This document is scheduled to be published in the
Federal Register on 11/27/2015 and available online at
http://federalregister.gov/a/2015-30057, and on FDsys.gov

**[6450-01-P]**

**DEPARTMENT OF ENERGY**

**10 CFR Parts 429 and 430**

**[Docket No. EERE-2014-BT-TP-0014]**

**RIN: 1904-AD22**

**Energy Conservation Program: Test Procedures for Portable Air Conditioners**

**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 modify the test procedure proposals for portable air conditioners (ACs), initially presented in a notice of proposed rulemaking (NOPR) published on February 25, 2015. Upon further analysis and review of the public comments received in response to the February 2015 NOPR, DOE proposes in this supplemental notice of proposed rulemaking (SNOPR) the following additions and clarifications to its proposed portable AC test procedure: (1) minor revisions to the indoor and outdoor cooling mode test conditions; (2) an additional test condition for cooling mode testing; (3) updated infiltration air and capacity calculations to account for the second cooling mode test condition; (4) removal of the measurement of case heat transfer; (5) a clarification of test unit placement within the test chamber; (6) removal of the heating mode test procedure; (7) a revision

to the CEER calculation to reflect the two cooling mode test conditions and removal of heating mode testing; and (8) additional technical corrections and clarifications. These proposals are to be combined with the initial NOPR proposals and would be codified in a newly created appendix CC to title 10 of the Code of Federal Regulations (CFR), part 430, subpart B. The test procedures would be used to determine capacities and energy efficiency metrics that would be the basis for any future energy conservation standards for portable ACs.

**DATES:** DOE will accept comments, data, and information regarding this SNOPR, submitted no later than [**INSERT DATE 30 DAYS AFTER DATE OF PUBLICATION IN THE FEDERAL REGISTER**]. See section V, "Public Participation," for details.

**ADDRESSES:** Any comments submitted must identify the SNOPR for Test Procedures for Portable Air Conditioners, and provide docket number EERE-2014–BT–TP–0014 and/or regulatory information number (RIN) number 1904-AD22. Comments may be submitted using any of the following methods:

1. Federal eRulemaking Portal: www.regulations.gov. Follow the instructions for submitting comments.
2. E-mail: PortableAC2014TP0014@ee.doe.gov. Include the docket number and/or RIN in the subject line of the message.
3. Mail: Ms. Brenda Edwards, U.S. Department of Energy, Building Technologies Program, Mailstop EE-5B, 1000 Independence Avenue, SW., Washington, DC, 20585-0121. If possible, please submit all items on a CD. It is not necessary to include printed copies.

4. Hand Delivery/Courier: Ms. Brenda Edwards, U.S. Department of Energy, Building Technologies Program, 950 L'Enfant Plaza, SW., Room 6094, Washington, DC, 20024. Telephone: (202) 586-2945. If possible, please submit all items on a CD. 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: http://www.regulations.gov/#!docketDetail;D=EERE-2014-BT-TP-0014 . 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, "Public Participation," for information on how to submit comments through www.regulations.gov.

For further information on how to submit a comment, or review other public comments and the docket, contact Ms. Brenda Edwards at (202) 586-2945 or by email: Brenda.Edwards@ee.doe.gov.

**FOR FURTHER INFORMATION CONTACT:**

Mr. Bryan Berringer, U.S. Department of Energy, Office of Energy Efficiency and Renewable Energy, Building Technology Office, EE-5B, 1000 Independence Ave., SW, Washington, DC 20585–0121. Telephone: 202-586-0371. E-mail: Bryan.Berringer@ee.doe.gov.

Ms. Sarah Butler, U.S. Department of Energy, Office of the General Counsel, Mailstop GC-33, 1000 Independence Ave., SW, Washington, D.C. 20585-0121. Telephone: 202-586-1777; E-mail: Sarah.Butler@hq.doe.gov.

**SUPPLEMENTARY INFORMATION:** DOE intends to incorporate by reference the following industry standard into 10 CFR parts 429 and 430: AHAM PAC-1-2015, Portable Air Conditioners. DOE also intends to incorporate by reference the following industry standard into 10 CFR part 430: ANSI/ASHRAE Standard 37-2009, Methods of Testing for Rating Electrically Driven Unitary Air-Conditioning and Heat Pump Equipment.

Copies of AHAM PAC-1-2015 can be obtained from the Association of Home Appliance Manufacturers 1111 19th Street NW, Suite 402, Washington, DC 20036, 202-872-5955, or by going to http://www.aham.org/ht/d/ProductDetails/sku/PAC12009/from/714/pid/.

Copies of ANSI/ASHRAE Standard 37-2009 can be obtained from the American National Standards Institute 25 W 43rd Street, 4th Floor, New York, NY, 10036, 212-642-4980, or by going to http://webstore.ansi.org/RecordDetail.aspx?sku=ANSI%2FASHRAE+Standard+37-2009.

See section IV.B. for a description of these standards.

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VI. Approval of the Office of the Secretary

## I. Authority and Background

Title III of the Energy Policy and Conservation Act (EPCA), as amended (42 U.S.C. 6291, et seq.; "EPCA" or, "the Act") sets forth various provisions designed to improve energy efficiency. Part A of title III of EPCA (42 U.S.C. 6291–6309) establishes the "Energy Conservation Program for Consumer Products Other Than Automobiles," which covers consumer products and certain commercial products (hereinafter referred to as "covered products").[1] EPCA authorizes DOE to establish technologically feasible, economically justified energy conservation standards for covered products or equipment that would be likely to result in significant national energy savings. (42 U.S.C. 6295(o)(2)(B)(i)(I)–(VII)) In addition to specifying a list of covered consumer and industrial products, EPCA contains provisions that enable the Secretary of Energy to classify additional types of consumer products as covered products. (42 U.S.C. 6292(a)(20)) For a given product to be classified as a covered product, the Secretary must determine that:

(1) Classifying the product as a covered product is necessary for the purposes of EPCA; and

(2) The average annual per-household energy use by products of each type is likely to exceed 100 kilowatt-hours (kWh) per year. (42 U.S.C. 6292(b)(1))

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

To prescribe an energy conservation standard pursuant to 42 U.S.C. 6295(o) and (p) for covered products added pursuant to 42 U.S.C. 6292(b)(1), the Secretary must also determine that:

(1) The average household energy use of the products has exceeded 150 kWh per household for a 12-month period;

(2) The aggregate 12-month energy use of the products has exceeded 4.2 terawatt-hours (TWh);

(3) Substantial improvement in energy efficiency is technologically feasible; and

(4) Application of a labeling rule under 42 U.S.C. 6294 is unlikely to be sufficient to induce manufacturers to produce, and consumers and other persons to purchase, covered products of such type (or class) that achieve the maximum energy efficiency that is technologically feasible and economically justified. (42 U.S.C. 6295(l)(1))

Under EPCA, the energy conservation program consists essentially of four parts: (1) testing, (2) labeling, (3) Federal energy conservation standards, and (4) certification and enforcement procedures. The testing requirements consist of test procedures that manufacturers of covered products must use as the basis for: (1) certifying to DOE that their products comply with the applicable energy conservation standards adopted under EPCA, and (2) making representations about the efficiency of those products. Similarly, DOE must use these test procedures to determine whether the products comply with any relevant standards promulgated under EPCA.

A. General Test Procedure Rulemaking Process

Under 42 U.S.C. 6293, EPCA sets forth the criteria and procedures DOE must follow when prescribing or amending test procedures for covered products. EPCA provides in relevant part that any test procedures prescribed or amended under this section shall be reasonably designed to produce test results that 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)) In addition, if DOE determines that a test procedure should be prescribed or amended, 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))

B. Test Procedure for Portable Air Conditioners

There are currently no DOE test procedures or energy conservation standards for portable ACs. On July 5, 2013, DOE issued a notice of proposed determination (NOPD) of coverage (hereinafter referred to as the "July 2013 NOPD"), in which DOE announced that it tentatively determined that portable ACs meet the criteria under 42 U.S.C. 6292(b)(1) to be classified as a covered product. 78 FR 40403. DOE estimated that approximately 974,000 portable AC units were shipped in North America in 2012, and projected that approximately 1.74 million units would be shipped in 2018, representing nearly 80-percent growth in 6 years.[2] Id. at 40404. In addition, DOE estimated the average per-household portable AC electricity consumption for those homes with portable ACs to be approximately 650 kWh per year. Id.

In response to the July 2013 NOPD, DOE received comments from interested parties on several topics regarding appropriate test procedures for portable ACs that DOE should consider if it issues a final determination classifying portable ACs as a covered product.

1. The May 2014 NODA

On May 9, 2014, DOE published in the Federal Register a notice of data availability (NODA) (hereinafter referred to as the "May 2014 NODA"), in which it agreed that a DOE test procedure for portable ACs would provide consistency and clarity for representations of energy use of these products. DOE evaluated available industry test procedures to determine whether such methodologies would be suitable for incorporation in a future DOE test procedure, should DOE determine to classify portable ACs as a covered product. DOE conducted testing on a range of portable ACs to determine typical cooling capacities and cooling energy efficiencies based on the existing industry test methods and other modified approaches for portable ACs. 79 FR 26639, 26640 (May 9, 2014).

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[2] Transparency Media Research, "Air Conditioning Systems Market - Global Scenario, Trends, Industry Analysis, Size, Share and Forecast, 2012 – 2018," January 2013.

2. The February 2015 NOPR

On February 25, 2015, DOE published in the Federal Register a notice of proposed rulemaking (NOPR) (hereinafter referred to as the "February 2015 NOPR"), in which it proposed test procedures for portable ACs that would provide a means of determining efficiency in various operating modes, including cooling mode, heating mode, off-cycle mode, standby mode, and off mode. 80 FR 10211. For cooling mode and heating mode, DOE proposed test procedures based on the then-current industry-accepted test procedure, Association of Home Appliance Manufacturers (AHAM) PAC-1-2014, "Portable Air Conditioners," with additional provisions to account for heat transferred to the indoor conditioned space from the case, ducts, and any infiltration air from unconditioned spaces. DOE also proposed various clarifications for cooling mode and heating mode testing, including: (1) test duct configuration; (2) instructions for condensate collection; (3) control settings for operating mode, fan speed, temperature set point, and louver oscillation; and (4) unit placement within the test chamber. For off-cycle mode, DOE proposed a test procedure that would measure portable AC energy use when the ambient dry-bulb temperature is at or below the setpoint. DOE also identified relevant low-power modes, proposed definitions for inactive mode and off mode, and proposed test procedures to determine representative energy consumption for these modes. Id.

In the February 2015 NOPR, DOE proposed to use a combined energy efficiency ratio (CEER) metric for representing the overall energy efficiency of single-duct and dual-duct portable ACs. The CEER metric would represent energy use in all available operating modes. DOE also proposed a cooling mode-specific CEER for units that do not provide a heating function to provide a basis for comparing performance with other cooling products such as room

ACs. In addition, DOE proposed separate energy efficiency ratio (EER) metrics for determining energy efficiency in cooling mode and heating mode only. 80 FR 10211, 10234–10235 (Feb. 25, 2015).

DOE also recently initiated a separate rulemaking to consider establishing energy conservation standards for portable ACs. Any new standards would be based on the same efficiency metrics derived from the test procedure that DOE would adopt in a final rule in this rulemaking.

## II. Synopsis of the Supplemental Notice of Proposed Rulemaking

Upon further analysis and review of the public comments received in response to the February 2015 NOPR, DOE proposes in this SNOPR the following additions and clarifications to its proposed portable AC test procedure: (1) minor revisions to the indoor and outdoor cooling mode test conditions; (2) an additional test condition for cooling mode testing; (3) updated infiltration air and capacity calculations to account for the second cooling mode test condition; (4) removal of the measurement of case heat transfer; (5) a clarification of test unit placement within the test chamber; (6) removal of the heating mode test procedure; (7) a revision to the CEER calculation to reflect the two cooling mode test conditions and removal of heating mode testing; and (8) additional technical corrections and clarifications.

Other than the specific amendments newly proposed in this SNOPR, DOE continues to propose the test procedure originally included in the February 2015 NOPR. For the reader's convenience, DOE has reproduced in this SNOPR the entire body of proposed regulatory text from the February 2015 NOPR, amended as appropriate according to these proposals. DOE's supporting analysis and discussion for the portions of the proposed regulatory text not affected by this SNOPR may be found in the February 2015 NOPR. 80 FR 10211 (Feb. 25, 2015).

## III. Discussion

### A. Active Mode

In the February 2015 NOPR, DOE proposed to define active mode, for purposes of the portable AC test procedure, as a mode in which the portable AC is connected to a mains power source, has been activated, and is performing the main functions of cooling or heating the conditioned space, circulating air through activation of its fan or blower without activation of the refrigeration system, or defrosting the refrigerant coil. 80 FR 10211, 10216 (Feb. 25, 2015). DOE has determined that the existing statutory definition of "active mode" is sufficient for purposes of this test procedure and therefore is no longer proposing a separate definition of "active mode" for portable ACs.

### B. Cooling Mode

In the February 2015 NOPR, DOE proposed that cooling mode is a mode in which a portable AC has activated the main cooling function according to the thermostat or temperature sensor signal, including activating the refrigeration system or the fan or blower without activation of the refrigeration system. 80 FR 10211, 10217 (Feb. 25, 2015). DOE determined that the existing industry standards used to measure portable AC cooling capacity and EER, which are based on air enthalpy methods, may not represent true portable AC performance. Additionally, DOE is aware that manufacturers may test according to different industry standards, causing confusion and variation in the reported cooling capacities and EERs for units currently on the market. DOE further concluded that varying infiltration air flow rates and heat losses would preclude a fixed translation factor that could be applied to the results of an air enthalpy measurement to account for the impact of these effects. Therefore, although DOE generally proposed a test procedure for portable ACs based on AHAM PAC-1-2014, the industry-accepted standard for testing portable ACs (which is based on an air enthalpy approach), the proposed test procedure incorporated infiltration air effects and heat losses to more accurately measure performance representative of typical operation and provide a clear and consistent basis for comparison of portable AC capacity and energy use. 80 FR 10211, 10222–10223 (Feb. 25, 2015).

The Appliance Standards Awareness Project (ASAP), Alliance to Save Energy (ASE), American Council for an Energy-Efficient Economy (ACEEE), National Consumer Law Center (NCLC), Natural Resources Defense Council (NRDC), and Northwest Energy Efficiency Alliance (NEEA) (hereinafter the “Joint Commenters”) and the Pacific Gas and Electric Company (PG&E), Southern California Gas Company (SCGC), Southern California Edison (SCE), and San Diego Gas and Electric Company (SDG&E) (hereinafter the “California IOUs”) supported DOE’s proposal to adopt AHAM PAC-1-2014 with modifications to account for the impacts of infiltration air and heat transfer from the duct(s) and case, as this would better reflect real-world performance of both single-duct and dual-duct portable ACs. (Joint Commenters, No. 19 at p. 1; California IOUs, No. 20 at p. 1)[3] The Joint Commenters further noted that in response to the NODA, they had encouraged DOE to adopt a test procedure based on the calorimeter approach. In light of the data presented in the February 2015 NOPR, the Joint Commenters now support the proposal to base a DOE portable AC test procedure on AHAM PAC-1-2014 as there is a good correlation with the calorimeter test results when the proposed adjustments that account for the impact of infiltration air and duct and case heat transfer are applied. (Joint Commenters, No. 19 at p. 2)

China WTO/TBT National Notification & Enquiry Center (China) noted that, compared to the industry-accepted and commonly used American National Standards Institute (ANSI)/American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE) Standard 128-2001, "Method of Rating Unitary Spot Air Conditioners," AHAM PAC-1-2014 is significantly more complex, increases the cost of testing, and would require laboratories to purchase new instrumentation and update or reconstruct their chambers. Further, China stated that DOE did not provide a comparison between AHAM PAC-1-2014 and

[3] A notation in the form "Joint Commenters, No. 19 at p. 1" identifies a written comment: (1) made by the Appliance Standards Awareness Project, Alliance to Save Energy, American Council for an Energy-Efficient Economy, National Consumer Law Center, Natural Resources Defense Council, and Northwest Energy Efficiency Alliance (the "Joint Commenters"); (2) recorded in document number 19 that is filed in the docket of this test procedure rulemaking (Docket No. EERE–2014– BT–TP–0014) and available for review at www.regulations.gov; and (3) which appears on page 1 of document number 19.

ANSI/ASHRAE 128-2001 based on test data. Without a comparison of the results, China does not believe that DOE can conclude there is a marked difference between the two, and cannot determine that testing according to AHAM PAC-1-2014 is necessary. China requested that DOE provide comparative data between the two test procedures. (China, No. 15 at pp. 3–4)

De' Longhi Appliances s.r.l. (De' Longhi) claimed that in the United States, most manufacturers are using the standard ANSI/ASHRAE 128-2001 to rate the performance of single-duct portable ACs. De' Longhi stated, however, that testing a single-duct portable AC according to AHAM PAC-1-2014 results in a cooling capacity about 25 percent lower than the rating obtained with ANSI/ASHRAE 128-2001. Despite this rated cooling capacity reduction, De' Longhi supports the use of AHAM PAC-1-2014 because it ensures more reliable and repeatable testing data. (De' Longhi, No. 16 at pp. 1–2)

AHAM and De' Longhi support the use of AHAM PAC-1-2014 as the basis for a DOE test procedure for portable ACs, albeit without the addition of certain test procedure provisions that DOE has proposed. (Public Meeting Transcript, AHAM, No. 13 at p. 31; Public Meeting Transcript, De' Longhi, No. 13 at pp. 13, 33; AHAM, No. 18 at p. 2; De' Longhi, No. 16 at p. 2)[4]

[4] A notation in the form "AHAM, Public Meeting Transcript, No. 13 at p. 31" identifies an oral comment that DOE received on March 18, 2015 during the NOPR public meeting, was recorded in the public meeting transcript in the docket for this test procedure rulemaking (Docket No. EERE-2014-BT-TP-0014). This particular notation refers to a comment (1) made by the Association of Home Appliance Manufacturers during the public meeting; (2) recorded in document number 13, which is the public meeting transcript that is filed in the docket of this test procedure rulemaking; and (3) which appears on page 31 of document number 13.

DOE agrees that certain portable ACs may be currently tested according to ANSI/ASHRAE 128-2001, but believes this is largely due to California's regulations for certifying spot coolers sold in that State. As discussed in the February 2015 NOPR, DOE is not proposing testing procedures for spot coolers at this time. 80 FR 10212, 10214–15 (Feb. 25, 2015). In addition, ANSI/ASHRAE 128-2001 is an obsolete version of that test standard, and DOE expects that manufacturers conducting testing of their portable ACs for reasons other than certification in California may be using a current, industry-accepted test standard such as AHAM PAC-1-2014 or ANSI/ASHRAE 128-2011, both of which were discussed and analyzed in the May 2014 NODA and February 2015 NOPR. For these reasons, and with the general support of interested parties, DOE continues to propose a test procedure for portable ACs that is based on the current version of AHAM PAC-1. DOE notes that AHAM issued a new version of PAC-1 in 2015, with no changes in language from the 2014 version. Therefore, although DOE previously proposed to adopt a test procedure for portable ACs that is based on AHAM PAC-1-2014, DOE now proposes in this SNOPR to reference the identical updated version, AHAM PAC-1-2015, in the proposed DOE portable AC test procedure. Accordingly, DOE refers to AHAM PAC-1-2015 for the remainder of this SNOPR when discussing its current proposals.

Additionally, this notice discusses other modifications to the test procedure proposed in the February 2015 NOPR to address commenters' concerns, improve repeatability, minimize test burden, and ensure the test procedure is representative of typical consumer usage.

1. Test Chamber and Infiltration Air Conditions

DOE proposed in the February 2015 NOPR to utilize the following ambient conditions presented in Table III.1 below, based on those test conditions specified in Table 3, "Standard Rating Conditions," of AHAM PAC-1-2014. DOE also proposed to determine test configurations according to Table 2 of AHAM PAC-1-2014, with Test Configuration 3 applicable to dual-duct portable ACs and Test Configuration 5 applicable to single-duct portable ACs. 80 FR 10211, 10226 (Feb. 25, 2015). For single-duct units, the condenser inlet conditions are the same as the evaporator inlet. For dual-duct units, the condenser inlet air conditions are monitored at the interface between the condenser inlet duct and outdoor test room.

**Table III.1 Standard Rating Conditions - Cooling Mode – NOPR Proposal**

| Test Configuration | Evaporator Inlet Air, °F (°C) | | Condenser Inlet Air, °F (°C) | |
|---|---|---|---|---|
| | Dry Bulb | Wet Bulb | Dry Bulb | Wet Bulb |
| 3 | 80.6 (27) | 66.2 (19) | 95.0 (35) | 75.2 (24) |
| 5 | 80.6 (27) | 66.2 (19) | 80.6 (27) | 66.2 (19) |

a. Test Chamber Conditions

In the February 2015 NOPR, DOE noted that the AHAM PAC-1-2014 test conditions are slightly different from the AHAM PAC-1-2009 test conditions, which AHAM revised to harmonize with the temperatures specified in Canadian Standards Association (CSA) C370-2013, "Cooling Performance of Portable Air Conditioners" and ANSI/ASHRAE Standard 128-2011, "Method of Rating Portable Air Conditioners." DOE's analysis and testing was conducted in accordance with AHAM PAC-1-2009, as the next version of the standard, AHAM PAC-1-2014, had not yet been finalized.  DOE tentatively determined that the test condition differences between the 2009 and 2014 versions of AHAM PAC-1 would not substantively impact test results. Therefore, DOE proposed to use the updated test conditions from AHAM PAC-1-2014.

DOE also noted in the February 2015 NOPR that these conditions are close, but not identical, to those required by the DOE room AC test procedure (80 degrees Fahrenheit (°F) dry-bulb temperature and 67 °F wet-bulb temperature on the indoor side, and 95 °F dry-bulb temperature and 75 °F wet-bulb temperature on the outdoor side, consistent with the AHAM PAC-1-2009 conditions). 80 FR 10211, 10226 (Feb. 25, 2015).

AHAM agreed that there are no major differences between the 2009 and 2014 versions, and that the main changes were editorial in nature to harmonize with the Canadian test procedure. AHAM stated that it is important that the North American and Canadian methods are harmonized. (Public Meeting Transcript, AHAM, No. 13 at pp. 31–32)

DENSO Products and Services Americas, Inc. (DENSO) commented that the room AC indoor test conditions in the DOE test procedure for those products correspond to about 50-percent relative humidity, whereas the AHAM PAC-1-2014 indoor test conditions are closer to 40-percent relative humidity. According to DENSO, this is a significant difference in test conditions and thus the AHAM PAC-1-2014 test conditions are not comparable to those for room ACs or other air conditioning products. DENSO also commented that the test conditions should be expressed in whole degrees instead of three-digit dry-bulb and wet-bulb temperatures in °F that are equivalent to whole degrees Celsius in other standards. (Public Meeting Transcript, DENSO, No. 13 at pp. 47–48, 69–70; DENSO, No. 14 at p. 2)

In response to the comments received regarding the chamber test conditions, DOE examined the relative impact of the varying latent heat differential between the indoor and outdoor conditions in the February 2015 NOPR proposal and in AHAM PAC-1-2009. The latent heat differential impacts cooling capacity primarily through the effects of infiltration air. Based on the average dry air mass flowrate for the single-duct and dual-duct units in DOE's test sample, DOE estimated that the change in test conditions from the 2009 to either the 2014 or 2015 version of AHAM PAC-1 would decrease cooling capacity by increasing the heating effect due to infiltration air by an average of 755 Btu/h and 330 Btu/h for the two configurations, respectively. With an average PAC-1-2009 cooling capacity (without accounting for infiltration air, case, or duct heat effects) of 7,650 Btu/h for single-duct units and 6,800 Btu/h for dual-duct units, adjusting the test conditions from the 2009 to 2015 version of AHAM PAC-1 would decrease cooling capacity by 5–10 percent, an amount which DOE considers to be significant. Therefore, DOE no longer concludes that the test condition differences between the 2009 and 2014 (and, thus, 2015) versions of AHAM PAC-1 would not substantively impact test results.

DOE further notes that the test conditions in AHAM PAC-1-2015, although harmonized with those in CSA C370-2013 and ANSI/ASHRAE Standard 128-2011, do not align with the test conditions in the DOE test procedures for other cooling products, particularly room ACs and central ACs. As noted earlier in this section, the AHAM PAC-1-2015 test approach is generally appropriate for portable ACs. However, DOE believes that the test conditions in AHAM PAC-1-2009, which align with the conditions used for testing other DOE covered products, are more appropriate for testing portable AC performance than those in AHAM PAC-1-2015. The temperatures specified in AHAM PAC-1-2015 were rounded to produce whole degrees Celsius, which results in a relative humidity on the indoor side (47.0 percent) that differs significantly from the relative humidity that DOE has previously determined for room ACs and central ACs is representative of a residential air-conditioned space (51.1 percent). To maintain consistency among products with similar functions, DOE proposes in this SNOPR to revise the test conditions proposed in the February 2015 NOPR to those presented in Table III.2 below, which would replace the test conditions specified in Table 3, "Standard Rating Conditions," of AHAM PAC-1-2015. As discussed in the next section, however, these revisions do not comprise the only changes that DOE is proposing in this SNOPR to the rating conditions for portable ACs.

**Table III.2 Revised Standard Rating Conditions - Cooling Mode**

| Test Configuration | Evaporator Inlet Air, °F (°C) | | Condenser Inlet Air, °F (°C) | |
|---|---|---|---|---|
| | Dry Bulb | Wet Bulb | Dry Bulb | Wet Bulb |
| 3 | 80 (26.7) | 67 (19.4) | 95 (35) | 75 (23.9) |
| 5 | 80 (26.7) | 67 (19.4) | 80 (26.7) | 67 (19.4) |

b. Infiltration Air Conditions

In the February 2015 NOPR, DOE noted that infiltration from outside the conditioned space occurs due to the negative pressure induced as condenser air is exhausted to the outdoor space. Although this effect is most pronounced for single-duct units, which draw all of their condenser air from within the conditioned space, dual-duct units also draw a portion of their condenser air from the conditioned space. DOE proposed calculating the infiltration air flow rate as the condenser exhaust flow rate to the outdoor chamber minus any condenser intake flow rate from the outdoor chamber. DOE proposed that the infiltration air conditions be 95 °F dry-bulb temperature and 75.2 °F wet-bulb temperature, consistent with the outdoor conditions specified in AHAM PAC-1-2014. 80 FR 10211, 10224–10225 (Feb. 25, 2015).

The Joint Commenters supported the proposal to use 95 °F dry-bulb temperature and 75 °F wet-bulb temperature outdoor air. (Public Meeting Transcript, ASAP, No. 13 at p. 44; Joint Commenters, No. 19 at p. 2) The Joint Commenters further stated that because AHAM PAC-1-2014 is conducted using these outdoor air conditions, it is important that the same conditions be used for the infiltration air to reflect the real-world performance of portable ACs under these outdoor air conditions. The Joint Commenters noted that all infiltration air is ultimately coming from the outdoors and adding heat to the home where the portable AC is installed. The Joint Commenters suspect that, in many cases, the bulk of the infiltration air will be coming directly from the outdoors due to imperfect installations, resulting in leaks through the window where the portable AC is installed. The Joint Commenters also suspect that over time, a greater portion of the infiltration air will come directly through the window where the portable AC is installed due to deterioration of the installation as the unit is repeatedly removed and re-installed. (Joint Commenters, No. 19 at p. 2)

De' Longhi did not agree with DOE's proposed approach to address infiltration air, stating that it would improperly represent the performance of single-duct products because the proposed infiltration air conditions of 95 °F dry-bulb temperature and 75.2 °F wet-bulb temperature represent worst-case outdoor conditions which occur for a negligible period of time during the cooling season. De' Longhi noted that according to ANSI/Air-Conditioning, Heating, and Refrigeration Institute (AHRI) 210/240, "Performance Rating of Unitary Air-Conditioning and Air-Source Heat Pump Equipment", outdoor temperatures ranging from 95 to 104 °F represent just 2.2 percent of the season while outdoor temperatures range from 65 to 80 °F during 66.1 percent of the season. De' Longhi stated that selection of an appropriate outdoor temperature for rating testing is critical for single-duct portable ACs. As a consequence, De' Longhi commented that DOE's proposed procedure overstates the impacts of infiltration air. (Public Meeting Transcript, De' Longhi, No. 13 at pp. 39–40; De' Longhi, No. 16 at p. 3)

The National Association of Manufacturers (NAM) stated that if the test procedure includes an infiltration air adjustment, the temperature must be representative and based on data. In NAM's view, given the uniqueness of homes, the proposed infiltration air temperatures are not practical, nor are they shown to be based on available data. (NAM, No. 17 at p. 2)

AHAM commented that portable ACs are not used just on the hottest summer days, but also during the transition periods before and after summer to cool only a certain room or rooms before central air conditioning or heating is turned on. According to AHAM, this use pattern suggests that an outdoor temperature representing the hottest days of summer is not representative of consumer use. AHAM commented that even if consumers use portable ACs only in the summer and only the outdoor air temperature is considered, a 95 °F infiltration air temperature would still be too high. (AHAM, No. 18 at p. 4)

De' Longhi and AHAM suggested that, should DOE include a numerical adjustment for infiltration air to the results of testing with AHAM PAC-1-2014, the proper temperature for the infiltration air would be 70 °F, based on available data. They noted that 70 °F is the representative average cooling season temperature that DOE found for the United States as a whole. They also claimed that according to ANSI/AHRI 210/240-2008, an outdoor temperature of 70 °F represents 50 percent of the total cooling season hours. (Public Meeting Transcript, De'

Longhi, No. 13 at p. 41; De' Longhi, No. 16 at p. 3; AHAM, No. 18 at p. 4) De' Longhi further stated that if DOE decides not to use 70 °F as the outdoor air temperature, this test condition should be no greater than 80.6 °F dry-bulb, the standard rating condition for single-duct portable ACs in AHAM PAC-1-2014 for both indoor and outdoor conditions. In order to compare single-duct and dual-duct portable ACs under the same conditions, De' Longhi would also accept 80.6 °F as the outdoor conditions for dual-duct units as well. (Public Meeting Transcript, De' Longhi, No. 13 at pp. 43–44; De' Longhi, No. 16 at p. 4)

Friedrich commented that 70 °F is low for an outdoor temperature that would necessitate AC use, and suggested DOE consider 80 °F as the outdoor condition. (Public Meeting Transcript, Friedrich, No. 13 at pp. 84–85)

In addition to the proposed temperatures for infiltration air, DOE received comments regarding the likely origin of the infiltration air to help inform the appropriate infiltration air conditions. De' Longhi noted that it is possible that some or all of the replacement air is drawn from a location other than the outdoors directly, such as a basement, attic, garage, or a space that is conditioned by other equipment. Thus, De' Longhi stated that DOE's proposed approach is unrealistic, as the building spaces from which infiltration air may be drawn and other inside air that may be cooled by central cooling systems must be taken into account. De' Longhi also commented that DOE's approach did not account for any internal heating loads, solar radiation, or thermal lag of the building itself. (Public Meeting Transcript, De' Longhi, No. 13 at pp. 41–43; De' Longhi, No. 16 at pp. 3–4)

AHAM agreed with De' Longhi, and noted that even if all air in a home originates from outdoors, the infiltration air may be cooled once indoors. Moreover, AHAM noted that the infiltration air could be at different temperatures for a portable AC that is moved from room to room—for example, the air in a garage is not likely the same temperature as the air in an attic or basement. AHAM commented that if DOE accounts for the effects of infiltration air, DOE must ensure that the temperature is representative and based on data. In AHAM's view, given the uniqueness of homes, that is not practical to do. (AHAM, No. 18 at pp. 3–4)

AHAM, NAM, and DENSO stated that should DOE nevertheless move forward with its proposal, it must ensure it selects a representative test temperature for that infiltration air. They commented that DOE's current proposal is not representative and should be revised. (AHAM, No. 18 at p. 1; NAM, No. 17 at p. 3; DENSO, No. 14 at p. 3)

In response to comments received on the February 2015 NOPR, DOE conducted additional analysis to ensure the DOE test procedure for portable ACs is representative of typical cooling product operation and consumer usage. On the matter of the source of infiltration air, DOE reviewed information developed on infiltration air flow rates and sources for room ACs, which encounter issues for sealing in windows similar to portable ACs.  In a study conducted by the National Renewable Energy Laboratory (NREL)[5], infiltration air flow rates around the louvers on either side of three room AC test units and the air flow rates through the units themselves were measured when the units were installed in a test chamber outfitted with two residential single-hung windows. The units, including the side louvers, were installed per manufacturer instructions (i.e., no additional sealing around the louvers was provided). A variable-speed blower was used to vary the differential pressure between the test chamber and ambient (outdoor condition) from 0 to 50 Pascals (Pa). NREL found that at 50 Pa, the infiltration air flow rates around the louvers ranged from approximately 50 to 90 standard cubic feet per minute (SCFM) among the three test units. These infiltration air flow rates represented as much as two thirds of the rated evaporator air flow rates at high fan speed, and similarly would also represent a substantial percentage of the infiltration air for a single-duct portable AC. NREL estimated that the infiltration air leakage path around the louvers was the equivalent of a 27 to 42 square-inch hole in the wall. Because DOE observed that the window brackets for mounting the portable AC duct(s) in its test sample typically did not include any gasket, tape, or other sealing material, it concludes that outdoor air leaking through the portable AC's window bracket likely also represents the source of a substantial percentage of the infiltration air for portable ACs. Additionally, because portable ACs that do not draw all of the condenser air from outside the conditioned space create net negative pressure within the conditioned space,

[5] Winkler, J., et al., 2013. "Laboratory Performance Testing of Residential Window Air Conditioners," National Renewable Energy Laboratory, Technical Report NREL/TP-5500-57617, March 2013.

infiltration air flow is likely greater than for room ACs. Therefore, DOE continues to conclude that infiltration air temperature is best represented as the outdoor test condition.

DOE also notes that the temperature of infiltration air from sources other than the window bracket cannot be definitively characterized because the air temperature in the other locations may be greater than (e.g., an attic) or less than (e.g., a basement) the outdoor temperature. In addition, infiltration air that is drawn from other conditioned space initially originated from locations that could also be direct sources of infiltration air for a portable AC, and thus DOE believes that the portable AC should not derive a de facto benefit by being rated at a lower infiltration air temperature achieved via the energy consumption of other conditioning equipment.

DOE next considered commenters' suggestion that the outdoor test condition in the current version of AHAM PAC-1 may not be representative of a significant portion of portable AC operation. DOE revisited its climate analysis from the February 2015 NOPR to determine the overall average dry-bulb temperature and relative humidity during hours allotted for cooling mode operation, in locations where portable ACs are likely to be used. DOE again performed this climate analysis using 2012 hourly ambient temperature data from the National Climatic Data Center (NCDC) of the National Oceanic and Atmospheric Administration (NOAA), collected at weather stations in 44 representative states. DOE determined the average temperature and humidity associated with the hottest 750 hours for each state for which there was data available. DOE then reviewed data from the 2009 Residential Energy Consumption Survey (RECS)[6] to identify room AC ownership in the different geographic regions because no portable AC-specific usage data were available. Based on the RECS ownership data, DOE used a weighted-average approach to combine the average temperature and humidity for each individual state into sub-regional, regional, and finally, the representative national average temperature and humidity for the hottest 750 hours in each state.[7] DOE found that the national average dry-bulb temperature and relative humidity associated with the hottest 750 hours are 83 °F and 45 percent, respectively.

To maintain harmonization with other cooling products and the AHAM PAC-1-2009 test conditions, as discussed previously, and to continue to consider cooling performance under a rating condition at which product performance is most important to consumers, DOE proposes to specify the outdoor test conditions and associated infiltration air conditions of 95 °F dry-bulb and 75 °F wet-bulb temperature. However, DOE also proposes in this SNOPR that a second cooling mode test be conducted for dual-duct units (Test Configuration 3) at outdoor test conditions that reflect the weighted-average temperature and humidity observed during the hottest 750 hours (the hours during which DOE expects portable ACs to operate in cooling

[6] RECS data are available online at: http://www.eia.gov/consumption/residential/data/2009/"www.eia.gov/consumption/residential/data/2009/.
[7] For more information on the weighted-average approach that DOE conducted for this analyses, see the February 2015 NOPR. 80 FR 10211, 10235–27 (Feb. 25, 2015).

mode): 83 °F dry-bulb temperature and 67.5 °F wet-bulb temperature. For single-duct units (Test Configuration 5), DOE would specify a second set of numerical calculations for cooling capacity and CEER based on adjustments for infiltration air at these same conditions, rather than providing for an additional test at the weighted-average outdoor temperature and humidity. In sum, Table III.3 shows the complete set of cooling mode rating conditions that DOE proposes for portable ACs in this SNOPR.

**Table III.3 Standard Rating Conditions – Cooling Mode – SNOPR Proposal**

| Test Configuration | Evaporator Inlet Air, °F (°C) | | Condenser Inlet Air, °F (°C) | |
|---|---|---|---|---|
| | Dry Bulb | Wet Bulb | Dry Bulb | Wet Bulb |
| 3 (Condition A) | 80 (26.7) | 67 (19.4) | 95 (35) | 75 (23.9) |
| 3 (Condition B) | 80 (26.7) | 67 (19.4) | 83 (28.3) | 67.5 (19.7) |
| 5 | 80 (26.7) | 67 (19.4) | 80 (26.7) | 67 (19.4) |

c. Infiltration Air Calculations

In the February 2015 NOPR, DOE proposed that the sensible and latent components of infiltration air heat transfer be calculated using the evaporator inlet conditions, to be representative of the indoor room's ambient conditions. As discussed above, DOE proposed that the nominal indoor test chamber conditions for portable AC testing would be 80 °F dry-bulb temperature and 67 °F wet-bulb temperature, resulting in a humidity ratio of 0.0112 pounds of water per pounds of dry air ($lb_w/lb_{da}$). DOE further proposed in the February 2015 NOPR that the indoor test chamber dry-bulb and wet-bulb temperature conditions be maintained within a range of 1.0 °F, with an average difference of 0.3 °F. 80 FR 10211, 10224, 10226 (Feb. 25, 2015).

DOE notes that the allowable tolerances for the indoor evaporator inlet conditions would permit variations in the humidity ratio of up to 8.6 percent. DOE reviewed its test data and found that the maximum variation between the measured and proposed humidity ratio was 4.5 percent. DOE believes that the proposal to use the measured evaporator inlet conditions (dry-bulb and wet-bulb temperatures and the resulting humidity ratio) when calculating the impacts of infiltration air heat transfer may introduce variability in the test results due to the sensitivity of infiltration air to the allowable evaporator inlet conditions variability and the resulting impact on overall cooling capacity. Therefore, DOE proposes in this SNOPR to calculate the sensible and latent heat contributions of infiltration air using the nominal test chamber temperatures and subsequent humidity ratio to reduce test variability.

DOE further notes that there was an error in the equations proposed in the February 2015 NOPR that divided the quantity of heat, in Btu/min, by 60 instead of multiplying by 60 to convert to Btu/h. 80 FR 10211, 10224 (Feb. 25, 2015). This SNOPR corrects the calculation error in DOE’s proposal.

Based on these changes, DOE proposes in this SNOPR to calculate the sensible and latent heat components of infiltration air, using the nominal test chamber temperatures and subsequent humidity ratio, as follows:

$$Q_s = \dot{m} \times 60 \times \left[\left(c_{p\_da} \times (T_{ia} - T_{indoor})\right) + c_{p\_wv} \times (\omega_{ia} \times T_{ia} - \omega_{indoor} \times T_{indoor})\right]$$

Where:

$Q_s$ is the sensible heat added to the room by infiltration air, in Btu/h;

$\dot{m}$ is the dry air mass flow rate of infiltration air for a single-duct or dual-dual duct unit, in lb/m;

$c_{p\_da}$ is the specific heat of dry air, 0.24 Btu/$lb_m$-°F.

$c_{p\_wv}$ is the specific heat of water vapor, 0.444 Btu/$lb_m$-°F.

$T_{indoor}$ is the indoor chamber dry-bulb temperature, 80 °F.

$T_{ia}$ is the infiltration air dry-bulb temperature, 95 °F.

$\omega_{ia}$ is the humidity ratio of the infiltration air, 0.0141 $lb_w/lb_{da}$.

$\omega_{indoor}$ is the humidity ratio of the indoor chamber air, 0.0112 $lb_w/lb_{da}$.

60 is the conversion factor from minutes to hours.

$$Q_l = \dot{m} \times 60 \times H_{fg} \times (\omega_{ia} - \omega_{indoor})$$

Where:

$Q_l$ is the latent heat added to the room by infiltration air, in Btu/h.

$\dot{m}$ is the mass flow rate of infiltration air for a single-duct or dual-dual duct unit, in lb/m.

$H_{fg}$ is the latent heat of vaporization for water vapor, 1061 Btu/$lb_m$.

$\omega_{ia}$ is the humidity ratio of the infiltration air, 0.0141 $lb_w/lb_{da}$.

$\omega_{indoor}$ is the humidity ratio of the indoor chamber air, 0.0112 $lb_w/lb_{da}$.

60 is the conversion factor from minutes to hours.

2. Test Duration

AHAM PAC-1-2015 specifies testing in accordance with certain sections of ANSI/ASHRAE Standard 37-2009, “Methods of Testing for Rating Electrically Driven Unitary

Air-Conditioning and Heat Pump Equipment" (ASHRAE 37-2009), but does not explicitly specify the test duration required when conducting portable AC active mode testing. Therefore, DOE proposes in this SNOPR that the active mode test duration shall be determined in accordance with section 8.7 of ASHRAE 37-2009.

3. Seasonally Adjusted Cooling Capacity

In the February 2015 NOPR, DOE proposed a calculation for adjusted cooling capacity, ACC, defined as the measured cooling capacity adjusted for case, duct, and infiltration air heat transfer impacts. 80 FR 10211, 10225 (Feb. 25, 2015).

With the proposal to add a second cooling mode test condition for dual-duct portable ACs and, similarly, a second numerically applied infiltration air condition for single-duct portable ACs, DOE proposes that the adjusted cooling capacities for both sets of conditions be combined to create a seasonally adjusted cooling capacity, SACC. The higher outdoor temperature condition is consistent with that used for testing other air conditioning equipment and ensures that products can operate when they are most needed, while the cooler condition represents the typical outdoor temperatures encountered during use. Because the performance of a portable AC is important under each of these scenarios, DOE proposes in this SNOPR to weight the adjusted cooling capacities obtained under the two cooling mode conditions to calculate the SACC as follows.

Using an analytical approach based on climate and RECS data that was similar to the method used to determine representative rating conditions, DOE estimated the percentage of portable AC operating hours that would be associated with each rating condition. From the climate data, DOE allocated the number of annual hours with temperatures that ranged from 80 °F (the indoor test condition) to 89 °F (a temperature mid-way between the two rating conditions) to the 83 °F rating condition. The hours in which the ambient temperature was greater than 89 °F were assigned to the 95 °F rating condition. DOE then performed the geographical weighted averaging using the RECS data as discussed in section III.1.b to determine weighting factors of 19.7 percent and 80.3 percent, respectively, for the 95 °F and 83 °F rating conditions. A similar approach was adopted for central ACs, where DOE specifies eight test conditions and corresponding weighting factors that are based on the distribution of fractional hours for representative temperature bins.[8] For portable ACs, DOE estimated hours per temperature bin from the climate data analysis, and proposes in this SNOPR to apply weighting factors of 20 percent and 80 percent to the results of its testing at 95 °F and 83 °F, respectively. DOE welcomes input on whether different weighting factors would be appropriate.

Therefore, DOE proposes to calculate SACC according to the following equation.

$$SACC = (ACC_{95} \times 0.2) + (ACC_{83} \times 0.8)$$

Where:

SACC is the seasonally adjusted cooling capacity, in Btu/h.

[8] The DOE test procedure for central ACs is codified at 10 CFR part 430, subpart B, appendix M.

$ACC_{95}$ and $ACC_{83}$ are the adjusted cooling capacities calculated at the 95 °F and 83 °F dry-bulb outdoor conditions, in Btu/h, respectively.

0.2 is the weighting factor for $ACC_{95}$.

0.8 is the weighting factor for $ACC_{83}$.

4. Duct Heat Transfer and Leakage

In the February 2015 NOPR, DOE presented its determination that duct heat losses and air leakage are non-negligible effects, and therefore proposed to account for heat transferred from the duct surface to the conditioned space in the portable AC test procedure. DOE proposed that four equally spaced thermocouples be adhered to the side of the entire length of the condenser exhaust duct for single-duct units and the condenser inlet and exhaust ducts for dual-duct units. DOE proposed to determine the duct heat transfer for each duct from the average duct surface temperature as measured by the four thermocouples, a convection heat transfer coefficient of 4 Btu/h per square foot per °F (Btu/h-ft$^{2}$-°F), and the calculated duct surface area based on the test setup. 80 FR 10211, 10227 (Feb. 25, 2015).

a. Duct Heat Transfer Impacts

ASAP supported incorporating the duct heat transfer effects into the measurement of cooling capacity, and noted that there was a reasonably good correlation between the results using the calorimeter method and the modified AHAM method, as presented in the February 2015 NOPR. (Public Meeting Transcript, ASAP, No. 13 at p. 56)

AHAM and De’ Longhi stated that DOE’s proposed test for duct heat transfer and leakage unnecessarily complicates the test procedure without a corresponding benefit. They also stated that the methodology for the temperature sensor placement and determination of overall heat losses may be interpreted differently. AHAM further commented that should DOE decide to include provisions for duct heat transfer and leakage, DOE should evaluate the impact of these effects on test procedure repeatability and reproducibility, preferably through a round robin test including manufacturers and third-party laboratories. (AHAM, No. 18 at p. 5; De’ Longhi, No. 16 at p. 4)

China commented that DOE did not present the percent of the total cooling capacity associated with the duct and case heat transfer, and that it would be necessary to consider such data before adopting an approach that accounts for these heat transfer effects. (China, No. 15 at p. 3)

In response to these comments, DOE conducted further analysis to quantify the impacts of duct heat transfer. Figure III.1 shows the impact of duct heat transfer as a percentage of the AHAM PAC-1-2009 cooling capacity measured in the February 2015 NOPR for each unit in DOE’s test sample. Exhaust duct heat transfer is presented for each single-duct unit, while a pair of values for inlet duct heat transfer and exhaust duct heat transfer are presented for each dual-duct unit.

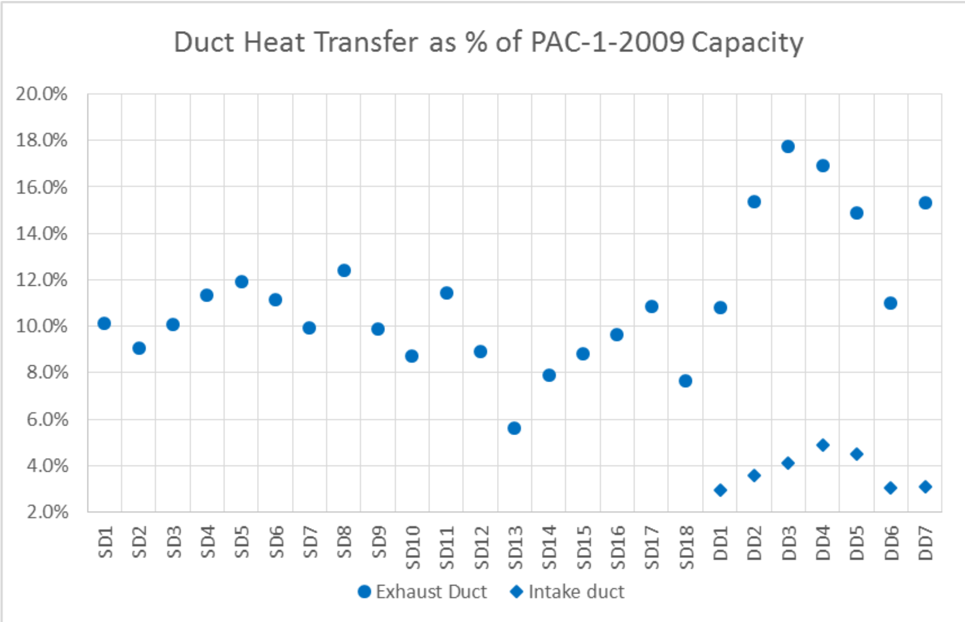


**Figure III.1 Duct Heat Transfer as a Percentage of AHAM PAC-1-2009 Cooling Capacity for each Test Unit in the February 2015 NOPR**

As shown in Figure III.1, the exhaust duct heat transfer determined according to the proposed methodology ranged from just below 6 percent to almost 18 percent of the AHAM PAC-1-2009 cooling capacity, with an average value of 11.1 percent. The intake duct heat transfer effect was lower than that of the exhaust duct due to the lower air temperature at the inlet, with values ranging from about 3 percent to almost 5 percent of the unadjusted cooling capacity and an average of 3.7 percent. DOE finds the exhaust and intake duct heat transfer impacts sufficiently significant to warrant the added test burdens associated with determining duct heat transfer. Therefore, DOE maintains the proposal from the February 2015 NOPR to measure and incorporate the duct heat transfer impacts into the overall seasonally adjusted cooling capacity.

b. Convection Coefficient

DENSO considered the 4 Btu/h-ft$^{2}$-°F convection coefficient proposed for the duct heat transfer calculation to be arbitrary, and recommended measuring the conditions of the air at the inlet and outlet of each duct to substantiate that factor. (Public Meeting Transcript, DENSO, No. 13 at p. 53; DENSO, No. 14 at p. 2) DOE recognizes that different test setups may have somewhat different convective heat transfer coefficients. However, when developing test procedures, DOE must consider the test burden and impact on manufacturers and test laboratories. Taking that into consideration, DOE proposed an approach in the February 2015 NOPR that would minimize burden while capturing the impact of heat transfer from ducts, which DOE determined to have a significant impact on overall net cooling capacity. DOE also notes that the approach proposed by DENSO to characterize heat loss to the conditioned space would significantly increase test burden, requiring additional thermocouples and modification of the test setup on the unit-side of the duct. Further, DOE notes that the convection heat transfer coefficient may vary among different laboratories and even for different chambers and test setups within each test laboratory. This would introduce variability from test to test, as the heat transfer coefficient may be highly sensitive to the specific test setup. To minimize the test burden and limit variability, DOE proposed one convection heat transfer coefficient for all units to provide a consistent estimate of the duct heat transfer.

In the February 2015 NOPR, DOE estimated the convection heat transfer coefficient to be 4 Btu/h-ft$^{2}$-°F based on a midpoint of values associated with free convection and forced convection, as recommended by the test laboratory that conducted testing for the NOPR. 80 FR 10211, 10219 (Feb. 25, 2015). The convection coefficient was based on values derived from coefficients listed in the 2013 ASHRAE Handbook—Fundamentals[9] for various types of assemblies in buildings. Depending on the orientation of the surface, direction of heat flow, and emissivity of the heat transfer surface, the typical coefficients for indoor assemblies, which DOE deduced would be subject primarily to free convection, ranged from 0.22 to 1.63 Btu/h-ft$^{2}$-°F. ASHRAE also provided coefficients for assemblies located outside and subject to wind speeds of 7.5 and 15 miles per hour (5.1 and 10.2 feet per second, respectively), which were 4.00 and 6.00 Btu/h-ft$^{2}$-°F, respectively. Because these speeds potentially correspond to air flow speeds over the portable AC duct(s) due to circulation of the conditioned air in the space, for example by the portable AC blower and infiltration air, DOE used these values as proxies for convective heat transfer coefficients for the duct surfaces. Therefore, DOE proposed in the February 2015 NOPR that the overall heat transfer coefficient for calculating duct heat losses would be 4 Btu/h-ft$^{2}$-°F, an approximate midpoint of the values described.

To further validate the proposed convection heat transfer coefficient for this notice, DOE re-examined the data it obtained from testing a sample of four single-duct and two dual-duct portables ACs with and without duct insulation for the May 2014 NODA. These tests were conducted using the calorimeter approach described in the May 2014 NODA, such that duct heat losses could be measured by subtracting the measured cooling capacity without insulation from

[9] ASHRAE Handbook—Fundamentals. American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Atlanta, GA. 2013.

the cooling capacity with insulation. Using the duct heat losses, duct surface area, and the differential between the indoor side ambient temperature and the average of the duct surface temperatures, an average duct heat transfer coefficient could be empirically determined for units in DOE's test sample. The results of this calculation are shown in Table III.4 below.

**Table III.4 Measured Duct Convection Heat Transfer Coefficients**

| Test Unit | Duct Convection Heat Transfer Coefficient (Btu/h-ft$^{2}$-°F) |
|---|---|
| SD1 | 2.74 |
| SD2 | 3.08 |
| SD3 | 1.70 |
| SD4 | 5.26 |
| DD1 (Test 1) | 4.10 |
| DD1 (Test 2) | 3.76 |
| DD2 (Test 1) | 2.11 |
| DD2 (Test 2) | 2.27 |
| Average | 3.13 |

SD = Single-duct
DD = Dual-duct

Although the average heat transfer coefficient calculated from DOE's test results was slightly lower than the value proposed in the February 2015 NOPR, DOE notes that there is variation in individual results that is likely due to different duct types, installation configurations, forced convection air flow patterns, and other factors. Therefore, DOE proposes to maintain the original duct heat transfer proposal from the February 2015 NOPR, including the convection heat transfer coefficient of 4 Btu/h-ft$^{2}$-°F.

c. Duct Surface Area Measurements

In the February 2015 NOPR, DOE proposed that the duct surface area be calculated using the outer duct diameter and extended length of the duct while under test. 80 FR 10211, 10227 (Feb. 25, 2015).

De' Longhi and AHAM commented that ducts often have a corrugated surface, so that the measure of the duct(s) surface area will have high uncertainty. (De' Longhi, No. 16 at p. 4; AHAM, No. 18 at p. 5) DOE further examined the surface area of the ducts in its test sample. DOE calculated the surface area in two ways, one with the ducts fully extended and the other with the duct setup as required in AHAM PAC-1-2015. DOE found that the average difference in surface area calculated using the fully extended duct versus using the test setup was 7.5 percent. With the average duct impact on cooling capacity of 11.1 percent and 3.7 percent for single-duct and dual-duct units, respectively, the overall variability that differences in duct surface area determinations would introduce into the cooling capacity would be no greater than 1 percent. Therefore, DOE concludes that any uncertainty in duct surface area measurements would not have a significant impact on test repeatability and reproducibility and maintains the surface area measurement as proposed in the February 2015 NOPR.

5. Case Heat Transfer

In the February 2015 NOPR, DOE proposed that case heat transfer be determined using a method similar to the approach proposed for duct heat transfer. DOE proposed that the surface area and average temperature of each side of the case be measured to determine the overall case heat transfer, which would be used to adjust the cooling capacity and efficiency. DOE noted that the case heat transfer methodology would impose additional test burden, but determined that the burdens were likely outweighed by the benefit of addressing the heat transfer effects of all internal heating components. 80 FR 10211, 10227–10229 (Feb. 25, 2015).

DENSO commented that DOE should incorporate the effects of evaporator fan heat rather than case heat transfer effects, because all of the fan motor power ends up in the evaporator exhaust air stream. DENSO also stated that heat transfer mechanics for all surfaces of the case are not the same. (DENSO, No. 14 at p. 2)

Friedrich believes that there is no need to measure heat loss from the electrical components inside the case because the end result of the test would be the total cooling capacity coming from the portable AC and the total measure of energy consumption. (Public Meeting Transcript, Friedrich, No. 13 at p. 34)

De' Longhi noted that because there is a wide range in unit design, each portable AC may have uniquely shaped faces on the case, and it would be very difficult or impossible to identify the front, back, right, left, top, and bottom of the case. De' Longhi stated that laboratories may measure the surface temperature differently, and therefore, the proposal in the February 2015 NOPR may lead to inconsistencies among different laboratories. De' Longhi further suggested that the convection coefficient should be different for each side of the case due to the different orientation of each surface, and commented that a small variation in the position of the temperature sensors can cause significant differences in the average temperatures of each case. (Public Meeting Transcript, De' Longhi, No. 13 at pp. 55–56; De' Longhi, No. 16 at p. 4)

AHAM stated that the proposed methodology for determining case heat transfer unnecessarily complicates the test procedure and will likely lead to variation. AHAM believes the impact of case heat transfer is negligible and does not justify the added burden and variation. According to AHAM, if DOE continues to consider case heat transfer, DOE should characterize the proposed test procedure's repeatability and reproducibility, preferably through a round robin test including manufacturers and third-party laboratories. (AHAM, No. 18 at pp. 5–6)

In response to these comments, DOE further investigated the effects of case heat transfer as a percentage of the overall cooling capacity by analyzing the data determined in accordance with AHAM PAC-1-2009 for the February 2015 NOPR. Figure III.2 shows, for each portable AC in its test sample, the heat transfer determined for each case side and the sum of all case sides as a percentage of the AHAM PAC-1-2009 cooling capacity.

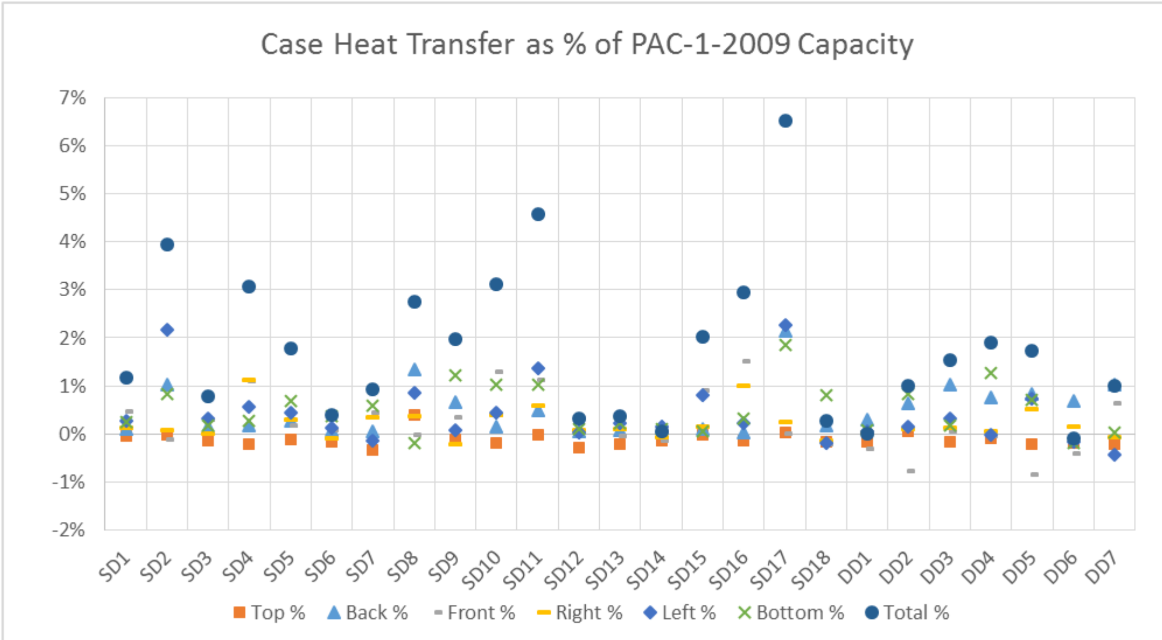


**Figure III.2 Case Heat Transfer as a Percentage of AHAM PAC-1-2009 Cooling Capacity for Each Test Unit in the February 2015 NOPR**

From the data in Figure III.2, DOE calculated that the average heat transfer for individual case sides was 0.29 percent of the AHAM PAC-1-2009 cooling capacity, and the maximum heat transfer observed for a single side was 2.27 percent. The total case heat transfer impact was, on average, 1.76 percent of the AHAM PAC-1-2009 cooling capacity, with a maximum of 6.53 percent. Because the total case heat transfer impact is, on average, less than 2 percent of the cooling capacity without adjustments for infiltration air and heat transfer effects, DOE proposes to remove the provisions for determining case heat transfer from the proposed portable AC test procedure.

6. Test Unit Placement

In the February 2015 NOPR, DOE proposed that for all portable AC configurations, there must be no less than 6 feet between the evaporator inlet and any chamber wall surface, and for single-duct units, there must be no less than 6 feet between the condenser inlet surface and any other wall surface. Additionally, DOE proposed that there be no less than 3 feet between the other surfaces of the portable AC with no air inlet or exhaust (other than the bottom of the unit) and any wall surfaces. 80 FR 10211, 10229–10230 (Feb. 25, 2015).

According to DENSO, the 6-foot minimum spacing would cause an unreasonable performance penalty when duct losses are incorporated into the efficiency rating. DENSO further noted that the ducted side of a portable AC is often located relatively close to the wall where the duct is mounted. (DENSO, No. 14 at p. 3)

AHAM objected to the proposed test unit placement, commenting that, due to duct length, it may not be feasible to maintain the proposed distances from the partition wall. AHAM stated that this particular distance is variable and unit-dependent, and should not be applicable for single-duct or dual-duct units. (AHAM, No. 18 at pp. 6–7)

De' Longhi requested clarification as to whether the back of the unit, or side with the duct attachments, is considered a side that must be placed at the minimum distance from the chamber or partition walls. If so, De' Longhi commented that the unit should be placed at least 6 feet from the partition wall and the ducts would likely not reach. (Public Meeting Transcript, De' Longhi, No. 13 at pp. 59–60; De' Longhi, No. 16 at p. 4)

DOE recognizes that the length of the duct and duct setup as outlined in AHAM PAC-1-2015 dictate the distance of the portable AC from the partition wall. Therefore, DOE proposes to adjust the February 2015 NOPR proposals for unit placement that would have required no less than 6 feet between the evaporator inlet and any chamber wall surfaces, and for single-duct units, no less than 6 feet between the condenser inlet surface and any other wall surface. Because AHAM PAC-1-2015 specifies the distance between the test unit and the partition wall, DOE proposes that the test unit be placed in such a way that there is no less than 3 feet between any test chamber wall and any surface on the portable AC, except the surface or surfaces that have a duct attachment, as prescribed by the AHAM PAC-1-2015 test setup requirements. DOE notes that this test unit placement would provide manufacturers and test laboratories more flexibility in the use of their test chambers than that proposed in the February 2015 NOPR, and would still provide sufficient space around the test unit to ensure free air flow with no air constriction.

C. Heating Mode

As discussed in the February 2015 NOPR, certain portable ACs, including some of the units in DOE’s test sample, incorporate a heating function in addition to cooling mode. DOE proposed to define heating mode as an active mode in which a portable AC has activated the main heating function according to the thermostat or temperature sensor signal, including activating a resistance heater, the refrigeration system with a reverse refrigerant flow valve, or the fan or blower without activation of the resistance heater or refrigeration system. 80 FR 10211, 10217 (Feb. 25, 2015). In the February 2015 NOPR, DOE concluded that a heating mode test to measure heating mode performance was feasible, and proposed a heating mode test procedure that utilized AHAM PAC-1-2014 at lower temperature ambient conditions and with comparable adjustments as were considered for cooling mode. 80 FR 10211, 10230–10231 (Feb. 25, 2015).

AHAM and De' Longhi opposed DOE's proposal to require testing in heating mode. They noted that heating mode is not the main consumer utility offered by portable ACs, and commented that it was not clear how often consumers use the heating feature and whether the burden of including this mode in the test procedure would be justified. AHAM, NAM, and De' Longhi commented that there are not sufficient heating mode data upon which to determine whether to include measurement of or assign annual operating hours to heating mode. AHAM and NAM further noted that in the heating analysis, DOE assumed that the consumer will use a portable AC in heating mode when the temperature has fallen below 45 °F, but presented no consumer data to support that assumption. According to AHAM, consumer usage of portable ACs in heating mode is extremely limited due to the seasonality of the product. AHAM, NAM, and De' Longhi commented that DOE should be consistent with its other analyses when considering heating mode. For example, they stated that DOE did not propose testing in fan-only mode because it would be impractical, nor did it propose testing in dehumidification mode because it is not the primary mode of operation for portable ACs. These commenters considered heating mode to be no different, and therefore concluded that DOE should not require it to be tested. (Public Meeting Transcript, AHAM, No. 13 at p. 64; AHAM, No. 18 at pp. 7, 10; De' Longhi, No. 16 at p. 5; NAM, No. 17 at p. 2)

AHAM noted that many of the comments submitted regarding cooling mode would also apply to heating mode where applicable. Specifically, should DOE require measurement of heating mode, AHAM would not object to DOE's proposal to use the unit and duct setup requirements and control settings of AHAM PAC-1-2014, as well as the test configurations referenced in Table 2 of AHAM PAC-1-2014. AHAM opposed the inclusion of infiltration air, duct heat transfer, case transfer, and test unit placement for heating mode as discussed for cooling mode. (AHAM, No. 18 at pp. 7–8)

DENSO stated that its cooling mode comments are generally applicable for heating mode as well. (DENSO, No. 14 at p. 3)

After considering stakeholder comments opposing the test procedure for heating mode and in light of the test burden that the heating mode test would impose, DOE proposes to remove the heating mode test provisions from the proposed DOE portable AC test procedure, including the definition of heating mode and calculations for $EER_{hm}$ and total combined energy efficiency ratio. Accordingly, the cooling-specific energy efficiency ratio, $EER_{cm}$, is no longer necessary, as the combined efficiency ratio, CEER, would appropriately represent energy efficiency in all modes under consideration. DOE expects that measuring performance in cooling mode, off-cycle mode, standby mode, and off mode would capture representative performance of portable ACs during the cooling season. DOE may reconsider including a test for heating mode in a future test procedure rulemaking.

D. Combined Energy Efficiency Ratio

In the February 2015 NOPR, DOE proposed a single energy conservation standard metric for portable ACs, in accordance with the requirements of EPCA. (42 U.S.C. 6295(gg)(3)(A)) The single integrated efficiency metric, CEER, weights the average power in each operating mode, as measured by the proposed test procedure, with estimated annual operating hours for each mode.

The modes considered in the February 2015 NOPR procedure were cooling mode, heating mode, off-cycle mode (with and without fan operation), inactive mode (including bucket-full mode), and off mode. 80 FR 10211, 10234–10235 (Feb. 25, 2015).

1. Annual Operating Mode Hours

As presented in the February 2015 NOPR, DOE developed several estimates of portable AC annual operating mode hours for cooling, heating, off-cycle, and inactive or off modes. However, the three estimates that addressed units with both cooling and heating mode operating hours are no longer applicable with the removal of the heating mode test procedure. Therefore, for this revised analysis, DOE considered the annual operating mode hours for all portable ACs to be those from the "Cooling Only" scenario presented in the February 2015 NOPR as follows:

**Table III.5 Proposed Annual Operating Hours by Mode**

| **Modes** | **Operating Hours** |
|---|---|
| Cooling Mode | 750 |
| Off-Cycle Mode | 880 |
| Off/Inactive Mode | 1,355 |

More information on the development of these annual hours for each operating mode can be found in the February 2015 NOPR. 80 FR 10211, 10235–10237 (Feb. 25, 2015).

Friedrich noted that it rates its portable AC energy consumption based on 750 hours, the same cooling mode operating hours as room ACs. Friedrich suggested that DOE maintain the proposal of 750 annual cooling mode operating hours for portable ACs to maintain harmonization with room ACs and properly reflect unit annual energy consumption. (Public Meeting Transcript, Friedrich, No. 13 at p. 84)

AHAM and NAM disagreed with DOE's proposals, stating that the majority of the analysis was based on outdated room AC data. They asserted that although portable ACs and room ACs are similar in some ways, the usage profiles and installation locations of the two products differ. AHAM and NAM urged DOE to obtain data on consumer usage of portable ACs or demonstrate that consumer use of portable ACs and room ACs are sufficiently comparable. (Public Meeting Transcript, AHAM, No. 13 at pp. 81–83; AHAM, No. 18 at p. 10; NAM, No. 17 at pp. 1–2).

AHAM and NAM also objected to DOE basing the proposed unplugged hours on assumptions, without any consumer study or supporting data. These commenters stated that DOE should obtain consumer use data in order to inform its proposal on the number of unplugged hours. (Public Meeting Transcript, AHAM, No. 13 at p. 81; AHAM, No. 18 at p. 10; NAM, No. 17 at p. 2)

AHAM further commented that it is not aware of consumer usage data for portable ACs, but would attempt to request that information from its members. AHAM urged DOE not to proceed in the absence of such consumer use data. (Public Meeting Transcript, AHAM, No. 13 at pp. 83–84)

Neither AHAM nor manufacturers provided additional consumer usage data, and no further data were available from RECS or other sources. Therefore, DOE continues to utilize the most relevant consumer use data available and proposes the annual operating hours in Table III.5, maintaining the analysis and approach described in the February 2015 NOPR. DOE welcomes any additional information and data regarding consumer use to further inform the proposed annual mode operating hours.

2. CEER Calculation

In addition to the CEER metric that incorporated energy consumption in all operating modes, including heating mode, DOE proposed a simplified CEER metric in the February 2015 NOPR for portable ACs that do not include a heating mode ($CEER_{cm}$). The CEER calculation in the February 2015 NOPR would equal $CEER_{cm}$ for units without heating mode. With the newly proposed removal of heating mode from the test procedure and addition of a second set of testing conditions for dual-duct units, DOE also proposes in this SNOPR to eliminate the $CEER_{cm}$ calculation and to revise the CEER metric calculation as follows, using the same weighting factors as were developed for SACC. The revised calculations also correctly divide energy consumption by annual cooling mode hours rather than total annual hours, as was initially proposed in the February 2015 NOPR.

$$CEER_{SD} = \left[\frac{(ACC_{95} \times 0.2 + ACC_{83} \times 0.8)}{\left(\frac{AEC_{SD} + AEC_T}{k \times t}\right)}\right]$$

$$CEER_{DD} = \left[\frac{ACC_{95}}{\left(\frac{AEC_{95} + AEC_T}{k \times t}\right)}\right] \times 0.2 + \left[\frac{ACC_{83}}{\left(\frac{AEC_{83} + AEC_T}{k \times t}\right)}\right] \times 0.8$$

Where:

$CEER_{SD}$ and $CEER_{DD}$ are the combined energy efficiency ratios for single-duct and dual duct units, respectively, in Btu/Wh.

$ACC_{95}$ and $ACC_{83}$ are the adjusted cooling capacities, tested at the 95°F and 83 °F dry-bulb outdoor conditions, respectively, in Btu/h.

$AEC_{SD}$ is the annual energy consumption in cooling mode for single-duct units, in kWh/year.

$AEC_{95}$ is the annual energy consumption in cooling mode for dual-duct units, assuming all cooling mode hours would be at the 95 °F dry-bulb outdoor conditions, in kWh/year.

$AEC_{83}$ is the annual energy consumption in cooling mode for dual-duct units, assuming all cooling mode hours would be at the 83 °F dry-bulb outdoor conditions, in kWh/year.

$AEC_T$ is the total annual energy consumption attributed to all modes except cooling, in kWh/year.

t is the number of cooling mode hours per year, 750.

k is 0.001 kWh/Wh conversion factor for watt-hours to kilowatt-hours.

0.2 is the weighting factor for the 95 °F dry-bulb outdoor condition test.

0.8 is the weighting factor for the 83 °F dry-bulb outdoor condition test.

The February 2015 NOPR included incorrect text stating that the representative CEER would be the mean of the test unit efficiencies. DOE proposes in this SNOPR to clarify that the representative CEER for a basic model is calculated based on the sampling plan instructions proposed in 10 CFR 429.62. DOE further maintains its proposal that the CEER would be rounded to the nearest 0.1 Btu/Wh.

E. 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)) In the February 2015 NOPR, DOE concluded that establishing a test procedure to measure the energy consumption of portable ACs in active mode, standby mode, and off mode would produce the required test results and would not be unduly burdensome to conduct. This determination was driven by the many similarities between the necessary testing equipment and facilities for portable ACs and other products, whose performance is currently certified through a DOE test procedure. Therefore, DOE concluded that manufacturers would not be required to make significant investment in test facilities and new equipment.

DOE notes that the modifications to the portable AC test procedures introduced in this notice, mainly the additional test condition in cooling mode for dual-duct units and the removal of heating mode testing and case heat transfer considerations, would not significantly increase the overall test burden compared to the test procedure proposed in the February 2015 NOPR. Further, because the added cooling mode test conditions are closer to those of the originally proposed cooling mode test than the test conditions for the heating mode test, DOE estimates that less time would be required to achieve and maintain the chamber conditions for the second cooling mode test than for a heating mode test, decreasing the test burden for dual-duct units with a heating mode. In addition, the outdoor test chamber would not be required to reach the low temperatures required for the proposed heating mode test, which may have presented difficulties for some manufacturers and test laboratories to achieve.

For dual-duct units without a heating mode, the proposals in this notice would introduce test burden by requiring a second cooling mode test. However, the removal of case surface temperature measurements would likely mitigate the increased burden associated with this second cooling mode test, resulting in similar overall test burden as for the test procedure proposed in the February 2015 NOPR.

DOE concludes that although this SNOPR introduces modifications to the test procedures proposed in the February 2015 NOPR, it does not significantly increase the test burden, and may instead reduce the overall test burden. Therefore, the determination in the February 2015 NOPR that the proposed portable AC test procedure would produce test results that measure energy consumption during representative use and that the test procedure would not be unduly burdensome to conduct still applies.

## IV. Procedural Issues and Regulatory Review

DOE has concluded that the determinations made pursuant to the various procedural requirements applicable to the February 2015 NOPR, set forth at 80 FR 10212, 10238–10241, remain unchanged for this SNOPR, except for the following additional analysis and determination DOE conducted in accordance with the Regulatory Flexibility Act (5 U.S.C. 601 et seq.).

A. 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 website: http://energy.gov/gc/office-general-counsel.

DOE reviewed this proposed rule under the provisions of the Regulatory Flexibility Act and the procedures and policies published on February 19, 2003. DOE's IRFA is set forth in the February 2015 NOPR, with additional analysis below based on the proposals in this SNOPR. DOE seeks comment on its analysis and the economic impacts of the rule on small manufacturers. In the February 2015 NOPR, DOE estimated that there is one small business that manufactures portable ACs. Since the February 2015 NOPR, DOE has determined that this small business no longer produces portable ACs and, therefore, DOE is unable to identify any small businesses that currently manufacture portable ACs. For this reason, DOE tentatively concludes and certifies that the proposed rule would not have a significant economic impact on a substantial number of small entities. Accordingly, DOE has not prepared a regulatory flexibility analysis for this rulemaking. DOE will transmit the certification and supporting statement of factual basis to the Chief Counsel for Advocacy of the Small Business Administration (SBA) for review under 5 U.S.C. 605(b).

In the alternative, should any small business manufacturers of portable ACs be identified, DOE evaluated the modifications proposed in this SNOPR to determine if these modification would have a significant economic impact on small businesses as compared to the proposals in the February 2015 NOPR. DOE believes that these modifications are likely to reduce overall test burden with respect to the proposals in the February 2015 NOPR, and therefore would not have a significant economic impact on small businesses, should any be identified.

In this SNOPR, DOE proposes to increase the number of cooling mode tests for dual-duct portable ACs from one test to two tests at different outdoor test conditions. Although this increase requires running the cooling mode test a second time, DOE notes that the test setup would not need to be modified between testing and as such would not significantly increase the test burden beyond that for a single cooling mode test. The remaining changes associated with the additional outdoor test condition impact the post-testing calculations and therefore do not increase test burden.

DOE further proposes in this SNOPR to remove the measurement of case heat transfer and the heating mode testing requirements that were originally proposed in the February 2015 NOPR. The removal of the case heat transfer measurement eliminates the added burden of determining surface area of each case surface and measuring the average temperature of each surface. In addition, the removal of the heating mode test significantly reduces test burden for dual-duct units with a heating mode, in that a substantial stabilization period is avoided that would require reducing the outdoor chamber conditions well below those for the cooling mode test.

In the February 2015 NOPR, DOE concluded that the costs associated with the February 2015 NOPR proposals were small compared to the overall financial investment needed to undertake the business enterprise of developing and testing consumer products. 80 FR 10211, 10239. Compared to the proposals in the February 2015 NOPR, there is no net change in the number of tests or power metering instrumentation required. In addition, the elimination of the case heat transfer requirement would avoid the potential need for setting up and purchasing additional temperature sensors, estimated to cost less than $500 for both equipment and labor.

On the basis of this analysis, DOE tentatively concludes that the proposed rule would not have a significant economic impact on a substantial number of small entities, should any small business manufacturers of portable ACs be identified.

DOE seeks comment on the determinations in this section and information on whether any small businesses manufacture portable ACs.

B. Description of Materials Incorporated by Reference

In this SNOPR, DOE proposes to incorporate by reference the test standard published by AHAM, titled "Portable Air Conditioners," AHAM PAC-1-2015. AHAM PAC-1-2015 is an industry accepted test procedure that measures portable AC performance in cooling mode and is applicable to products sold in North America. AHAM PAC-1-2015 specifies testing conducted in accordance with other industry accepted test procedures (already incorporated by reference) and determines energy efficiency metrics for various portable AC configurations. The test procedure proposed in this SNOPR references various sections of AHAM PAC-1-2015 that address test setup, instrumentation, test conduct, calculations, and rounding. AHAM PAC-1-2015 is readily available on AHAM's website at http://www.aham.org/ht/d/ProductDetails/sku/PAC12009/from/714/pid/.

In this SNOPR, DOE also proposes to incorporate by reference the test standard ASHRAE Standard 37-2009, titled "Methods of Testing for Rating Electrically Driven Unitary Air-Conditioning and Heat Pump Equipment," (ANSI Approved). ANSI/ASHRAE Standard 37-2009 is an industry-accepted test standard referenced by AHAM PAC-1-2015 that defines various uniform methods for measuring performance of air conditioning and heat pump equipment. Although AHAM PAC-1-2015 references a number of sections in ANSI/ASHRAE Standards 37-2009, the test procedure proposed in this SNOPR additionally references one section in ANSI/ASHRAE Standards 37-2009 that addresses test duration. ANSI/ASHRAE Standards 37-2009 is readily available on ANSI's website at http://webstore.ansi.org/RecordDetail.aspx?sku=ANSI%2FASHRAE+Standard+37-2009.

## V. Public Participation

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 www.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 www.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 website 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).

## VI. Approval of the Office of the Secretary

The Secretary of Energy has approved publication of this supplemental notice of proposed rulemaking.

**List of Subjects**

10 CFR Part 429

Confidential business information, Energy conservation, Household appliances, Imports, Incorporation by reference, Reporting and recordkeeping requirements.

10 CFR Part 430

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

Issued in Washington, DC, on November 17, 2015.

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Kathleen B. Hogan
Deputy Assistant Secretary for Energy Efficiency
Energy Efficiency and Renewable Energy

For the reasons stated in the preamble, DOE proposes to amend parts 429 and 430 of Chapter II of Title 10, Code of Federal Regulations as set forth below:

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

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

**Authority:** 42 U.S.C. 6291–6317.

2. Section 429.4 is amended by adding paragraph (b)(3) to read as follows:

**§ 429.4 Materials incorporated by reference**.

*     *     *     *     *

(b)     *     *     *

(3)     AHAM PAC-1-2015, Portable Air Conditioners, 2015, IBR approved for § 429.62.

*     *     *     *     *

3. Add § 429.62 to read as follows:

**§ 429.62 Portable air conditioners.**

(a) <u>Sampling plan for selection of units for testing</u>. (1) The requirements of § 429.11 are applicable to portable air conditioners; and

(2) For each basic model of portable air conditioner, a sample of sufficient size shall be randomly selected and tested to ensure that—

(i) Any represented value of energy consumption or other measure of energy consumption of a basic model for which consumers would favor lower values shall be greater than or equal to the higher of:

(A) The mean of the sample:

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

Where:

$\bar{x}$ is the sample mean;

$x_i$ is the i$^{th}$ sample; and

n is the number of units in the test sample.

Or,

(B) The upper 95 percent confidence limit (UCL) of the true mean divided by 1.10:

$$UCL = \bar{x} + t_{0.95}\left(\frac{s}{\sqrt{n}}\right)$$

Where:

$\bar{x}$ is the sample mean;

s is the sample standard deviation;

n is the number of units in the test sample; and

$t_{0.95}$ is the t statistic for a 95% one-tailed confidence interval with n-1 degrees of freedom.

And,

(ii) Any represented value of the combined energy efficiency ratio or other measure of energy consumption of a basic model for which consumers would favor higher values shall be less than or equal to the lower of:

(A) The mean of the sample:

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

Where:

$\bar{x}$ is the sample mean;

$x_i$ is the $i^{th}$ sample; and

n is the number of units in the test sample.

Or,

(B) The lower 95 percent confidence limit (LCL) of the true mean divided by 0.90:

$$LCL = \bar{x} - t_{0.95}\left(\frac{s}{\sqrt{n}}\right)$$

Where:

$\bar{x}$ is the sample mean;

s is the sample standard deviation;

n is the number of units in the test sample; and

$t_{0.95}$ is the t statistic for a 95% one-tailed confidence interval with n-1 degrees of freedom.

And,

(3) The value of seasonally adjusted cooling capacity of a basic model shall be the mean of the seasonally adjusted cooling capacities for each tested unit of the basic model. Round the mean capacity value to the nearest 50, 100, 200, or 500 Btu/h, depending on the value being rounded, in accordance with Table 1 of AHAM PAC-1-2015, (incorporated by reference, see § 429.4), "Multiples for reporting Dual Duct Cooling

Capacity, Single Duct Cooling Capacity, Spot Cooling Capacity, Water Cooled Condenser Capacity and Power Input Ratings."

(4) Round the value of combined energy efficiency ratio of a basic model to the nearest 0.1 Btu/Wh.

(b) Certification reports. [Reserved]

**PART 430 -- ENERGY CONSERVATION PROGRAM FOR CONSUMER PRODUCTS**

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

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

5. Section 430.2 is amended by adding the definition of "portable air conditioner" in alphabetical order to read as follows:

**§ 430.2 Definitions**.

* * * * *

Portable air conditioner means an encased assembly, other than a "packaged terminal air conditioner," "room air conditioner," or "dehumidifier," designed as a portable unit for delivering cooled, conditioned air to an enclosed space, that is powered by single-phase electric current, and which may rest on the floor or other elevated surface. It includes a source of refrigeration and may include additional means for air circulation and heating.

* * * * *

6. Section 430.3 is amended by:

a. Revising paragraph (g)(4);

b. Redesignating paragraph (i)(8) as (i)(9), and adding a new paragraph (i)(8); and

c. Revising paragraph (p)(4).

The revisions read as follows:

**§430.3 Materials incorporated by reference.**

*   *   *   *   *

(g) *   *   *

(4) ANSI/ASHRAE Standard 37-2009, ("ASHRAE 37-2009"), Methods of Testing for Rating Electrically Driven Unitary Air-Conditioning and Heat Pump Equipment, ANSI approved June 25, 2009, IBR approved for appendix AA and CC to subpart B.

*   *   *   *   *

(i) *   *   *

(8) AHAM PAC-1-2015, Portable Air Conditioners, 2015, IBR approved for appendix CC to subpart B.

*   *   *   *   *

(p) *   *   *

(4) IEC 62301 ("IEC 62301"), Household electrical appliances—Measurement of standby power, (Edition 2.0, 2011-01), IBR approved for appendices C1, D1, D2, G, H, I, J2, N, O, P, X, X1, Z and CC to subpart B.

*   *   *   *   *

7. Section 430.23 is amended by adding paragraph (dd) to read as follows:

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

* * * * *

(dd) Portable air conditioners. (1) For portable air conditioners, measure the seasonally adjusted cooling capacity, expressed in British thermal units per hour (Btu/h), and the combined energy efficiency ratio, expressed in British thermal units per watt-hour (Btu/Wh) in accordance with section 5 of appendix CC of this subpart.

(2) Determine the estimated annual operating cost for portable air conditioners, expressed in dollars per year, by multiplying the following two factors:

(i) For dual-duct portable air conditioners, the sum of $AEC_{95}$ multiplied by 0.2, $AEC_{83}$ multiplied by 0.8, and $AEC_T$ as measured in accordance with section 5.3 of appendix CC of this subpart; or for single-duct portable air conditioners, the sum of $AEC_{SD}$ and $AEC_T$ as measured in accordance with section 5.3 of appendix CC of this subpart; and

(ii) A representative average unit cost of electrical energy in dollars per kilowatt-hour as provided by the Secretary.

(iii) Round the resulting product to the nearest dollar per year.

7. Add appendix CC to subpart B of part 430 to read as follows:

## Appendix CC to Subpart B of Part 430–Uniform Test Method for Measuring the Energy Consumption of Portable Air Conditioners

1. Scope

This appendix covers the test requirements used to measure the energy performance of single-duct and dual-duct portable air conditioners. It does not contain testing provisions for measuring the energy performance of spot coolers at this time.

2. Definitions

2.1 AHAM PAC-1 means the test standard published by the Association of Home Appliance Manufacturers, titled "Portable Air Conditioners," AHAM PAC-1-2015 (incorporated by reference; see § 430.3).

2.2 Combined energy efficiency ratio is the energy efficiency of a portable air conditioner as measured in accordance with this test procedure in Btu per watt-hours (Btu/Wh) and determined in section 5.4.

2.3 Cooling mode means a mode in which a portable air conditioner has activated the main cooling function according to the thermostat or temperature sensor signal, including activating the refrigeration system or the fan or blower without activation of the refrigeration system.

2.4 Dual-duct portable air conditioner means a portable air conditioner that draws some or all of the condenser inlet air from outside the conditioned space through a duct, and may draw additional condenser inlet air from the conditioned space. The condenser outlet air is discharged outside the conditioned space by means of a separate duct. 2.6 IEC 62301 means the test standard published by the International Electrotechnical Commission, titled "Household electrical appliances–Measurement of standby power,” Publication 62301 (Edition 2.0 2011-01) (incorporated by reference; see § 430.3).

2.5 Inactive mode means a standby mode that facilitates the activation of an active mode or off-cycle mode by remote switch (including remote control), internal sensor, or timer, or that provides continuous status display.

2.6 Off-cycle mode means a mode in which a portable air conditioner:

(1) Has cycled off its main cooling or heating function by thermostat or temperature sensor signal;

(2) May or may not operate its fan or blower; and

(3) Will reactivate the main function according to the thermostat or temperature sensor signal.

2.7 Off mode means a mode in which a portable air conditioner is connected to a mains power source and is not providing any active mode, off-cycle mode, or standby mode function, and where the mode may persist for an indefinite time. An indicator that only shows the user that the portable air conditioner is in the off position is included within the classification of an off mode.

2.8 Seasonally adjusted cooling capacity means a measure of the cooling, measured in Btu/h, provided to the indoor conditioned space, measured under the specified ambient conditions.

2.9 Single-duct portable air conditioner means a portable air conditioner that draws all of the condenser inlet air from the conditioned space without the means of a duct, and discharges the condenser outlet air outside the conditioned space through a single duct.

2.10 Spot cooler means a portable air conditioner that draws condenser inlet air from and discharges condenser outlet air to the conditioned space, and draws evaporator inlet air from and discharges evaporator outlet air to a localized zone within the conditioned space.

2.11 Standby mode means any mode where a portable air conditioner is connected to a mains power source and offers one or more of the following user-oriented or protective functions which may persist for an indefinite time:

(1) To facilitate the activation of other modes (including activation or deactivation of cooling mode) by remote switch (including remote control), internal sensor, or timer; or

(2) Continuous functions, including information or status displays (including clocks) or sensor-based functions. A timer is a continuous clock function (which may or may not be associated with a display) that provides regular scheduled tasks (e.g., switching) and that operates on a continuous basis.

## 3. Test Apparatus and General Instructions

3.1 Active mode.

3.1.1 Test conduct. The test apparatus and instructions for testing portable air conditioners in cooling mode and off-cycle mode shall conform to the requirements specified in Section 4, “Definitions” and Section 7, “Tests,” of AHAM PAC-1-2015 (incorporated by reference; see § 430.3), except as otherwise specified in this appendix. Where applicable, measure duct heat transfer and infiltration air heat transfer according to section 4.1.1.1 and section 4.1.1.2 of this appendix, respectively.

3.1.1.1 Duct setup.  Use ducting components provided by the manufacturer, including, where provided by the manufacturer, ducts, connectors for attaching the duct(s) to the test unit, and window mounting fixtures. Do not apply additional sealing or insulation.

3.1.1.2 Single-duct evaporator inlet test conditions.  When testing single-duct portable air conditioners, maintain the evaporator inlet dry-bulb temperature within a range of 1.0 °F with an average difference within 0.3 °F.

3.1.1.3 Condensate Removal. Setup the test unit in accordance with manufacturer instructions.  If the unit has an auto-evaporative feature, keep any provided drain plug installed as shipped and do not provide other means of condensate removal. If the internal condensate collection bucket fills during the test, halt the test, remove the drain plug, install a gravity drain line, and start the test from the beginning. If no auto-evaporative feature is available, remove the drain plug and install a gravity drain line. If no auto-evaporative feature or gravity drain is available and a condensate pump is included, or if the manufacturer specifies the use of an included condensate pump during cooling mode operation, then test the portable air conditioner with the condensate pump enabled.  For units tested with a condensate pump, apply the provisions in Section 7.1.2 of AHAM PAC-1-2015 (incorporated by reference; see § 430.3) if the pump cycles on and off.

3.1.1.4 Unit Placement. There shall be no less than 3 feet between any test chamber wall surface and any surface on the portable air conditioner, except the surface or surfaces of the portable air conditioner that include a duct attachment. The distance between the test chamber wall and a surface with one or more duct attachments is prescribed by the test setup requirements in Section 7.3.7 of AHAM PAC-1-2015 (incorporated by reference; see § 430.3).

3.1.1.5 Electrical supply. Maintain the input standard voltage at 115 V ±1 percent. Test at the rated frequency, maintained within ±1 percent.

3.1.1.6 Duct temperature measurements. Measure the surface temperatures of each duct using four equally spaced thermocouples per duct, adhered to the outer surface of the entire length of the duct. Temperature measurements must have an error no greater than ±0.5 °F over the range being measured.

3.1.2 Control settings. Set the controls to the lowest available temperature setpoint for cooling mode. If the portable air conditioner has a user-adjustable fan speed, select the maximum fan speed setting. If the portable air conditioner has an automatic louver oscillation feature, disable that feature throughout testing. If the louver oscillation feature is included but there is no option to disable it, testing shall proceed with the louver oscillation enabled. If the portable air conditioner has adjustable louvers, position the louvers parallel with the airflow to maximize air flow and minimize static pressure loss.

3.1.3 Measurement resolution and rounding. Record measurements at the resolution of the test instrumentation. Round the seasonally adjusted cooling capacity value in accordance with Table 1 of AHAM PAC-1-2015 (incorporated by reference; see § 430.3). Round CEER as calculated in section 5 of this appendix, to the nearest 0.1 Btu/Wh.

3.2 Standby mode and off mode.

3.2.1 Installation requirements. For the standby mode and off mode testing, install the portable air conditioner in accordance with Section 5, Paragraph 5.2 of IEC 62301 (incorporated by reference; see § 430.3), disregarding the provisions regarding batteries and the determination, classification, and testing of relevant modes.

3.2.2 Electrical energy supply.

3.2.2.1 Electrical supply. For the standby mode and off mode testing, maintain the input standard voltage at 115 V ±1 percent. Maintain the electrical supply at the rated frequency ±1 percent.

3.2.2.2 Supply voltage waveform. For the standby mode and off mode testing, maintain the electrical supply voltage waveform indicated in Section 4, Paragraph 4.3.2 of IEC 62301 (incorporated by reference; see § 430.3).

3.2.3 Standby mode and off mode wattmeter. The wattmeter used to measure standby mode and off mode power consumption must meet the requirements specified in Section 4, Paragraph 4.4 of IEC 62301 (incorporated by reference; see § 430.3).

3.2.4 Standby mode and off mode ambient temperature. For standby mode and off mode testing, maintain room ambient air temperature conditions as specified in Section 4, Paragraph 4.2 of IEC 62301 (incorporated by reference; see § 430.3).

## 4. Test Measurement

4.1 Cooling mode. Measure the indoor room cooling capacity and overall power input in cooling mode in accordance with Section 7.1.b and 7.1.c of AHAM PAC-1-2015 (incorporated by reference; see § 430.3), respectively. The test duration shall be determined in accordance with Section 8.7 of ASHRAE 37-2009 (incorporated by reference; § 430.3). Substitute the test conditions in Table 3 of AHAM PAC-1-2015 with the test conditions for single-duct and dual-duct portable air conditioners presented in Table 1 of this appendix. For single-duct units, measure the indoor room cooling capacity, $Capacity_{SD}$, and overall power input in cooling mode, $P_{SD}$, in accordance with the ambient conditions for test configuration 5, presented in Table 1 of this appendix. For dual-duct units, measure the indoor room cooling capacity and overall power input in accordance with ambient conditions for test configuration 3, condition A ($Capacity_{95}$, $P_{95}$), and a second time in accordance with the ambient conditions for test configuration 3, condition B ($Capacity_{83}$, $P_{83}$), presented in Table 1 of this appendix.

Table 1: Evaporator and Condenser Inlet Test Conditions

| Test Configuration | Evaporator Inlet Air, °F (°C) | | Condenser Inlet Air, °F (°C) | |
|---|---|---|---|---|
| | Dry Bulb | Wet Bulb | Dry Bulb | Wet Bulb |
| 3 (Condition A) | 80 (26.7) | 67 (19.4) | 95 (35.0) | 75 (23.9) |
| 3 (Condition B) | 80 (26.7) | 67 (19.4) | 83 (28.3) | 67.5 (19.7) |
| 5 | 80 (26.7) | 67 (19.4) | 80 (26.7) | 67 (19.4) |

4.1.1. Duct Heat Transfer. Measure the surface temperature of the condenser exhaust duct and condenser inlet duct, where applicable, throughout the cooling mode test. Calculate the average temperature at each individual location, and then calculate the average surface temperature of each duct by averaging the four average temperature measurements taken on that duct. Calculate the surface area ($A_{duct\_j}$) of each duct according to the following:

$$A_{duct\_j} = \pi \times d_j \times L_j$$

Where:

$d_j$ = the outer diameter of duct "j".

$L_j$ = the extended length of duct "j" while under test.

j represents the condenser exhaust duct and, for dual-duct units, condenser inlet duct.

Calculate the total heat transferred from the surface of the duct(s) to the indoor conditioned space while operating in cooling mode for the outdoor test conditions in Table 1 of this appendix, as follows. For single-duct portable air conditioners:

$Q_{duct\_SD} = h \times A_{duct\_j} \times (T_{duct\_SD\_j} - T_{ei})$

For dual-duct portable air conditioners:

$Q_{duct\_95} = \sum_j \{h \times A_{duct\_j} \times (T_{duct\_95\_j} - T_{ei})\}$

$Q_{duct\_83} = \sum_j \{h \times A_{duct\_j} \times (T_{duct\_83\_j} - T_{ei})\}$

Where:

$Q_{duct\_SD}$ = for single-duct portable air conditioners, the total heat transferred from the duct to the indoor conditioned space in cooling mode when tested according to the test conditions in Table 1 of this appendix, in Btu/h.

$Q_{duct\_95}$ and $Q_{duct\_83}$ = for dual-duct portable air conditioners, the total heat transferred from the ducts to the indoor conditioned space in cooling mode when tested according to the 95 °F dry-bulb and 83 °F dry-bulb outdoor test conditions in Table 1 of this appendix, in Btu/h.

h = convection coefficient, 4 Btu/h per square foot per °F.

$A_{duct\_j}$ = surface area of duct “j”, in square feet.

$T_{duct\_SD\_j}$ = average surface temperature for the condenser exhaust duct of single-duct portable air conditioners, as measured during testing according to the test condition in Table 1 of this appendix, in °F.

$T_{duct\_95\_j}$ and $T_{duct\_83\_j}$ = average surface temperature for duct “j” of dual-duct portable air conditioners, as measured during testing according to the two outdoor test conditions in Table 1 of this appendix, in °F.

j represents the condenser exhaust duct and, for dual-duct units, condenser inlet duct.

$T_{ei}$ = average evaporator inlet air dry-bulb temperature, in °F.

4.1.2 Infiltration Air Heat Transfer. Measure the heat contribution from infiltration air for single-duct portable air conditioners and dual-duct portable air conditioners that draw at least part of the condenser air from the conditioned space. Calculate the heat contribution from infiltration air for single-duct and dual-duct portable air conditioners for both cooling mode outdoor test conditions, as described in this section. The dry air mass flow rate of infiltration air shall be calculated according to the following equations. For single-duct portable air conditioners:

$$\dot{m}_{SD} = \frac{V_{co\_SD} \times \rho_{co\_SD}}{\left(1 + \omega_{co\_SD}\right)}$$

For dual-duct portable air conditioners:

$$\dot{m}_{95} = \left[\frac{V_{co\_95} \times \rho_{co\_95}}{\left(1 + \omega_{co\_95}\right)}\right] - \left[\frac{V_{ci\_95} \times \rho_{ci\_95}}{\left(1 + \omega_{ci\_95}\right)}\right]$$

$$\dot{m}_{83} = \left[\frac{V_{co\_83} \times \rho_{co\_83}}{\left(1 + \omega_{co\_83}\right)}\right] - \left[\frac{V_{ci\_83} \times \rho_{ci\_83}}{\left(1 + \omega_{ci\_83}\right)}\right]$$

Where:

$\dot{m}_{SD}$ = dry air mass flow rate of infiltration air for single-duct portable air conditioners, in pounds per minute (lb/m).

$\dot{m}_{95}$ and $\dot{m}_{83}$ = dry air mass flow rate of infiltration air for dual-duct portable air conditioners, as calculated based on testing according to the test conditions in Table 1 of this appendix, in lb/m.

$V_{co\_SD}$, $V_{co\_95}$, and $V_{co\_83}$ = average volumetric flow rate of the condenser outlet air during cooling mode testing for single-duct portable air conditioners; and at the 95 °F and 83 °F dry-bulb outdoor conditions for dual-duct portable air conditioners, respectively, in cubic feet per minute (cfm).

$V_{ci\_95}$, and $V_{ci\_83}$ = average volumetric flow rate of the condenser inlet air during cooling mode testing at the 95 °F and 83 °F dry-bulb outdoor conditions for dual-duct portable air conditioners, respectively, in cfm.

$\rho_{co\_SD}$, $\rho_{co\_95}$, and $\rho_{co\_83}$ = average density of the condenser outlet air during cooling mode testing for single-duct portable air conditioners, and at the 95 °F and 83 °F dry-bulb outdoor conditions for dual-duct portable air conditioners, respectively, in pounds mass per cubic foot ($lb_m/ft^3$).

$\rho_{ci\_95}$, and $\rho_{ci\_83}$ = average density of the condenser inlet air during cooling mode testing at the 95 °F and 83 °F dry-bulb outdoor conditions for dual-duct portable air conditioners, respectively, in $lb_m/ft^3$.

$\omega_{co\_SD}$, $\omega_{co\_95}$, and $\omega_{co\_83}$ = average humidity ratio of condenser outlet air during cooling mode testing for single-duct portable air conditioners, and at the 95 °F and 83 °F dry-bulb outdoor conditions for dual-duct portable air conditioners, respectively, in pounds mass of water vapor per pounds mass of dry air ($lb_w/lb_{da}$).

$\omega_{ci\_95}$, and $\omega_{ci\_83}$ = average humidity ratio of condenser inlet air during cooling mode testing at the 95 °F and 83 °F dry-bulb outdoor conditions for dual-duct portable air conditioners, respectively, in $lb_w/lb_{da}$.

For single-duct and dual-duct portable air conditioners, calculate the sensible component of infiltration air heat contribution according to the following:

$$Q_{s\_95} = \dot{m} \times 60 \times \left[\left(c_{p\_da} \times \left(T_{ia\_95} - T_{indoor}\right)\right) + c_{p\_wv} \times \left(\omega_{ia\_95} \times T_{ia\_95} - \omega_{indoor} \times T_{indoor}\right)\right]$$

$$Q_{s\_83} = \dot{m} \times 60 \times \left[\left(c_{p\_da} \times \left(T_{ia\_83} - T_{indoor}\right)\right) + c_{p\_wv} \times \left(\omega_{ia\_83} \times T_{ia\_83} - \omega_{indoor} \times T_{indoor}\right)\right]$$

Where:

$Q_{s\_95}$ and $Q_{s\_83}$ = sensible heat added to the room by infiltration air, calculated at the 95 °F and 83 °F dry-bulb outdoor conditions in Table 1 of this appendix, in Btu/h.

$\dot{m}$ = dry air mass flow rate of infiltration air, $\dot{m}_{SD}$ or $\dot{m}_{95}$ when calculating $Q_{s\_95}$ and $\dot{m}_{SD}$ or $\dot{m}_{83}$ when calculating $Q_{s\_83}$, in lb/m.

$c_{p\_da}$ = specific heat of dry air, 0.24 Btu/$lb_m$-°F.

$c_{p\_wv}$ = specific heat of water vapor, 0.444 Btu/$lb_m$-°F.

$T_{indoor}$ = indoor chamber dry-bulb temperature, 80 °F.

$T_{ia\_95}$ and $T_{ia\_83}$ = infiltration air dry-bulb temperatures for the two test conditions in Table 1 of this appendix, 95 °F and 83 °F, respectively.

$\omega_{ia\_95}$ and $\omega_{ia\_83}$= humidity ratios of the 95 °F and 83 °F dry-bulb infiltration air, 0.0141 and 0.01086 $lb_w/lb_{da}$, respectively.

$\omega_{indoor}$ = humidity ratio of the indoor chamber air, 0.0112 $lb_w/lb_{da}$.

60 = conversion factor from minutes to hours.

Calculate the latent heat contribution of the infiltration air according to the following:

$$Q_{l\_95} = \dot{m} \times 60 \times H_{fg} \times (\omega_{ia\_95} - \omega_{indoor})$$

$$Q_{l\_83} = \dot{m} \times 60 \times H_{fg} \times (\omega_{ia\_83} - \omega_{indoor})$$

Where:

$Q_{l\_95}$ and $Q_{l\_83}$ = latent heat added to the room by infiltration air, calculated at the 95°F and 83 °F dry-bulb outdoor conditions in Table 1 of this appendix, in Btu/h.

$\dot{m}$ = mass flow rate of infiltration air, $\dot{m}_{SD}$ or $\dot{m}_{95}$ when calculating $Q_{l,95}$ and $\dot{m}_{SD}$ or $\dot{m}_{83}$ when calculating $Q_{l\_83}$, in lb/m.

$H_{fg}$ = latent heat of vaporization for water vapor, 1061 Btu/$lb_m$.

$\omega_{ia\_95}$ and $\omega_{ia\_83}$ = humidity ratios of the 95 °F and 83 °F dry-bulb infiltration air, 0.0141 and 0.01086 $lb_w/lb_{da}$, respectively.

$\omega_{indoor}$ = humidity ratio of the indoor chamber air, 0.0112 $lb_w/lb_{da}$.

60 = conversion factor from minutes to hours.

The total heat contribution of the infiltration air is the sum of the sensible and latent heat:

$$Q_{infiltration\_95} = Q_{s\_95} + Q_{l\_95}$$

$$Q_{infiltration\_83} = Q_{s\_83} + Q_{l\_83}$$

Where:

$Q_{infiltration\_95}$ and $Q_{infiltration\_83}$ = total infiltration air heats in cooling mode, calculated at the 95°F and 83 °F dry-bulb outdoor conditions in Table 1 of this appendix, in Btu/h.

$Q_{s\_95}$ and $Q_{s\_83}$ = sensible heat added to the room by infiltration air, calculated at the 95 °F and 83 °F dry-bulb outdoor conditions in Table 1 of this appendix, in Btu/h.

$Q_{l\_95}$ and $Q_{l\_83}$ = latent heat added to the room by infiltration air, calculated at the 95 °F and 83 °F dry-bulb outdoor conditions in Table 1 of this appendix, in Btu/h.

4.2 Off-cycle mode. Establish the test conditions specified in section 3.1.1 of this appendix for off-cycle mode, except that the duct measurements in section 3.1.1.6 shall not be used and the wattmeter specified in section 3.2.3 of this appendix shall be used. Begin the off-cycle mode test period 5 minutes following the cooling mode test period. Adjust the setpoint higher than the ambient temperature to ensure the product will not enter cooling mode and begin the test 5 minutes after the compressor cycles off due to the change in setpoint. The off-cycle mode test period shall be 2 hours in duration, during which the power consumption is recorded at the same intervals as recorded for cooling mode testing. Measure and record the average off-cycle mode power of the portable air conditioner, $P_{oc}$, in watts.

4.3 Standby mode and off mode. Establish the testing conditions set forth in section 3.2 of this appendix, ensuring that the portable air conditioner does not enter any active modes during the test. For portable air conditioners that take some time to enter a stable state from a higher power state as discussed in Section 5, Paragraph 5.1, Note 1 of IEC 62301, (incorporated by reference; see § 430.3), allow sufficient time for the portable air conditioner to reach the lowest power state before proceeding with the test measurement. Follow the test procedure specified in Section 5, Paragraph 5.3.2 of IEC 62301 for testing in each possible mode as described in sections 4.3.1 and 4.3.2 of this appendix.

4.3.1 If the portable air conditioner has an inactive mode, as defined in section 2.5 of this appendix, but not an off mode, as defined in section 2.7 of this appendix, measure and record the average inactive mode power of the portable air conditioner, $P_{ia}$, in watts.

4.3.2 If the portable air conditioner has an off mode, as defined in section 2.7 of this appendix, measure and record the average off mode power of the portable air conditioner, $P_{om}$, in watts.

## 5. Calculation of Derived Results From Test Measurements

5.1 Adjusted Cooling Capacity. Calculate the adjusted cooling capacities for portable air conditioners, $ACC_{95}$ and $ACC_{83}$, expressed in Btu/h, according to the following equations. For single-duct portable air conditioners:

$$ACC_{95} = Capacity_{SD} - Q_{duct\_SD} - Q_{infiltration\_95}$$

$$ACC_{83} = Capacity_{SD} - Q_{duct\_SD} - Q_{infiltration\_83}$$

For dual-duct portable air conditioners:

$$ACC_{95} = Capacity_{95} - Q_{duct\_95} - Q_{infiltration\_95}$$

$$ACC_{83} = Capacity_{83} - Q_{duct\_83} - Q_{infiltration\_83}$$

Where:

$Capacity_{SD}$, $Capacity_{95}$, and $Capacity_{83}$ = cooling capacity measured in section 4.1.1 of this appendix.

$Q_{duct\_SD}$, $Q_{duct\_95}$, and $Q_{duct\_83}$ = duct heat transfer while operating in cooling mode, calculated in section 4.1.1.1 of this appendix.

$Q_{infiltration\_95}$ and $Q_{infiltration\_83}$ = total infiltration air heat transfer in cooling mode, calculated in section 4.1.1.2 of this appendix.

5.2 Seasonally Adjusted Cooling Capacity. Calculate the seasonally adjusted cooling capacity for portable air conditioners, SACC, expressed in Btu/h, according to the following:

$$SACC = ACC_{95} \times 0.2 + ACC_{83} \times 0.8$$

Where:

$ACC_{95}$ and $ACC_{83}$ = adjusted cooling capacity, in Btu/h, calculated in section 5.1 of this appendix.

0.2 = weighting factor for $ACC_{95}$.

0.8 = weighting factor for $ACC_{83}$.

5.3 Annual Energy Consumption. Calculate the annual energy consumption in each operating mode, $AEC_m$, expressed in kilowatt-hours per year (kWh/year). The annual hours of operation in each mode are estimated as follows:

| Operating Mode | Annual Operating Hours |
|---|---|
| Cooling Mode, Dual-Duct 95 °F[1] | 750 |
| Cooling Mode, Dual-Duct 83 °F[1] | 750 |
| Cooling Mode, Single-Duct | 750 |
| Off-Cycle | 880 |
| Inactive or Off | 1,355 |

[1] These operating mode hours are for the purposes of calculating annual energy consumption under different ambient conditions for dual-duct portable air conditioners, and are not a division of the total cooling mode operating hours. The total dual-duct cooling mode operating hours are 750 hours.

$$AEC_m = P_m \times t_m \times k$$

Where:

$AEC_m$ = annual energy consumption in each mode, in kWh/year.

$P_m$ = average power in each mode, in watts.

m represents the operating mode ("95" and "83" cooling mode at the 95 °F and 83 °F dry-bulb outdoor conditions, respectively for dual-duct portable air conditioners, "SD" cooling mode for single-duct portable air conditioners, "oc" off-cycle, and "ia" inactive or "om" off mode).

t = number of annual operating time in each mode, in hours.

k = 0.001 kWh/Wh conversion factor from watt-hours to kilowatt-hours.

Total annual energy consumption in all modes except cooling, is calculated according to the following:

$$AEC_T = \sum_m AEC_m$$

Where:

$AEC_T$ = total annual energy consumption attributed to all modes except cooling, in kWh/year;

$AEC_m$ = total annual energy consumption in each mode, in kWh/year.

m represents the operating modes included in $AEC_T$ ("oc" off-cycle, and "im" inactive or "om" off mode).

5.4 Combined Energy Efficiency Ratio. Using the annual operating hours, as outlined in section 5.3 of this appendix, calculate the combined energy efficiency ratio, CEER, expressed in Btu/Wh, according to the following:

$$CEER_{SD} = \left[\frac{(ACC_{95} \times 0.2 + ACC_{83} \times 0.8)}{\left(\frac{AEC_{SD} + AEC_T}{k \times t}\right)}\right]$$

$$CEER_{DD} = \left[\frac{ACC_{95}}{\left(\frac{AEC_{95} + AEC_T}{k \times t}\right)}\right] \times 0.2 + \left[\frac{ACC_{83}}{\left(\frac{AEC_{83} + AEC_T}{k \times t}\right)}\right] \times 0.8$$

Where:

$CEER_{SD}$ and $CEER_{DD}$ = combined energy efficiency ratio for single-duct and dual-duct portable air conditioners, respectively, in Btu/Wh.

$ACC_{95}$ and $ACC_{83}$ = adjusted cooling capacity, tested at the 95 °F and 83 °F dry-bulb outdoor conditions in Table 1 of this appendix, in Btu/h, calculated in section 5.1 of this appendix.

$AEC_{SD}$ = annual energy consumption in cooling mode for single-duct portable air conditioners, in kWh/year, calculated in section 5.3 of this appendix.

$AEC_{95}$ and $AEC_{83}$ = annual energy consumption for the two cooling mode test conditions in Table 1 of this appendix for dual-duct portable air conditioners, in kWh/year, calculated in section 5.3 of this appendix.

$AEC_T$ = total annual energy consumption attributed to all modes except cooling, in kWh/year, calculated in section 5.3 of this appendix.

t = number of cooling mode hours per year, 750.

k = 0.001 kWh/Wh conversion factor for watt-hours to kilowatt-hours.

0.2 = weighting factor for the 95 °F dry-bulb outdoor condition test.

0.8 = weighting factor for the 83 °F dry-bulb outdoor condition test.

[FR Doc. 2015-30057 Filed: 11/25/2015 8:45 am; Publication Date:  11/27/2015]