi-circuit systems

(c) Single-package air conditioners

(d) Single-package heat pumps

(e) Small-duct, high-velocity systems (including VRF)

(f) Space-constrained products—air conditioners

(g) Space-constrained products—heat pumps

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

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

1.2 Definitions.

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

Airflow prevention device denotes a device(s) 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.

Annual performance factor means the total heating and cooling done by a heat pump in a particular region in one year divided by the total electric energy used in one year.

Blower coil indoor unit means the indoor unit of a split-system central air conditioner or heat pump that includes a refrigerant-to-air heat exchanger coil, may include a cooling-mode expansion device, and includes either an indoor blower housed with the coil or 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.

CFR means Code of Federal Regulations.

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

Coil-only indoor unit means the indoor unit of a split-system central air conditioner or heat pump that includes a refrigerant-to-air heat exchanger coil and may include a cooling-mode expansion device, but does not include an indoor blower housed with the coil, and does not include a separate designated air mover such as a furnace or a modular blower (as defined in Appendix AA to this subpart. A coil-only indoor unit is designed to use a separately-installed furnace or a modular blower for indoor air movement. *Coil-only system* refers to a system that includes one or more coil-only indoor units.

Condensing unit removes the heat absorbed by the refrigerant to transfer it to the outside environment, and which 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 5 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. The denominator is the total cooling that would be delivered, given the same ambient conditions, had the unit operated continuously at its steady-state, space-cooling capacity for the same total time (ON + OFF) interval.

Crankcase heater means any electrically powered device or mechanism for intentionally generating heat within and/or around the compressor sump volume often done to minimize the dilution of the compressor's refrigerant oil by condensed refrigerant. 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.

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 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. One acceptable alternative to the criterion given in the prior sentence is a feedback system that measures the length of the defrost period and adjusts defrost frequency accordingly. In all cases, when the frost parameter(s) reaches a predetermined value, the system initiates a defrost. In a demand-defrost control system, defrosts are terminated based on monitoring a parameter(s) that indicates that frost has been eliminated from the coil. (**Note:** Systems that vary defrost intervals according to outdoor dry-bulb temperature are not demand-defrost systems.) A demand-defrost control system, which otherwise meets the above requirements, may allow time-initiated defrosts if, and only if, such defrosts occur after 6 hours of compressor operating time.

Design heating requirement (DHR) predicts the space heating load of a residence when subjected to outdoor design conditions. Estimates for the minimum and maximum

DHR are provided for six generalized U.S. climatic regions in section 4.2.

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

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. These rate quantities must be determined from a single test or, if derived via interpolation, must be determined at a single set of operating conditions. EER is expressed in units of

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

When determined for a ducted unit tested without an indoor blower installed, EER must include the section 3.3 and 3.5.1 default values for the heat output and power input of a fan motor.

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

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

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

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 space heating season, expressed in Btu's, divided by the total electrical energy consumed by the heat pump system during the same season, expressed in watt-hours. The HSPF used to evaluate compliance with the Energy Conservation Standards (see 10 CFR 430.32(c)) is based on Region IV, the minimum standardized design heating requirement, 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 transfers heat between the refrigerant and the indoor air and consists of an indoor coil and casing and may include a cooling mode expansion device and/or an air moving device.

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

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

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

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, the shoulder season and the entire heating season; and for heat pumps, the shoulder season only.

Outdoor unit 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, could include a heating mode expansion device, reversing valve, and 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 energy efficiency ratio (coefficient of performance) to the steady-state energy efficiency ratio (coefficient of performance), where both energy efficiency ratios (coefficients of performance) 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.

Short ducted system means a ducted split system whose one or more indoor sections produce greater than zero but no greater than 0.1 inches (of water) of external static pressure when operated at the full-load air volume not exceeding 450 cfm per rated ton of cooling.

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 that has one indoor coil-only or indoor blower coil unit connected to its other component(s) with a single refrigeration circuit.

Single-zone-multiple-coil 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.

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

Split system means any air conditioner or heat pump that has one or more of the major assemblies separated from the others. 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 single-zone-multiple-coil, 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 shall:

(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) Represent the highest sales volume model family that can meet the 95 percent nominal cooling capacity of the outdoor unit [**Note:** another indoor model family may be used if five indoor units from the highest sales volume model family do not provide sufficient capacity to meet the 95 percent threshold level].

(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.

(vi) Where referenced, ''nominal cooling capacity'' is to be interpreted for indoor units as 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 as the lowest cooling capacity listed in published product literature for these conditions. If incomplete or no operating conditions are reported, the highest (for indoor units) or lowest (for outdoor units) such cooing capacity shall be used.

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

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

In a second application of the control scheme, one or more parameters are measured (*e.g.,* air and/or refrigerant temperatures) at the predetermined, cumulative, compressor ON-time. A defrost is initiated only if the measured parameter(s) falls within a predetermined range. The ON-time counter is reset regardless of whether or not a defrost is initiated. If systems of this second type use cumulative ON-time intervals of 10 minutes or less, then the heat pump may qualify as having a demand defrost control system (see definition).

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

Triple-split system means a central air conditioner or heat pump that is composed of three separate components: An outdoor fan coil section, an indoor blower coil section, 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 for heating mode tests may be the same or different from the cooling mode value.

For such systems, high capacity means the compressor(s) operating at low 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 certified indoor coil model number should reflect whether the ratings pertain to the lockout enabled option via the inclusion of an extra identifier, such as ''+LO''. When testing as a two-capacity, northern heat pump, the lockout feature must remain enabled for all tests.

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. Single-phase VRF systems less than 65,000 Btu/h are a kind of 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.

For such a system, maximum speed means the maximum operating speed, measured by RPM or frequency (Hz), that the unit is designed to operate in cooling mode or heating mode. Maximum speed does not change with ambient temperature, and it can be different from cooling mode to heating mode. Maximum speed does not necessarily mean maximum capacity.

For such systems, minimum speed means the minimum speed, measured by RPM or frequency (Hz), that the unit is designed to operate in cooling mode or heating mode. Minimum speed does not change with ambient temperature, and it can be different from cooling mode to heating mode. Minimum speed does not necessarily mean minimum capacity.

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 ANSI/AHRI Standard 1230–2010 sections 3 (except 3.8, 3.9, 3.13, 3.14, 3.15, 3.16, 3.23, 3.24, 3.26, 3.27, 3.28, 3.29, 3.30, and 3.31), 5.1.3, 5.1.4, 6.1.5 (except Table 8), 6.1.6, and 6.2 (incorporated by reference, see § 430.3) and Appendix M. Where ANSI/AHRI Standard 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 ANSI/AHRI Standard 1230–2010.

For definitions use section 1 of Appendix M and section 3 of ANSI/AHRI Standard 1230–2010, excluding sections 3.8, 3.9, 3.13, 3.14, 3.15, 3.16, 3.23, 3.24, 3.26, 3.27, 3.28, 3.29, 3.30, and 3.31. For rounding requirements refer to § 430.23 (m). For determination of certified rating requirements refer to § 429.16.

For test room requirements, refer to section 2.1 from Appendix M. 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 from Appendix M, and sections 5.1.3 and 5.1.4 of ANSI/AHRI Standard 1230–2010.

For general requirements for the test procedure refer to section 3.1 of Appendix M, 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 Table 8) and 6.1.6 of ANSI/AHRI Standard 1230–2010. For external static pressure requirements, refer to Table 3 in Appendix M.

For the test procedure, refer to sections 3.3 to 3.5 and 3.7 to 3.13 in Appendix M. For cooling mode and heating mode test conditions, refer to section 6.2 of ANSI/AHRI Standard 1230–2010. For calculations of seasonal performance descriptors use section 4 of Appendix M.

(B) For systems other than VRF, only a subset of the sections listed in this test procedure apply when testing and rating 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. 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.

BILLING CODE 6450–01–P

			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; 3.1.10; 3.7a,b,d; 3.8a,d; 3.8.1; 3.9; 3.10	4.4; 4.5; 4.6	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;2.4d; 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; 2.4.1; 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		3.1.7	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 VRF SDHV†	2.1; 2.2.a; 2.2.b; 2.2.c; 2.2.1; 2.2.2; 2.2.3(a); 2.2.3(c);, 2.2.4; 2.2.5; 2.4-2.12	3.1 (except 3.1.3, 3.1.4) 3.1.4.1.1c; 3.11-3.13	3.3-3.5	3.7–3.10	4.4; 4.5; 4.6	4.1	4.2
	Modulation Capability	Single speed compressor, fixed speed fan			3.2.1	3.6.1		4.1.1	4.2.1
		Single speed compressor, VAV fan		3.1.7	3.2.2	3.6.2		4.1.2	4.2.2
		Two-capacity compressor		3.1.10	3.2.3	3.6.3		4.1.3	4.2.3
		Variable speed compressor			3.2.4	3.6.4		4.1.4	4.2.4
	Special Features	Heat pump with heat comfort controller		3.1.9		3.6.5			4.2.5
		Units with a multi-speed outdoor fan	2.2.2						
		Single indoor unit having multiple blowers			3.26	3.6.2; 3.6.7		4.1.5	4.2.7

BILLING CODE 6450–01–C

* Does not apply to heating-only heat pumps.

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

† Use ANSI/AHRI Standard 1230–2010 with Addendum 2, 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 rating VRF multiple-split and VRF SDHV systems.

NOTE: For all units, use section 3.13 for off mode testing procedures and section 4.3 for off mode calculations. For all units subject to an EER standard, use section 4.7 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 available indoor test rooms as needed to accommodate the total number of indoor units. These rooms must comply with the requirements specified in sections 8.1.2 and 8.1.3 of ASHRAE Standard 37–2009 (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. When applied, cycle the heater located in the same room as the test unit evaporator coil ON and OFF when the test unit cycles ON and OFF. Cycle the heater located in the same room as the test unit condensing coil ON and OFF when the test unit cycles OFF and ON.

2.2 Test unit installation requirements.

a. Install the unit according to section 8.2 of ASHRAE Standard 37–2009 (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) with Addendum 1 and 2. For the vapor refrigerant line(s), use the insulation included with the unit; if no insulation is provided, refer to 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, refer to 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;

(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 thermostatic expansion valve with internal pressure equalization that the valve manufacturer's product literature indicates is appropriate for the system.

(3) When testing triple-split systems (see section 1.2, Definitions), use the tubing length specified in section 6.1.3.5 of AHRI 210/240–2008 (incorporated by reference, see § 430.3) with Addendum 1 and 2 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; or

(4) When testing split systems having multiple indoor coils, connect each indoor blower-coil to the outdoor unit using: (a) 25 feet of tubing, or (b) tubing furnished by the manufacturer, whichever is longer.

If they are needed to make a secondary measurement of capacity, install refrigerant pressure measuring instruments as described in section 8.2.5 of ASHRAE Standard 37–2009 (incorporated by reference, see § 430.3). Refer to section 2.10 of this appendix to learn which secondary methods require refrigerant pressure measurements. At a minimum, insulate the low-pressure line(s) of a split system with insulation having an inside diameter that matches the refrigerant tubing and a nominal thickness of 0.5 inch.

b. For units designed for both horizontal and vertical installation or for both up-flow and down-flow vertical installations, the manufacturer must use the orientation for testing specified 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, 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). Except as noted in section 3.1.10, prevent the indoor air supplementary heating coils from operating during all tests. For coil-only indoor units that are supplied without an enclosure, create an enclosure using 1 inch fiberglass ductboard having a nominal density of 6 pounds per cubic foot. Or alternatively, use some other insulating material having a thermal resistance (''R'' value) between 4 and 6 hr·ft^2· °F/Btu. For units where the coil is housed within an enclosure or cabinet, no extra insulating or sealing is allowed.

d. When testing coil-only central air conditioners and heat pumps, install a toroidal-type transformer to power the system's low-voltage components, complying with any additional requirements for this transformer mentioned in the installation manuals included with the unit by the 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 that results in the transformer being loaded at a level that is between 25 and 90 percent based on the highest power value expected and then confirmed during the off mode test; (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. The power consumption of the components connected to the transformer must be included as part of the total system power consumption during the off mode tests, less if included the power consumed by the transformer when no load is connected to it.

e. An outdoor unit with no match (*i.e.,* that is not sold with indoor units) shall be tested without an indoor blower installed, with a single cooling air volume rate, using an indoor unit 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 1.15 square inches per British thermal unit per hour (sq. in./Btu/hr). NGIFS is calculated as follows:

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

where,

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

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 for information on region IV.) For heat pumps that use a time-adaptive defrost control system (see section 1.2, Definitions), the manufacturer must specify the frosting interval to be used during Frost Accumulation tests and provide the procedure for manually initiating the defrost at the specified time. To ease testing of any unit, the manufacturer should provide information and any necessary hardware to manually initiate a defrost cycle.

2.2.2 Special requirements for units having a multiple-speed outdoor fan.

Configure the multiple-speed outdoor fan according to the 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, systems composed of multiple single-zone-multiple-coil split-system units (having multiple outdoor units located side-by-side), and ducted systems using a single indoor section containing multiple 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 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 and systems composed of multiple single-zone-multiple-coil split-system units. For any test where the system is operated at part load (*i.e.,* one or more compressors ''off'', operating at the intermediate or minimum compressor speed, or at low compressor capacity), the manufacturer shall designate the indoor coil(s) that are not providing heating or cooling during the test such that the sum of the nominal heating or cooling capacity of the operational indoor units is within 5 percent of the intended part load heating or cooling capacity. For variable-speed systems, the manufacturer must designate 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 shall choose to turn off zero, one, two, or more indoor units. The chosen configuration shall remain unchanged for all tests conducted at the same compressor speed/capacity. For any indoor coil that is 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 systems with a single indoor section containing multiple blowers where the blowers are designed to cycle on and off independently of one another and are not controlled such that all blowers are modulated to always operate at the same air volume rate or speed. This Appendix covers systems with a single-speed compressor or systems offering two fixed stages of compressor capacity (*e.g.,* a two-speed compressor, two single-speed compressors). 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—blowers accounting for at least one-third of the full-load air volume rate must be turned off unless prevented by the controls of the unit. In such cases, turn off as many blowers as permitted by the unit's controls. Where more than one option exists for meeting this ''off'' blower requirement, the manufacturer shall include in its installation manuals included with the unit which 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 an ''off'' blower.

c. For test setups where it is physically impossible for the laboratory to use the required line length listed in Table 3 of ANSI/AHRI Standard 1230–2010 (incorporated by reference, see § 430.3) with Addendum 2, 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 ANSI/AHRI Standard 1230–2010 with Addendum 2 are applied.

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

2.2.4.1 Cooling mode tests. For wet-coil cooling mode tests, regulate the water vapor content of the air entering the indoor unit to the applicable wet-bulb temperature listed in Tables 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. For dry coil tests on such units, it may be necessary to limit the moisture content of the air entering the outdoor side of the unit to meet the requirements of section 3.4.

2.2.4.2 Heating mode tests.

For heating mode tests, regulate the water vapor content of the air entering the outdoor unit to the applicable wet-bulb temperature listed in Tables 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 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 The ''manufacturer's published instructions,'' as stated in section 8.2 of ASHRAE Standard 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 are shipped with the unit shall take precedence over installation instructions that appear in the labels applied to the unit.

2.2.5.2 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 the refrigerant charge shall be adjusted per the outdoor installation instructions.

c. For systems consisting of an outdoor unit manufacturer's outdoor section and an independent coil manufacturer's indoor section with differing charging procedures the refrigerant charge shall be adjusted per the indoor installation instructions.

2.2.5.3 Test(s) to Use for Charging.

a. Use the tests or operating conditions specified in the manufacturer's installation instructions for charging.

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 function in the H1 or $H1_2$ test with the charge set for the A or A_2 test and for heating-only heat pumps, use the H1 or $H1_2$ test.

2.2.5.4 Parameters to Set and Their Target Values.

a. Consult the manufacturer's installation instructions regarding which parameters 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 (defined as 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
- (iii) High side temperature
- (iv) 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.

2.2.5.5 Charging Tolerances.

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

b. Otherwise, use the following tolerances for the different charging parameters:

1. Superheat: +/−2.0 °F
2. Subcooling: +/−0.6 °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

2.2.5.6 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 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 this test.

b. Single-Package Systems

Unless otherwise directed by the manufacturer's installation instructions, install one or more refrigerant line pressure gauges during the setup of the unit if setting of refrigerant charge is based on certain operating parameters:

(1) Install a pressure gauge 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 on the suction line if charging is on the basis of superheat, or low side pressure or corresponding saturation or dew point temperature. If manufacturer's installation instructions indicate that pressure gauges are not to be installed, setting of charge shall not be based on any of the parameters listed in b.(1) and (2) of this section.

2.2.5.7 Near-azeotropic and zeotropic refrigerants.

Charging of near-azeotropic and zeotropic refrigerants shall only be performed with refrigerant in the liquid state.

2.2.5.8 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.

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 installation instructions that are provided with the equipment while meeting the airflow requirements that are specified in section 3.1.4 of this appendix. If the manufacturer installation instructions do not provide guidance on the airflow-control settings for a system tested with the indoor blower installed, select the lowest speed that will satisfy the minimum external static pressure specified in section 3.1.4.1.1 of this appendix with an air volume rate at or higher than the rated full-load cooling air volume rate while meeting the maximum air flow requirement.

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

2.3.2 Heating tests.

a. If needed, set the indoor blower airflow-control settings (*e.g.,* fan motor pin settings, fan motor speed) according to the installation instructions that are provided with the equipment. Do this set-up while meeting all applicable airflow requirements specified in sections 3.1.4 of this appendix. For a cooling and heating heat pump tested with an indoor blower installed, if the manufacturer installation instructions do not provide guidance on the fan airflow-control settings, use the same airflow-control settings used for the cooling test. If the manufacturer installation instructions do not provide guidance on the airflow-control settings for a heating-only heat pump tested with the indoor blower installed, select the lowest speed that will satisfy the minimum external static pressure specified in section 3.1.4.4.3 of this appendix with an air volume rate at or higher than the rated heating full-load air volume rate.

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 described in section 2.4.1 of this appendix and, if installed, the inlet plenum described in section 2.4.2 of this appendix with thermal insulation having a nominal overall resistance (R-value) of at least 19 hr·ft²·°F/Btu.

2.4.1 Outlet plenum for the indoor unit.

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

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

c. For small-duct, high-velocity systems, install an outlet plenum that has a diameter that is equal to or less than the value listed below. The limit depends only on the Cooling Full-Load Air Volume Rate (see section 3.1.4.1.1 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. Figures 7a, 7b, 7c of ASHRAE Standard 37–2009 (incorporated by reference, see § 430.3) shows two of the three options allowed for the manifold configuration; the third option is the broken-ring, four-to-one manifold configuration that is shown in Figure 7a of ASHRAE Standard 37–2009. See Figures 7a, 7b, 7c, and 8 of ASHRAE Standard 37–2009 for the cross-sectional dimensions and minimum length of the (each) plenum and the locations for adding the static pressure taps for units tested with and without an indoor blower installed.

TABLE 2—SIZE OF OUTLET PLENUM

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 or a packaged system where the indoor coil is located in the outdoor test room. Add static pressure taps at the center of each face of this plenum, if rectangular, or at four evenly distributed locations along the circumference of an oval or round plenum. Make a manifold that connects the four static-pressure taps using one of the three configurations specified in section 2.4.1. See Figures 7b, 7c, and Figure 8 of ASHRAE Standard 37–2009 (incorporated by reference, see § 430.3) for cross-sectional dimensions, the minimum length of the inlet plenum, and the locations of the static-pressure taps. When testing a ducted unit having an indoor blower (and the indoor coil is in the indoor test room), test with an inlet plenum installed unless physically prohibited by space limitations within the test room. If used, construct the inlet plenum and add the four static-pressure taps as shown in Figure 8 of ASHRAE Standard 37–2009. If used, the inlet duct size shall equal the size of the inlet opening of the air-handling (blower coil) unit or furnace, with a minimum length of 6 inches. Manifold the four static-pressure taps using one of the three configurations specified in section 2.4.1.d. Never use an inlet plenum when testing a non-ducted system.

2.5 Indoor coil air property measurements and air damper box applications.

Follow instructions for indoor coil air property measurements as described in AHRI 210/240-Draft, appendix E, section E4, 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 ASHRAE Standard 41.1–2013 (incorporated by reference, see § 430.3) for guidance on constructing an air sampling device. No part of the air sampling device or the tubing transferring the sampled air to the sensor shall be within two inches of the test chamber floor, and the transfer tubing shall be insulated. The sampling device may also divert air to a remotely located sensor(s) that measures dry bulb temperature. The air sampling device and the remotely located temperature sensor(s) may be used to determine the entering air dry bulb temperature during any test. The air sampling device and the remotely located leaving air dry bulb temperature sensor(s) may be used for all tests except:

(1) Cyclic tests; and

(2) Frost accumulation tests.

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

c. Use an inlet and outlet air damper box, an inlet upturned duct, or any combination thereof when conducting one or both of the cyclic tests listed in sections 3.2 and 3.6 on ducted systems. Otherwise if not conducting one or both of said cyclic tests, install an outlet air damper box when testing ducted and non-ducted heat pumps that cycle off the indoor blower during defrost cycles if no other means is available 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 a non-ducted system. An inlet upturned duct is a length of ductwork so 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 the variation of the dry bulb temperature at this location, measured at least every minute during the compressor OFF period of the cyclic test, does not exceed 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; install a dry-bulb temperature sensor at a centerline location not higher than the lowest elevation of the duct edges at the device inlet.

2.5.1.1 If the section 2.4.2 inlet plenum is installed.

Construct the airflow prevention device having a cross-sectional flow area equal to or greater than the flow area of the inlet plenum. Install the airflow prevention device upstream of the inlet plenum and construct ductwork connecting it to the inlet plenum. If needed, use an adaptor plate or a transition duct section to connect the airflow prevention device with the inlet plenum. Insulate the ductwork and inlet plenum with thermal insulation that has a nominal overall resistance (R-value) of at least 19 hr · ft^2 · °F/Btu.

2.5.1.2 If the section 2.4.2 inlet plenum is not installed.

Construct the airflow prevention device having a cross-sectional flow area equal to or greater than the flow area of the air inlet of the indoor unit. Install the airflow prevention device immediately upstream of the inlet of the indoor unit. If needed, use an adaptor plate or a short transition duct section to connect the airflow prevention device with the unit's air inlet. Add static pressure taps at the center of each face of a rectangular airflow prevention device, or at four evenly distributed locations along the circumference of an oval or round airflow prevention device. Locate the pressure taps between the airflow prevention device and the inlet of the indoor unit. Make a manifold that connects the four static pressure taps. Insulate the ductwork with thermal insulation that has a nominal overall resistance (R-value) of at least 19 hr · ft^2 · °F/Btu.

2.5.2 Test set-up on the inlet side of the indoor unit: For cases where no airflow prevention device is installed.

If using the section 2.4.2 inlet plenum and a grid of dry bulb temperature sensors, mount the grid at a location upstream of the static pressure taps described in section 2.4.2, preferably at the entrance plane of the inlet plenum. If the section 2.4.2 inlet plenum is not used, but a grid of dry bulb temperature sensors is used, locate the grid approximately 6 inches upstream from the inlet of the indoor coil. Or, in the case of non-ducted units having multiple indoor coils, locate a grid approximately 6 inches upstream from the inlet of each indoor coil. Position an air sampling device, or the sensor used to measure the water vapor content of the inlet air, immediately upstream of the (each) entering air dry-bulb temperature sensor grid. If a grid of sensors is not used, position the entering air sampling device (or the sensor used to measure the water vapor content of the inlet air) as if the grid were present.

2.5.3 Indoor coil static pressure difference measurement.

Section 6.5.2 of ASHRAE Standard 37–2009 (incorporated by reference, see § 430.3) describes the method for fabricating static-pressure taps. Also refer to Figure 2A of ASHRAE Standard 51–07/AMCA Standard 210–07 (incorporated by reference, see § 430.3). 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. If an inlet plenum or inlet airflow prevention device is not used, leave the inlet side of the differential pressure instrument open to the surrounding atmosphere. For non-ducted systems that are tested with multiple outlet plenums, measure the static pressure within each outlet plenum relative to the surrounding atmosphere.

2.5.4 Test set-up on the outlet side of the indoor coil.

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

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

(1) Downstream of the air sampling device;

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

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

2.5.4.1 Outlet air damper box placement and requirements.

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

2.5.4.2 Procedures to minimize temperature maldistribution.

Use these procedures if necessary to correct temperature maldistributions. Install a mixing device(s) upstream of the outlet air, dry-bulb temperature grid (but downstream of the outlet plenum static pressure taps). Use a perforated screen located between the mixing device and the dry-bulb temperature grid, with a maximum open area of 40 percent. One or both items should help to meet the maximum outlet air temperature distribution specified in section 3.1.8. Mixing devices are described in sections 5.3.2 and 5.3.3 of ASHRAE Standard 41.1–2013 and section 5.2.2 of ASHRAE Standard 41.2–87 (RA 92) (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. 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, 7.2, and 7.3 of ASHRAE Standard 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, 7.4, and 7.5 of ASHRAE Standard 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, and 7.1 of ASHRAE Standard 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), the air damper box(es) must be capable of being completely opened or completely closed within 10 seconds for each action.

2.6 Airflow measuring apparatus.

a. Fabricate and operate an Air Flow Measuring Apparatus as specified in section 6.2 and 6.3 of ASHRAE Standard 37–2009 (incorporated by reference, see § 430.3). Refer to Figure 12 of ASHRAE Standard 51–07/AMCA Standard 210–07 or Figure 14 of ASHRAE Standard 41.2–87 (RA 92) (incorporated by reference, see § 430.3) for guidance on placing the static pressure taps and positioning the diffusion baffle (settling means) relative to the chamber inlet. When measuring the static pressure difference across nozzles and/or velocity pressure at nozzle throats using electronic pressure transducers and a data acquisition system, if high frequency fluctuations cause measurement variations to exceed the test tolerance limits specified in section 9.2 and Table 2 of ASHRAE Standard 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. See sections 6.1.1, 6.1.2, and 6.1.4, and Figures 1, 2, and 4 of ASHRAE Standard 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 ASHRAE Standard 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) with Addendum 1 and 2 for ''Standard Rating Tests.'' If the voltage on the nameplate of indoor and outdoor units differs, the voltage supply on the outdoor unit shall be selected for testing. Measure the supply voltage at the terminals on the test unit using a volt meter that provides a reading that is accurate to within ±1.0 percent of the measured quantity.

2.8 Electrical power and energy measurements.

a. Use an integrating power (watt-hour) measuring system to determine the electrical energy or average electrical power supplied to all components of the air conditioner or heat pump (including auxiliary components such as controls, transformers, crankcase heater, integral condensate pump on non-ducted indoor units, etc.). The watt-hour measuring system must give readings that are accurate to within ±0.5 percent. For cyclic tests, this accuracy is required during both the ON and OFF cycles. Use either two different scales on the same watt-hour meter or two separate watt-hour meters. Activate the scale or meter having the lower power rating within 15 seconds after beginning an OFF cycle. Activate the scale or meter having the higher power rating active within 15 seconds prior to beginning an ON cycle. For ducted units tested with a fan installed, the ON cycle lasts from compressor ON to indoor blower OFF. For ducted units tested without an indoor blower installed, 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 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. 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),

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

(3) A duct section that connects these two components and itself contains the

instrumentation for measuring the dry-bulb temperature and water vapor content of the air leaving the outdoor coil (sections 2.5.4, 2.5.5, and 2.5.6), and

(4) On the inlet side, a sampling device and temperature grid (section 2.11b.).

c. During the preliminary tests described in sections 3.11.1 and 3.11.1.1, measure the evaporator and condenser temperatures or pressures. On both the outdoor coil and the indoor coil, solder a thermocouple onto a return bend located at or near the midpoint of each coil or at points not affected by vapor superheat or liquid subcooling. Alternatively, if the test unit is not sensitive to the refrigerant charge, 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 Standard 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, measure refrigerant properties, and adjust the refrigerant charge according to section 7.4.2 and 8.2.5 of ASHRAE Standard 37–2009 (incorporated by reference, see § 430.3). Use refrigerant temperature and pressure measuring instruments that meet the specifications given in sections 5.1.1 and 5.2 of ASHRAE Standard 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 ASHRAE Standard 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 ASHRAE Standard 37–2009. Refrigerant flow measurement device(s), if used, must be 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, unless the device(s) are elevated at least two feet from the floor.

2.11 Measurement of test room ambient conditions.

Follow instructions for measurement of test room ambient conditions as described in AHRI 210/240-Draft, appendix E, section E4, 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 ASHRAE Standard 37–2009), add instrumentation to permit measurement of the indoor test room dry-bulb temperature.

b. For the outdoor side, install a grid of evenly-distributed sensors on every air-permitting face on the inlet of the outdoor unit, such that each measurement represents an air-inlet area of no more than one square foot. This grid must be constructed and applied as per section 5.3 of ASHRAE Standard 41.1–2013 (incorporated by reference, see § 430.3). The maximum and minimum temperatures measured by these sensors may differ by no more than 1.5 °F—otherwise adjustments to the test room must be made to improve temperature uniformity. The outdoor conditions shall be verified with the air collected by air sampling device. Air collected by an air sampling device at the air inlet of the outdoor unit for transfer to sensors for measurement of temperature and/or humidity shall be protected from temperature change as follows: Any surface of the air conveying tubing in contact with surrounding air at a different temperature than the sampled air shall be insulated with thermal insulation with a nominal thermal resistance (R-value) of at least 19 hr $\cdot$ft^2 $\cdot$ °F/Btu, no part of the air sampling device or the tubing conducting the sampled air to the sensors shall be within two inches of the test chamber floor, and pairs of measurements (*e.g.* dry bulb temperature and wet bulb temperature) used to determine water vapor content of sampled air shall be measured in the same location. Take steps (*e.g.,* add or re-position a lab circulating fan), as needed, to maximize temperature uniformity within the outdoor test room. However, ensure that any fan used for this purpose does not cause air velocities in the vicinity of the test unit to exceed 500 feet per minute.

c. Measure dry bulb temperatures as specified in sections 4, 5, 7.2, 6, and 7.3 of ASHRAE Standard 41.1–2013. Measure water vapor content as stated in section 2.5.6.

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 ASHRAE Standard 37–2009 (incorporated by reference, see § 430.3).

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 to determine indoor space conditioning capacity. Calculate this secondary check of capacity according to section 3.11. The two capacity measurements must agree to within 6 percent to constitute a valid test. For this capacity comparison, use the Indoor Air Enthalpy Method capacity that is calculated in section 7.3 of ASHRAE Standard 37–2009 (and, if testing a coil-only system, do not make the after-test fan heat adjustments described in section 3.3, 3.4, 3.7, and 3.10 of this appendix). However, include the appropriate section 3.3 to 3.5 and 3.7 to 3.10 fan heat adjustments within the Indoor Air Enthalpy Method capacities used for the section 4 seasonal calculations.

3.1.2 Manufacturer-provided equipment overrides.

Where needed, the manufacturer must provide a means for overriding the controls of the test unit so that the compressor(s) operates at the specified speed or capacity and the indoor 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) with Addendum 1 and 2 when obtaining the airflow through the outdoor coil.

3.1.3.1 Double-ducted. For products intended to be installed with the outdoor airflow ducted, the unit shall be installed with outdoor coil ductwork installed per manufacturer installation instructions and shall operate between 0.10 and 0.15 in H_2O external static pressure. External static pressure measurements shall be made in accordance with ASHRAE Standard 37–2009 Section 6.4 and 6.5.

3.1.4 Airflow through the indoor coil.

Airflow setting(s) shall be determined before testing begins. Unless otherwise specified within this or its subsections, no changes shall be made to the airflow setting(s) after initiation of testing.

3.1.4.1 Cooling Full-load Air Volume Rate.

3.1.4.1.1 Cooling Full-Load Air Volume Rate for Ducted Units.

The manufacturer must specify the cooling full-load air volume rate and the instructions for setting fan speed or controls. Adjust the cooling full-load air volume rate if needed to satisfy the additional requirements of this section. First, when conducting the A or A_2 Test (exclusively), the measured air volume rate, when divided by the measured indoor air-side total cooling capacity must not exceed 37.5 cubic feet per minute of standard air (scfm) per 1000 Btu/h. If this ratio is exceeded, reduce the air volume rate until this ratio is equaled. Use this reduced air volume rate for all tests that call for using the Cooling Full-load Air Volume Rate. Pressure requirements are as follows:

a. For all ducted units tested with an indoor blower installed, except those having a constant-air-volume-rate indoor blower:

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

2. Measure the external static pressure;

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

4. If the Table 3 minimum is not equaled or exceeded,

4a. reduce the air volume rate and increase the external static pressure by adjusting the exhaust fan of the airflow measuring apparatus until the applicable Table 3 minimum is equaled or

4b. until the measured air volume rate equals 90 percent of the air volume rate from step 1, whichever occurs first.

5. If the conditions of step 4a occur first, the pressure requirement is satisfied. Use the step 4a reduced air volume rate for all tests that require the Cooling Full-load Air Volume Rate.

6. If the conditions of step 4b occur first, make an incremental change to the set-up of the indoor blower (*e.g.,* next highest fan motor pin setting, next highest fan motor speed) and repeat the evaluation process beginning at above step 1. If the indoor blower set-up cannot be further changed, reduce the air volume rate and increase the external static pressure by adjusting the exhaust fan of the airflow measuring apparatus until the applicable Table 3 minimum is equaled. Use this reduced air volume rate for all tests that require the Cooling Full-load Air Volume Rate.

b. For ducted units that are tested with a constant-air-volume-rate indoor blower installed. For all tests that specify the Cooling Full-load Air Volume Rate, obtain an external static pressure as close to (but not less than) the applicable Table 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 ducted units that are tested without an indoor fan installed. For the A or A_2 Test, (exclusively), the pressure drop across the indoor coil assembly must not exceed 0.30 inches of water. If this pressure drop is exceeded, reduce the air volume rate until the measured pressure drop equals the specified maximum. Use this reduced air volume rate for all tests that require the Cooling Full-load Air Volume Rate.

TABLE 3—MINIMUM EXTERNAL STATIC PRESSURE FOR DUCTED SYSTEMS TESTED WITH AN INDOOR BLOWER INSTALLED

Rated cooling [1] or heating [2] capacity (Btu/h)	Minimum external resistance [3] (inches of water)		
	Short ducted systems [4]	Small-duct, high-velocity systems [4 5]	All other systems
Up Thru 28,800	0.03	1.10	0.10
29,000 to 42,500	0.05	1.15	0.15
43,000 and Above	0.07	1.20	0.20

[1] For air conditioners and heat pumps, the value cited by the manufacturer in published literature for the unit's capacity when operated at the *A* or A_2 Test conditions.
[2] For heating-only heat pumps, the value the manufacturer cites in published literature for the unit's capacity when operated at the *H1* or $H1_2$ Test conditions.
[3] For ducted units tested without an air filter installed, increase the applicable tabular value by 0.08 inches of water.
[4] See section 1.2, Definitions, to determine if the equipment qualifies as a short-ducted or a small-duct, high-velocity system.
[5] If a closed-loop, air-enthalpy test apparatus is used on the indoor side, limit the resistance to airflow on the inlet side of the indoor blower coil to a maximum value of 0.1 inch of water. Impose the balance of the airflow resistance on the outlet side of the indoor blower.

d. For ducted systems having multiple indoor blowers within a single indoor section, obtain the full-load air volume rate with all blowers operating unless prevented by the controls of the unit. In such cases, turn on the maximum number of blowers permitted by the unit's controls. Where more than one option exists for meeting this ''on'' blower requirement, which blower(s) are turned on must match that specified by the manufacturer in the installation manuals included with the unit. Conduct section 3.1.4.1.1 setup steps for each blower separately. If two or more indoor blowers are connected to a common duct as per section 2.4.1, either turn off the other indoor blowers connected to the same common duct or temporarily divert their air volume to the test room when confirming or adjusting the setup configuration of individual blowers. If the indoor blowers are all the same size or model, the target air volume rate for each blower plenum equals the full-load air volume rate divided by the number of ''on'' blowers. If different size blowers are used within the indoor section, the allocation of the system's full-load air volume rate assigned to each ''on'' blower must match that specified by the manufacturer in the installation manuals included with the unit.

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.

The manufacturer must specify the cooling minimum air volume rate and the instructions for setting fan speed or controls. 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.

a. For ducted units tested with an indoor blower installed that is not a constant-air-volume indoor blower, adjust for external static pressure as follows.

1. Achieve the manufacturer-specified cooling minimum air volume rate;

2. Measure the external static pressure;

3. If this pressure is equal to or greater than the target minimum external static pressure calculated as described above, use the

current air volume rate for all tests that require the cooling minimum air volume rate.

4. If the target minimum is not equaled or exceeded,

4a. reduce the air volume rate and increase the external static pressure by adjusting the exhaust fan of the airflow measuring apparatus until the applicable target minimum is equaled or

4b. until the measured air volume rate equals 90 percent of the air volume rate from step 1, whichever occurs first.

5. If the conditions of step 4a occur first, use the step 4a reduced air volume rate for all tests that require the cooling minimum air volume rate.

6. If the conditions of step 4b occur first, make an incremental change to the set-up of the indoor fan (*e.g.,* next highest fan motor pin setting, next highest fan motor speed) and repeat the evaluation process beginning at above step 1. If the indoor fan set-up cannot be further changed, reduce the air volume rate and increase the external static pressure by adjusting the exhaust fan of the airflow measuring apparatus until the applicable target minimum is equaled. Use this reduced air volume rate for all tests that require 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, 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 units that are tested without an indoor blower installed, 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) unit, obtain this Cooling Minimum Air Volume Rate regardless of the pressure drop across the indoor coil assembly.

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

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 for the minimum number of blowers that must be turned off. Adjust for external static pressure and if necessary adjust air volume rates as described in section 3.1.4.2.a if the indoor fan is not a constant-air-volume indoor fan or as described in section 3.1.4.2.b if the indoor fan is a constant-air-volume indoor fan. The sum of the individual ''on'' blowers' air volume rates is the cooling minimum air volume rate for the system.

3.1.4.3 Cooling Intermediate Air Volume Rate.

The manufacturer must specify the cooling intermediate 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.

a. For ducted units tested with an indoor blower, installed that is not a constant-air-volume indoor blower, adjust for external static pressure as described in section 3.1.4.2.a for cooling minimum air volume rate.

b. For ducted units tested with constant-air-volume indoor blowers installed, 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, 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 heat pumps tested with an indoor blower installed that is not a constant-air-volume indoor blower that operates at the same airflow-control setting during both the A (or A_2) and the H1 (or $H1_2$) Tests;

2. Ducted heat pumps tested with constant-air-flow indoor blowers installed that provide the same air flow for the A (or A_2) and the H1 (or $H1_2$) Tests; and

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

b. For heat pumps that meet the above criteria ''1'' and ''3,'' no minimum requirements apply to the measured external or internal, respectively, static pressure. For heat pumps that meet the above criterion ''2,'' test at an external static pressure that does not cause 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 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 indoor blower operation.

The manufacturer must specify the heating full-load 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.

a. For ducted heat pumps tested with an indoor blower installed that is not a constant-air-volume indoor blower, adjust for external static pressure as described in section 3.1.4.2.a 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, 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 northern heat pumps (see section 1.2, Definitions), use the appropriate approach of the above two cases for units that are tested with an indoor blower installed. For coil-only 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'' blowers as used for the cooling full-load air volume rate. For systems where individual blowers regulate the speed (as opposed to the cfm) of the indoor blower, use the first section 3.1.4.2 equation for each blower individually. Sum the individual blower air volume rates to obtain the heating full-load air volume rate for the system.

3.1.4.4.3 Ducted heating-only heat pumps.

The manufacturer must specify the Heating Full-load Air Volume Rate.

a. For all ducted heating-only heat pumps tested with an indoor blower installed, 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.

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

2. Measure the external static pressure.

3. If this pressure is equal to or greater than the Table 3 minimum external static pressure that applies given the heating-only heat pump's rated heating capacity, use the

current air volume rate for all tests that require the Heating Full-load Air Volume Rate.

4. If the Table 3 minimum is not equaled or exceeded,

4a. reduce the air volume rate and increase the external static pressure by adjusting the exhaust fan of the airflow measuring apparatus until the applicable Table 3 minimum is equaled or

4b. until the measured air volume rate equals 90 percent of the manufacturer-specified Full-load Air Volume Rate, whichever occurs first.

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

6. If the conditions of step 4b occur first, make an incremental change to the set-up of the indoor blower (*e.g.,* next highest fan motor pin setting, next highest fan motor speed) and repeat the evaluation process beginning at above step 1. If the indoor blower set-up cannot be further changed, reduce the air volume rate until the applicable Table 3 minimum is equaled. Use this reduced air volume rate for all tests that require the Heating Full-load Air Volume Rate.

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

3.1.4.4.4 Non-ducted heat pumps, including non-ducted heating-only heat pumps.

For non-ducted heat pumps, the Heating Full-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 heat pumps tested with an indoor blower installed that is not a constant-air-volume indoor blower that operates at the same airflow-control setting during both the A_1 and the $H1_1$ tests;

2. Ducted heat pumps tested with constant-air-flow indoor blowers installed that provide the same air flow for the A_1 and the $H1_1$ Tests; and

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

b. For heat pumps that meet the above criteria ''1'' and ''3,'' no minimum requirements apply to the measured external or internal, respectively, static pressure. For heat pumps that meet the above criterion ''2,'' test at an external static pressure that does not cause 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.

The manufacturer must specify the heating minimum 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.

a. For ducted heat pumps tested with an indoor blower installed that is not a constant-air-volume indoor blower, adjust for external static pressure as described in section 3.1.4.2.a 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 thanor 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 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 northern heat pumps that are tested with an indoor blower installed, use the appropriate approach of the above two cases.

d. For ducted two-capacity heat pumps that are tested without an indoor blower installed, use the Cooling Minimum Air Volume Rate as the Heating Minimum Air Volume Rate. For ducted two-capacity northern heat pumps that are tested without an indoor blower installed, use the Cooling Full-load Air Volume Rate as the Heating Minimum Air Volume Rate. For ducted two-capacity heating-only heat pumps that are tested without an indoor blower installed, 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'' blowers as used for the cooling minimum air volume rate. For systems where individual blowers regulate the speed (as opposed to the cfm) of the indoor blower, use the first section 3.1.4.5 equation for each blower individually. Sum the individual blower air volume rates to obtain the heating minimum air volume rate for the system.

3.1.4.6 Heating Intermediate Air Volume Rate.

The manufacturer must specify the heating intermediate 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.

a. For ducted heat pumps tested with an indoor blower installed that is not a constant-air-volume indoor blower, adjust for external static pressure as described in section 3.1.4.2.a 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, 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. Make adjustments as described in section 3.14.6 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 ASHRAE Standard 37–2009), maintain the dry bulb temperature within the test room within ±5.0 °F of the applicable sections 3.2 and 3.6 dry bulb temperature test condition for the air entering the indoor unit. Dew point shall be within 2 °F of the required inlet conditions.

3.1.6 Air volume rate calculations.

For all steady-state tests and for Frost Accumulation ($H2$, $H2_1$, $H2_2$, $H2_V$) tests, calculate the air volume rate through the indoor coil as specified in sections 7.7.2.1 and 7.7.2.2 of ASHRAE Standard 37–2009 (incorporated by reference, see § 430.3). When using the Outdoor Air Enthalpy Method, follow sections 7.7.2.1 and 7.7.2.2 to calculate the air volume rate through the outdoor coil. To express air volume rates in terms of standard air, use:

$$\text{Equation 3-1} \qquad \overline{\dot{V}_s} = \frac{\overline{\dot{V}_{mx}}}{0.075\frac{lbm_{da}}{ft^3} * v_n' * [1+W_n]} = \frac{\overline{\dot{V}_{mx}}}{0.075\frac{lbm_{da}}{ft^3} * v_n}$$

Where,

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

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

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

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

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

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

(**Note:** In the first printing of ASHRAE Standard 37–2009, the second IP equation for Q_{mi} should read,

$$Q_{mi} = 1097CA_n\sqrt{P_V v_n'}$$

3.1.7 Test sequence.

Manufacturers may optionally operate the equipment under test for a ''break-in'' period, not to exceed 20 hours, prior to conducting the test method specified in this section. A manufacturer who elects to use this optional compressor break-in period in its certification testing should record this information (including the duration) in the test data underlying the certified ratings that are required to be maintained under 10 CFR 429.71. 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 an 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. For the 30-minute data collection interval used to determine capacity, the maximum spread among the outlet dry bulb temperatures from any data sampling must not exceed 1.5 °F. Install the mixing devices described in section 2.5.4.2 to minimize the temperature spread.

3.1.9 Requirement for the air temperature distribution entering the outdoor coil.

Monitor the temperatures of the air entering the outdoor coil using the grid of temperature sensors described in section 2.11. For the 30-minute data collection interval used to determine capacity, the maximum difference between dry bulb temperatures measured at any of these locations must not exceed 1.5 °F.

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, the short test follows the H1 or, if conducted, the H1C Test. For two-capacity heat pumps and heat pumps covered under section 3.6.2, the short test follows the $H1_2$ Test. Set the heat comfort controller to provide the maximum supply air temperature. With the heat pump operating and while maintaining the Heating Full-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 unit having a single-speed compressor, or a multi-circuit system, that is tested with a fixed-speed indoor blower installed, with a constant-air-volume-rate indoor blower installed, or with no indoor blower installed.

Conduct two steady-state wet coil tests, the A and B Tests. Use the two 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 testing outdoor units of central air conditioners or heat pumps that are not sold with indoor units, assign C_D^c the default value of 0.2. 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-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)		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—required (steady, dry coil)	80	([3])	82		Cooling full-load.[2]
D Test—required (cyclic, dry coil)	80	([3])	82		([4])

[1] The specified test condition only applies if the unit rejects condensate to the outdoor coil.

[2] Defined in section 3.1.4.1.

[3] The entering air must have a low enough moisture content so no condensate forms on the indoor coil. (It is recommended that an indoor wet-bulb temperature of 57 °F or less be used.)

[4] Maintain the airflow nozzles static pressure difference or velocity pressure during the ON period at the same pressure difference or velocity pressure as measured during the C Test.

3.2.2 Tests for a unit having a single-speed compressor where the indoor section uses a single variable-speed variable-air-volume rate indoor blower or multiple 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 blowers.

Conduct four steady-state wet coil tests: The A_2, A_1, B_2, and B_1 Tests. Use the two dry-coil tests, the steady-state C_1 Test and the d D_1 Test, to determine the cooling mode cyclic degradation coefficient, C_D^c.

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 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]—required (steady, dry coil)	80	(4)	82		Cooling minimum.[3]
D_1 Test [4]—required (cyclic, dry coil)	80	(4)	82	(5)	

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

3.2.3 Tests for a unit having a two-capacity compressor. (see section 1.2, Definitions)

a. Conduct four steady-state wet coil tests: The A_2, B_2, B_1, and F_1 Tests. Use the two 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. 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, 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 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, $C_D^c(k=2)$. 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—required (steady, dry-coil).	80	(4)	82	High	Cooling Full-Load[2].	
D_2 Test—required (cyclic, dry-coil)	80	(4)	82	High	(5)	
C_1 Test—required (steady, dry-coil).	80	(4)	82	Low	Cooling Minimum[3].	
D_1 Test—required (cyclic, dry-coil)	80	(4)	82	Low	(6)	
F_1 Test—required (steady, wet coil).	80	67	67	[1]53.5	Low	Cooling Minimum.[3]

[1] The specified test condition only applies if the unit rejects condensate to the outdoor coil.
[2] Defined in section 3.1.4.1.
[3] Defined in section 3.1.4.2.
[4] The entering air must have a low enough moisture content so no condensate forms on the indoor coil. DOE recommends using an indoor air wet-bulb temperature of 57 °F or less.

[5] Maintain the airflow nozzle(s) static pressure difference or velocity pressure during the ON period at the same pressure or velocity as measured during the C_2 Test.
[6] Maintain the airflow nozzle(s) static pressure difference or velocity pressure during the ON period at the same pressure or velocity as measured during the C_1 Test.

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

a. Conduct five steady-state wet coil tests: The A_2, E_V, B_2, B_1, and F_1 Tests. Use the two 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. Table 7 specifies test conditions for these seven tests. Determine the intermediate compressor speed cited in Table 7 using:

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

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

b. For units that modulate the indoor 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, at least one indoor unit must be turned off. The manufacturer shall designate the particular indoor unit(s) that is turned off. The manufacturer must also specify the compressor speed used for the Table 7 E_V Test, a cooling-mode intermediate compressor speed that falls within ¼ and ¾ of the difference between the maximum and minimum cooling-mode speeds. The manufacturer should prescribe an intermediate speed that is expected to yield the highest EER for the given E_V Test conditions and bracketed compressor speed range. The manufacturer can designate that one or more indoor units are turned off for the E_V Test.

TABLE 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	Maximum	Cooling Full-Load.[2]
B_2 Test—required (steady, wet coil).	80	67	82	[1] 65	Maximum	Cooling Full-Load.[2]
E_V Test—required (steady, wet coil).	80	67	87	[1] 69	Intermediate	Cooling Intermediate.[3]
B_1 Test—required (steady, wet coil).	80	67	82	[1] 65	Minimum	Cooling Minimum.[4]
F_1 Test—required (steady, wet coil).	80	67	67	[1] 53.5	Minimum	Cooling Minimum.[4]
G_1 Test[5]—required (steady, dry-coil).	80	([6])	67	Minimum	Cooling Minimum[4].	
I_1 Test[5]—required (cyclic, dry-coil)	80	([6])	67	Minimum	([6]).	

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

3.2.5 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 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 blowers and offering two stages of compressor modulation.

Conduct the cooling mode tests specified in section 3.2.3.

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, 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 cases where its control is required, the water vapor content of the air entering the outdoor coil.

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

b. After satisfying the pretest equilibrium requirements, make the measurements specified in Table 3 of ASHRAE Standard 37–2009 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.,* four consecutive 10-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 ASHRAE Standard 37–2009 (incorporated by reference, see § 430.3). 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 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 maximum speed, k=1 to denote low capacity or minimum speed, and k=v to denote the intermediate speed.

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

$$\frac{1250\ Btu/h}{1000\ scfm} * \overline{\dot{V}_s}$$

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

$$\frac{365\ W}{1000\ scfm} * \overline{\dot{V}_s}$$

where $\dot{V}_s$ 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		
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	0.12	[5] 0.02
Electrical voltage, % of rdg.	2.0	1.5
Nozzle pressure drop, % of rdg.	8.0	

[1] See section 1.2, 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 for wet coil tests. Prior to recording data during the steady-state dry coil test, operate the unit at least one hour after achieving dry coil conditions. Drain the drain pan and plug the drain opening. Thereafter, the drain pan should remain completely dry.

b. Denote the resulting total space cooling capacity and electrical power derived from the test as $\dot{Q}_{ss,dry}$ and $\dot{E}_{ss,dry}$. With regard to a section 3.3 deviation, do not adjust $\dot{Q}_{ss,dry}$ for duct losses (*i.e.,* do not apply section 7.3.3.3 of ASHRAE Standard 37–2009). In preparing for the section 3.5 cyclic tests, record the average indoor-side air volume rate, $\acute{V}$, specific heat of the air, Cp,a (expressed on dry air basis), specific volume of the air at the nozzles, v'_n, humidity ratio at the nozzles, W_n, and either pressure difference or velocity pressure for the flow nozzles. For units having a variable-speed indoor fan (that provides either a constant or variable air volume rate) that will or may be tested during the cyclic dry coil cooling mode test with the indoor fan turned off (see section

3.5), include the electrical power used by the indoor fan motor among the recorded parameters from the 30-minute test.

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).

a. After completing the steady-state dry-coil test, remove the Outdoor Air Enthalpy method test apparatus, if connected, and begin manual OFF/ON cycling of the unit's compressor. The test set-up should otherwise be identical to the set-up used during the steady-state dry coil test. When testing heat pumps, leave the reversing valve during the compressor OFF cycles in the same position as used for the compressor ON cycles, unless automatically changed by the controls of the unit. For units having a variable-speed indoor 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.

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 specify airflow requirements through the indoor coil of ducted and non-ducted systems, respectively. In all cases, use the exhaust fan of the airflow measuring apparatus (covered under section 2.6) along with the indoor 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 blower, temporarily remove the blower.

e. Conduct a minimum of six complete compressor OFF/ON cycles for a unit with a single-speed or two-speed compressor, and a minimum of five complete compressor OFF/ON cycles for a unit with a variable speed compressor. The first three cycles for a unit with a single-speed compressor or two-speed compressor and the first two cycles for a unit with a unit with a variable speed compressor are the warm-up period—the later cycles are called the active cycles. Calculate the degradation coefficient C_D for each complete active cycle if the test tolerances given in Table 9 are satisfied. If the average C_D for the first three active cycles is within 0.02 of the average C_D for the first two active cycles, use the average C_D of the three active cycles as the final result. If these averages differ by more than 0.02, continue the test to get C_D for the fourth cycle. If the average C_D of the last three cycles is lower than or no more than 0.02 greater than the average C_D of the first three cycles, use the average C_D of all four active cycles as the final result. Otherwise, continue the test with a fifth cycle. If the average C_D of the last three cycles is 0.02 higher than the average for the previous three cycles, use the default C_D, otherwise use the average C_D of all five active cycles. If the test tolerances given in Table 9 are not satisfied, use default C_D value. The default C_D value for cooling is 0.2.

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 units tested with an indoor blower installed and operating, 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	0.12	
Airflow nozzle pressure difference or velocity pressure [2], % of reading	8.0	[4] 2.0
Electrical voltage [5], % of rdg.	2.0	1.5

[1] See section 1.2, Definitions.

[2] Applies during the interval that air flows through the indoor (outdoor) coil except for the first 30 seconds after flow initiation. For units having a variable-speed indoor blower that ramps, the tolerances listed for the external resistance to airflow apply from 30 seconds after achieving full speed until ramp down begins.

[3] Shall at no time exceed a wet-bulb temperature that results in condensate forming on the indoor coil.

[4] The test condition shall be the average nozzle pressure difference or velocity pressure measured during the steady-state dry coil test.

[5] Applies during the interval when at least one of the following—the compressor, the outdoor fan, or, if applicable, the indoor blower—are operating except for the first 30 seconds after compressor start-up.

i. 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,

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

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

$T_{a1}(\tau)$ = dry bulb temperature of the air entering the indoor coil at time τ, °F.

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

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

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

3.5.1 Procedures when testing ducted systems.

The automatic controls that are normally installed with the test unit must govern the OFF/ON cycling of the air moving equipment on the indoor side (exhaust fan of the airflow measuring apparatus and, if installed, the indoor blower of the test unit). For example, for ducted units tested without an indoor blower installed but rated based on using a fan time delay relay, control the indoor coil airflow according to the rated ON and/or OFF delays provided by the relay. For ducted units having a variable-speed indoor 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 units tested without an indoor blower installed, cycle the indoor coil airflow in unison with the cycling of the compressor. If air damper boxes are used, close them on the inlet and outlet side during the OFF period. Airflow through the indoor coil should stop within 3 seconds after the automatic controls of the test unit (act to) de-energize the indoor blower. For ducted units tested without an indoor blower installed (excluding the special case where a variable-speed fan is temporarily removed), increase $e_{cyc,dry}$ by the quantity,

$$\text{Equation 3.5-2.} \quad \frac{365\, W}{1000\, scfm} * \overline{\dot{V}}_s * [\tau_2 - \tau_1]$$

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

$$\text{Equation 3.5-3.} \quad \frac{1250\, Btu/h}{1000\, scfm} * \overline{\dot{V}}_s * [\tau_2 - \tau_1]$$

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

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

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

1. Measure the electrical power consumed by the variable-speed indoor blower at a minimum of three operating conditions: At the speed/air volume rate/external static pressure that was measured during the steady-state test, at operating conditions associated with the midpoint of the ramp-up interval, and at conditions associated with the midpoint of the ramp-down interval. For these measurements, the tolerances on the airflow volume or the external static pressure are the same as required for the section 3.4 steady-state test.

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

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

3.5.2 Procedures when testing non-ducted systems.

Do not use airflow prevention devices when conducting cyclic tests on non-ducted units. Until the last OFF/ON compressor cycle, airflow through the indoor coil must cycle off and on in unison with the compressor. For the last OFF/ON compressor cycle—the one used to determine $e_{cyc,dry}$ and $q_{cyc,dry}$—use the exhaust fan of the airflow measuring apparatus and the indoor 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 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 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. The default value for two-capacity units cycling at high capacity, however, is the low-capacity coefficient, *i.e.,* $C_D^c(k=2) = C_D^c$. Evaluate C_D^c using the above results and those from the section 3.4 dry-coil steady-state test.

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

where,

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

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

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

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

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

the cooling load factor dimensionless

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

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

3.6.1 Tests for a heat pump having a single-speed compressor that is tested with a fixed speed indoor blower installed, with a constant-air-volume-rate indoor blower installed, or with no indoor blower installed. Conduct the High Temperature Cyclic (H1C) Test to determine the heating mode cyclic-degradation coefficient, C_D^h. Test conditions for the four tests are specified in Table 10.

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]
H1C Test (required, 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.

[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 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 High Temperature Cyclic ($H1C_1$) Test to determine the heating mode cyclic-degradation coefficient, C_D^h. 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:

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

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

where,

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

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

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

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

3.6.3 Tests for a heat pump having a two-capacity compressor (see section 1.2, Definitions), including two-capacity, northern heat pumps (see section 1.2, 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 seasonal performance calculations; and

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

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

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

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

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

b. Conduct the High Temperature Cyclic Test ($H1C_1$) to determine the heating mode cyclic-degradation coefficient, C_D^h. 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). 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 (required,[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 (required, 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]

TABLE 12—HEATING MODE TEST CONDITIONS FOR UNITS HAVING A TWO-CAPACITY COMPRESSOR—Continued

Test description	Air entering indoor unit temperature (°F)		Air entering outdoor unit temperature (°F)		Compressor capacity	Heating air volume rate
	Dry bulb	Wet bulb	Dry bulb	Wet bulb		
H3$_1$ Test [5] (required, steady)	70	60 (max)	17	15	Low	Heating Minimum.[1]

[1] Defined in section 3.1.4.5.
[2] Defined in section 3.1.4.4.
[3] Maintain the airflow nozzle(s) static pressure difference or velocity pressure during the ON period at the same pressure or velocity as measured during the *H1*$_2$ Test.
[4] Maintain the airflow nozzle(s) static pressure difference or velocity pressure during the ON period at the same pressure or velocity as measured during the *H1*$_1$ Test.
[5] Required only if the heat pump's performance when operating at low compressor capacity and outdoor temperatures less than 37 °F is needed to complete the section 4.2.3 *HSPF* calculations.
[6] If table note #5 applies, the section 3.6.3 equations for $\dot{Q}_h^{k=1}$ (35) and $\dot{E}_h^{k=1}$ (17) may be used in lieu of conducting the H2$_1$ Test.
[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$_2$ and H1$_1$), one Frost Accumulation Test (H2$_V$), and one Low Temperature Test (H3$_2$). Conducting one or both of the following tests is optional: An additional High Temperature Test (H1$_N$) and an additional Frost Accumulation Test (H2$_2$). Conduct the Maximum Temperature Cyclic (H0C$_1$) Test to determine the heating mode cyclic-degradation coefficient, C_D^h. Test conditions for the eight tests are specified in Table 13. Determine the intermediate compressor speed cited in Table 13 using the heating mode maximum and minimum compressors speeds and:

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

Where a tolerance of plus 5 percent or the next higher inverter frequency step from that calculated is allowed. If the H2$_2$Test is not done, use the following equations to approximate the capacity and electrical power at the H2$_2$ test conditions:

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

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

b. Determine the quantities $\dot{Q}_h^{k=2}(47)$ and from $\dot{E}_h^{k=2}(47)$ from the H1$_2$ Test and evaluate them according to section 3.7. Determine the quantities $\dot{Q}_h^{k=2}(17)$ and $\dot{E}_h^{k=2}(17)$ from the H3$_2$ Test and evaluate them according to section 3.10. For heat pumps where the heating mode maximum compressor speed exceeds its cooling mode maximum compressor speed, conduct the H1$_N$ Test if the manufacturer requests it. If the H1$_N$ Test is done, operate the heat pump's compressor at the same speed as the speed used for the cooling mode A$_2$ Test. Refer to the last sentence of section 4.2 to see how the results of the H1$_N$ Test may be used in calculating the heating seasonal performance factor.

TABLE 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	Minimum	Heating Minimum.[1]
H0C$_1$ Test (required, steady)	70	60 (max)	62	56.5	Minimum	([2]).
H1$_2$ Test (required, steady)	70	60 (max)	47	43	Maximum	Heating Full-Load.[3]
H1$_1$ Test (required, steady)	70	60 (max)	47	43	Minimum	Heating Minimum.[1]
H1$_N$ Test (optional, steady)	70	60 (max)	47	43	Cooling Mode Maximum.	Heating Nominal.[4]
H2$_2$ Test (optional)	70	60 (max)	35	33	Maximum	Heating Full-Load.[3]
H2$_V$ Test (required)	70	60 (max)	35	33	Intermediate	Heating Intermediate.[5]
H3$_2$ Test (required, steady)	70	60 (max)	17	15	Maximum	Heating Full-Load.[3]

[1] Defined in section 3.1.4.5.
[2] Maintain the airflow nozzle(s) static pressure difference or velocity pressure during an ON period at the same pressure or velocity as measured during the H0$_1$ Test.

[3] Defined in section 3.1.4.4.
[4] Defined in section 3.1.4.7.
[5] Defined in section 3.1.4.6.

c. For multiple-split heat pumps (only), the following procedures supersede the above requirements. For all Table 13 tests specified for a minimum compressor speed, at least one indoor unit must be turned off. The manufacturer shall designate the particular indoor unit(s) that is turned off. The manufacturer must also specify the compressor speed used for the Table 13 $H2_V$ Test, a heating mode intermediate compressor speed that falls within ¼ and ¾ of the difference between the maximum and minimum heating mode speeds. The manufacturer should prescribe an intermediate speed that is expected to yield the highest COP for the given $H2_V$ Test conditions and bracketed compressor speed range. The manufacturer can designate that one or more specific indoor units are turned off for the $H2_V$ Test.

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

Test any heat pump that has a heat comfort controller (see section 1.2, 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 with the heat comfort controller active to determine the system's maximum supply air temperature. (Note: Heat pumps having a variable speed compressor and a heat comfort controller are not covered in the test procedure at this time.)

3.6.6 Heating mode tests for northern heat pumps with triple-capacity compressors.

Test triple-capacity, northern heat pumps for the heating mode as follows:

(a) Conduct one maximum-temperature test ($H0_1$), two high-temperature tests ($H1_2$ and $H1_1$), one Frost Accumulation test ($H2_2$), two low-temperature tests ($H3_2$, $H3_3$), and one minimum-temperature test ($H4_3$). Conduct an additional Frost Accumulation test ($H2_1$) and low- temperature test ($H3_1$) if both of the following conditions exist: (1) Knowledge of the heat pump's capacity and electrical power at low compressor capacity for outdoor temperatures of 37 °F and less is needed to complete the section 4.2.6 seasonal performance calculations; and (2) the heat pump's controls allow low-capacity operation at outdoor temperatures of 37 °F and less. If the above two conditions are met, an alternative to conducting the $H2_1$ Frost Accumulation Test to determine $\dot{Q}_h^{k=1}(35)$ and $\dot{E}_h^{k=1}(35)$ is to use the following equations to approximate this capacity and electrical power:

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

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

In evaluating the above equations, determine the quantities $\dot{Q}_h^{k=1}(47)$ from the $H1_1$ Test and evaluate them according to section 3.7. Determine the quantities $\dot{Q}_h^{k=1}(17)$ and $\dot{E}_h^{k=1}(17)$ from the $H3_1$ Test and evaluate them according to section 3.10. 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 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 $Q_h^{k=3}(35)$ and $E_h^{k=3}(35)$ using the following equations to approximate this capacity and electrical power:

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

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

where:

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

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

Determine the quantities $\dot{Q}_h^{k=2}(47)$ and $\dot{E}_h^{k=2}(47)$ from the $H1_2$ Test and evaluate them according to section 3.7. Determine the quantities $\dot{Q}_h^{k=2}(35)$ and $\dot{E}_h^{k=2}(35)$ from the $H2_2$Test and evaluate them according to section 3.9.1. Determine the quantities $\dot{Q}_h^{k=2}(17)$ and $\dot{E}_h^{k=2}(17)$ from the $H3_2$Test, determine the quantities $\dot{Q}_h^{k=3}(17)$ and $\dot{E}_h^{k=3}(17)$ from the $H3_3$Test, and determine the quantities $\dot{Q}_h^{k=3}(2)$ and $\dot{E}_h^{k=3}(2)$ from the $H4_3$Test. Evaluate all six quantities according to section 3.10. Use the paired values of $\dot{Q}_h^{k=3}(35)$ and $\dot{E}_h^{k=3}(35)$ derived from conducting the $H2_3$Frost Accumulation Test and calculated as specified in section 3.9.1 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 high-temperature cyclic test ($H1C_1$) to determine the heating mode cyclic-degradation coefficient, C_D^h. 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 (required,[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 (required, 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] (required, cyclic)	70	60 (max)	17	15	Booster	[7].
$H3_2$ Test (required, steady)	70	60 (max)	17	15	High	Heating Full-Load.[2]
$H3_1$ Test [5] (required, steady)	70	60 (max)	17	15	Low	Heating Minimum.[1]
$H4_3$ Test (required, steady)	70	60 (max)	2	1	Booster	Heating Full-Load.[2]

[1] Defined in section 3.1.4.5.

[2] Defined in section 3.1.4.4.

[3] Maintain the airflow nozzle(s) static pressure difference or velocity pressure during the ON period at the same pressure or velocity as measured during the $H1_2$ Test.

[4] Maintain the airflow nozzle(s) static pressure difference or velocity pressure during the ON period at the same pressure or velocity as measured during the $H1_1$Test.

[5] Required only if the heat pump's performance when operating at low compressor capacity and outdoor temperatures less than 37 °F is needed to complete the section 4.2.6 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 blowers and offering two stages of compressor modulation. Conduct the heating mode tests specified in section 3.6.3.

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 for additional requirements that depend on the selected secondary test method. After satisfying the pretest equilibrium requirements, make the measurements specified in Table 3 of ASHRAE Standard 37–2009 (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.,* four consecutive 10-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:		

TABLE 15—TEST OPERATING AND TEST CONDITION TOLERANCES FOR SECTION 3.7 AND SECTION 3.10 STEADY-STATE HEATING MODE TESTS—Continued

	Test operating tolerance [1]	Test condition tolerance [1]
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	0.12	[3] 0.02
Electrical voltage, % of rdg	2.0	1.5
Nozzle pressure drop, % of rdg	8.0	

[1] See section 1.2, 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 ASHRAE Standard 37–2009 (incorporated by reference, see § 430.3). Do not adjust the parameters used in calculating capacity for the permitted variations in test conditions. Assign the average space heating capacity and electrical power over the 30-minute data collection interval to the variables $\dot{Q}_h^k$ and $\dot{E}_h^k(T)$ respectively. The "T" and superscripted "k" are the same as described in section 3.3. Additionally, for the heating mode, use the superscript to denote results from the optional $H1_N$ Test, if conducted.

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

$$\frac{1250\,BTU/h}{1000\,scfm} * \bar{\dot{V}}_s$$

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

$$\frac{365\,W}{1000\,scfm} * \bar{\dot{V}}_s$$

where $\bar{\dot{V}}_s$ is the average measured indoor air volume rate expressed in units of cubic feet per minute of standard air (scfm). During the 30-minute data collection interval of a High Temperature Test, pay attention to preventing a defrost cycle. Prior to this time, allow the heat pump to perform a defrost cycle if automatically initiated by its own controls. As in all cases, wait for the heat pump's defrost controls to automatically terminate the defrost cycle. Heat pumps that undergo a defrost should operate in the heating mode for at least 10 minutes after defrost termination prior to beginning the 30-minute data collection interval. For some heat pumps, frost may accumulate on the outdoor coil during a High Temperature test. If the indoor coil leaving air temperature or the difference between the leaving and entering air temperatures decreases by more than 1.5 °F over the 30-minute data collection interval, then do not use the collected data to determine capacity. Instead, initiate a defrost cycle. Begin collecting data no sooner than 10 minutes after defrost termination. Collect 30 minutes of new data during which the Table 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, record the average indoor-side air volume rate, $\bar{\dot{V}}$, 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. 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 for the outdoor coil entering dry-bulb temperature. Drop the subscript ''dry'' used in variables cited in section 3.5 when referring to quantities from the cyclic heating mode test., The default C_D value for heating is 0.25. If available, use electric resistance heaters (see section 2.1) 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 except for making the following changes:

(1) When evaluating Equation 3.5–1, use the values of $\bar{V}$, $C_{p,a}$,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,

(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 heat pumps tested without an indoor blower installed (excluding the special case where a variable-speed fan is temporarily removed), increase q_{cyc} by the amount calculated using Equation 3.5–3. Additionally, increase e_{cyc} by the amount calculated using Equation 3.5–2. In making these calculations, use the average indoor air volume rate ($\bar{V}_s$) 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 ASHRAE Standard 37–2009 (incorporated by reference, see § 430.3) in determining $\dot{Q}_h^k(T_{cyc})$ (or q_{cyc}). 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 $C_{Dh}$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	0.12	
Airflow nozzle pressure difference or velocity pressure,[2]% of reading	2.0	[3] 2.0
Electrical voltage,[4]% of rdg	8.0	1.5

[1] See section 1.2, Definitions.
[2] Applies during the interval that air flows through the indoor (outdoor) coil except for the first 30 seconds after flow initiation. For units having a variable-speed indoor blower that ramps, the tolerances listed for the external resistance to airflow shall apply from 30 seconds after achieving full speed until ramp down begins.
[3] The test condition shall be the average nozzle pressure difference or velocity pressure measured during the steady-state test conducted at the same test conditions.
[4] Applies during the interval that at least one of the following—the compressor, the outdoor fan, or, if applicable, the indoor blower—are operating, except for the first 30 seconds after compressor start-up.

3.9 Test procedures for Frost Accumulation heating mode tests (the H2, $H2_2$, $H2_V$, and $H2_1$ Tests).

a. Confirm that the defrost controls of the heat pump are set as specified in section 2.2.1. Operate the test room reconditioning apparatus and the heat pump for at least 30 minutes at the specified section 3.6 test conditions before starting the ''preliminary'' test period. The preliminary test period must immediately precede the ''official'' test period, which is the heating and defrost interval over which data are collected for evaluating average space heating capacity and average electrical power consumption.

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

c. The official test period begins when the preliminary test period ends, at defrost termination. The official test period ends at the termination of the next occurring automatic defrost cycle. When testing a heat pump that uses a time-adaptive defrost control system (see section 1.2, 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 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 heat pumps tested without an indoor blower installed, determine the corresponding cumulative time (in hours) of indoor coil airflow, $\Delta\tau_a$. Sample measurements used in calculating the air volume rate (refer to sections 7.7.2.1 and 7.7.2.2 of ASHRAE Standard 37–2009) at equal intervals that span 10 minutes or less. (Note: In the first printing of ASHRAE Standard 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
Outdoor entering wet-bulb temperature, °F	1.5		0.5
External resistance to airflow, inches of water	0.12		[5] 0.02
Electrical voltage, % of rdg	2.0		1.5

[1] See section 1.2, Definitions.
[2] Applies when the heat pump is in the heating mode, except for the first 10 minutes after termination of a defrost cycle.
[3] Applies during a defrost cycle and during the first 10 minutes after the termination of a defrost cycle when the heat pump is operating in the heating mode.
[4] For heat pumps that turn off the indoor blower during the defrost cycle, the noted tolerance only applies during the 10 minute interval that follows defrost termination.
[5] Only applies when testing non-ducted heat pumps.

3.9.1 Average space heating capacity and electrical power calculations.

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

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

Where,

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

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

v_n' = specific volume of the air-water vapor mixture at the nozzle, ft³/lbmmx.

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

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

$$\Gamma = \int_{\tau_1}^{\tau_2} [T_{a2}(\tau) - T_{a1}(\tau)] d\tau, \; hr * {}^\circ F$$

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

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

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

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

v_n = specific volume of the dry air portion of the mixture evaluated at the dry-bulb temperature, vapor content, and barometric pressure existing at the nozzle, ft³ per lbm of dry air.

To account for the effect of duct losses between the outlet of the indoor unit and the section 2.5.4 dry-bulb temperature grid, adjust $\dot{Q}_h^k(35)$ in accordance with section 7.3.4.3 of ASHRAE Standard 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 heat pumps tested without an indoor blower installed, increase $\dot{Q}_h^k(35)$ by,

$$\frac{1250\, Btu/h}{1000\, scfm} * \bar{\dot{V}}_s * \frac{\Delta\tau_a}{\Delta\tau_{FR}}$$

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

$$\frac{365\,W}{1000\,scfm} * \bar{\dot{V}}_s * \frac{\Delta\tau_a}{\Delta\tau_{FR}}$$

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

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

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

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

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

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

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

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

3.9.2 Demand defrost credit.

a. Assign the demand defrost credit, F_{def}, that is used in section 4.2 to the value of 1 in all cases except for heat pumps having a demand-defrost control system (see section 1.2, Definitions). For such qualifying heat pumps, evaluate F_{def} using,

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

Where,

$\Delta\tau_{def}$ = the time between defrost terminations (in hours) or 1.5, whichever is greater. A value of 6 must be assigned to $\Delta\tau_{def}$ if this limit is reached during a Frost Accumulation test and the heat pump has not completed a defrost cycle.

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

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

3.10 Test procedures for steady-state Low Temperature heating mode tests (the H3, $H3_2$, and $H3_1$ Tests).

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

3.11 Additional requirements for the secondary test methods.

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

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

3.11.1.1 If a preliminary test precedes the official test.

a. The test conditions for the preliminary test are the same as specified for the official test. Connect the indoor air-side test apparatus to the indoor coil; disconnect the outdoor air-side test apparatus. Allow the test room reconditioning apparatus and the unit being tested to operate for at least one hour. After attaining equilibrium conditions, measure the following quantities at equal intervals that span 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.,* four consecutive 10-minute samples) is obtained where the Table 8 or Table 15, whichever applies, test tolerances are satisfied.

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

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

Connect the outdoor-side test apparatus to the unit. Adjust the exhaust fan of the outdoor airflow measuring apparatus to achieve the same external static pressure as measured during the prior preliminary test conducted with the unit operating in the same cooling or heating mode at the same outdoor fan speed.

3.11.1.3 Official test.

a. Continue (preliminary test was conducted) or begin (no preliminary test) the official test by making measurements for both the Indoor and Outdoor Air Enthalpy Methods at equal intervals that span 5 minutes or less. Discontinue these measurements only after obtaining a 30-minute period where the specified test condition and test operating tolerances are satisfied. To constitute a valid official test:

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

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

b. For space cooling tests, calculate capacity from the outdoor air-enthalpy measurements as specified in sections 7.3.3.2 and 7.3.3.3 of ASHRAE Standard 37–2009 (incorporated by reference, see § 430.3). Calculate heating capacity based on outdoor air-enthalpy measurements as specified in sections 7.3.4.2 and 7.3.3.4.3 of the same ASHRAE Standard. Adjust the outdoor-side capacity according to section 7.3.3.4 of ASHRAE Standard 37–2009 to account for line losses when testing split systems. Use the outdoor unit fan power as measured during the official test and not the value measured during the preliminary test, as described in section 8.6.2 of ASHRAE Standard 37–2009, when calculating the capacity.

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

a. Conduct separate calibration tests using a calorimeter to determine the refrigerant flow rate. Or for cases where the superheat of the refrigerant leaving the evaporator is less than 5 °F, use the calorimeter to measure total capacity rather than refrigerant flow rate. Conduct these calibration tests at the same test conditions as specified for the tests in this appendix. Operate the unit for at least one hour or until obtaining equilibrium conditions before collecting data that will be used in determining the average refrigerant flow rate or total capacity. Sample the data at equal intervals that span 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 Standard 23.1–2010 (incorporated by reference, see § 430.3); sections 5, 6, 7, 8, 9, and 11 of ASHRAE Standard 41.9–2011 (incorporated by reference, see § 430.3); and section 7.4 of ASHRAE Standard 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 ASHRAE Standard 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 ASHRAE Standard 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 section 4 calculations, however, round only to the nearest integer.

3.13 Laboratory testing to determine off mode average power ratings.

Conduct one of the following tests after the completion of the B, B_1, or B_2 Test, whichever comes last: If the central air conditioner or heat pump lacks a compressor crankcase heater, perform the test in Section 3.13.1; if the central air conditioner or heat pump has compressor crankcase heater that lacks controls, perform the test in Section 3.13.1; if the central air conditioner or heat pump has a compressor crankcase heater equipped with controls, perform the test in Section 3.13.2.

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 heater that lacks controls.

a. 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. This particular test contains no requirements as to ambient conditions within the test rooms, and room conditions are allowed to change during the test. Ensure that the low-voltage transformer and low-voltage components are connected.

b. Measure $P1_x$: 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.

c. Measure P_x for coil-only split systems (that would be installed in the field with a furnace having a dedicated board for indoor controls) and for blower-coil split systems for which a furnace 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.

d. Calculate $P1$:

Single-package systems and blower coil split systems for which the designated air mover is not a furnace: 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. If the compressor is a modulating-type, assign a value of 1.5 for the number of compressors. Round $P1$ to the nearest watt and record as both $P1$ and $P2$, the latter of which is the heating season per-compressor off mode power. The expression for calculating $P1$ is as follows:

$$P1 = \frac{P1_x}{number\ of\ compressors}.$$

Coil-only split systems (that would be installed in the field with a furnace having a dedicated board for indoor controls) and blower-coil split systems for which a furnace 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. If the compressor is a modulating-type, assign a value of 1.5 for the number of compressors. Round $P1$ to the nearest watt and record as both $P1$ and $P2$, the latter of which is the heating season per-compressor off mode power. 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 that have a compressor crankcase heater equipped with controls.

a. 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 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. Ensure that the low-voltage transformer and low-voltage components are connected. Adjust the outdoor temperature at a rate of change of no more than 20 °F per hour and achieve an outdoor dry-bulb temperature of 72 °F. Maintain this temperature within ±2 °F for at least 5 minutes, while maintaining an indoor dry-bulb temperature of between 75 °F and 85 °F.

b. Measure $P1_x$: 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.

c. Reconfigure Controls: In the process of reaching the target outdoor dry-bulb temperature, adjust the outdoor temperature at a rate of change of no more than 20 °F per hour. This target temperature is the temperature specified by the manufacturer in the DOE Compliance Certification Database at which the crankcase heater turns on, minus five degrees Fahrenheit. Maintain this temperature within ±2 °F for at least 5 minutes, while maintaining an indoor dry-bulb temperature of between 75 °F and 85 °F.

d. Measure $P2_x$: 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.

e. Measure P_x for coil-only split systems (that would be installed in the field with a furnace having a dedicated board for indoor controls) and for blower-coil split systems for which a furnace 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.

f. Calculate *P1*:

Single-package systems and blower coil split systems for which the air mover is not a furnace: 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. If the compressor is a modulating-type, assign a value of 1.5 for the number of compressors. The expression for calculating *P1* is as follows:

$$P1 = \frac{P1_x}{number\ of\ compressors}.$$

Coil-only split systems (that would be installed in the field with a furnace having a dedicated board for indoor controls) and blower-coil split systems for which a furnace 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. If the compressor is a modulating-type, assign a value of 1.5 for the number of compressors. The expression for calculating *P1* is as follows:

$$P1 = \frac{P1_x - P_x}{number\ of\ compressors}.$$

h. Calculate *P2*:

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. If the compressor is a modulating-type, assign a value of 1.5 for the number of compressors. The expression for calculating *P2* is as follows:

$$P2 = \frac{P2_x}{number\ of\ compressors}.$$

Coil-only split systems (that would be installed in the field with a furnace having a dedicated board for indoor controls) and blower-coil split systems for which a furnace 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. If the compressor is a modulating-type, assign a value of 1.5 for the number of compressors. The expression for calculating *P2* is as follows:

$$P2 = \frac{P2_x - P_x}{number\ of\ compressors}.$$

4. Calculations of Seasonal Performance Descriptors

4.1 Seasonal Energy Efficiency Ratio (SEER) Calculations. SEER must be calculated as follows: For equipment covered under sections 4.1.2, 4.1.3, and 4.1.4, 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, use a building cooling load, $BL(T_j)$. When referenced, evaluate $BL(T_j)$ for cooling using,

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

where,

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

1.1 = sizing factor, dimensionless.

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

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

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, Btu/h per watt.

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

b. Refer to section 3.3 regarding the definition and calculation of $\dot{Q}_c(82)$ and $\dot{E}_c(82)$.

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 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,

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

$X(T_j) = \begin{Bmatrix} BL(T_j)/\dot{Q}_c(T_j) \\ or \\ 1 \end{Bmatrix}$ whichever is less; the cooling mode load factor for temperature bin j, dimensionless.

where,

$\dot{Q}_c(T_j)$ = 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,

$$\text{Equation 4.1.2-2} \quad \dot{Q}_c(T_j) = \dot{Q}_c^{k=1}(T_j) + \frac{\dot{Q}_c^{k=2}(T_j) - \dot{Q}_c^{k=1}(T_j)}{\mathrm{FP}_c^{k=2} - \mathrm{FP}_c^{k=1}} * \left[\mathrm{FP}_c(T_j) - \mathrm{FP}_c^{k=1}\right]$$

where,

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

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

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

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

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

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

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

$$\dot{E}_c(T_j) = \dot{E}_c^{k=1}(T_j) + \frac{\dot{E}_c^{k=2}(T_j) - \dot{E}_c^{k=1}(T_j)}{FP_c^{k=2} - FP_c^{k=1}} * [FP_c(T_j) - FP_c^{k=1}]$$

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 regarding the definitions and calculations of $\dot{E}_c^{k=1}(82)$, $\dot{E}_c^{k=1}(95)$, $\dot{E}_c^{k=2}(82)$, and $\dot{E}_c^{k=2}(95)$.

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

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

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

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

where $\dot{Q}_c^{k=1}(82)$ and $\dot{E}_c^{k=1}(82)$ are determined from the B_1 Test, $\dot{Q}_c^{k=1}(67)$ and $\dot{E}_c^{k=1}(67)$ are determined from the F_1 Test, and all four quantities are calculated as specified in section 3.3. Evaluate the space cooling capacity, $\dot{Q}_c^{k=2}(T_j)$, and electrical power consumption, $\dot{E}_c^{k=2}(T_j)$, of the test unit when operating at high compressor capacity and outdoor temperature T_j using, Equation 4.1.3-3 $\dot{Q}_c^{k=2}(T_j) = \dot{Q}_c^{k=2}(82) + \frac{\dot{Q}_c^{k=2}(95) - \dot{Q}_c^{k=2}(82)}{95-82} * (T_j - 82)$

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

where $\dot{Q}_c^{k=2}(95)$ and $\dot{E}_c^{k=2}(95)$ are determined from the A_2 Test, $\dot{Q}_c^{k=2}(82)$, and $\dot{E}_c^{k=2}(82)$, are determined from the B_2Test, and all are calculated as specified in section 3.3.

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

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

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

where,

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

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

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

Obtain the fractional bin hours for the cooling season, n_j/N, from Table 18. Use Equations 4.1.3-1 and 4.1.3-2, respectively, to evaluate $\dot{Q}_c^{k=1}(T_j)$ and $\dot{E}_c^{k=1}(T_j)$.

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 $\dot{E}_c^{k=1}(T_j)$.

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 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, $BL(T_j) < Q_c^{k=2}(T_j)$. This section applies to units that lock out low compressor capacity operation at higher outdoor temperatures.

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

where,

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

$PLF_j = 1 - C_D^c(k = 2) * [1 - X^{k=2}(T_j)$ the part load factor, dimensionless.

Obtain the fraction bin hours for the cooling season, $\frac{n_j}{N}$, from Table 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 are not conducted, set C_D^c (k=2) equal to the default value specified in section 3.5.3

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. Evaluate the space cooling capacity, $\dot{Q}_c^{k=2}(T_j)$, and electrical power consumption, $\dot{E}_c^{k=2}(T_j)$, of the test unit when operating at maximum compressor speed and outdoor temperature T_j. Use Equations 4.1.3–3 and 4.1.3–4, respectively, where $\dot{Q}_c^{k=2}(95)$ and $\dot{E}_c^{k=2}(95)$ are determined from the A_2 Test, $\dot{Q}_c^{k=2}(82)$ and $\dot{E}_c^{k=2}(82)$ are determined from the B_2 Test, and all four quantities are calculated as specified in section 3.3. Calculate the space cooling capacity, $\dot{Q}_c^{k=v}(T_j)$, and electrical power consumption, $\dot{E}_c^{k=v}(T_j)$, of the test unit when operating at outdoor temperature T_j and the intermediate compressor speed used during the section 3.2.4 (and Table 7) E_V Test using,

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

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

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

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

where,

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

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

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

Obtain the fractional bin hours for the cooling season, n_j/N, from Table 18. Use Equations 4.1.3–1 and 4.1.3–2, respectively, to evaluate $\dot{Q}_c^{k=1}(T_j)$ and $\dot{E}_c^{k=1}(T_j)$.

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, $\dot{Q}_c^{k=1}(T_j) < BL(T_j) < \dot{Q}_c^{k=2}(T_j)$.

$$\frac{q_c(T_j)}{N} = \dot{Q}_c^{k=1}(T_j) * \frac{n_j}{N} \qquad \frac{e_c(T_j)}{N} = \dot{E}_c^{k=1}(T_j) * \frac{n_j}{N}$$

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.

$\dot{E}_c^{k=1}(T_j) = \frac{\dot{Q}_c^{k=1}(T_j)}{EER^{k=1}(T_j)}$ the electrical power input required by the test unit when operating at a compressor speed of k = i and temperature T_j, W.

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,

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

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

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

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

where,

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

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

T_2 = the outdoor temperature at which the unit, when operating at maximum compressor speed, provides a space cooling capacity that is equal to the building load ($\dot{Q}_c^{k=2}(T_2) = BL(T_2)$), °F. Determine T_2 by equating Equations 4.1.3–3 and 4.1–2 and solving for outdoor temperature.

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

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

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

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

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

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

4.1.5 SEER calculations for an air conditioner or heat pump having a single indoor unit with multiple blowers. Calculate SEER using Eq. 4.1–1, where $q_c(Tj)/N$ and $e_c(Tj)/N$ are evaluated as specified in applicable below subsection.

4.1.5.1 For multiple blower systems that are connected to a lone, single-speed outdoor unit. a. Calculate the space cooling capacity, $\dot{Q}^{k=1}(Tj)$, and electrical power consumption, $\dot{E}^{k=1}(Tj)$, 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. Calculate the space cooling capacity, $\dot{Q}^{k=2}(Tj)$, and electrical power consumption, $\dot{E}^{k=2}(Tj)$, 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. In evaluating the section 4.1.2.1 equations, determine the quantities $\dot{Q}^{k=1}(82)$ and $\dot{E}^{k=1}(82)$ from the B1 Test, $\dot{Q}^{k=1}(95)$ and $\dot{E}^{k=1}(82)$ from the Al Test, $\dot{Q}^{k=2}(82)$ and $\dot{E}^{k=2}(82)$ from the B2 Test, and $\dot{Q}^{k=2}(95)$ and $\dot{E}^{k=2}(95)$ 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, CD_c, as per sections 3.2.2.1 and 3.5 to 3.5.3. Assign this same value to $CD_c(K=2)$. c. Except for using the above values of $\dot{Q}^{k=1}(Tj)$, $\dot{E}^{k=1}(Tj)$, $\dot{E}^{k=2}(Tj)$, $\dot{Q}^{k=2}(Tj)$, CD_c, and CD_c (K=2), calculate the quantities $q_c(T_j)/N$ and $e_c(T_j)/N$ as specified in section 4.1.3.1 for cases where $\dot{Q}^{k=1}(Tj) \geq BL(T_j)$. 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 if $\dot{Q}^{k=2}(Tj) > BL(T_j)$ or as specified in section 4.1.3.4 if $\dot{Q}^{k=2}(Tj) \leq BL(T_j)$.

4.1.5.2 For multiple blower systems that are connected to either a lone outdoor unit having a two-capacity compressor or to 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.

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

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

Where,

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

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

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

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

*Pacific Coast Region.

Evaluate the building heating load using

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

Where,

T_{OD} = the outdoor design temperature, °F. An outdoor design temperature is specified for each generalized climatic region in Table 19.

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

DHR = the design heating requirement (see section 1.2, Definitions), Btu/h.

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

$$DHR_{min} = \begin{cases} \dot{Q}_h^k(47) * \left[\dfrac{65 - T_{OD}}{60}\right], & for\ Regions\ I, II, III, IV, \&\ VI \\ \dot{Q}_h^k(47), & for\ Region\ V \end{cases}$$

and

$$DHR_{max} = \begin{cases} 2 * \dot{Q}_h^k(47) * \left[\dfrac{65 - T_{OD}}{60}\right], & for\ Regions\ I, II, III, IV, \&\ VI \\ 2.2 * \dot{Q}_h^k(47), & for\ Region\ V \end{cases}$$

Rounded to the nearest standardized DHR given in Table 19

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

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

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

3. For two-capacity, northern heat pumps (see section 1.2, Definitions), $\dot{Q}^k{}_h(47) = \dot{Q}^{k=1}{}_h(47)$, the space heating capacity determined from the $H1_1$ Test.

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

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

For heat pumps with heat comfort controllers (see section 1.2, 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 for the additional steps required for calculating the HSPF.

TABLE 20—STANDARDIZED DESIGN HEATING REQUIREMENTS (BTU/H)

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

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

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

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

where,

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

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

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

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

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

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

Use Equation 4.2–2 to determine $BL(T_j)$. Obtain fractional bin hours for the heating season, n_j/N, from Table 19.

Determine the low temperature cut-out factor using

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

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,

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

Equation 4.2.1-5

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

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

4.2.2 Additional steps for calculating the HSPF of a heat pump having a single-speed compressor and a variable-speed, variable-air-volume-rate indoor 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 with the exception of replacing references to the H1C Test and section 3.6.1 with the $H1C_1$ Test and section 3.6.2. In addition, evaluate the space heating capacity and electrical power consumption of the heat pump $\dot{Q}_h(T_j)$ and $\dot{E}_h(T_j)$ using

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

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

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

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

Equation 4.2.2-4

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

For units where indoor blower speed is the primary control variable, $\text{FP}_h^{k=1}$ denotes the fan speed used during the required $H1_1$ and $H3_1$ Tests (see Table 11), $\text{FP}_h^{k=2}$ denotes the fan speed used during the required $H1_2$, $H2_2$, and $H3_2$ Tests, and $\text{FP}_h(T_j)$ denotes the fan speed used by the unit when the outdoor temperature equals T_j. For units where indoor air volume rate is the primary control variable, the three FP_h's are similarly defined only now being expressed in terms of air volume rates rather than fan speeds. Determine $\dot{Q}_h^{k=1}(47)$ and $\dot{E}_h^{k=1}(47)$ from the $H1_1$ Test, and $\dot{Q}_h^{k=2}(47)$ and $\dot{E}_h^{k=2}(47)$ from the $H1_2$ Test. Calculate all four quantities as specified in section 3.7. Determine $\dot{Q}_h^{k=1}(35)$ and $\dot{E}_h^{k=1}(35)$ as specified in section 3.6.2; determine $\dot{Q}_h^{k=2}(35)$ and $\dot{E}_h^{k=2}(35)$ and from the $H2_2$ Test and the calculation specified in section 3.9. Determine $\dot{Q}_h^{k=1}(17)$ and $\dot{E}_h^{k=1}(17$ from the $H3_1$ Test, and $\dot{Q}_h^{k=2}(17)$ and $\dot{E}_h^{k=2}(17)$ from the $H3_2$ Test. Calculate all four quantities as specified in section 3.10.

4.2.3 Additional steps for calculating the HSPF of a heat pump having a two-capacity compressor. The calculation of the Equation 4.2–1 quantities differ depending upon whether the heat pump would operate at low capacity (section 4.2.3.1), cycle between low and high capacity (Section 4.2.3.2), or operate at high capacity (sections 4.2.3.3 and 4.2.3.4) in responding to the building load. For heat pumps that lock out low capacity operation at low outdoor temperatures, the manufacturer must supply information regarding the cutoff temperature(s) so that the appropriate equations can be selected.

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

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

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

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

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

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

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

$$\text{Equation 4.2.3-2} \quad \frac{RH(T_j)}{N} = \frac{BL(T_j) * [1 - \delta(T_j)]}{3.413 \frac{Btu/h}{W}} * \frac{n_j}{N}$$

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.

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. Use the calculations given in section 4.2.3.3, and not the above, if:

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

(b) T_j is below this lockout threshold temperature.

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

Calculate $\frac{RH(T_j)}{N}$ using Equation 4.2.3-2. Evaluate $\frac{e_h(T_j)}{N}$ using

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

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 is not conducted, set C_D^h (k=2) equal to the default value specified in section 3.8.1.

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

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

$$\frac{e_h(T_j)}{N} = \dot{E}_h^{k=2}(T_j) * \delta'(T_j) * \frac{n_j}{N} \qquad \frac{RH(T_j)}{N} = \frac{BL(T_j)*[\dot{Q}_h^{k=2}(T_j)*\delta\prime(T_j)]}{3.413\frac{Btu/h}{W}} * \frac{n_j}{N}$$

Where

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

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

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

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

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

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

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

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. Approximate the slopes of the k=v intermediate speed heating capacity and electrical power input curves, M_Q and M_E, as follows:

$$M_Q = \left[\frac{\dot{Q}_h^{k=1}(62) - \dot{Q}_h^{k=1}(47)}{62-47} * (1 - N_Q)\right] + \left[N_Q * \frac{\dot{Q}_h^{k=2}(35) - \dot{Q}_h^{k=2}(17)}{35-17}\right]$$

$$M_E = \left[\frac{\dot{E}_h^{k=1}(62) - \dot{E}_h^{k=1}(47)}{62-47} * (1 - N_E)\right] + \left[N_E * \frac{\dot{E}_h^{k=2}(35) - \dot{E}_h^{k=2}(17)}{35-17}\right]$$

where,

$$N_Q = \frac{\dot{Q}_h^{k=v}(35) - \dot{Q}_h^{k=1}(35)}{\dot{Q}_h^{k=2}(35) - \dot{Q}_h^{k=1}(35)} \qquad N_E = \frac{\dot{E}_h^{k=v}(35) - \dot{E}_h^{k=1}(35)}{\dot{E}_h^{k=2}(35) - \dot{E}_h^{k=1}(35)}$$

Use Equations 4.2.4-1 and 4.2.4-2, respectively, to calculate $\dot{Q}_h^{k=1}(35)$ and $\dot{E}_h^{k=1}(35)$.

The calculation of Equation 4.2-1 quantities $\frac{RH(T_j)}{N}$ and $\frac{e_h(T_j)}{N}$ differs depending upon whether the heat pump would operate at minimum speed (section 4.2.4.1), operate at an intermediate speed (section 4.2.4.2), or operate at maximum speed (section 4.2.4.3) in responding to the building load.

4.2.4.1 Steady-state space heating capacity when operating at minimum compressor speed is greater than or equal to the building heating load at temperature T_j, $\dot{Q}_h^{k=1}(T_j \geq BL(T_j)$. Evaluate the Equation 4.2–1 quantities

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

as specified in section 4.2.3.1. Except now use Equations 4.2.4–1 and 4.2.4–2 to evaluate $\dot{Q}_h^{k=1}(T_j)$ and $\dot{E}_h^{k=1}(T_j)$, respectively, and replace section 4.2.3.1 references to ''low capacity'' and section 3.6.3 with ''minimum speed'' and section 3.6.4. Also, the last sentence of section 4.2.3.1 does not apply.

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

$\frac{RH(T_j)}{N}$ using Equation 4.2.3-2 while evaluating $\frac{e_h(T_j)}{N}$ using,

$$\frac{e_h(T_j)}{N} = \dot{E}_h^{k=1}(T_j) * \delta(T_j) * \frac{n_j}{N}$$

where,

$$\dot{E}_h^{k=1}(T_j) = \frac{\dot{Q}_h^{k=1}(T_j)}{3.413\frac{Btu/h}{W} * \ COP^{k=1}(T_j)}$$

and $\delta(T_j)$ is evaluated using Equation 4.2.3–3 while,

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

$COP^{k=i}(T_j)$ = the steady-state coefficient of performance of the heat pump when operating at compressor speed k=i and temperature T_j, dimensionless.

For each temperature bin where the heat pump operates at an intermediate compressor speed, determine $COP^{k=i}(T_j)$ using,

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

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

$$D = \frac{T_3^2 - T_4^2}{T_{vh}^2 - T_4^2} \qquad B = \frac{COP^{k=2}(T_4) - COP^{k=1}(T_3) - D * [COP^{k=2}(T_4) - COP^{k=v}(T_{vh})]}{T_4 - T_3 - D * (T_4 - T_{vh})}$$

Where,=

T_3 = the outdoor temperature at which the heat pump, when operating at minimum compressor speed, provides a space heating capacity that is equal to the building load ($\dot{Q}_h^{k=1}(T_3) = BL(T_3)$), °F. Determine T_3 by equating Equations 4.2.4–1 and 4.2–2 and solving for:

$$C = \frac{COP^{k=2}(T_4) - COP^{k=2}(T_3) - B * (T_4 - T_3)}{T_4^2 - T_3^2} \qquad A = COP^{k=2}(T_4) - B * T_4 - C * T_4^2$$

outdoor temperature.

T_{vh} = the outdoor temperature at which the heat pump, when operating at the intermediate compressor speed used during the section 3.6.4 $H2_V$ Test, provides a space heating capacity that is equal to the building load ($\dot{Q}_h^{k=v}(T_{vh}) = BL(T_{vh})$), °F. Determine T_{vh} by equating Equations 4.2.4–3 and 4.2–2 and solving for outdoor temperature.

T_4 = the outdoor temperature at which the heat pump, when operating at maximum compressor speed, provides a space heating capacity that is equal to the building load ($\dot{Q}_h^{k=2}(T_4) = BL(T_4)$), °F. Determine T_4 by equating Equations 4.2.2–3 (k=2) and 4.2–2 and solving for outdoor temperature.

$$COP^{k=1}(T_3) = \frac{\dot{Q}_h^{k=1}(T_3)[Eqn. 4.2.4 - 1, substituting\ T_3\ for\ T_j]}{3.413\frac{Btu/h}{W} * \dot{E}_h^{k=1}(T_3)[Eqn. 4.2.4 - 2, substituting\ T_3\ for\ T_j]}$$

$$COP^{k=v}(T_{vh}) = \frac{\dot{Q}_h^{k=v}(T_{vh})[Eqn. 4.2.4 - 3, substituting\ T_{vh}\ for\ T_j]}{3.413\frac{Btu/h}{W} * \dot{E}_h^{k=v}(T_{vh})[Eqn. 4.2.4 - 4, substituting\ T_{vh}\ for\ T_j]}$$

$$COP^{k=2}(T_4) = \frac{\dot{Q}_h^{k=2}(T_4)[Eqn. 4.2.2 - 3, substituting\ T_4\ for\ T_j]}{3.413\frac{Btu/h}{W} * \dot{E}_h^{k=2}(T_4)[Eqn. 4.2.2 - 4, substituting\ T_4\ for\ T_j]}$$

For multiple-split heat pumps (only), the following procedures supersede the above requirements for calculating $COP_h^{k=i}(T_j)$. For each temperature bin where $T_3 > T_j > T_{vh}$,

$$COP_h^{k=1}(T_j) = COP_h^{k=1}(T_3) + \frac{COP_h^{k=v}(T_{vh}) - COP_h^{k=1}(T_3)}{T_{vh} - T_3} * (T_j - T_3)$$

For each temperature bin where $T_{vh} \geq T_j > T_4$,

$$COP_h^{k=1}(T_j) = COP_h^{k=v}(T_{vh}) + \frac{COP_h^{k=2}(T_4) - COP_h^{k=v}(T_{vh})}{T_4 - T_{vh}} * (T_j - T_{vh})$$

4.2.4.3 Heat pump must operate continuously at maximum (k=2) compressor speed at temperature T_j, $BL(T_j) \geq \dot{Q}_h^{k=2}(T_j)$. Evaluate the Equation 4.2–1 quantities

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

as specified in section 4.2.3.4 with the understanding that $\dot{Q}_h^{k=2}(T_j)$ and $\dot{E}_h^{k=2}(T_j)$ correspond to maximum compressor speed operation and are derived from the results of the specified section 3.6.4 tests.

4.2.5 Heat pumps having a heat comfort controller. Heat pumps having heat comfort controllers, when set to maintain a typical minimum air delivery temperature, will cause the heat pump condenser to operate less because of a greater contribution from the resistive elements. With a conventional heat pump, resistive heating is only initiated if the heat pump condenser cannot meet the building load (*i.e.,* is delayed until a second stage call from the indoor thermostat). With a heat comfort controller, resistive heating can occur even though the heat pump condenser has adequate capacity to meet the building load (*i.e.,* both on during a first stage call from the indoor thermostat). As a result, the outdoor temperature where the heat pump compressor no longer cycles (*i.e.,* starts to run continuously), will be lower than if the heat pump did not have the heat comfort controller.

4.2.5.1 Heat pump having a heat comfort controller: additional steps for calculating the HSPF of a heat pump having a single-speed compressor that was tested with a fixed-speed indoor blower installed, a constant-air-volume-rate indoor blower installed, or with no indoor blower installed. Calculate the space heating capacity and electrical power of the heat pump without the heat comfort controller being active as specified in section 4.2.1 (Equations 4.2.1–4 and 4.2.1–5) for each outdoor bin temperature, T_j, that is listed in Table 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 $Btu/lbm_{da} \cdot °F$) from the results of the H1 Test using:

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

where $\overline{\dot{V}}_s$, $\overline{\dot{V}}_{mx}$, v'_n (or v_n), and W_n are defined following Equation 3–1. For each outdoor bin temperature listed in Table 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. For each bin calculation, use the space heating capacity and electrical power from Case 1 or Case 2, whichever applies.

Case 1. For outdoor bin temperatures where $T_o(T_j)$ is equal to or greater than T_{CC} (the maximum supply temperature determined according to section 3.1.9), determine $\dot{Q}_h(T_j)$ and $\dot{E}_h(T_j)$ as specified in section 4.2.1 (*i.e.,* $\dot{Q}_h(T_j) = \dot{Q}_{hp}(T_j)$ and $\dot{E}_{hp}(T_j) = \dot{E}_{hp}(T_j)$). Note: Even though $T_o(T_j) \geq T_{cc}$, resistive heating may be required; evaluate Equation 4.2.1–2 for all bins.

Case 2. For outdoor bin temperatures where $T_o(T_j) > T_{cc}$, determine $\dot{Q}_h(T_j)$ and $\dot{E}_h(T_j)$ using,

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

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

Where,

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

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 (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:

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

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

Where $\overline{\dot{V}}_s$, $\overline{\dot{V}}_{mx}$, v'_n (or v_n), and W_n are defined following Equation 3–1. For each outdoor bin temperature listed in Table 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 with the exception of replacing references to the H1C Test and section 3.6.1 with the H1C$_1$ Test and section 3.6.2. For each bin calculation, use the space heating capacity and electrical power from Case 1 or Case 2, whichever applies.

Case 1. For outdoor bin temperatures where $T_o(T_j)$ is equal to or greater than T_{CC} (the maximum supply temperature determined according to section 3.1.9), determine $\dot{Q}_h(T_j)$ and $\dot{E}_h(T_j)$ as specified in section 4.2.2 (*i.e.* $\dot{Q}_h(T_j) = \dot{Q}_{hp}(T_j)$ and $\dot{E}_h(T_j) = \dot{E}_{hp}(T_j)$). Note: Even though $T_o(T_j) \geq T_{CC}$, resistive heating may be required; evaluate Equation 4.2.1–2 for all bins.

Case 2. For outdoor bin temperatures where $T_o(T_j) < T_{CC}$, determine $\dot{Q}_h(T_j)$ and $\dot{E}_h(T_j)$ using,

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

Where,

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

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 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:

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

$$C_{p,da}^{k=1} = 0.24 + 0.444 * W_n$$

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

Where

$\dot{V}_s$, $\dot{V}_{mx}$, 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, whichever applies, for each temperature bin. To evaluate these quantities, use the low-capacity space heating capacity and the low-capacity electrical power from Case 1 or Case 2, whichever applies; use the high-capacity space heating capacity and the high-capacity electrical power from Case 3 or Case 4, whichever applies.

Case 1. For outdoor bin temperatures where $T_o^{k=1}(T_j)$ is equal to or greater than T_{CC} (the maximum supply temperature determined according to section 3.1.9), determine $\dot{Q}_h^{k=1}(T_j)$ and $\dot{E}_h^{k=1}(T_j)$ as specified in section 4.2.3 (*i.e.,* $\dot{Q}_h^{k=1}(T_j) = \dot{Q}_{hp}^{k=1}(T_j)$ and $\dot{E}_h^{k=1}(T_j) = \dot{E}_{hp}^{k=1}(T_j)$.

Note: Even though $T_o^{k=1}(T_j) \geq T_{CC}$, resistive heating may be required; evaluate $RH(T_j)/N$ for all bins.

Case 2. For outdoor bin temperatures where $T_o^{k=1}(T_j) < T_{CC}$, determine $\dot{Q}_h^{k=1}(T_j)$ and $\dot{E}_h^{k=1}(T_j)$ using,

$$\dot{Q}_h^{k=1}(T_j) = \dot{Q}_{hp}^{k=1}(T_j) + \dot{Q}_{CC}^{k=1}(T_j) \qquad \dot{E}_h^{k=1}(T_j) = \dot{E}_{hp}^{k=1}(T_j) + \dot{E}_{CC}^{k=1}(T_j)$$

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 (*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), cycle on and off at high capacity (section 4.2.6.2), cycle on and off at booster capacity (4.2.6.3), cycle between low and high capacity (section 4.2.6.4), cycle between high and booster capacity (section 4.2.6.5), operate continuously at low capacity (4.2.6.6), operate continuously at high capacity (section 4.2.6.7), operate continuously at booster capacity (4.2.6.8), or heat solely using resistive heating (also section 4.2.6.8) 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 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. If, in accordance with section 3.6.6, 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 and determine $\dot{Q}_h^{k=1}(35)$ and $\dot{E}_h^{k=1}(35)$ as specified in section 3.6.6.

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. Determine the equation input for $\dot{Q}_h^{k=2}(35)$ and $\dot{E}_h^{k=2}(35)$ from the $H2_2$, evaluated as specified in section 3.9.1. 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.

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

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

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

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. 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.

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.

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. Determine the equation inputs $X^{k=2}(T_j)$, PLF_j, and $\delta'(T_j)$ as specified in section 4.2.3.3. 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.

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

$$\frac{e_h(T_j)}{N} = \frac{X^{k=3}(T_j) * \dot{E}_h^{k=3}(T_j) * \delta'(T_j)}{PLF_j} * \frac{n_j}{N}$$

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.

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. 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.

4.2.6.5 Heat pump alternates between high (k=2) and booster (k=3) compressor capacity to satisfy the building heating load at a temperature T_j, $\dot{Q}_h^{k=2}(T_j) < BL(T_j) < \dot{Q}_h^{k=3}(T_j)$.

Calculate $\frac{RH(T_j)}{N}$ and using Eq. 4.2.3-2. Evaluate $\frac{e_h(T_j)}{N}$ using

$$\frac{e_h(T_j)}{N} = [X^{k=2}(T_j) * \dot{E}_h^{k=2}(T_j) + X^{k=3}(T_j) * \dot{E}_h^{k=3}(T_j)] * \delta'(T_j) * \frac{n_j}{N}$$

where:

$$X^{k=2}(T_j) = \frac{\dot{Q}_h^{k=3}(T_j) - BL(T_j)}{\dot{Q}_h^{k=3}(T_j) - \dot{Q}_h^{k=2}(T_j)}$$

and $X^{k=3}(T_j) = X^{k=2}(T_j)$ = the heating mode, booster capacity load factor for temperature bin j, dimensionless. Determine the low temperature cut-out factor, $\delta'(T_j)$, using Eq. 4.2.3–3.

4.2.6.6 Heat pump only operates at low (k=1) capacity at temperature T_j and its capacity is less than the building heating load, $BL(T_j) > \dot{Q}_h^{k=1}(T_j)$.

$$\frac{e_h(T_j)}{N} = \dot{E}_h^{k=1}(T_j) * \delta'(T_j) * \frac{n_j}{N} \quad \text{and} \frac{RH(T_j)}{N} = \frac{BL(T_j) - [\dot{Q}_h^{k=1}(T_j) * \delta'(T_j)]}{3.413 \frac{Btu/h}{W}} * \frac{n_j}{N}$$

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. Calculate $\delta''(Tj)$ using the equation given in section 4.2.3.4.

4.2.6.8 Heat pump only operates at booster (k = 3) capacity at temperature Tj and its capacity is less than the building heating load, $BL(T_j) > \dot{Q}_h^{k=3}(T_j)$. or the system converts to using only resistive heating.

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

Where $\delta''(Tj)$ is calculated as specified in section 4.2.3.4 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 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 applicable below subsection.

4.2.7.1 For multiple 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. Determine the quantities $\dot{Q}_h^{k=1}(35)$ and $\dot{E}_h^{k=1}(35)$ as specified in section 3.6.2. Determine $\dot{Q}_h^{k=2}(35)$ and $\dot{E}_h^{k=2}(35)$ from the $H2_2$ Frost Accumulation Test as calculated according to section 3.9.1. 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. Refer to section 3.6.2 and Table 11 for additional information on the referenced laboratory tests.

b. Determine the heating mode cyclic degradation coefficient, CD_h, as per sections 3.6.2 and 3.8 to 3.8.1. Assign this same value to $CD_h(k = 2)$.

c. Except for using the above values of $\dot{Q}_h^{k=1}(Tj)$, $\dot{E}_h^{k=1}(Tj)$, $\dot{Q}_h^{k=2}(Tj)$, $\dot{E}_h^{k=2}(Tj)$, CD_h, and $CD_h(k = 2)$, calculate the quantities $e_h(T_j)/N$ as specified in section 4.2.3.1 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 if $\dot{Q}_h^{k=2}(Tj) > BL(Tj)$ or as specified in section 4.2.3.4 if $\dot{Q}_h^{k=2}(Tj) \leq BL(T_j)$

4.2.7.2 For multiple blower heat pumps connected to either a lone 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.

4.3 Calculations of Off-mode Seasonal Power and Energy Consumption.

4.3.1 For central air conditioners and heat pumps with a cooling capacity of:

less than 36,000 Btu/h, determine the off mode rating, $P_{W,OFF}$, with the following equation:

$$P_{W,OFF} = \begin{cases} P1 & if\ P2 = 0 \\ \frac{P1+P2}{2} & otherwise \end{cases};$$

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 rating, $P_{W,OFF}$, with the following equation: $P_{W,OFF} = \begin{cases} \frac{P1}{F_{scale}} & if\ P2 = 0 \\ \frac{(P1+P2)/2}{F_{scale}} & otherwise \end{cases};$

4.3.2 Calculate the off mode energy consumption for both central air conditioner and heat pumps for the shoulder season, *E1,* using: $E1 = P1 \cdot SSH$; and the off mode energy consumption of a CAC, only, for the heating season, *E2,* using: $E2 = P2 \cdot HSH$; where *P1* and *P2* is determined in Section 3.13. *HSH* can be determined by multiplying the heating season-hours from Table 21 with the fractional Bin-hours, from Table 19, that pertain to the range of temperatures at which the crankcase heater operates. If the crankcase heater is controlled to disable for the heating season, the temperature range at which the crankcase heater operates is defined to be from 72 °F to five degrees Fahrenheit below a turn-off temperature specified by the manufacturer in the DOE Compliance Certification Database. If the crankcase heater is operated during the heating season, the temperature range at which the crankcase heater operates is defined to be from 72 °F to −23 °F, the latter of which is a temperature that sets the range of Bin-hours to encompass all outside air temperatures in the heating season.

SSH can be determined by multiplying the shoulder season-hours from Table 21 with the fractional Bin-hours in Table 22.

TABLE 21—REPRESENTATIVE COOLING AND HEATING LOAD HOURS AND THE CORRESPONDING SET OF SEASONAL HOURS FOR EACH GENERALIZED CLIMATIC REGION

Climatic region	Cooling load hours CLH_R	Heating load hours HLH_R	Cooling season hours CSH_R	Heating season hours HSH_R	Shoulder season hours SSH_R
I	2400	750	6731	1826	203
II	1800	1250	5048	3148	564
III	1200	1750	3365	4453	942
IV	800	2250	2244	5643	873
Rating Values	1000	2080	2805	5216	739
V	400	2750	1122	6956	682
VI	200	2750	561	6258	1941

HSH is evaluated as: $HSH = \frac{HLH \cdot (65 - T_{OD})}{\sum_{j=1}^{J} (65 - T_j) \cdot \frac{n_j}{N}},$

where T_{OD} and $\frac{n_j}{N}$ are listed in Table 18 and depend on the location of interest relative to Figure 1. For the six generalized climatic regions, this equation simplifies to the following set of equations:

Region I: $HSH = 2.4348 HLH$;

Region II: $HSH = 2.5182 HLH$;

Region III: $HSH = 2.5444 HLH$;

Region IV: $HSH = 2.5078HLH$;
Region V: $HSH = 2.5295HLH$;
Region VI: $HSH = 2.2757HLH$.

SSH is evaluated: $SSH = 8760 - (CSH + HSH)$, where CSH = the cooling season hours calculated using $CSH = 2.8045 \cdot CLH$

TABLE 22—FRACTIONAL BIN HOURS FOR THE SHOULDER SEASON HOURS FOR ALL REGIONS

Tj(°F)	Fractional bin hours	
	Air conditioners	Heat pumps
72	0.333	0.167
67	0.667	0.333
62	0	0.333
57	0	0.167

4.3.3 If a shoulder season crankcase heater time delay and/or a heating season crankcase heater time delay is specified by the manufacturer, multiply E1 and/or E2, by $\left(1 - \frac{t_{delay,i}}{60}\right)$, where $t_{delay,1}$ is the time delay for operation during the shoulder season and $t_{delay,2}$ is the time delay for operation during the heating season, in minutes. If a time delay is not specified, $t_{delay,i}$ is 15 minutes.

4.3.4 For air conditioners, the annual off mode energy consumption, E_{TOTAL}, is: $E_{TOTAL} = E1 + E2$.

4.3.5 For heat pumps, the annual off mode energy consumption, E_{TOTAL}, is $E1$.

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

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

$$APF_A = \frac{CLH_A \cdot \dot{Q}_c^k(95) + HLH_A \cdot DHR \cdot C}{\frac{CLH_A \cdot \dot{Q}_c^k(95)}{SEER} + \frac{HLH_A \cdot DHR \cdot C}{HSPF} + P1 \cdot SSH + P2 \cdot HSH},$$

Where,

CLH_A = the actual cooling hours for a particular location as determined using the map given in Figure 2, hr.

$\dot{Q}_c^k(95)$ = the space cooling capacity of the unit as determined from the A or A_2 Test, whichever applies, Btu/h.

HLH_A = the actual heating hours for a particular location as determined using the map given in Figure 1, hr.

DHR = the design heating requirement used in determining the HSPF; refer to section 4.2 and see section 1.2, Definitions, Btu/h.

C = defined in section 4.2 following Equation 4.2–2, dimensionless.

SEER = the seasonal energy efficiency ratio calculated as specified in section 4.1, Btu/W·h.

HSPF = the heating seasonal performance factor calculated as specified in section 4.2 for the generalized climatic region that includes the particular location of interest (see Figure 1), Btu/W·h. The HSPF should correspond to the actual design heating requirement (DHR), if known. If it does not, it may correspond to one of the standardized design heating requirements referenced in section 4.2.

P1 is the shoulder season per-compressor off mode power, as determined in section 3.13, W.

SSH is the shoulder season hours, hr.

P2 is the heating season per-compressor off mode power, as determined in section 3.13, W.

HSH is the heating season hours, hr.

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

$$APF_R = \frac{CLH_R \cdot \dot{Q}_c^k(95) + HLH_R \cdot DHR \cdot C}{\frac{CLH_R \cdot \dot{Q}_c^k(95)}{SEER} + \frac{HLH_R \cdot DHR \cdot C}{HSPF} + P1 \cdot SSH + P2 \cdot HSH},$$

Where,

CLH_R = the representative cooling hours for each generalized climatic region, Table 23, hr.

HLH_R = the representative heating hours for each generalized climatic region, Table 23, hr.

HSPF = the heating seasonal performance factor calculated as specified in section 4.2 for the each generalized climatic region and for each standardized design heating requirement within each region, Btu/W.h.

The SEER, $\dot{Q}_c^k(95)$, DHR, and C are the same quantities as defined in section 4.3.1. Figure 1 shows the generalized climatic regions. Table 20 lists standardized design heating requirements.

TABLE 23—REPRESENTATIVE COOLING AND HEATING LOAD HOURS FOR EACH GENERALIZED CLIMATIC REGION

Region	CLH_R	HLH_R
I	2400	750
II	1800	1250
III	1200	1750

TABLE 23—REPRESENTATIVE COOLING AND HEATING LOAD HOURS FOR EACH GENERALIZED CLIMATIC REGION—Continued

Region	CLH_R	HLH_R
IV	800	2250
V	400	2750
VI	200	2750

4.5. Rounding of SEER, HSPF, and APF for reporting purposes. After calculating SEER according to section 4.1, HSPF according to section 4.2, and APF according to section 4.3, round the values off as specified in subpart B 430.23(m) of Title 10 of the Code of Federal Regulations.

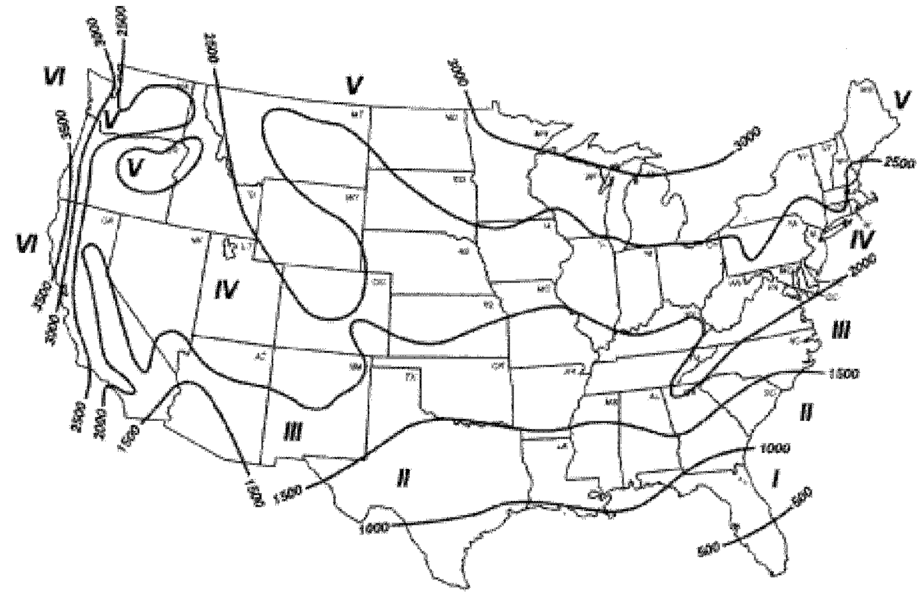

Figure 1—Heating Load Hours (HLH_A) for the United States

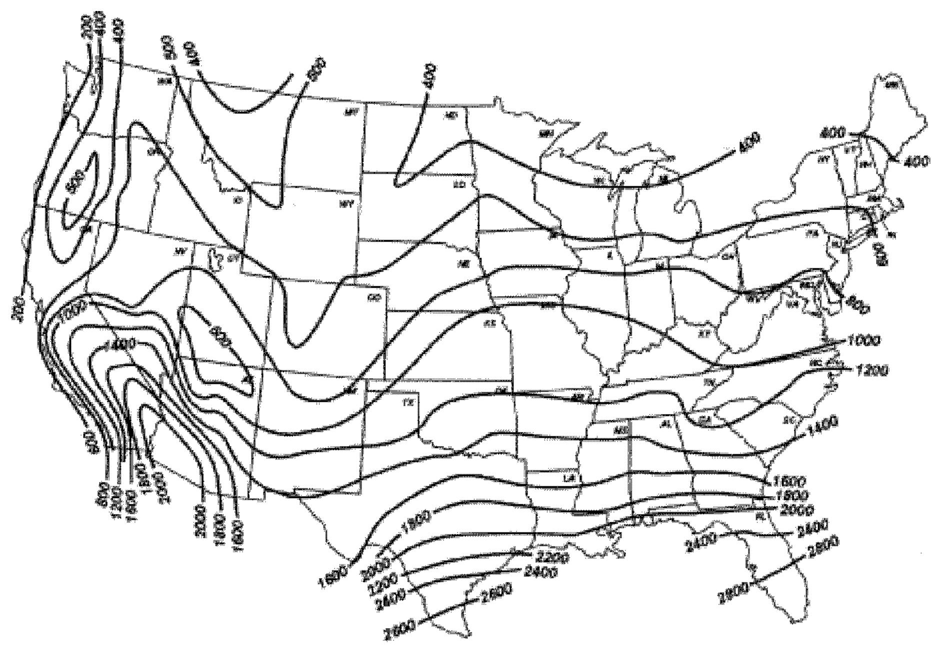

Figure 2—Cooling Load Hours (CLH_A) for the United States

4.6 Calculations of the SHR, which should be computed for different equipment configurations and test conditions specified in Table 24.

TABLE 24—APPLICABLE TEST CONDITIONS FOR CALCULATION OF THE SENSIBLE HEAT RATIO

Equipment configuration	Reference Table No. 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.

4.7 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. Add the letter identification for each steady-state test as a subscript (*e.g.,* EER_{A2}) to differentiate among the resulting EER values.

■ 11. 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

Note: Prior to May 9, 2016, any representations, including compliance certifications, made with respect to the energy use, power, or efficiency of central air conditioners and central air conditioning heat pumps must be based on the results of testing pursuant to either Appendix M or the procedures in Appendix M as it appeared at 10 CFR part 430, subpart B, Appendix M, in the 10 CFR parts 200 to 499 edition revised as of January 1, 2015. Any representations made with respect to the energy use or efficiency of such central air conditioners and central air conditioning heat pumps must be in accordance with whichever version is selected.

On or after May 9, 2016 and prior to the compliance date for any amended energy conservation standards, 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.

On or after the compliance date for any amended energy conservation standards, 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 (Appendix M1).

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

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

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

1.2. Definitions

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

Airflow prevention device denotes a device(s) 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.

Annual performance factor means the total heating and cooling done by a heat pump in a particular region in one year divided by the total electric energy used in one year.

Blower coil indoor unit means the indoor unit of a split-system central air conditioner or heat pump that includes a refrigerant-to-air heat exchanger coil, may include a cooling-mode expansion device, and includes either an indoor blower housed with the coil or a separate designated air mover such as a furnace or a modular blower (as defined in Appendix AA). *Blower coil system* refers to a split-system that includes one or more blower coil indoor units.

CFR means Code of Federal Regulations.

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

Coil-only indoor unit means the indoor unit of a split-system central air conditioner or heat pump that includes a refrigerant-to-air heat exchanger coil and may include a cooling-mode expansion device, but does not include an indoor blower housed with the coil, and does not include a separate designated air mover such as a furnace or a modular blower (as defined in Appendix AA). A coil-only indoor unit is designed to use a separately-installed furnace or a modular blower for indoor air movement.

Coil-only system refers to a system that includes one or more coil-only indoor units.

Condensing unit removes the heat absorbed by the refrigerant to transfer it to the outside environment, and which 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 5 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. The denominator is the total cooling that would be delivered, given the same ambient conditions, had the unit operated continuously at its steady-state, space-cooling capacity for the same total time (ON + OFF) interval.

Crankcase heater means any electrically powered device or mechanism for intentionally generating heat within and/or around the compressor sump volume often done to minimize the dilution of the compressor's refrigerant oil by condensed refrigerant. 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.

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 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. One acceptable alternative to the criterion given in the prior sentence is a feedback system that measures the length of the defrost period and adjusts defrost frequency accordingly. In all cases, when the frost parameter(s) reaches a predetermined value, the system initiates a defrost. In a demand-defrost control system, defrosts are terminated based on monitoring a parameter(s) that indicates that frost has been eliminated from the coil. (Note: Systems that vary defrost intervals according to outdoor dry-bulb temperature are not demand-defrost systems.) A demand-defrost control system, which otherwise meets the above requirements, may allow time-initiated defrosts if, and only if, such defrosts occur after 6 hours of compressor operating time.

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

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

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. These rate quantities must be determined from a single test or, if derived via interpolation, must be determined at a single set of operating conditions. EER is expressed in units of

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

When determined for a ducted unit tested without an indoor blower installed, EER must include the section 3.3 and 3.5.1 default values for the heat output and power input of a fan motor.

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

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

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

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 space heating season, expressed in Btu's, divided by the total electrical energy consumed by the heat pump system during the same season, expressed in watt-hours. The HSPF used to evaluate compliance with the Energy Conservation Standards (see 10 CFR 430.32(c)) is based on Region IV, the design heating requirement, 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 transfers heat between the refrigerant and the indoor air and consists of an indoor coil and casing and may include a cooling mode expansion device and/or an air moving device.

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 indoor coil-only or indoor blower coil units connected to its other component(s) 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 in the product name plate. Nominal cooling capacity is approximate to the air conditioner cooling capacity tested at A or A2 condition. Nominal heating capacity is approximate to the heat pump heating capacity tested in H12 test (or the optional H1N test).

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

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, the shoulder season and the entire heating season; and for heat pumps, the shoulder season only.

Outdoor unit 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, could include a heating mode expansion device, reversing valve, and 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 energy efficiency ratio (coefficient of performance) to the steady-state energy efficiency ratio (coefficient of performance), where both energy efficiency ratios (coefficients of performance) 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.

Short ducted system means a ducted split system whose one or more indoor sections produce greater than zero but no greater than 0.1 inches (of water) of external static pressure when operated at the full-load air volume not exceeding 450 cfm per rated ton of cooling.

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 that has one indoor coil-only or indoor blower coil unit connected to its other component(s) with a single refrigeration circuit.

Single-zone-multiple-coil 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.

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

Split system means any air conditioner or heat pump that has one or more of the major assemblies separated from the others. 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 single-zone-multiple-coil, 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 shall:

(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) Represent the highest sales volume model family that can meet the 95 percent nominal cooling capacity of the outdoor unit [Note: another indoor model family may be used if five indoor units from the highest sales volume model family do not provide sufficient capacity to meet the 95 percent threshold level].

(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.

(vi) Where referenced, ''nominal cooling capacity'' is to be interpreted for indoor units as 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 as the lowest cooling capacity listed in published product literature for these conditions. If incomplete or no operating conditions are reported, the highest (for indoor units) or lowest (for outdoor units) such cooing capacity shall be used.

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

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

In a second application of the control scheme, one or more parameters are measured (*e.g.,* air and/or refrigerant temperatures) at the predetermined, cumulative, compressor ON-time. A defrost is initiated only if the measured parameter(s) falls within a predetermined range. The ON-time counter is reset regardless of whether or not a defrost is initiated. If systems of this second type use cumulative ON-time intervals of 10 minutes or less, then the heat pump may qualify as having a demand defrost control system (see definition).

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

Triple-split system means a central air conditioner or heat pump that is composed of three separate components: An outdoor fan coil section, an indoor blower coil section, 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 for heating mode tests may be the same or different from the cooling mode value.

For such systems, high capacity means the compressor(s) operating at low 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 certified indoor coil model number should reflect whether the ratings pertain to the lockout enabled option via the inclusion of an extra identifier, such as ''+LO''. When testing as a two-capacity, northern heat pump, the lockout feature must remain enabled for all tests.

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. Single-phase VRF systems less than 65,000 Btu/h are a kind of 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.

For such a system, maximum speed means the maximum operating speed, measured by RPM or frequency (Hz), that the unit is designed to operate in cooling mode or heating mode. Maximum speed does not change with ambient temperature, and it can be different from cooling mode to heating mode. Maximum speed does not necessarily mean maximum capacity.

For such systems, minimum speed means the minimum speed, measured by RPM or frequency (Hz), that the unit is designed to operate in cooling mode or heating mode. Minimum speed does not change with ambient temperature, and it can be different from cooling mode to heating mode. Minimum speed does not necessarily mean minimum capacity.

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 ANSI/AHRI Standard 1230–2010 sections 3 (except 3.8, 3.9, 3.13, 3.14, 3.15, 3.16, 3.23, 3.24, 3.26, 3.27, 3.28, 3.29, 3.30, and 3.31), 5.1.3, 5.1.4, 6.1.5 (except Table 8), 6.1.6, and 6.2 (incorporated by reference, see § 430.3) and Appendix M. Where ANSI/AHRI Standard 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 ANSI/AHRI Standard 1230–2010.

For definitions use section 1 of Appendix M and section 3 of ANSI/AHRI Standard 1230–2010, excluding sections 3.8, 3.9, 3.13, 3.14, 3.15, 3.16, 3.23, 3.24, 3.26, 3.27, 3.28, 3.29, 3.30, and 3.31. For rounding requirements refer to § 430.23 (m). For determination of certified rating requirements refer to § 429.16.

For test room requirements, refer to section 2.1 from Appendix M. 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 from Appendix M, and sections 5.1.3 and 5.1.4 of ANSI/AHRI Standard 1230–2010.

For general requirements for the test procedure refer to section 3.1 of Appendix M, 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 Table 8) and 6.1.6 of ANSI/AHRI Standard 1230–2010. For external static pressure requirements, refer to Table 3 in Appendix M.

For the test procedure, refer to sections 3.3 to 3.5 and 3.7 to 3.13 in Appendix M. For cooling mode and heating mode test conditions, refer to section 6.2 of ANSI/AHRI Standard 1230–2010. For calculations of seasonal performance descriptors use section 4 of Appendix M.

(B) For systems other than VRF, only a subset of the sections listed in this test procedure apply when testing and rating 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. 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 A 1

			Testing conditions	Testing procedures			Calculations		
			General	General	Cooling*	Heating**	General	Cooling*	Heating**
Requirements for All units (except VRF)			2.1; 2.2a-c; 2.2.1; 2.2.4; 2.2.4.1; 2.2.4.1 (1); 2.2.4.2; 2.2.5.1-5; 2.2.5.7-8; 2.3; 2.3.1; 2.3.2; 2.4; 2.4.1a,d; 2.5a-c; 2.5.1; 2.5.2 - 2.5.4.2; 2.5.5 – 2.13	3.1; 3.1.1-3; 3.1.5-9; 3.11; 3.12	3.3; 3.4; 3.5a-i	3.1.4.7; 3.1.10; 3.7a,b,d; 3.8a,d; 3.8.1; 3.9; 3.10	4.4; 4.5; 4.6	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,c,2.4d; 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; 2.4.1; 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		3.1.7	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 VRF SDHV†	2.1; 2.2.a; 2.2.b; 2.2.c; 2.2.1; 2.2.2; 2.2.3(a); 2.2.3(c);, 2.2.4; 2.2.5; 2.4-2.12	3.1 (except 3.1.3, 3.1.4) 3.1.4.1.1c; 3.11-3.13	3.3-3.5	3.7–3.10	4.4; 4.5; 4.6	4.1	4.2
	Modulation Capability	Single speed compressor, fixed speed fan			3.2.1	3.6.1		4.1.1	4.2.1
		Single speed compressor, VAV fan		3.1.7	3.2.2	3.6.2		4.1.2	4.2.2
		Two-capacity compressor		3.1.10	3.2.3	3.6.3		4.1.3	4.2.3
		Variable speed compressor			3.2.4	3.6.4		4.1.4	4.2.4
	Special Features	Heat pump with heat comfort controller		3.1.9		3.6.5			4.2.5
		Units with a multi-speed outdoor fan	2.2.2						
		Single indoor unit having multiple blowers			3.26	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 ANSI/AHRI Standard 1230-2010 with Addendum 2, 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 rating VRF multiple-split and VRF SDHV systems.

NOTE: For all units, use section 3.13 for off mode testing procedures and section 4.3 for off mode calculations. For all units subject to an EER standard, use section 4.7 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 available indoor test rooms as needed to accommodate the total number of indoor units. These rooms must comply with the requirements specified in sections 8.1.2 and 8.1.3 of ASHRAE Standard 37–2009 (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. When applied, cycle the heater located in the same room as the test unit evaporator coil ON and OFF when the test unit cycles ON and OFF. Cycle the heater located in the same room as the test unit condensing coil ON and OFF when the test unit cycles OFF and ON.

2.2 Test unit installation requirements.

a. Install the unit according to section 8.2 of ASHRAE Standard 37–2009 (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) with Addendum 1 and 2. For the vapor refrigerant line(s), use the insulation included with the unit; if no insulation is provided, refer to 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, refer to 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;

(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 thermostatic expansion valve with internal pressure equalization that the valve manufacturer's product literature indicates is appropriate for the system.

(3) When testing triple-split systems (see section 1.2, Definitions), use the tubing length specified in section 6.1.3.5 of AHRI 210/240–2008 (incorporated by reference, see § 430.3) with Addendum 1 and 2 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; or

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

(a) 25 feet of tubing, or

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

If they are needed to make a secondary measurement of capacity, install refrigerant pressure measuring instruments as described in section 8.2.5 of ASHRAE Standard 37–2009(incorporated by reference, see § 430.3). Refer to section 2.10 of this appendix to learn which secondary methods require refrigerant pressure measurements. At a minimum, insulate the low-pressure line(s) of a split system with insulation having an inside diameter that matches the refrigerant tubing and a nominal thickness of 0.5 inch.

b. For units designed for both horizontal and vertical installation or for both up-flow and down-flow vertical installations, the manufacturer must use the orientation for testing specified 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, 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). Except as noted in section 3.1.10, prevent the indoor air supplementary heating coils from operating during all tests. For coil-only indoor units that are supplied without an enclosure, create an enclosure using 1 inch fiberglass ductboard having a nominal density of 6 pounds per cubic foot. Or alternatively, use some other insulating material having a thermal resistance (''R'' value) between 4 and 6 hr·ft²· °F/Btu. For units where the coil is housed within an enclosure or cabinet, no extra insulating or sealing is allowed.

d. When testing oil-only central air conditioners and heat pumps, install a toroidal-type transformer to power the system's low-voltage components, complying with any additional requirements for this transformer mentioned in the installation manuals included with the unit by the 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 that results in the transformer being loaded at a level that is between 25 and 90 percent based on the highest power value expected and then confirmed during the off mode test;

(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. The power consumption of the components connected to the transformer must be included as part of the total system power consumption during the off mode tests, less if included the power consumed by the transformer when no load is connected to it.

e. An outdoor unit with no match (*i.e.,* that is not sold with indoor units) shall be tested without an indoor blower installed, with a single cooling air volume rate, using an indoor unit 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 1.15 square inches per British thermal unit per hour (sq. in./Btu/hr). NGIFS is calculated as follows:

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

Where,

L_f = Indoor coil fin length in inches, also height of the coil transverse to the tubes.

W_f = Indoor coil fin width in inches, also depth of the coil.

N_f = Number of fins.

$\dot{Q}_c(95)$ = the measured space cooling capacity of the tested outdoor unit/indoor unit combination as determined from the A2 or A Test whichever applies, Btu/h.

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 for information on region IV.) For heat pumps that use a time-adaptive defrost control system (see section 1.2, Definitions), the manufacturer must specify the frosting interval to be used during Frost Accumulation tests and provide the procedure for manually initiating the defrost at the specified time. To ease testing of any unit, the manufacturer should provide information and any necessary hardware to manually initiate a defrost cycle.

2.2.2 Special requirements for units having a multiple-speed outdoor fan.

Configure the multiple-speed outdoor fan according to the 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, systems composed of multiple single-zone-multiple-coil split-system units (having multiple outdoor units located side-by-side), and ducted systems using a single indoor section containing multiple 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 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 and systems composed of multiple single-zone-multiple-coil split-system units. For any test where the system is operated at part load (*i.e.,* one or more compressors ''off'', operating at the intermediate or minimum compressor speed, or at low compressor capacity), the manufacturer shall designate the indoor coil(s) that are not providing heating or cooling during the test such that the sum of the nominal heating or cooling capacity of the operational indoor units is within 5 percent of the intended part load heating or cooling capacity. For variable-speed systems, the manufacturer must designate 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 shall choose to turn off zero, one, two, or more indoor units. The chosen configuration shall remain unchanged for all tests conducted at the same compressor speed/capacity. For any indoor coil that is 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 systems with a single indoor section containing multiple blowers where the blowers are designed to cycle on and off independently of one another and are not controlled such that all blowers are modulated to always operate at the same air volume rate or speed. This Appendix covers systems with a single-speed compressor or systems offering two fixed stages of compressor capacity (*e.g.,* a two-speed compressor, two single-speed compressors). 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—blowers accounting for at least one-third of the full-load air volume rate must be turned off unless prevented by the controls of the unit. In such cases, turn off as many blowers as permitted by the unit's controls. Where more than one option exists for meeting this ''off'' blower requirement, the manufacturer shall include in its installation manuals included with the unit which 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 an ''off'' blower.

c. For test setups where it is physically impossible for the laboratory to use the required line length listed in Table 3 of ANSI/AHRI Standard 1230–2010 (incorporated by reference, see § 430.3) with Addendum 2, 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 ANSI/AHRI Standard 1230–2010 with Addendum 2 are applied.

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

2.2.4.1 Cooling mode tests.

For wet-coil cooling mode tests, regulate the water vapor content of the air entering the indoor unit to the applicable wet-bulb temperature listed in Tables 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. For dry coil tests on such units, it may be necessary to limit the moisture content of the air entering the outdoor side of the unit to meet the requirements of section 3.4.

2.2.4.2 Heating mode tests.

For heating mode tests, regulate the water vapor content of the air entering the outdoor unit to the applicable wet-bulb temperature listed in Tables 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 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 The ''manufacturer's published instructions,'' as stated in section 8.2 of ASHRAE Standard 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 are shipped with the unit shall take precedence over installation instructions that appear in the labels applied to the unit.

2.2.5.2 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 the refrigerant charge shall be adjusted per the outdoor installation instructions.

c. For systems consisting of an outdoor unit manufacturer's outdoor section and an independent coil manufacturer's indoor section with differing charging procedures the refrigerant charge shall be adjusted per the indoor installation instructions.

2.2.5.3 Test(s) to Use for Charging

a. Use the tests or operating conditions specified in the manufacturer's installation instructions for charging.

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 function in the H1 or $H1_2$ test with the charge set for the A or A_2 test and for heating-only heat pumps, use the H1 or $H1_2$ test.

2.2.5.4 Parameters to Set and Their Target Values

a. Consult the manufacturer's installation instructions regarding which parameters 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 (defined as 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.

2.2.5.5 Charging Tolerances

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

b. Otherwise, use the following tolerances for the different charging parameters:

1. Superheat: ±2.0 °F
2. Subcooling: ±0.6 °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

2.2.5.6 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 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 this test.

b. Single-Package Systems

Unless otherwise directed by the manufacturer's installation instructions, install one or more refrigerant line pressure gauges during the setup of the unit if setting of refrigerant charge is based on certain operating parameters:

(1) Install a pressure gauge 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 on the suction line if charging is on the basis of superheat, or low side pressure or corresponding saturation or dew point temperature. If manufacturer's installation instructions indicate that pressure gauges are not to be installed, setting of charge shall not be based on any of the parameters listed in b.(1) and (2) of this section.

2.2.5.7 Near-azeotropic and zeotropic refrigerants.

Charging of near-azeotropic and zeotropic refrigerants shall only be performed with refrigerant in the liquid state.

2.2.5.8 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.

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 installation instructions that are provided with the equipment while meeting the airflow requirements that are specified in section 3.1.4. If the manufacturer installation instructions do not provide guidance on the airflow-control settings for a system tested with the indoor blower installed, select the lowest speed that will satisfy the minimum external static pressure specified in section 3.1.4.1.1 with an air volume rate at or higher than the rated full-load cooling air volume rate while meeting the maximum air flow requirement.

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

2.3.2 Heating tests.

a. If needed, set the indoor blower airflow-control settings (*e.g.,* fan motor pin settings, fan motor speed) according to the installation instructions that are provided with the equipment. Do this set-up while meeting all applicable airflow requirements specified in sections 3.1.4. For a cooling and heating heat pump tested with an indoor blower installed, if the manufacturer installation instructions do not provide guidance on the fan airflow-control settings, use the same airflow-control settings used for the cooling test. If the manufacturer installation instructions do not provide guidance on the airflow-control settings for a heating-only heat pump tested with the indoor blower installed, select the lowest speed that will satisfy the minimum external static pressure specified in section 3.1.4.4.3 with an air volume rate at or higher than the rated heating full-load air volume rate.

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 described in section 2.4.1 and, if installed, the inlet plenum described in section 2.4.2 with thermal insulation having a nominal overall resistance (R-value) of at least 19 hr·ft²· °F/Btu.

2.4.1 Outlet plenum for the indoor unit.

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

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

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

d. Add a static pressure tap to each face of the (each) outlet plenum, if rectangular, or at four evenly distributed locations along the circumference of an oval or round plenum. Create a manifold that connects the four static pressure taps. Figures 7a, 7b, 7c of ASHRAE Standard 37–2009 (incorporated by reference, see § 430.3) shows two of the three options allowed for the manifold configuration; the third option is the broken-ring, four-to-one manifold configuration that is shown in Figure 7a of ASHRAE Standard 37–2009. See Figures 7a, 7b, 7c, and 8 of ASHRAE Standard 37–2009 for the cross-sectional dimensions and minimum length of the (each) plenum and the locations for adding the static pressure taps for units tested with and without an indoor blower installed.

TABLE 2—SIZE OF OUTLET PLENUM

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

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

2.4.2 Inlet plenum for the indoor unit.

Install an inlet plenum when testing a coil-only indoor unit or a packaged system where the indoor coil is located in the outdoor test room. Add static pressure taps at the center of each face of this plenum, if rectangular, or at four evenly distributed locations along the circumference of an oval or round plenum. Make a manifold that connects the four static-pressure taps using one of the three configurations specified in section 2.4.1. See Figures 7b, 7c, and Figure 8 of ASHRAE Standard 37–2009 (incorporated by reference, see § 430.3) for cross-sectional dimensions, the minimum length of the inlet plenum, and the locations of the static-pressure taps. When testing a ducted unit having an indoor blower (and the indoor coil is in the indoor test room), test with an inlet plenum installed unless physically prohibited by space limitations within the test room. If used, construct the inlet plenum and add the four static-pressure taps as shown in Figure 8 of ASHRAE Standard 37–2009. If used, the inlet duct size shall equal the size of the inlet opening of the air-handling (blower coil) unit or furnace, with a minimum length of 6 inches. Manifold the four static-pressure taps using one of the three configurations specified in section 2.4.1.d. Never use an inlet plenum when testing a non-ducted system.

2.5 Indoor coil air property measurements and air damper box applications.

Follow instructions for indoor coil air property measurements as described in AHRI 210/240-Draft, appendix E, section E4, 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 ASHRAE Standard 41.1–2013 (incorporated by reference, see § 430.3) for guidance on constructing an air sampling device. No part of the air sampling device or the tubing transferring the sampled air to the sensor shall be within two inches of the test chamber floor, and the transfer tubing shall be insulated. The sampling device may also divert air to a remotely located sensor(s) that measures dry bulb temperature. The air sampling device and the remotely located temperature sensor(s) may be used to determine the entering air dry bulb temperature during any test. The air sampling device and the remotely located leaving air dry bulb temperature sensor(s) may be used for all tests except:

(1) Cyclic tests; and

(2) Frost accumulation tests.

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

c. Use an inlet and outlet air damper box, an inlet upturned duct, or any combination thereof when conducting one or both of the cyclic tests listed in sections 3.2 and 3.6 on ducted systems. Otherwise if not conducting one or both of said cyclic tests, install an outlet air damper box when testing ducted and non-ducted heat pumps that cycle off the indoor blower during defrost cycles if no other means is available 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 a non-ducted system. An inlet upturned duct is a length of ductwork so 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 the variation of the dry bulb temperature at this location, measured at least every minute during the compressor OFF period of the cyclic test, does not exceed 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, 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; install a dry-bulb temperature sensor at a centerline location not higher than the lowest elevation of the duct edges at the device inlet.

2.5.1.1 If the section 2.4.2 inlet plenum is installed.

Construct the airflow prevention device having a cross-sectional flow area equal to or greater than the flow area of the inlet plenum. Install the airflow prevention device upstream of the inlet plenum and construct ductwork connecting it to the inlet plenum. If needed, use an adaptor plate or a transition duct section to connect the airflow prevention device with the inlet plenum. Insulate the ductwork and inlet plenum with thermal insulation that has a nominal overall resistance (R-value) of at least 19 hr · ft^2 · °F/Btu.

2.5.1.2 If the section 2.4.2 inlet plenum is not installed.

Construct the airflow prevention device having a cross-sectional flow area equal to or greater than the flow area of the air inlet of the indoor unit. Install the airflow prevention device immediately upstream of the inlet of the indoor unit. If needed, use an adaptor plate or a short transition duct section to connect the airflow prevention device with the unit's air inlet. Add static pressure taps at the center of each face of a rectangular airflow prevention device, or at four evenly distributed locations along the circumference of an oval or round airflow prevention device. Locate the pressure taps between the airflow prevention device and the inlet of the indoor unit. Make a manifold that connects the four static pressure taps. Insulate the ductwork with thermal insulation that has a nominal overall resistance (R-value) of at least 19 hr·ft^2 · °F/Btu.

2.5.2 Test set-up on the inlet side of the indoor unit: For cases where no airflow prevention device is installed.

If using the section 2.4.2 inlet plenum and a grid of dry bulb temperature sensors, mount the grid at a location upstream of the static pressure taps described in section 2.4.2, preferably at the entrance plane of the inlet plenum. If the section 2.4.2 inlet plenum is not used, but a grid of dry bulb temperature sensors is used, locate the grid approximately 6 inches upstream from the inlet of the indoor coil. Or, in the case of non-ducted units having multiple indoor coils, locate a grid approximately 6 inches upstream from the inlet of each indoor coil. Position an air sampling device, or the sensor used to measure the water vapor content of the inlet air, immediately upstream of the (each) entering air dry-bulb temperature sensor grid. If a grid of sensors is not used, position the entering air sampling device (or the sensor used to measure the water vapor content of the inlet air) as if the grid were present.

2.5.3 Indoor coil static pressure difference measurement.

Section 6.5.2 of ASHRAE Standard 37–2009 describes the method for fabricating static-pressure taps. Also refer to Figure 2A of ASHRAE Standard 51–07/AMCA Standard 210–07 (incorporated by reference, see § 430.3). 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. If an inlet plenum or inlet airflow prevention device is not used, leave the inlet side of the differential pressure instrument open to the surrounding atmosphere. For non-ducted systems that are tested with multiple outlet plenums, measure the static pressure within each outlet plenum relative to the surrounding atmosphere.

2.5.4 Test set-up on the outlet side of the indoor coil.

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

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

(1) Downstream of the air sampling device;

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

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

2.5.4.1 Outlet air damper box placement and requirements.

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

2.5.4.2 Procedures to minimize temperature maldistribution.

Use these procedures if necessary to correct temperature maldistributions. Install a mixing device(s) upstream of the outlet air, dry-bulb temperature grid (but downstream of the outlet plenum static pressure taps). Use a perforated screen located between the mixing device and the dry-bulb temperature grid, with a maximum open area of 40 percent. One or both items should help to meet the maximum outlet air temperature distribution specified in section 3.1.8. Mixing devices are described in sections 5.3.2 and 5.3.3 of ASHRAE Standard 41.1–2013 (incorporated by reference, see § 430.3) and section 5.2.2 of ASHRAE Standard 41.2–87 (RA 92) (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. 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 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, 7.2, and 7.3 of ASHRAE Standard 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, 7.4, and 7.5 of ASHRAE Standard 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, and 7.1 of ASHRAE Standard 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), the air damper box(es) must be capable of being completely opened or completely closed within 10 seconds for each action.

2.6 Airflow measuring apparatus.

a. Fabricate and operate an Air Flow Measuring Apparatus as specified in section 6.2 and 6.3 of ASHRAE Standard 37–2009. Refer to Figure 12 of ASHRAE Standard 51–07/AMCA Standard 210–07 or Figure 14 of ASHRAE Standard 41.2–87 (RA 92) (incorporated by reference, see § 430.3) for guidance on placing the static pressure taps and positioning the diffusion baffle (settling means) relative to the chamber inlet. When measuring the static pressure difference across nozzles and/or velocity pressure at nozzle throats using electronic pressure transducers and a data acquisition system, if high frequency fluctuations cause measurement variations to exceed the test tolerance limits specified in section 9.2 and Table 2 of ASHRAE Standard 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. See sections 6.1.1, 6.1.2, and 6.1.4, and Figures 1, 2, and 4 of ASHRAE Standard 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 ASHRAE Standard 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 with Addendum 1 and 2 for ''Standard Rating Tests.'' If the voltage on the nameplate of indoor and outdoor units differs, the voltage supply on the outdoor unit shall be selected for testing. Measure the supply voltage at the terminals on the test unit using a volt meter that provides a reading that is accurate to within ±1.0 percent of the measured quantity.

2.8 Electrical power and energy measurements.

a. Use an integrating power (watt-hour) measuring system to determine the electrical energy or average electrical power supplied to all components of the air conditioner or heat pump (including auxiliary components such as controls, transformers, crankcase heater, integral condensate pump on non-ducted indoor units, etc.). The watt-hour measuring system must give readings that are accurate to within ±0.5 percent. For cyclic tests, this accuracy is required during both the ON and OFF cycles. Use either two different scales on the same watt-hour meter or two separate watt-hour meters. Activate the scale or meter having the lower power rating within 15 seconds after beginning an OFF cycle. Activate the scale or meter having the higher power rating active within 15 seconds prior to beginning an ON cycle. For ducted units tested with a fan installed, the ON cycle lasts from compressor ON to indoor blower OFF. For ducted units tested without an indoor blower installed, 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, and/or 3.10, this same instrumentation requirement applies when testing air conditioners and heat pumps having a variable-speed constant-air-volume-rate indoor 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. In addition, for all steady-state tests, conduct a second, independent measurement of capacity as described in section 3.1.1. For split systems, use one of the following secondary measurement methods: Outdoor Air Enthalpy Method, Compressor Calibration Method, or Refrigerant Enthalpy Method. For single-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),

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

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

(4) On the inlet side, a sampling device and temperature grid (section 2.11b.).

c. During the preliminary tests described in sections 3.11.1 and 3.11.1.1, measure the evaporator and condenser temperatures or pressures. On both the outdoor coil and the indoor coil, solder a thermocouple onto a return bend located at or near the midpoint of each coil or at points not affected by vapor superheat or liquid subcooling. Alternatively, if the test unit is not sensitive to the refrigerant charge, 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 Standard 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, measure refrigerant properties, and adjust the refrigerant charge according to section 7.4.2 and 8.2.5 of ASHRAE Standard 37–2009 (incorporated by reference, see § 430.3). Use refrigerant temperature and pressure measuring instruments that meet the specifications given in sections 5.1.1 and 5.2 of ASHRAE Standard 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 ASHRAE Standard 37–2009 for the requirements for this method, including the additional instrumentation requirements, and information on placing the flow meter and a sight glass. Use refrigerant temperature, pressure, and flow measuring instruments that meet the specifications given in sections 5.1.1, 5.2, and 5.5.1 of ASHRAE Standard 37–2009. Refrigerant flow measurement device(s), if used, must be 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, unless the device(s) are elevated at least two feet from the floor.

2.11 Measurement of test room ambient conditions.

Follow instructions for measurement of test room ambient conditions as described in AHRI 210/240-Draft, appendix E, section E4, (incorporated by reference, see § 430.3) 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 ASHRAE Standard 37–2009), add instrumentation to permit measurement of the indoor test room dry-bulb temperature.

b. For the outdoor side, install a grid of evenly-distributed sensors on every air-permitting face on the inlet of the outdoor unit, such that each measurement represents an air-inlet area of no more than one square foot. This grid must be constructed and applied as per section 5.3 of ASHRAE Standard 41.1–2013 (incorporated by reference, see § 430.3). The maximum and minimum temperatures measured by these sensors may differ by no more than 1.5 °F—otherwise adjustments to the test room must be made to improve temperature uniformity. The outdoor conditions shall be verified with the air collected by air sampling device. Air collected by an air sampling device at the air inlet of the outdoor unit for transfer to sensors for measurement of temperature and/or humidity shall be protected from temperature change as follows: Any surface of the air conveying tubing in contact with surrounding air at a different temperature than the sampled air shall be insulated with thermal insulation with a nominal thermal resistance (R-value) of at least 19 hr · ft² · °F/Btu, no part of the air sampling device or the tubing conducting the sampled air to the sensors shall be within two inches of the test chamber floor, and pairs of measurements (*e.g.* dry bulb temperature and wet bulb temperature) used to determine water vapor content of sampled air shall be measured in the same location. Take steps (*e.g.,* add or re-position a lab circulating fan), as needed, to maximize temperature uniformity within the outdoor test room. However, ensure that any fan used for this purpose does not cause air velocities in the vicinity of the test unit to exceed 500 feet per minute.

c. Measure dry bulb temperatures as specified in sections 4, 5, 7.2, 6, and 7.3 of ASHRAE Standard 41.1–2013. Measure water vapor content as stated above in section 2.5.6.

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 ASHRAE Standard 37–2009.

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 to determine indoor space conditioning capacity. Calculate this secondary check of capacity according to section 3.11. The two capacity measurements must agree to within 6 percent to constitute a valid test. For this capacity comparison, use the Indoor Air Enthalpy Method capacity that is calculated in section 7.3 of ASHRAE Standard 37–2009 (and, if testing a coil-only system, do not make the after-test fan heat adjustments described in section 3.3, 3.4, 3.7, and 3.10 of this appendix). However, include the appropriate section 3.3 to 3.5 and 3.7 to 3.10 fan heat adjustments within the Indoor Air Enthalpy Method capacities used for the section 4 seasonal calculations.

3.1.2 Manufacturer-provided equipment overrides.

Where needed, the manufacturer must provide a means for overriding the controls of the test unit so that the compressor(s) operates at the specified speed or capacity and the indoor 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) with Addendum 1 and 2 when obtaining the airflow through the outdoor coil.

3.1.3.1 Double-ducted.

For products intended to be installed with the outdoor airflow ducted, the unit shall be installed with outdoor coil ductwork installed per manufacturer installation instructions and shall operate between 0.10 and 0.15 in H_2O external static pressure. External static pressure measurements shall be made in accordance with ASHRAE Standard 37–2009 Section 6.4 and 6.5.

3.1.4 Airflow through the indoor coil.

Airflow setting(s) shall be determined before testing begins. Unless otherwise specified within this or its subsections, no changes shall be made to the airflow setting(s) after initiation of testing.

3.1.4.1 Cooling Full-load Air Volume Rate.

3.1.4.1.1 Cooling Full-Load Air Volume Rate for Ducted Units.

The manufacturer must specify the cooling full-load air volume rate and the instructions for setting fan speed or controls. Adjust the cooling full-load air volume rate if needed to satisfy the additional requirements of this section. First, when conducting the A or A_2 Test (exclusively), the measured air volume rate, when divided by the measured indoor air-side total cooling capacity must not exceed 37.5 cubic feet per minute of standard air (scfm) per 1000 Btu/h. If this ratio is exceeded, reduce the air volume rate until this ratio is equaled. Use this reduced air volume rate for all tests that call for using the Cooling Full-load Air Volume Rate. Pressure requirements are as follows:

a. For all ducted units tested with an indoor blower installed, except those having a constant-air-volume-rate indoor blower:

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

2. Measure the external static pressure;

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

4. If the Table 3 minimum is not equaled or exceeded,

4a. reduce the air volume rate and increase the external static pressure by adjusting the exhaust fan of the airflow measuring apparatus until the applicable Table 3 minimum is equaled or

4b. until the measured air volume rate equals 90 percent of the air volume rate from step 1, whichever occurs first.

5. If the conditions of step 4a occur first, the pressure requirement is satisfied. Use the step 4a reduced air volume rate for all tests that require the Cooling Full-load Air Volume Rate.

6. If the conditions of step 4b occur first, make an incremental change to the set-up of the indoor blower (*e.g.,* next highest fan motor pin setting, next highest fan motor speed) and repeat the evaluation process beginning at above step 1. If the indoor blower set-up cannot be further changed, reduce the air volume rate and increase the external static pressure by adjusting the exhaust fan of the airflow measuring apparatus until the applicable Table 3 minimum is equaled. Use this reduced air volume rate for all tests that require the Cooling Full-load Air Volume Rate.

b. For ducted units that are tested with a constant-air-volume-rate indoor blower installed. For all tests that specify the Cooling Full-load Air Volume Rate, obtain an external static pressure as close to (but not less than) the applicable Table 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 ducted units that are tested without an indoor blower installed. For the A or A_2 Test, (exclusively), the pressure drop across the indoor coil assembly must not exceed 0.30 inches of water. If this pressure drop is exceeded, reduce the air volume rate until the measured pressure drop equals the specified maximum. Use this reduced air volume rate for all tests that require the Cooling Full-load Air Volume Rate.

TABLE 3—MINIMUM EXTERNAL STATIC PRESSURE FOR DUCTED SYSTEMS TESTED WITH AN INDOOR BLOWER INSTALLED

Rated Cooling [1] or heating [2] capacity (Btu/h)	Minimum external static pressure [3] (Inches of water)		
	Short ducted systems [6]	Small-duct, high-velocity systems [4] [5]	All other systems
≤28,800	0.03	1.10	0.45
≥29,000 and ≤42,500	0.05	1.15	0.50
≥43,000	0.07	1.20	0.55

[1] For air conditioners and heat pumps, the value cited by the manufacturer in published literature for the unit's capacity when operated at the *A* or A_2 Test conditions.
[2] For heating-only heat pumps, the value the manufacturer cites in published literature for the unit's capacity when operated at the *H1* or $H1_2$ Test conditions.
[3] For ducted units tested without an air filter installed, increase the applicable tabular value by 0.08 inches of water. For ducted units for which the indoor blower installed for testing is the fan of a condensing gas furnace, decrease the applicable tabular value by 0.10 inches of water (make both adjustments if they both apply). If the adjusted value is less than zero, readjust it to zero.
[4] See section 1.2, Definitions, to determine if the equipment qualifies as a small-duct, high-velocity system.
[5] If a closed-loop, air-enthalpy test apparatus is used on the indoor side, limit the resistance to airflow on the inlet side of the indoor blower coil to a maximum value of 0.1 inch of water. Impose the balance of the airflow resistance on the outlet side of the indoor blower.
[6] See section 1.2. Definitions.

d. For ducted systems having multiple indoor blowers within a single indoor section, obtain the full-load air volume rate with all blowers operating unless prevented by the controls of the unit. In such cases, turn on the maximum number of blowers permitted by the unit's controls. Where more than one option exists for meeting this ''on'' blower requirement, which blower(s) are turned on must match that specified by the manufacturer in the installation manuals included with the unit. Conduct section 3.1.4.1.1 setup steps for each blower separately. If two or more indoor blowers are connected to a common duct as per section 2.4.1, either turn off the other indoor blowers connected to the same common duct or temporarily divert their air volume to the test room when confirming or adjusting the setup configuration of individual blowers. If the indoor blowers are all the same size or model, the target air volume rate for each blower plenum equals the full-load air volume rate divided by the number of ''on'' blowers. If different size blowers are used within the indoor section, the allocation of the system's full-load air volume rate assigned to each ''on'' blower must match that specified by the manufacturer in the installation manuals included with the unit.

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.

The manufacturer must specify the cooling minimum air volume rate and the instructions for setting fan speed or controls. 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.

a. For ducted units tested with an indoor blower installed that is not a constant-air-volume indoor blower, adjust for external static pressure as follows.

1. Achieve the manufacturer-specified cooling minimum air volume rate;

2. Measure the external static pressure;

3. If this pressure is equal to or greater than the target minimum external static pressure calculated as described above, use the current air volume rate for all tests that require the cooling minimum air volume rate.

4. If the target minimum is not equaled or exceeded,

4a. reduce the air volume rate and increase the external static pressure by adjusting the exhaust fan of the airflow measuring apparatus until the applicable target minimum is equaled or

4b. until the measured air volume rate equals 90 percent of the air volume rate from step 1, whichever occurs first.

5. If the conditions of step 4a occur first, use the step 4a reduced air volume rate for all tests that require the cooling minimum air volume rate.

6. If the conditions of step 4b occur first, make an incremental change to the set-up of the indoor fan (*e.g.,* next highest fan motor pin setting, next highest fan motor speed) and repeat the evaluation process beginning at above step 1. If the indoor fan set-up cannot be further changed, reduce the air volume rate and increase the external static pressure by adjusting the exhaust fan of the airflow measuring apparatus until the applicable target minimum is equaled. Use this reduced air volume rate for all tests that require 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, 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 units that are tested without an indoor blower installed, 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) unit, obtain this Cooling Minimum Air Volume Rate regardless of the pressure drop across the indoor coil assembly.

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

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 for the minimum number of blowers that must be turned off. Adjust for external static pressure and if necessary adjust air volume rates as described in section 3.1.4.2.a if the indoor fan is not a constant-air-volume indoor fan or as described in section 3.1.4.2.b if the indoor fan is a constant-air-volume indoor fan. The sum of the individual ''on'' blowers' air volume rates is the cooling minimum air volume rate for the system.

3.1.4.3 Cooling Intermediate Air Volume Rate.

The manufacturer must specify the cooling intermediate 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.

a. For ducted units tested with an indoor blower, installed that is not a constant-air-volume indoor blower, adjust for external static pressure as described in section 3.1.4.2.a for cooling minimum air volume rate.

b. For ducted units tested with constant-air-volume indoor blowers installed, 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, 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 heat pumps tested with an indoor blower installed that is not a constant-air-volume indoor blower that operates at the same airflow-control setting during both the A (or A_2) and the H1 (or $H1_2$) Tests;

2. Ducted heat pumps tested with constant-air-flow indoor blowers installed that provide the same air flow for the A (or A_2) and the H1 (or $H1_2$) Tests; and

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

b. For heat pumps that meet the above criteria ''1'' and ''3,'' no minimum requirements apply to the measured external or internal, respectively, static pressure. For heat pumps that meet the above criterion ''2,'' test at an external static pressure that does not cause 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 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 indoor blower operation.

The manufacturer must specify the heating full-load 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.

a. For ducted heat pumps tested with an indoor blower installed that is not a constant-air-volume indoor blower, adjust for external static pressure as described in section 3.1.4.2.a 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, 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 northern heat pumps (see section 1.2, Definitions), use the appropriate approach of the above two cases for units that are tested with an indoor blower installed. For coil-only 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'' blowers as used for the cooling full-load air volume rate. For systems where individual blowers regulate the speed (as opposed to the cfm) of the indoor blower, use the first section 3.1.4.2 equation for each blower individually. Sum the individual blower air volume rates to obtain the heating full-load air volume rate for the system.

3.1.4.4.3 Ducted heating-only heat pumps.

The manufacturer must specify the Heating Full-load Air Volume Rate.

a. For all ducted heating-only heat pumps tested with an indoor blower installed, 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.

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

2. Measure the external static pressure.

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

4. If the Table 3 minimum is not equaled or exceeded,

4a. reduce the air volume rate and increase the external static pressure by adjusting the exhaust fan of the airflow measuring apparatus until the applicable Table 3 minimum is equaled or

4b. until the measured air volume rate equals 90 percent of the manufacturer-specified Full-load Air Volume Rate, whichever occurs first.

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

6. If the conditions of step 4b occur first, make an incremental change to the set-up of the indoor blower (*e.g.,* next highest fan motor pin setting, next highest fan motor speed) and repeat the evaluation process beginning at above step 1. If the indoor blower set-up cannot be further changed, reduce the air volume rate until the applicable Table 3 minimum is equaled. Use this reduced air volume rate for all tests that require the Heating Full-load Air Volume Rate.

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

3.1.4.4.4 Non-ducted heat pumps, including non-ducted heating-only heat pumps.

For non-ducted heat pumps, the Heating Full-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 heat pumps tested with an indoor blower installed that is not a constant-air-volume indoor blower that operates at the same airflow-control setting during both the A_1 and the $H1_1$ tests; 2. Ducted heat pumps tested with constant-air-flow indoor blowers installed that provide the same air flow for the A_1 and the $H1_1$ Tests; and

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

b. For heat pumps that meet the above criteria ''1'' and ''3,'' no minimum requirements apply to the measured external or internal, respectively, static pressure. For heat pumps that meet the above criterion ''2,'' test at an external static pressure that does not cause 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.

The manufacturer must specify the heating minimum 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.

a. For ducted heat pumps tested with an indoor blower installed that is not a constant-air-volume indoor blower, adjust for external static pressure as described in section 3.1.4.2.a 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 thanor 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 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 northern heat pumps that are tested with an indoor blower installed, use the appropriate approach of the above two cases.

d. For ducted two-capacity heat pumps that are tested without an indoor blower installed, use the Cooling Minimum Air Volume Rate as the Heating Minimum Air Volume Rate. For ducted two-capacity northern heat pumps that are tested without an indoor blower installed, use the Cooling Full-load Air Volume Rate as the Heating Minimum Air Volume Rate. For ducted two-capacity heating-only heat pumps that are tested without an indoor blower installed, 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'' blowers as used for the cooling minimum air volume rate. For systems where individual blowers regulate the speed (as opposed to the cfm) of the indoor blower, use the first section 3.1.4.5 equation for each blower individually. Sum the individual blower air volume rates to obtain the heating minimum air volume rate for the system.

3.1.4.6 Heating Intermediate Air Volume Rate.

The manufacturer must specify the heating intermediate 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.

a. For ducted heat pumps tested with an indoor blower installed that is not a constant-air-volume indoor blower, adjust for external static pressure as described in section 3.1.4.2.a 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, 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. Make adjustments as described in section 3.14.6 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 ASHRAE Standard 37–2009), maintain the dry bulb temperature within the test room within ±5.0 °F of the applicable sections 3.2 and 3.6 dry bulb temperature test condition for the air entering the indoor unit. Dew point shall be within 2 °F of the required inlet conditions.

3.1.6 Air volume rate calculations.

For all steady-state tests and for Frost Accumulation (H2, $H2_1$, $H2_2$, $H2_V$) tests, calculate the air volume rate through the indoor coil as specified in sections 7.7.2.1 and 7.7.2.2 of ASHRAE Standard 37–2009. When using the Outdoor Air Enthalpy Method, follow sections 7.7.2.1 and 7.7.2.2 to calculate the air volume rate through the outdoor coil. To express air volume rates in terms of standard air, use:

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 ASHRAE Standard 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.

Manufacturers may optionally operate the equipment under test for a ‘‘break-in’’ period, not to exceed 20 hours, prior to conducting the test method specified in this section. A manufacturer who elects to use this optional compressor break-in period in its certification testing should record this information (including the duration) in the test data underlying the certified ratings that are required to be maintained under 10 CFR 429.71. 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 an 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. For the 30-minute data collection interval used to determine capacity, the maximum spread among the outlet dry bulb temperatures from any data sampling must not exceed 1.5 °F. Install the mixing devices described in section 2.5.4.2 to minimize the temperature spread.

3.1.9 Requirement for the air temperature distribution entering the outdoor coil.

Monitor the temperatures of the air entering the outdoor coil using the grid of temperature sensors described in section 2.11. For the 30-minute data collection interval used to determine capacity, the maximum difference between dry bulb temperatures measured at any of these locations must not exceed 1.5 °F.

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, the short test follows the H1 or, if conducted, the H1C Test. For two-capacity heat pumps and heat pumps covered under section 3.6.2, the short test follows the $H1_2$ Test. Set the heat comfort controller to provide the maximum supply air temperature. With the heat pump operating and while maintaining the Heating Full-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 unit having a single-speed compressor, or a system comprised of independently circuited single-speed compressors, that is tested with a fixed-speed indoor blower installed, with a constant-air-volume-rate indoor blower installed, or with no indoor blower installed.

Conduct two steady-state wet coil tests, the A and B Tests. Use the two 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 testing outdoor units of central air conditioners or heat pumps that are not sold with indoor units, assign C_D^c the default value of 0.2. 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-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)		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—required (steady, dry coil)	80	([3])	82		Cooling full-load.[2]
D Test—required (cyclic, dry coil)	80	([3])	82		([4]).

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

3.2.2 Tests for a unit having a single-speed compressor where the indoor section uses a single variable-speed variable-air-volume rate indoor blower or multiple 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 blowers.

Conduct four steady-state wet coil tests: The A_2, A_1, B_2, and B_1 Tests. Use the two 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.

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 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]—required (steady, dry coil)	80	([4])	82		Cooling minimum.[3]
D_1 Test[4]—required (cyclic, dry coil)	80	([4])	82	([5])	

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

3.2.3 Tests for a unit having a two-capacity compressor. (see section 1.2, Definitions)

a. Conduct four steady-state wet coil tests: The A_2, B_2, B_1, and F_1 Tests. Use the two 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. 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, 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 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, $C_D^c(k{=}2)$. The default $C_O^c(k{=}2)$ 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—required (steady, dry-coil)	80	([4])	82	High	Cooling Full-Load.[2]	
D_2 Test—required (cyclic, dry-coil)	80	([4])	82	High	([5])	
C_1 Test—required (steady, dry-coil)	80	([4])	82	Low	Cooling Minimum.[3]	
D_1 Test—required (cyclic, dry-coil)	80	([4])	82	Low	([6])	
F_1 Test—required (steady, wet coil)	80	67	67	[1] 53.5	Low	Cooling Minimum.[3]

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

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

a. Conduct five steady-state wet coil tests: The A_2, E_V, B_2, B_1, and F_1 Tests. Use the two dry-coil tests, the steady-state G_1 Test and the cyclic I_1 Test, to determine the cooling mode cyclic degradation coefficient, $C_D{}^c$..-Table-7 specifies test conditions for these seven tests. Determine the intermediate compressor speed cited in Table 7 using:

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

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

b. For units that modulate the indoor 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, at least one indoor unit must be turned off. The manufacturer shall designate the particular indoor unit(s) that is turned off. The manufacturer must also specify the compressor speed used for the Table 7 E_V Test, a cooling-mode intermediate compressor speed that falls within ¼ and ¾ of the difference between the maximum and minimum cooling-mode speeds. The manufacturer should prescribe an intermediate speed that is expected to yield the highest EER for the given E_V Test conditions and bracketed compressor speed range. The manufacturer can designate that one or more indoor units are turned off for the E_V Test.

TABLE 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	Maximum	Cooling Full-Load.[2]
B_2 Test—required (steady, wet coil)	80	67	82	[1] 65	Maximum	Cooling Full-Load.[2]
E_V Test—required (steady, wet coil)	80	67	87	[1] 69	Intermediate	Cooling Intermediate.[3]
B_1 Test—required (steady, wet coil)	80	67	82	[1] 65	Minimum	Cooling Minimum.[4]
F_1 Test—required (steady, wet coil)	80	67	67	[1] 53.5	Minimum	Cooling Minimum.[4]
G_1 Test [5]—required (steady, dry-coil)	80	([6])	67	Minimum	Cooling Minimum.[4].	
I_1 Test [5]—required (cyclic, dry-coil)	80	([6])	67	Minimum	([6]).	

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

3.2.5 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 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 blowers and offering two stages of compressor modulation.

Conduct the cooling mode tests specified in section 3.2.3.

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, 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 cases where its control is required, the water vapor content of the air entering the outdoor coil.

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

b. After satisfying the pretest equilibrium requirements, make the measurements specified in Table 3 of ASHRAE Standard 37–2009 (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 reaching a 30-minute period (*e.g.,* four consecutive 10-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 ASHRAE Standard 37–2009. 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 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 $\dot{Q}ck(T)$, $\dot{Q}sck(T)$ and $\dot{E}ck(T)$, 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 maximum speed, k=1 to denote low capacity or minimum speed, and k=v to denote the intermediate speed.

d. For units tested without an indoor blower installed, 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:		
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	0.12	[5] 0.02
Electrical voltage, % of rdg.	2.0	1.5
Nozzle pressure drop, % of rdg.	8.0	

[1] See section 1.2, 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 for wet coil tests. Prior to recording data during the steady-state dry coil test, operate the unit at least one hour after achieving dry coil conditions. Drain the drain pan and plug the drain opening. Thereafter, the drain pan should remain completely dry.

b. Denote the resulting total space cooling capacity and electrical power derived from the test as $\dot{Q}_{ss,dry}$ and $\dot{E}_{ss,dry}$. With regard to a section 3.3 deviation, do not adjust $\dot{Q}_{ss,dry}$ for duct losses (*i.e.,* do not apply section 7.3.3.3 of ASHRAE Standard 37–2009). In preparing for the section 3.5 cyclic tests, record the average indoor-side air volume rate, $\dot{V}$, specific heat of the air, Cp,a (expressed on dry air basis), specific volume of the air at the nozzles, v'_n, humidity ratio at the nozzles, W_n, and either pressure difference or velocity pressure for the flow nozzles. For units having a variable-speed indoor fan (that provides either a constant or variable air volume rate) that will or may be tested during the cyclic dry coil cooling mode test with the indoor fan turned off (see section 3.5), include the electrical power used by the indoor fan motor among the recorded parameters from the 30-minute test.

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).

a. After completing the steady-state dry-coil test, remove the Outdoor Air Enthalpy method test apparatus, if connected, and begin manual OFF/ON cycling of the unit's compressor. The test set-up should otherwise be identical to the set-up used during the steady-state dry coil test. When testing heat pumps, leave the reversing valve during the compressor OFF cycles in the same position as used for the compressor ON cycles, unless automatically changed by the controls of the unit. For units having a variable-speed indoor 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.

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 specify airflow requirements through the indoor coil of ducted and non-ducted systems, respectively. In all cases, use the exhaust fan of the airflow measuring apparatus (covered under section 2.6) along with the indoor 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 blower, temporarily remove the blower.

e. Conduct a minimum of six complete compressor OFF/ON cycles for a unit with a single-speed or two-speed compressor, and a minimum of five complete compressor OFF/ON cycles for a unit with a variable speed compressor. The first three cycles for a unit with a single-speed compressor or two-speed compressor and the first two cycles for a unit with a unit with a variable speed compressor are the warm-up period—the later cycles are called the active cycles. Calculate the degradation coefficient C_D for each complete active cycle if the test tolerances given in Table 9 are satisfied. If the average C_D for the first three active cycles is within 0.02 of the average C_D for the first two active cycles, use the average C_D of the three active cycles as the final result. If these averages differ by more than 0.02, continue the test to get C_D for the fourth cycle. If the average C_D of the last three cycles is lower than or no more than 0.02 greater than the average C_D of the first three cycles, use the average C_D of all four active cycles as the final result. Otherwise, continue the test with a fifth cycle. If the average C_D of the last three cycles is 0.02 higher than the average for the previous three cycles, use the default C_D, otherwise use the average C_D of all five active cycles. If the test tolerances given in Table 9 are not satisfied, use default C_D value. The default C_D value for cooling is 0.2.

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 units tested with an indoor blower installed and operating, 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	0.12	....
Airflow nozzle pressure difference or velocity pressure,[2] % of reading	8.0	[4] 2.0
Electrical voltage,[5] % of rdg	2.0	1.5

[1] See section 1.2, Definitions.

[2] Applies during the interval that air flows through the indoor (outdoor) coil except for the first 30 seconds after flow initiation. For units having a variable-speed indoor blower that ramps, the tolerances listed for the external resistance to airflow apply from 30 seconds after achieving full speed until ramp down begins.

[3] Shall at no time exceed a wet-bulb temperature that results in condensate forming on the indoor coil.

[4] The test condition shall be the average nozzle pressure difference or velocity pressure measured during the steady-state dry coil test.

[5] Applies during the interval when at least one of the following—the compressor, the outdoor fan, or, if applicable, the indoor blower—are operating except for the first 30 seconds after compressor start-up.

i. 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,

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

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

$T_{a1}(\tau)$ = dry bulb temperature of the air entering the indoor coil at time τ, °F.

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

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

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

3.5.1 Procedures when testing ducted systems.

The automatic controls that are normally installed with the test unit must govern the OFF/ON cycling of the air moving equipment on the indoor side (exhaust fan of the airflow measuring apparatus and, if installed, the indoor blower of the test unit). For example, for ducted units tested without an indoor blower installed but rated based on using a fan time delay relay, control the indoor coil airflow according to the rated ON and/or OFF delays provided by the relay. For ducted units having a variable-speed indoor 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 units tested without an indoor blower installed, cycle the indoor coil airflow in unison with the cycling of the compressor. If air damper boxes are used, close them on the inlet and outlet side during the OFF period. Airflow through the indoor coil should stop within 3 seconds after the automatic controls of the test unit (act to) de-energize the indoor blower. For ducted units tested without an indoor blower installed (excluding the special case where a variable-speed fan is temporarily removed), increase $e_{cyc,dry}$ by the quantity,

Equation 3.5–2 $(441 W \div 1000 scfm) * \bar{\dot{V}} * [\tau_2 - \tau_1]$

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

Equation 3.5–3 $(1505\ Btu/h \div 1000 scfm) * \bar{\dot{V}} * [\tau_2 - \tau_1]$

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 systems.

Do not use airflow prevention devices when conducting cyclic tests on non-ducted units. Until the last OFF/ON compressor cycle, airflow through the indoor coil must cycle off and on in unison with the compressor. For the last OFF/ON compressor cycle—the one used to determine $e_{cyc,dry}$ and $q_{cyc,dry}$—use the exhaust fan of the airflow measuring apparatus and the indoor 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 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 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. 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

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

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

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

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

Conduct the High Temperature Cyclic (H1C) Test to determine the heating mode cyclic-degradation coefficient, C_D^h. Test conditions for the four tests are specified in Table 10.

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]
H1C Test (required, 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.
[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 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 High Temperature Cyclic ($H1C_1$) Test to determine the heating mode cyclic-degradation coefficient, $C_D{}^h$. 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:

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

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

where,

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

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

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

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

3.6.3 Tests for a heat pump having a two-capacity compressor (see section 1.2, Definitions), including two-capacity, northern heat pumps (see section 1.2, 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 seasonal performance calculations; and

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

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

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

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

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

b. Conduct the High Temperature Cyclic Test ($H1C_1$) to determine the heating mode cyclic-degradation coefficient, C_D^h. 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). 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 (required,[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 (required, 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.
[2] Defined in section 3.1.4.4.
[3] Maintain the airflow nozzle(s) static pressure difference or velocity pressure during the ON period at the same pressure or velocity as measured during the *$H1_2$* Test.
[4] Maintain the airflow nozzle(s) static pressure difference or velocity pressure during the ON period at the same pressure or velocity as measured during the *$H1_1$* Test.
[5] Required only if the heat pump's performance when operating at low compressor capacity and outdoor temperatures less than 37 °F is needed to complete the section 4.2.3 *HSPF* calculations.
[6] If table note #5 applies, the section 3.6.3 equations for $\dot{Q}_h^{k=1}(35)$ and $\dot{E}_h^{k=1}(17)$ may be used in lieu of conducting the $H2_1$ Test.
[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. (1) Conduct one Maximum Temperature Test ($H0_1$), two High Temperature Tests ($H1_2$ and $H1_1$), one Frost Accumulation Test ($H2_V$), and one Low Temperature Test ($H3_2$). Conducting one or all of the following tests is optional: An additional High Temperature Test ($H1_N$), an additional Frost Accumulation Test ($H2_2$), and an additional Low Temperature Test ($H4_2$). Conduct the High Temperature Cyclic ($H1C_1$) Test to determine the heating mode cyclic-degradation coefficient, C_D^h. (2) The optional low ambient temperature test ($H4_2$) may be conducted in place of $H1_2$ to allow representation of heating performance below 17 °F ambient temperature using the results of $H4_2$ and $H3_2$ rather than the results of $H3_2$ and $H1_2$. This option may not be used for units which have a cutoff temperature preventing compressor operation below 12 °F. If $H4_2$ is conducted, it is optional to conduct the $H1_2$ test for heating capacity rating purposes—$H1_N$ can be conducted for heating capacity rating purposes. If $H1_2$ is not conducted, $H2_2$ must be conducted.

Test conditions for the nine tests are specified in Table 13. Determine the intermediate compressor speed cited in Table 13 using the heating mode maximum and minimum compressors speeds and:

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

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

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

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

b. Determine the quantities $\dot{Q}_h^{k=2}(47)$ and from $\dot{E}_h^{k=2}(47)$ from the $H1_2$ Test and evaluate them according to section 3.7. Determine the quantities $\dot{Q}_h^{k=2}(17)$ and $\dot{E}_h^{k=2}(17)$ from the $H3_2$ Test and evaluate them according to section 3.10. Determine the quantities $\dot{Q}_h^{k=2}(T_L)$ and $\dot{E}_h^{k=2}(T_L)$ from the $H4_2$ Test and evaluate them according to section 3.10. For heat pumps where the heating mode maximum compressor speed exceeds its cooling mode maximum compressor speed, conduct the $H1_N$ Test if the manufacturer requests it. If the $H1_N$ Test is done, operate the heat pump's compressor at the same speed as the speed used for the cooling mode A_2 Test.

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	Minimum	Heating Minimum.[1]
$H1C_1$ Test (required, cyclic)	70	60 (max)	47	43	Minimum	([2]).
$H1_2$ Test (required, steady)	70	60 (max)	47	43	Maximum	Heating Full-Load.[3]
$H1_1$ Test (required, steady)	70	60 (max)	47	43	Minimum	Heating Minimum.[1]
$H1_N$ Test (optional, steady)	70	60 (max)	47	43	Cooling Mode Maximum.	Heating Nominal.[4]
$H2_2$ Test (optional)	70	60 (max)	35	33	Maximum	Heating Full-Load.[3]
$H2_V$ Test (required)	70	60 (max)	35	33	Intermediate	Heating Intermediate.[5]
$H3_2$ Test (required, steady)	70	60 (max)	17	15	Maximum	Heating Full-Load.[3]
$H4_2$ Test (optional, steady)[6]	70	60 (max)	[7] 2	[7] 1	Maximum[8]	Heating Full-Load.[3]

[1] Defined in section 3.1.4.5.
[2] Maintain the airflow nozzle(s) static pressure difference or velocity pressure during an ON period at the same pressure or velocity as measured during the $H0_1$ Test.
[3] Defined in section 3.1.4.4.
[4] Defined in section 3.1.4.7.
[5] Defined in section 3.1.4.6.
[6] If the maximum speed is limited below 17 °F, this test becomes required.
[7] If the cutoff temperature is higher than 2 °F, run at the cutoff temperature.
[8] If maximum speed is limited by unit control, this test should run at the maximum speed allowed by the control, in such case, the speed is different from the maximum speed defined in the definition section.

c. For multiple-split heat pumps (only), the following procedures supersede the above requirements. For all Table 13 tests specified for a minimum compressor speed, at least one indoor unit must be turned off. The manufacturer shall designate the particular indoor unit(s) that is turned off. The manufacturer must also specify the compressor speed used for the Table 13 $H2_V$ Test, a heating mode intermediate compressor speed that falls within ¼ and ¾ of the difference between the maximum and minimum heating mode speeds. The manufacturer should prescribe an intermediate speed that is expected to yield the highest COP for the given $H2_V$ Test conditions and bracketed compressor speed range. The manufacturer can designate that one or more specific indoor units are turned off for the $H2_V$ Test.

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

Test any heat pump that has a heat comfort controller (see section 1.2, 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 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 northern heat pumps with triple-capacity compressors.

Test triple-capacity, northern heat pumps for the heating mode as follows:

a. Conduct one maximum-temperature test ($H0_1$), two high-temperature tests ($H1_2$ and $H1_1$), one Frost Accumulation test ($H2_2$), two low-temperature tests ($H3_2$, $H3_3$), and one minimum-temperature test ($H4_3$). Conduct an additional Frost Accumulation test ($H2_1$) and low-temperature test ($H3_1$) if both of the following conditions exist: (1) Knowledge of the heat pump's capacity and electrical power at low compressor capacity for outdoor temperatures of 37 °F and less is needed to complete the section 4.2.6 seasonal performance calculations; and (2) the heat pump's controls allow low-capacity operation at outdoor temperatures of 37 °F and less. If the above two conditions are met, an alternative to conducting the $H2_1$ Frost Accumulation Test to determine $\dot{Q}_h^{k=1}(35)$ and $\dot{E}_h^{k=1}(35)$ is to use the following equations to approximate this capacity and electrical power:

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

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

In evaluating the above equations, determine the quantities $\dot{Q}_h^{k=1}(47)$ from the $H1_1$ Test and evaluate them according to section 3.7. Determine the quantities $\dot{Q}_h^{k=1}(17)$ and $\dot{E}_h^{k=1}(17)$ from the $H3_1$ Test and evaluate them according to section 3.10. 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 or use the paired values calculated using the above default equations, whichever contribute to a higher Region IV HSPF based on the DHR.

b. Conducting a Frost Accumulation Test ($H2_3$) with the heat pump operating at its booster capacity is optional. If this optional test is not conducted, determine $\dot{Q}_h^{k=3}(35)$ and $\dot{E}_h^{k=3}(35)$ using the following equations to approximate this capacity and electrical power:

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

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

where,

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

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

Determine the quantities $\dot{Q}_h^{k=2}(47)$ and $\dot{E}_h^{k=2}(47)$ from the $H1_2$ Test and evaluate them according to section 3.7. Determine the quantities $\dot{Q}_h^{k=2}(35)$ and $\dot{E}_h^{k=2}(35)$ from the $H2_2$Test and evaluate them according to section 3.9.1. Determine the quantities $\dot{Q}_h^{k=2}(17)$ and $\dot{E}_h^{k=2}(17)$ from the $H3_2$Test, determine the quantities $\dot{Q}_h^{k=3}(17)$ and $\dot{E}_h^{k=3}(17)$ from the $H3_3$Test, and determine the quantities $\dot{Q}_h^{k=3}(2)$ and $\dot{E}_h^{k=3}(2)$ from the $H4_3$Test. Evaluate all six quantities according to section 3.10. Use the paired values of $\dot{Q}_h^{k=3}(35)$ and $\dot{E}_h^{k=3}(35)$ derived from conducting the $H2_3$Frost Accumulation Test and calculated as specified in section 3.9.1 or use the paired values calculated using the above default equations, whichever contribute to a higher Region IV HSPF based on the DHR.

c. Conduct the high-temperature cyclic test ($H1C_1$) to determine the heating mode cyclic-degradation coefficient, C_D^h. 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 (required, cyclic).	70	60 (max)	47	43	High	([3])
$H1_1$ Test (required)	70	60 (max)	47	43	Low	Heating Minimum.[1]
$H1C_1$ Test (required, 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 (max)[5] [6] (required, cyclic).	70	60 (max)	17	15	Booster	([7])
$H3_2$ Test (required, steady).	70	60 (max)	17	15	High	Heating Full-Load.[2]
$H3_1$ Test[5] (required, steady).	70	60 (max)	17	15	Low	Heating Minimum.[1]
$H4_3$ Test (required, steady).	70	60 (max)	2	1	Booster	Heating Full-Load.[2]

[1] Defined in section 3.1.4.5.
[2] Defined in section 3.1.4.4.
[3] Maintain the airflow nozzle(s) static pressure difference or velocity pressure during the ON period at the same pressure or velocity as measured during the $H1_2$ Test.
[4] Maintain the airflow nozzle(s) static pressure difference or velocity pressure during the ON period at the same pressure or velocity as measured during the $H1_1$Test.
[5] Required only if the heat pump's performance when operating at low compressor capacity and outdoor temperatures less than 37°F is needed to complete the section 4.2.6 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 $H2_I$ 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.

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 for additional requirements that depend on the selected secondary test method. After satisfying the pretest equilibrium requirements, make the measurements specified in Table 3 of ASHRAE Standard 37–2009 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 a 30-minute period (*e.g.,* four consecutive 10-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	0.12	[3] 0.02
Electrical voltage, % of rdg	2.0	1.5
Nozzle pressure drop, % of rdg	8.0	

[1] See section 1.2, 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 ASHRAE Standard 37–2009. Do not adjust the parameters used in calculating capacity for the permitted variations in test conditions. Assign the average space heating capacity and electrical power over the 30-minute data collection interval to the variables $\dot{Q}_h^k$ and $\dot{E}_h^k(T)$ respectively. The "T" and superscripted "k" are the same as described in section 3.3. Additionally, for the heating mode, use the superscript to denote results from the optional $H1_N$ Test, if conducted.c. For heat pumps tested without an indoor blower installed, increase $\dot{Q}_h^k(T)$ by

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

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

$$\frac{441\,W}{1000\,scfm} * \bar{\dot{V}}_s$$

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

where $\bar{V}_s$ is the average measured indoor air volume rate expressed in units of cubic feet per minute of standard air (scfm). During the 30-minute data collection interval of a High Temperature Test, pay attention to preventing a defrost cycle. Prior to this time, allow the heat pump to perform a defrost cycle if automatically initiated by its own controls. As in all cases, wait for the heat pump's defrost controls to automatically terminate the defrost cycle. Heat pumps that undergo a defrost 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 $\dot{E}_h^k(47)$.

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

$$\frac{1250\,BTU/h}{1000\,scfm} * \bar{\dot{V}}_s$$

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

$$\frac{365\,W}{1000\,scfm} * \bar{\dot{V}}_s$$

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

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, record the average indoor-side air volume rate, $\bar{\dot{V}}$, 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. 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 for the outdoor coil entering dry-bulb temperature. Drop the subscript ''dry'' used in variables cited in section 3.5 when referring to quantities from the cyclic heating mode test. The default C_D value for heating is 0.25. If available, use electric resistance heaters (see section 2.1) 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 except for making the following changes:

(1) When evaluating Equation 3.5–1, use the values of $\bar{V}$, $C_{p,a}$,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 heat pumps tested without an indoor blower installed (excluding the special case where a variable-speed fan is temporarily removed), increase q_{cyc} by the amount calculated using Equation 3.5–3. Additionally, increase e_{cyc} by the amount calculated using Equation 3.5–2. In making these calculations, use the average indoor air volume rate ($\bar{V}_s$) 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 cyclic test and the required steady-state test that were conducted at the same test conditions to determine the heating mode cyclic-degradation coefficient C_D^h. Add ''(k=2)'' to the coefficient if it corresponds to a two-capacity unit cycling at high capacity. For the below calculation of the heating mode cyclic degradation coefficient, do not include the duct loss correction from section 7.3.3.3 of ASHRAE Standard 37–2009 in determining $\dot{Q}_h^k(T_{cyc})$ (or q_{cyc}). The tested C_D^h is calculated as follows:

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

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	0.12	
Airflow nozzle pressure difference or velocity pressure,[2]% of reading	2.0	[3] 2.0
Electrical voltage,[4]% of rdg	8.0	1.5

[1] See section 1.2, Definitions.

[2] Applies during the interval that air flows through the indoor (outdoor) coil except for the first 30 seconds after flow initiation. For units having a variable-speed indoor blower that ramps, the tolerances listed for the external resistance to airflow shall apply from 30 seconds after achieving full speed until ramp down begins.

[3] The test condition shall be the average nozzle pressure difference or velocity pressure measured during the steady-state test conducted at the same test conditions.

[4] Applies during the interval that at least one of the following—the compressor, the outdoor fan, or, if applicable, the indoor blower—are operating, except for the first 30 seconds after compressor start-up.

3.9 Test procedures for Frost Accumulation heating mode tests (the H2, $H2_2$, $H2_V$, and $H2_1$ Tests).

a. Confirm that the defrost controls of the heat pump are set as specified in section 2.2.1. Operate the test room reconditioning apparatus and the heat pump for at least 30 minutes at the specified section 3.6 test conditions before starting the ''preliminary'' test period. The preliminary test period must immediately precede the ''official'' test period, which is the heating and defrost interval over which data are collected for evaluating average space heating capacity and average electrical power consumption.

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

c. The official test period begins when the preliminary test period ends, at defrost termination. The official test period ends at the termination of the next occurring automatic defrost cycle. When testing a heat pump that uses a time-adaptive defrost control system (see section 1.2, 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 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 heat pumps tested without an indoor blower installed, determine the corresponding cumulative time (in hours) of indoor coil airflow, $\Delta\tau_a$. Sample measurements used in calculating the air volume rate (refer to sections 7.7.2.1 and 7.7.2.2 of ASHRAE Standard 37–2009) at equal intervals that span 10 minutes or less. (Note: In the first printing of ASHRAE Standard 37–2009, the second IP equation for Q_{mi} should read:

$$Q_{mi} = 1097CA_n\sqrt{P_V v_n'.})$$

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
Outdoor entering wet-bulb temperature, °F	1.5		0.5
External resistance to airflow, inches of water	0.12		[5] 0.02
Electrical voltage, % of rdg	2.0		1.5

[1] See section 1.2, Definitions.
[2] Applies when the heat pump is in the heating mode, except for the first 10 minutes after termination of a defrost cycle.
[3] Applies during a defrost cycle and during the first 10 minutes after the termination of a defrost cycle when the heat pump is operating in the heating mode.
[4] For heat pumps that turn off the indoor blower during the defrost cycle, the noted tolerance only applies during the 10 minute interval that follows defrost termination.
[5] Only applies when testing non-ducted heat pumps.

3.9.1 Average space heating capacity and electrical power calculations.

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

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

Where,

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

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

v_n' = specific volume of the air-water vapor mixture at the nozzle, ft³/lbm_{mx}.

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

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

$$\Gamma = \int_{\tau_1}^{\tau_2} [T_{a2}(\tau) - T_{a1}(\tau)]d\tau,\ hr * °F$$

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

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

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

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

v_n = specific volume of the dry air portion of the mixture evaluated at the dry-bulb temperature, vapor content, and barometric pressure existing at the nozzle, ft³ per lbm of dry air.

To account for the effect of duct losses between the outlet of the indoor unit and the section 2.5.4 dry-bulb temperature grid, adjust $\dot{Q}_h^k(35)$ in accordance with section 7.3.4.3 of ASHRAE Standard 37–2009.

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

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

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

$$\frac{1505 \frac{Btu}{h}}{1000\, scfm} * \bar{\dot{V}}_s * \frac{\Delta\tau_a}{\Delta\tau_{FR}}$$

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

$$\frac{441W}{1000\, scfm} * \bar{\dot{V}}_s * \frac{\Delta\tau_a}{\Delta\tau_{FR}}$$

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

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

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

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

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

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

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

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

3.9.2 Demand defrost credit.

a. Assign the demand defrost credit, F_{def}, that is used in section 4.2 to the value of 1 in all cases except for heat pumps having a demand-defrost control system (see section 1.2, Definitions). For such qualifying heat pumps, evaluate F_{def} using,

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

Where,

$\Delta\tau_{def}$ = the time between defrost terminations (in hours) or 1.5, whichever is greater. A value of 6 must be assigned to $\Delta\tau_{def}$ if this limit is reached during a Frost Accumulation test and the heat pump has not completed a defrost cycle.

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

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

3.10 Test procedures for steady-state Low Temperature heating mode tests (the H3, $H3_2$, $H3_1$ and $H4_2$ Tests).

Except for the modifications noted in this section, conduct the Low Temperature heating mode test using the same approach as specified in section 3.7 for the Maximum and High Temperature tests. After satisfying the section 3.7 requirements for the pretest interval but before beginning to collect data to determine $\dot{Q}_h^k(17)$ and $\dot{E}_h^k(17)$, conduct a defrost cycle. This defrost cycle may be manually or automatically initiated. The defrost sequence must be terminated by the action of the heat pump's defrost controls. Begin the 30-minute data collection interval described in section 3.7, from which $\dot{Q}_h^k(17)$ and $\dot{E}_h^k(17)$ are determined, no sooner than 10 minutes after defrost termination. Defrosts should be prevented over the 30-minute data collection interval. Defrost cycle is not required for $H4_2$ Test.

3.11 Additional requirements for the secondary test methods.

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

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

3.11.1.1 If a preliminary test precedes the official test.

a. The test conditions for the preliminary test are the same as specified for the official test. Connect the indoor air-side test apparatus to the indoor coil; disconnect the outdoor air-side test apparatus. Allow the test room reconditioning apparatus and the unit being tested to operate for at least one hour. After attaining equilibrium conditions, measure the following quantities at equal intervals that span 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.,* four consecutive 10-minute samples) is obtained where the Table 8 or Table 15, whichever applies, test tolerances are satisfied.

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

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

Connect the outdoor-side test apparatus to the unit. Adjust the exhaust fan of the outdoor airflow measuring apparatus to achieve the same external static pressure as measured during the prior preliminary test conducted with the unit operating in the same cooling or heating mode at the same outdoor fan speed.

3.11.1.3 Official test.

a. Continue (preliminary test was conducted) or begin (no preliminary test) the official test by making measurements for both the Indoor and Outdoor Air Enthalpy Methods at equal intervals that span 5 minutes or less. Discontinue these measurements only after obtaining a 30-minute period where the specified test condition and test operating tolerances are satisfied. To constitute a valid official test:

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

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

b. For space cooling tests, calculate capacity from the outdoor air-enthalpy measurements as specified in sections 7.3.3.2 and 7.3.3.3 of ASHRAE Standard 37–2009. Calculate heating capacity based on outdoor air-enthalpy measurements as specified in sections 7.3.4.2 and 7.3.3.4.3 of the same ASHRAE Standard. Adjust the outdoor-side capacity according to section 7.3.3.4 of ASHRAE Standard 37–2009 to account for line losses when testing split systems. Use the outdoor unit fan power as measured during the official test and not the value measured during the preliminary test, as described in section 8.6.2 of ASHRAE Standard 37–2009, when calculating the capacity.

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

a. Conduct separate calibration tests using a calorimeter to determine the refrigerant flow rate. Or for cases where the superheat of the refrigerant leaving the evaporator is less than 5 °F, use the calorimeter to measure total capacity rather than refrigerant flow rate. Conduct these calibration tests at the same test conditions as specified for the tests in this appendix. Operate the unit for at least one hour or until obtaining equilibrium conditions before collecting data that will be used in determining the average refrigerant flow rate or total capacity. Sample the data at equal intervals that span 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 Standard 23.1–2010; sections 5, 6, 7, 8, 9, and 11 of ASHRAE Standard 41.9–2011; and section 7.4 of ASHRAE Standard 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 ASHRAE Standard 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 ASHRAE Standard 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 10 CFR 430.23 (for a single unit) and in 10 CFR 429.16 (for a sample).

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

3.13 Laboratory testing to determine off mode average power ratings.

Conduct one of the following tests after the completion of the B, B_1, or B_2 Test, whichever comes last: If the central air conditioner or heat pump lacks a compressor crankcase heater, perform the test in section 3.13.1; if the central air conditioner or heat pump has compressor crankcase heater that lacks controls, perform the test in section 3.13.1; if the central air conditioner or heat pump has a compressor crankcase heater equipped with controls, perform the test in section 3.13.2.

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 heater that lacks controls.

a. 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. This particular test contains no requirements as to ambient conditions within the test rooms, and room conditions are allowed to change during the test. Ensure that the low-voltage transformer and low-voltage components are connected.

b. Measure $P1_x$: 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.

c. Measure P_x for coil-only split systems (that would be installed in the field with a furnace having a dedicated board for indoor controls) and for blower-coil split systems for which a furnace 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.

d. Calculate *P1:*

Single-package systems and blower coil split systems for which the designated air mover is not a furnace: 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. If the compressor is a modulating-type, assign a value of 1.5 for the number of compressors. Round *P1* to the nearest watt and record as both *P1* and *P2*, the latter of which is the heating season per-compressor off mode power. The expression for calculating *P1* is as follows:

$$P1 = \frac{P1_x}{number\ of\ compressors}.$$

Coil-only split systems (that would be installed in the field with a furnace having a dedicated board for indoor controls) and blower-coil split systems for which a furnace 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. If the compressor is a modulating-type, assign a value of 1.5 for the number of compressors. Round *P1* to the nearest watt and record as both *P1* and *P2,* the latter of which is the heating season per-compressor off mode power. 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 that have a compressor crankcase heater equipped with controls.

a. 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 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. Ensure that the low-voltage transformer and low-voltage components are connected. Adjust the outdoor temperature at a rate of change of no more than 20 °F per hour and achieve an outdoor dry-bulb temperature of 72 °F. Maintain this temperature within ±2 °F for at least 5 minutes, while maintaining an indoor dry-bulb temperature of between 75 °F and 85 °F.

b. Measure $P1_x$: 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.

c. Reconfigure Controls: In the process of reaching the target outdoor dry-bulb temperature, adjust the outdoor temperature at a rate of change of no more than 20 °F per hour. This target temperature is the temperature specified by the manufacturer in the DOE Compliance Certification Database at which the crankcase heater turns on, minus five degrees Fahrenheit. Maintain this temperature within ±2 °F for at least 5 minutes, while maintaining an indoor dry-bulb temperature of between 75 °F and 85 °F.

d. Measure $P2_x$: 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.

e. Measure P_x for coil-only split systems (that would be installed in the field with a furnace having a dedicated board for indoor controls) and for blower-coil split systems for which a furnace 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

f. Calculate *P1:*

Single-package systems and blower coil split systems for which the air mover is not a furnace: 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. If the compressor is a modulating-type, assign a value of 1.5 for the number of compressors. The expression for calculating *P1* is as follows:

$$P1 = \frac{P1_x}{number\ of\ compressors}.$$

Coil-only split systems (that would be installed in the field with a furnace having a dedicated board for indoor controls) and blower-coil split systems for which a furnace 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. If the compressor is a modulating-type, assign a value of 1.5 for the number of compressors. The expression for calculating *P1* is as follows:

$$P1 = \frac{P1_x - P_x}{number\ of\ compressors}.$$

h. Calculate *P2*:

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. If the compressor is a modulating-type, assign a value of 1.5 for the number of compressors. The expression for calculating *P2* is as follows:

$$P2 = \frac{P2_x}{number\ of\ compressors}.$$

Coil-only split systems (that would be installed in the field with a furnace having a dedicated board for indoor controls) and blower-coil split systems for which a furnace 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. If the compressor is a modulating-type, assign a value of 1.5 for the number of compressors. The expression for calculating *P2* is as follows:

$$P2 = \frac{P2_x - P_x}{number\ of\ compressors}.$$

4. Calculations of Seasonal Performance Descriptors

4.1 Seasonal Energy Efficiency Ratio (SEER) Calculations. SEER must be calculated as follows: For equipment covered under sections 4.1.2, 4.1.3, and 4.1.4, 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, use a building cooling load, $BL(T_j)$. When referenced, evaluate $BL(T_j)$ for cooling using,

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

where,

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

1.1 = sizing factor, dimensionless.

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

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

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, Btu/h per watt.

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

b. Refer to section 3.3 regarding the definition and calculation of $\dot{Q}_c(82)$ and $\dot{E}_c(82)$.

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 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,

Equation 4.1.2-1 $$\frac{q_c(T_j)}{N} = X(T_j) * \dot{Q}_c(T_j) * \frac{n_j}{N}$$

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

where,

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

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

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

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

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

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

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

$$\dot{E}_c(T_j) = \dot{E}_c^{k=1}(T_j) + \frac{\dot{E}_c^{k=2}(T_j) - \dot{E}_c^{k=1}(T_j)}{FP_c^{k=2} - FP_c^{k=1}} * [FP_c(T_j) - FP_c^{k=1}]$$

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 regarding the definitions and calculations of $\dot{E}_c^{k=1}(82)$, $\dot{E}_c^{k=1}(95)$, $\dot{E}_c^{k=2}(82)$, and $\dot{E}_c^{k=2}(95)$.

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

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

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

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

where $\dot{Q}_c^{k=1}(82)$ and $\dot{E}_c^{k=1}(82)$ are determined from the B_1 Test, $\dot{Q}_c^{k=1}(67)$ and $\dot{E}_c^{k=1}(67)$ are determined from the F_1Test, and all four quantities are calculated as specified in section 3.3. Evaluate the space cooling capacity, $\dot{Q}_c^{k=2}(T_j)$, and electrical power consumption, $\dot{E}_c^{k=2}(T_j)$, of the test unit when operating at high compressor capacity and outdoor temperature T_j using,

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

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

where $\dot{Q}_c^{k=2}(95)$ and $\dot{E}_c^{k=2}(95)$ are determined from the A_2 Test, $\dot{Q}_c^{k=2}(82)$, and $\dot{E}_c^{k=2}(82)$, are determined from the B_2Test, and all are calculated as specified in section 3.3.

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

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

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

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, the 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 $\dot{E}_c^{k=1}(T_j)$.

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.

$PLF_j = 1 - C_D^{(k=2)} * [1 - X^{k=2}(T_j)]$, the part load factor, dimensionless.

Obtain the fraction bin hours for the cooling season, $\frac{n_j}{N}$, from Table 18. Use Equations 4.1.3-3 and 4.1.3-4, respectively, to evaluate $\dot{Q}_c^{k=2}(T_j)$ and $\dot{E}_c^{k=2}(T_j)$.

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

Calculate the space cooling capacity, $\dot{Q}_c^{k=v}(T_j)$, and electrical power consumption, $\dot{E}_c^{k=v}(T_j)$, of the test unit when operating at outdoor temperature T_j and the intermediate compressor speed used during the section 3.2.4 (and Table 7) E_V Test using,

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

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

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

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

where,

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

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

n_j/N, the 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 $\dot{Q}_c^{k=l}(T_j)$ and $\dot{E}_c^{k=l}(T_j)$.

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

$$\frac{q_c(T_j)}{N} = \dot{Q}_c^{k=1}(T_j) * \frac{n_j}{N} \qquad \frac{e_c(T_j)}{N} = \dot{E}_c^{k=1}(T_j) * \frac{n_j}{N}$$

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.

$\dot{E}_c^{k=1}(T_j) = \frac{\dot{Q}_c^{k=1}(T_j)}{EER^{k=1}(T_j)}$ 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,

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

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

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

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

where,

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

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

T_2 = the outdoor temperature at which the unit, when operating at maximum compressor speed, provides a space cooling capacity that is equal to the building load ($\dot{Q}_c^{k=2}(T_2) = BL(T_2)$), °F. Determine T_2 by equating Equations 4.1.3–3 and 4.1–2 and solving for outdoor temperature.

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

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

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

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

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

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

4.1.5 SEER calculations for an air conditioner or heat pump having a single indoor unit with multiple blowers. Calculate SEER using Eq. 4.1–1, where $q_c(Tj)/N$ and $e_c(Tj)/N$ are evaluated as specified in applicable below subsection.

4.1.5.1 For multiple blower systems that are connected to a lone, 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. 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. 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, CD_c, as per sections 3.2.2.1 and 3.5 to 3.5.3. Assign this same value to $CD_c(K=2)$. c. Except for using the above values of $\dot{Q}_c^{k=1}(T_j)$, $\dot{E}_c^{k=1}(T_j)$, $\dot{E}_c^{k=2}(T_j)$, $\dot{Q}_c^{k=2}(T_j)$, CD_c, and CD_c (K=2), calculate the quantities $q_c(T_j)/N$ and $e_c(T_j)/N$ as specified in section 4.1.3.1 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 if $\dot{Q}_c^{k=2}(T_j) > BL(T_j)$ or as specified in section 4.1.3.4 if $\dot{Q}_c^{k=2}(T_j) \leq BL(T_j)$.

4.1.5.2 For multiple blower systems that are connected to either a lone outdoor unit having a two-capacity compressor or to 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.

4.2 Heating Seasonal Performance Factor (HSPF) Calculations.

Unless an approved alternative efficiency determination method is used, as set forth in 10 CFR 429.70(e), HSPF must be calculated as follows: Six generalized climatic regions are depicted in Figure 1 and otherwise defined in Table 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 space heating season when the outdoor temperature fell within the range represented by bin temperature T_j to the total number of hours in the heating season (N), W. For heat pumps having a heat comfort controller, this ratio may also include electrical energy used by resistive elements to maintain a minimum air delivery temperature (see 4.2.5).

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

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

n_j/N, the 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, 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 number	I	II	III	IV	V	VI
Heating Load Hours	562	909	1,363	1,701	2,202	1,974 *
Outdoor Design Temperature, T_{OD}	37	27	17	5	−10	30
Zero Load Temperature, T_{ZL}	60	58	57	55	55	58
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

Equation 4.2-2 $$BL(T_j) = \frac{(T_{zl} - T_j)}{T_{zl} - T_{OD}} * DHR$$

where,

T_{OD}, the outdoor design temperature, °F. An outdoor design temperature is specified for each generalized climatic region in Table 19.

DHR, the design heating requirement (see section 1.2, Definitions), Btu/h.

T_{zl}, the zero load temperature, °F

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

For a heat pump that delivers both cooling and heating,

$$DHR = \dot{Q}_c^{k=2}(95) * C * \frac{T_{zl} - T_{OD}}{T_{zl} - 5}$$

where,

C = 1.3, a multiplier to provide the appropriate slope for the heating load line, dimensionless.

T_{zl}, the zero load temperature, °F

$\dot{Q}_c^{k=2}(95)$, the space cooling capacity of the unit as determined from the A or A_2 Test, whichever applies, Btu/h.

For a heating-only heat pump,

$$DHR = \dot{Q}_h^{k}(47) * C * \frac{T_{zl} - T_{OD}}{T_{zl} - 5}$$

where,

C = 1.3, a multiplier to provide the appropriate slope for the heating load line, dimensionless.

T_{zl}, the zero load temperature, °F

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

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

2. For a variable-speed heating only heat pump, a section 3.6.2 single-speed heating only heat pump, or a two-capacity heating only heat pump, $\dot{Q}_h^k(47) = \dot{Q}_h^{k=2}(47)$, the space heating capacity determined from the $H1_2$ Test.

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

For heat pumps with heat comfort controllers (see section 1.2, 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 for the additional steps required for calculating the HSPF.

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

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

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

where,

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

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

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

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

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

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

Use Equation 4.2–2 to determine $BL(T_j)$. Obtain fractional bin hours for the heating season, n_j/N, from Table 19.

Determine the low temperature cut-out factor using

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

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,

Equation 4.2.1-4

$$\dot{Q}_h(T_j) = \begin{cases} \dot{Q}_h(17) + \frac{[\dot{Q}_h(47) - \dot{Q}_h(17)] * (T_j - 17)}{47 - 17}, & if\ T_j \geq 45\ °F\ or\ T_j \leq 17\ °F \\ \dot{Q}_h(17) + \frac{[\dot{Q}_h(35) - \dot{Q}_h(17)] * (T_j - 17)}{35 - 17}, & if\ 17\ °F < T_j < 45\ °F \end{cases}$$

Equation 4.2.1-5

$$\dot{E}_h(T_j) = \begin{cases} \dot{E}_h(17) + \frac{[\dot{E}_h(47) - \dot{E}_h(17)] * (T_j - 17)}{47 - 17}, & if\ T_j \geq 45\ °F\ or\ T_j \leq 17\ °F \\ \dot{E}_h(17) + \frac{[\dot{E}_h(35) - \dot{E}_h(17)] * (T_j - 17)}{35 - 17}, & if\ 17\ °F < T_j < 45\ °F \end{cases}$$

where,

$\dot{Q}_h(47)$ and $\dot{E}_h(47)$ are determined from the H1 Test and calculated as specified in section 3.7

$\dot{Q}_h(35)$ and $\dot{E}_h(35)$ are determined from the H2 Test and calculated as specified in section 3.9.1

$\dot{Q}_h(17)$ and $\dot{E}_h(17)$ are determined from the H3 Test and calculated as specified in section 3.10.

4.2.2 Additional steps for calculating the HSPF of a heat pump having a single-speed compressor and a variable-speed, variable-air-volume-rate indoor 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 with the exception of replacing references to the H1C Test and section 3.6.1 with the $H1C_1$ Test and section 3.6.2. In addition, evaluate the space heating capacity and electrical power consumption of the heat pump $\dot{Q}_h(T_j)$ and $\dot{E}_h(T_j)$ using

Equation 4.2.2-1 $$\dot{Q}_h(T_j) = \dot{Q}_h^{k=1}(T_j) + \frac{\dot{Q}_h^{k=2}(T_j) - \dot{Q}_h^{k=1}(T_j)}{FP_h^{k=2} - FP_h^{k=1}} * [FP_h(T_j) - FP_h^{k=1}]$$

Equation 4.2.2-2 $$\dot{E}_h(T_j) = \dot{E}_h^{k=1}(T_j) + \frac{\dot{E}_h^{k=2}(T_j) - \dot{E}_h^{k=1}(T_j)}{FP_h^{k=2} - FP_h^{k=1}} * [FP_h(T_j) - FP_h^{k=1}]$$

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

Equation 4.2.2-3

$$\dot{Q}_h^k(T_j) = \begin{cases} \dot{Q}_h^k(17) + \frac{[\dot{Q}_h^k(47) - \dot{Q}_h^k(17)] * (T_j - 17)}{47 - 17}, & if\ T_j \geq 45\ °F\ or\ T_j \leq 17\ °F \\ \dot{Q}_h^k(17) + \frac{[\dot{Q}_h^k(35) - \dot{Q}_h^k(17)] * (T_j - 17)}{35 - 17}, & if\ 17\ °F < T_j < 45\ °F \end{cases}$$

Equation 4.2.2-4

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

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. Determine $\dot{Q}_h^{k=1}(35)$ and $\dot{E}_h^{k=1}(35)$ as specified in section 3.6.2; determine $\dot{Q}_h^{k=2}(35)$ and $\dot{E}_h^{k=2}(35)$ and from the $H2_2$ Test and the calculation specified in section 3.9. Determine $\dot{Q}_h^{k=1}(17)$ and $\dot{E}_h^{k=1}(17$ from the $H3_1$ Test, and $\dot{Q}_h^{k=2}(17)$ and

$\dot{E}_h^{k=2}(17)$ from the $H3_2$ Test. Calculate all four quantities as specified in section 3.10.

4.2.3 Additional steps for calculating the HSPF of a heat pump having a two-capacity compressor. The calculation of the Equation 4.2–1 quantities differ depending upon whether the heat pump would operate at low capacity (section 4.2.3.1), cycle between low and high capacity (Section 4.2.3.2), or operate at high capacity (sections 4.2.3.3 and 4.2.3.4) in responding to the building load. For heat pumps that lock out low capacity operation at low outdoor temperatures, the manufacturer must supply information regarding the cutoff temperature(s) so that the appropriate equations can be selected.

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

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. Determine $\dot{Q}_h^{k=2}(35)$ and $\dot{E}_h^{k=2}(35)$ from the $H2_2$ Test and, if required as described in section 3.6.3, determine $\dot{Q}_h^{k=1}(35)$ and $\dot{E}_h^{k=1}(35)$ from the $H2_1$ Test. Calculate the required 35 °F quantities as specified in section 3.9. Determine $\dot{Q}_h^{k=2}(17)$ and $\dot{E}_h^{k=2}(17)$ from the $H3_2$ Test and, if required as described in section 3.6.3, determine $\dot{Q}_h^{k=1}(17)$ and $\dot{E}_h^{k=1}(17)$ from the $H3_1$ Test. Calculate the required 17 °F quantities as specified in section 3.10.

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

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

$$\text{Equation 4.2.3-2} \quad \frac{RH(T_j)}{N} = \frac{BL(T_j) * [1 - \delta(T_j)]}{3.413 \frac{Btu/h}{W}} * \frac{n_j}{N}$$

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.

Determine the low temperature cut-out factor using

$$\delta(T_j) = \begin{cases} 0, & if\ T_j \leq T_{off} \\ 1/2, & if\ T_{off} < T_j \leq T_{on} \\ 1, & if\ T_j > T_{on} \end{cases}$$

Where,

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

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

(b) T_j is below this lockout threshold temperature.

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

Calculate $\frac{RH(T_j)}{N}$ using Equation 4.2.3-2. Evaluate $\frac{e_h(T_j)}{N}$ using

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

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,

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

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

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

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

$$\frac{e_h(T_j)}{N} = \dot{E}_h^{k=2}(T_j) * \delta'(T_j) * \frac{n_j}{N} \qquad \frac{RH(T_j)}{N} = \frac{BL(T_j) * [\dot{Q}_h^{k=2}(T_j) * \delta'(T_j)]}{3.413 \frac{Btu/h}{W}} * \frac{n_j}{N}$$

Where

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

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

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

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

Where,

$\dot{Q}_h^{k=1}(62)$ and $\dot{E}_h^{k=1}(62)$ are determined from the $H0_1$ Test

$\dot{Q}_h^{k=1}(47)$ and $\dot{E}_h^{k=1}(47)$ are determined from the $H1_1$ Test,

and all four quantities are calculated as specified in section 3.7.

Evaluate the space heating capacity, $\dot{Q}_h^{k=2}(T_j)$, and electrical power consumption, $\dot{E}_h^{k=2}(T_j)$, of the heat pump when operating at maximum compressor speed and outdoor temperature T_j by solving Equations 4.2.2–3 and 4.2.2–4, respectively, for k=2. Determine the Equation 4.2.2–3 and 4.2.2–4 quantities $\dot{Q}_h^{k=2}(47)$ and $\dot{E}_h^{k=2}(47)$ from the $H1_2$ Test and the calculations specified in section 3.7. Determine $\dot{Q}_h^{k=2}(35)$ and $\dot{E}_h^{k=2}(35)$ from the $H2_2$ Test and the calculations specified in section 3.9 or, if the $H2_2$ Test is not conducted, by conducting the calculations specified in section 3.6.4. Determine $\dot{Q}_h^{k=2}(17)$ and $\dot{E}_h^{k=2}(17)$ from the $H3_2$ Test and the calculations specified in section 3.10. If $H4_2$ test is 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 maximum compressor speed and outdoor temperature T_j by using the following equation instead of Equations 4.2.2–3 and 4.2.2–4. Determine the quantities $\dot{Q}_h^{k=2}(T_l)$ and $\dot{E}_h^{k=2}(T_l)$ from the $H4_2$ Test and the calculations specified in section 3.7.

$$\dot{Q}_h^2(T_j) = \begin{cases} \dot{Q}_h^2(17) + \frac{[\dot{Q}_h^2(T_l) - \dot{Q}_h^2(17)] * (T_j - 17)}{T_l - 17}, & if\ T_j \geq 45\ °\mathrm{F}\ or\ T_j \leq 17\ °\mathrm{F} \\ \dot{Q}_h^2(17) + \frac{[\dot{Q}_h^2(35) - \dot{Q}_h^2(17)] * (T_j - 17)}{35 - 17}, & if\ 17\ °\mathrm{F} < T_j < 45\ °\mathrm{F} \end{cases}$$

$$\dot{E}_h^2(T_j) = \begin{cases} \dot{E}_h^2(17) + \frac{[\dot{E}_h^2(T_l) - \dot{E}_h^2(17)] * (T_j - 17)}{T_l - 17}, & if\ T_j \geq 45\ °\mathrm{F}\ or\ T_j \leq 17\ °\mathrm{F} \\ \dot{E}_h^2(17) + \frac{[\dot{E}_h^2(35) - \dot{E}_h^2(17)] * (T_j - 17)}{35 - 17}, & if\ 17\ °\mathrm{F} < T_j < 45\ °\mathrm{F} \end{cases}$$

Where T_l is the outdoor temperature where the $H4_2$ test is conducted.

Calculate the space heating capacity, $\dot{Q}_h^{k=v}(T_j)$, and electrical power consumption, $\dot{E}_h^{k=v}(T_j)$, of the heat pump when operating at outdoor temperature T_j and the intermediate compressor speed used during the section 3.6.4 $H2_V$ Test using

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

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

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. 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,

$$N_Q = \frac{\dot{Q}_h^{k=v}(35) - \dot{Q}_h^{k=1}(35)}{\dot{Q}_h^{k=2}(35) - \dot{Q}_h^{k=1}(35)} \quad N_E = \frac{\dot{E}_h^{k=v}(35) - \dot{E}_h^{k=1}(35)}{\dot{E}_h^{k=2}(35) - \dot{E}_h^{k=1}(35)}$$

Use Equations 4.2.4–1 and 4.2.4–2, respectively, to calculate $\dot{Q}_h^{k=1}(35)$ and $\dot{E}_h^{k=1}(35)$. The calculation of Equation 4.2–1 quantities $e_h(T_j)/N$ and $RH(T_j)/N$ differs depending upon whether the heat pump would operate at minimum speed (section 4.2.4.1), operate at an intermediate speed (section 4.2.4.2), or operate at maximum speed (section 4.2.4.3) in responding to the building load.

4.2.4.1 Steady-state space heating capacity when operating at minimum compressor speed is greater than or equal to the building heating load at temperature T_j, $\dot{Q}_h^{k=1}(T_j \geq BL(T_j)$. Evaluate the Equation 4.2-1 quantities $\frac{e_h(T_j)}{N}$ and $\frac{RH(T_j)}{N}$ as specified in section 4.2.3.1. Except now use Equations 4.2.4-1 and 4.2.4-2 to evaluate $\dot{Q}_h^{k=1}(T_j)$ and $\dot{E}_h^{k=1}(T_j)$, respectively, and replace section 4.2.3.1 references to "low capacity" and section 3.6.3 with "minimum speed" and section 3.6.4. Also, the last sentence of section 4.2.3.1 does not apply.

4.2.4.2 Heat pump operates at an intermediate compressor speed (k=i) in order to match the building heating load at a temperature T_j, $\dot{Q}_h^{k=1}(T_j) < BL(T_j) < \dot{Q}_h^{k=2}(T_j)$. Calculate $\frac{RH(T_j)}{N}$ using Equation 4.2.3-2 while evaluating $\frac{e_h(T_j)}{N}$ using:

$$\frac{e_h(T_j)}{N} = \dot{E}_h^{k=1}(T_j) * \delta(T_j) * \frac{n_j}{N}$$

where,

$$\dot{E}_h^{k=1}(T_j) = \frac{\dot{Q}_h^{k=1}(T_j)}{3.413\frac{Btu/h}{W} * \ COP^{k=1}(T_j)}$$

and $\delta(T_j)$ is evaluated using Equation 4.2.3–3 while,

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

$COP^{k=i}(T_j)$, the steady-state coefficient of performance of the heat pump when operating at compressor speed k=i and temperature T_j, dimensionless.

For each temperature bin where the heat pump operates at an intermediate compressor speed, determine $COP^{k=i}(T_j)$ using,

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

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

$$D = \frac{T_3^2 - T_4^2}{T_{vh}^2 - T_4^2} B = \frac{COP^{k=2}(T_4) - COP^{k=1}(T_3) - D*[COP^{k=2}(T_4) - COP^{k=v}(T_{vh})]}{T_4 - T_3 - D*(T_4 - T_{vh})}$$

Where,

T_3, the outdoor temperature at which the heat pump, when operating at minimum compressor speed, provides a space heating capacity that is equal to the building load ($\dot{Q}_h^{k=1}(T_3) = BL(T_3)$), °F. Determine T_3 by equating Equations 4.2.4–1 and 4.2–2 and solving for outdoor temperature:

$$C = \frac{COP^{k=2}(T_4) - COP^{k=1}(T_3) - B*(T_4 - T_3)}{T_4^2 - T_3^2} \quad A = COP^{k=2}(T_4) - B * T_4 - C * T_4^2$$

T_{vh}, the outdoor temperature at which the heat pump, when operating at the intermediate compressor speed used during the section 3.6.4 $H2_V$ Test, provides a space heating capacity that is equal to the building load ($\dot{Q}_h^{k=v}(T_{vh}) = BL(T_{vh})$), °F. Determine T_{vh} by equating Equations 4.2.4–3 and 4.2–2 and solving for outdoor temperature.

T_4, the outdoor temperature at which the heat pump, when operating at maximum compressor speed, provides a space heating capacity that is equal to the building load ($\dot{Q}_h^{k=2}(T_4) = BL(T_4)$), °F.

Determine T_4 by equating Equations 4.2.2–3 (k=2) and 4.2–2 and solving for outdoor temperature.

$$COP^{k=1}(T_3) = \frac{\dot{Q}_h^{k=1}(T_3)\left[Eqn.\,4.2.4-1, substituting\ T_3\ for\ T_j\right]}{3.413\frac{Btu/h}{W} * \dot{E}_h^{k=1}(T_3)\left[Eqn.\,4.2.4-2, substituting\ T_3\ for\ T_j\right]}$$

$$COP^{k=v}(T_{vh}) = \frac{\dot{Q}_h^{k=v}(T_{vh})\left[Eqn.\,4.2.4-3, substituting\ T_{vh}\ for\ T_j\right]}{3.413\frac{Btu/h}{W} * \dot{E}_h^{k=v}(T_{vh})\left[Eqn.\,4.2.4-4, substituting\ T_{vh}\ for\ T_j\right]}$$

$$COP^{k=2}(T_4) = \frac{\dot{Q}_h^{k=2}(T_4)\left[Eqn.\,4.2.2-3, substituting\ T_4\ for\ T_j\right]}{3.413\frac{Btu/h}{W} * \dot{E}_h^{k=2}(T_4)\left[Eqn.\,4.2.2-4, substituting\ T_4\ for\ T_j\right]}$$

For multiple-split heat pumps (only), the following procedures supersede the above requirements for calculating $COP_h^{k=i}(T_j)$. For each temperature bin where $T_3 > T_j > T_{vh}$,

$$COP_h^{k=i}(T_j) = COP_h^{k=1}(T_3) + \frac{COP_h^{k=v}(T_{vh}) - COP_h^{k=1}(T_3)}{T_{vh} - T_3} * (T_j - T_3)$$

For each temperature bin where $T_{vh} \geq T_j > T_4$,

$$COP_h^{k=1}(T_j) = COP_h^{k=v}(T_{vh}) + \frac{COP_h^{k=2}(T_4) - COP_h^{k=v}(T_{vh})}{T_4 - T_{vh}} * (T_j - T_{vh})$$

4.2.4.3 Heat pump must operate continuously at maximum (k=2) compressor speed at temperature T_j, $BL(T_j) \geq \dot{Q}_h^{k=2}(T_j)$. Evaluate the Equation 4.2–1 quantities

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

as specified in section 4.2.3.4 with the understanding that $\dot{Q}_h^{k=2}(T_j)$ and $\dot{E}_h^{k=2}(T_j)$ correspond to maximum compressor speed operation and are derived from the results of the specified section 3.6.4 tests. If $H4_2$ test is conducted in place of $H1_2$, evaluate $\dot{Q}_h^{k=2}(T_j)$ and $\dot{E}_h^{k=2}(T_j)$ using the following equation instead of equations 4.2.2–3 and 4.2.2–4.

$$\dot{Q}_h^2(T_j) = \dot{Q}_h^2(T_L) + \frac{[\dot{Q}_h^2(17) - \dot{Q}_h^2(T_L)] \cdot (T_j - T_L)}{17 - T_L}$$

$$\dot{E}_h^2(T_j) = \dot{E}_h^2(T_L) + \frac{[\dot{E}_h^2(17) - \dot{E}_h^2(T_L)] \cdot (T_j - T_L)}{17 - T_L}$$

Where, T_L is the ambient dry bulb temperature where $H4_2$ test is conducted.

4.2.5 Heat pumps having a heat comfort controller. Heat pumps having heat comfort controllers, when set to maintain a typical minimum air delivery temperature, will cause the heat pump condenser to operate less because of a greater contribution from the resistive elements. With a conventional heat pump, resistive heating is only initiated if the heat pump condenser cannot meet the building load (*i.e.,* is delayed until a second stage call from the indoor thermostat). With a heat comfort controller, resistive heating can occur even though the heat pump condenser has adequate capacity to meet the building load (*i.e.,* both on during a first stage call from the indoor thermostat). As a result, the outdoor temperature where the heat pump compressor no longer cycles (*i.e.,* starts to run continuously), will be lower than if the heat pump did not have the heat comfort controller.

4.2.5.1 Heat pump having a heat comfort controller: additional steps for calculating the HSPF of a heat pump having a single-speed compressor that was tested with a fixed-speed indoor blower installed, a constant-air-volume-rate indoor blower installed, or with no indoor blower installed. Calculate the space heating capacity and electrical power of the heat pump without the heat comfort controller being active as specified in section 4.2.1 (Equations 4.2.1–4 and 4.2.1–5) for each outdoor bin temperature, T_j, that is listed in Table 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 $Btu/lbm_{da} \cdot {}^{\circ}F$) from the results of the H1 Test using:

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

Where $\overline{\dot{V}}_s$, $\overline{\dot{V}}_{mx}$, v'_n (or v_n), and W_n are defined following Equation 3–1. For each outdoor bin temperature listed in Table 19, calculate the nominal temperature of the air leaving the heat pump condenser coil using,

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

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. For each bin calculation, use the space heating capacity and electrical power from Case 1 or Case 2, whichever applies.

Case 1. For outdoor bin temperatures where $T_o(T_j)$ is equal to or greater than T_{CC} (the maximum supply temperature determined according to section 3.1.9), determine $\dot{Q}_h(T_j)$ and $\dot{E}_h(T_j)$ as specified in section 4.2.1 (*i.e.,* $\dot{Q}_h(T_j) = \dot{Q}_{hp}(T_j)$ and $\dot{E}_{hp}(T_j) = \dot{E}_{hp}(T_j)$). Note: Even though $T_o(T_j) \geq T_{cc}$, resistive heating may be required; evaluate Equation 4.2.1–2 for all bins.

Case 2. For outdoor bin temperatures where $T_o(T_j) > T_{cc}$, determine $\dot{Q}_h(T_j)$ and $\dot{E}_h(T_j)$ using,

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

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 (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 $Btu/lbm_{da} \cdot {}^{\circ}F$) from the results of the $H1_2$ Test using:

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

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

Where $\overline{\dot{V}}_S$, $\overline{\dot{V}}_{mx}$, v'_n (or v_n), and W_n are defined following Equation 3–1. For each outdoor bin temperature listed in Table 19, calculate the nominal temperature of the air leaving the heat pump condenser coil using,

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

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 with the exception of replacing references to the H1C Test and section 3.6.1 with the $H1C_1$ Test and section 3.6.2. For each bin calculation, use the space heating capacity and electrical power from Case 1 or Case 2, whichever applies.

Case 1. For outdoor bin temperatures where $T_o(T_j)$ is equal to or greater than T_{CC} (the maximum supply temperature determined according to section 3.1.9), determine $\dot{Q}_h(T_j)$ and $\dot{E}_h(T_j)$ as specified in section 4.2.2 (*i.e.* $\dot{Q}_h(T_j) = \dot{Q}_{hp}(T_j)$ and $\dot{E}_h(T_j) = \dot{E}_{hp}(T_j)$). Note: Even though $T_o(T_j) \geq T_{CC}$, resistive heating may be required; evaluate Equation 4.2.1–2 for all bins.

Case 2. For outdoor bin temperatures where $T_o(T_j) < T_{CC}$, determine $\dot{Q}_h(T_j)$ and $\dot{E}_h(T_j)$ using,

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

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 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 $Btu/lbm_{da} \cdot °F$) from the results of the $H1_1$ Test using:

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

$$C_{p,da}^{k=1} = 0.24 + 0.444 * W_n$$

Where $\overline{\dot{V}}_s$, $\overline{\dot{V}}_{mx}$, v'_n (or v_n), and W_n are defined following Equation 3–1. For each outdoor bin temperature listed in Table 19, calculate the nominal temperature of the air leaving the heat pump condenser coil when operating at low capacity using,

$$T_0^{k=1}(T_j) = 70°F + \frac{\dot{Q}_{hp}^{k=1}(T_j)}{\dot{m}_{da}^{k=1} * C_{p,da}^{k=1}}$$

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,

$$T_0^{k=2}(T_j) = 70°F + \frac{\dot{Q}_{hp}^{k=2}(T_j)}{\dot{m}_{da}^{k=2} * C_{p,da}^{k=2}}$$

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, whichever applies, for each temperature bin. To evaluate these quantities, use the low-capacity space heating capacity and the low-capacity electrical power from Case 1 or Case 2, whichever applies; use the high-capacity space heating capacity and the high-capacity electrical power from Case 3 or Case 4, whichever applies.

Case 1. For outdoor bin temperatures where $T_o^{k=1}(T_j)$ is equal to or greater than T_{CC} (the maximum supply temperature determined according to section 3.1.9), determine $\dot{Q}_h^{k=1}(T_j)$ and $\dot{E}_h^{k=1}(T_j)$ as specified in section 4.2.3 (*i.e.,* $\dot{Q}_h^{k=1}(T_j) = \dot{Q}_{hp}^{k=1}(T_j)$ and $\dot{E}_h^{k=1}(T_j) = \dot{E}_{hp}^{k=1}(T_j)$.

Note: Even though $T_o^{k=1}(T_j) \geq T_{CC}$, resistive heating may be required; evaluate $RH(T_j)/N$ for all bins.

Case 2. For outdoor bin temperatures where $T_o^{k=1}(T_j) < T_{CC}$, determine $\dot{Q}_h^{k=1}(T_j)$ and $\dot{E}_h^{k=1}(T_j)$ using,

$$\dot{Q}_h^{k=1}(T_j) = \dot{Q}_{hp}^{k=1}(T_j) + \dot{Q}_{CC}^{k=1}(T_j) \qquad \dot{E}_h^{k=1}(T_j) = \dot{E}_{hp}^{k=1}(T_j) + \dot{E}_{CC}^{k=1}(T_j)$$

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 (*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,

$$\dot{Q}_h^{k=2}(T_j) = \dot{Q}_{hp}^{k=2}(T_j) + \dot{Q}_{CC}^{k=2}(T_j) \quad \dot{E}_h^{k=2}(T_j) = \dot{E}_{hp}^{k=2}(T_j) + \dot{E}_{CC}^{k=2}(T_j)$$

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), cycle on and off at high capacity (section 4.2.6.2), cycle on and off at booster capacity (4.2.6.3), cycle between low and high capacity (section 4.2.6.4), cycle between high and booster capacity (section 4.2.6.5), operate continuously at low capacity (4.2.6.6), operate continuously at high capacity (section 4.2.6.7), operate continuously at booster capacity (4.2.6.8), or heat solely using resistive heating (also section 4.2.6.8) 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 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. If, in accordance with section 3.6.6, 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 and determine $\dot{Q}_h^{k=1}(35)$ and $\dot{E}_h^{k=1}(35)$ as specified in section 3.6.6.

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. Determine the equation input for $\dot{Q}_h^{k=2}(35)$ and $\dot{E}_h^{k=2}(35)$ from the $H2_2$, evaluated as specified in section 3.9.1. 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.

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

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

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

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. 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.

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, $\dot{Q}_h^{k=1}(T_j) \ge BL(T_j)$., 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.

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. Determine the equation inputs $X^{k=2}(T_j)$, PLF_j, and $\delta'(T_j)$ as specified in section 4.2.3.3. 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.

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, $BL(T_j) \le \dot{Q}_h^{k=3}(T_j)$.

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

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

Determine the low temperature cut-out factor, $\delta'(T_j)$, using Eq. 4.2.3–3. Use the booster-capacity cyclic-degradation coefficient, $C_D^h(k=3)$ determined in accordance with section 3.6.6.

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. 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.

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, $BL(T_j) > \dot{Q}_h^{k=1}(T_j)$.

$$\frac{e_h(T_j)}{N} = \dot{E}_h^{k=1}(T_j) * \delta'(T_j) * \frac{n_j}{N} \quad \text{and} \quad \frac{RH(T_j)}{N} = \frac{BL(T_j) - [\dot{Q}_h^{k=1}(T_j) * \delta(T_j)]}{3.413 \frac{Btu/h}{W}} * \frac{n_j}{N}$$

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, $BL(T_j) > \dot{Q}_h^{k=2}(T_j)$. Evaluate the quantities

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

as specified in section 4.2.3.4. Calculate $\delta''(Tj)$ using the equation given in section 4.2.3.4.

4.2.6.8 Heat pump only operates at booster (k = 3) capacity at temperature Tj and its capacity is less than the building heating load, $BL(T_j) > \dot{Q}_h^{k=3}(T_j)$. or the system converts to using only resistive heating.

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

Where $\delta''(Tj)$ is calculated as specified in section 4.2.3.4 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 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 applicable below subsection.

4.2.7.1 For multiple 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. Determine the quantities $\dot{Q}_h^{k=1}(35)$ and $\dot{E}_h^{k=1}(35)$ as specified in section 3.6.2. Determine $\dot{Q}_h^{k=2}(35)$ and $\dot{E}_h^{k=2}(35)$ from the $H2_2$ Frost Accumulation Test as calculated according to section 3.9.1. 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. Refer to section 3.6.2 and Table 11 for additional information on the referenced laboratory tests.

b. Determine the heating mode cyclic degradation coefficient, CD_h, as per sections 3.6.2 and 3.8 to 3.8.1. Assign this same value to $CD_h(k = 2)$.

c. Except for using the above values of $\dot{Q}_h^{k=1}(Tj)$, $\dot{E}_h^{k=1}(Tj)$, $\dot{Q}_h^{k=2}(Tj)$, $\dot{E}_h^{k=2}(Tj)$, CD_h, and $CD_h(k = 2)$, calculate the quantities $e_h(T_j)/N$ as specified in section 4.2.3.1 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 if $\dot{Q}_h^{k=2}(Tj) > BL(Tj)$ or as specified in section 4.2.3.4 if $\dot{Q}_h^{k=2}(Tj) \leq BL(T_j)$.

4.2.7.2 For multiple blower heat pumps connected to either a lone 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.

4.3 Calculations of Off-mode Seasonal Power and Energy Consumption.

4.3.1 For central air conditioners and heat pumps with a cooling capacity of:

less than 36,000 Btu/h, determine the off mode rating, $P_{W,OFF}$, with the following equation:

$$P_{W,OFF} = \begin{cases} P1 & if\ P2 = 0 \\ \frac{P1+P2}{2} & otherwise \end{cases};$$

4.3.2 Calculate the off mode energy consumption for both central air conditioner and heat pumps for the shoulder season, $E1$, using: $E1 = P1 \cdot SSH$; and the off mode energy consumption of a CAC, only, for the heating season, $E2$, using: $E2 = P2 \cdot HSH$; where $P1$ and $P2$ is determined in Section 3.13. *HSH* can be determined by multiplying the heating season-hours from Table 20 with the fractional Bin-hours, from Table 19, that pertain to the range of temperatures at which the crankcase heater operates. If the crankcase heater is controlled to disable for the heating season, the temperature range at which the crankcase heater operates is defined to be from 72 °F to five degrees Fahrenheit below a turn-off temperature specified by the manufacturer in the DOE Compliance Certification Database. If the crankcase heater is operated during the heating season, the temperature range at which the crankcase heater operates is defined to be from 72 °F to −23 °F, the latter of which is a temperature that sets the range of Bin-hours to encompass all outside air temperatures in the heating season.

SSH can be determined by multiplying the shoulder season-hours from Table 20 with the fractional Bin-hours in Table 21.

TABLE 20—REPRESENTATIVE COOLING AND HEATING LOAD HOURS AND THE CORRESPONDING SET OF SEASONAL HOURS FOR EACH GENERALIZED CLIMATIC REGION

Climatic region	Cooling load hours CLH_R	Heating load hours HLH_R	Cooling season hours CSH_R	Heating season hours HSH_R	Shoulder season hours SSH_R
I	2,400	750	6,731	1,826	203
II	1,800	1,250	5,048	3,148	564
III	1,200	1,750	3,365	4,453	942
IV	800	2,250	2,244	5,643	873
Rating Values	1,000	2,080	2,805	5,216	739
V	400	2,750	1,122	6,956	682

TABLE 20—REPRESENTATIVE COOLING AND HEATING LOAD HOURS AND THE CORRESPONDING SET OF SEASONAL HOURS FOR EACH GENERALIZED CLIMATIC REGION—Continued

Climatic region	Cooling load hours CLH_R	Heating load hours HLH_R	Cooling season hours CSH_R	Heating season hours HSH_R	Shoulder season hours SSH_R
VI	200	2,750	561	6,258	1,941

HSH is evaluated as: $HSH = \frac{HLH \cdot (65 - T_{OD})}{\sum_{j=1}^{J}(65 - T_j) \cdot \frac{n_j}{N}}$,

where T_{OD} and $\frac{n_j}{N}$ are listed in Table 18 and depend on the location of interest relative to Figure 1. For the six generalized climatic regions, this equation simplifies to the following set of equations:

Region I: $HSH = 2.4348 HLH$;
Region II: $HSH = 2.5182 HLH$;
Region III: $HSH = 2.5444 HLH$;
Region IV: $HSH = 2.5078 HLH$;
Region V: $HSH = 2.5295 HLH$;
Region VI: $HSH = 2.2757 HLH$;

SSH is evaluated: $SSH = 8760 - (CSH + HSH)$. where *CSH* = the cooling season hours calculated using $CSH = 2.8045 \cdot CLH$.

TABLE 21—FRACTIONAL BIN HOURS FOR THE SHOULDER SEASON HOURS FOR ALL REGIONS

Tj(°F)	Fractional bin hours	
	Air conditioners	Heat pumps
72	0.333	0.167
67	0.667	0.333
62	0	0.333
57	0	0.167

4.3.3 If a shoulder season crankcase heater time delay and/or a heating season crankcase heater time delay is specified by the manufacturer, multiply E1 and/or E2, by $\left(1 - \frac{t_{delay,i}}{60}\right)$, where $t_{delay,1}$ is the time delay for operation during the shoulder season and $t_{delay,2}$ is the time delay for operation during the heating season, in minutes. If a time delay is not specified, $t_{delay,i}$ is 15 minutes.

4.3.4 For air conditioners, the annual off mode energy consumption, E_{TOTAL}, is: $E_{TOTAL} = E1 + E2$.

4.3.5 For heat pumps, the annual off mode energy consumption, E_{TOTAL}, is *E1*.

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

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

$$APF_A = \frac{CLH_A \cdot \dot{Q}_c^k(95) + HLH_A \cdot DHR \cdot C}{\frac{CLH_A \cdot \dot{Q}_c^k(95)}{SEER} + \frac{HLH_A \cdot DHR \cdot C}{HSPF} + P1 \cdot SSH + P2 \cdot HSH},$$

Where,

CLH_A = the actual cooling hours for a particular location as determined using the map given in Figure 2, hr.

$\dot{Q}_c^k(95)$ = the space cooling capacity of the unit as determined from the A or A_2 Test, whichever applies, Btu/h.

HLH_A = the actual heating hours for a particular location as determined using the map given in Figure 1, hr.

DHR = the design heating requirement used in determining the HSPF; refer to section 4.2 and see section 1.2, Definitions, Btu/h.

C = defined in section 4.2 following Equation 4.2–2, dimensionless.

SEER = the seasonal energy efficiency ratio calculated as specified in section 4.1, Btu/W·h.

HSPF = the heating seasonal performance factor calculated as specified in section

4.2 for the generalized climatic region that includes the particular location of interest (see Figure 1), Btu/W·h. The HSPF should correspond to the actual design heating requirement (DHR), if known. If it does not, it may correspond to one of the standardized design heating requirements referenced in section 4.2.

P1 is the shoulder season per-compressor off mode power, as determined in section 3.13, W.

SSH is the shoulder season hours, hr.

P2 is the heating season per-compressor off mode power, as determined in section 3.13, W.

HSH is the heating season hours, hr.

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

$$APF_R = \frac{CLH_R \cdot \dot{Q}_c^k(95) + HLH_R \cdot DHR \cdot C}{\frac{CLH_R \cdot \dot{Q}_c^k(95)}{SEER} + \frac{HLH_R \cdot DHR \cdot C}{HSPF} + P1 \cdot SSH + P2 \cdot HSH},$$

Where,

CLH_R = the representative cooling hours for each generalized climatic region, Table 22, hr.

HLH_R = the representative heating hours for each generalized climatic region, Table 22, hr.

HSPF = the heating seasonal performance factor calculated as specified in section 4.2 for the each generalized climatic region and for each standardized design heating requirement within each region, Btu/W.h.

The SEER, $\dot{Q}_c^k(95)$, DHR, and C are the same quantities as defined in section 4.4.1. Figure 1 shows the generalized climatic regions.

TABLE 22—REPRESENTATIVE COOLING AND HEATING LOAD HOURS FOR EACH GENERALIZED CLIMATIC REGION

Region	CLH_R	HLH_R
I	2400	750
II	1800	1250
III	1200	1750
IV	800	2250
V	400	2750
VI	200	2750

4.5. Rounding of SEER, HSPF, and APF for reporting purposes. After calculating SEER according to section 4.1, HSPF according to section 4.2, and APF according to section 4.4, round the values off as specified in subpart B 430.23(m) of Title 10 of the Code of Federal Regulations.

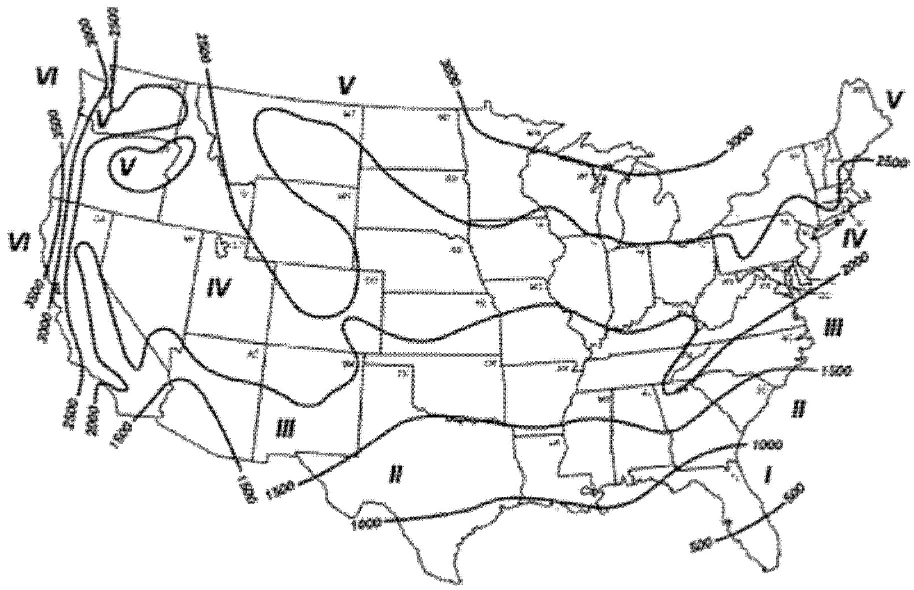

Figure 1—Heating Load Hours (HLH_A) for the United States

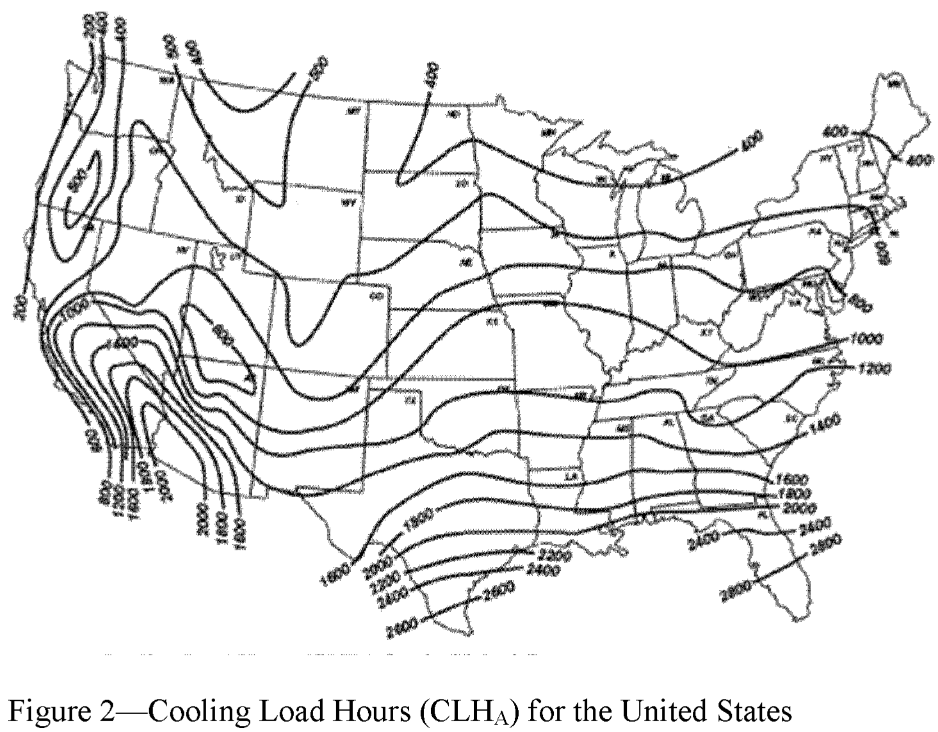

Figure 2—Cooling Load Hours (CLH_A) for the United States

4.6 Calculations of the SHR, which should be computed for different equipment configurations and test conditions specified in Table 23.

TABLE 23 APPLICABLE TEST CONDITIONS FOR CALCULATION OF THE SENSIBLE HEAT RATIO

Equipment configuration	Reference Table No. 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.

4.7 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. Add the letter identification for each steady-state test as a subscript (*e.g.,* EER_{A_2}) to differentiate among the resulting EER values.

■ 12. Section 430.32 is amended by revising paragraph (c) to read as follows:

§ 430.32 Energy and water conservation standards and their compliance dates.

* * * * *

(c) *Central air conditioners and heat pumps.* The energy conservation standards defined in terms of the heating seasonal performance factor are based on Region IV, the minimum standardized design heating requirement, and the provisions of 10 CFR 429.16 of this chapter.

(1) Each basic model of single-package central air conditioners and central air conditioning heat pumps and each individual combination of split-system central air conditioners and central air conditioning heat pumps manufactured on or after January 1, 2015, shall have a Seasonal Energy Efficiency Ratio and Heating Seasonal Performance Factor not less than:

Product class	Seasonal energy efficiency ratio (SEER)	Heating seasonal performance factor (HSPF)
(i) Split-system air conditioners	13	
(ii) Split-system heat pumps	14	8.2
(iii) Single-package air conditioners	14	
(iv) Single-package heat pumps	14	8.0
(v) Small-duct, high-velocity systems	12	7.2
(vi)(A) Space-constrained products—air conditioners	12	
(vi)(B) Space-constrained products—heat pumps	12	7.4

(2) In addition to meeting the applicable requirements in paragraph (c)(2) of this section, products in product class (i) of that paragraph (*i.e.,* split-system air conditioners) that are installed on or after January 1, 2015, and

installed in the States of Alabama, Arkansas, Delaware, Florida, Georgia, Hawaii, Kentucky, Louisiana, Maryland, Mississippi, North Carolina, Oklahoma, South Carolina, Tennessee, Texas, or Virginia, or in the District of Columbia, shall have a Seasonal Energy Efficiency Ratio not less than 14. The least efficient combination of each basic model must comply with this standard.

(3) In addition to meeting the applicable requirements in paragraphs (c)(2) of this section, products in product classes (i) and (iii) of paragraph (c)(2) (*i.e.,* split-system air conditioners and single-package air conditioners) that are installed on or after January 1, 2015, and installed in the States of Arizona, California, Nevada, or New Mexico shall have a Seasonal Energy Efficiency Ratio not less than 14 and have an Energy Efficiency Ratio (at a standard rating of 95 °F dry bulb outdoor temperature) not less than the following:

Product class	Energy efficiency ratio (EER)
(i) Split-system rated cooling capacity less than 45,000 Btu/hr	12.2
(ii) Split-system rated cooling capacity equal to or greater than 45,000 Btu/hr	11.7
(iii) Single-package systems	11.0

The least efficient combination of each basic model must comply with this standard.

(4) Each basic model of single-package central air conditioners and central air conditioning heat pumps and each individual combination of split-system central air conditioners and central air conditioning heat pumps manufactured on or after January 1, 2015, shall have an average off mode electrical power consumption not more than the following:

Product class	Average off mode power consumption $P_{W,OFF}$ (watts)
(i) Split-system air conditioners	30
(ii) Split-system heat pumps	33
(iii) Single-package air conditioners	30
(iv) Single-package heat pumps	33
(v) Small-duct, high-velocity systems	30
(vi) Space-constrained air conditioners	30
(vii) Space-constrained heat pumps	33

* * * * *

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