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[6450-01-P]

DEPARTMENT OF ENERGY

10 CFR Part 430

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

RIN: 1904 – AD22

Energy Conservation Program for Consumer Products: Test Procedure for Portable Air Conditioners

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

ACTION: Notice of data availability; request for comment.

SUMMARY: In a notice of proposed determination (NOPD) published on July 5, 2013, the U.S. Department of Energy (DOE) tentatively determined that portable air conditioners (ACs) qualify as a covered product under Part B of Title III of the Energy Policy and Conservation Act (EPCA), as amended. To assist in a final determination and to consider approaches for a future DOE test procedure for these products, should DOE determine that portable ACs are covered products, DOE conducted investigative testing to evaluate industry test procedures that could be used to measure cooling capacity and energy use for portable ACs. In today's notice, DOE discusses various industry test procedures and presents results from its investigative testing that evaluated existing methodologies and alternate approaches adapted from these methodologies for portable ACs. DOE requests comment and additional information regarding the testing and

results presented in this NODA. DOE also encourages interested parties to provide comment on any alternate approaches for testing portable ACs and information that may improve the analysis.

DATES: DOE will accept comments, data, and information regarding this notice of data availability (NODA) submitted no later than [**INSERT DATE 30 DAYS AFTER DATE OF PUBLICATION IN THE FEDERAL REGISTER**].

ADDRESSES: Any comments submitted must identify the Notice of Data Availability for Portable Air Conditioners, and provide docket number EERE-2014-BT-TP-0014 and/or RIN 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 docket EERE-2014-BT-TP-0014 and/or RIN 1904 – AD22 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 compact disc (CD), in which case it is not necessary to include printed copies.
4. Hand Delivery/Courier: Ms. Brenda Edwards, U.S. Department of Energy, Building Technologies Program, 6th Floor, 950 L'Enfant Plaza SW., Washington, DC 20024. Telephone: (202) 586-2945. If possible, please submit all items on a CD, in which case it is not necessary to include printed copies.

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

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 contains a link to the docket for this notice on the www.regulations.gov site. The www.regulations.gov web page contains simple instructions on how to access all documents, including public comments, in the docket.

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 email: Brenda.Edwards@ee.doe.gov.

FOR FURTHER INFORMATION CONTACT:

Mr. Bryan Berringer, U.S. Department of Energy, Office of Building Technology Office, EE-5B, 950 L'Enfant Plaza SW. Room 603, 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-71, 1000 Independence Ave., SW, Washington, D.C. 20585-0121. Telephone: 202-586-2902; E-mail: Sarah.Butler@hq.doe.gov.

SUPPLEMENTARY INFORMATION:

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I. Authority and Background

Title III, Part B[1] of the Energy Policy and Conservation Act of 1975 (EPCA), Pub. L. 94-163, (42 U.S.C. 6291–6309, as codified), sets forth a variety of provisions designed to improve energy efficiency and established the Energy Conservation Program for Consumer Products Other Than Automobiles, a program covering most major household appliances (hereinafter

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

referred to as “covered products”).[2] In addition to specifying a list of covered products, EPCA contains provisions that enable the Secretary of Energy to classify additional types of consumer products as covered products. 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 such type is likely to exceed 100 kilowatt-hours (kWh) per year. (42 U.S.C. 6292(b)(1)).

In order 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 any 12-month period ending before such determination;

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

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

(4) Application of a labeling rule under 42 U.S.C. 6294 is not likely 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)).

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[2] All references to EPCA in this document refer to the statute as amended through the American Manufacturing Technical Corrections Act (AEMTCA), Public Law 112-210 (Dec. 18, 2012).

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 for covered products. In reaching this tentative determination, DOE found that classifying products of such type as covered products is necessary or appropriate to carry out the purposes of EPCA, and the average U.S. household energy use for portable ACs is likely to exceed 100 kWh per year. 78 FR 40403–07.

In response to the July 2013 NOPD, DOE received comments from interested parties on several topics, including appropriate test procedures for portable ACs that DOE should consider if it issues a final determination that classifies portable ACs as covered products. Consumer Reports recommended that portable ACs be tested similar to, and performance compared with, room ACs because they are seen by consumers as comparable products that perform nearly identical functions. (Consumer Reports, No. 2 at p.2).[3] In addition, the Appliance Standards Awareness Project (ASAP), American Council for an Energy-Efficient Economy (ACEEE), Consumers Union (CU), Natural Resources Defense Council (NRDC), and Northwest Energy Efficiency Alliance (NEEA), (hereinafter referred to as the "Joint Commenters"), commented that any portable AC test procedure must facilitate a realistic comparison with room ACs, and that a portable AC test procedure must reflect actual installation and operation to determine a meaningful and applicable cooling capacity and Energy Efficiency Ratio (EER). (Joint Commenters, No. 4 at p. 2).

[3] A notation in the form "Consumer Reports, No. 2 at p. 2" identifies a written comment: (1) made by Consumer Reports; (2) recorded in document number 2 that is filed in the docket of the portable AC determination of coverage rulemaking (Docket No. EERE–2013– BT–STD–0033) and available for review at www.regulations.gov; and (3) which appears on page 2 of document number 2.

The Pacific Gas and Electric Company (PG&E), Southern California Gas Company (SCGC), San Diego Gas and Electric (SDG&E), and Southern California Edison (SCE), (hereinafter referred to as the "California IOUs"), commented that based on Consumer Reports' testing, the published ratings for portable ACs may underestimate actual performance in the field by approximately 50 percent. The California IOUs recommended establishing a standardized test procedure to ensure that representations of portable AC energy use would better reflect actual usage and be more meaningful for consumers making purchasing decisions. (California IOUs, No. 5 at p. 3)

The Association of Home Appliance Manufacturers (AHAM) commented that a DOE test procedure would ensure that all manufacturers test and rate their products according to the same test procedure. AHAM also suggested that DOE incorporate current test procedures by reference, particularly the version of AHAM's portable AC test procedure which is currently under development to harmonize with the Canadian Standards Association (CSA) test procedure. AHAM commented that DOE should work with Natural Resources Canada (NRCan) and CSA to harmonize the U.S. and Canadian test procedures for portable ACs. (AHAM, No. 6 at pp. 2–4)

DOE agrees that a DOE test procedure for portable ACs would provide consistency and clarity for representations of energy use of these products. DOE is evaluating available industry test procedures to determine whether their methodologies are suitable for incorporation in a future DOE test procedure, should DOE determine that portable ACs are a covered product.

II. Discussion

In the July 2013 NOPD, DOE proposed defining a portable AC as "a consumer product, other than a 'packaged terminal air conditioner,' which is powered by a single-phase electric current and which is an encased assembly designed as a portable unit that may rest on the floor or other elevated surface for the purpose of providing delivery of conditioned air to an enclosed space. It includes a prime source of refrigeration and may include a means for ventilating and heating." 78 FR 40403, 40404 (Jul. 5, 2013). The most common type of portable AC configuration in the United States utilizes a single condenser air exhaust duct that removes heat to the unconditioned space. Other configurations include dual-duct, which intakes and exhausts unconditioned air to cool the condenser and remove moisture, and spot coolers, which have no ducting on the condenser side and may utilize small directional ducts on the evaporator exhaust.

In response to comments from interested parties, DOE conducted testing to determine typical portable AC cooling capacities and energy efficiencies based on the existing industry test methods and to investigate their applicability to a possible DOE test procedure for portable ACs. DOE is aware of three test procedures that measure portable AC performance and that are applicable to products sold in North America.

(1) American National Standards Institute (ANSI)/AHAM PAC-1-2009 "Portable Air Conditioners"[4] (ANSI/AHAM PAC-1-2009) specifies cooling mode testing conducted in accordance with ANSI/American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE) Standard 37-2005 "Methods of Testing for

[4] ANSI/AHAM test procedures are available for purchase online at: www.aham.org.

Rating Electrically Driven Unitary Air-Conditioning and Heat Pump Equipment" (ANSI/ASHRAE Standard 37-2005).[5] The metrics incorporated in ANSI/AHAM PAC-1-2009 include cooling capacity and EER for the following configurations: Single-Duct, Dual-Duct, Spot Cooling, and Water Cooled Condenser.

(2) CSA C370-2013 "Cooling Performance of Portable Air Conditioners"[6] (CSA C370) is harmonized with ANSI/AHAM PAC-1-2009, and thus also incorporates testing provisions from ANSI/ASHRAE Standard 37, although it specifies the later 2009 version.

(3) ANSI/ASHRAE Standard 128-2011 "Method of Rating Unitary Spot Air Conditioners" (ANSI/ASHRAE Standard 128-2011) is adapted from the previous 2009 version of CSA C370. It too references ANSI/ASHRAE Standard 37-2009. The previous version of ANSI/ASHRAE Standard 128, published in 2001, is required by California regulations to be used to certify spot cooler performance for such products sold in that State. A key difference between ANSI/ASHRAE Standard 128-2011 and ANSI/ASHRAE Standard 128-2001 is that the older version specifies a higher indoor ambient testing temperature, which increases measured cooling capacity and EER.

DOE found no significant differences that would provide varying results among the AHAM, CSA, and ASHRAE test procedures. In reviewing the current versions of these test procedures, DOE observed that each measures cooling capacity and EER based on an air

[5] ANSI/ASHRAE Standard 37 was updated in 2009. DOE reviewed the 2005 and 2009 versions and concluded there would be no measurable difference in portable AC results obtained from each. Therefore, DOE utilized ANSI/ASHRAE Standard 37-2009 when testing according to ANSI/AHAM PAC-1-2009. ANSI/ASHRAE test procedures are available for purchase online at: www.techstreet.com.

[6] CSA test procedures are available for purchase online at: www.csagroup.org.

enthalpy approach that measures the airflow rate, dry-bulb temperature, and water vapor content of air at the inlet and outlet of the indoor (evaporator) side. In addition, for air-cooled portable ACs with cooling capacities less than 135,000 British thermal units per hour (Btu/h), which include the products that are the subject of today's notice, the indoor air enthalpy results must be validated by additionally measuring cooling capacity by either an outdoor air enthalpy method or a compressor calibration method. In its testing, DOE selected the outdoor air enthalpy method to minimize its test burden because that approach only requires additional metering components, similar to those used for the indoor air enthalpy method. The compressor calibration method requires monitoring refrigerant conditions with additional equipment that was not available at the time in the test laboratory. DOE expects that using either approach would produce equivalent results because the compressor calibration approach measures the heat transferred to the refrigerant from the evaporator side and the outdoor air enthalpy approach measures that same heat when it is transferred from the refrigerant and rejected at the condenser side.

DOE conducted initial testing according to ANSI/AHAM PAC-1-2009 to establish baseline cooling capacities and efficiencies of the test units according to the existing industry test procedures. As noted previously, although ANSI/AHAM PAC-1-2009 references ANSI/ASHRAE Standard 37-2005, DOE determined there were no differences in the relevant provisions between this version and the current version, ANSI/ASHRAE Standard 37-2009.[7] DOE, therefore, used ANSI/ASHRAE Standard 37-2009 for all testing according to ANSI/AHAM PAC-1-2009.

[7] Both versions of ANSI/ASHRAE Standard 37 are available for purchase online at: www.techstreet.com.

In addition, DOE reviewed information suggesting that certain operational factors not addressed in existing test procedures could have a significant effect on portable AC performance. For example, a Consumer Reports buying guide indicates that units tested as part of a field study delivered only half of the rated cooling capacity.[8] DOE observed that when condenser air is drawn from the conditioned space and exhausted to the unconditioned space, a pressure gradient is created that results in replacement air infiltrating into the conditioned space. If this infiltration air is drawn from unconditioned locations, including possibly directly from outdoors through leaky windows or mounting brackets, the net cooling capacity and EER of the portable AC would be reduced. DOE notes that this air infiltration likely has the largest effect on the performance of single-duct units because these units intake all condenser air from the conditioned space. Dual-duct units may intake a portion of condenser air from the conditioned space; the remainder, which may be all of the condenser air, is drawn from outdoors through the condenser inlet duct. If air infiltration is not accounted for, testing may suggest that a single-duct unit would perform better than a dual-duct unit with comparable components. Single-duct units utilize lower-temperature air from the conditioned space to cool the condenser and it would appear that these units are able to operate more efficiently than equivalent dual-duct units.

Portable AC performance may also be reduced due to the heat transfer to the room through leaks in the product case and manufacturer-provided ducting that is not addressed in current test procedures. The portable AC and all associated equipment are located in the conditioned space and the ducting is typically flexible plastic with no additional insulation. Further, the connection between the duct and the case and the connection between the duct and

[8] Consumer Reports, Buying Advice: Portable Air Conditioners, June 20, 2008. Available online at: www.consumerreports.org/cro/news/2008/06/buying-advice-portable-air-conditioners/index.htm.

the manufacturer-supplied window fixture may not be tightly sealed, allowing some condenser-side air to leak into the room. Finally, DOE observed that mixing may occur between the condenser air exhaust and intake for dual-duct units because the window fixtures typically locate the air intake and exhaust connections adjacent to one another, allowing some of the hotter exhaust air to potentially short-circuit and enter the intake duct.

To investigate the contribution of these operational factors on the apparent reduction in cooling capacity observed for units in the field, DOE compared the results of ANSI/AHAM PAC-1-2009 testing with the results of additional testing using a test room calorimeter approach based on ANSI/ASHRAE Standard 16-1983 (RA 99), “Method of Testing for Rating Room Air Conditioners and Packaged Terminal Air Conditioners” (ANSI/ASHRAE Standard 16-1983), with certain modifications as explained below to allow testing of portable ACs. The room calorimeter approach would allow DOE to determine the cooling capacity and associated EER of a portable AC that accounts for any air infiltration effects and heat transfer to the conditioned space through gaps in the product case and seams in the duct connections. Values of these performance metrics measured accordingly may more accurately reflect real-world portable AC operation. In this test series, DOE also investigated cooling capacity and EER as a function of the infiltration air temperature for single-duct and dual-duct units, and the effect of condenser exhaust air entrainment at the intake for dual-duct portable ACs.

The following sections detail the units in DOE’s test sample, the baseline test results obtained using ANSI/AHAM PAC-1-2009, and the results from the investigative tests using the modified ANSI/ASHRAE Standard 16-1983 to estimate the effects of infiltration air, case and duct heat transfer, and condenser duct air mixing.

A. Test Units

For its portable AC testing, DOE selected a sample of units that are representative of products and configurations currently available on the U.S. market. The test sample included four single-duct, two dual-duct, and two spot-cooling portable ACs, covering a range of rated cooling capacities (8,000–13,500 Btu/h) and EERs (7.0–11.2 Btu per watt-hour (Btu/Wh)). Because DOE does not currently require manufacturers to certify portable ACs to any energy conservation standards, manufacturers may advertise or market their products using any available test procedure. For models that are included in the California Energy Commission (CEC) product database and that are sold in California, however, manufacturers must report cooling capacity and EER according to ANSI/ASHRAE Standard 128-2001. DOE notes that the cooling capacities and EERs obtained from using ANSI/ASHRAE Standard 128-2001 are higher than those obtained using the current ANSI/ASHRAE Standard 128-2011, primarily due to higher temperature evaporator inlet air.[9]

Due to the consistent method of reporting, DOE selected units for its test sample largely from cooling capacities and EERs listed in the CEC product database. Where values were not available in the CEC product database, DOE utilized information from manufacturer literature to

[9] ANSI/ASHRAE Standard 128-2011 specifies 80.6 degrees Fahrenheit (°F) dry-bulb temperature and 66.2 °F wet-bulb temperature for the standard rating conditions for the evaporator inlet of dual-duct portable ACs and both the evaporator and condenser inlets of single-duct units. It also specifies standard rating conditions of 95 °F dry-bulb temperature and 75.2 °F wet-bulb temperature for the condenser inlet side of dual-duct portable ACs and both the evaporator and condenser inlets of spot coolers. ANSI/ASHRAE Standard 128-2001 specified 95 °F dry-bulb temperature and 83 °F wet-bulb temperature for the standard rating conditions for both the evaporator and condenser inlets of all portable ACs, including spot coolers.

inform its selection. However, due to the difference in testing temperature, DOE expected that these values would differ from the cooling capacities and EERs that would be obtained using any one of the three industry test methods. The eight test units and their key features are presented in Table II.1, with cooling capacity expressed in Btu/h and EER expressed in Btu/Wh. DOE included two spot coolers in the test sample that, unlike the majority of spot coolers which are designed for commercial applications, have supply power requirements that would allow them to be used in residential applications.

Table II.1 Portable Air Conditioner Test Units and Features

Test Unit	**Duct Type**	**Rated Cooling Capacity** (Btu/h)	**Rated EER** (Btu/Wh)
SD1	Single	8,000	7.0
SD2	Single	9,500	9.6
SD3	Single	12,000	8.7
SD4	Single	13,000	9.7
DD1	Dual	9,500	9.4
DD2	Dual	13,000	8.9
SC1	Spot Cooler	10,000	10.1
SC2	Spot Cooler	13,500	11.2

B. Baseline Testing

DOE performed testing according to ANSI/AHAM PAC-1-2009 to determine baseline performance when using the industry standards. ANSI/AHAM PAC-1-2009 requires two-chamber air enthalpy testing for single-duct and dual-duct units, and a single-chamber setup for spot coolers. For each ducted configuration, the portable AC and any associated ducting is located entirely within a chamber held at “indoor” standard rating conditions at the evaporator inlet of 80 degrees Fahrenheit (°F) dry-bulb temperature and 67 °F wet-bulb temperature, which correspond to 51-percent relative humidity. For the condenser side exhaust on single-duct and dual-duct units, the manufacturer-supplied or manufacturer-specified flexible ducting connects the unit under test to a separate test chamber maintained at "outdoor" standard rating conditions. The outdoor conditions specify 95 °F dry-bulb temperature[10] and 75 °F wet-bulb temperature (40-percent relative humidity) at the condenser inlet for dual-duct units. The outdoor conditions for single-duct units, however, are not explicitly specified. ANSI/AHAM PAC-1-2009 only requires that the condenser inlet conditions, which would be set by air intake from the indoor side chamber, be maintained at 80 °F dry-bulb temperature and 67 °F wet-bulb temperature. Because the single-duct condenser air is discharged to the outdoor side with no intake air from that location, DOE does not believe that the results obtained using ANSI/AHAM PAC-1-2009 would be measurably affected by the conditions in the outdoor side chamber. Nonetheless, for consistency with the testing of dual-duct units, DOE chose to maintain the outdoor side conditions, measured near to the condenser exhaust but not close enough to be affected by that airflow, at 95 °F dry-bulb temperature and 75 °F wet-bulb temperature. For spot coolers, ANSI/AHAM PAC-1-2009 specifies testing the unit in a chamber maintained at the outdoor standard rating conditions of 95 °F dry-bulb temperature and 75 °F wet-bulb temperature.

Section 6.1 of ANSI/AHAM PAC-1-2009, "Method of Test," instructs that the details of testing are as specified in ANSI/ASHRAE Standard 37-2005, with references in Section 8.5.1 of ANSI/ASHRAE Standard 37-2005 to the indoor side (the cooling, or evaporator, side) of the portable AC under test and references to the outdoor side (the heat rejection, or condenser, side). No additional instructions regarding the specific provisions to use in ANSI/ASHRAE Standard

[10] Table 3 of ANSI/AHAM PAC-1-2009, which specifies standard rating conditions, lists a dry-bulb temperature of 94 °F and a wet-bulb temperature of 75 °F for single-duct and dual-duct portable ACs. DOE expects this to be a typographical error, and that the correct dry-bulb temperature is 95 °F.

37-2005 are included. As discussed previously, DOE utilized the latest version of ANSI/ASHRAE Standard 37, published in 2009. The following paragraphs describe the clauses from ANSI/ASHRAE Standard 37-2009 that DOE decided were appropriate for conducting its baseline tests.

The test apparatus (i.e., ducts, air flow-measurement nozzle, and additional instrumentation) were adjusted according to Section 8.6, "Additional Requirements for the Outdoor Air Enthalpy Method," of ANSI/ASHRAE Standard 37-2009, which ensures that air flow rate and static pressure in the condenser exhaust air stream, and condenser inlet air stream for dual-duct units, are representative of actual installations. The test room conditioning apparatus and the units under test were then operated until steady-state performance was achieved according to the specified test tolerances in Section 8.7, "Test Procedure for Cooling Capacity Tests," of ANSI/ASHRAE Standard 37-2009. Airflow rate, dry-bulb temperature, and water vapor content were recorded to evaluate cooling capacity at equal intervals that spanned five minutes or less until readings over one-half hour were within the same tolerances, as required by that section.

These collected data were then used to calculate total, sensible, and latent indoor cooling capacity based on the equations in Section 7.3.3, "Cooling Calculations," of ANSI/ASHRAE Standard 37-2009. This section provides calculations to determine indoor cooling capacity based on both the indoor and outdoor air enthalpy methods. As described in Section 7.3.3.3 of ANSI/ASHRAE Standard 37-2009, the indoor air enthalpy cooling capacity calculation was adjusted for heat transferred from the surface of the duct(s) to the conditioned space. DOE estimated a convective heat transfer coefficient of 4 Btu/h per square foot per °F, based on a midpoint of values for forced convection and free convection as recommended by the test laboratory for this specific test and setup. Four thermocouples were placed in a grid on the surface of the condenser duct(s). The heat transfer was determined by multiplying the estimated heat transfer coefficient by the surface area of each component and by the average temperature difference between the duct surface and test chamber air.

Although ANSI/AHAM PAC-1-2009 specifies that the evaporator circulating fan heat shall be included in the total cooling capacity, DOE did not meter the fan power for testing. Rather, for ducted units, DOE estimated the heat transferred to the conditioned space based on the temperature differential between the case surfaces and the indoor room, with measurements and calculations similar to those used for the ducts. This estimate was made by placing four thermocouples on each surface of the case and measuring the surface area to determine the heat transfer. This approach directly estimates the heating contribution of all internal components within the case to the cooling capacity, while making no assumption regarding whether the heat from individual components is transferred to the cooling or heat rejection side. Although ANSI/AHAM PAC-1-2009 requires the evaporator circulating fan heat be addressed in the cooling capacity for all portable ACs including spot coolers, DOE decided not to include case heat transfer for spot coolers because these units reject all heat directly to the space where the unit sits. That rejected heat does not impact the cooling provided by the unit to the specific conditioned spot.

Section 10.1.2 of ANSI/ASHRAE Standard 37-2009 requires that the calculated indoor cooling capacities from each method shall agree within 6.0 percent for a valid test. From the calculated cooling capacity, DOE determined the associated EER consistent with the definitions in Sections 3.8 to 3.10 and ratings requirements in Sections 5.3 to 5.5 of ANSI/AHAM PAC-1-2009. Table II.2 shows the results of the baseline testing for all test units according to ANSI/AHAM PAC-1-2009.

Table II.2 Baseline Test Results

Test Unit	Cooling Capacity (Btu/h)			EER (Btu/Wh)		
	Rated	**Baseline**	**Reduction (%)**	**Rated**	**Baseline**	**Reduction (%)**
SD1	8,000	5,842.7	27.0	7.0	6.84	2.3
SD2	9,500	6,599.8	30.5	9.6	7.41	22.8
SD3	12,000	10,947.6	8.8	8.7	7.47	14.1
SD4	13,000	9,505.6	26.9	9.7	6.59	32.0
DD1	9,500	8,597.2	9.5	9.4	7.41	21.2
DD2	13,000	7,211.2	44.5	8.9	5.50	38.2
SC1	10,000	10,225.7	-2.3	10.1	9.62	4.7
SC2	13,500	10,774.7	20.19	11.2	6.72	39.9

For all units, the tested cooling capacity and EER were on average 19 percent and 21 percent lower, respectively, when compared with the rated values. However, the difference between tested and rated cooling capacity ranged from an increase of 2.3 percent to a decrease of over 44 percent, while the tested EERs showed a reduction from 2.3 to 40 percent compared to the rated values. DOE notes that cooling capacity and EER for single-duct units were lower on average than the rated values by 23 and 18 percent, respectively; the cooling capacity and EER for dual-duct units were lower on average by 27 and 30 percent, respectively; and the cooling capacity and EER for spot coolers were lower on average by 9 and 22 percent, respectively. Although the results were generally consistent for the different product types, DOE notes that these data are based on a small sample of test units, and a larger sample may provide more representative trends for each configuration.

Due to lack of information available regarding typical spot cooler operating locations and conditions, DOE also tested the two spot coolers at reduced ambient conditions consistent with the "indoor" conditions for single-duct and dual-duct units, at 80 °F dry-bulb temperature and 67 °F wet-bulb temperature. The test results at both conditions and percent reductions in cooling capacity and EER at the indoor conditions are shown in Table II.3.

Table II.3 Baseline Spot Cooler Performance at Reduced Conditions

Test Unit	**Cooling Capacity** (Btu/h)			**EER** (Btu/Wh)		
	Baseline 95/75 °F	**Indoor 80/67 °F**	**Reduction** (%)	**Baseline 95/75 °F**	**Indoor 80/67 °F**	**Reduction** (%)
SC1	10,225.7	10,061.9	1.60	9.62	10.80	-12.28
SC2	10,774.7	9,557.5	11.30	6.72	6.68	0.64

DOE notes that the SC1 test unit tested within 3 percent of its rated cooling capacity and within 7 percent of its rated EER for both tests. The tested cooling capacity and EER for the SC2 test unit were within 12 percent of the tested values at the baseline test conditions, but still roughly 30 percent and 40 percent, respectively, below the rated values.

Issue 1. DOE seeks comment on the suitability of current industry standards for a potential DOE portable AC test procedure; specifically:

(1) ANSI/AHAM PAC-1-2009;

(2) ANSI/ASHRAE Standard 128-2001, which although not current is required for reporting in California;

(3) ANSI/ASHRAE Standard 128-2011; and

(4) CSA C370-13.

Issue 2. DOE seeks comment on whether the metrics for cooling capacity and EER as determined in these industry test procedures measure representative performance of the different portable AC product types (i.e., single-duct, dual-duct, and spot cooler).

Issue 3. DOE seeks comment on the approach used to estimate case and duct heat transfer to the conditioned space.

C. Investigative Testing

1. Calorimeter Approach

In response to the comments mentioned previously, suggesting a testing approach for portable ACs comparable to that for room ACs, and to further investigate heat transfer effects not currently captured in available portable AC test procedures, DOE conducted testing according to a room calorimeter approach adapted from ANSI/ASHRAE Standard 16-1983. DOE tested all of the single-duct and dual-duct units in its test sample by this approach, which used two test chambers, one maintained at the indoor conditions and the other adjusted to maintain the outdoor conditions as specified below. Rather than installing the test unit in the wall between the indoor and outdoor test rooms, as for a room AC, the portable AC under test was located within the indoor test room with the condenser duct(s) interfacing with the outdoor test room by means of the manufacturer-supplied or manufacturer-recommended mounting fixture. Unless otherwise noted, no sealing other than that recommended in manufacturer instructions was made at the duct connections or around the mounting fixture during the tests.

DOE used a pressure-equalizing device placed between the indoor chamber and outdoor chamber to maintain a static pressure differential of less than 0.005 inches of water between the chambers throughout testing, as specified in Section 4.2.3 of ANSI/ASHRAE Standard 16-1983. Consistent with the ambient conditions required by ANSI/AHAM PAC-1-2009, DOE maintained the indoor conditions at 80 °F dry-bulb and 67 °F wet-bulb (51-percent relative humidity) and the outdoor conditions at 95 °F dry-bulb and 75 °F wet-bulb (40-percent relative humidity). For some units, significant infiltration air flow from the outdoor chamber to the indoor chamber was required to maintain the required static pressure differential between the two test chambers. The calorimeter approach consisted of monitoring all energy consumed by the indoor chamber components to maintain the required ambient conditions while the portable AC under test operated continuously at its maximum fan speed. Following a period of no less than 1 hour with stabilized conditions under continuous portable AC operation, the data of a subsequent 1-hour stable period were analyzed to sum all heating and cooling contributions to the indoor chamber, including: chamber cooling, heat transferred through the chamber wall, air circulation fans, dehumidifiers, humidifiers, and scales. These instruments, conditioning equipment, and heat transfer components were all necessary to maintain the indoor chamber conditions throughout testing. The net indoor chamber cooling was recorded as the portable AC's cooling capacity. This approach encompasses all cooling and heating effects generated by the portable AC, including air infiltration effects that are not captured or estimated by the air enthalpy approach.

For the first set of calorimeter tests, the test units were installed with the manufacturer-provided ducting, duct attachment collar, and mounting fixture. This was done in order to include the impacts of heat transfer from the ducts and air leaks in the duct connections and mounting fixture, in addition to heat leakage through the case and infiltration air. Table II.4 shows the measured net cooling capacities and EERs for single-duct and dual-duct units tested according to the calorimeter approach when the infiltration air dry-bulb temperature was 95 °F. The results are compared to rated values.

Table II.4 Calorimeter Approach Results

Test Unit	Cooling Capacity (Btu/h)			EER (Btu/Wh)		
	Rated	**Calorimeter**	**Reduction (%)**	**Rated**	**Calorimeter**	**Reduction (%)**
SD1	8,000	-470.8	105.9	7.0	-0.54	107.7
SD2	9,500	-641.4	106.8	9.6	-0.70	107.3
SD3	12,000	3475.5	71.0	8.7	2.30	73.5
SD4	13,000	1841.4	85.8	9.7	1.34	86.2
DD1	9,500	3379.9	64.4	9.4	2.89	69.2
DD2	13,000	3442.4	73.5	8.9	2.60	70.8

DOE notes the significant difference between the rated cooling capacity and the results measured according to the calorimeter approach for both single-duct and dual-duct units. As expected, due to the larger effect of air infiltration, the difference was greater for single-duct units than for dual-duct ones. On average for single-duct units, cooling capacity was reduced by 92.4 percent and EER was reduced by 93.7 percent. For single-duct units SD1 and SD2, however, the net effects captured by the calorimeter approach resulted in negative cooling capacities; that is, there was overall heating in the indoor-side chamber. For dual-duct units, the average reductions in cooling capacity and EER were 69 percent and 70 percent, respectively.

As discussed previously, the calorimeter approach requires monitoring the energy consumption of all heating and cooling components required to maintain stable chamber conditions, while accounting for the heat transferred between the indoor and outdoor chambers. To quantify the combined impact of the heat transfer from leaks in the case and ducts and the enthalpy added from the infiltration air, DOE compared these calorimeter test results with the baseline results, as shown in Table II.5.

Table II.5 Comparison of Baseline Results and Calorimeter Approach Results

Test Unit	Cooling Capacity (Btu/h)			EER (Btu/Wh)		
	Baseline	**Calorimeter**	**Reduction (%)**	**Baseline**	**Calorimeter**	**Reduction (%)**
SD1	5,842.7	-470.8	108.1	6.84	-0.54	107.9
SD2	6,599.8	-641.4	109.7	7.41	-0.70	109.4
SD3	10,947.6	3475.5	68.3	7.47	2.30	69.2
SD4	9,505.6	1841.4	80.6	6.59	1.34	79.7
DD1	8,597.2	3379.9	60.7	7.41	2.89	60.9
DD2	7,211.2	3442.4	52.3	5.50	2.60	52.7

The percent reduction from baseline results to those measured using the calorimeter approach range from 52 percent to over 100 percent for both cooling capacity and EER.

Issue 4. DOE requests feedback on the applicability of the calorimeter approach for measuring the performance of portable ACs, and the associated testing burden.

Issue 5. DOE seeks comment on other possible testing methods or alternate approaches to measure representative portable AC performance.

DOE performed additional investigative testing to quantify the individual impacts on performance due to each of the factors discussed previously in this section of today's notice. The test setup, approach, and data collected for each of these investigations is presented below.

2. Duct Heat Loss and Leakage

To quantify the heat transfer to the conditioned space through the minimally insulated condenser duct(s) and from any leaks at the duct connections or mounting fixture, DOE repeated the calorimeter testing with insulation surrounding the condenser ducts to benchmark results without this heat transfer. DOE used insulation having a nominal R value of 6 (in units of hours-°F-square feet per Btu), with seams around the duct, adapter, and mounting bracket sealed with tape to minimize air leakage. To determine duct losses and air leakage effects, DOE compared results from these tests to the results from the initial calorimeter approach tests with no insulation, as shown in Table II.6.

Table II.6 Duct Loss and Air Leakage Effects

Test Unit	**Cooling Capacity** (Btu/h)			**EER** (Btu/Wh)		
	Uninsulated	**Insulated**	**Change***	**Uninsulated**	**Insulated**	**Change***
SD1	-470.8	-5.0	465.8	-0.54	-0.006	0.54
SD2	-641.4	-32.3	609.0	-0.70	-0.035	0.66
SD3	3475.5	4,091.8	616.3	2.30	2.723	0.42
SD4	1841.4	3,024.8	1,183.4	1.34	2.17	0.83
DD1	3379.9	4,682.0	1,302.1	2.89	3.94	1.04
DD2	3442.4	4,209.4	767.0	2.60	3.14	0.53

*Change in performance in the table above may not align with the performance values listed due to rounding considerations.

For all units in the test sample, insulating the ducts and sealing any potential leak locations improved the measured cooling capacity and EER results; however, the magnitude of the change varied from unit to unit.

Issue 6. DOE requests feedback on the potential performance impacts related to all components of duct heat losses, and whether and how a test procedure should account for them.

3. Infiltration Air

DOE investigated the impacts of air infiltration from outside the conditioned space in which the portable AC is located 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 may also draw a portion of their condenser air from the conditioned space.

a. Infiltration Air Flowrate

DOE estimated the infiltration air flow rate as equal to the condenser exhaust flow rate to the outdoor chamber minus any condenser intake flow rate from the outdoor chamber. DOE concluded, based on review of the test chamber configurations, that air leakage from the outdoor chamber to locations other than the indoor chamber was negligible. The net flow rate into the outdoor chamber was thus estimated to entirely be transferred into the indoor chamber through the pressure regulating apparatus during calorimeter testing. For accurate measurement of condenser air flow rates, the inlet and outlet air flow rates were measured during baseline testing using the duct instrumentation necessary for the air enthalpy method.

For a single-duct unit, the air balance equation results in the infiltration air flow rate being equal to the condenser exhaust air flow rate. For dual-duct units, the condenser exhaust duct flow rate may be higher than the inlet duct flow rate. This is due to some intake air being drawn from the indoor chamber via louvers or leakage through the case, duct connections, or between the evaporator and condenser sections. The estimated infiltration air flow rate for all single-duct and dual-duct units in DOE's test sample are presented in Table II.7.

Table II.7 Infiltration Air Flow Rate

Test Unit	Condenser Outlet Air Flow Rate (CFM)	Condenser Inlet Air Flow Rate (CFM)*	Net Infiltration Air Flow Rate (CFM)
SD1	268.0	N/A	268.0
SD2	262.6	N/A	262.6
SD3	285.5	N/A	285.5
SD4	254.3	N/A	254.3
DD1	271.9	170.8	101.1
DD2	214.8	128.1	86.8

*Condenser inlet air flow rate is only applicable for dual-duct units.

b. Effect of Infiltration Air Temperature

In its initial calorimeter test, DOE maintained the outdoor test chamber conditions at 95 °F dry-bulb temperature and 75 °F wet-bulb temperature. Infiltration air was provided by means of a pressure-regulated connection between the outdoor and indoor test chambers, thereby resulting in infiltration air at those temperatures. Such conditions would be representative of outdoor air being drawn directly into the conditioned space to replace any condenser inlet air from that same conditioned space. However, 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. Because varying infiltration air temperature would have a significant impact on cooling capacity and EER when using the calorimeter test method, and because DOE was unable to identify information on a representative infiltration air temperature and relative humidity, DOE performed calorimeter testing over a range of dry-bulb temperatures for the infiltration air that spanned 78 °F to 95 °F, all at the 40-percent relative humidity specified at the 95 °F condition. DOE selected conditions at 87 °F and 82 °F dry-bulb temperature based on outdoor test conditions among those specified for cooling mode tests in the ANSI/Air-Conditioning, Heating, and Refrigeration Institute (AHRI) Standard 210/240-2008, "Performance Rating of Unitary Air-Conditioning & Air-Source Heat Pump Equipment." The 78 °F test condition was selected based on the lowest temperature maintainable by the third-party test laboratory conducting testing.[11] Dual-duct units were not tested at this lowest-temperature test condition because DOE estimated that infiltration effects are not as significant for dual-duct units as they are for single-duct units and therefore did not warrant additional testing.

DOE tested two single-duct and two dual-duct units at the infiltration air conditions shown in Table II.8.

Table II.8 Infiltration Air Temperature Test Series

Infiltration Air Test Series	Infiltration Air Temperature (Dry/Wet Bulb)	Single-duct	Dual-duct
Test 1	95 °F / 75 °F	SD2, SD4	DD1, DD2
Test 2	87 °F / 69 °F	SD2, SD4	DD1, DD2
Test 3	82 °F / 65 °F	SD2, SD4	DD1, DD2
Test 4	78 °F / 62 °F	SD2, SD4	N/A

Infiltration air conditions at the lower end of the tested temperature range were similar to the ambient conditions being maintained in the indoor test chamber, and therefore would result in the smallest air infiltration effect on the measurement of cooling capacity and EER. Test results obtained under those conditions could potentially be similar to those obtained by the use of the current industry test procedures, after accounting for case and duct heat losses.

[11] The lowest maintainable temperature varied depending upon the test unit's capacity and air flow configuration. The 78 °F dry-bulb test condition was selected as the lowest maintainable condition for all units in the test sample.

Table II.9 shows the cooling capacity and EER results for single-duct and dual-duct units at the various infiltration temperatures.

Table II.9 Cooling Capacity and EER at Varying Infiltration Air Temperature

Test Unit	Cooling Capacity (Btu/h)				EER (Btu/Wh)			
	Test 1 (95/75 °F)	**Test 2** (87/69 °F)	**Test 3** (82/65 °F)	**Test 4** (78/62 °F)	**Test 1** (95/75 °F)	**Test 2** (87/69 °F)	**Test 3** (82/65 °F)	**Test 4** (78/62 °F)
SD2	-614.4	4,048.3	7,039.5	9,584.0	-0.70	4.51	7.88	10.66
SD4	1,841.4	7,808.2	10,468.9	12,247.4	1.34	5.47	7.51	9.00
DD1	3,379.9	6,268.8	7,801.0	N/A	2.89	5.53	7.07	N/A
DD2	3,442.4	6,396.1	8,147.3	N/A	2.60	4.99	6.40	N/A

These results confirm that single-duct unit performance as determined using the calorimeter approach is highly dependent on infiltration air temperature. The dual-duct units tested also showed significant variation of performance with infiltration air temperature because of the portion of condenser air that is drawn from the indoor chamber. Table II.10 lists the calorimeter test results at each infiltration air temperature as a percentage of the results obtained during baseline testing. At temperatures representative of many likely real-world infiltration air temperatures, it can be seen that the actual performance of portable ACs may be substantially lower than values obtained using the air enthalpy method would suggest.

Table II.10 Comparison of Baseline and Calorimeter Testing for Varying Infiltration Air

Test Unit	Calorimeter Cooling Capacity as a Percentage of Baseline Capacity (%)				Calorimeter EER as a Percentage of Baseline EER (%)			
	Test 1 (95/75 °F)	Test 2 (87/69 °F)	Test 3 (82/65 °F)	Test 4 (78/62 °F)	Test 1 (95/75 °F)	Test 2 (87/69 °F)	Test 3 (82/65 °F)	Test 4 (78/62 °F)
SD2	-9.7	61.30	106.7	145.2	-9.4	60.9	106.3	143.8
SD4	19.4	82.1	110.1	128.8	20.3	83.0	113.9	136.5
DD1	39.3	72.9	90.7	N/A	39.1	74.7	95.4	N/A
DD2	47.7	88.7	113.0	N/A	47.3	90.6	116.2	N/A

DOE notes that the test results with infiltration air at 82 °F dry-bulb temperature and 65 °F wet-bulb temperature were most similar to the baseline tests conducted according to the air enthalpy method.

DOE next quantified the total heat added to the room by the infiltration air at each reduced temperature test. DOE used the following equation to calculate the sensible heat contribution of the infiltration air as:

$$Q_s = \frac{\left(\frac{V \times \delta}{1 + \omega}\right) \times \left[(c_{p\_da} \times \Delta T) + \omega \times (c_{p_{wv}} \times \Delta T)\right]}{60}$$

Where:

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

V is the volumetric flow rate of infiltration air, in cubic feet per minute (cfm),

δ is the density of the air mixture, in pounds mass per cubic feet (lb_m/ft^3),

c_{p_da} is the specific heat of dry air, in Btu/lb_m-°F,

ω is the humidity ratio, in pounds mass of water vapor per pounds of dry air,

c_{p_wv} is the specific heat of water vapor, in Btu/lb_m-°F, 60 is the conversion factor from minutes to hours, and

ΔT is the difference between the infiltration air and indoor chamber dry-bulb temperatures, in °F.

DOE used the following equation for the latent heat contribution of the infiltration air:

$$Q_l = \frac{\left(\frac{V \times \delta}{1 + \omega}\right) \times (\omega \times h_{fg})}{60}$$

Where:

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

V is the volumetric flow rate of infiltration air, in cfm,

δ is the density of the air mixture, in lb_m/ft^3,

ω is the humidity ratio, in pounds mass of water vapor per pounds of dry air,

60 is the conversion factor from minutes to hours, and

h_{fg} is the latent heat of vaporization for water vapor, in Btu/lb_m.

The total heat contribution of the infiltration air is the sum of the sensible and latent heat. Table II.11 presents results for the total heat input from the infiltration air at various temperatures for each test unit, along with a comparison to the baseline cooling capacity.

Table II.11 Heat Input from Infiltration Air

Test Unit	Total Heat Transferred (Btu/h)			Heat Transferred as a Percentage of Baseline Cooling Capacity (%)		
	Test 1 (95/75 °F)	**Test 2** (87/69 °F)	**Test 3*** (82/65 °F)	**Test 1** (95/75 °F)	**Test 2** (87/69 °F)	**Test 3** (82/65 °F)
SD2	6,391.6	885.7	-2,294.0	96.8	13.4	-34.8
SD4	5,523.5	587.0	-2,263.9	58.1	6.2	-23.8
DD1	2,070.5	327.0	-679.9	24.1	3.8	-7.9
DD2	1,707.4	259.5	-576.8	23.7	3.6	-8.0

[*]DOE notes that at an infiltration air dry-bulb temperature slightly higher than the indoor 80 °F dry-bulb standard test condition, a net cooling effect is achieved because the latent heat of the infiltration air is less than the latent heat of the indoor test condition.

Table II.11 shows that infiltration air heat input is significant, almost 97 percent for one single-duct unit, when compared with the overall cooling capacity measured with current industry test procedures that do not address this heating effect. As expected, infiltration air at higher temperatures has a larger impact on performance than at lower temperatures, and is therefore a larger percentage of the baseline cooling capacity.

Issue 7. DOE seeks comment on whether infiltration air should be accounted for as part of a future DOE test procedure for portable ACs, should DOE determine to include portable ACs as a covered product, and if so, what test method would be appropriate to account for the infiltration air.

Issue 8. DOE seeks comment and information on whether the current industry standard outdoor air conditions of 95 °F dry-bulb temperature and 75 °F wet-bulb temperature are representative for real-world infiltration air, and if not, on what would be representative infiltration air temperatures.

Issue 9. DOE requests feedback on the effects of heat input from infiltration air and the performance differences that are observed between the results of testing according to the air enthalpy approach and the calorimeter approach.

4. Mixing Between the Condenser Inlet and Exhaust for Dual-Duct Portable Air Conditioners

The current industry test procedures specify the condenser inlet conditions for single-duct and dual-duct portable ACs, but do not address potential air mixing between the condenser inlet and exhaust air streams for the dual-duct configuration. Manufacturers typically provide a single mounting fixture for both the condenser inlet and exhaust ducts to minimize installation time and optimize the use of window space. However, this approach typically positions the condenser inlet and exhaust directly adjacent to one another. During operation when installed in the field, short-circuiting may occur between some of the condenser exhaust air (typically above 110 °F) and the outdoor ambient air (95 °F according to current industry test procedures). Elevated condenser inlet air temperature reduces the efficiency of the refrigeration system because it limits the ability of the condenser to reject heat from the conditioned space.

To investigate the effects of potential condenser inlet and exhaust mixing, DOE tested both dual-duct units according to two different approaches for maintaining the outdoor room conditions. The first approach was to maintain the overall outdoor chamber conditions at 95 °F dry-bulb temperature and 75 °F wet-bulb temperature as measured at the infiltration air inlet, allowing for mixing of condenser inlet and outlet air and thereby possibly increasing the condenser inlet temperature. Additionally, DOE notes that test chamber dimensions resulted in the duct fixture being located approximately four feet from the opposite wall of the outdoor chamber, which would likely be a worst-case configuration in terms of condenser air mixing for real-world installations.

The second approach was to monitor the condenser inlet dry-bulb and wet-bulb temperatures and adjust the chamber conditions to maintain the 95 °F/75 °F conditions at that location. Condenser exhaust and inlet air mixing would result in a lower temperature being maintained in the outdoor chamber.

Table II.12 shows the condenser inlet air and infiltration air dry-bulb temperatures when testing the two dual-duct units according to both test approaches. DOE tested each unit in two different configurations, once with manufacturer provided ducting and the second time with sealed and insulated ducts as described in section II.C.2 of today's notice.

Table II.12 Condenser Mixing Effects on Air Flow Temperatures

Test Units	**Infiltration Air at 95 °F**		**Condenser Inlet Air at 95 °F**	
	Condenser Inlet (°F)	**Infiltration Air** (°F)	**Condenser Inlet** (°F)	**Infiltration Air** (°F)
DD1 Uninsulated	95.5	95.1	94.8	94.3
DD1 Insulated	95.9	95.0	95.0	94.1
DD2 Uninsulated	95.1	95.1	95.0	95.0
DD2 Insulated	95.0	94.6	95.3	95.0

As shown in Table II.12 the difference between the condenser inlet air temperature and infiltration air temperature for both test approaches is at most 0.9 °F, regardless of duct heat losses. These results indicated that there was minimal mixing between the condenser exhaust and inlet air flows. Further confirming this observation were data for cooling capacity and EER shown in Table II.13, also showing that the difference between the two test approaches was minimal.

Table II.13 Condenser Mixing Effects on Performance

Test Units	Infiltration Air at 95 °F		Condenser Inlet Air at 95 °F		Percent Change	
	Cooling Capacity (Btu/h)	**EER** (Btu/Wh)	**Cooling Capacity** (Btu/h)	**EER** (Btu/Wh)	**Cooling Capacity** (%)	**EER** (%)
DD1 Uninsulated	3,379.9	2.89	3,447.7	2.96	2.01	2.18
DD1 Insulated	4,682.0	3.94	4,640.1	3.93	-0.90	-0.23
DD2 Uninsulated	3,442.4	2.60	3,413.8	2.58	-0.83	-1.02
DD2 Insulated	4,209.4	3.14	4,242.5	3.16	0.79	0.90

*Percent reduction in the table above may not align with the performance values listed due to rounding considerations.

Issue 10. DOE requests feedback regarding measures that should be considered in a portable AC test procedure to address any condenser exhaust air and inlet air mixing in dual-duct units.

D. Alternate Testing Approach

Based on the investigative testing, DOE considered whether another approach that utilizes the existing test procedures with numerical adjustments for infiltration air would accurately reflect portable AC performance. As described above in section II.C.3.b of this notice, DOE calculated the infiltration heat effects from the air flow rate and humidity ratio of the infiltration air. Subtracting the infiltration air heat from the cooling capacity as determined by the baseline test could be close enough to results obtained from the calorimeter method to provide a representative measure of portable AC performance. Table II.14 displays the cooling capacity as determined by combining the estimated infiltration air heat transfer with the baseline results, and the cooling capacity as determined by the calorimeter method.

Table II.14 Alternate Testing Approach Performance

Test Unit	Cooling Capacity (Btu/h)			EER (Btu/Wh)		
	Calorimeter	**Baseline - Infiltration Air**	**Increase (%)**	**Calorimeter**	**Baseline - Infiltration Air**	**Increase (%)**
SD1	-470.8	-878.4	-86.6	-0.54	-1.03	-90.0
SD2	-641.4	208.2	132.5	-0.70	0.23	133.5
SD3	3475.5	4,032.9	16.0	2.30	2.75	19.5
SD4	1841.4	3,982.1	116.3	1.34	2.76	106.1
DD1	3379.9	6,526.7	93.1	2.89	5.62	94.3
DD2	3442.4	5,503.8	59.9	2.60	4.20	61.5

The data in Table II.14 indicate that there is no consistent difference between the two test approaches. The increase in cooling capacity from the calorimeter approach to the alternate approach for single-duct units ranged between negative 87 percent and over 133 percent, while the two dual-duct units in the test sample had a smaller range in cooling capacity change, from 60 to 93 percent. A larger sample size may further show the trends for difference unit configurations.

Issue 11. DOE welcomes comment on this alternate testing approach, and in particular on the testing burden associated with it.

E. Additional Issues on which DOE Seeks Comment

Should DOE issue a final determination that portable ACs are a covered product, DOE may prescribe test procedures and energy conservation standards for portable ACs. As part of that effort, DOE may propose a new portable AC test procedure. In addition to the specific issues discussed above for which DOE is seeking comment, DOE welcomes comment on any aspect of this NODA and is also interested in receiving comments and views from interested parties on the following issues:

Issue 12. DOE welcomes general comments about the various investigative test approaches DOE conducted as discussed and presented above in this notice, including whether any of these approaches are currently utilized by manufacturers and test facilities. DOE also welcomes comment on any testing methodologies appropriate for consideration as an alternative to the industry accepted methodologies and those performed by DOE.

Issue 13. DOE requests data on the repeatability and reproducibility of such testing methods. DOE also welcomes additional data on the repeatability and reproducibility of testing results using the test methods presented in this notice.

The purpose of this NODA is to solicit feedback from industry, manufacturers, academia, consumer groups, efficiency advocates, government agencies, and other interested parties on issues related to a potential DOE portable AC test procedure. DOE is specifically interested in information and additional data on the current industry test procedures for portable ACs and alternate test approaches discussed in today's notice. Respondents are advised that DOE is under no obligation to acknowledge receipt of the information received or provide feedback to respondents with respect to any information submitted under this NODA. Responses to this NODA do not bind DOE to any further actions related to this topic.

III. Public Participation

DOE is interested in receiving comments on all aspects of the data and analysis presented in the NODA and supporting documentation that can be found at: http://www1.eere.energy.gov/buildings/appliance_standards/product.aspx/productid/79.

Submission of Comments

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

Submitting comments via www.regulations.gov. The www.regulations.gov webpage 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.

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Include contact information each time you submit comments, data, documents, and other information to DOE. If you submit via mail or hand delivery/courier, please provide all items on a CD, if feasible, in which case 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, that are written in English, and that are 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. Pursuant 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 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. 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).

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Energy Efficiency and Renewable Energy