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DEPARTMENT OF ENERGY

10 CFR Part 430

[Docket Number EERE-2014-BT-TP-0014]

RIN 1904–AD22

Energy Conservation Program: Test Procedures for Portable Air Conditioners; Correction

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

ACTION: Correcting amendments.

SUMMARY: The U.S. Department of Energy (DOE) published a final rule in the Federal Register on June 1, 2016, establishing test procedures for portable air conditioners. This correction addresses typographical errors in that final rule that were included in Title 10 of the Code of Federal Regulations (CFR) part 430, subpart B, appendix CC. Neither the errors nor the corrections in this document affect the substance of the rulemaking or any of the conclusions reached in support of the final rule.

DATES: This correction is effective **[INSERT DATE OF PUBLICATION IN THE FEDERAL REGISTER]**.

FOR FURTHER INFORMATION CONTACT:

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SUPPLEMENTARY INFORMATION: On June 1, 2016, DOE published a final rule (the "June 2016 final rule") to establish test procedures for portable air conditioners. 81 FR 35241. DOE has since found that the June 2016 final rule contained minor typographical errors in Title 10 of the Code of Federal Regulations (CFR) part 430, subpart B, appendix CC. This final rule correction revises appendix CC to subpart B of 10 CFR part 430, to correct these typographical errors. Specifically, in section 4.1.1, DOE is correcting the following errors: an incorrect subscript for the variable $T_{duct\_SD\_j}$ in the Q_{duct_SD} equation and missing subscripts "j" on the T_{duct} variables in the equations for Q_{duct_95} and Q_{duct_83}. In section 4.1.2, DOE is correcting the following errors: a missing equals sign and parenthesis; incorrect subscripts for the variable C_{p_da} and the infiltration air variables in the Q_{s_95} equation; incorrect subscripts in the infiltration air variables in the Q_{s_83} equation; missing equals signs in the Q_{l_95} and Q_{l_83} equations; and missing "Q" variables and incorrect subscripts for the Q_{l_95} and Q_{l_83} variables in the $Q_{infiltration\_95}$ and $Q_{infiltration\_83}$ equations.

DOE also found that the summation symbols in the two dual-duct Q_{duct} equations in section 4.1.1 were not properly represented in the Electronic Code of Federal Regulations (eCFR).

Neither the errors nor the corrections in this document affect the substance of the rulemaking or any of the conclusions reached in support of the final rule. Accordingly, DOE finds that there is good cause under 5 U.S.C. 553(b)(B) to not issue a separate notice to solicit public comment on the corrections contained in this final rule as doing so would be impractical, unnecessary, and contrary to the public interest. For the same reasons and pursuant to 5 U.S.C. 553(d), DOE finds good cause to waive the 30-day delay in effective date.

Procedural Issues and Regulatory Review

DOE has concluded that the determinations made pursuant to the various procedural requirements to the June 2016 final rule that originally codified DOE's test procedures for portable air conditioners remain unchanged for this final rule technical correction. 81 FR 35241. The amendments from that final rule became effective July 1, 2016. Id.

List of Subjects in 10 CFR Part 430

Administrative practice and procedure, Confidential business information, Energy conservation, Household appliances, Imports, Intergovernmental relations, Reporting and recordkeeping requirements, and Small businesses.

Issued in Washington, DC, on October 7, 2016.

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

For the reasons set forth in the preamble, DOE amends part 430 of title 10, Code of Federal Regulations by making the following correcting amendments:

PART 430 -- ENERGY CONSERVATION PROGRAM FOR CONSUMER PRODUCTS

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

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

2. Appendix CC to subpart B of part 430 is amended by revising sections 4.1.1 and 4.1.2 to read as follows:

APPENDIX CC TO SUBPART B OF PART 430– UNIFORM TEST METHOD FOR MEASURING THE ENERGY CONSUMPTION OF PORTABLE AIR CONDITIONERS

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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:

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

Where:

d_j = the outer diameter of duct "j", including any manufacturer-supplied insulation.

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

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

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

For dual-duct portable air conditioners:

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

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

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, in Btu/h, when tested according to the 95 °F dry-bulb and 83 °F dry-bulb outdoor test conditions in Table 1 of this appendix, respectively.

h = convection coefficient, 3 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, the condenser exhaust duct and the 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. Calculate the dry air mass flow rate of infiltration air according to the following equations:

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

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.

ρ_{co_SD}, ρ_{co_95}, and ρ_{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).

ρ_{ci_95}, and ρ_{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.

ω_{co_SD}, ω_{co_95}, and ω_{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}).

ω_{ci_95}, and ω_{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:

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

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

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.

ω_{ia_95} and ω_{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.

ω_{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:

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

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

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.

ω_{ia_95} and ω_{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.

ω_{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 heat 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.

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