DEPARTMENT OF ENERGY

Office of Energy Efficiency and Renewable Energy

10 CFR Part 430

[Docket No. EE–RM/TP–97–440]

RIN 1904–AA46

Energy Conservation Program for Consumer Products: Test Procedure for Residential Central Air Conditioners and Heat Pumps

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

ACTION: Final rule.

SUMMARY: The Department of Energy (DOE, or the Department) amends its test procedures for residential central air conditioners and heat pumps. This final rule adds new sections and revises several sections of the test procedure to bring it up-to-date by eliminating the need for several test procedure waivers and making it more complete. The Department also re-organized the test procedure to be more chronological in its progression. The revisions to the test procedure do not alter the minimum energy conservation standards currently in effect for central air conditioners and heat pumps.

DATES: This rule is effective April 10, 2006. The incorporation by reference of certain publications listed in this rule is approved by the Director of the Federal Register as of April 10, 2006.

ADDRESSES: You may review copies of all materials related to this rulemaking at the U.S. Department of Energy, Forrestal Building, Room 1J–018 (Resource Room of the Building Technologies Program), 1000 Independence Avenue, SW., Washington, DC, (202) 586–9127, between 9 a.m. and 4 p.m., Monday through Friday, except Federal holidays. Please call Ms. Brenda Edwards-Jones at the above telephone number for additional information regarding visiting the Resource Room. **Please note:** The Department's Freedom of Information Reading Room (formerly Room 1E–190 at the Forrestal Building) is no longer housing rulemaking materials.

FOR FURTHER INFORMATION CONTACT:

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SUPPLEMENTARY INFORMATION: The final rule incorporates, by reference, into Subpart B of Part 430 seven test-method standards published by the American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc. (ASHRAE), as follows:

- Standard 23–1993, ''Methods of Testing for Rating Positive Displacement Refrigerant Compressors and Condensing Units;''
- Standard 37–1988, ''Methods of Testing for Rating Unitary Air-Conditioning and Heat Pump Equipment;''
- Standard 41.1–1986 (Reaffirmed 2001), ''Standard Method for Temperature Measurement;''
- Standard 41.2–1987 (Reaffirmed 1992), ''Standard Methods for Laboratory Airflow Measurement;''
- Standard 41.6–1994 (Reaffirmed 2001), ''Standard Method for Measurement of Moist Air Properties;''
- Standard 41.9–2000, ''Calorimeter Test Methods for Mass Flow Measurements of Volatile Refrigerants;'' and
- Standard 116–1995, ''Methods of Testing for Rating for Seasonal Efficiency of Unitary Air Conditioners and Heat Pumps.''

The following joint test-method standard of ASHRAE and the Air Movement and Control Association International, Inc. (ASHRAE/AMCA) is incorporated by reference into subpart B of Part 430:

- Standard 51–1999/210–1999, ''Laboratory Methods of Testing Fans for Aerodynamic Performance Rating.''

The following test-and-rating standard of the Air-Conditioning and Refrigeration Institute (ARI) is incorporated by reference into Subpart B of Part 430:

- Standard 210/240–2003, ''Unitary Air-Conditioning and Air-Source Heat Pump Equipment.''

Copies of these standards are available for public review at the Department of Energy's Building Technologies Program Resource Room described above. Copies of the ASHRAE, ASHRAE/AMCA and ARI Standards are available from the American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc., 1971 Tullie Circle, NE., Atlanta, GA 30329, *http://www.ashrae.org*; the Air Movement and Control Association International, Inc., 30 West University Drive, Arlington Heights, IL 60004–1893, *http://www.amca.org*; and the Air-Conditioning and Refrigeration Institute, 4100 North Fairfax Drive, Suite 200, Arlington, VA 22203–1629, *http://www.ari.org.*

I. Introduction

A. Authority

Part B of Title III of the Energy Policy and Conservation Act (EPCA or Act) (42 U.S.C. 6291 *et seq.*), established the Energy Conservation Program for Consumer Products Other Than Automobiles (Program). The products currently subject to this Program (''covered products'') include central air conditioners and heat pumps, the subject of today's final rule.

Under the Act, the Program consists of three parts: Testing, labeling, and the Federal energy conservation standards. The Department, in consultation with the National Institute of Standards and Technology (NIST), is authorized to establish or amend test procedures as appropriate for each of the covered products. (42 U.S.C. 6293) The purpose of the test procedures is to measure energy efficiency, energy use, or estimated annual operating cost of a covered product during a representative, average use cycle or period of use. The test procedure must not be unduly burdensome to conduct. (42 U.S.C. 6293(b)(3))

If a test procedure is amended, DOE is required to determine to what extent, if any, the proposed new test procedure would alter the measured energy efficiency of any covered product as determined under the existing test procedure. (42 U.S.C. 6293(e)(1)) If DOE determines that an amended test procedure would alter the measured energy efficiency of a covered product, DOE is required to amend the applicable energy conservation standard with respect to such test procedure. In determining any such amended energy conservation standard, DOE is required to measure the energy efficiency or energy use of a representative sample of covered products that minimally comply with the existing standard. The average efficiency or energy use of these representative samples, tested using the amended test procedure, constitutes the amended standard. (42 U.S.C. 6293(e)(2)) The Department has determined that today's amended test procedure does not alter the measured efficiency or measured energy use of central air conditioners and heat pumps.

Beginning 180 days after a test procedure for a covered product is prescribed, no manufacturer, distributor, retailer, or private labeler may make representations with respect to the energy use, efficiency, or cost of energy consumed by such product, except as reflected in tests conducted according to the DOE procedure. (42 U.S.C. 6293(c)(2))

B. Background

On January 22, 2001, the Department published a Notice of Proposed Rulemaking (hereafter referred to as the January 22, 2001, proposed rule) that proposed a revised test procedure for central air conditioners and heat pumps. (66 FR 6768) As summarized in the January 22, 2001, proposed rule, the Department initiated several interactions, including a DOE workshop, phone conferences, and the release of multiple drafts for review and comment between DOE and stakeholders prior to preparing the revised test procedure.

Most of the existing test procedure dates back to its original publication in the **Federal Register** on December 27, 1979. (44 FR 76700) The Department modified the test procedure on March 14, 1988, to cover variable-speed air conditioners and heat pumps, to address testing of split-type non-ducted units, and to change the method used for crediting heat pumps that provide a demand defrost capability. (53 FR 8304)

The January 22, 2001, proposed rule specified dates for holding a public hearing and for submitting written comments. At the request of ARI, the Department changed these specified dates. (66 FR 15203, March 16, 2001) Prior to the public hearing and at the invitation of ARI, a NIST representative attended a meeting of the ARI Unitary Small Equipment Engineering Committee on February 27, 2001, at ARI headquarters. The public hearing was held on March 29, 2001, at DOE headquarters.[1] At the public hearing, the participants spent the majority of the time discussing the list of items from the proposed rulemaking for which the Department solicited stakeholder comment. One manufacturer, the Carrier Corporation, presented a prepared oral statement. On May 1, 2001, DOE and NIST personnel met with representatives of the Carrier Corporation at DOE headquarters.

During the comment period, stakeholders, DOE, and NIST held several phone and e-mail discussions about issues associated with the proposed test procedure (a revision of 10 CFR part 430, Subpart B, Appendix M) and about rating untested split-system combinations (a separate test procedure issue not covered in Appendix M, but in 10 CFR 430.24(m)). The issue of rating untested split-system combinations is not part of this rulemaking and will be the subject of a future rulemaking.

II. Discussion of Comments

A. General Discussion

Nine different stakeholders submitted a total of fourteen comments on the January 22, 2001, proposed rule. Concurrent with this rulemaking, the Department also conducted a rulemaking to issue new energy conservation standards for central air conditioners and heat pumps. Both rulemakings covered, among other consumer products, small-duct, high-velocity (SDHV) systems. In the standards rulemaking (66 FR 7197), DOE stated that concerns for SDHV systems had been addressed by modifying the test procedure for SDHV products. This test procedure modification would have given SDHV systems a higher tested value of the Seasonal Energy Efficiency Ratio (SEER). (DOE later rejected this test procedure modification for reasons discussed in section II.A.2 of this preamble). As a result, the Department considered comments received on October 18, 2001, from SDHV manufacturers SpacePak and Unico, Inc. (Unico) as part of the energy conservation standards rulemaking in today's final rule on the test procedure.

[1] The Department held a public workshop on issues that would not be considered for the current revision to the test procedure (*i.e.*, alternative rating method for untested combinations, promoting devices that compensate for installation problems, metrification of the DOE test procedure) on the day immediately following the close of the public hearing.

(SpacePak, No. 21, Unico, No. 22) [2] The Department also considered during this rulemaking amended comments from ARI, dated October 30, 2001, that addressed the SDHV issue. (ARI, No. 20) A discussion of the comments and the actions taken in response to them follows.

1. Adopting References Updated Since Public Hearing

The January 22, 2001, proposed rule referenced seven ASHRAE standards, as well as ASHRAE Standard 51–99/AMCA Standard 210–99, and ARI standard 210/240. Since the publication of the proposed rule, however, two of these standards have been reaffirmed and two have been revised. The two reaffirmed standards are ASHRAE Standard 41.1–1986 (Reaffirmed 2001) and ASHRAE Standard 41.6–1994 (Reaffirmed 2001). When a standard is reaffirmed within ASHRAE, no substantive changes are permitted to the document. In the ASHRAE Project Committee Manual of Procedures, substantive change is defined as

a change that involves an important (has value, weight or consequences), fundamental (is the foundation, without which it would collapse), or essential (belongs to the very nature of a thing) part or changes the meaning of the material or that directly and materially affects the use of the standard. Following are example changes that may be found substantive when examined in context;

- ''shall'' to ''should'' or ''should'' to ''shall;'
- addition, deletion or revision of mandatory requirements, regardless of the number of changes;
- or addition of mandatory compliance with referenced standards.

Thus, today's final rule references ASHRAE Standards 41.1–1986 (Reaffirmed 2001) and 41.6–1994 (Reaffirmed 2001), whereas the January 22, 2001, proposed rule had referenced ASHRAE Standards 41.1–1986 (Reaffirmed 1991) and 41.6–1994. These changes have no effect on the test procedure itself nor on the reported energy efficiency ratings of the tested equipment.

The two revised standards are ASHRAE Standard 41.9–2000 and ARI Standard 210/240–2003. A revision of ASHRAE Standard 41.9, ''Calorimeter Test Methods for Mass Flow Measurements of Volatile Refrigerants,'' was published in 2000. The previous version, Standard 41.9–1988, was referenced in the proposed rulemaking. This particular standard is only referenced in section 3.11.2 of the test procedure. Section 3.11.2 pertains to one of three allowed secondary test methods, the Compressor Calibration Method. These secondary test methods do not affect the reported performance ratings. Instead, these secondary test methods are used to provide a check of the primary method, *i.e.*, the Indoor Air Enthalpy Method. Specifically, the cooling or heating capacity determined using the approved primary method and the user selected secondary test method must agree within six percent to constitute a valid test set-up. The revised version of ASHRAE Standard 41.9 is referenced in today's test procedure both because it does not affect the reported ratings and because it provides the most current methods for making refrigerant calorimeter measurements.

The other revised standard is ARI Standard 210/240–2003. The main impetus behind the 2003 revision of ARI Standard 210/240 was a desire to narrow the scope of the equipment covered by the standard. Whereas the 1994 version of Standard 210/240 covered equipment up to 135,000 Btu/h, the 2003 version is limited to equipment having rated capacities less than 65,000 Btu/h. With regard to the DOE test procedure, the January 22, 2001, proposed rule referenced four sections within ARI Standard 210/240–1994. In the 2003 version of the standard, no substantive changes were made to these four sections. The numbering/lettering of the sections, however, did change slightly. For example, section 5.1.3.5 in the 1994 document became section 6.1.3.5 in the 2003 document. Today's test procedure maintains the approach taken in the proposed rule of only referencing the four particular sections of 210/240. Because of this consistency, the DOE test procedure is unaffected by referencing ARI Standard 210/240–2003 rather than Standard 210/240–1994. The reported energy efficiency ratings of the tested equipment are unaffected as well.

2. Small-Duct, High-Velocity (SDHV) Systems

As discussed in the January 22, 2001, proposed rule, Unico, a manufacturer of SDHV systems, argued for creating a separate SDHV product class that was subject to a lower future energy conservation standard than the level established for conventional units. (66 FR 6768) However, in the energy standards rulemaking, a majority of industry members opposed the separate-product-class option. DOE did not include a separate SDHV class in the January 22, 2001, proposed rule. Instead, DOE proposed testing SDHV systems as coil-only units. Testing as coil-only units would give SDHV units an immediate SEER and Heating Seasonal Performance Factor (HSPF) boost, as long as the default fan power was less than the actual blower wattage. The SEER and HSPF boost eliminated the need for a separate product class. Both Unico and ARI at first endorsed this approach. (Unico, No. 10; ARI, No. 19 at p. 3) But SpacePak, Trane, and ultimately ARI, disagreed with the coil-only testing approach. (SpacePak, No. 15; Trane, No. 12 at p. 1, ARI, No. 20) These comments noted that SDHV systems would be tested in a manner that would never occur in real applications and, as a result, give energy efficiency and cost-of-operation results that are not representative of the unit's true energy performance. Furthermore, SDHV manufacturers would have no incentive to use high-efficiency blowers if systems were tested without the indoor blower. Finally, there is no technical basis for setting the default fan-power level. For these reasons, DOE has determined that its proposal to test SDHV systems as coil-only units is unacceptable. As a result, today's final rule does not amend the test procedures to test SDHV systems as coil-only units.

DOE considered another alternative for SDHV systems which it also ultimately rejected. This alternative was to make no changes at all. In other words, test SDHV systems as they are currently tested and require them to meet the same future energy conservation standards as conventional units. The Department rejected this option because it risked the continued existence of SDHV systems. The Department explained its position at the public hearing on March 29, 2001: The Department cannot set standards in a way that removes from the market a product which offers special utility. (Public Hearing Tr., p. 44)

Because today's final rule does not amend the test procedures for SDHV units, DOE recognizes, as it did in the January 22, 2001, energy standards final rule, that SDHV units will have difficulty in meeting the 13 SEER standard. In the May 23, 2002, final rule on central air conditioner and heat pump standards, DOE further discussed how the special characteristics of SDHV systems would make it unlikely such systems could even meet the 12 SEER/7.4 HSPF standard established for space constrained products. (67 FR 36396) However, because of the ruling by the

[2] These comments were received in the course of the standards rulemaking, Docket Number EE–RM–98–440, but are relevant to this test procedure rulemaking. SpacePak's comments are item 267 in that docket; Unico's comments are item 251.

U.S. Court of Appeals for the Second Circuit in January, 2004, 355 F.3d 179 (2d Cir. 2004), that bars DOE from adopting a standard of less than 13 SEER for SDHV systems, the 13 SEER standard applies to SDHV systems, despite DOE's later conclusion that it is unlikely such systems can meet that standard or even the lower 12 SEER standard for space constrained systems. (69 FR 50997) Nonetheless, the inability of SDHV systems to meet the applicable energy efficiency standards is not a new problem created by the amendments to the test procedure in today's rulemaking. Instead, these products were unable to meet the standard under the old test procedures. As a result, DOE need not amend the applicable test procedure or standard to mitigate this noncompliance. DOE has advised the two manufacturers of these systems of the procedure available to affected persons under section 504 of the Department of Energy Organization Act (42 U.S.C. 7194), which allows them to request relief from hardship or inequity caused by a regulation issued under EPCA.

3. Non-Defrost Heat Pumps

The January 22, 2001, proposed rule included steps for calculating the HSPF of a non-defrost heat pump. This proposal addressed the test procedure waiver granted to Enviromaster International (EMI). In 1992, the Department granted EMI a waiver for its line of non-defrost, multi-split heat pumps. Under the waiver, the Department did not require EMI to report an HSPF and instead required EMI to include in its printed materials for the product the following sentence, ''No HSPF value has been measured since the heat pump cannot be operated at temperatures below 35°F.'' EMI finally applied to the Department's Office of Hearing and Appeals (OHA) on January 23, 2003, for exception relief from the HSPF efficiency standards. OHA granted the exception relief on April 1, 2003. Thus, EMI has never calculated HSPF because of its waiver, and will not do so in the future because of OHA exception relief.

Since there are no manufacturers of products on the market which would actually use the proposed procedure for calculating the HSPF of a non-defrost heat pump, the Department has removed from the test procedure all references to non-defrost heat pumps and the special caveats for calculating an HSPF for such units.

4. Two-Capacity, Northern Heat Pumps

The January 22, 2001, proposed rule applied to a two-capacity heat pump configured to use only low capacity when cooling, while using both low and high capacities when heating. (66 FR 6768) The proposed test procedure identified such units as ''two-capacity heat pumps that lock out high capacity when cooling.'' At the March 29, 2001, public hearing, York expressed concern regarding the use of the term ''lockout.'' (Public Hearing Tr., p. 54) York felt the term was too restrictive, since it could be interpreted to mean that the lockout feature must be hard-wired, whereas DOE intended the meaning to include factory or field-selectable lockout.

At the March 29, 2001, public hearing, ARI commented that such units would typically have two different indoor coil identifiers and, as a result, two different sets of ratings. (Public Hearing Tr., p. 53) The ARI comment was supported by many of the other participants at the public hearing. ARI and York submitted written comments that supported the consensus reached at the public hearing. (ARI, No. 19 at p. 2; York, No. 9 at p. 2) The Department chose to adopt the public comment consensus and now defines these types of systems as ''two-capacity, northern heat pumps.'' The Department included a requirement in the definition of ''two-capacity, northern heat pump'' that the manufacturer must clearly state that the feature is factory or field-selectable and that manufacturers must publish two sets of ratings. Finally, the definition indicates that the lockout feature is to remain enabled for all tests. The northern heat pump is allowed to operate at high capacity during its defrost cycle, an issue that arose at the public hearing. (Public Hearing Tr., p. 55)

5. Heat Pumps Having a Heat Comfort Controller

The January 22, 2001, proposed rule included an algorithm for calculating the HSPF for most single-speed heat pumps having a heat comfort controller. (66 FR 6768) At the March 29, 2001, public hearing, Trane commented that the wording in the test procedure on the calculation of the energy consumed for resistive heating by a heat comfort controller needed clarification. Trane suggested that one use the higher of: (1) The resistive heating based on meeting the heat comfort controller's temperature setting; or (2) the resistive heating based on meeting the building load deficit (when operating below the balance point). (Public Hearing Tr., p. 30) Later, Trane submitted written comments that the algorithm, as interpreted, would overstate the HSPF at heat-comfort-controller set points beginning around 90°F and get progressively worse as the set point was reduced. (Trane, No. 12)

Battelle offered three general recommendations. The first recommendation was to emphasize that comfort controllers operate both above and below the normal balance point temperature. The second recommendation was to account for the fact that conventional heat pumps and, to a lesser extent, heat pumps with comfort controllers, will cycle below the system balance point. The third recommendation was that DOE perform a parametric calculation to determine ''HSPF deficits'' due to the operation of a comfort controller. (Battelle, No. 11) The end product could potentially be a table listing the reduction in HSPF that results from operating the comfort controller at different temperature settings.

The American Gas Association (AGA) comments paralleled those from Battelle. Both AGA and Battelle recommended that the definition of HSPF specify that for heat pumps with heat comfort controllers, HSPF accounts for resistive heating contributed when operating either above or below the balance point as a result of maintaining a minimum supply temperature. Both also recommended that the equation for the heating load factor in section 4.2.1 be changed to the following:

$$X(T_j) = \frac{BL(T_j)}{\dot{Q}_h(T_j) + n(RH_b)}$$

where,

$X(T_j)$ = the heating mode load factor for temperature bin j, dimensionless

$BL(T_j)$ = the building space conditioning load corresponding to an outdoor temperature of T_j

$\dot{Q}_h(T_j)$ = the space heating capacity of the heat pump when operating at outdoor temperature T_j, Btu/h

RH_b = the size of each resistance heat bank

n = the number of banks needed to exceed the building load at each bin temperature.

Finally, in a slight variation from Battelle, AGA recommended that ''DOE provide direction in the test procedure for evaluating performance of heat pumps retrofitted with heat comfort controllers in the field, including a parametric table of HSPF by DOE region for various delivered air temperatures.'' (AGA, No. 18, Battelle, No. 11)

Given the general support for covering those heat pumps having heat comfort controllers, today's test procedure covers all heat pumps having heat comfort controllers, except when a heat comfort controller is used with a heat pump having a variable-speed compressor. Test procedure section 4.2.5.4 is reserved for a variable-speed heat pump having a heat comfort controller.

The algorithm for calculating the HSPF of a heat pump having a heat comfort controller is covered in sections 4.2.5.1 to 4.2.5.3 of today's final rule. The algorithm captures the fact that the balance point temperature (*i.e.*, where the compressor first runs continuously) for a heat pump with a heat comfort controller will be less than, or equal to, the balance point temperature of that same heat pump without the heat comfort controller. In response to Trane's comments (Public Hearing Tr., p. 30; Trane, No. 12), today's test procedure includes editorial additions that alert the user to evaluate Equation 4.2.1–2 for all temperature bins. The test procedure then accounts for the resistive heating needed to satisfy the minimum air delivery temperature of the heat comfort controller and the (additional) resistive heating needed to give an overall heating capacity that matches the building load.[3]

In considering AGA and Battelle's recommended definition change, the key point is to emphasize the downward shift in the balance point and the associated lower contribution by the heat pump. The Department doesn't believe that a single sentence referenced to heat comfort controllers within the HSPF definition, even when modified as recommended, is sufficient. Therefore, the definition of ''Heat pumps having a heat comfort controller,'' emphasizes the downward shift in the balance point and the associated lower contribution by the heat pump.

The Department is amending the definition of HSPF by moving the following language from the definition text in the proposed rule to the main text of the test procedure, specifically, to the end of Section 4.2, ''Heating Seasonal Performance Factor (HSPF) Calculations.''

> For all heat pumps, HSPF accounts for the heating delivered and the energy consumed by auxiliary resistive elements when operating below the balance point. This condition occurs when the building load exceeds the space heating capacity of the heat pump condenser. For heat pumps with heat comfort controllers (see Definition 1.26), in addition, HSPF also accounts for resistive heating contributed when operating above the balance point as a result of maintaining a minimum supply temperature.

This moved text includes the one sentence from the HSPF definition in the proposed rule that specifically addressed heat comfort controllers. This sentence is the same one that both AGA and Battelle recommended changing. Coupled with the additional paragraph in Section 4.2.5, ''Heat pumps having a heat comfort controller,'' the Department believes the revisions more accurately convey the operating changes caused by adding a heat comfort controller.

The Department did not adopt AGA and Battelle's recommendation for changing the calculation of the heating-mode-load factor. (AGA, No. 18, Battelle, No. 11) The Department agrees with AGA and Battelle that resistive heating initiated as the result of a second stage call of the indoor thermostat can, under the right conditions, cause a conventional heat pump to cycle below its balance point. Even though a conventional heat pump terminates resistive heating once the second stage setpoint is met, the concentrated burst of resistive heating coupled with the capacity of the continuously operating heat pump may cause the first stage of the thermostat to be met shortly after the second stage is met. An overshoot occurs and the heat pump cycles off. The overshoot is more likely to occur near the balance point where only a small amount of resistive heating is needed.

The existing test procedure makes the implicit assumption that an overshoot never occurs. AGA and Battelle's proposed change assumes that an overshoot always occurs. The frequency of this overshoot is unknown. Until data become available showing that overshoot occurs more often than the case where the heat pump runs continuously and the resistive elements cycle on and off at the second stage, the Department will leave the heating-load-factor calculation unchanged. The AGA and Battelle recommendation would be more appropriate if resistive heating, once initiated as the result of a second-stage call, stayed on until the first stage setpoint was met. The Department is not aware of conventional heat pumps that use this strategy, so it did not change the calculation of the heating-mode-load factor.

Heat pumps with heat comfort controllers operate differently from conventional heat pumps following a second-stage-thermostat call for resistive heating. When the second-stage setpoint is satisfied, heat comfort controllers reduce the resistive heating rather than cycling it off. In this manner, the heat comfort controller attempts to modulate the resistive heating so that additional second-stage calls are reduced while also avoiding satisfying the first-stage setpoint. The goal is for the heat pump to operate continuously below the balance point while having the resistive heating regulated to provide a more uniform delivery temperature than that provided by a conventional heat pump. The heat comfort controller's operation when responding to a second-stage-thermostat call is believed to provide a more comfortable environment for the homeowner, while not causing an energy penalty. The one field study cited by both AGA and Battelle[4] supports this assertion. Therefore, as was decided for conventional heat pumps, the Department did not adopt the AGA and Battelle recommended heating-load-factor equation within the section 4.2.5 calculations that only apply to heat pumps having a heat comfort controller.

Finally, with regard to the Battelle and AGA recommendations that the test procedure contain information on the impact of heat comfort controllers for different temperature setpoints and/or quantify the impact from an after-market retro-fit installation of a heat comfort controller, the Department agrees that such information is probably warranted but judges it inappropriate for inclusion in the test procedure. The scope of the test procedure is to test and rate new, factory-supplied equipment. Addressing the impact of after-market products on the performance of covered products is not within the purview of EPCA. However, as pointed out at the March 29, 2001, pubic hearing, the test procedure may provide a framework for building code officials' consideration when deciding how to handle the after-market sale of heat comfort controllers. (Public Hearing Tr., p. 32)

[3] When calculating the HSPF for a conventional heat pump, the section 4.2 variable $\dot{E}_h(T_j)$ and $\dot{Q}_h(T_j)$ represent the electrical power and heating capacity provided exclusively by the heat pump, while the variable $RH(T_j)$ applies exclusively to any resistive heating contribution. When calculating the HSPF of a heat pump having a heat comfort controller, by comparison, the variables $\dot{E}_h(T_j)$ and $\dot{Q}_h(T_j)$ represent the electrical power and heating capacity provided by the heat pump and any supplemental resistive heating needed to provide the comfort-controller-set-point air delivery temperature. The variable $RH(T_j)$, in this case, reflects any additional resistive heating if the combined capacity of heat pump and the resistive heating associated with achieving the set-point air delivery temperature is nonetheless insufficient to meet the building load. Electrical resistive heating for a heat pump having a heat comfort controller is thus allocated among two variables ($\dot{E}_h(T_j)$ and $RH(T_j)$) rather than one ($RH(T_j)$). This redefining allows the calculation procedure to capture the reduced heat pump contribution, the shift to a lower balance point, and the negative impact on HSPF.

[4] ''Improving Occupant Comfort Without an Energy Penalty in Homes Heated by Electric Heat Pumps,'' Yuill, G.K., and Musser, A., ASHRAE Paper 4162, ASHRAE Transactions 1998 V. 104, Pt. 1.

B. Definitions

In addition to the amendments to the definitions discussed above in section II.A.1 of this preamble, today's final rule modifies definitions and references as described below.

An editorial correction was made to the citation for ASHRAE Standard 51–99/AMCA Standard 210–99. In the proposed rule the words ''AMCA Standard'' were wrongly omitted.

The definitions of ''heating seasonal performance factor (HSPF),'' and ''seasonal energy efficiency ratio (SEER)'' have been modified to move some text to later sections of the test procedure. The moved text provided complementary information that was better placed in the main text of the test procedure rather than in a definition. Sentences from the definition of HSPF were moved to Section 4.2, ''Heating Seasonal Performance Factor (HSPF) Calculations.'' Similarly, one sentence from the definition of SEER became the first sentence in Section 4.1, ''Seasonal Energy Efficiency Ratio (SEER) Calculations.''

C. Testing Conditions

1. Section 2.2.4 Wet-Bulb Temperature Requirements for Air Entering the Indoor and Outdoor Coils

The January 22, 2001, proposed rule included a requirement that applied to wet-coil cooling tests of single-packaged units where all or part of the indoor section is located in the outdoor test room. The requirement was that the average dew point temperature of the air entering the outdoor coil must be within ±3.0°F of the average dew point temperature of the air entering the indoor coil. This requirement was added to address concerns about equipment leakage affecting capacity measurements. The water vapor content of the outdoor air could affect the repeatability of the measurements. Similarly, leakage could present a problem when using the Outdoor Air Enthalpy test method for testing a single-packaged heat pump where all or part of its outdoor section is located in the indoor test room.

In comments made at the March 29, 2001, public hearing and in written comments received thereafter, York and ARI agreed with the proposed requirements. (Public Hearing Tr., p. 79; York, No. 9 at p. 4; ARI, No. 19 at p. 2) The Department has adopted the proposed test requirement in today's final rule without alteration.

2. Section 2.2.5 Additional Refrigerant Charging Requirements

Existing testing procedures require that the unit be installed in accordance with the manufacturer's installation instructions. The ARI, as part of its certification program, occasionally makes decisions on what is and is not within the spirit of the requirement. Thus, a policy has evolved wherein ARI certification testing allows procedures such as break-in times for compressors and washing the oil residue from the coils prior to testing. ARI does not allow disconnecting an electrical component, such as a crankcase heater. For the most part, the Department chose to defer to ARI to maintain consistency in the test set-ups. However, the Department proposed additional limits on the specific issue of the refrigerant-charging procedure. In the section 2.2.5 of the January 22, 2001, proposed rule, the Department proposed two additional requirements. First, the Department sought to avoid a gray area of defining when an independent test laboratory should consult with the manufacturer on how to charge a unit. The proposed section included the sentence: ''For third party testing, for example, do not consult the manufacturer about how to charge the unit.'' This requirement was thought to place extra responsibility on the manufacturer to publish accurate and clear charging instructions.

The second requirement was to promote the ideal of testing the unit in a manner that is similar to its actual installation in the field. The Department proposed amendments to section 2.2.5 to include the following sentence: ''Where the manufacturer's installation instructions contain two sets of refrigerant charging criteria, one for field installations and one for lab testing, use the field installation criteria.''

At the March 29, 2001, public hearing, ARI, ITS, and ACEEE spoke in favor of allowing the independent test laboratory to contact the manufacturer if it had any charging questions. (Public Hearing Tr., pages 101 to 112) This discussion noted the value of feedback in assisting the manufacturer to identify mistakes or incompleteness in its published instructions. Such feedback, if acted upon by the manufacturer, could benefit the eventual field installer. At the public hearing, attendees also came to the realization that the attempt to prevent special lab-only charging criteria could likely be circumvented by having a single criteria that listed wide ranges for such charging parameters as the targeted superheat or subcooling level(s).

The Department considered deleting the proposed section 2.2.5. However, today's final rule contains a revised version of the January 22, 2001, proposed rule language. (66 FR 6792) In the proposed rule, for third-party testing, the test laboratory was not to consult with the manufacturer about how to charge a unit. Based on the public hearing comments discussed above, today's final rule has modified this requirement. The test laboratory may consult with the manufacturer about the refrigerant-charging procedure and make changes that do not contradict the published installation instructions. The manufacturer may specify an alternative charging criteria to the third-party laboratory if the manufacturer then revises the published installation instructions accordingly. DOE decided to keep the section in an effort to convey the side benefit of the allowed feedback mechanism and to emphasize that the goal is a lab set-up as consistent as possible with a field installation.

D. Testing Procedures

1. Section 3.1.4 Airflow Through the Indoor Coil: Systems Having a Variable-Speed, Constant Airflow Blower

The January 22, 2001, proposed rule included additions to the test procedure for systems having a variable-speed, constant airflow (often called constant CFM (cubic foot per minute)) blower. These additions included:

(1) Controlling the exhaust fan of the airflow measuring apparatus to obtain a specified external static pressure. DOE received no comments on this addition.

(2) Specifying an additional test and algorithm to correct the fan power in cases where the specified external static pressure cannot be achieved during testing due to blower instabilities. ITS and York commented in favor of this addition. (Public Hearing Tr., ITS, p. 72–73, York, p. 73)

(3) Making use of the fan laws if a unit must be tested at an air volume rate other than the (cooling or heating) Certified Air Volume Rate. DOE received no comments on this addition.

(4) Allowing cyclic tests to be conducted with or without the indoor fan enabled and using a step profile for the air volume rate during cyclic tests. DOE received no comments on this addition.

(5) Imposing an 8-percent tolerance for the difference between the lab-measured and manufacturer-Certified Air Volume Rates.

At the March 29, 2001, public hearing, ARI, Trane, and York spoke in favor of making a change to eliminate the eight percent tolerance. (Public Hearing Tr., ARI, p. 69, Trane, p. 70, and York, p. 70) ARI and York submitted written comments to the same effect. (ARI, No. 19 at p. 2; York, No. 9 at p. 2) Opposition to the eight percent tolerance was based on the industry's not wanting another certified parameter. ARI recommended that DOE limit its focus to rated capacity and seasonal performance, SEER and HSPF, and not include parameters that affect those values. (ARI, No. 19 at p. 2)

DOE proposed the tolerance to provide manufacturers with assurance that any third-party testing would employ a representative air volume rate. However, these blowers have a level of variability which may occasionally exceed the proposed eight percent tolerance. The eight-percent tolerance could cause several unnecessary stoppages in testing where the impact on rated capacity and seasonal performance would be negligible. Given the foreseeable unfavorable trade-off from imposing the tolerance, the Department has eliminated the eight-percent tolerance in today's final rule.

2. Sections 3.1.4.2, 3.1.4.5, 3.3, 3.5.1, 3.7, and 3.9.1. Testing a Two-Capacity Compressor System: Coil-Only Units Tested at Low Capacity and Differences in High/Low Cycling

The proposed test procedure sections 3.1.4.2 and 3.1.4.5 specified that the air volume rate used when testing two-capacity, coil-only units at low capacity (*i.e.*, at the Minimum Air Volume Rate) is the higher of:

(1) The rate specified by the manufacturer, or

(2) 75 percent of the air volume rate used for the high capacity tests.

At both the public hearing and in its written comments, York opposed the proposed 75-percent limit. (Public Hearing Tr., pp. 81–86; York, No. 9 at p. 3) York argued that the limit was ''arbitrarily derived, is unnecessary, and restrictive towards applying existing and future technologies in motor speed controls. * * *'' (York, No. 9 at p. 3) Conversely, at both the public hearing and in their written comments, both Copeland Corporation and ARI supported the defining of a lower limit. Their written comments specifically endorsed assigning the limit at 75 percent. (Public Hearing Tr., pp 86–90; Copeland Corporation, No. 13 at p. 2; ARI, No. 19 at p. 2)

This 75-percent value is based on the assumption that the two-capacity coil-only unit would most often be used with an existing multi-tap furnace blower. The low range offered from typical multi-tap motors can vary considerably. Nonetheless, the limited data collected by NIST and by industry supports the proposed 75-percent value, and DOE has included it in today's final rule.

The proposed test procedure sections 3.3, 3.5.1, 3.7 and 3.9.1 did not differentiate between the default fan power values for high capacity and low capacity. The value of 365 watts per 1000 standard cubic feet per minute (SCFM) was used in all cases. Only York commented on this issue, and York's comment supported the proposed test procedure. (Public Hearing Tr., p. 94, York, No. 9 at p. 3) York commented that the proposed low capacity default causes a conservative prediction of fan power, with a resulting error too insignificant to warrant a change. (York, No. 9 at p. 3) Today's final rule maintains the changes on this subject incorporated into the proposed test procedure.

The final two-capacity, compressor-system issue was whether there is a significant performance difference between compressors (systems) that can switch between low and high stages over a very short time interval versus those having to turn off for a short period and take longer overall to make the transition. (This issue is included because DOE received comments about it. It does not appear in the proposed rule, nor in today's final rule.) Copeland Corporation noted that it has experience manufacturing both types of compressors and that it has ''observed that shutting a system down for greater than one minute has nearly the same cyclic loss impact as a typical on/off C_D penalty, since the evaporator warms up almost completely.'' Copeland encouraged the Department to study the issue further and stated that an appropriate action may be to conduct a test program at Intertek Testing Services (ITS). (Copeland Corporation, No. 13 at p. 1) York, on the other hand, expressed its opinion that the difference in technology was not significant enough to warrant a change in the test procedure. (York, No. 9 at p. 3) The Department has been unable to identify test procedure changes that could capture a performance difference, assuming that its overall impact significantly alters the SEER and HSPF ratings. The Department would have to make assumptions about the frequency of high/low transitions as a function of the magnitudes of the low and high stage capacities relative to each temperature bin building load. Also, data are needed to determine whether the cooling and heating mode on/off degradation coefficients could act as substitutes for the high/low transition degradation or whether a separate optional test and/or separate transition default values are warranted. In general, the Department is willing to consider future changes to the test procedure but asks that interested industry members take the lead in quantifying the impact on SEER and HSPF before making specific recommendations on how to alter the test procedure calculations.

III. Summary of Other Additions and Changes to the DOE Residential Central Air Conditioner and Heat Pump Test Procedure

Today's final rule contains numerous changes that were proposed in the January 22, 2001, proposed rule, for which the Department received no adverse comments.

A. Update and Add References for ASHRAE and ARI Standards

The current test procedure references ASHRAE Standard 37–78 and ASHRAE Standard 41.1 (no year), ARI Standard 210–79, ARI Standard 240–77, and ARI Standard 320–76. Today's final rule also includes references to ARI Standard 210/240–03, ASHRAE Standard 23–93, ASHRAE Standard 37–88, ASHRAE Standard 41.1–86 (RA 01), ASHRAE Standard 41.2–87 (RA 92), ASHRAE Standard 41.6–94 (RA 01), ASHRAE Standard 41.9–00, ASHRAE Standard 51–99/AMCA Standard 210–99, and ASHRAE Standard 116–95. The additional commercial standards are necessary to more completely inform manufacturers and testers about the multiple test options, especially for the secondary test method, and to address as many of the small details of lab testing as possible. The additional commercial standards were all included in the January 22, 2001, proposed rule. (66 FR 6768) Some of the commercial standards have been updated since the publication of the proposed rule as discussed in section II.A.1 of this preamble.

B. Air Volume Rates

The current test procedure references ARI Standard 240–77. Now, rather than referencing ARI Standard 210/240–03, which replaced ARI Standard 240–77, the Department has added its own sections to the test procedure. The main reason for no longer referencing ARI Standard 210/240 is that it does not cover variable-speed and constant CFM blowers. In addition, ARI Standard 210/240 does not directly address two-capacity and variable-speed systems. The Department believes it is preferable to have the overall issue of air volume rates covered in one place rather than in two.

The test procedure set forth in this final rule no longer references ASHRAE Standard 37–78 (or ASHRAE Standard 37–88, its replacement) for the equation used to calculate the air volume rate of standard air, because the referenced equation is incorrect. The factor ''1 $+W_n$'' is missing from the denominator of the pertinent equation in both versions of ASHRAE Standard 37. Today's test procedure includes what DOE believes to be the correct version of the equation.

Today's test procedure also adopts the approach used in the ISO Standard 5151 of conducting each test at zero external static pressure when testing a non-ducted unit.

All of these ''air volume rate'' substantive changes were originally published in the proposed rulemaking (66 FR 6778) and are included in today's final rule.

C. Cyclic Testing

The Department is today adopting standard industry practice and the method described in ASHRAE Standard 116. Sections 4.1.1.2, 4.1.2, 4.2.2.2, and 5.1 of the current (1988) test procedure require measurement of the air volume rate during cyclic tests and use of this measurement in determining the total cooling (heating) delivered. Standard laboratory practice, by comparison, is to achieve and maintain the same velocity pressure or nozzle static pressure drop that was obtained during the comparable steady-state test. The total cooling (heating) delivered during a cyclic test, in addition, is calculated using the air volume rate measured during the comparable steady-state test. Changes to adopt this industry practice and become consistent with ASHRAE Standard 116 were introduced in the proposed rulemaking and are included in today's final rule in section 3.1.

When testing split-type non-ducted (ductless) systems, section 4.1.1.5 of the current test procedure provides, ''The integration time for capacity and power shall be from compressor cut-on time to indoor fan cutoff time.'' The indoor fan is operated for three minutes prior to compressor cut-on and for three minutes after compressor cutoff during the final OFF/ON interval. In sections 3.5 and 3.5.2, today's final rule adopts industry practice and integrates power from compressor OFF to compressor OFF and subtracts the electrical energy associated with operating the indoor fan during the initial three-minute fan-only period. Space cooling capacity is integrated from compressor ON to indoor fan OFF. As with the current test procedure, fan energy for the three minutes after compressor cutoff is added to the integrated cooling capacity.

The current test procedure does not contain specific information regarding the air dampers: where to install them, how well they should seal, and how quickly they should respond. Appendix D of ARI Standard 210/240–03 contains much of this information. Today's final rule incorporates the required information in sections 2.5.4.1 and 2.5.7 rather than make specific references to each pertinent section of Appendix D of the ARI Standard.

For dry coil tests, today's test procedure final rule adopts, in section 3.4, the language in ARI Standard 210/240–03 Appendix D with regard to the requirements that the drain pan be plugged and completely dry.

Today's final rule clarifies in section 2.8 that the requirement of making electrical energy measurements using an instrument having an accuracy of ±0.5 percent of reading applies during both the ON and OFF intervals of cyclic tests.

Today's final rule deletes the current section 4.1.3.1, ''The indoor and outdoor average dry-bulb temperature for the cyclic dry coil test D shall both be within 1.0 °F of the indoor and outdoor average dry bulb temperature for the steady-state dry coil test C, respectively.'' This requirement is automatically met given the 0.5 °F test condition tolerance associated with each test. (Today's amended test procedure is substantially re-organized; the section 4.1.3.1 in today's final rule has no relation to the deleted section 4.1.3.1.)

For units having a variable-speed indoor fan, the manufacturer will have the option of conducting the cyclic tests with the indoor fan either enabled or disabled, the latter being the default option if an attempt at testing with the fan enabled is unsuccessful. See section 3.5 of today's final rule. Specifically, if the test is performed with the indoor fan operating, and the fan automatically reverses, shuts down, or operates at an uncharacteristically high external static pressure, then the test must be repeated using a pull-thru method, with the fan disabled.

Although a unit having a variable-speed indoor fan may be designed to ramp its fan speed when cycling on and/or off, a step response in air volume rate is nonetheless required during cyclic tests. See section 3.5 of today's final rule. The work associated with moving the additional air during the ramp periods is performed by the exhaust fan of the air flow measuring apparatus. The step response begins at the initiation of ramp up and ends at the termination of ramp down. The rationale for imposing the step change is mainly due to the difficulty in obtaining the ramp response and then making an accurate measurement of the space conditioning delivered. Systems having indoor fans that ramp are expected to have low cyclic degradation coefficients (C_D) regardless of whether the ramp feature is used, thus the absolute improvement in C_D is expected to be minor.

D. Fanless (Coil-Only) Units

Section 4.1 of the current test procedure calls for corrections to capacity and power based on air flow measured in cubic feet per minute (CFM). Section 4.2 of the current test procedure calls for corrections to capacity and power based on air flow measured in cubic feet per minute under standard conditions (SCFM). To avoid confusion, the test procedure should base corrections on either CFM or SCFM, but not both. ITS, which tests for both the industry and ARI, uses SCFM in all cases. Therefore, in consideration of the above, today's test procedure adopts, in sections 3.3, 3.5.1, and 3.7, the practice of specifying all corrections in terms of SCFM.

The test procedure also adopts in section 2.2 the requirement in ARI Standard 210/240–03, Appendix D, that an enclosure be constructed using one-inch ductboard for testing a coil-only unit that does not employ an enclosure.

E. Frost Accumulation Test

Today's final rule adopts the convention in ASHRAE Standard 116–95 and ARI 210/240–03 of specifying the outdoor wet bulb temperature (33 °F) in place of the presently specified dew point temperature (30 °F). Sections 3.6.1, 3.6.2, 3.6.3, and 3.6.4.

F. Test Tolerance Tables

The current test procedure contains tables covering all tests except steady-state cooling-mode tests, for which Table III in ASHRAE Standard 37–78 is referenced. Since the test procedure includes all other tables, the Department chose to add the needed parts of Table III (Table 7 of this document).

The test condition tolerance for external resistance to air flow now applies only when testing non-ducted units. (See Table 7). Also, DOE has added in Table 7 a test condition tolerance for electrical supply voltage (previously, only a test operating tolerance was specified). The existing test procedure lacked a clarification that the test condition tolerance for the indoor inlet wet bulb temperature in Table III of ASHRAE Standard 37–78 does not apply for dry coil tests. Therefore, today's final rule includes a footnote to Table 7 that makes this clarification. In a similar attempt to clarify when particular tolerances apply, today's final rule also includes a footnote to tables stating that the test tolerances given for the outdoor outlet dry and wet bulb temperatures only apply when using the Outdoor Air Enthalpy Method to provide the secondary capacity measurement.

For the Frost Accumulation Test, DOE modified slightly the intervals considered to be heating versus defrosting. Specifically, in the current test procedure in section 4.2.3.3, the first five minutes after a defrost termination was included in the defrost interval. In today's final rule, the time interval has been increased to ten minutes in section 3.7. This is a better approximation of the time needed for temperatures to reach equilibrium after defrost termination. Also, in making the test condition conversion of 30 °F dew point to 33 °F wet bulb, the test operating tolerance and test condition tolerance convert to wet bulb temperature tolerances of 0.6 °F and 0.3 °F, respectively. This 0.6 °F test operating tolerance on outdoor wet bulb temperature is more stringent than the value allowed for the steady-state tests. The 0.3 °F test condition tolerance is the same as required for steady-state tests. Because these tolerances should be less stringent that those required of a steady-state test, the test procedure adopts in Table 15 the values given in ASHRAE Standard 37: 1.5 °F and 0.5 °F.

G. Pretest Intervals

1. Wet Coil Tests

The following change makes the test conditions more specific than they are in the current test procedure:

Current: ''The test room reconditioning apparatus and the equipment under test shall be operated until equilibrium conditions are attained.'' (Section 4.1.1.1)

Today's final rule: ''For the pretest interval, operate the test room reconditioning apparatus and the unit to be tested until maintaining equilibrium conditions for at least 30 minutes at the specified section 3.2 test conditions.'' (Section 3.3)

2. Dry Coil Steady-State Test

The following change also makes the test conditions more specific than they are in the current test procedure. The industry realized the merits of this improved wording several years ago. The added text is taken from a prescriptive methodology that appears within an appendix of ARI Standard 210/240–2003.

Current: ''The test room reconditioning apparatus and the equipment under test shall be operated until equilibrium conditions are attained, but not for less than one hour before data for test C are recorded.'' (Section 4.1.1.2)

Today's final rule: Same as proposed for section 3.3 wet coil tests with the additional requirement to ''operate the unit at least one hour after achieving dry coil conditions.'' (Section 3.4)

3. Dry Coil Cyclic Test

The following change makes the test conditions more specific than they are in the current test procedure. The existing language is weaker because the phrase ''until steadily repeating ambient conditions are again achieved'' is comparatively subjective.

Current: ''[T]est unit shall be manually cycled 'off' and 'on'* * * until steadily repeating ambient conditions are again achieved in both the indoor and outdoor test chambers, but for not less than two complete 'off/on' cycles.'' (Section 4.1.1.2)

Today's final rule: ''After completing a minimum of two complete compressor OFF/ON cycles, determine the overall cooling delivered and total electrical energy consumption during any subsequent data collection interval where the test tolerances given in Table 8 are satisfied.'' (Section 3.5)

4. Maximum and High Temperature Heating Mode Tests

The requirement for the test apparatus and the test unit to operate for at least one hour was dropped based on industry comments that it had no bearing on the outcome of the testing—the key is to have steady operation at the specified test conditions for an interval (30 minutes) prior to starting the test.

Current: ''The test room apparatus and test units must be operated for at least one hour with at least one-half hour at equilibrium and at the specified test conditions prior to starting the test.'' (Section 4.2.1.1)

Today's final rule: ''For the pretest interval, operate the test room reconditioning apparatus and the heat pump until equilibrium conditions are maintained for at least 30 minutes at the specified section 3.6 test conditions.'' (Section 3.7)

5. Heating Mode Cyclic Test

The new language is more definitive and easier for a test laboratory to understand and implement. The existing language is weaker because the phrase ''until steadily repeating ambient conditions are again achieved'' is comparatively subjective.

Current: ''[A]nd be cycled 'on' and 'off' as specified in 3.2.1.2 until steadily repeating ambient conditions are achieved for both the indoor and outdoor test chambers, but for not less than two complete 'off'/'on' cycles.'' (Section 4.2.1.2)

Today's final rule: ''After completing a minimum of two complete compressor OFF/ON cycles, determine the overall cooling delivered and total electrical energy consumption during any subsequent data collection interval where the test tolerances given in Table 8 are satisfied.'' (Section 3.5)

6. Frost Accumulation Test

The new wording is clearer about the goal of getting the test room to achieve and maintain the specified test conditions. It clarifies the 30-minute requirement as a period that starts after the test conditions are first achieved.

Current: ''The test room reconditioning equipment and the unit under test shall be operated for at least one-half hour prior to the start of a 'preliminary' test period.'' (Section 4.2.1.3)

Today's final rule: ''Operate the test room reconditioning apparatus and the heat pump for at least 30 minutes at the specified section 3.6 test conditions before starting the 'preliminary' test period.'' (Section 3.9)

7. Low Temperature Test

The existing language can be interpreted to mean that one only needs to achieve the test conditions immediately prior to starting the test as opposed to maintaining the test conditions for at least 30 minutes prior to starting the test. The new wording is clearer. The new wording also clarifies the sequential process for having the heat pump conduct a defrost.

Current: ''The test room reconditioning equipment shall first be operated in a steady-state manner for at least one-half hour at equilibrium and at the specified test conditions. The unit shall then undergo a defrost, either automatic or manually induced.'' (Section 4.2.1.4)

Today's final rule: ''For the pretest interval, operate the test room reconditioning apparatus and the heat pump until equilibrium conditions are maintained for at least 30 minutes at the specified section 3.6 test conditions.'' (Section 3.7) ''After satisfying the section 3.7 requirements for the pretest interval, but before beginning to collect data to determine $\dot{Q}_h^k(17)$ and $\dot{E}_h^k(17)$, conduct a defrost cycle. This defrost cycle may be manually or automatically initiated.'' (Section 3.10)

H. Multi-Capacity Systems

1. Two-Capacity Heat Pumps That Lock Out Low Capacity at Higher Outdoor Temperatures

The current test procedure in section 2.2.2 covers two-capacity units that operate exclusively at high capacity when the building load exceeds the unit's low capacity. The Department is unaware of any two-capacity units that implement such a control strategy, and so DOE is not including coverage of them in today's final rule. However, the Department is adding coverage in section 3.2.3 to address units that lock out low capacity operation at low (heating) or high (cooling) outdoor temperatures. Today's test procedure uses the C_D determined based on cycling at low capacity (or the appropriate default) in all cases.

2. Systems Having a Single-Speed Compressor and a Variable-Speed Indoor Fan Where Fan Speed or Air Volume Rate Depends on Outdoor Temperature

Today's final rule requires two additional steady-state tests for the cooling mode (see section 3.2.2.1 and Table 4) and two additional steady-state tests for the heating mode (see section 3.6.2 and Table 10). The additional tests, at a different air volume rate, are required to calculate the effect of the variable-speed indoor fan. An additional frost accumulation test is optional.

I. Triple-Split Systems

The current DOE test procedure, in sections 4.1 and 4.2.1, refers to ASHRAE Standard 37–78 on the issue of laboratory set up procedures. Section 3.1.3 of ASHRAE Standard 37–78 requires using the calorimeter air-enthalpy method arrangement when testing units where the compressor is in the indoor section and separately ventilated. For this arrangement, an enclosure must be built around the equipment within the indoor chamber. The present requirement is burdensome, and DOE has learned no one uses it when testing triple-splits. Furthermore, the heat loss from the indoor compressor section should be reflected, if at all, in an adjusted output capacity and not by a raised entering-air temperature because the lost heat is transferred to the surrounding ambient, not dissipated within the return air duct. The surrounding ambient, in this case, may or may not be part of the conditioned space.

The amount of heat dissipated to the ambient by the indoor compressor section of such units is usually minimized as a result of the insulated enclosure of the third section (mainly in an effort to reduce the operating noise). Based on the limited information currently available, DOE believes that the amount of heat lost from the indoor compressor section is on the order of two percent or less of the unit's space conditioning capacity.

Today's final rule reflects the assumption that the heat loss from the indoor compressor section contributes nothing to the unit's overall delivered capacity if the compressor section is located in an unconditioned space. If the compressor section is located in the conditioned space, it still contributes only a negligible amount. Today's final rule specifies that triple-split systems are not to be tested using the calorimeter air-enthalpy method arrangement (see note in section 2.6 of the test procedure in today's final rule). The final rule does not provide for any adjustment to capacity, or any algorithm or method for assigning/determining the heat loss from the indoor compressor section. If triple-split systems become more popular and if information becomes available indicating the heat loss from the indoor compressor section exceeds two percent of the air-side capacity, then DOE will revisit the option of having a capacity adjustment.

J. Time-Adaptive Defrost Control Systems

When conducting a frost accumulation test on a heat pump having a time-adaptive defrost control system, repeatable frosting and defrosting intervals typically require (if obtainable at all) an excessive number of cycles. The tester must manually initiate defrosts during the ''preliminary'' test and the ''official'' test. Under today's final rule, the manufacturer must provide information as to how long the unit would optimally frost before it initiates a defrost, and on how to initiate a defrost cycle at the appropriate elapsed time. See section 2.2.1. However, the controls of the unit will still control the duration of the defrost cycle after its initiation.

K. Test Unit Installation

For the most part, equipment installation requirements under today's final rule will continue according to the manufacturer's field installation instructions. However, today's final rule adopts the lab and field practice of insulating the low pressure line(s) of a split system. See section 2.2.

L. Test Apparatus and Measurement/Sampling Frequency

1. Inlet Plenum for Blower Coils

The current DOE test procedure does not require an inlet plenum when testing blower coil units. (Lab ceiling height on vertical installation is a limitation.) In today's final rule, the manufacturer has the option to test with or without an inlet plenum installed when testing a ducted unit having an indoor fan. Space limitations within the test room may dictate that the manufacturer choose the latter option. (Section 2.4.2)

2. Manifolded Static Pressure Taps

The current (1988) test procedure does not discuss methods of manifolding static pressure taps. Today's final rule allows three configurations: The triple-T configuration; the complete ring, four-to-one manifold configuration; and the broken-ring, four-to-one manifold configuration. (Section 2.4.1) A 1976 study found the triple-T configuration to be the preferred method for manifolding static pressure taps.[5] The broken-ring, four-to-one manifold configuration is generally considered to be the least accurate of the three methods.

3. Temperature Measurement Intervals

Today's final rule (Definition 1.15) specifies dry-bulb temperature measurements at the intervals specified in ASHRAE Standard 41.1–86 (RA01). The tester must measure wet bulb temperature, dew point temperature, or relative humidity at the minimum sampling interval specified in the definition of the term ''Continuously recorded.''

4. Temperature Measurement Accuracies

Today's final rule (sections 2.5.5, 2.5.6, 2.11) incorporates the accuracy and precision requirements of temperature measurement from ASHRAE Standard 41.1–86 (RA 01).

5. Grid of Individual Temperature Sensors Within the Indoor-Side Outlet Plenum

Today's final rule adopts the requirements in ARI Standard 210/240–03, Appendix D, that a temperature spread of 1.5 °F or less be obtained, and that a minimum of 9 sensors compose the outlet temperature grid. (Section 2.5.5.) The January 22, 2001, proposed rule contained these DOE recommendations (66 FR 6796):

[5] ''The Design of Piezometer Rings'' by K. A. Blake, *Journal of Fluid Mechanics,* Vol. 78, 1976, part 2, pp. 415–428.

DOE recommends using 16 temperature sensors within each temperature grid. DOE recommends installing redundant inlet and outlet dry bulb temperature sensors and particularly a thermopile. If using thermocouples, DOE recommends the following:

(1) Use 24 gauge wire;

(2) Remove approximately 1 inch of insulation from each lead when preparing to make a junction; and

(3) Use no more than two bonded turns per junction.

The Department believes these recommendations to be sound, but today's final rule omits them because recommendations are not appropriate in a regulatory test procedure.

6. Duct Loss Correction

Today's final rule includes a correction for the heat transfer between the test room and an outlet duct sandwiched between the coil and the outlet temperature grid. (Section 3.11) This correction is already an industry practice.

7. Water Vapor Measurements Using a Dew-Point Hygrometer, a Relative Humidity Meter, or Any Other Alternative Instrument

Today's final rule explicitly permits alternatives to using wet bulb temperature sensors. To ease instrumentation selection, the rule specifies required instrument accuracies for dew point hygrometers and relative humidity meters. (Section 2.5.6)

8. Voltmeter Accuracy

The required accuracy of voltage measurements has been changed from ±2 percent to ±1 percent. (Section 2.7)

9. Electrical Power Measurement

Adjustable-speed-driven motors, as used in a variable-speed compressor, distort the input current and, to a lesser degree, voltage waveforms. For reasons that were outlined in the preamble of the January 22, 2001, proposed rule (66 FR 6779), today's final rule (Section 2.8) eschews the use of induction type meters for measuring such non-sinusoidal power. The January 22, 2001, proposed rule included a recommendation to use a meter capable of sampling up to the 50th harmonic. Sampling up to the 50th harmonic reduces the chances for measurement errors, but the extra expense for such a piece of equipment may not be justified, so today's final rule does not require its use.

M. Different Compressor Speeds and Indoor Fan Capacities Between Cooling and Heating

The existing test procedure covers variable-speed systems that operate at higher speeds when heating than when cooling. Today's final rule extrapolates this allowance to coverage of two-capacity, northern heat pumps (see section 4.2). Today's rule covers any case where the heat pump uses different fan speeds or air volume rates for cooling versus when heating. (Section 3.1.4.4.2)

N. Secondary Test Requirements

When using the Outdoor Air Enthalpy test method, the tester must conduct a preliminary test to compensate, if necessary, for any performance impact resulting from the outdoor air-side test apparatus. (Section 3.11.1) In the existing test procedure, a preliminary test is conducted prior to all steady-state tests (*i.e.*, those tests that require a secondary measurement of capacity). Today's final rule relaxes this requirement. Section 3.11.1 indicates that the number of preliminary tests can be reduced in most cases to one (for air conditioners or heating-only heat pumps) or two (for heat pumps): One for the first cooling mode steady-state test and one for the first heating mode steady-state test. The above ''test apparatus and measurement/sampling frequency'' substantive changes were introduced in the proposed rulemaking and are maintained in today's final rule. (Section 3.11.1)

O. HSPF Calculations

Today's final rule does not include the final paragraph of sections 5.2.1 and 5.2.2 of the current test procedure. The paragraph in question reads ''Once the maximum and minimum HSPF and operating cost values have been obtained for each region, the HSPF and operating cost shall be determined for each standardized design heating requirement (see section 6.2.6) between the maximum and minimum design heating requirements by means of interpolation.'' The number of required HSPF calculations is covered in 10 CFR Subpart B, 430.23(m)(3)(ii). In today's final rule, this section of the CFR is noted in the Definition (1.27) for HSPF. Because of the relative ease of automating the calculation process, and the nonlinearity of the HSPF-versus-design-heating-requirement relationship, today's final rule makes no reference to obtaining HSPF or operating cost via interpolation.

P. Effect of Test Procedure Revisions on SEER and HSPF

The most significant revisions to the test procedure in this final rule adopt industry practices and clear up gray areas with more precise instructions. No existing requirements are changed, but new requirements are added. Based on its development, review and analysis of the test procedure revisions being published today, the Department believes that these test procedure revisions will have no material impact on the measured values of SEER and HSPF, and thus it has satisfied the requirement of 42 U.S.C. 6293(e)(1): ''In the case of any amended test procedure which is prescribed pursuant to this section, the Secretary shall determine, in the rulemaking carried out with respect to prescribing such procedure, to what extent, if any, the proposed test procedure would alter the measured energy efficiency, measured energy use, or measured water use of any covered product as determined under the existing test procedure.'' In the January 22, 2001, proposed rule, the Department asked for comments on this issue (66 FR 6782), and received no comments contending that these revisions would impact measured values of SEER and HSPF.

IV. Procedural Requirements

A. Review Under Executive Order 12866

It has been determined that today's regulatory action is not a ''significant regulatory action'' under Executive Order 12866, ''Regulatory Planning and Review,'' 58 FR 51735 (October 4, 1993). Accordingly, this action was not subject to review under the Executive Order by the Office of Information and Regulatory Affairs (OIRA) of the Office of Management and Budget (OMB) .

B. Review Under the Regulatory Flexibility Act

The Regulatory Flexibility Act (5 U.S.C. 601 et seq.) requires preparation of an initial regulatory flexibility analysis 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 rulemaking process. (68 FR 7990) DOE has made its procedures and policies available on the Office of General Counsel's Web site: *http://www.gc.doe.gov.*

DOE reviewed today's rule under the provisions of the Regulatory Flexibility Act and the procedures and policies published on February 19, 2003. DOE certified in the January 22, 2001, proposed rule that the proposed rule would not impose a significant economic impact on a substantial number of small entities. (66 FR 6780) DOE received no comments on this issue, and after considering the potential small entity impact of this final rule, DOE affirms the certification that this rule will not have a significant economic impact on a substantial number of small entities.

C. Review Under the Paperwork Reduction Act

This rulemaking imposes no new information or record keeping requirements under the Paperwork Reduction Act. (44 U.S.C. 3501 et seq.)

D. Review Under the National Environmental Policy Act

DOE has determined that this rule falls into a class of actions that are categorically excluded from review under the National Environmental Policy Act of 1969 (42 U.S.C. 4321 et seq.) and the Department's implementing regulations at 10 CFR part 1021. This rule amends an existing rule without changing its environmental effect, and, therefore, is covered by the Categorical Exclusion in paragraph A5 to subpart D, 10 CFR part 1021. Accordingly, neither an environmental assessment nor an environmental impact statement is required.

E. Review Under Executive Order 13132

Executive Order 13132, ''Federalism,'' 64 FR 43255 (August 4, 1999) imposes certain requirements on agencies formulating and implementing policies or regulations that preempt State law or that have federalism implications. The Executive Order requires agencies to examine the constitutional and statutory authority supporting any action that would limit the policymaking discretion of the States and to carefully assess the necessity for such actions. The Executive Order also requires agencies to have an accountable process to ensure meaningful and timely input by State and local officials in the development of regulatory policies that have federalism implications. On March 14, 2000, DOE published a statement of policy describing the intergovernmental consultation process it will follow in the development of such regulations. (65 FR 13735) DOE has examined today's rule and has determined that it does not preempt State law and does not have a substantial direct effect on the States, on the relationship between the national government and the States, or on the distribution of power and responsibilities among the various levels of government. No further action is required by Executive Order 13132.

F. Review Under Executive Order 12988

With respect to the review of existing regulations and the promulgation of new regulations, section 3(a) of Executive Order 12988, ''Civil Justice Reform'' (61 FR 4729, February 7, 1996), imposes on Federal agencies the general duty to adhere to the following requirements: (1) Eliminate drafting errors and ambiguity; (2) write regulations to minimize litigation; and (3) provide a clear legal standard for affected conduct rather than a general standard and promote simplification and burden reduction. Section 3(b) of Executive Order 12988 specifically requires that Executive agencies make every reasonable effort to ensure that the regulation: (1) Clearly specifies the preemptive effect, if any; (2) clearly specifies any effect on existing Federal law or regulation; (3) provides a clear legal standard for affected conduct while promoting simplification and burden reduction; (4) specifies the retroactive effect, if any; (5) adequately defines key terms; and (6) addresses other important issues affecting clarity and general draftsmanship under any guidelines issued by the Attorney General. Section 3(c) of Executive Order 12988 requires Executive agencies to review regulations in light of applicable standards in section 3(a) and section 3(b) to determine whether they are met or it is unreasonable to meet one or more of them. DOE has completed the required review and determined that, to the extent permitted by law, this rule meets the relevant standards of Executive Order 12988.

G. Review Under the Unfunded Mandates Reform Act of 1995

Title II of the Unfunded Mandates Reform Act of 1995 (Pub. L. 104–4) (UMRA) requires each Federal agency to assess the effects of Federal regulatory actions on State, local, and Tribal governments and the private sector. For a proposed regulatory action that may result in the expenditure by State, local and Tribal governments, in the aggregate, or by the private sector of $100 million or more (adjusted annually for inflation), section 202 of UMRA requires a Federal agency to publish estimates of the resulting costs, benefits, and other effects on the national economy. (2 U.S.C. 1532(a), (b)) UMRA also requires a Federal agency to develop an effective process to permit timely input by elected officers of State, local, and Tribal governments on a proposed ''significant intergovernmental mandate,'' and requires an agency plan for giving notice and opportunity for timely input to potentially affected small governments before establishing any requirements that might significantly or uniquely affect small governments. On March 18, 1997, DOE published a statement of policy on its process for intergovernmental consultation under UMRA (62 FR 12820) (also available at *http://www.gc.doe.gov*). The rule published today contains neither an intergovernmental mandate, nor a mandate that may result in an expenditure of $100 million or more in any year, so these requirements do not apply.

H. Review Under the Treasury and General Government Appropriations Act of 1999

Section 654 of the Treasury and General Government Appropriations Act, 1999 (Pub. L. 105–277) requires Federal agencies to issue a Family Policymaking Assessment for any rule that may affect family well-being. This rule would not have any impact on the autonomy or integrity of the family as an institution. Accordingly, DOE has concluded that it is not necessary to prepare a Family Policymaking Assessment.

I. Review Under Executive Order 12630

DOE has determined, under Executive Order 12630, ''Governmental Actions and Interference with Constitutionally Protected Property Rights,'' 53 FR 8859 (March 18, 1988) that this regulation would not result in any takings which might require compensation under the Fifth Amendment to the United States Constitution.

J. Review Under the Treasury and General Government Appropriations Act of 2001

The Treasury and General Government Appropriations Act, 2001 (44 U.S.C. 3516, note) provides for agencies to review most disseminations of information to the public under guidelines established by each agency pursuant to general guidelines issued by OMB. OMB's guidelines were published at 67 FR 8452 (February 22, 2002), and DOE's guidelines were published at 67 FR 62446 (October 7, 2002). DOE has reviewed today's notice under the OMB and DOE guidelines and has concluded that it is consistent with applicable policies in those guidelines.

K. Review Under Executive Order 13211

Executive Order 13211, ''Actions Concerning Regulations That Significantly Affect Energy Supply, Distribution, or Use,'' 66 FR 28355 (May 22, 2001) requires Federal agencies to prepare and submit to OIRA, a Statement of Energy Effects for any proposed significant energy action. A ''significant energy action'' is defined as any action by an agency that promulgated or is expected to lead to promulgation of a final rule, and that: (1) Is a significant regulatory action under Executive Order 12866, or any successor order; and (2) is likely to have a significant adverse effect on the supply, distribution, or use of energy, or (3) is designated by the Administrator of OIRA as a significant energy action. For any proposed significant energy action, the agency must give a detailed statement of any adverse effects on energy supply, distribution, or use should the proposal be implemented, and of reasonable alternatives to the action and their expected benefits on energy supply, distribution, and use. Today's regulatory action would not have a significant adverse effect on the supply, distribution, or use of energy and, therefore, is not a significant energy action. Accordingly, DOE has not prepared a Statement of Energy Effects.

L. Review Under Section 32 of the Federal Energy Administration Act of 1974

Under section 301 of the Department of Energy Organization Act (Pub. L. 95–91), the Department of Energy must comply with section 32 of the Federal Energy Administration Act of 1974 (FEAA), as amended by the Federal Energy Administration Authorization Act of 1977. (15 U.S.C. 788) Section 32 provides in essence that, where a proposed rule contains or involves use of commercial standards, the notice of proposed rulemaking must inform the public of the use and background of such standards. This final rule incorporates nine commercial standards as discussed in section II.A.1 of this preamble.

The Department has evaluated these standards and is unable to conclude whether they fully comply with the requirements of section 32(b) of the FEAA, *i.e.*, that they were developed in a manner which fully provides for public participation, comment and review. As required by Section 32(c) of the FEAA, the Department has consulted with the Attorney General and the Chairman of the Federal Trade Commission concerning the impact of these two standards on competition, and neither recommended against incorporation of these standards.

M. Congressional Notification

As required by 5 U.S.C. 801, DOE will report to Congress on the promulgation of today's rule prior to its effective date. The report will state that it has been determined that the rule is not a ''major rule'' as defined by 5 U.S.C. 804(2).

N. Approval of the Office of the Secretary

The Secretary of Energy has approved publication of today's rule.

List of Subjects in 10 CFR Part 430

Administrative practice and procedure, Energy conservation, Household appliances, Incorporation by reference.

Issued in Washington, DC, on July 21, 2005.

Douglas L. Faulkner,

Acting Assistant Secretary, Energy Efficiency and Renewable Energy.

■ For the reasons set forth in the preamble, Part 430 of Chapter II of Title 10, Code of Federal Regulations is amended as set forth below.

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. Section 430.22 is amended:
■ a. In paragraph (b)(1) by adding paragraph (b)(1)8.
■ b. In paragraph (b)(5) by removing paragraph (b)(5)2., and adding new paragraphs (b)(5)2. through (b)(5)9.
■ c. By adding paragraph (b)(8).

The additions specified above read as follows:

§ 430.22 Reference Sources.

* * * * *

(b) * * *

(1) * * *

8. ANSI Standard Z21.56–1994, ''Gas-Fired Pool Heaters,'' section 2.9.

* * * * *

(5) * * *

2. American Society of Heating, Refrigerating, and Air-Conditioning Engineers Standard 23–1993, ''Methods of Testing for Rating Positive Displacement Refrigerant Compressors and Condensing Units.''

3. American Society of Heating, Refrigerating, and Air-Conditioning Engineers Standard 37–1988, ''Methods of Testing for Rating Unitary Air-Conditioning and Heat Pump Equipment.''

4. American Society of Heating, Refrigerating, and Air-Conditioning Engineers Standard 41.1–1986 (Reaffirmed 2001), ''Standard Method for Temperature Measurement.''

5. American Society of Heating, Refrigerating, and Air-Conditioning Engineers Standard 41.2–1987 (Reaffirmed 1992), ''Standard Methods for Laboratory Airflow Measurement.''

6. American Society of Heating, Refrigerating, and Air-Conditioning Engineers Standard 41.6–1994 (Reaffirmed 2001), ''Standard Method for Measurement of Moist Air Properties.''

7. American Society of Heating, Refrigerating, and Air-Conditioning Engineers Standard 41.9–2000, ''Calorimeter Test Methods for Mass Flow Measurements of Volatile Refrigerants.''

8. American Society of Heating, Refrigerating, and Air-Conditioning Engineers Standard 116–1995, ''Methods of Testing for Rating for Seasonal Efficiency of Unitary Air Conditioners and Heat Pumps.''

9. American Society of Heating, Refrigerating, and Air-Conditioning Engineers/Air Movement and Control Association International, Inc. Standard 51–1999/210–1999, ''Laboratory Methods of Testing Fans for Aerodynamic Performance Rating.''

* * * * *

(8) Air-Conditioning and Refrigeration Institute (ARI), 4100 North Fairfax Drive, Suite 200, Arlington, Virginia 22203–1629, (703) 524–8800, ARI Standard 210/240–2003, ''Unitary Air-Conditioning and Air-Source Heat Pump Equipment.''

* * * * *

■ 3. Section 430.23 of subpart B is amended by revising the section heading, paragraph (m) introductory heading and paragraph (m)(1), (2), and (3) to read as follows:

§ 430.23 Test procedure for measures of energy consumption.

* * * * *

(m) *Central air conditioners and heat pumps.* (1) The estimated annual operating cost for cooling-only units and air-source heat pumps shall be one of the following:

(i) For cooling-only units or the cooling portion of the estimated annual operating cost for air-source heat pumps which provide both heating and cooling, the product of:

(A) The quotient of the cooling capacity, in Btu's per hour, determined from the steady-state wet-coil test (A or A_2 Test), as described in section 3.2 of appendix M to this subpart, divided by the seasonal energy efficiency ratio (SEER), in Btu's per watt-hour, determined from section 4.1 of appendix M to this subpart;

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

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

(D) The representative average unit cost of electricity in dollars per kilowatt-hour as provided pursuant to section 323(b)(2) of the Act, the resulting product then being rounded off to the nearest dollar per year.

(ii) For air-source heat pumps which provide only heating or the heating portion of the estimated annual operating cost for air-source heat pumps which provide both heating and cooling, the product of:

(A) The quotient of the standardized design heating requirement, in Btu's per hour, nearest to the heating Region IV minimum design heating requirement, determined in section 4.2 of appendix M to this subpart, divided by the heating seasonal performance factor (HSPF), in Btu's per watt-hour, calculated for heating Region IV corresponding to the above-mentioned standardized design heating requirement and determined in section 4.2 of appendix M to this subpart;

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

(C) The adjustment factor of 0.77 which serves to adjust the calculated design heating requirement and heating load hours to the actual load experienced by a heating system;

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

(E) The representative average unit cost of electricity in dollars per kilowatt-hour as provided pursuant to section 323(b)(2) of the Act, the resulting product then being rounded off to the nearest dollar per year.

(iii) For air-source heat pumps which provide both heating and cooling, the estimated annual operating cost is the sum of the quantity determined in paragraph (m)(1)(i) of this section added to the quantity determined in paragraph (m)(1)(ii) of this section.

(2) The estimated regional annual operating cost for cooling-only units and for air-source heat pumps shall be one of the following:

(i) For cooling-only units or the cooling portion of the estimated regional annual operating cost for air-source heat pumps which provide both heating and cooling, the product of:

(A) The quotient of the cooling capacity, in Btu's per hour, determined from the steady-state wet-coil test (A or A_2 Test), as described in section 3.2 of appendix M to this subpart, divided by the seasonal energy efficiency ratio (SEER), in Btu's per watt-hour, determined from section 4.1 of appendix M to this subpart;

(B) The estimated number of regional cooling load hours per year determined from Figure 3 in section 4.3 of appendix M to this subpart;

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

(D) The representative average unit cost of electricity in dollars per kilowatt-hour as provided pursuant to section 323(b)(2) of the Act, the resulting product then being rounded off to the nearest dollar per year.

(ii) For air-source heat pumps which provide only heating or the heating portion of the estimated regional annual operating cost for air-source heat pumps which provide both heating and cooling, the product of:

(A) The estimated number of regional heating load hours per year determined from Figure 2 in section 4.3 of appendix M to this subpart;

(B) The quotient of the standardized design heating requirement, in Btu's per hour, for the appropriate generalized climatic region of interest (*i.e.*, corresponding to the regional heating load hours from ''A'') and determined in section 4.2 of appendix M to this subpart, divided by the heating seasonal performance factor (HSPF), in Btu's per watt-hour, calculated for the appropriate generalized climatic region of interest and corresponding to the above-mentioned standardized design heating requirement while being determined in section 4.2 of appendix M to this subpart;

(C) The adjustment factor of 0.77 which serves to adjust the calculated design heating requirement and heating load hours to the actual load experienced by a heating system;

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

(E) The representative average unit cost of electricity in dollars per kilowatt-hour as provided pursuant to section 323(b)(2) of the Act, the resulting product then being rounded off to the nearest dollar per year.

(iii) For air-source heat pumps which provide both heating and cooling, the estimated regional annual operating cost is the sum of the quantity determined in paragraph (m)(3)(i) of this section added to the quantity determined in paragraph (m)(3)(ii) of this section.

(3) The measure(s) of efficiency of performance for cooling-only units and air-source heat pumps shall be one or more of the following:

(i) The cooling mode efficiency measure for cooling-only units and air-source heat pumps which provide cooling shall be the seasonal energy efficiency ratio (SEER), in Btu's per watt-hour, determined according to section 4.1 of appendix M to this subpart, rounded off to the nearest 0.05.

(ii) The heating mode efficiency measure for air-source heat pumps shall be the heating seasonal performance factors (HSPF), in Btu's per watt-hour, determined according to section 4.2 of appendix M to this subpart for each applicable standardized design heating requirement within each climatic region, rounded off to the nearest 0.05.

(iii) The annual efficiency measure for air-source heat pumps which provide heating and cooling, shall be the annual performance factors (APF), in Btu's per watt-hour, determined according to section 4.3 of appendix M to this subpart for each standardized design heating requirement within each climatic region, rounded off to the nearest 0.05.

* * * * *

■ 4. Section 430.24 of subpart B is amended by revising the introductory text for paragraph (m)(1) to read as follows:

§ 430.24 Units to be tested.

* * * * *

(m)(1) For central air conditioners and heat pumps, each condensing unit (outdoor unit) shall have a condenser-evaporator (outdoor coil-indoor coil) combination selected and a sample of sufficient size tested in accordance with applicable provisions of this subpart such that

* * * * *

■ 5. Appendix M to Subpart B is revised to read as follows:

Appendix M to Subpart B of Part 430—Uniform Test Method for Measuring the Energy Consumption of Central Air Conditioners and Heat Pumps

4.1.4.2 Unit operates at an intermediate compressor speed (k=i) in order to match the building cooling load at temperature T_j, $\dot{Q}_c^{k=1}(T_j) < BL(T_j) < \dot{Q}_c^{k=2}(T_j)$.

4.1.4.3 Unit must operate continuously at maximum (k=2) compressor speed at temperature T_j, $BL(T_j) \geq \dot{Q}_c^{k=2}(T_j)$.

4.2 Heating Seasonal Performance Factor (HSPF) Calculations.

4.2.1 Additional steps for calculating the HSPF of a heat pump having a single-speed compressor that was tested with a fixed-speed indoor fan installed, a constant-air-volume-rate indoor fan installed, or with no indoor fan installed.

4.2.2 Additional steps for calculating the HSPF of a heat pump having a single-speed compressor and a variable-speed, variable-air-volume-rate indoor fan.

4.2.3 Additional steps for calculating the HSPF of a heat pump having a two-capacity compressor.

4.2.3.1 Steady-state space heating capacity when operating at low compressor capacity is greater than or equal to the building heating load at temperature T_j, $\dot{Q}_h^{k=1}(T_j) \geq BL(T_j)$.

4.2.3.2 Heat pump alternates between high (k=2) and low (k=1) compressor capacity to satisfy the building heating load at a temperature T_j, $\dot{Q}_h^{k=1}(T_j)$ BL $(T_j) < \dot{Q}_h^{k=2}(T_j)$.

4.2.3.3 Heat pump only operates at high (k=2) compressor capacity at temperature T_j and its capacity is greater than the building heating load, $BL(T_j) < \dot{Q}_h^{k=2}(T_j)$.

4.2.3.4 Heat pump must operate continuously at high (k=2) compressor capacity at temperature T_j, $BL(T_j) \geq \dot{Q}_h^{k=2}(T_j)$.

4.2.4 Additional steps for calculating the HSPF of a heat pump having a variable-speed compressor.

4.2.4.1 Steady-state space heating capacity when operating at minimum compressor speed is greater than or equal to the building heating load at temperature T_j, $\dot{Q}_h^{k=1}(T_j) \geq BL(T_j)$.

4.2.4.2 Heat pump operates at an intermediate compressor speed (k=i) in order to match the building heating load at a temperature T_j, $\dot{Q}_h^{k=1}(T_j) < BL(T_j) < \dot{Q}_h^{k=2}(T_j)$.

4.2.4.3 Heat pump must operate continuously at maximum (k=2) compressor speed at temperature T_j, $BL(T_j) \geq \dot{Q}_h^{k=2}(T_j)$.

4.2.5 Heat pumps having a heat comfort controller.

4.2.5.1 Heat pump having a heat comfort controller: Additional steps for calculating the HSPF of a heat pump having a single-speed compressor that was tested with a fixed-speed indoor fan installed, a constant-air-volume-rate indoor fan installed, or with no indoor fan installed.

4.2.5.2 Heat pump having a heat comfort controller: Additional steps for calculating the HSPF of a heat pump having a single-speed compressor and a variable-speed, variable-air-volume-rate indoor fan.

4.2.5.3 Heat pumps having a heat comfort controller: Additional steps for calculating the HSPF of a heat pump having a two-capacity compressor.

4.2.5.4 Heat pumps having a heat comfort controller: Additional steps for calculating the HSPF of a heat pump having a variable-speed compressor. [Reserved]

4.3 Calculations of the Actual and Representative Regional Annual Performance Factors for Heat Pumps.

4.3.1 Calculation of actual regional annual performance factors (APF_A) for a particular location and for each standardized design heating requirement.

4.3.2 Calculation of representative regional annual performance factors (APF_R) for each generalized climatic region and for each standardized design heating requirement.

4.4 Rounding of SEER, HSPF, and APF for reporting purposes.

1. Definitions

1.1 Annual performance factor means the total heating and cooling done by a heat pump in a particular region in one year divided by the total electric energy used in one year. Paragraph (m)(3)(iii) of § 430.23 of the Code of Federal Regulations states the calculation requirements for this rating descriptor.

1.2 ARI means Air-Conditioning and Refrigeration Institute.

1.3 ARI Standard 210/240–2003 means the test standard "Unitary Air-Conditioning and Air-Source Heat Pump Equipment" published in 2003 by ARI.

1.4 ASHRAE means the American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

1.5 ASHRAE Standard 23–93 means the test standard "Methods of Testing for Rating Positive Displacement Refrigerant Compressors and Condensing Units" published in 1993 by ASHRAE.

1.6 ASHRAE Standard 37–88 means the test standard "Methods of Testing for Rating Unitary Air-Conditioning and Heat Pump Equipment" published in 1988 by ASHRAE.

1.7 ASHRAE Standard 41.1–86 (RA 01) means the test standard "Standard Method for Temperature Measurement" published in 1986 and reaffirmed in 2001 by ASHRAE.

1.8 ASHRAE Standard 41.2–87 (RA 92) means the test standard "Standard Methods for Laboratory Airflow Measurement" published in 1987 and reaffirmed in 1992 by ASHRAE.

1.9 ASHRAE Standard 41.6–94 (RA 01) means the test standard "Method for Measurement of Moist Air Properties" published in 1994 and reaffirmed in 2001 by ASHRAE.

1.10 ASHRAE Standard 41.9–00 means the test standard "Calorimeter Test Methods for Mass Flow Measurements of Volatile Refrigerants" published in 2000 by ASHRAE.

1.11 ASHRAE Standard 51–99/AMCA Standard 210–1999 means the test standard "Laboratory Methods of Testing Fans for Aerodynamic Performance Rating" published in 1999 by ASHRAE and the Air Movement and Control Association International, Inc.

1.12 ASHRAE Standard 116–95 means the test standard "Methods of Testing for Rating for Seasonal Efficiency of Unitary Air Conditioners and Heat Pumps" published in 1995 by ASHRAE.

1.13 CFR means Code of Federal Regulations.

1.14 Constant-air-volume-rate indoor fan means a fan that varies its operating speed to provide a fixed air-volume-rate from a ducted system.

1.15 Continuously recorded, when referring to a dry bulb measurement, means that the specified temperature must be sampled at regular intervals that are equal to or less than the maximum intervals specified in section 4.3 part "a" of ASHRAE Standard 41.1–86 (RA 01). If such dry bulb temperatures are used only for test room control, it means that one samples at regular intervals equal to or less than the maximum intervals specified in section 4.3 part "b" of the same ASHRAE Standard. Regarding wet bulb temperature, dew point temperature, or relative humidity measurements, continuously recorded means that the measurements must be made at regular intervals that are equal to or less than 1 minute.

1.16 Cooling load factor (CLF) means the ratio having as its numerator the total cooling delivered during a cyclic operating interval consisting of one ON period and one OFF period. The denominator is the total cooling that would be delivered, given the same ambient conditions, had the unit operated continuously at its steady-state space cooling capacity for the same total time (ON + OFF) interval.

1.17 Coefficient of Performance (COP) means the ratio of the average rate of space heating delivered to the average rate of electrical energy consumed by the heat pump. These rate quantities must be determined from a single test or, if derived via interpolation, must be tied to a single set of operating conditions. COP is a dimensionless quantity. When determined for a ducted unit tested without an indoor fan installed, COP must include the section 3.7, 3.8, and 3.9.1 default values for the heat output and power input of a fan motor.

1.18 Cyclic Test means a test where the unit's compressor is cycled on and off for specific time intervals. A cyclic test provides half the information needed to calculate a degradation coefficient.

1.19 Damper box means a short section of duct having an air damper that meets the performance requirements of section 2.5.7.

1.20 Degradation coefficient (C_D) means a parameter used in calculating the part load factor. The degradation coefficient for cooling is denoted by C_D^c. The degradation coefficient for heating is denoted by C_D^h.

1.21 Demand-defrost control system means a system that defrosts the heat pump outdoor coil only when measuring a predetermined degradation of performance. The heat pump's controls monitor one or more parameters that always vary with the amount of frost accumulated on the outdoor coil (*e.g.*, coil to air differential temperature, coil differential air pressure, outdoor fan power or current, optical sensors, etc.) at least once for every ten minutes of compressor ON-time when space heating. One acceptable alternative to the criterion given in the prior sentence is a feedback system that measures the length of the defrost period and adjusts defrost frequency accordingly.[1] In all cases, when the frost parameter(s) reaches a predetermined value,

[1] Systems that vary defrost intervals according to outdoor dry-bulb temperature are not demand defrost systems.

the system initiates a defrost. In a demand-defrost control system, defrosts are terminated based on monitoring a parameter(s) that indicates that frost has been eliminated from the coil.

A demand-defrost control system, which otherwise meets the above requirements, may allow time-initiated defrosts if, and only if, such defrosts occur after 6 hours of compressor operating time.

1.22 Design heating requirement (DHR) predicts the space heating load of a residence when subjected to outdoor design conditions. Estimates for the minimum and maximum DHR are provided for six generalized U.S. climatic regions in section 4.2.

1.23 Dry-coil tests are cooling mode tests where the wet-bulb temperature of the air supplied to the indoor coil is maintained low enough that no condensate forms on this coil.

1.24 Ducted system means an air conditioner or heat pump that is designed to be permanently installed equipment and delivers conditioned air to the indoor space through a duct(s). The air conditioner or heat pump may be either a split system or a single-packaged unit.

1.25 Energy efficiency ratio (EER) means the ratio of the average rate of space cooling delivered to the average rate of electrical energy consumed by the air conditioner or heat pump. These rate quantities must be determined from a single test or, if derived via interpolation, must be tied to a single set of operating conditions. EER is expressed in units of

$$\frac{\text{Btu/h}}{\text{W}}.$$

When determined for a ducted unit tested without an indoor fan installed, EER must include the section 3.3 and 3.5.1 default values for the heat output and power input of a fan motor.

1.26 Heating load factor (HLF) means the ratio having as its numerator the total heating delivered during a cyclic operating interval consisting of one ON period and one OFF period. The denominator is the total heating that would be delivered, given the same ambient conditions, if the unit operated continuously at its steady-state space heating capacity for the same total time (ON plus OFF) interval.

1.27 Heating seasonal performance factor (HSPF) means the total space heating required during the space heating season, expressed in Btu's, divided by the total electrical energy consumed by the heat pump system during the same season, expressed in watt-hours. The HSPF used to evaluate compliance with the Energy Conservation Standards (see 10 CFR 430.32(c), Subpart C) is based on Region IV, the minimum standardized design heating requirement, and the sampling plan stated in 10 CFR 430.24(m), Subpart B.

1.28 Heat pump having a heat comfort controller means equipment that regulates the operation of the electric resistance elements to assure that the air temperature leaving the indoor section does not fall below a specified temperature. This specified temperature is usually field adjustable. Heat pumps that actively regulate the rate of electric resistance heating when operating below the balance point (as the result of a second stage call from the thermostat) but do not operate to maintain a minimum delivery temperature are not considered as having a heat comfort controller.

1.29 Mini-split air conditioners and heat pumps means systems that have a single outdoor section and one or more indoor sections. The indoor sections cycle on and off in unison in response to a single indoor thermostat.

1.30 Multiple-split air conditioners and heat pumps means systems that have two or more indoor sections. The indoor sections operate independently and can be used to condition multiple zones in response to multiple indoor thermostats.

1.31 Non-ducted system means an air conditioner or heat pump that is designed to be permanently installed equipment and directly heats or cools air within the conditioned space using one or more indoor coils that are mounted on room walls and/or ceilings. The unit may be of a modular design that allows for combining multiple outdoor coils and compressors to create one overall system. Non-ducted systems covered by this test procedure are all split systems.

1.32 Part-load factor (PLF) means the ratio of the cyclic energy efficiency ratio (coefficient of performance) to the steady-state energy efficiency ratio (coefficient of performance). Evaluate both energy efficiency ratios (coefficients of performance) based on operation at the same ambient conditions.

1.33 Seasonal energy efficiency ratio (SEER) means the total heat removed from the conditioned space during the annual cooling season, expressed in Btu's, divided by the total electrical energy consumed by the air conditioner or heat pump during the same season, expressed in watt-hours. The SEER calculation in section 4.1 of this Appendix and the sampling plan stated in 10 CFR Subpart B, 430.24(m) are used to evaluate compliance with the Energy Conservation Standards. (See 10 CFR 430.32(c), Subpart C.)

1.34 Single-packaged unit means any central air conditioner or heat pump that has all major assemblies enclosed in one cabinet.

1.35 Small-duct, high-velocity system means a system that contains a blower and indoor coil combination that is designed for, and produces, at least 1.2 inches (of water) of external static pressure when operated at the certified air volume rate of 220–350 cfm per rated ton of cooling. When applied in the field, small-duct products use high-velocity room outlets (*i.e.*, generally greater than 1000 fpm) having less than 6.0 square inches of free area.

1.36 Split system means any air conditioner or heat pump that has one or more of the major assemblies separated from the others.

1.37 Standard Air means dry air at 70 °F and 14.696 psia. Under these conditions, dry air has a mass density of 0.075 lb/ft^3.

1.38 Steady-state test means a test where the test conditions are regulated to remain as constant as possible while the unit operates continuously in the same mode.

1.39 Temperature bin means the 5 °F increments that are used to partition the outdoor dry-bulb temperature ranges of the cooling (≥ 65 °F) and heating (< 65 °F) seasons.

1.40 Test condition tolerance means the maximum permissible difference between the average value of the measured test parameter and the specified test condition.

1.41 Test operating tolerance means the maximum permissible range that a measurement may vary over the specified test interval. The difference between the maximum and minimum sampled values must be less than or equal to the specified test operating tolerance.

1.42 Time adaptive defrost control system is a demand-defrost control system (see definition 1.21) that measures the length of the prior defrost period(s) and uses that information to automatically determine when to initiate the next defrost cycle.

1.43 Time-temperature defrost control systems initiate or evaluate initiating a defrost cycle only when a predetermined cumulative compressor ON-time is obtained. This predetermined ON-time is generally a fixed value (*e.g.*, 30, 45, 90 minutes) although it may vary based on the measured outdoor dry-bulb temperature. The ON-time counter accumulates if controller measurements (*e.g.*, outdoor temperature, evaporator temperature) indicate that frost formation conditions are present, and it is reset/remains at zero at all other times. In one application of the control scheme, a defrost is initiated whenever the counter time equals the predetermined ON-time. The counter is reset when the defrost cycle is completed.

In a second application of the control scheme, one or more parameters are measured (*e.g.*, air and/or refrigerant temperatures) at the predetermined, cumulative, compressor ON-time. A defrost is initiated only if the measured parameter(s) falls within a predetermined range. The ON-time counter is reset regardless of whether a defrost is initiated. If systems of this second type use cumulative ON-time intervals of 10 minutes or less, then the heat pump may qualify as having a demand defrost control system (see definition 1.21).

1.44 Triple-split system means an air conditioner or heat pump that is composed of three separate components: An outdoor fan coil section, an indoor fan coil section, and an indoor compressor section.

1.45 Two-capacity (or two-stage) compressor means an air conditioner or heat pump that has one of the following:

(1) A two-speed compressor,

(2) Two compressors where only one compressor ever operates at a time,

(3) Two compressors where one compressor (Compressor #1) operates at low loads and both compressors (Compressors #1 and #2) operate at high loads but Compressor #2 never operates alone, or

(4) A compressor that is capable of cylinder or scroll unloading.

For such systems, low capacity means:

(1) Operating at low compressor speed,

(2) Operating the lower capacity compressor,

(3) Operating Compressor #1, or

(4) Operating with the compressor unloaded (*e.g.*, operating one piston of a two-piston reciprocating compressor, using a

fixed fractional volume of the full scroll, etc.).

For such systems, high capacity means:

(1) Operating at high compressor speed,

(2) Operating the higher capacity compressor,

(3) Operating Compressors #1 and #2, or

(4) Operating with the compressor loaded (*e.g.*, operating both pistons of a two-piston reciprocating compressor, using the full volume of the scroll).

1.46 Two-capacity, northern heat pump means a heat pump that has a factory or field-selectable lock-out feature to prevent space cooling at high-capacity. Two-capacity heat pumps having this feature will typically have two sets of ratings, one with the feature disabled and one with the feature enabled. The indoor coil model number should reflect whether the ratings pertain to the lockout enabled option via the inclusion of an extra identifier, such as ''+LO.'' When testing as a two-capacity, northern heat pump, the lockout feature must remain enabled for all tests.

1.47 Wet-coil test means a test conducted at test conditions that typically cause water vapor to condense on the test unit evaporator coil.

2. Testing Conditions

This test procedure covers split-type and single-packaged ducted units and split-type non-ducted units. Except for units having a variable-speed compressor, ducted units tested without an indoor fan installed are covered.

a. Only a subset of the sections listed in this test procedure apply when testing and rating a particular unit. Tables 1–A through 1–C show which sections of the test procedure apply to each type of equipment. In each table, look at all four of the Roman numeral categories to see what test sections apply to the equipment being tested.

1. The first category, Rows I–1 through I–4 of the Tables, pertains to the compressor and indoor fan features of the equipment. After identifying the correct ''I'' row, find the table cells in the same row that list the type of equipment being tested: Air conditioner (AC), heat pump (HP), or heating-only heat pump (HH). Use the test section(s) listed above each noted table cell for testing and rating the unit.

2. The second category, Rows II–1 and II–2, pertains to the presence or absence of ducts. Row II–1 shows the test procedure sections that apply to ducted systems, and Row II–2 shows those that apply to non-ducted systems.

3. The third category is for special features that may be present in the equipment. When testing units that have one or more of the three (special) equipment features described by the Table legend for Category III, use Row III to find test sections that apply.

4. The fourth category is for the secondary test method to be used. If the secondary method for determining the unit's cooling and/or heating capacity is known, use Row IV to find the appropriate test sections. Otherwise, include all of the test sections referenced by Row IV cell entries—*i.e.*, sections 2.10 to 2.10.3 and 3.11 to 3.11.3—among those sections consulted for testing and rating information.

b. Obtain a complete listing of all pertinent test sections by recording those sections identified from the four categories above.

c. The user should note that, for many sections, only part of a section applies to the unit being tested. In a few cases, the entire section may not apply. For example, sections 3.4 to 3.5.3 (which describe optional dry coil tests), are not relevant if the allowed default value for the cooling mode cyclic degradation coefficient is used rather than determining it by testing.

Example for Using Tables 1–A to 1–C

Equipment Description: A ducted air conditioner having a single-speed compressor, a fixed-speed indoor fan, and a multi-speed outdoor fan.

Secondary Test Method: Refrigerant Enthalpy Method

Step 1. Determine which of four listed Row ''I'' options applies ==> Row I–2

Table 1–A: ''AC'' in Row I–2 is found in the columns for sections 1.1 to 1.47, 2.1 to 2.2, 2.2.4 to 2.2.4.1, 2.2.5, 2.3 to 2.3.1, 2.4 to 2.4.1, 2.5, 2.5.2 to 2.10, and 2.11 to 2.13.

Table 1–B: ''AC'' is listed in Row I–2 for sections 3 to 3.1.4, 3.1.5 to 3.1.8, 3.2.1, 3.3 to 3.5, 3.5.3, 3.11 and 3.12.

Table 1–C: ''AC'' is listed in Row I–2 for sections 4.1.1 and 4.4.

Step 2. Equipment is ducted ==> Row II–1

Table 1–A: ''AC'' is listed in Row II–1 for sections 2.4.2 and 2.5.1 to 2.5.1.2.

Table 1–B: ''AC'' is listed in Row II–1 for sections 3.1.4.1 to 3.1.4.1.1 and 3.5.1.

Table 1–C: no ''AC'' listings in Row II–1.

Step 3. Equipment Special Features include multi-speed outdoor fan ==> Row III, M

Table 1–A: ''M'' is listed in Row III for section 2.2.2

Tables 1–B and 1–C: no ''M'' listings in Row III.

Step 4. Secondary Test Method is Refrigerant Enthalpy Method ==> Row IV, R

Table 1–A: ''R'' is listed in Row IV for section 2.10.3

Table 1–B: ''R'' is listed in Row IV for section 3.11.3

Table 1–C: no ''R'' listings in Row IV.

Step 5. Cumulative listing of applicable test procedure sections 1.1 to 1.47, 2.1 to 2.2, 2.2.2, 2.2.4 to 2.4.1, 2.2.5, 2.3 to 2.3.1, 2.4 to 2.4.1, 2.4.2, 2.5, 2.5.1 to 2.5.1.2, 2.5.2 to 2.10, 2.10.3, 2.11 to 2.13, 3. to 3.1.4, 3.1.4.1 to 3.1.4.1.1, 3.1.5 to 3.1.8, 3.2.1, 3.3 to 3.5, 3.5.1, 3.5.3, 3.11, 3.11.3, 3.12, 4.1.1, and 4.4.

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Table 1A. Selection of Test Procedure Sections: Section 1 (Definitions) and Section 2 (Testing Conditions)

Key Equipment Features and Secondary Test Method / Sections From the Test Procedure	1.1 to 1.47	2.1 to 2.2	2.2.1	2.2.2	2.2.3	2.2.4 to 2.2.4.1	2.2.4.2	2.2.5	2.3 to 2.3.1	2.3.2	2.4 to 2.4.1	2.4.2	2.5	2.5.1 to 2.5.1.2	2.5.2 to 2.10	2.10.1	2.10.2	2.10.3	2.11 to 2.13
I-1. Single-speed Compressor; Variable-Speed Variable Air Volume Indoor Fan	AC HP HH	AC HP HH	HP HH			AC HP	HP HH	AC HP HH	AC HP	HP HH	AC HP HH		AC HP HH		AC HP HH				AC HP HH
I-2. Single-speed Compressor Except as Covered by "I-1"	AC HP HH	AC HP HH	HP HH			AC HP	HP HH	AC HP HH	AC HP	HP HH	AC HP HH		AC HP HH		AC HP HH				AC HP HH
I-3. Two-capacity Compressor	AC HP HH	AC HP HH	HP HH			AC HP	HP HH	AC HP HH	AC HP	HP HH	AC HP HH		AC HP HH		AC HP HH				AC HP HH
I-4. Variable-speed Compressor	AC HP HH	AC HP HH	HP HH			AC HP	HP HH	AC HP HH	AC HP	HP HH	AC HP HH		AC HP HH		AC HP HH				AC HP HH
II-1. Ducted												AC HP HH		AC HP HH					
II-2. Non-Ducted																			
III. Special Features				M	G														
IV. Secondary Test Method																O	C	R	

Legend for Table Entries:

Categories I and II: AC = applies for an Air Conditioner that meets the corresponding Column 1 "Key Equipment . . ." criterion
HP = applies for a Heat Pump that meets the corresponding Column 1 "Key Equipment . . ." criterion
HH = applies for a Heating-only Heat pump that meets the corresponding Column 1 "Key Equipment . . ." criterion

Category III: G = ganged mini-splits or multi-splits;
H = heat pump with a heat comfort controller;
M = units with a multi-speed outdoor fan.

Category IV: O = Outdoor Air Enthalpy Method; C = Compressor Calibration Method; R = Refrigerant Enthalpy Method

Table 1B. Selection of Test Procedure Sections: Section 3 (Testing Procedures)

Key Equipment Features and Secondary Test Method \ Sections From the Test Procedure	3. to 3.1.4	3.1.4.1 to 3.1.4.1.1	3.1.4.1.2	3.1.4.2	3.1.4.3	3.1.4.4 to 3.1.4.4.2	3.1.4.4.3	3.1.4.4.4	3.1.4.5	3.1.4.6 to 3.1.4.7	3.1.5 to 3.1.8	3.1.9	3.2.1	3.2.2 to 3.2.2.2	3.2.3	3.2.4	3.3 to 3.5	3.5.1	3.5.2	3.5.3
I-1. Single-speed Compressor; Variable-speed, Variable Air Volume Indoor Fan	AC HP HH			AC HP					HP HH		AC HP HH	HP HH		AC HP			AC HP			AC HP
I-2. Single-speed Compressor Except as Covered by "I-1"	AC HP HH										AC HP HH	HP HH	AC HP				AC HP			AC HP
I-3. Two-capacity Compressor	AC HP HH			AC HP					HP HH		AC HP HH	HP HH			AC HP		AC HP			AC HP
I-4. Variable-speed Compressor	AC HP HH			AC HP	AC HP				HP HH	HP HH	AC HP HH	HP HH				AC HP	AC HP			AC HP
II-1. Ducted		AC HP				HP	HH											AC HP		
II-2. Non-Ducted			AC HP					HP HH											AC HP	
III. Special Features												H								
IV. Secondary Test Method																				

Legend for Table Entries:

Categories I and II:
- AC = applies for an Air Conditioner that meets the corresponding Column 1 "Key Equipment . . ." criterion
- HP = applies for a Heat Pump that meets the corresponding Column 1 "Key Equipment . . ." criterion
- HH = applies for a Heating-only Heat pump that meets the corresponding Column 1 "Key Equipment . . ." criterion

Category III:
- G = ganged mini-splits or multi-splits;
- H = heat pump with a heat comfort controller;
- M = units with a multi-speed outdoor fan.

Category IV:
- O = Outdoor Air Enthalpy Method; C = Compressor Calibration Method; R = Refrigerant Enthalpy Method

Table 1B. Selection of Test Procedure Sections: Section 3 (Testing Procedures) (continued)

Key Equipment Features and Secondary Test Method \ Sections From the Test Procedure	3.6.1	3.6.2	3.6.3	3.6.4	3.6.5	3.7 to 3.8.1	3.9 to 3.10	3.11	3.11.1 to 3.11.1.3	3.11.2	3.11.3	3.12
I-1. Single-speed Compressor; Variable-speed, Variable Air Volume Indoor Fan		HP HH				HP HH	HP HH	AC HP HH				AC HP HH
I-2. Single-speed Compressor Except as Covered by "I-1"	HP HH					HP HH	HP HH	AC HP HH				AC HP HH
I-3. Two-capacity Compressor			HP HH			HP HH	HP HH	AC HP HH				AC HP HH
I-4. Variable-speed Compressor				HP HH		HP HH	HP HH	AC HP HH				AC HP HH
II-1. Ducted												
II-2. Non-Ducted												
III. Special Features					H							
IV. Secondary Test Method									O	C	R	

Legend for Table Entries:

Categories I and II: AC = applies for an Air Conditioner that meets the corresponding Column 1 "Key Equipment . . ." criterion
HP = applies for a Heat Pump that meets the corresponding Column 1 "Key Equipment . . ." criterion
HH = applies for a Heating-only Heat pump that meets the corresponding Column 1 "Key Equipment . . ." criterion

Category III: G = ganged mini-splits or multi-splits;
H = heat pump with a heat comfort controller;
M = units with a multi-speed outdoor fan.

Category IV: O = Outdoor Air Enthalpy Method; C = Compressor Calibration Method; R = Refrigerant Enthalpy Method

Table 1C. Selection of Test Procedure Sections: Section 4 (Calculations of Seasonal Performance Descriptors)

Key Equipment Features and Secondary Test Method \ Sections From the Test Procedure	4 to 4.1	4.1.1	4.1.2 to 4.1.2.2	4.1.3 to 4.1.3.4	4.1.4 to 4.1.4.3	4.2	4.2.1	4.2.2	4.2.3 to 4.2.3.4	4.2.4 to 4.2.4.3	4.2.5 to 4.2.5.4	4.3 to 4.3.2	4.4
I-1. Single-speed Compressor; Variable-speed Variable Air Volume Indoor Fan	AC HP		AC HP			HP HH		HP HH				HP	AC HP HH
I-2. Single-speed Compressor Except as Covered by "I-1"		AC HP				HP HH	HP HH					HP	AC HP HH
I-3. Two-capacity Compressor	AC HP			AC HP		HP HH			HP HH			HP	AC HP HH
I-4. Variable-speed Compressor	AC HP				AC HP	HP HH				HP HH		HP	AC HP HH
II-1. Ducted													
II-2. Non-Ducted													
III. Special Features						H					H		
IV. Secondary Test Method													

Legend for Table Entries:

Categories I and II: AC = applies for an Air Conditioner that meets the corresponding Column 1 "Key Equipment . . ." criterion

HP = applies for a Heat Pump that meets the corresponding Column 1 "Key Equipment . . ." criterion

HH = applies for a Heating-only Heat pump that meets the corresponding Column 1 "Key Equipment . . ." criterion

Category III: G = ganged mini-splits or multi-splits;

H = heat pump with a heat comfort controller;

M = units with a multi-speed outdoor fan.

Category IV: O = Outdoor Air Enthalpy Method; C = Compressor Calibration Method; R = Refrigerant Enthalpy Method

2.1 Test room requirements. a. Test using two side-by-side rooms, an indoor test room and an outdoor test room. These rooms must comply with the requirements specified in sections 8.1.2 and 8.1.3 of ASHRAE Standard 37–88 (incorporated by reference, see § 430.22).

b. Inside these test rooms, use artificial loads during cyclic tests and frost accumulation tests, if needed, to produce stabilized room air temperatures. For one room, select an electric resistance heater(s) having a heating capacity that is approximately equal to the heating capacity of the test unit's condenser. For the second room, select a heater(s) having a capacity that is close to the sensible cooling capacity of the test unit's evaporator. When applied, cycle the heater located in the same room as the test unit evaporator coil ON and OFF when the test unit cycles ON and OFF. Cycle the heater located in the same room as the test unit condensing coil ON and OFF when the test unit cycles OFF and ON.

2.2 Test unit installation requirements. a. Install the unit according to section 8.6 of ASHRAE Standard 37–88 (incorporated by reference, see § 430.22). With respect to interconnecting tubing used when testing split systems, however, follow the requirements given in section 6.1.3.5 of ARI Standard 210/240–2003 (incorporated by reference, see § 430.22). When testing triple-split systems (see Definition 1.44), use the tubing length specified in section 6.1.3.5 of ARI Standard 210/240–2003 (incorporated by reference, see § 430.22) to connect the outdoor coil, indoor compressor section, and indoor coil while still meeting the requirement of exposing 10 feet of the tubing to outside conditions. When testing non-ducted systems having multiple indoor coils, connect each indoor fan-coil to the outdoor unit using: a. 25 feet of tubing, or b. tubing furnished by the manufacturer, whichever is longer. If they are needed to make a secondary measurement of capacity, install refrigerant pressure measuring instruments as described in section 8.6.5 of ASHRAE Standard 37–88 (incorporated by reference, see § 430.22). Refer to section 2.10 of this Appendix to learn which secondary methods require refrigerant pressure measurements. At a minimum, insulate the low pressure line(s) of a split system with foam insulation having an inside diameter that matches the refrigerant tubing and a nominal thickness of ½ inch.

b. For units designed for both horizontal and vertical installation or for both up-flow and down-flow vertical installations, the manufacturer must specify the orientation used for testing. Conduct testing with the following installed:

(1) The most restrictive filter(s);

(2) Supplementary heating coils; and

(3) Other equipment specified as part of the unit, including all hardware used by a heat comfort controller if so equipped (see Definition 1.28).

c. Testing a ducted unit without having an indoor air filter installed is permissible as long as the minimum external static pressure requirement is adjusted as stated in Table 2, note 3 (see section 3.1.4). Except as noted in section 3.1.9, prevent the indoor air supplementary heating coils from operating during all tests. For coil-only indoor units that are supplied without an enclosure, create an enclosure using 1 inch fiberglass ductboard having a nominal density of 6 pounds per cubic foot. Or alternatively, use some other insulating material having a thermal resistance (''R'' value) between 4 and 6 hr·ft²·°F/Btu. For units where the coil is housed within an enclosure or cabinet, no extra insulating or sealing is allowed.

2.2.1 Defrost control settings. Set heat pump defrost controls at the normal settings which most typify those encountered in generalized climatic region IV. (Refer to Figure 2 and Table 17 of section 4.2 for information on region IV.) For heat pumps that use a time-adaptive defrost control system (see Definition 1.42), the manufacturer must specify the frosting interval to be used during Frost Accumulation tests and provide the procedure for manually initiating the defrost at the specified time. To ease testing of any unit, the manufacturer should provide information and any necessary hardware to manually initiate a defrost cycle.

2.2.2 Special requirements for units having a multiple-speed outdoor fan. Configure the multiple-speed outdoor fan according to the manufacturer's specifications, and thereafter, leave it unchanged for all tests. The controls of the unit must regulate the operation of the outdoor fan during all lab tests except dry coil cooling mode tests. For dry coil cooling mode tests, the outdoor fan must operate at the same speed used during the required wet coil test conducted at the same outdoor test conditions.

2.2.3 Special requirements for multi-split air conditioners and heat pumps, and systems composed of multiple mini-split units (outdoor units located side-by-side) that would normally operate using two or more indoor thermostats. During the steady-state tests, shunt all thermostats to make all indoor fan-coil units operate simultaneously. To ease the testing burden of cyclic tests, consider creating a single control circuit that allows simultaneous cycling of all compressor systems. For these systems, the test procedure references to a single indoor fan, outdoor fan, and compressor means all indoor fans, all outdoor fans, and all compressor systems.

2.2.4 Wet-bulb temperature requirements for the air entering the indoor and outdoor coils.

2.2.4.1 Cooling mode tests. For wet-coil cooling mode tests, regulate the water vapor content of the air entering the indoor unit to the applicable wet-bulb temperature listed in Tables 3 to 6. As noted in these same tables, achieve a wet-bulb temperature during dry-coil cooling mode tests that results in no condensate forming on the indoor coil. Controlling the water vapor content of the air entering the outdoor side of the unit is not required for cooling mode tests except when testing:

(1) Units that reject condensate to the outdoor coil during wet coil tests. Tables 3–6 list the applicable wet-bulb temperatures.

(2) Single-packaged units where all or part of the indoor section is located in the outdoor test room. The average dew point temperature of the air entering the outdoor coil during wet coil tests must be within ±3.0°F of the average dew point temperature of the air entering the indoor coil over the 30-minute data collection interval described in section 3.3. For dry coil tests on such units, it may be necessary to limit the moisture content of the air entering the outdoor side of the unit to meet the requirements of section 3.4.

2.2.4.2 Heating mode tests. For heating mode tests, regulate the water vapor content of the air entering the outdoor unit to the applicable wet-bulb temperature listed in Tables 9 to 12. The wet-bulb temperature entering the indoor side of the heat pump must not exceed 60°F. Additionally, if the Outdoor Air Enthalpy test method is used while testing a single-packaged heat pump where all or part of the outdoor section is located in the indoor test room, adjust the wet-bulb temperature for the air entering the indoor side to yield an indoor-side dew point temperature that is as close as reasonably possible to the dew point temperature of the outdoor-side entering air.

2.2.5 Additional refrigerant charging requirements. Charging according to the ''manufacturer's instructions,'' as stated in section 8.6 of ASHRAE Standard 37–88 (incorporated by reference, see § 430.22), means the manufacturer's installation instructions that come packaged with the unit. If a unit requires charging but the installation instructions do not specify a charging procedure, then evacuate the unit and add the nameplate refrigerant charge. Where the manufacturer's installation instructions contain two sets of refrigerant charging criteria, one for field installations and one for lab testing, use the field installation criteria. For third-party testing, the test laboratory may consult with the manufacturer about the refrigerant charging procedure and make any needed corrections so long as they do not contradict the published installation instructions. The manufacturer may specify an alternative charging criteria to the third-party laboratory so long as the manufacturer thereafter revises the published installation instructions accordingly.

2.3 Indoor air volume rates. If a unit's controls allow for overspeeding the indoor fan (usually on a temporary basis), take the necessary steps to prevent overspeeding during all tests.

2.3.1 Cooling tests. a. Set indoor fan control options (*e.g.*, fan motor pin settings, fan motor speed) according to the published installation instructions that are provided with the equipment while meeting the airflow requirements that are specified in sections 3.1.4.1 to 3.1.4.3.

b. Express the Cooling Certified Air Volume Rate, the Cooling Minimum Air Volume Rate, and the Cooling Intermediate Air Volume Rate in terms of standard air.

2.3.2 Heating tests. a. If needed, set the indoor fan control options (*e.g.*, fan motor pin settings, fan motor speed) according to the published installation instructions that are provided with the equipment. Do this set-up while meeting all applicable airflow requirements specified in sections 3.1.4.4 to 3.1.4.7.

b. Express the Heating Certified Air Volume Rate, the Heating Minimum Air Volume Rate, the Heating Intermediate Air Volume Rate, and the Heating Nominal Air Volume Rate in terms of standard air.

2.4 Indoor coil inlet and outlet duct connections. Insulate and/or construct the outlet plenum described in section 2.4.1 and, if installed, the inlet plenum described in section 2.4.2 with thermal insulation having a nominal overall sistance (R-value) of at least 19 hr·ft²·°F/Btu.

2.4.1 Outlet plenum for the indoor unit. Attach a plenum to the outlet of the indoor coil. (Note: for some packaged systems, the indoor coil may be located in the outdoor test room.) For non-ducted systems having multiple indoor coils, attach a plenum to each indoor coil outlet. Add a static pressure tap to each face of the (each) outlet plenum, if rectangular, or at four evenly distributed locations along the circumference of an oval or round plenum. Create a manifold that connects the four static pressure taps. Figure 1 shows two of the three options allowed for the manifold configuration; the third option is the broken-ring, four-to-one manifold configuration that is shown in Figure 7 of ASHRAE Standard 37–88 (incorporated by reference, see § 430.22). See Figures 7 and 8 of ASHRAE Standard 37–88 (incorporated by reference, see § 430.22) for the cross-sectional dimensions and minimum length of the (each) plenum and the locations for adding the static pressure taps for units tested with and without an indoor fan installed. For a non-ducted system having multiple indoor coils, have all outlet plenums discharge air into a single common duct. At the plane where each plenum enters the common duct, install an adjustable airflow damper and use it to equalize the static pressure in each plenum.

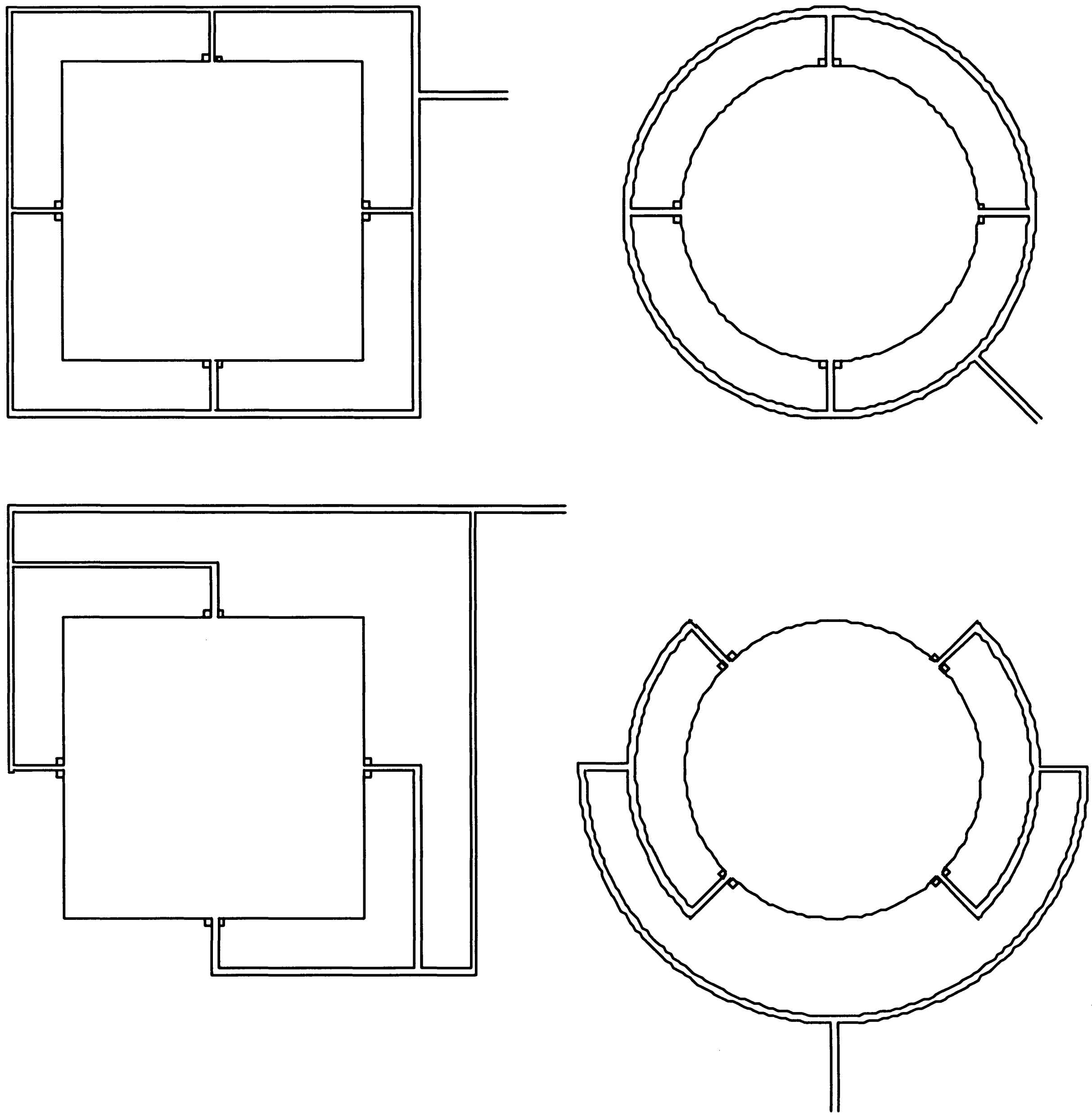

Figure 1. Configurations for manifolding the static pressure taps. The top two diagrams show the complete ring, four-to-one configuration. The lower two diagrams show the triple-T configuration.

2.4.2 Inlet plenum for the indoor unit. Install an inlet plenum when testing a coil-only indoor unit or a packaged system where the indoor coil is located in the outdoor test room. Add static pressure taps at the center of each face of this plenum, if rectangular, or at four evenly distributed locations along the circumference of an oval or round plenum. Make a manifold that connects the four static pressure taps using one of the three configurations specified in section 2.4.1. See Figure 8 of ASHRAE Standard 37–88 (incorporated by reference, see § 430.22) for cross-sectional dimensions, the minimum length of the inlet plenum, and the locations of the static pressure taps. When testing a ducted unit having an indoor fan (and the indoor coil is in the indoor test room), the manufacturer has the option to test with or without an inlet plenum installed. Space limitations within the test room may dictate that the manufacturer choose the latter option. If used, construct the inlet plenum and add the four static pressure taps as shown in Figure 8 of ASHRAE Standard 37–88 (incorporated by reference, see § 430.22). Manifold the four static pressure taps using one of the three configurations specified in section 2.4.1. Never use an inlet plenum when testing a non-ducted system.

2.5 Indoor coil air property measurements and air damper box applications. a. Measure the dry-bulb temperature and water vapor content of the air entering and leaving the indoor coil. If needed, use an air sampling device to divert air to a sensor(s) that measures the water vapor content of the air. See Figure 2 of ASHRAE Standard 41.1–86 (RA 01) (incorporated by reference, see § 430.22) for guidance on constructing an air sampling device. The sampling device may also divert air to a remotely located sensor(s) that measures dry bulb temperature. The air sampling device and the remotely located temperature sensor(s) may be used to determine the entering air dry bulb temperature during any test. The air sampling device and the remotely located leaving air dry bulb temperature sensor(s) may be used for all tests except:

(1) Cyclic tests; and

(2) Frost accumulation tests.

b. An acceptable alternative in all cases, including the two special cases noted above, is to install a grid of dry bulb temperature sensors within the outlet and inlet ducts. Use a temperature grid to get the average dry bulb temperature at one location, leaving or entering, or when two grids are applied as a thermopile, to directly obtain the temperature difference. A grid of temperature sensors (which may also be used for determining average leaving air dry bulb temperature) is required to measure the temperature distribution within a cross-section of the leaving airstream.

c. Use an inlet and outlet air damper box when testing ducted systems if conducting one or both of the cyclic tests listed in sections 3.2 and 3.6. Otherwise, install an outlet air damper box when testing heat pumps, both ducted and non-ducted, that cycle off the indoor fan during defrost cycles if no other means is available for preventing natural or forced convection through the indoor unit when the indoor fan is off. Never use an inlet damper box when testing a non-ducted system.

2.5.1 Test set-up on the inlet side of the indoor coil: for cases where the inlet damper box is installed. a. Install the inlet side damper box as specified in section 2.5.1.1 or 2.5.1.2, whichever applies. Insulate or construct the ductwork between the point where the air damper is installed and where the connection is made to either the inlet plenum (section 2.5.1.1 units) or the indoor unit (section 2.5.1.2 units) with thermal insulation that has a nominal overall resistance (R-value) of at least 19 hr·ft^2·°F/Btu.

b. Locate the grid of entering air dry-bulb temperature sensors, if used, at the inlet of the damper box. Locate the air sampling device, or the sensor used to measure the water vapor content of the inlet air, at a location immediately upstream of the damper box inlet.

2.5.1.1 If the section 2.4.2 inlet plenum is installed. Install the inlet damper box upstream of the inlet plenum. The cross-sectional flow area of the damper box must be equal to or greater than the flow area of the inlet plenum. If needed, use an adaptor plate or a transition duct section to connect the damper box with the inlet plenum.

2.5.1.2 If the section 2.4.2 inlet plenum is not installed. Install the damper box immediately upstream of the air inlet of the indoor unit. The cross-sectional dimensions of the damper box must be equal to or greater than the dimensions of the indoor unit inlet. If needed, use an adaptor plate or a short transition duct section to connect the damper box with the unit's air inlet. Add static pressure taps at the center of each face of the damper box, if rectangular, or at four evenly distributed locations along the circumference, if oval or round. Locate the pressure taps between the inlet damper and the inlet of the indoor unit. Make a manifold that connects the four static pressure taps.

2.5.2 Test set-up on the inlet side of the indoor unit: for cases where no inlet damper box is installed. If using the section 2.4.2 inlet plenum and a grid of dry bulb temperature sensors, mount the grid at a location upstream of the static pressure taps described in section 2.4.2, preferably at the entrance plane of the inlet plenum. If the section 2.4.2 inlet plenum is not used, but a grid of dry bulb temperature sensors is used, locate the grid approximately 6 inches upstream from the inlet of the indoor coil. Or, in the case of non-ducted units having multiple indoor coils, locate a grid approximately 6 inches upstream from the inlet of each indoor coil. Position an air sampling device, or the sensor used to measure the water vapor content of the inlet air, immediately upstream of the (each) entering air dry-bulb temperature sensor grid. If a grid of sensors is not used, position the entering air sampling device (or the sensor used to measure the water vapor content of the inlet air) as if the grid were present.

2.5.3 Indoor coil static pressure difference measurement. Section 6.4.4.1 of ASHRAE Standard 37–88 (incorporated by reference, see § 430.22) describes the method for fabricating static pressure taps. Also refer to Figure 2A of ASHRAE Standard 51–99/AMCA Standard 210–99 (incorporated by reference, see § 430.22). Use a differential pressure measuring instrument that is accurate to within ±0.01 inches of water and has a resolution of at least 0.01 inches of water to measure the static pressure difference between the indoor coil air inlet and outlet. Connect one side of the differential pressure instrument to the manifolded pressure taps installed in the outlet plenum. Connect the other side of the instrument to the manifolded pressure taps located in either the inlet plenum or incorporated within the air damper box. If an inlet plenum or inlet damper box are not used, leave the inlet side of the differential pressure instrument open to the surrounding atmosphere. For non-ducted systems that are tested with multiple outlet plenums, measure the static pressure within each outlet plenum relative to the surrounding atmosphere.

2.5.4 Test set-up on the outlet side of the indoor coil. a. Install an interconnecting duct between the outlet plenum described in section 2.4.1 and the airflow measuring apparatus described below in section 2.6. The cross-sectional flow area of the interconnecting duct must be equal to or greater than the flow area of the outlet plenum or the common duct used when testing non-ducted units having multiple indoor coils. If needed, use adaptor plates or transition duct sections to allow the connections. To minimize leakage, tape joints within the interconnecting duct (and the outlet plenum). Construct or insulate the entire flow section with thermal insulation having a nominal overall resistance (R-value) of at least 19 hr·ft^2·°F/Btu.

b. Install a grid(s) of dry-bulb temperature sensors inside the interconnecting duct. Also, install an air sampling device, or the sensor(s) used to measure the water vapor content of the outlet air, inside the interconnecting duct. Locate the dry-bulb temperature grid(s) upstream of the air sampling device (or the in-duct sensor(s) used to measure the water vapor content of the outlet air). Air that circulates through an air sampling device and past a remote water-vapor-content sensor(s) must be returned to the interconnecting duct at a point:

(1) Downstream of the air sampling device;

(2) Upstream of the outlet air damper box, if installed; and

(3) Upstream of the section 2.6 airflow measuring apparatus.

2.5.4.1 Outlet air damper box placement and requirements. If using an outlet air damper box (see section 2.5), install it within the interconnecting duct at a location downstream of the location where air from the sampling device is reintroduced or downstream of the in-duct sensor that measures water vapor content of the outlet air. The leakage rate from the combination of the outlet plenum, the closed damper, and the duct section that connects these two components must not exceed 20 cubic feet per minute when a negative pressure of 1 inch of water column is maintained at the plenum's inlet.

2.5.4.2 Procedures to minimize temperature maldistribution. Use these procedures if necessary to correct temperature maldistributions. Install a mixing device(s) upstream of the outlet air, dry-bulb temperature grid (but downstream of the outlet plenum static pressure taps). Use a perforated screen located between the mixing device and the dry-bulb temperature grid, with a maximum open area of 40 percent. One or both items should help to meet the maximum outlet air temperature distribution specified in section 3.1.8. Mixing devices are described in sections 6.3—6.5 of ASHRAE Standard 41.1–86 (RA 01) (incorporated by reference, see § 430.22) and section 5.2.2 of ASHRAE Standard 41.2–87 (RA 92) (incorporated by reference, see § 430.22).

2.5.5 Dry bulb temperature measurement. a. Measure dry bulb temperatures as specified in sections 4, 5, 6.1–6.10, 9, 10, and 11 of ASHRAE Standard 41.1–86 (RA 01) (incorporated by reference, see § 430.22). The transient testing requirements cited in section 4.3 of ASHRAE Standard 41.1–86 (RA 01) (incorporated by reference, see § 430.22) apply if conducting a cyclic or frost accumulation test.

b. Distribute the sensors of a dry-bulb temperature grid over the entire flow area. The required minimum is 9 sensors per grid.

2.5.6 Water vapor content measurement. Determine water vapor content by measuring dry-bulb temperature combined with the air wet-bulb temperature, dew point temperature, or relative humidity. If used, construct and apply wet-bulb temperature sensors as specified in sections 4, 5, 6, 9, 10, and 11 of ASHRAE Standard 41.1–86 (RA 01) (incorporated by reference, see § 430.22). As specified in ASHRAE 41.1–86 (RA 01) (incorporated by reference, see § 430.22), the temperature sensor (wick removed) must be accurate to within ±0.2 °F. If used, apply dew point hygrometers as specified in sections 5 and 8 of ASHRAE Standard 41.6–94 (RA 01) (incorporated by reference, see § 430.22). The dew point hygrometers must be accurate to within ±0.4 °F when operated at conditions that result in the evaluation of dew points above 35 °F. If used, a relative humidity (RH) meter must be accurate to within ±0.7% RH. Other means to determine the psychrometric state of air may be used as long as the measurement accuracy is equivalent to or better than the accuracy achieved from using a wet-bulb temperature sensor that meets the above specifications.

2.5.7 Air damper box performance requirements. If used (see section 2.5), the air damper box(es) must be capable of being completely opened or completely closed within 10 seconds for each action.

2.6 Airflow measuring apparatus. a. Fabricate and operate an Air Flow Measuring Apparatus as specified in section 6.6 of ASHRAE Standard 116–95 (incorporated by reference, see § 430.22). Refer to Figure 12 of ASHRAE Standard 51–99/AMCA Standard 210–99 (incorporated by reference, see § 430.22) or Figure 14 of ASHRAE Standard 41.2–87 (RA 92) (incorporated by reference, see § 430.22) for guidance on placing the static pressure taps and positioning the diffusion baffle (settling means) relative to the chamber inlet.

b. Connect the airflow measuring apparatus to the interconnecting duct section described in section 2.5.4. See sections 6.1.1, 6.1.2, and 6.1.4, and Figures 1, 2, and 4 of ASHRAE Standard 37–88 (incorporated by reference, see § 430.22), and Figures D1, D2, and D4 of ARI Standard 210/240–2003 (incorporated by reference, see § 430.22) for illustrative examples of how the test apparatus may be applied within a complete laboratory set-up. Instead of following one of these examples, an alternative set-up may be used to handle the air leaving the airflow measuring apparatus and to supply properly conditioned air to the test unit's inlet. The alternative set-up, however, must not interfere with the prescribed means for measuring airflow rate, inlet and outlet air temperatures, inlet and outlet water vapor contents, and external static pressures, nor create abnormal conditions surrounding the test unit. (Note: Do not use an enclosure as described in section 6.1.3 of ASHRAE Standard 37–88 (incorporated by reference, see § 430.22) when testing triple-split units.)

2.7 Electrical voltage supply. Perform all tests at the voltage specified in section 6.1.3.2 of ARI Standard 210/240–2003 (incorporated by reference, see § 430.22) for ''Standard Rating Tests.'' Measure the supply voltage at the terminals on the test unit using a volt meter that provides a reading that is accurate to within ±1.0 percent of the measured quantity.

2.8 Electrical power and energy measurements. a. Use an integrating power (watt-hour) measuring system to determine the electrical energy or average electrical power supplied to all components of the air conditioner or heat pump (including auxiliary components such as controls, transformers, crankcase heater, integral condensate pump on non-ducted indoor units, etc.). The watt-hour measuring system must give readings that are accurate to within ±0.5 percent. For cyclic tests, this accuracy is required during both the ON and OFF cycles. Use either two different scales on the same watt-hour meter or two separate watt-hour meters. Activate the scale or meter having the lower power rating within 15 seconds after beginning an OFF cycle. Activate the scale or meter having the higher power rating active within 15 seconds prior to beginning an ON cycle. For ducted units tested with a fan installed, the ON cycle lasts from compressor ON to indoor fan OFF. For ducted units tested without an indoor fan installed, the ON cycle lasts from compressor ON to compressor OFF. For non-ducted units, the ON cycle lasts from indoor fan ON to indoor fan OFF. When testing air conditioners and heat pumps having a variable-speed compressor, avoid using an induction watt/watt-hour meter.

b. When performing section 3.5 and/or 3.8 cyclic tests on non-ducted units, provide instrumentation to determine the average electrical power consumption of the indoor fan motor to within ±1.0 percent. If required according to sections 3.3, 3.4, 3.7, 3.9.1, and/or 3.10, this same instrumentation requirement applies when testing air conditioners and heat pumps having a variable-speed constant-air-volume-rate indoor fan or a variable-speed, variable-air-volume-rate indoor fan.

2.9 Time measurements. Make elapsed time measurements using an instrument that yields readings accurate to within ±0.2 percent.

2.10 Test apparatus for the secondary space conditioning capacity measurement. For all tests, use the Indoor Air Enthalpy Method to measure the unit's capacity. This method uses the test set-up specified in sections 2.4 to 2.6. In addition, for all steady-state tests, conduct a second, independent measurement of capacity as described in section 3.1.1. For split systems, use one of the following secondary measurement methods: Outdoor Air Enthalpy Method, Compressor Calibration Method, or Refrigerant Enthalpy Method. For single packaged units, use either the Outdoor Air Enthalpy Method or the Compressor Calibration Method as the secondary measurement.

2.10.1 Outdoor Air Enthalpy Method. a. To make a secondary measurement of indoor space conditioning capacity using the Outdoor Air Enthalpy Method, do the following:

(1) Measure the electrical power consumption of the test unit;

(2) Measure the air-side capacity at the outdoor coil; and

(3) Apply a heat balance on the refrigerant cycle.

b. The test apparatus required for the Outdoor Air Enthalpy Method is a subset of the apparatus used for the Indoor Air Enthalpy Method. Required apparatus includes the following:

(1) An outlet plenum containing static pressure taps (sections 2.4, 2.4.1, and 2.5.3),

(2) An airflow measuring apparatus (section 2.6),

(3) A duct section that connects these two components and itself contains the instrumentation for measuring the dry-bulb temperature and water vapor content of the air leaving the outdoor coil (sections 2.5.4, 2.5.5, and 2.5.6), and

(4) On the inlet side, a sampling device and optional temperature grid (sections 2.5 and 2.5.2).

c. During the preliminary tests described in sections 3.11.1 and 3.11.1.1, measure the evaporator and condenser temperatures or pressures. On both the outdoor coil and the indoor coil, solder a thermocouple onto a return bend located at or near the midpoint of each coil or at points not affected by vapor superheat or liquid subcooling. Alternatively, if the test unit is not sensitive to the refrigerant charge, connect pressure gages to the access valves or to ports created from tapping into the suction and discharge lines. Use this alternative approach when testing a unit charged with a zeotropic refrigerant having a temperature glide in excess of 1 °F at the specified test conditions.

2.10.2 Compressor Calibration Method. Measure refrigerant pressures and temperatures to determine the evaporator superheat and the enthalpy of the refrigerant that enters and exits the indoor coil. Determine refrigerant flow rate or, when the superheat of the refrigerant leaving the evaporator is less than 5 °F, total capacity from separate calibration tests conducted under identical operating conditions. When using this method, install instrumentation, measure refrigerant properties, and adjust the refrigerant charge according to section 7.4.2 of ASHRAE Standard 37–88 (incorporated by reference, see § 430.22). Use refrigerant temperature and pressure measuring instruments that meet the specifications given in sections 5.1.1 and 5.2 of ASHRAE Standard 37–88 (incorporated by reference, see § 430.22).

2.10.3 Refrigerant Enthalpy Method. For this method, calculate space conditioning capacity by determining the refrigerant enthalpy change for the indoor coil and directly measuring the refrigerant flow rate. Use section 7.6.2 of ASHRAE Standard 37–88 (incorporated by reference, see § 430.22) for the requirements for this method, including the additional instrumentation requirements, and information on placing the flow meter and a sight glass. Use refrigerant temperature, pressure, and flow measuring instruments that meet the specifications given in sections 5.1.1, 5.2, and 5.5.1 of ASHRAE Standard 37–88 (incorporated by reference, see § 430.22).

2.11 Measurement of test room ambient conditions. a. If using a test set-up where air is ducted directly from the conditioning apparatus to the indoor coil inlet (see Figure 2, Loop Air-Enthalpy Test Method Arrangement, of ASHRAE Standard 37–88 (incorporated by reference, see § 430.22)), add instrumentation to permit measurement of the indoor test room dry-bulb temperature.

b. If the Outdoor Air Enthalpy Method is not used, add instrumentation to measure the dry-bulb temperature and the water vapor content of the air entering the outdoor coil. If an air sampling device is used, construct and apply the device as per section 6 of ASHRAE Standard 41.1–86 (RA 01) (incorporated by reference, see § 430.22). Take steps (*e.g.*, add or re-position a lab circulating fan), as needed, to minimize the magnitude of the temperature distribution non-uniformity. Position any fan in the outdoor test room while trying to keep air velocities in the vicinity of the test unit below 500 feet per minute.

c. Measure dry bulb temperatures as specified in sections 4, 5, 6.1–6.10, 9, 10, and 11 of ASHRAE Standard 41.1–86 (RA 01) (incorporated by reference, see § 430.22). Measure water vapor content as stated above in section 2.5.6.

2.12 Measurement of indoor fan speed. When required, measure fan speed using a revolution counter, tachometer, or stroboscope that gives readings accurate to within ±1.0 percent.

2.13 Measurement of barometric pressure. Determine the average barometric pressure during each test. Use an instrument that meets the requirements specified in section 5.2 of ASHRAE Standard 37–88 (incorporated by reference, see § 430.22).

3. Testing Procedures

3.1 General Requirements. If, during the testing process, an equipment set-up adjustment is made that would alter the performance of the unit when conducting an already completed test, then repeat all tests affected by the adjustment. For cyclic tests, instead of maintaining an air volume rate, for each airflow nozzle, maintain the static pressure difference or velocity pressure during an ON period at the same pressure difference or velocity pressure as measured during the steady-state test conducted at the same test conditions.

3.1.1 Primary and secondary test methods. For all tests, use the Indoor Air Enthalpy Method test apparatus to determine the unit's space conditioning capacity. The procedure and data collected, however, differ slightly depending upon whether the test is a steady-state test, a cyclic test, or a frost accumulation test. The following sections described these differences. For all steady-state tests (*i.e.*, the A, A_2, A_1, B, B_2, B_1, C, C_1, EV, F_1, G_1, $H0_1$, H_1, $H1_2$, $H1_1$, HI_N, H_3, $H3_2$, and $H3_1$ Tests), in addition, use one of the acceptable secondary methods specified in section 2.10 to determine indoor space conditioning capacity. Calculate this secondary check of capacity according to section 3.11. The two capacity measurements must agree to within 6 percent to constitute a valid test. For this capacity comparison, use the Indoor Air Enthalpy Method capacity that is calculated in section 7.3 of ASHRAE Standard 37–88 (incorporated by reference, see § 430.22) (and do not make the after-test fan heat adjustments described in sections 3.3, 3.4, 3.7, and 3.10 of this Appendix). However, include the appropriate section 3.3 to 3.5 and 3.7 to 3.10 fan heat adjustments within the Indoor Air Enthalpy Method capacities used for the section 4 seasonal calculations.

3.1.2 Manufacturer-provided equipment overrides. Where needed, the manufacturer must provide a means for overriding the controls of the test unit so that the compressor(s) operates at the specified speed or capacity and the indoor fan operates at the specified speed or delivers the specified air volume rate.

3.1.3 Airflow through the outdoor coil. For all tests, meet the requirements given in section 6.1.3.4 of ARI Standard 210/240–2003 (incorporated by reference, see § 430.22) when obtaining the airflow through the outdoor coil.

3.1.4 Airflow through the indoor coil.

3.1.4.1 Cooling Certified Air Volume Rate.

3.1.4.1.1 Cooling Certified Air Volume Rate for Ducted Units. The manufacturer must specify the Cooling Certified Air Volume Rate. Use this value as long as the following two requirements are satisfied. First, when conducting the A or A_2 Test (exclusively), the measured air volume rate, when divided by the measured indoor air-side total cooling capacity, must not exceed 37.5 cubic feet per minute of standard air (scfm) per 1000 Btu/h. If this ratio is exceeded, reduce the air volume rate until this ratio is equaled. Use this reduced air volume rate for all tests that call for using the Cooling Certified Air Volume Rate. The second requirement is as follows:

a. For ducted units that are tested with a fixed-speed, multi-speed, or variable-speed variable-air-volume-rate indoor fan installed. For the A or A_2 Test (exclusively), the measured external static pressure must be equal to or greater than the applicable minimum external static pressure cited in Table 2. If the Table 2 minimum is not equaled or exceeded, incrementally change the set-up of the indoor fan (*e.g.*, fan motor pin settings, fan motor speed) until the Table 2 requirement is met while maintaining the same air volume rate. If the indoor fan set-up changes cannot provide the minimum external static, then reduce the air volume rate until the correct Table 2 minimum is equaled. For the last scenario, use the reduced air volume rate for all tests that require the Cooling Certified Air Volume Rate.

b. For ducted units that are tested with a constant-air-volume-rate indoor fan installed. For all tests that specify the Cooling Certified Air Volume Rate, obtain an external static pressure as close to (but not less than) the applicable Table 2 value that does not cause instability or an automatic shutdown of the indoor blower.

c. For ducted units that are tested without an indoor fan installed. For the A or A_2 Test, (exclusively), the pressure drop across the indoor coil assembly must not exceed 0.30 inches of water. If this pressure drop is exceeded, reduce the air volume rate until the measured pressure drop equals the specified maximum. Use this reduced air volume rate for all tests that require the Cooling Certified Air Volume Rate.

TABLE 2.—MINIMUM EXTERNAL STATIC PRESSURE FOR DUCTED SYSTEMS TESTED WITH AN INDOOR FAN INSTALLED

Rated Cooling [1] or Heating [2] Capacity (Btu/h)	Minimum External Resistance [3] (Inches of Water)
Up Thru 28,800	0.10
29,000 to 42,500	0.15
43,000 and Above	0.20

[1] For air conditioners and heat pumps, the value cited by the manufacturer in published literature for the unit's capacity when operated at the A or A_2 Test conditions.

[2] For heating-only heat pumps, the value the manufacturer cites in published literature for the unit's capacity when operated at the H1 or $H1_2$ Test conditions.

[3] For ducted units tested without an air filter installed, increase the applicable tabular value by 0.08 inches of water.

3.1.4.1.2 Cooling Certified Air Volume Rate for Non-ducted Units. For non-ducted units, the Cooling Certified Air Volume Rate is the air volume rate that results during each test when the unit is operated at an external static pressure of zero inches of water.

3.1.4.2 Cooling Minimum Air Volume Rate. a. For ducted units that regulate the speed (as opposed to the cfm) of the indoor fan,

$$\text{Cooling Minimum Air Vol. Rate} = \text{Cooling Certified Air Vol. Rate} \times \frac{\text{Cooling Minimum Fan Speed}}{\text{A}_2 \text{ Test Fan Speed}},$$

where ''Cooling Minimum Fan Speed'' corresponds to the fan speed used when operating at low compressor capacity (two-capacity system), the fan speed used when operating at the minimum compressor speed (variable-speed system), or the lowest fan speed used when cooling (single-speed compressor and a variable-speed variable-air-volume-rate indoor fan). For such systems, obtain the Cooling Minimum Air Volume Rate regardless of the external static pressure.

b. For ducted units that regulate the air volume rate provided by the indoor fan, the manufacturer must specify the Cooling Minimum Air Volume Rate. For such systems, conduct all tests that specify the Cooling Minimum Air Volume Rate—(*i.e.*, the A_1, B_1, C_1, F_1, and G_1 Tests)—at an external static pressure that does not cause instability or an automatic shutdown of the indoor blower while being as close to, but not less than,

$$A_1,\ B_1,\ C_1,\ F_1,\ \&\ G_1 \text{ Test } \Delta P_{st} = \Delta P_{st,A_2} \times \left[\frac{\text{Cooling Minimum Air Volume Rate}}{\text{Cooling Certified Air Volume Rate}}\right]^2,$$

where $\Delta P_{st,A_2}$ is the applicable Table 2 minimum external static pressure that was targeted during the A_2 (and B_2) Test.

c. For ducted two-capacity units that are tested without an indoor fan installed, the Cooling Minimum Air Volume Rate is the higher of (1) the rate specified by the manufacturer or (2) 75 percent of the Cooling Certified Air Volume Rate. During the laboratory tests on a coil-only (fanless) unit, obtain this Cooling Minimum Air Volume Rate regardless of the pressure drop across the indoor coil assembly.

d. For non-ducted units, the Cooling Minimum Air Volume Rate is the air volume rate that results during each test when the unit operates at an external static pressure of zero inches of water and at the indoor fan setting used at low compressor capacity (two-capacity system) or minimum compressor speed (variable-speed system). For units having a single-speed compressor and a variable-speed variable-air-volume-rate indoor fan, use the lowest fan setting allowed for cooling.

3.1.4.3 Cooling Intermediate Air Volume Rate. a. For ducted units that regulate the speed of the indoor fan,

$$\text{Cooling Intermediate Air Volume Rate} = \text{Cooling Certified Air Volume Rate} \times \frac{E_v \text{ Test Fan Speed}}{A_2 \text{ Test Fan Speed}}.$$

For such units, obtain the Cooling Intermediate Air Volume Rate regardless of the external static pressure.

b. For ducted units that regulate the air volume rate provided by the indoor fan, the manufacturer must specify the Cooling Intermediate Air Volume Rate. For such systems, conduct the E_V Test at an external static pressure that does not cause instability or an automatic shutdown of the indoor blower while being as close to, but not less than,

$$E_v \text{ Test } \Delta P_{st} = \Delta P_{st,A_2} \times \left[\frac{\text{Cooling Intermediate Air Volume Rate}}{\text{Cooling Certified Air Volume Rate}}\right]^2,$$

where $\Delta P_{st,A_2}$ is the applicable Table 2 minimum external static pressure that was targeted during the A_2 (and B_2) Test.

c. For non-ducted units, the Cooling Intermediate Air Volume Rate is the air volume rate that results when the unit operates at an external static pressure of zero inches of water and at the fan speed selected by the controls of the unit for the E_V Test conditions.

3.1.4.4 Heating Certified Air Volume Rate.

3.1.4.4.1 Ducted heat pumps where the Heating and Cooling Certified Air Volume Rates are the same. a. Use the Cooling Certified Air Volume Rate as the Heating Certified Air Volume Rate for:

1. Ducted heat pumps that operate at the same indoor fan speed during both the A (or A_2) and the H1 (or $H1_2$) Tests;

2. Ducted heat pumps that regulate fan speed to deliver the same constant air volume rate during both the A (or A_2) and the H1 (or $H1_2$) Tests; and

3. Ducted heat pumps that are tested without an indoor fan installed (except two-capacity northern heat pumps that are tested only at low capacity cooling—see 3.1.4.4.2).

b. For heat pumps that meet the above criteria ''1'' and ''3,'' no minimum requirements apply to the measured external or internal, respectively, static pressure. For heat pumps that meet the above criterion ''2,'' test at an external static pressure that does not cause instability or an automatic shutdown of the indoor blower while being as close to, but not less than, the same Table 2 minimum external static pressure as was specified for the A (or A_2) cooling mode test.

3.1.4.4.2 Ducted heat pumps where the Heating and Cooling Certified Air Volume Rates are different due to indoor fan operation. a. For ducted heat pumps that regulate the speed (as opposed to the cfm) of the indoor fan,

$$\text{Heating Certified Air Volume Rate} = \text{Cooling Certified Air Volume Rate} \times \frac{\text{H1 or H1}_2 \text{ Test Fan Speed}}{\text{A or A}_2 \text{ Test Fan Speed}}.$$

For such heat pumps, obtain the Heating Certified Air Volume Rate without regard to the external static pressure.

b. For ducted heat pumps that regulate the air volume rate delivered by the indoor fan, the manufacturer must specify the Heating Certified Air Volume Rate. For such heat pumps, conduct all tests that specify the Heating Certified Air Volume Rate at an external static pressure that does not cause instability or an automatic shutdown of the indoor blower while being as close to, but not less than,

$$\text{Heating Certified } \Delta P_{st} = \text{Cooling Certified } \Delta P_{st} \times \left[\frac{\text{Heating Certified Air Volume Rate}}{\text{Cooling Certified Air Volume Rate}}\right]^2$$

where the Cooling Certified ΔP_{st} is the applicable Table 2 minimum external static pressure that was specified for the A or A_2 Test.

c. When testing ducted, two-capacity northern heat pumps (see Definition 1.46), use the appropriate approach of the above two cases for units that are tested with an indoor fan installed. For coil-only (fanless) northern heat pumps, the Heating Certified Air Volume Rate is the lesser of the rate specified by the manufacturer or 133 percent of the Cooling Certified Air Volume Rate. For this latter case, obtain the Heating Certified Air Volume Rate regardless of the pressure drop across the indoor coil assembly.

3.1.4.4.3 Ducted heating-only heat pumps. The manufacturer must specify the Heating Certified Air Volume Rate. Use this value when the following two requirements are satisfied. First, when conducting the H1 or $H1_2$ Test (exclusively), the measured air volume rate, when divided by the measured indoor air-side total heating capacity, must not exceed 37.5 cubic feet per minute of standard air (scfm) per 1000 Btu/h. If this ratio is exceeded, reduce the air volume rate until this ratio is equaled. Use this reduced air volume rate for all tests of heating-only heat pumps that call for the Heating Certified Air Volume Rate. The second requirement is as follows:

a. For heating-only heat pumps that are tested with a fixed-speed, multi-speed, or variable-speed variable-air-volume-rate indoor fan installed. For the H1 or $H1_2$ Test (exclusively), the measured external static pressure must be equal to or greater than the Table 2 minimum external static pressure that applies given the heating-only heat pump's rated heating capacity. If the Table 2 minimum is not equaled or exceeded, incrementally change the set-up of the indoor fan until the Table 2 requirement is met while maintaining the same air volume rate. If the indoor fan set-up changes cannot provide the necessary external static pressure, then reduce the air volume rate until the correct Table 2 minimum is equaled. For the last scenario, use the reduced air volume rate for all tests that require the Heating Certified Air Volume Rate.

b. For ducted heating-only heat pumps having a constant-air-volume-rate indoor fan. For all tests that specify the Heating Certified Air Volume Rate, obtain an external static pressure that does not cause instability or an automatic shutdown of the indoor blower while being as close to, but not less than, the applicable Table 2 minimum.

c. For ducted heating-only heat pumps that are tested without an indoor fan installed. For the H1 or $H1_2$ Test, (exclusively), the pressure drop across the indoor coil assembly must not exceed 0.30 inches of water. If this pressure drop is exceeded, reduce the air volume rate until the measured pressure drop equals the specified maximum. Use this reduced air volume rate for all tests that require the Heating Certified Air Volume Rate.

3.1.4.4.4 Non-ducted heat pumps, including non-ducted heating-only heat pumps. For non-ducted heat pumps, the Heating Certified Air Volume Rate is the air volume rate that results during each test when the unit operates at an external static pressure of zero inches of water.

3.1.4.5 Heating Minimum Air Volume Rate. a. For ducted heat pumps that regulate the speed (as opposed to the cfm) of the indoor fan,

$$\text{Heating Minimum Air Volume Rate} = \text{Heating Certified Air Volume Rate} \times \frac{\text{Heating Minimum Fan Speed}}{\text{H1}_2 \text{ Test Fan Speed}}$$

where "Heating Minimum Fan Speed" corresponds to the fan speed used when operating at low compressor capacity (two-capacity system), the lowest fan speed used at any time when operating at the minimum compressor speed (variable-speed system), or the lowest fan speed used when heating (single-speed compressor and a variable-speed variable-air-volume-rate indoor fan). For such heat pumps, obtain the Heating Minimum Air Volume Rate without regard to the external static pressure.

b. For ducted heat pumps that regulate the air volume rate delivered by the indoor fan, the manufacturer must specify the Heating Minimum Air Volume Rate. For such heat pumps, conduct all tests that specify the Heating Minimum Air Volume Rate—(*i.e.*, the $H0_1$, $H1_1$, $H2_1$, and $H3_1$ Tests)—at an external static pressure that does not cause instability or an automatic shutdown of the indoor blower while being as close to, but not less than,

$$\text{H0}_1,\ \text{H1}_1,\ \text{H2}_1,\ \text{H3}_1,\ \text{Test } \Delta P_{st} = \Delta P_{st,H1_2} \times \left[\frac{\text{Htg Minimum Air Vol. Rate}}{\text{Htg Certified Air Vol. Rate}}\right]^2,$$

where $\Delta P_{st,H1_2}$ is the minimum external static pressure that was targeted during the $H1_2$ Test.

c. For ducted two-capacity northern heat pumps that are tested with an indoor fan installed, use the appropriate approach of the above two cases.

d. For ducted two-capacity heat pumps that are tested without an indoor fan installed, use the Cooling Minimum Air Volume Rate as the Heating Minimum Air Volume Rate. For ducted two-capacity northern heat pumps that are tested without an indoor fan installed, use the Cooling Certified Air Volume Rate as the Heating Minimum Air Volume Rate. For ducted two-capacity heating-only heat pumps that are tested without an indoor fan installed, the Heating Minimum Air Volume Rate is the higher of the rate specified by the manufacturer or 75 percent of the Heating Certified Air Volume Rate. During the laboratory tests on a coil-only (fanless) unit, obtain the Heating Minimum Air Volume Rate without regard to the pressure drop across the indoor coil assembly.

e. For non-ducted heat pumps, the Heating Minimum Air Volume Rate is the air volume rate that results during each test when the unit operates at an external static pressure of zero inches of water and at the indoor fan setting used at low compressor capacity (two-capacity system) or minimum compressor speed (variable-speed system). For units having a single-speed compressor and a variable-speed, variable-air-volume-rate indoor fan, use the lowest fan setting allowed for heating.

3.1.4.6 Heating Intermediate Air Volume Rate. a. For ducted heat pumps that regulate the speed of the indoor fan,

$$\text{Heating Intermediate Air Volume Rate} = \text{Heating Certified Air Volume Rate} \times \frac{H2_V \text{ Test Fan Speed}}{H1_2 \text{ Test Fan Speed}}.$$

For such heat pumps, obtain the Heating Intermediate Air Volume Rate without regard to the external static pressure.

b. For ducted heat pumps that regulate the air volume rate delivered by the indoor fan, the manufacturer must specify the Heating Intermediate Air Volume Rate. For such heat pumps, conduct the $H2_V$ Test at an external static pressure that does not cause instability or an automatic shutdown of the indoor blower while being as close to, but not less than,

$$H2_V \text{ Test } \Delta P_{st} = \Delta P_{st,H1_2} \times \left[\frac{\text{Heating Intermediate Air Volume Rate}}{\text{Heating Certified Air Volume Rate}}\right]^2,$$

where $\Delta P_{st,H1_2}$

is the minimum external static pressure that was specified for the $H1_2$ Test.

c. For non-ducted heat pumps, the Heating Intermediate Air Volume Rate is the air volume rate that results when the heat pump operates at an external static pressure of zero inches of water and at the fan speed selected by the controls of the unit for the $H2_V$ Test conditions.

3.1.4.7 Heating Nominal Air Volume Rate. Except for the noted changes, determine the Heating Nominal Air Volume Rate using the approach described in section 3.1.4.6. Required changes include substituting ''$H1_N$ Test'' for $H2_V$ Test'' within the first section 3.1.4.6 equation, substituting ''$H1_N$ Test ΔP_{st}'' for ''$H2_V$ Test ΔP_{st}'' in the second section 3.1.4.6 equation, substituting ''$H1_N$ Test'' for each ''$H2_V$ Test'', and substituting ''Heating Nominal Air Volume Rate'' for each ''Heating Intermediate Air Volume Rate.''

$$\text{Heating Nominal Air Volume Rate} = \text{Heating Certified Air Volume Rate} \times \frac{H1_N \text{ Test Fan Speed}}{H1_2 \text{ Test Fan Speed}}.$$

$$H1_N \text{ Test } \Delta P_{st} = \Delta P_{st,H1_2} \times \left[\frac{\text{Heating Nominal Air Volume Rate}}{\text{Heating Certified Air Volume Rate}}\right]^2.$$

3.1.5 Indoor test room requirement when the air surrounding the indoor unit is not supplied from the same source as the air entering the indoor unit. If using a test set-up where air is ducted directly from the air reconditioning apparatus to the indoor coil inlet (see Figure 2, Loop Air-Enthalpy Test Method Arrangement, of ASHRAE Standard 37–88) (incorporated by reference, see § 430.22), maintain the dry bulb temperature within the test room within ±5.0 °F of the applicable sections 3.2 and 3.6 dry bulb temperature test condition for the air entering the indoor unit.

3.1.6 Air volume rate calculations. For all steady-state tests and for frost accumulation ($H2$, $H2_1$, $H2_2$, $H2_V$) tests, calculate the air volume rate through the indoor coil as specified in sections 7.8.3.1 and 7.8.3.2 of ASHRAE Standard 37–88 (incorporated by reference, see § 430.22). When using the Outdoor Air Enthalpy Method, follow sections 7.8.3.1 and 7.8.3.2 to calculate the air volume rate through the outdoor coil. To express air volume rates in terms of standard air, use:

$$\overline{\dot{V}}_s = \frac{\overline{\dot{V}}_{mx}}{0.075 \frac{lbm_{da}}{ft^3} \cdot v_n' \cdot [1 + W_n]} = \frac{\overline{\dot{V}}_{mx}}{0.075 \frac{lbm_{da}}{ft^3} \cdot v_n} \qquad (3\text{-}1)$$

where,

$\overline{\dot{V}}_s$ = air volume rate of standard (dry) air, $(ft^3/min)_{da}$

$\overline{\dot{V}}_{mx}$ = air volume rate of the air-water vapor mixture, $(ft^3/min)_{mx}$

v_n' = specific volume of air-water vapor mixture at the nozzle, ft^3 per lbm of the air-water vapor mixture

W_n = humidity ratio at the nozzle, lbm of water vapor per lbm of dry air

0.075 = the density associated with standard (dry) air, (lbm/ft^3)

v_n = specific volume of the dry air portion of the mixture evaluated at the dry-bulb temperature, vapor content, and barometric pressure existing at the nozzle, ft^3 per lbm of dry air.

3.1.7 Test sequence. When testing a ducted unit (except if a heating-only heat pump), conduct the A or A_2 Test first to establish the Cooling Certified Air Volume Rate. For ducted heat pumps where the Heating and Cooling Certified Air Volume Rates are different, make the first heating mode test one that requires the Heating Certified Air Volume Rate. For ducted heating-only heat pumps, conduct the H1 or $H1_2$ Test first to establish the Heating Certified Air Volume Rate. When conducting an optional cyclic test, always conduct it immediately after the steady-state test that requires the same test conditions. For variable-speed systems, the first test using the Cooling Minimum Air Volume Rate should precede the E_V Test if one expects to adjust the indoor fan control options when preparing for the first Minimum Air Volume Rate test. Under the same circumstances, the first test using the Heating Minimum Air Volume Rate should precede the $H2_V$ Test. The test laboratory makes all other decisions on the test sequence.

3.1.8 Requirement for the air temperature distribution leaving the indoor coil. For at least the first cooling mode test and the first heating mode test, monitor the temperature distribution of the air leaving the indoor coil using the grid of individual sensors described in sections 2.5 and 2.5.4. For the 30-minute data collection interval used to determine capacity, the maximum spread among the outlet dry bulb temperatures from any data sampling must not exceed 1.5 °F. Install the mixing devices described in section 2.5.4.2 to minimize the temperature spread.

3.1.9 Control of auxiliary resistive heating elements. Except as noted, disable heat pump resistance elements used for heating indoor air at all times, including during defrost cycles and if they are normally regulated by a heat comfort controller. For heat pumps equipped with a heat comfort controller, enable the heat pump resistance elements only during the below-described, short test. For single-speed heat pumps covered under section 3.6.1, the short test follows the H1 or, if conducted, the H1C Test. For two-capacity heat pumps and heat pumps covered under section 3.6.2, the short test follows the $H1_2$ Test. Set the heat comfort controller to provide the maximum supply air temperature. With the heat pump operating and while maintaining the Heating Certified Air Volume Rate, measure the temperature of the air leaving the indoor-side beginning 5 minutes after activating the heat comfort controller. Sample the outlet dry-bulb temperature at regular intervals that span 5 minutes or less. Collect data for 10 minutes, obtaining at least 3 samples. Calculate the average outlet temperature over the 10-minute interval, T_{CC}.

3.2 Cooling mode tests for different types of air conditioners and heat pumps.

3.2.1 Tests for a unit having a single-speed compressor that is tested with a fixed-speed indoor fan installed, with a constant-air-volume-rate indoor fan installed, or with no indoor fan installed. Conduct two steady-state wet coil tests, the A and B Tests. Use the two optional dry-coil tests, the steady-state C Test and the cyclic D Test, to determine the cooling mode cyclic degradation coefficient, C_D^c. If the two optional tests are not conducted, assign C_D^c the default value of 0.25. Table 3 specifies test conditions for these four tests.

TABLE 3.—COOLING MODE TEST CONDITIONS FOR UNITS HAVING A SINGLE-SPEED COMPRESSOR AND A FIXED-SPEED INDOOR FAN, A CONSTANT AIR VOLUME RATE INDOOR FAN, OR NO INDOOR FAN

Test description	Air entering indoor unit temperature (°F)		Air entering outdoor unit temperature (°F)		Cooling air volume rate
	Dry bulb	Wet bulb	Dry bulb	Wet bulb	
A Test—required (steady, wet coil)	80	67	95	[1] 75	Cooling certified [2]
B Test—required (steady, wet coil)	80	67	82	[1] 65	Cooling certified [2]
C Test—optional (steady, dry coil)	80	([3])	82	....	Cooling certified [2]
D Test—optional (cyclic, dry coil)	80	([3])	82	....	([4])

[1] The specified test condition only applies if the unit rejects condensate to the outdoor coil.
[2] Defined in section 3.1.4.1.
[3] The entering air must have a low enough moisture content so no condensate forms on the indoor coil. (It is recommended that an indoor wet-bulb temperature of 57 °F or less be used.)
[4] Maintain the airflow nozzles static pressure difference or velocity pressure during the ON period at the same pressure difference or velocity pressure as measured during the C Test.

3.2.2 Tests for a unit having a single-speed compressor and a variable-speed variable-air-volume-rate indoor fan installed.

3.2.2.1 Indoor fan capacity modulation that correlates with the outdoor dry bulb temperature. Conduct four steady-state wet coil tests: The A_2, A_1 , B_2, and B_1 Tests. Use the two optional dry-coil tests, the steady-state C_1 Test and the cyclic D_1 Test, to determine the cooling mode cyclic degradation coefficient, C_D^c. If the two optional tests are not conducted, assign C_D^c the default value of 0.25. Table 4 specifies test conditions for these six tests.

3.2.2.2 Indoor fan capacity modulation based on adjusting the sensible to total (S/T) cooling capacity ratio. The testing requirements are the same as specified in section 3.2.1 and Table 3. Use a Cooling Certified Air Volume Rate that represents a normal residential installation. If performed, conduct the steady-state C Test and the cyclic D Test with the unit operating in the same S/T capacity control mode as used for the B Test.

TABLE 4.—COOLING MODE TEST CONDITIONS FOR UNITS HAVING A SINGLE-SPEED COMPRESSOR AND A VARIABLE AIR VOLUME RATE INDOOR FAN THAT CORRELATES WITH THE OUTDOOR DRY BULB TEMPERATURE (SEC. 3.2.2.1)

Test description	Air entering indoor unit temperature (°F)		Air entering outdoor unit temperature (°F)		Cooling air volume rate
	Dry bulb	Wet bulb	Dry bulb	Wet bulb	
A_2 Test—required (steady, wet coil)	80	67	95	[1] 75	Cooling certified [2]
A_1 Test—required (steady, wet coil)	80	67	95	[1] 75	Cooling minimum [3]
B_2 Test—required (steady, wet coil)	80	67	82	[1] 65	Cooling certified [2]
B_1 Test—required (steady, wet coil)	80	67	82	[1] 65	Cooling minimum [3]
C_1 Test [4]—optional (steady, dry coil)	80	([4])	82	....	Cooling minimum [3]
D_1 Test [4]—optional (cyclic, dry coil)	80	([4])	82	....	([5])

[1] The specified test condition only applies if the unit rejects condensate to the outdoor coil.
[2] Defined in section 3.1.4.1.
[3] Defined in section 3.1.4.2.
[4] The entering air must have a low enough moisture content so no condensate forms on the indoor coil. (It is recommended that an indoor wet-bulb temperature of 57 °F or less be used.)
[5] Maintain the airflow nozzles static pressure difference or velocity pressure during the ON period at the same pressure difference or velocity pressure as measured during the C_1 Test.

3.2.3 Tests for a unit having a two-capacity compressor. (See Definition 1.45.) a. Conduct four steady-state wet coil tests: The A_2, A_1, B_2, and B_1 Tests. Use the two optional dry-coil tests, the steady-state C_1 Test and the cyclic D_1 Test, to determine the cooling mode cyclic degradation coefficient, C_D^c. If the two optional tests are not conducted, assign C_D^c the default value of 0.25. Table 5 specifies test conditions for these six tests.

b. For units having a variable speed indoor fan that is modulated to adjust the sensible to total (S/T) cooling capacity ratio, use Cooling Certified and Cooling Minimum Air Volume Rates that represent a normal residential installation. Additionally, if conducting the optional dry-coil tests, operate the unit in the same S/T capacity control mode as used for the B_1 Test.

c. Test two-capacity, northern heat pumps (see Definition 1.46) in the same way as a single speed heat pump with the unit operating exclusively at low compressor capacity (see section 3.2.1 and Table 3).

d. If a two-capacity air conditioner or heat pump locks out low capacity operation at outdoor temperatures that are less than 95 °F, conduct the A_1 Test using the outdoor temperature conditions listed for the F_1 Test in Table 6 rather than using the outdoor temperature conditions listed in Table 5 for the A_1 Test.

TABLE 5.—COOLING MODE TEST CONDITIONS FOR UNITS HAVING A TWO-CAPACITY COMPRESSOR

Test description	Air entering indoor unit temperature (°F)		Air entering outdoor unit temperature (°F)		Compressor capacity	Cooling air volume rate
	Dry bulb	Wet bulb	Dry bulb	Wet bulb		
A_2 Test—required (steady, wet coil)	80	67	95	[1] 75	High	Cooling Certified [2]
A_1 Test—required (steady, wet coil)	80	67	95	[1] 75	Low	Cooling Minimum [3]
B_2 Test—required (steady, wet coil)	80	67	82	[1] 65	High	Cooling Certified [2]
B_1 Test—required (steady, wet coil)	80	67	82	[1] 65	Low	Cooling Minimum [3]
C_1 Test [4]—optional (steady, dry coil)	80	([4])	82	....	Low	Cooling Minimum [3]
D_1 Test [4]—optional (cyclic, dry coil)	80	([4])	82	....	Low	([5])

[1] The specified test condition only applies if the unit rejects condensate to the outdoor coil.
[2] Defined in section 3.1.4.1.
[3] Defined in section 3.1.4.2.
[4] The entering air must have a low enough moisture content so no condensate forms on the indoor coil. (It is recommended that an indoor wet-bulb temperature of 57 °F or less be used.)
[5] Maintain the airflow nozzles static pressure difference or velocity pressure during the ON period at the same pressure difference or velocity pressure as measured during the C_1 Test.

3.2.4 Tests for a unit having a variable-speed compressor. a. Conduct five steady-state wet coil tests: The A_2, E_V, B_2, B_1, and F_1 Tests. Use the two optional dry-coil tests, the steady-state G_1 Test and the cyclic I_1 Test, to determine the cooling mode cyclic degradation coefficient, C_D^c. If the two optional tests are not conducted, assign C_D^c the default value of 0.25. Table 6 specifies test conditions for these seven tests. Determine the intermediate compressor speed cited in Table 6 using:

$$\text{Intermediate speed} = \text{Minimum speed} + \frac{\text{Maximum speed} - \text{Minimum speed}}{3}$$

where a tolerance of plus 5 percent or the next higher inverter frequency step from that calculated is allowed.

b. For units that modulate the indoor fan speed to adjust the sensible to total (S/T) cooling capacity ratio, use Cooling Certified, Cooling Intermediate, and Cooling Minimum Air Volume Rates that represent a normal residential installation. Additionally, if conducting the optional dry-coil tests, operate the unit in the same S/T capacity control mode as used for the F_1 Test.

TABLE 6.—COOLING MODE TEST CONDITIONS FOR UNITS HAVING A VARIABLE-SPEED COMPRESSOR

Test description	Air entering indoor unit Temperature (°F)		Air entering outdoor unit Temperature (°F)		Compressor speed	Cooling air volume rate
	Dry bulb	Wet bulb	Dry bulb	Wet bulb		
A_2 Test—required (steady, wet coil)	80	67	95	[1] 75	Maximum	Cooling Certified [2]
B_2 Test—required (steady, wet coil)	80	67	82	[1] 65	Maximum	Cooling Certified [2]
E_V Test—required (steady, wet coil)	80	67	87	[1] 69	Intermediate	Cooling Intermediate [3]
B_1 Test—required (steady, wet coil)	80	67	82	[1] 65	Minimum	Cooling Minimum [4]
F_1 Test—required (steady, wet coil)	80	67	67	[1] 53.5	Minimum	Cooling Minimum [4]
G_1 Test [5]—optional (steady, dry coil)	80	([5])	67	....	Minimum	Cooling Minimum [4]
I_1 Test [5]—optional (cyclic, dry coil)	80	([5])	67	....	Minimum	([6])

[1] The specified test condition only applies if the unit rejects condensate to the outdoor coil.
[2] Defined in section 3.1.4.1.
[3] Defined in section 3.1.4.3.
[4] Defined in section 3.1.4.2.
[5] The entering air must have a low enough moisture content so no condensate forms on the indoor coil. (It is recommended that an indoor wet-bulb temperature of 57 °F or less be used.)
[6] Maintain the airflow nozzles static pressure difference or velocity pressure during the ON period at the same pressure difference or velocity pressure as measured during the G_1 Test.

3.3 Test procedures for steady-state wet coil cooling mode tests (the A, A_2, A_1, B, B_2, B_1, E_V, and F_1 Tests). a. For the pretest interval, operate the test room reconditioning apparatus and the unit to be tested until maintaining equilibrium conditions for at least 30 minutes at the specified section 3.2 test conditions. Use the exhaust fan of the airflow measuring apparatus and, if installed, the indoor fan of the test unit to obtain and then maintain the indoor air volume rate and/or external static pressure specified for the particular test. Continuously record (see Definition 1.15):

(1) The dry-bulb temperature of the air entering the indoor coil,

(2) The water vapor content of the air entering the indoor coil,

(3) The dry-bulb temperature of the air entering the outdoor coil, and

(4) For the section 2.2.4 cases where its control is required, the water vapor content of the air entering the outdoor coil.

Refer to section 3.11 for additional requirements that depend on the selected secondary test method.

b. After satisfying the pretest equilibrium requirements, make the measurements specified in Table 5 of ASHRAE Standard 37–88 (incorporated by reference, see § 430.22) for the Indoor Air Enthalpy method and the user-selected secondary method. Except for external static pressure, make the Table 5 measurements at equal intervals that span 10 minutes or less. Measure external static pressure every 5 minutes or less. Continue data sampling until reaching a 30-minute period (*e.g.*, four consecutive 10-minute samples) where the test tolerances specified in Table 7 are satisfied. For those continuously recorded parameters, use the entire data set from the 30-minute interval to evaluate Table 7 compliance. Determine the average electrical power consumption of the air conditioner or heat pump over the same 30-minute interval.

c. Calculate indoor-side total cooling capacity as specified in section 7.3.3.1 of ASHRAE Standard 37–88 (incorporated by reference, see § 430.22). Do not adjust the parameters used in calculating capacity for the permitted variations in test conditions. Evaluate air enthalpies based on the measured barometric pressure. Assign the average total space cooling capacity and electrical power consumption over the 30-minute data collection interval to the variables $\dot{Q}_c^k(T)$ and $\dot{E}_c^k(T)$, respectively. For these two variables, replace the "T" with the nominal outdoor temperature at which the test was conducted. The superscript k is used only when testing multi-capacity units. Use the superscript k=2 to denote a test with the unit operating at high capacity or maximum speed, k=1 to denote low capacity or minimum speed, and k=v to denote the intermediate speed.

d. For units tested without an indoor fan installed, decrease $\dot{Q}_c^k(T)$ by

$$\frac{1250\ \text{Btu/h}}{1000\ \text{scfm}} \cdot \overline{\dot{V}_s},$$

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

$$\frac{365\ \text{W}}{1000\ \text{scfm}} \cdot \overline{\dot{V}_s},$$

where $\overline{\dot{V}_s}$ is the average measured indoor air volume rate expressed in units of cubic feet per minute of standard air (scfm).

TABLE 7.—TEST OPERATING AND TEST CONDITION TOLERANCES FOR SECTION 3.3 STEADY-STATE WET COIL COOLING MODE TESTS AND SECTION 3.4 DRY COIL COOLING MODE TESTS

	Test operating tolerance [1]	Test condition tolerance [2]
Indoor dry-bulb, °F		
Entering temperature	2.0	0.5
Leaving temperature	2.0	
Indoor wet-bulb, °F		
Entering temperature	1.0	[3] 0.3
Leaving temperature	[3] 1.0	
Outdoor dry-bulb, °F		
Entering temperature	2.0	0.5
Leaving temperature	[4] 2.0	
Outdoor wet-bulb, °F		
Entering temperature	1.0	[5] 0.3
Leaving temperature	[4] 1.0	
External resistance to airflow, inches of water	0.05	[6] 0.02
Electrical voltage, % of rdg.	2.0	1.5
Nozzle pressure drop, % of rdg.	2.0	

[1] See Definition 1.41.
[2] See Definition 1.40.
[3] Only applies during wet coil tests; does not apply during steady-state, dry coil cooling mode tests.
[4] Only applies when using the Outdoor Air Enthalpy Method.
[5] Only applies during wet coil cooling mode tests where the unit rejects condensate to the outdoor coil.
[6] Only applies when testing non-ducted units.

d. For air conditioners and heat pumps having a constant-air-volume-rate indoor fan, the five additional steps listed below are required if the average of the measured external static pressures exceeds the applicable sections 3.1.4 minimum (or target) external static pressure (ΔP_{min}) by 0.03 inches of water or more.

1. Measure the average power consumption of the indoor fan motor ($\dot{E}_{fan,1}$) and record the corresponding external static pressure (ΔP_1) during or immediately following the 30-minute interval used for determining capacity.

2. After completing the 30-minute interval and while maintaining the same test conditions, adjust the exhaust fan of the airflow measuring apparatus until the external static pressure increases to approximately $\Delta P_1 + (\Delta P_1 - \Delta P_{min})$.

3. After re-establishing steady readings of the fan motor power and external static pressure, determine average values for the indoor fan power ($\dot{E}_{fan,2}$) and the external static pressure (ΔP_2) by making measurements over a 5-minute interval.

4. Approximate the average power consumption of the indoor fan motor at ΔP_{min} using linear extrapolation:

$$\dot{E}_{fan,min} = \frac{\dot{E}_{fan,2} - \dot{E}_{fan,1}}{\Delta P_2 - \Delta P_1}\left(\Delta P_{min} - \Delta P_1\right) + \dot{E}_{fan,1}.$$

5. Increase the total space cooling capacity, $\dot{Q}_c^k(T)$, by the quantity $(\dot{E}_{fan,1} - \dot{E}_{fan,min})$, when expressed on a Btu/h basis. Decrease the total electrical power, $\dot{E}_c^k(T)$, by the same fan power difference, now expressed in watts.

3.4 Test procedures for the optional steady-state dry coil cooling mode tests (the C, C_1, and G_1 Tests). a. Except for the modifications noted in this section, conduct the steady-state dry coil cooling mode tests as specified in section 3.3 for wet coil tests. Prior to recording data during the steady-state dry coil test, operate the unit at least one hour after achieving dry coil conditions. Drain the drain pan and plug the drain opening. Thereafter, the drain pan should remain completely dry.

b. Denote the resulting total space cooling capacity and electrical power derived from the test as $\dot{Q}_{ss,dry}$ and $\dot{E}_{ss,dry}(T)$. In preparing for the section 3.5 cyclic test, record the average indoor-side air volume rate, $\bar{\dot{V}}$, specific heat of the air, $C_{p,a}$ (expressed on dry air basis), specific volume of the air at the nozzles, v'_n, humidity ratio at the nozzles, W_n, and either pressure difference or velocity pressure for the flow nozzles. For units having a variable-speed indoor fan (that provides either a constant or variable air volume rate) that will or may be tested during the cyclic dry coil cooling mode test with the indoor fan turned off (see section 3.5), include the electrical power used by the indoor fan motor among the recorded parameters from the 30-minute test.

3.5 Test procedures for the optional cyclic dry coil cooling mode tests (the D, D_1, and I_1 Tests). a. After completing the steady-state dry-coil test, remove the Outdoor Air Enthalpy method test apparatus, if connected, and begin manual OFF/ON cycling of the unit's compressor. The test set-up should otherwise be identical to the set-up used during the steady-state dry coil test. When testing heat pumps, leave the reversing valve during the compressor OFF cycles in the same position as used for the compressor ON cycles, unless automatically changed by the controls of the unit. For units having a variable-speed indoor fan, the manufacturer has the option of electing at the outset whether to conduct the cyclic test with the indoor fan enabled or disabled. Always revert to testing with the indoor fan disabled if cyclic testing with the fan enabled is unsuccessful.

b. For units having a single-speed or two-capacity compressor, cycle the compressor OFF for 24 minutes and then ON for 6 minutes ($\Delta\tau_{cyc,dry}$ = 0.5 hours). For units having a variable-speed compressor, cycle the compressor OFF for 48 minutes and then ON for 12 minutes ($\Delta\tau_{cyc,dry}$ = 1.0 hours). Repeat the OFF/ON compressor cycling pattern until the test is completed. Allow the controls of the unit to regulate cycling of the outdoor fan.

c. Sections 3.5.1 and 3.5.2 specify airflow requirements through the indoor coil of ducted and non-ducted systems, respectively. In all cases, use the exhaust fan of the airflow measuring apparatus (covered under section 2.6) along with the indoor fan of the unit, if installed and operating, to approximate a step response in the indoor coil airflow. Regulate the exhaust fan to quickly obtain and then maintain the flow nozzle static pressure difference or velocity pressure at the same value as was measured during the steady-state dry coil test. The pressure difference or velocity pressure should be within 2 percent of the value from the steady-state dry coil test within 15 seconds after airflow initiation. For units having a variable-speed indoor fan that ramps when cycling on and/or off, use the exhaust fan of the airflow measuring apparatus to impose a step response that begins at the initiation of ramp up and ends at the termination of ramp down.

d. For units having a variable-speed indoor fan, conduct the cyclic dry coil test using the pull-thru approach described below if any of the following occur when testing with the fan operating:

(1) The test unit automatically cycles off;

(2) Its blower motor reverses; or

(3) The unit operates for more than 30 seconds at an external static pressure that is 0.1 inches of water or more higher than the value measured during the prior steady-state test.

For the pull-thru approach, disable the indoor fan and use the exhaust fan of the airflow measuring apparatus to generate the specified flow nozzles static pressure difference or velocity pressure. If the exhaust fan cannot deliver the required pressure difference because of resistance created by the unpowered blower, temporarily remove the blower.

e. After completing a minimum of two complete compressor OFF/ON cycles, determine the overall cooling delivered and total electrical energy consumption during any subsequent data collection interval where the test tolerances given in Table 8 are satisfied. If available, use electric resistance heaters (see section 2.1) to minimize the variation in the inlet air temperature.

f. With regard to the Table 8 parameters, continuously record the dry-bulb temperature of the air entering the indoor and outdoor coils during periods when air flows through the respective coils. Sample the water vapor content of the indoor coil inlet air at least every 2 minutes during periods when air flows through the coil. Record external static pressure and the air volume rate indicator (either nozzle pressure difference or velocity pressure) at least every minute during the interval that air flows through the indoor coil. (These regular measurements of the airflow rate indicator are in addition to the required measurement at 15 seconds after flow initiation.) Sample the electrical voltage at least every 2 minutes beginning 30 seconds after compressor start-up. Continue until the compressor, the outdoor fan, and the indoor fan (if it is installed and operating) cycle off.

g. For ducted units, continuously record the dry-bulb temperature of the air entering (as noted above) and leaving the indoor coil. Or if using a thermopile, continuously record the difference between these two temperatures during the interval that air flows through the indoor coil. For non-ducted units, make the same dry-bulb temperature measurements beginning when the compressor cycles on and ending when indoor coil airflow ceases.

h. Integrate the electrical power over complete cycles of length $\Delta\tau_{cyc,dry}$. For ducted units tested with an indoor fan installed and operating, integrate electrical power from indoor fan OFF to indoor fan OFF. For all other ducted units and for non-ducted units, integrate electrical power from compressor OFF to compressor OFF. (Some cyclic tests will use the same data collection intervals to determine the electrical energy and the total space cooling. For other units, terminate data collection used to determine the electrical energy before terminating data collection used to determine total space cooling.)

TABLE 8.—TEST OPERATING AND TEST CONDITION TOLERANCES FOR CYCLIC DRY COIL COOLING MODE TESTS

	Test Operating Tolerance [1]	Test Condition Tolerance [2]
Indoor entering dry-bulb temperature [3], °F	2.0	0.5
Indoor entering wet-bulb temperature, °F	....	([4])
Outdoor entering dry-bulb temperature [3], °F	2.0	0.5
External resistance to airflow [3], inches of water	0.05	....
Airflow nozzle pressure difference or velocity pressure [3], % of reading	2.0	[5] 2.0
Electrical voltage [6], % of rdg.	2.0	1.5

[1] See Definition 1.41.

[2] See Definition 1.40.

[3] Applies during the interval that air flows through the indoor (outdoor) coil except for the first 30 seconds after flow initiation. For units having a variable-speed indoor fan that ramps, the tolerances listed for the external resistance to airflow apply from 30 seconds after achieving full speed until ramp down begins.

[4] Shall at no time exceed a wet-bulb temperature that results in condensate forming on the indoor coil.

[5] The test condition shall be the average nozzle pressure difference or velocity pressure measured during the steady-state dry coil test.

[6] Applies during the interval when at least one of the following—the compressor, the outdoor fan, or, if applicable, the indoor fan—are operating except for the first 30 seconds after compressor start-up.

i. If the Table 8 tolerances are satisfied over the complete cycle, record the measured electrical energy consumption as $e_{cyc,dry}$ and express it in units of watt-hours. Calculate the total space cooling delivered, $q_{cyc,dry}$, in units of Btu using,

$$q_{cyc,dry} = \frac{60 \cdot \bar{\dot{V}} \cdot C_{p,a} \cdot \Gamma}{\left[v'_n \cdot \left(1 + W_n\right)\right]} = \frac{60 \cdot \bar{\dot{V}} \cdot C_{p,a} \cdot \Gamma}{v_n} \qquad (3.5\text{-}1)$$

where $\bar{\dot{V}}$, $C_{p,a}$, v_n' (or v_n), and W_n are the values recorded during the section 3.4 dry coil steady-state test and,

$$\Gamma = \int_{\tau_1}^{\tau_2} \left[T_{a1}(\tau) - T_{a2}(\tau)\right] d\tau, \text{ hr} \cdot {}^{\circ}\text{F}.$$

$T_{a1}(\tau)$ = dry bulb temperature of the air entering the indoor coil at time τ, °F.
$T_{a2}(\tau)$ = dry bulb temperature of the air leaving the indoor coil at time τ, °F.
τ_1 = for ducted units, the elapsed time when airflow is initiated through the indoor coil; for non-ducted units, the elapsed time when the compressor is cycled on, hr.
τ_2 = the elapsed time when indoor coil airflow ceases, hr.

3.5.1 Procedures when testing ducted systems. The automatic controls that are normally installed with the test unit must govern the OFF/ON cycling of the air moving equipment on the indoor side (exhaust fan of the airflow measuring apparatus and, if installed, the indoor fan of the test unit). For example, for ducted units tested without an indoor fan installed but rated based on using a fan time delay relay, control the indoor coil airflow according to the rated ON and/or OFF delays provided by the relay. For ducted units having a variable-speed indoor fan that has been disabled (and possibly removed), start and stop the indoor airflow at the same instances as if the fan were enabled. For all other ducted units tested without an indoor fan installed, cycle the indoor coil airflow in unison with the cycling of the compressor. Close air dampers on the inlet (section 2.5.1) and outlet side (sections 2.5 and 2.5.4) during the OFF period. Airflow through the indoor coil should stop within 3 seconds after the automatic controls of the test unit (act to) de-energize the indoor fan. For ducted units tested without an indoor fan installed (excluding the special case where a variable-speed fan is temporarily removed), increase $e_{cyc,dry}$ by the quantity,

$$\frac{365 \text{ W}}{1000 \text{ scfm}} \cdot \bar{\dot{V}}_s \cdot \left[\tau_2 - \tau_1\right], \qquad (3.5\text{-}2)$$

and decrease $q_{cyc,dry}$ by,

$$\frac{1250 \text{ Btu/h}}{1000 \text{ scfm}} \cdot \bar{\dot{V}}_s \cdot \left[\tau_2 - \tau_1\right], \qquad (3.5\text{-}3)$$

where $\bar{\dot{V}}_s$ is the average indoor air volume rate from the section 3.4 dry coil steady-state test and is expressed in units of cubic feet per minute of standard air (scfm). For units having a variable-speed indoor fan that is disabled during the cyclic test, increase $e_{cyc,dry}$ and decrease $q_{cyc,dry}$ based on:

a. The product of $[\tau_2 - \tau_1]$ and the indoor fan power measured during or following the dry coil steady-state test; or,

b. The following algorithm if the indoor fan ramps its speed when cycling.

1. Measure the electrical power consumed by the variable-speed indoor fan at a minimum of three operating conditions: at the speed/air volume rate/external static pressure that was measured during the steady-state test, at operating conditions associated with the midpoint of the ramp-up interval, and at conditions associated with the midpoint of the ramp-down interval. For these measurements, the tolerances on the airflow volume or the external static pressure are the same as required for the section 3.4 steady-state test.

2. For each case, determine the fan power from measurements made over a minimum of 5 minutes.

3. Approximate the electrical energy consumption of the indoor fan if it had operated during the cyclic test using all three power measurements. Assume a linear profile during the ramp intervals. The manufacturer must provide the durations of the ramp-up and ramp-down intervals. If a manufacturer-supplied ramp interval exceeds 45 seconds, use a 45-second ramp interval nonetheless when estimating the fan energy.

The manufacturer is allowed to choose option a, and forego the extra testing burden of option b, even if the unit ramps indoor fan speed when cycling.

3.5.2 Procedures when testing non-ducted systems. Do not use air dampers when conducting cyclic tests on non-ducted units. Until the last OFF/ON compressor cycle, airflow through the indoor coil must cycle off and on in unison with the compressor. For the last OFF/ON compressor cycle—the one used to determine $e_{cyc,dry}$ and $q_{cyc,dry}$—use the exhaust fan of the airflow measuring apparatus and the indoor fan of the test unit to have indoor airflow start 3 minutes prior to compressor cut-on and end three minutes after compressor cutoff. Subtract the electrical energy used by the indoor fan during the 3 minutes prior to compressor cut-on from the integrated electrical energy, $e_{cyc,dry}$. Add the electrical energy used by the indoor fan during the 3 minutes after compressor cutoff to the integrated cooling capacity, $q_{cyc,dry}$. For the case where the non-ducted unit uses a variable-speed indoor fan which is disabled during the cyclic test, correct $e_{cyc,dry}$ and $q_{cyc,dry}$ using the same approach as prescribed in section 3.5.1 for ducted units having a disabled variable-speed indoor fan.

3.5.3 Cooling mode cyclic degradation coefficient calculation. Use two optional dry-coil tests to determine the cooling mode cyclic degradation coefficient, C_D^c. If the two optional tests are not conducted, assign C_D^c the default value of 0.25. Evaluate C_D^c using the above results and those from the section 3.4 dry coil steady-state test.

$$C_D^c = \frac{1 - \dfrac{EER_{cyc,dry}}{EER_{ss,dry}}}{1 - CLF}$$

where,

$$EER_{cyc,dry} = \frac{q_{cyc,dry}}{e_{cyc,dry}},$$

the average energy efficiency ratio during the cyclic dry coil cooling mode test, Btu/W·h

$$EER_{ss,dry} = \frac{\dot{Q}_{ss,dry}}{\dot{E}_{ss,dry}},$$

the average energy efficiency ratio during the steady-state dry coil cooling mode test, Btu/W·h

$$CLF = \frac{q_{cyc,dry}}{Q_{ss,dry} \cdot \Delta\tau_{cyc,dry}},$$

the cooling load factor dimensionless.

Round the calculated value for C_D^c to the nearest 0.01. If C_D^c is negative, then set it equal to zero.

3.6 Heating mode tests for different types of heat pumps, including heating-only heat pumps.

3.6.1 Tests for a heat pump having a single-speed compressor that is tested with a fixed speed indoor fan installed, with a constant-air-volume-rate indoor fan installed, or with no indoor fan installed. Conduct three tests: The High Temperature (H1) Test, the Frost Accumulation (H2) Test, and the Low Temperature (H3) Test. Conduct the optional High Temperature Cyclic (H1C) Test to determine the heating mode cyclic degradation coefficient, C_D^h. If this optional test is not conducted, assign C_D^h the default value of 0.25. Test conditions for these four tests are specified in Table 9.

TABLE 9.—HEATING MODE TEST CONDITIONS FOR UNITS HAVING A SINGLE-SPEED COMPRESSOR AND A FIXED-SPEED INDOOR FAN, A CONSTANT AIR VOLUME RATE INDOOR FAN, OR NO INDOOR FAN

Test description	Air entering indoor unit Temperature (°F)		Air entering outdoor unit Temperature (°F)		Heating air volume rate
	Dry bulb	Wet bulb	Dry bulb	Wet bulb	
H1 Test (required, steady)	70	60(max)	47	43	Heating Certified [1]
H1C Test (optional, cyclic)	70	60(max)	47	43	([2])
H2 Test (required)	70	60(max)	35	33	Heating Certified [1]
H3 Test (required, steady)	70	60(max)	17	15	Heating Certified [1]

[1] Defined in section 3.1.4.4.
[2] Maintain the airflow nozzles static pressure difference or velocity pressure during the ON period at the same pressure difference or velocity pressure as measured during the H1 Test.

3.6.2 Tests for a heat pump having a single-speed compressor and a variable-speed, variable-air-volume-rate indoor fan: capacity modulation correlates with outdoor dry bulb temperature. Conduct five tests: two High Temperature Tests ($H1_2$ and $H1_1$), one Frost Accumulation Test ($H2_2$), and two Low Temperature Tests ($H3_2$ and $H3_1$). Conducting an additional Frost Accumulation Test ($H2_1$) is optional. Conduct the optional High Temperature Cyclic ($H1C_1$) Test to determine the heating mode cyclic degradation coefficient, C_D^h. If this optional test is not conducted, assign C_D^h the default value of 0.25. Table 10 specifies test conditions for these seven tests. If the optional $H2_1$ Test is not done, use the following equations to approximate the capacity and electrical power of the heat pump at the $H2_1$ test conditions:

$$\dot{Q}_h^{k=1}(35) = QR_h^{k=2}(35) \cdot \left\{\dot{Q}_h^{k=1}(17) + 0.6 \cdot \left[\dot{Q}_h^{k=1}(47) - \dot{Q}_h^{k=1}(17)\right]\right\}$$

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

where,

$$\dot{Q}R_h^{k=2}(35) = \frac{\dot{Q}_h^{k=2}(35)}{\dot{Q}^{k=2}(17) + 0.6 \cdot \left[\dot{Q}_h^{k=2}(47) - \dot{Q}_h^{k=2}(17)\right]}$$

$$PR_h^{k=2}(35) = \frac{\dot{E}_h^{k=2}(35)}{\dot{E}_h^{k=2}(17) + 0.6 \cdot \left[\dot{E}_h^{k=2}(47) - \dot{E}_h^{k=2}(17)\right]}.$$

The quantities $\dot{Q}_h^{k=2}(47)$, $\dot{E}_h^{k=2}(47)$, $\dot{Q}_h^{k=1}(47)$, and $\dot{E}_h^{k=1}(47)$ are determined from the $H1_2$ and $H1_1$ Tests and evaluated as specified in section 3.7; the quantities $\dot{Q}_h^{k=2}(35)$ and $\dot{E}_h^{k=2}(35)$ are determined from the $H2_2$ Test and evaluated as specified in section 3.9; and the quantities $\dot{Q}_h^{k=2}(17)$, $\dot{E}_h^{k=2}(17)$, $\dot{Q}_h^{k=1}(17)$, and $\dot{E}_h^{k=1}(17)$, are determined from the $H3_2$ and $H3_1$ Tests and evaluated as specified in section 3.10.

TABLE 10.—HEATING MODE TEST CONDITIONS FOR UNITS HAVING A SINGLE-SPEED COMPRESSOR AND A VARIABLE AIR VOLUME RATE INDOOR FAN

Test description	Air entering indoor unit temperature (°F)		Air entering outdoor unit temperature (°F)		Heating air volume rate
	Dry bulb	Wet bulb	Dry bulb	Wet bulb	
$H1_2$ Test (required, steady)	70	60(max)	47	43	Heating Certified.[1]
$H1_1$ Test (required, steady)	70	60(max)	47	43	Heating Minimum.[2]
$H1C_1$ Test (optional, cyclic)	70	60(max)	47	43	([3])
$H2_2$ Test (required)	70	60(max)	35	33	Heating Certified.[1]
$H2_1$ Test (optional)	70	60(max)	35	33	Heating Minimum.[2]
$H3_2$ Test (required, steady)	70	60(max)	17	15	Heating Certified.[1]
$H3_1$ Test (required, steady)	70	60(max)	17	15	Heating Minimum.[2]

[1] Defined in section 3.1.4.4.
[2] Defined in section 3.1.4.5.
[3] Maintain the airflow nozzles static pressure difference or velocity pressure during the ON period at the same pressure difference or velocity pressure as measured during the $H1_1$ Test.

3.6.3 Tests for a heat pump having a two-capacity compressor (see Definition 1.45), including two-capacity, northern heat pumps (see Definition 1.46). a. Conduct one Maximum Temperature Test ($H0_1$), two High Temperature Tests ($H1_2$ and $H1_1$), one Frost Accumulation Test ($H2_2$), and one Low Temperature Test ($H3_2$). Conduct an additional Frost Accumulation Test ($H2_1$) and Low Temperature Test ($H3_1$) if both of the following conditions exist:

1. Knowledge of the heat pump's capacity and electrical power at low compressor capacity for outdoor temperatures of 37°F and less is needed to complete the section 4.2.3 seasonal performance calculations, and
2. The heat pump's controls allow low capacity operation at outdoor temperatures of 37°F and less.

b. Conduct the optional Maximum Temperature Cyclic Test ($H0C_1$) to determine the heating mode cyclic degradation coefficient, C_D^h. If this optional test is not conducted, assign C_D^h the default value of 0.25. Table 11 specifies test conditions for these eight tests.

TABLE 11.—HEATING MODE TEST CONDITIONS FOR UNITS HAVING A TWO-CAPACITY COMPRESSOR

Test description	Air entering indoor unit Temperature (°F)		Air entering outdoor unit Temperature (°F)		Compressor capacity	Heating air volume rate
	Dry Bulb	Wet Bulb	Dry Bulb	Wet Bulb		
$H0_1$ Test (required, steady)	70	60(max)	62	56.5	Low	Heating Minimum[1]
$H0C_1$ Test (optional, cyclic)	70	60(max)	62	56.5	Low	([2])
$H1_2$ Test (required, steady)	70	60(max)	47	43	High	Heating Certified[3]

TABLE 11.—HEATING MODE TEST CONDITIONS FOR UNITS HAVING A TWO-CAPACITY COMPRESSOR—Continued

Test description	Air entering indoor unit Temperature (°F)		Air entering outdoor unit Temperature (°F)		Compressor capacity	Heating air volume rate
	Dry Bulb	Wet Bulb	Dry Bulb	Wet Bulb		
$H1_1$ Test (required, steady)	70	$60^{(max)}$	47	43	Low	Heating Minimum [1]
$H2_2$ Test (required)	70	$60^{(max)}$	35	33	High	Heating Certified [3]
$H2_1$ Test [4] (required)	70	$60^{(max)}$	35	33	Low	Heating Minimum [1]
$H3_2$ Test (required, steady)	70	$60^{(max)}$	17	15	High	Heating Certified [3]
$H3_1$ Test [4] (required, steady)	70	$60^{(max)}$	17	15	Low	Heating Minimum [1]

[1] Defined in section 3.1.4.5.
[2] Maintain the airflow nozzles static pressure difference or velocity pressure during the ON period at the same pressure difference or velocity pressure as measured during the $H0_1$ Test.
[3] Defined in section 3.1.4.4.
[4] Required only if the heat pump's performance when operating at low compressor capacity and outdoor temperatures less than 37 °F is needed to complete the section 4.2.3 HSPF calculations.

3.6.4 Tests for a heat pump having a variable-speed compressor. a. Conduct one Maximum Temperature Test ($H0_1$), two High Temperature Tests ($H1_2$ and $H1_1$), one Frost Accumulation Test ($H2_V$), and one Low Temperature Test ($H3_2$). Conducting one or both of the following tests is optional: An additional High Temperature Test ($H1_N$) and an additional Frost Accumulation Test ($H2_2$). Conduct the optional Maximum Temperature Cyclic ($H0C_1$) Test to determine the heating mode cyclic degradation coefficient, C_D^h. If this optional test is not conducted, assign C_D^h the default value of 0.25. Table 12 specifies test conditions for these eight tests. Determine the intermediate compressor speed cited in Table 12 using the heating mode maximum and minimum compressors speeds and:

$$\text{Intermediate speed} = \text{Minimum speed} + \frac{\text{Maximum speed} - \text{Minimum speed}}{3}$$

where a tolerance of plus 5 percent or the next higher inverter frequency step from that calculated is allowed. If the $H2_2$ Test is not done, use the following equations to approximate the capacity and electrical power at the $H2_2$ test conditions:

$$\dot{Q}_h^{k=2}(35) = 0.90 \cdot \left\{\dot{Q}_h^{k=2}(17) + 0.6 \cdot \left[\dot{Q}_h^{k=2}(47) - \dot{Q}_h^{k=2}(17)\right]\right\}$$

$$\dot{E}_h^{k=2}(35) = 0.985 \cdot \left\{\dot{E}_h^{k=2}(17) + 0.6 \cdot \left[\dot{E}_h^{k=2}(47) - \dot{E}_h^{k=2}(17)\right]\right\}.$$

b. Determine the quantities $\dot{Q}_h^{k=2}(47)$ and from $\dot{E}_h^{k=2}(47)$ from the $H1_2$ Test and evaluate them according to section 3.7. Determine the quantities $\dot{Q}_h^{k=2}(17)$ and $\dot{E}_h^{k=2}(17)$ from the $H3_2$ Test and evaluate them according to section 3.10. For heat pumps where the heating mode maximum compressor speed exceeds its cooling mode maximum compressor speed, conduct the $H1_N$ Test if the manufacturer requests it. If the $H1_N$ Test is done, operate the heat pump's compressor at the same speed as the speed used for the cooling mode A_2 Test. Refer to the last sentence of section 4.2 to see how the results of the $H1_N$ Test may be used in calculating the heating seasonal performance factor.

TABLE 12.—HEATING MODE TEST CONDITIONS FOR UNITS HAVING A VARIABLE-SPEED COMPRESSOR

Test description	Air entering indoor unit temperature (°F)		Air entering outdoor unit temperature (°F)		Compressor speed	Heating air volume rate
	Dry bulb	Wet bulb	Dry bulb	Wet bulb		
$H0_1$ Test (required, steady)	70	$60^{(max)}$	62	56.5	Minimum	Heating Minimum.[1]
$H0C_1$ Test (optional, cyclic)	70	$60^{(max)}$	62	56.5	Minimum	([2])
$H1_2$ Test (required, steady)	70	$60^{(max)}$	47	43	Maximum	Heating Certified.[3]
$H1_1$ Test (required, steady)	70	$60^{(max)}$	47	43	Minimum	Heating Minimum.[1]
$H1_N$ Test (optional, steady)	70	$60^{(max)}$	47	43	Cooling Mode Maximum.	Heating Nominal.[4]
$H2_2$ Test (optional)	70	$60^{(max)}$	35	33	Maximum	Heating Certified.[3]
$H2_V$ Test (required)	70	$60^{(max)}$	35	33	Intermediate	Heating Intermediate.[5]
$H3_2$ Test (required, steady)	70	$60^{(max)}$	17	15	Maximum	Heating Certified.[3]

[1] Defined in section 3.1.4.5.
[2] Maintain the airflow nozzles static pressure difference or velocity pressure during the ON period at the same pressure difference or velocity pressure as measured during the $H0_1$ Test.
[3] Defined in section 3.1.4.4.
[4] Defined in section 3.1.4.7.
[5] Defined in section 3.1.4.6.

3.6.5 Additional test for a heat pump having a heat comfort controller. Test any heat pump that has a heat comfort controller (see Definition 1.28) according to section 3.6.1, 3.6.2, or 3.6.3, whichever applies, with the heat comfort controller disabled. Additionally, conduct the abbreviated test described in section 3.1.9 with the heat comfort controller active to determine the system's maximum supply air temperature. (Note: heat pumps having a variable speed compressor and a heat comfort controller are not covered in the test procedure at this time.)

3.7 Test procedures for steady-state Maximum Temperature and High Temperature heating mode tests (the $H0_1$, H1, $H1_2$, $H1_1$, and $H1_N$ Tests). a. For the pretest interval, operate the test room reconditioning apparatus and the heat pump until equilibrium conditions are maintained for at least 30 minutes at the specified section 3.6 test conditions. Use the exhaust fan of the airflow measuring apparatus and, if installed, the indoor fan of the heat pump to obtain and then maintain the indoor air volume rate and/or the external static pressure specified for the particular test. Continuously record the dry-bulb temperature of the air entering the indoor coil, and the dry-bulb temperature and water vapor content of the air entering the outdoor coil. Refer to section 3.11 for additional requirements that depend on the selected secondary test method. After satisfying the pretest equilibrium requirements, make the measurements specified in Table 5 of ASHRAE Standard 37–88 (incorporated by reference, see § 430.22) for the Indoor Air Enthalpy method and the user-selected secondary method. Except for external static pressure, make the Table 5 measurements at equal intervals that span 10 minutes or less. Measure external static pressure every 5 minutes or less. Continue data sampling until a 30-minute period (*e.g.*, four consecutive 10-minute samples) is reached where the test tolerances specified in Table 13 are satisfied. For those continuously recorded parameters, use the entire data set for the 30-minute interval when evaluating Table 13 compliance. Determine the average electrical power consumption of the heat pump over the same 30-minute interval.

TABLE 13.—TEST OPERATING AND TEST CONDITION TOLERANCES FOR SECTION 3.7 AND SECTION 3.10 STEADY-STATE HEATING MODE TESTS

	Test operating tolerance [1]	Test condition tolerance [2]
Indoor dry-bulb, °F:		
Entering temperature	2.0	0.5
Leaving temperature	2.0	
Indoor wet-bulb, °F:		
Entering temperature	1.0	
Leaving temperature	1.0	
Outdoor dry-bulb, °F:		
Entering temperature	2.0	0.5
Leaving temperature	[2] 2.0	
Outdoor wet-bulb, °F:		
Entering temperature	1.0	0.3
Leaving temperature	[3] 1.0	
External resistance to airflow, inches of water	0.05	[4] 0.02
Electrical voltage, % of rdg	2.0	1.5
Nozzle pressure drop, % of rdg	2.0	

[1] See Definition 1.41.
[2] See Definition 1.40.
[3] Only applies when the Outdoor Air Enthalpy Method is used.
[4] Only applies when testing non-ducted units.

b. Calculate indoor-side total heating capacity as specified in section 7.3.4.1 of ASHRAE Standard 37–88 (incorporated by reference, see § 430.22). Do not adjust the parameters used in calculating capacity for the permitted variations in test conditions. Assign the average space heating capacity and electrical power over the 30-minute data collection interval to the variables $\dot{Q}_h^k$ and $\dot{E}_h^k(T)$ respectively. The "T" and superscripted "k" are the same as described in section 3.3. Additionally, for the heating mode, use the superscript to denote results from the optional $H1_N$ Test, if conducted.

c. For heat pumps tested without an indoor fan installed, increase $\dot{Q}_h^k(T)$ by

$$\frac{1250 \text{ Btu/h}}{1000 \text{ scfm}} \cdot \overline{\dot{V}}_s,$$

and increase $\dot{E}_h^k(T)$ by,

$$\frac{365 \text{ W}}{1000 \text{ scfm}} \cdot \overline{\dot{V}}_s,$$

where $\overline{\dot{V}}_s$ is the average measured indoor air volume rate expressed in units of cubic feet per minute of standard air (scfm). During the 30-minute data collection interval of a High Temperature Test, pay attention to preventing a defrost cycle. Prior to this time, allow the heat pump to perform a defrost cycle if automatically initiated by its own controls. As in all cases, wait for the heat pump's defrost controls to automatically terminate the defrost cycle. Heat pumps that undergo a defrost should operate in the heating mode for at least 10 minutes after defrost termination prior to beginning the 30-minute data collection interval. For some heat pumps, frost may accumulate on the outdoor coil during a High Temperature test. If the indoor coil leaving air temperature or the difference between the leaving and entering air temperatures decreases by more than 1.5 °F over the 30-minute data collection interval, then do not use the collected data to determine capacity. Instead, initiate a defrost cycle. Begin collecting data no sooner than 10 minutes after defrost termination. Collect 30 minutes of new data during which the Table 13 test tolerances are satisfied. In this case, use only the results from the second 30-minute data collection interval to evaluate $\dot{Q}_h^k(47)$ and $\dot{E}_h^k(47)$.

d. If conducting the optional cyclic heating mode test, which is described in section 3.8, record the average indoor-side air volume rate, $\dot{V}$, specific heat of the air, $C_{p,a}$ (expressed on dry air basis), specific volume of the air at the nozzles, v_n' (or v_n), humidity ratio at the nozzles, W_n, and either pressure difference or velocity pressure for the flow nozzles. If either or both of the below criteria apply, determine the average, steady-state, electrical power consumption of the indoor fan motor ($\dot{E}_{fan,1}$):

1. The section 3.8 cyclic test will be conducted and the heat pump has a variable-speed indoor fan that is expected to be disabled during the cyclic test; or

2. The heat pump has a (variable-speed) constant-air volume-rate indoor fan and during the steady-state test the average external static pressure (ΔP_1) exceeds the applicable section 3.1.4.4 minimum (or targeted) external static pressure (ΔP_{min}) by 0.03 inches of water or more.

Determine $\dot{E}_{fan,1}$ by making measurements during the 30-minute data collection interval, or immediately following the test and prior to changing the test conditions. When the above "2" criteria applies, conduct the following four steps after determining $\dot{E}_{fan,1}$ (which corresponds to ΔP_1):

i. While maintaining the same test conditions, adjust the exhaust fan of the airflow measuring apparatus until the external static pressure increases to approximately $\Delta P_1 + (\Delta P_1 - \Delta P_{min})$.

ii. After re-establishing steady readings for fan motor power and external static pressure, determine average values for the indoor fan power ($\dot{E}_{fan,2}$) and the external static pressure (ΔP_2) by making measurements over a 5-minute interval.

iii. Approximate the average power consumption of the indoor fan motor if the 30-minute test had been conducted at ΔP_{min} using linear extrapolation:

$$\dot{E}_{fan,min} = \frac{\dot{E}_{fan,2} - \dot{E}_{fan,1}}{\Delta P_2 - \Delta P_1}\left(\Delta P_{min} - \Delta P_1\right) + \dot{E}_{fan,1}.$$

iv. Decrease the total space heating capacity, $\dot{Q}_h^k(T)$, by the quantity ($\dot{E}_{fan,1} - \dot{E}_{fan,min}$), when expressed on a Btu/h basis. Decrease the total electrical power, $\dot{E}_h^k(T)$ by the same fan power difference, now expressed in watts.

3.8 Test procedures for the optional cyclic heating mode tests (the H0C$_1$, H1C, and H1C$_1$ Tests). a. Except as noted below, conduct the cyclic heating mode test as specified in section 3.5. As adapted to the heating mode, replace section 3.5 references to ''the steady-state dry coil test'' with ''the heating mode steady-state test conducted at the same test conditions as the cyclic heating mode test.'' Use the test tolerances in Table 14 rather than Table 8. Record the outdoor coil entering wet-bulb temperature according to the requirements given in section 3.5 for the outdoor coil entering dry-bulb temperature. Drop the subscript ''dry'' used in variables cited in section 3.5 when referring to quantities from the cyclic heating mode test. Determine the total space heating delivered during the cyclic heating test, q_{cyc}, as specified in section 3.5 except for making the following changes:

(1) When evaluating Equation 3.5–1, use the values of $\bar{V}$, $C_{p,a}$, v_n', (or v_n), and W_n that were recorded during the section 3.7 steady-state test conducted at the same test conditions.

(2) Calculate Γ using,

$$\Gamma = \int_{\tau_1}^{\tau_2}\left[T_{a2}(\tau) - T_{a1}(\tau)\right]\delta\tau,\ \text{hr}\cdot{}^\circ\text{F}.$$

b. For ducted heat pumps tested without an indoor fan installed (excluding the special case where a variable-speed fan is temporarily removed), increase q_{cyc} by the amount calculated using Equation 3.5–3. Additionally, increase e_{cyc} by the amount calculated using Equation 3.5–2. In making these calculations, use the average indoor air volume rate ($\bar{V}_s$) determined from the section 3.7 steady-state heating mode test conducted at the same test conditions.

c. For non-ducted heat pumps, subtract the electrical energy used by the indoor fan during the 3 minutes after compressor cutoff from the non-ducted heat pump's integrated heating capacity, q_{cyc}.

d. If a heat pump defrost cycle is manually or automatically initiated immediately prior to or during the OFF/ON cycling, operate the heat pump continuously until 10 minutes after defrost termination. After that, begin cycling the heat pump immediately or delay until the specified test conditions have been re-established. Pay attention to preventing defrosts after beginning the cycling process. For heat pumps that cycle off the indoor fan during a defrost cycle, make no effort here to restrict the air movement through the indoor coil while the fan is off. Resume the OFF/ON cycling while conducting a minimum of two complete compressor OFF/ON cycles before determining q_{cyc} and e_{cyc}.

3.8.1 Heating mode cyclic degradation coefficient calculation. Use the results from the optional cyclic test and the required steady-state test that were conducted at the same test conditions to determine the heating mode cyclic degradation coefficient, C_D^h. If the optional test is not conducted, assign C_D^h the default value of 0.25.

$$C_D^h = \frac{1 - \dfrac{COP_{cyc}}{COP_{ss}\left(T_{cyc}\right)}}{1 - HLF}$$

where,

$$COP_{cyc} = \frac{q_{cyc}}{3.413\dfrac{Btu/h}{W}\cdot e_{cyc}},$$

the average coefficient of performance during the cyclic heating mode test, dimensionless.

$$COP_{ss}\left(T_{cyc}\right) = \frac{\dot{Q}_h^k\left(T_{cyc}\right)}{3.413\dfrac{Btu/h}{W}\cdot \dot{E}_h^k\left(T_{cyc}\right)},$$

the average coefficient of performance during the steady-state heating mode test conducted at the same test conditions—*i.e.*, same outdoor dry bulb temperature, T_{cyc}, and speed/capacity, k, if applicable—as specified for the cyclic heating mode test, dimensionless.

$$HLF = \frac{q_{cyc}}{\dot{Q}_h^k\left(T_{cyc}\right)\cdot \Delta\tau_{cyc}},$$

the heating load factor, dimensionless.

T_{cyc} = the nominal outdoor temperature at which the cyclic heating mode test is conducted, 62 or 47 °F.

$\Delta\tau_{cyc}$ = the duration of the OFF/ON intervals; 0.5 hours when testing a heat pump having a single-speed or two-capacity compressor and 1.0 hour when testing a heat pump having a variable-speed compressor.

Round the calculated value for C_D^h to the nearest 0.01. If C_D^h is negative, then set it equal to zero.

TABLE 14.—TEST OPERATING AND TEST CONDITION TOLERANCES FOR CYCLIC HEATING MODE TESTS.

	Test operating tolerance [1]	Test condition tolerance [2]
Indoor entering dry-bulb temperature,[3] °F	2.0	0.5
Indoor entering wet-bulb temperature,[3] °F	1.0	
Outdoor entering dry-bulb temperature,[3] °F	2.0	0.5
Outdoor entering wet-bulb temperature,[3] °F	2.0	1.0
External resistance to air-flow,[3] inches of water	0.05	
Airflow nozzle pressure difference or velocity pressure,[3] % of reading	2.0	[4] 2.0
Electrical voltage,[5] % of rdg	2.0	1.5

[1] See Definition 1.41.
[2] See Definition 1.40.
[3] Applies during the interval that air flows through the indoor (outdoor) coil except for the first 30 seconds after flow initiation. For units having a variable-speed indoor fan that ramps, the tolerances listed for the external resistance to airflow shall apply from 30 seconds after achieving full speed until ramp down begins.
[4] The test condition shall be the average nozzle pressure difference or velocity pressure measured during the steady-state test conducted at the same test conditions.
[5] Applies during the interval that at least one of the following—the compressor, the outdoor fan, or, if applicable, the indoor fan—are operating, except for the first 30 seconds after compressor start-up.

3.9 Test procedures for Frost Accumulation heating mode tests (the H2, H2$_2$, H2$_V$, and H2$_1$ Tests). a. Confirm that the defrost controls of the heat pump are set as specified in section 2.2.1. Operate the test room reconditioning apparatus and the heat pump for at least 30 minutes at the specified section 3.6 test conditions before starting the ''preliminary'' test period. The preliminary test period must immediately precede the ''official'' test period, which is the heating and defrost interval over which data are collected for evaluating average space heating capacity and average electrical power consumption.

b. For heat pumps containing defrost controls which are likely to cause defrosts at intervals less than one hour, the preliminary test period starts at the termination of an automatic defrost cycle and ends at the termination of the next occurring automatic defrost cycle. For heat pumps containing defrost controls which are likely to cause defrosts at intervals exceeding one hour, the preliminary test period must consist of a heating interval lasting at least one hour followed by a defrost cycle that is either manually or automatically initiated. In all cases, the heat pump's own controls must govern when a defrost cycle terminates.

c. The official test period begins when the preliminary test period ends, at defrost termination. The official test period ends at the termination of the next occurring automatic defrost cycle. When testing a heat pump that uses a time-adaptive defrost control system (see Definition 1.42), however, manually initiate the defrost cycle that ends the official test period at the instant indicated by instructions provided by the manufacturer. If the heat pump has not undergone a defrost after 12 hours, immediately conclude the test and use the results from the full 12-hour period to calculate the average space heating capacity and average electrical power consumption. For heat pumps that turn the indoor fan off during the defrost cycle, take steps to cease forced airflow through the indoor coil and block the outlet duct whenever the heat pump's controls cycle off the indoor fan. If it is installed, use the outlet damper box described in section 2.5.4.1 to affect the blocked outlet duct.

d. Defrost termination occurs when the controls of the heat pump actuate the first change in converting from defrost operation to normal heating operation. Defrost initiation occurs when the controls of the heat pump first alter its normal heating operation in order to eliminate possible accumulations of frost on the outdoor coil.

e. To constitute a valid Frost Accumulation test, satisfy the test tolerances specified in Table 15 during both the preliminary and official test periods. As noted in Table 15, test operating tolerances are specified for two sub-intervals: (1) When heating, except for the first 10 minutes after the termination of a defrost cycle (Sub-interval H, as described in Table 15) and (2) when defrosting, plus these same first 10 minutes after defrost termination (Sub-interval D, as described in Table 15). Evaluate compliance with Table 15 test condition tolerances and the majority of the test operating tolerances using the averages from measurements recorded only during Sub-interval H. Continuously record the dry bulb temperature of the air entering the indoor coil, and the dry bulb temperature and water vapor content of the air entering the outdoor coil. Sample the remaining parameters listed in Table 15 at equal intervals that span 10 minutes or less.

f. For the official test period, collect and use the following data to calculate average space heating capacity and electrical power. During heating and defrosting intervals when the controls of the heat pump have the indoor fan on, continuously record the dry-bulb temperature of the air entering (as noted above) and leaving the indoor coil. If using a thermopile, continuously record the difference between the leaving and entering dry-bulb temperatures during the interval(s) that air flows through the indoor coil. For heat pumps tested without an indoor fan installed, determine the corresponding cumulative time (in hours) of indoor coil airflow, $\Delta\tau_a$. Sample measurements used in calculating the air volume rate (refer to sections 7.8.3.1 and 7.8.3.2 of ASHRAE Standard 37–88 (incorporated by reference, see § 430.22)) at equal intervals that span 10 minutes or less. Record the electrical energy consumed, expressed in watt-hours, from defrost termination to defrost termination, $e_{DEF}^{k}(35)$, as well as the corresponding elapsed time in hours, $\Delta\tau_{FR}$.

TABLE 15.—TEST OPERATING AND TEST CONDITION TOLERANCES FOR FROST ACCUMULATION HEATING MODE TESTS.

	Test operating tolerance [1]		Test condition tolerance [2]
	Sub-interval H [3]	Sub-interval D [4]	Sub-interval H [3]
Indoor entering dry-bulb temperature, °F	2.0	[5] 4.0	0.5
Indoor entering wet-bulb temperature, °F	1.0	....	....
Outdoor entering dry-bulb temperature, °F	2.0	10.0	1.0
Outdoor entering wet-bulb temperature, °F	1.5	....	0.5
External resistance to airflow, inches of water	0.05	....	0.02 [6]
Electrical voltage, % of rdg	2.0	....	1.5

[1] See Definition 1.41.
[2] See Definition 1.40.
[3] Applies when the heat pump is in the heating mode, except for the first 10 minutes after termination of a defrost cycle.
[4] Applies during a defrost cycle and during the first 10 minutes after the termination of a defrost cycle when the heat pump is operating in the heating mode.
[5] For heat pumps that turn off the indoor fan during the defrost cycle, the noted tolerance only applies during the 10 minute interval that follows defrost termination.
[6] Only applies when testing non-ducted heat pumps.

3.9.1 Average space heating capacity and electrical power calculations. a. Evaluate average space heating capacity, $\dot{Q}_h^k(35)$, when expressed in units of Btu per hour, using:

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

where,

$\overline{\dot{V}}$ = the average indoor air volume rate measured during Sub-interval H, cfm.

$C_{p,a} = 0.24 + 0.444 \cdot W_n$, the constant pressure specific heat of the air-water vapor mixture that flows through the indoor coil and is expressed on a dry air basis, Btu / lbm_{da} · °F.

v_n' = specific volume of the air-water vapor mixture at the nozzle, ft³ / lbm_{mx}.

W_n = humidity ratio of the air-water vapor mixture at the nozzle, lbm of water vapor per lbm of dry air.

$\Delta\tau_{FR} = \tau_2 - \tau_1$, the elapsed time from defrost termination to defrost termination, hr.

$$\Gamma = \int_{\tau_1}^{\tau_2} \left[T_{a2}(\tau) - T_{a1}(\tau)\right] d\tau, \ \text{hr} \cdot {}^\circ\text{F}.$$

$T_{al}(\tau)$ = dry bulb temperature of the air entering the indoor coil at elapsed time τ, °F; only recorded when indoor coil airflow occurs; assigned the value of zero during periods (if any) where the indoor fan cycles off.

$T_{a2}(\tau)$ = dry bulb temperature of the air leaving the indoor coil at elapsed time τ, °F; only recorded when indoor coil airflow occurs; assigned the value of zero during periods (if any) where the indoor fan cycles off.

τ_1 = the elapsed time when the defrost termination occurs that begins the official test period, hr.

τ_2 = the elapsed time when the next automatically occurring defrost termination occurs, thus ending the official test period, hr.

v_n = specific volume of the dry air portion of the mixture evaluated at the dry-bulb temperature, vapor content, and barometric pressure existing at the nozzle, ft^3 per lbm of dry air.

b. Evaluate average electrical power, $\dot{E}_h^k(35)$, when expressed in units of watts, using:

$$\dot{E}_h^k(35) = \frac{e_{def}(35)}{\Delta\tau_{FR}}.$$

For heat pumps tested without an indoor fan installed, increase $\dot{Q}_h^k(35)$ by,

$$\frac{1250 \text{ Btu/h}}{1000 \text{ scfm}} \cdot \bar{\dot{V}}_s \cdot \frac{\Delta\tau_a}{\Delta\tau_{FR}},$$

and increase $\dot{E}_h^k(35)$ by,

$$\frac{365 \text{ W}}{1000 \text{ scfm}} \cdot \bar{\dot{V}}_s \cdot \frac{\Delta\tau_a}{\Delta\tau_{FR}},$$

where $\bar{\dot{V}}_s$ is the average indoor air volume rate measured during the Frost Accumulation heating mode test and is expressed in units of cubic feet per minute of standard air (scfm).

c. For heat pumps having a constant-air-volume-rate indoor fan, the five additional steps listed below are required if the average of the external static pressures measured during sub-Interval H exceeds the applicable section 3.1.4.4, 3.1.4.5, or 3.1.4.6 minimum (or targeted) external static pressure (ΔP_{min}) by 0.03 inches of water or more:

1. Measure the average power consumption of the indoor fan motor ($\dot{E}_{fan,1}$) and record the corresponding external static pressure (ΔP_1) during or immediately following the Frost Accumulation heating mode test. Make the measurement at a time when the heat pump is heating, except for the first 10 minutes after the termination of a defrost cycle.

2. After the Frost Accumulation heating mode test is completed and while maintaining the same test conditions, adjust the exhaust fan of the airflow measuring apparatus until the external static pressure increases to approximately $\Delta P_1 + (\Delta P_1 - \Delta P_{min})$.

3. After re-establishing steady readings for the fan motor power and external static pressure, determine average values for the indoor fan power ($\dot{E}_{fan,2}$) and the external static pressure (ΔP_2) by making measurements over a 5-minute interval.

4. Approximate the average power consumption of the indoor fan motor had the Frost Accumulation heating mode test been conducted at ΔP_{min} using linear extrapolation:

$$\dot{E}_{fan,min} = \frac{\dot{E}_{fan,2} - \dot{E}_{fan,1}}{\Delta P_2 - \Delta P_1}\left(\Delta P_{min} - \Delta P_1\right) + \dot{E}_{fan,1}.$$

5. Decrease the total heating capacity, $\dot{Q}_h^k(35)$, by the quantity $[(\dot{E}_{fan,1} - \dot{E}_{fan,min}) \cdot (\Delta\tau_a/\Delta\tau_{FR}]$, when expressed on a Btu/h basis. Decrease the total electrical power, $E_h^k(35)$, by the same quantity, now expressed in watts.

3.9.2 Demand defrost credit. a. Assign the demand defrost credit, F_{def}, that is used in section 4.2 to the value of 1 in all cases except for heat pumps having a demand-defrost control system (Definition 1.21). For such qualifying heat pumps, evaluate F_{def} using,

$$F_{def} = 1 + 0.03 \cdot \left[1 - \frac{\Delta\tau_{def} - 1.5}{\Delta\tau_{max} - 1.5}\right],$$

where,

$\Delta\tau_{def}$ = the time between defrost terminations (in hours) or 1.5, whichever is greater.

$\Delta\tau_{max}$ = maximum time between defrosts as allowed by the controls (in hours) or 12, whichever is less.

b. For two-capacity heat pumps and for section 3.6.2 units, evaluate the above equation using the $\Delta\tau_{def}$ that applies based on the Frost Accumulation Test conducted at high capacity and/or at the Heating Certified Air Volume Rate. For variable-speed heat pumps, evaluate $\Delta\tau_{def}$ based on the required Frost Accumulation Test conducted at the intermediate compressor speed.

3.10 Test procedures for steady-state Low Temperature heating mode tests (the H3, H3$_2$, and H3$_1$ Tests). Except for the modifications noted in this section, conduct the Low Temperature heating mode test using the same approach as specified in section 3.7 for the Maximum and High Temperature tests. After satisfying the section 3.7 requirements for the pretest interval but before beginning to collect data to determine $\dot{Q}_h^k(17)$ and $\dot{E}_h^k(17)$, conduct a defrost cycle. This defrost cycle may be manually or automatically initiated. The defrost sequence must be terminated by the action of the heat pump's defrost controls. Begin the 30-minute data collection interval described in section 3.7, from which $\dot{Q}_h^k(17)$ and $\dot{E}_h^k(17)$ are determined, no sooner than 10 minutes after defrost termination. Defrosts should be prevented over the 30-minute data collection interval.

3.11 Additional requirements for the secondary test methods. Prior to evaluating if the energy balance specified in section 3.1.1 is obtained, make an adjustment to account for the energy loss within the air duct that connects the indoor coil and the location where the outlet dry-bulb temperature is measured. If using the Outdoor Air Enthalpy Method, make an adjustment to account for the energy loss within the air duct that connects the outdoor coil and the location where the outlet temperature is measured. In all cases, apply the correction to the indoor space conditioning capacity that is determined using the secondary test method.

3.11.1 If using the Outdoor Air Enthalpy Method as the secondary test method. During the ''official'' test, the outdoor air-side test apparatus described in section 2.10.1 is connected to the outdoor unit. To help compensate for any effect that the addition of this test apparatus may have on the unit's performance, conduct a ''preliminary'' test where the outdoor air-side test apparatus is disconnected. Conduct a preliminary test prior to the first section 3.2 steady-state cooling mode test and prior to the first section 3.6 steady-state heating mode test. No other preliminary tests are required so long as the unit operates the outdoor fan during all cooling mode steady-state tests at the same speed and all heating mode steady-state tests at the same speed. If using more than one outdoor fan speed for the cooling mode steady-state tests, however, conduct a preliminary test prior to each cooling mode test where a different fan speed is first used. This same requirement applies for the heating mode tests.

3.11.1.1 If a preliminary test precedes the official test. a. The test conditions for the preliminary test are the same as specified for the official test. Connect the indoor air-side test apparatus to the indoor coil; disconnect the outdoor air-side test apparatus. Allow the test room reconditioning apparatus and the unit being tested to operate for at least one hour. After attaining equilibrium conditions, measure the following quantities at equal intervals that span 10 minutes or less:

1. The section 2.10.1 evaporator and condenser temperatures or pressures;

2. Parameters required according to the Indoor Air Enthalpy Method.

Continue these measurements until a 30-minute period (*e.g.*, four consecutive 10-minute samples) is obtained where the Table 7 or Table 13, whichever applies, test tolerances are satisfied.

b. After collecting 30 minutes of steady-state data, reconnect the outdoor air-side test apparatus to the unit. Adjust the exhaust fan of the outdoor airflow measuring apparatus until averages for the evaporator and condenser temperatures, or the saturated temperatures corresponding to the measured pressures, agree within ±0.5 °F of the averages achieved when the outdoor air-side test apparatus was disconnected. Calculate the averages for the reconnected case using five or more consecutive readings taken at one minute intervals. Make these consecutive readings after re-establishing equilibrium conditions and before initiating the official test.

3.11.1.2 If a preliminary test does not precede the official test. Connect the outdoor-side test apparatus to the unit. Adjust the exhaust fan of the outdoor airflow measuring apparatus to achieve the same external static pressure as measured during the prior preliminary test conducted with the unit operating in the same cooling or heating mode at the same outdoor fan speed.

3.11.1.3 Official test. a. Continue (preliminary test was conducted) or begin (no preliminary test) the official test by making measurements for both the Indoor and Outdoor Air Enthalpy Methods at equal intervals that span 10 minutes or less.

Discontinue these measurement only after obtaining a 30-minute period where the specified test condition and test operating tolerances are satisfied. To constitute a valid official test:

(1) Achieve the energy balance specified in section 3.1.1; and,

(2) For cases where a preliminary test is conducted, the capacities determined using the Indoor Air Enthalpy Method from the official and preliminary test periods must agree within 2.0 percent.

b. For space cooling tests, calculate capacity from the outdoor air enthalpy measurements as specified in section 7.3.3.2 of ASHRAE Standard 37–88 (incorporated by reference, see § 430.22). Calculate heating capacity based on outdoor air enthalpy measurements as specified in section 7.3.4.2 of the same ASHRAE Standard. Adjust outdoor side capacities according to section 7.3.3.3 of ASHRAE Standard 37–88 (incorporated by reference, see § 430.22) to account for line losses when testing split systems. Do not correct the average electrical power measurement as described in section 8.5.3 of ASHRAE Standard 37–88 (incorporated by reference, see § 430.22).

3.11.2 If using the Compressor Calibration Method as the secondary test method.

a. Conduct separate calibration tests using a calorimeter to determine the refrigerant flow rate. Or for cases where the superheat of the refrigerant leaving the evaporator is less than 5 °F, use the calorimeter to measure total capacity rather than refrigerant flow rate. Conduct these calibration tests at the same test conditions as specified for the tests in this Appendix. Operate the unit for at least one hour or until obtaining equilibrium conditions before collecting data that will be used in determining the average refrigerant flow rate or total capacity. Sample the data at equal intervals that span 10 minutes or less. Determine average flow rate or average capacity from data sampled over a 30-minute period where the Table 7 (cooling) or the Table 13 (heating) tolerances are satisfied. Otherwise, conduct the calibration tests according to ASHRAE Standard 23–93 (incorporated by reference, see § 430.22), ASHRAE Standard 41.9–00 (incorporated by reference, see § 430.22), and section 7.5 of ASHRAE Standard 37–88 (incorporated by reference, see § 430.22).

b. Calculate space cooling and space heating capacities using the compressor calibration method measurements as specified in sections 7.5.7 and 7.5.8, respectively, of ASHRAE Standard 37–88 (incorporated by reference, see § 430.22).

3.11.3 If using the Refrigerant Enthalpy Method as the secondary test method. Conduct this secondary method according to section 7.6 of ASHRAE Standard 37–88 (incorporated by reference, see § 430.22). Calculate space cooling and space heating capacities using the refrigerant enthalpy method measurements as specified in sections 7.6.4 and 7.6.5, respectively, of the same ASHRAE Standard.

3.12 Rounding of space conditioning capacities for reporting purposes.

a. When reporting rated capacities, round them off as follows:

1. For capacities less than 20,000 Btu/h, round to the nearest 100 Btu/h.

2. For capacities between 20,000 and 37,999 Btu/h, round to the nearest 200 Btu/h.

3. For capacities between 38,000 and 64,999 Btu/h, round to the nearest 500 Btu/h.

b. For the capacities used to perform the section 4 calculations, however, round only to the nearest integer.

4. CALCULATIONS OF SEASONAL PERFORMANCE DESCRIPTORS

4.1 Seasonal Energy Efficiency Ratio (SEER) Calculations. SEER must be calculated as follows: For equipment covered under sections 4.1.2, 4.1.3, and 4.1.4, evaluate the seasonal energy efficiency ratio,

$$\text{SEER} = \frac{\sum_{j=1}^{8} q_c(T_j)}{\sum_{j=1}^{8} e_c(T_j)} = \frac{\sum_{j=1}^{8} \frac{q_c(T_j)}{N}}{\sum_{j=1}^{8} \frac{e_c(T_j)}{N}} \qquad (4.1\text{-}1)$$

where,

$$\frac{q_c(T_j)}{N} =$$

the ratio of the total space cooling provided during periods of the space cooling season when the outdoor temperature fell within the range represented by bin temperature T_j to the total number of hours in the cooling season (N), Btu/h.

$$\frac{e_c(T_j)}{N} =$$

the electrical energy consumed by the test unit during periods of the space cooling season when the outdoor temperature fell within the range represented by bin temperature T_j to the total number of hours in the cooling season (N), W.

T_j = the outdoor bin temperature, °F. Outdoor temperatures are grouped or ''binned.'' Use bins of 5 °F with the 8 cooling season bin temperatures being 67, 72, 77, 82, 87, 92, 97, and 102 °F.

j = the bin number. For cooling season calculations, j ranges from 1 to 8.

Additionally, for sections 4.1.2, 4.1.3, and 4.1.4, use a building cooling load, $BL(T_j)$. When referenced, evaluate $BL(T_j)$ for cooling using,

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

where,

$\dot{Q}_c^{k=2}(95)$ = the space cooling capacity determined from the A_2 Test and calculated as specified in section 3.3, Btu/h.

1.1 = sizing factor, dimensionless.

The temperatures 95 °F and 65 °F in the building load equation represent the selected outdoor design temperature and the zero-load base temperature, respectively.

4.1.1 SEER calculations for an air conditioner or heat pump having a single-speed compressor that was tested with a fixed-speed indoor fan installed, a constant-air-volume-rate indoor fan installed, or with no indoor fan installed. a. Evaluate the seasonal energy efficiency ratio, expressed in units of Btu/watt-hour, using:

$\text{SEER} = PLF(0.5) \cdot EER_B$

where,

$$EER_B = \frac{\dot{Q}_c(82)}{\dot{E}_c(82)},$$

the energy efficiency ratio determined from the B Test described in sections 3.2.1, 3.1.4.1, and 3.3, Btu/h per watt.

$PLF(0.5) = 1 - 0.5 \cdot C_D^c$, the part-load performance factor evaluated at a cooling load factor of 0.5, dimensionless.

b. Refer to section 3.3 regarding the definition and calculation of $\dot{Q}_c(82)$ and $\dot{E}_c(82)$. If the optional tests described in section 3.2.1 are not conducted, set the cooling mode cyclic degradation coefficient, C_D^c, to the default value specified in section

3.5.3. If these optional tests are conducted, set C_D^c to the lower of:

1. The value calculated as per section 3.5.3; or

2. The section 3.5.3 default value of 0.25.

4.1.2 SEER calculations for an air conditioner or heat pump having a single-speed compressor and a variable-speed variable-air-volume-rate indoor fan.

4.1.2.1 Units covered by section 3.2.2.1 where indoor fan capacity modulation correlates with the outdoor dry bulb temperature. The manufacturer must provide information on how the indoor air volume rate or the indoor fan speed varies over the outdoor temperature range of 67 °F to 102 °F. Calculate SEER using Equation 4.1–1. Evaluate the quantity $q_c(T_j)/N$ in Equation 4.1–1 using,

$$\frac{q_c(T_j)}{N} = X(T_j) \cdot \dot{Q}_c(T_j) \cdot \frac{n_j}{N} \qquad (4.1.2\text{-}1)$$

where,

$$X(T_j) = \begin{Bmatrix} BL(T_j)/\dot{Q}_c(T_j) \\ \text{or} \\ 1 \end{Bmatrix};$$

whichever is less; the cooling mode load factor for temperature bin j, dimensionless.

$\dot{Q}_c(T_j)$ = the space cooling capacity of the test unit when operating at outdoor temperature, T_j, Btu/h.

n_j/N = fractional bin hours for the cooling season; the ratio of the number of hours during the cooling season when the outdoor temperature fell within the range represented by bin temperature T_j to the total number of hours in the cooling season, dimensionless.

a. For the space cooling season, assign n_j/N as specified in Table 16. Use Equation 4.1–2 to calculate the building load, $BL(T_j)$. Evaluate $\dot{Q}_c(T_j)$ using,

$$\dot{Q}_c(T_j) = \dot{Q}_c^{k=1}(T_j) + \frac{\dot{Q}_c^{k=2}(T_j) - \dot{Q}_c^{k=1}(T_j)}{FP_c^{k=2} - FP_c^{k=1}} \cdot \left[FP_c(T_j) - FP_c^{k=1}\right] \qquad (4.1.2\text{-}2)$$

where,

$$\dot{Q}_c^{k=1}(T_j) = \dot{Q}_c^{k=1}(82) + \frac{\dot{Q}_c^{k=1}(95) - \dot{Q}_c^{k=1}(82)}{95 - 82} \cdot (T_j - 82),$$

the space cooling capacity of the test unit at outdoor temperature T_j if operated at the Cooling Minimum Air Volume Rate, Btu/h.

$$\dot{Q}_c^{k=2}(T_j) = \dot{Q}_c^{k=2}(82) + \frac{\dot{Q}_c^{k=2}(95) - \dot{Q}_c^{k=2}(82)}{95 - 82} \cdot (T_j - 82),$$

the space cooling capacity of the test unit at outdoor temperature T_j if operated at the Cooling Certified Air Volume Rate, Btu/h.

b. For units where indoor fan speed is the primary control variable, $FP_c^{k=1}$ denotes the fan speed used during the required A_1 and B_1 Tests (see section 3.2.2.1), $FP_c^{k=2}$ denotes the fan speed used during the required A_2 and B_2 Tests, and $FP_c(T_j)$ denotes the fan speed used by the unit when the outdoor temperature equals T_j. For units where indoor air volume rate is the primary control variable, the three FP_c's are similarly defined only now being expressed in terms of air volume rates rather than fan speeds. Refer to sections 3.2.2.1, 3.1.4 to 3.1.4.2, and 3.3 regarding the definitions and calculations of $\dot{Q}_c^{k=1}(82)$, $\dot{Q}_c^{k=1}(95)$, $\dot{Q}_c^{k=2}(82)$, and $\dot{Q}_c^{k=2}(95)$. Calculate $e_c(T_j)/N$ in Equation 4.1–1 using,

$$\frac{e_c(T_j)}{N} = \frac{X(T_j) \cdot \dot{E}_c(T_j)}{PLF_j} \cdot \frac{n_j}{N} \qquad (4.1.2\text{-}3)$$

where,

$PLF_j = 1 - C_D^c \cdot [1 - X(T_j)]$, the part load factor, dimensionless.

$\dot{E}_c(T_j)$ = the electrical power consumption of the test unit when operating at outdoor temperature T_j, W.

c. The quantities $X(T_j)$ and n_j/N are the same quantities as used in Equation 4.1.2–1. If the optional tests described in section 3.2.2.1 and Table 4 are not conducted, set the cooling mode cyclic degradation coefficient, C_D^c, to the default value specified in section 3.5.3. If these optional tests are conducted, set C_D^c to the lower of:

1. The value calculated as per section 3.5.3; or

2. The section 3.5.3 default value of 0.25.

d. Evaluate $\dot{E}_c(T_j)$ using,

$$\dot{E}_c(T_j) = \dot{E}_c^{k=1}(T_j) + \frac{\dot{E}_c^{k=2}(T_j) - \dot{E}_c^{k=1}(T_j)}{FP_c^{k=2} - FP_c^{k=1}} \cdot \left[FP_c(T_j) - FP_c^{k=1}\right] \qquad (4.1.2\text{-}4)$$

where

$$\dot{E}_c^{k=1}(T_j) = \dot{E}_c^{k=1}(82) + \frac{\dot{E}_c^{k=1}(95) - \dot{E}_c^{k=1}(82)}{95 - 82} \cdot (T_j - 82),$$

the electrical power consumption of the test unit at outdoor temperature T_j if operated at the Cooling Minimum Air Volume Rate, W.

$$\dot{E}_c^{k=2}(T_j) = \dot{E}_c^{k=2}(82) + \frac{\dot{E}_c^{k=2}(95) - \dot{E}_c^{k=2}(82)}{95 - 82} \cdot (T_j - 82),$$

the electrical power consumption of the test unit at outdoor temperature T_j if operated at the Cooling Certified Air Volume Rate, W.

e. The parameters $FP_c^{k=1}$, and $FP_c^{k=2}$, and $FP_c(T_j)$ are the same quantities that are used when evaluating Equation 4.1.2–2. Refer to sections 3.2.2.1, 3.1.4 to 3.1.4.2, and 3.3 regarding the definitions and calculations of $\dot{E}_c^{k=1}(82)$, $\dot{E}_c^{k=1}(95)$, $\dot{E}_c^{k=2}(82)$, and $\dot{E}_c^{k=2}(95)$.

4.1.2.2 Units covered by section 3.2.2.2 where indoor fan capacity modulation is used to adjust the sensible to total cooling capacity ratio. Calculate SEER as specified in section 4.1.1.

4.1.3 SEER calculations for an air conditioner or heat pump having a two-capacity compressor. Calculate SEER using Equation 4.1–1. Evaluate the space cooling capacity, $\dot{Q}_c^{k=1}(T_j)$, and electrical power consumption, $\dot{E}_c^{k=1}(T_j)$, of the test unit when operating at low compressor capacity and outdoor temperature T_j using,

$$\dot{Q}_c^{k=1}(T_j) = \dot{Q}_c^{k=1}(82) + \frac{\dot{Q}_c^{k=1}(95) - \dot{Q}_c^{k=1}(82)}{95 - 82} \cdot (T_j - 82) \qquad (4.1.3\text{-}1)$$

$$\dot{E}_c^{k=1}(T_j) = \dot{E}_c^{k=1}(82) + \frac{\dot{E}_c^{k=1}(95) - \dot{E}_c^{k=1}(82)}{95 - 82} \cdot (T_j - 82) \qquad (4.1.3\text{-}2)$$

where $\dot{Q}_c^{k=1}(95)$ and $\dot{E}_c^{k=1}(95)$ are determined from the A_1 Test, $\dot{Q}_c^{k=1}(82)$ and $\dot{E}_c^{k=1}(82)$ are determined from the B_1 Test, and all are calculated as specified in section 3.3. For two-capacity units that lock out low capacity operation at outdoor temperatures less than 95 °F (but greater than 82 °F), use Equations 4.1.4–1 and 4.1.4–2 rather than Equations 4.1.3–1 and 4.1.3–2 for estimating performance at low compressor capacity.

Evaluate the space cooling capacity, $\dot{Q}_c^{k=2}(T_j)$, and electrical power consumption, $\dot{E}_c^{k=2}(T_j)$, of the test unit when operating at high compressor capacity and outdoor temperature T_j using,

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

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

where $\dot{Q}_c^{k=2}(95)$ and $\dot{E}_c^{k=2}(95)$ are determined from the A_2 Test, $\dot{Q}_c^{k=2}(82)$, and $\dot{E}_c^{k=2}(82)$, are determined from the B_2 Test, and all are calculated as specified in section 3.3.

The calculation of Equation 4.1–1 quantities $q_c(T_j)/N$ and $e_c(T_j)/N$ differs depending on whether the test unit would operate at low capacity (section 4.1.3.1), cycle between low and high capacity (section 4.1.3.2), or operate at high capacity (sections 4.1.3.3 and 4.1.3.4) in responding to the building load. For units that lock out low capacity operation at higher outdoor temperatures, the manufacturer must supply information regarding this temperature so that the appropriate equations are used. Use Equation 4.1–2 to calculate the building load, $BL(T_j)$, for each temperature bin.

4.1.3.1 Steady-state space cooling capacity at low compressor capacity is greater than or equal to the building cooling load at temperature T_j, $\dot{Q}_c^{k=1}(T_j) \geq BL(T_j)$.

$$\frac{q_c(T_j)}{N} = X^{k=1}(T_j) \cdot \dot{Q}_c^{k=1}(T_j) \cdot \frac{n_j}{N}$$

$$\frac{e_c(T_j)}{N} = \frac{X^{k=1}(T_j) \cdot \dot{E}_c^{k=1}(T_j)}{PLF_j} \cdot \frac{n_j}{N}$$

where,

$X^{k=1}(T_j) = BL(T_j)/\dot{Q}_c^{k=1}(T_j)$, the cooling mode low capacity load factor for temperature bin j, dimensionless.

$PLF_j = 1 - C_D^c \cdot [1 - X^{k=1}(T_j)]$, the part load factor, dimensionless.

$\frac{n_j}{N} =$ fractional bin hours for the cooling season; the ratio of the number of hours during the cooling season when the outdoor temperature fell within the range represented by bin temperature T_j to the total number of hours in the cooling season, dimensionless.

Obtain the fractional bin hours for the cooling season, n_j/N, from Table 16. Use Equations 4.1.3–1 and 4.1.3–2, respectively, to evaluate $\dot{Q}_c^{k=1}(T_j)$ and $\dot{E}_c^{k=1}(T_j)$. If the optional tests described in section 3.2.3 and Table 5 are not conducted, set the cooling mode cyclic degradation coefficient, C_D^c, to the default value specified in section 3.5.3. If these optional tests are conducted, set C_D^c to the lower of:

a. The value calculated according to section 3.5.3; or

b. The section 3.5.3 default value of 0.25.

TABLE 16.—DISTRIBUTION OF FRACTIONAL HOURS WITHIN COOLING SEASON TEMPERATURE BINS

Bin number, j	Bin temperature range °F	Representative temperature for bin °F	Fraction of of total temperature bin hours, n_j/N
1	65–69	67	0.214
2	70–74	72	0.231
3	75–79	77	0.216
4	80–84	82	0.161
5	85–89	87	0.104
6	90–94	92	0.052
7	95–99	97	0.018
8	100–104	102	0.004

4.1.3.2 Unit alternates between high (k=2) and low (k=1) compressor capacity to satisfy the building cooling load at temperature T_j, $\dot{Q}_c^{k=1}(T_j) < BL(T_j) < \dot{Q}_c^{k=2}(T_j)$.

$$\frac{q_c(T_j)}{N} = \left[X^{k=1}(T_j) \cdot \dot{Q}_c^{k=1}(T_j) + X^{k=2}(T_j) \cdot \dot{Q}_c^{k=2}(T_j)\right] \cdot \frac{n_j}{N}$$

$$\frac{e_c(T_j)}{N} = \left[X^{k=1}(T_j) \cdot \dot{E}_c^{k=1}(T_j) + X^{k=2}(T_j) \cdot \dot{E}_c^{k=2}(T_j)\right] \cdot \frac{n_j}{N}$$

where,

$$X^{k=1}(T_j) = \frac{\dot{Q}_c^{k=2}(T_j) - BL(T_j)}{\dot{Q}_c^{k=2}(T_j) - \dot{Q}_c^{k=1}(T_j)},$$

the cooling mode, low capacity load factor for temperature bin j, dimensionless.

$X^{k=2}(T_j) = 1 - X^{k=1}(T_j)$, the cooling mode, high capacity load factor for temperature bin j, dimensionless.

Obtain the fractional bin hours for the cooling season, n_j/N, from Table 16. Use Equations 4.1.3–1 and 4.1.3–2, respectively, to evaluate $\dot{Q}_c^{k=1}(T_j)$ and $\dot{E}_c^{k=1}(T_j)$. Use Equations 4.1.3–3 and 4.1.3–4, respectively, to evaluate $\dot{Q}_c^{k=2}(T_j)$ and $\dot{E}_c^{k=2}(T_j)$.

4.1.3.3 Unit only operates at high (k=2) compressor capacity at temperature T_j and its capacity is greater than the building cooling load, $BL(T_j) < \dot{Q}_c^{k=2}(T_j)$. This section applies to units that lock out low compressor capacity operation at higher outdoor temperatures.

$$\frac{q_c(T_j)}{N} = X^{k=2}(T_j) \cdot \dot{Q}_c^{k=2}(T_j) \cdot \frac{n_j}{N}$$

$$\frac{e_c(T_j)}{N} = \frac{X^{k=2}(T_j) \cdot \dot{E}_c^{k=2}(T_j)}{PLF_j} \cdot \frac{n_j}{N}$$

where,

$X^{k=2}(T_j) = BL(T_j)/\dot{Q}_c^{k=2}(T_j)$, the cooling mode high capacity load factor for temperature bin j, dimensionless.

$PLF_j = 1 - C_D^c \cdot [1 - X^{k=2}(T_j)]$, the part load factor, dimensionless.

Obtain the fractional bin hours for the cooling season, n_j/N, from Table 16. Use Equations 4.1.3–3 and 4.1.3–4, respectively, to evaluate $\dot{Q}_c^{k=2}(T_j)$ and $\dot{E}_c^{k=2}(T_j)$. When evaluating the above equation for part load factor at high capacity, use the same value of C_D^c as used in the section 4.1.3.1 calculations.

4.1.3.4 Unit must operate continuously at high (k=2) compressor capacity at temperature T_j, $BL(T_j) \geq \dot{Q}_c^{k=2}(T_j)$.

$$\frac{q_c(T_j)}{N} = \dot{Q}_c^{k=2}(T_j) \cdot \frac{n_j}{N}$$

$$\frac{e_c(T_j)}{N} = \dot{E}_c^{k=2}(T_j) \cdot \frac{n_j}{N}.$$

Obtain the fractional bin hours for the cooling season, n_j/N, from Table 16. Use Equations 4.1.3–3 and 4.1.3–4, respectively, to evaluate $\dot{Q}_c^{k=2}(T_j)$ and $\dot{E}_c^{k=2}(T_j)$.

4.1.4 SEER calculations for an air conditioner or heat pump having a variable-speed compressor. Calculate SEER using Equation 4.1–1. Evaluate the space cooling capacity, $\dot{Q}_c^{k=1}(T_j)$, and electrical power consumption, $\dot{E}_c^{k=1}(T_j)$, of the test unit when operating at minimum compressor speed and outdoor temperature T_j. Use,

$$\dot{Q}_c^{k=1}(T_j) = \dot{Q}_c^{k=1}(67) + \frac{\dot{Q}_c^{k=1}(82) - \dot{Q}_c^{k=1}(67)}{82 - 67} \cdot (T_j - 67) \qquad (4.1.4\text{-}1)$$

$$\dot{E}_c^{k=1}(T_j) = \dot{E}_c^{k=1}(67) + \frac{\dot{E}_c^{k=1}(82) - \dot{E}_c^{k=1}(67)}{82 - 67} \cdot (T_j - 67) \qquad (4.1.4\text{-}2)$$

where $\dot{Q}_c^{k=1}(82)$ and $\dot{E}_c^{k=1}(82)$ are determined from the B_1 Test, $\dot{Q}_c^{k=1}(67)$ and $\dot{E}_c^{k=1}(67)$ are determined from the F1 Test, and all four quantities are calculated as specified in section 3.3. Evaluate the space cooling capacity, $\dot{Q}_c^{k=2}(T_j)$, and electrical power consumption, $\dot{E}_c^{k=2}(T_j)$, of the test unit when operating at maximum compressor speed and outdoor temperature T_j. Use Equations 4.1.3–3 and 4.1.3–4, respectively, where $\dot{Q}_c^{k=2}(95)$ and $\dot{E}_c^{k=2}(95)$ are determined from the A_2 Test, $\dot{Q}_c^{k=2}(82)$ and $\dot{E}_c^{k=2}(82)$ are determined from the B_2 Test, and all four quantities are calculated as specified in section 3.3. Calculate the space cooling capacity, $\dot{Q}_c^{k=v}(T_j)$, and electrical power consumption, $\dot{E}_c^{k=v}(T_j)$, of the test unit when operating at outdoor temperature T_j and the intermediate compressor speed used during the section 3.2.4 (and Table 6) E_V Test using,

$$\dot{Q}_c^{k=v}(T_j) = \dot{Q}_c^{k=v}(87) + M_Q \cdot (T_j - 87) \qquad (4.1.4\text{-}3)$$

$$\dot{E}_c^{k=v}(T_j) = \dot{E}_c^{k=v}(87) + M_E \cdot (T_j - 87) \qquad (4.1.4\text{-}4)$$

where $\dot{Q}_c^{k=v}(87)$ and $\dot{E}_c^{k=v}(87)$ are determined from the E_V Test and calculated as specified in section 3.3. Approximate the slopes of the k = v intermediate speed cooling capacity and electrical power input curves, M_Q and M_E, as follows:

$$M_Q = \left[\frac{\dot{Q}_c^{k=1}(82) - \dot{Q}_c^{k=1}(67)}{82-67} \cdot (1 - N_Q)\right] + \left[N_Q \cdot \frac{\dot{Q}_c^{k=2}(95) - \dot{Q}_c^{k=2}(82)}{95-82}\right]$$

$$M_E = \left[\frac{\dot{E}_c^{k=1}(82) - \dot{E}_c^{k=1}(67)}{82-67} \cdot (1 - N_E)\right] + \left[N_E \cdot \frac{\dot{E}_c^{k=2}(95) - \dot{E}_c^{k=2}(82)}{95-82}\right]$$

where,

$$N_Q = \frac{\dot{Q}_c^{k=v}(87) - \dot{Q}_c^{k=1}(87)}{\dot{Q}_c^{k=2}(87) - \dot{Q}_c^{k=1}(87)}, \text{ and}$$

$$N_E = \frac{\dot{E}_c^{k=v}(87) - \dot{E}_c^{k=1}(87)}{\dot{E}_c^{k=2}(87) - \dot{E}_c^{k=1}(87)}.$$

Calculating Equation 4.1–1 quantities

$$\frac{q_c(T_j)}{N} \text{ and } \frac{e_c(T_j)}{N}$$

differs depending upon whether the test unit would operate at minimum speed (section 4.1.4.1), operate at an intermediate speed (section 4.1.4.2), or operate at maximum speed (section 4.1.4.3) in responding to the building load. Use Equation 4.1–2 to calculate the building load, $BL(T_j)$, for each temperature bin.

4.1.4.1 Steady-state space cooling capacity when operating at minimum compressor speed is greater than or equal to the building cooling load at temperature T_j, $\dot{Q}_c^{k=1}(T_j) \ge BL(T_j)$.

$$\frac{q_c(T_j)}{N} = X^{k=1}(T_j) \cdot \dot{Q}_c^{k=1}(T_j) \cdot \frac{n_j}{N}$$

$$\frac{e_c(T_j)}{N} = \frac{X^{k=1}(T_j) \cdot \dot{E}_c^{k=1}(T_j)}{PLF_J} \cdot \frac{n_j}{N}$$

where,

$X^{k=1}(T_j) = BL(T_j) / \dot{Q}_c^{k=1}(T_j)$, the cooling mode minimum speed load factor for temperature bin j, dimensionless.

$PLF_j = 1 - C_D^c \cdot [1 - X^{k=1}(T_j)]$, the part load factor, dimensionless.

n_j/N = fractional bin hours for the cooling season; the ratio of the number of hours during the cooling season when the outdoor temperature fell within the range represented by bin temperature T_j to the total number of hours in the cooling season, dimensionless.

Obtain the fractional bin hours for the cooling season, n_j/N, from Table 16. Use Equations 4.1.4–1 and 4.1.4–2, respectively, to evaluate $\dot{Q}_c^{k=1}(T_j)$ and $\dot{E}_c^{k=1}(T_j)$. If the optional tests described in section 3.2.4 and Table 6 are not conducted, set the cooling mode cyclic degradation coefficient, C_D^c, to the default value specified in section 3.5.3. If these optional tests are conducted, set C_D^c to the lower of:

a. The value calculated according to section 3.5.3; or

b. The section 3.5.3 default value of 0.25.

4.1.4.2 Unit operates at an intermediate compressor speed (k=i) in order to match the building cooling load at temperature T_j, $\dot{Q}_c^{k=1}(T_j) < BL(T_j) < \dot{Q}_c^{k=2}(T_j)$.

$$\frac{q_c(T_j)}{N} = \dot{Q}_c^{k=i}(T_j) \cdot \frac{n_j}{N}$$

$$\frac{e_c(T_j)}{N} = \dot{E}_c^{k=i}(T_j) \cdot \frac{n_j}{N}$$

where,

$\dot{Q}_c^{k=i}(T_j) = BL(T_j)$, the space cooling capacity delivered by the unit in matching the building load at temperature T_j, Btu/h. The matching occurs with the unit operating at compressor speed k = i.

$$\dot{E}_c^{k=i}(T_j) = \frac{\dot{Q}_c^{k=i}(T_j)}{EER^{k=i}(T_j)},$$

the electrical power input required by the test unit when operating at a compressor speed of k = i and temperature T_j, W.

$EER^{k=i}(T_j)$ = the steady-state energy efficiency ratio of the test unit when operating at a compressor speed of k = i and temperature T_j, Btu/h per W.

Obtain the fractional bin hours for the cooling season, n_j/N, from Table 16. For each temperature bin where the unit operates at an intermediate compressor speed, determine the energy efficiency ratio $EER^{k=i}(T_j)$ using,

$EER^{k=i}(T_j) = A + B \cdot T_j + C \cdot T_j^2$.

For each unit, determine the coefficients A, B, and C by conducting the following calculations once:

$$D = \frac{T_2^2 - T_1^2}{T_v^2 - T_1^2}$$

$$B = \frac{EER^{k=1}(T_1) - EER^{k=2}(T_2) - D \cdot \left[EER^{k=1}(T_1) - EER^{k=v}(T_v)\right]}{T_1 - T_2 - D \cdot (T_1 - T_v)}$$

$$C = \frac{EER^{k=1}(T_1) - EER^{k=2}(T_2) - B \cdot (T_1 - T_2)}{T_1^2 - T_2^2}$$

$$A = EER^{k=2}(T_2) - B \cdot T_2 - C \cdot T_2^2$$

where,

T_1 = the outdoor temperature at which the unit, when operating at minimum compressor speed, provides a space cooling capacity that is equal to the building load ($\dot{Q}_c^{k=1}(T_1) = BL(T_1)$), °F. Determine T_1 by equating Equations 4.1.4–1 and 4.1–2 and solving for outdoor temperature.

T_v = the outdoor temperature at which the unit, when operating at the intermediate compressor speed used during the section 3.2.4 E_V Test, provides a space cooling capacity that is equal to the building load ($\dot{Q}_c^{k=v}(T_v) = BL(T_v)$), °F. Determine T_v by equating Equations 4.1.4–3 and 4.1–2 and solving for outdoor temperature.

T_2 = the outdoor temperature at which the unit, when operating at maximum compressor speed, provides a space cooling capacity that is equal to the building load ($\dot{Q}_c^{k=2}(T_2) = BL(T_2)$), °F. Determine T_2 by equating Equations 4.1.3–3 and 4.1–2 and solving for outdoor temperature.

$$EER^{k=1}(T_1) = \frac{\dot{Q}_c^{k=1}(T_1)\left[\text{Eqn. 4.1.4-1, substituting } T_1 \text{ for } T_j\right]}{\dot{E}_c^{k=1}(T_1)\left[\text{Eqn. 4.1.4-2, substituting } T_1 \text{ for } T_j\right]}, \text{ Btu/h per W.}$$

$$EER^{k=v}(T_v) = \frac{\dot{Q}_c^{k=v}(T_v)\left[\text{Eqn. 4.1.4-3, substituting } T_v \text{ for } T_j\right]}{\dot{E}_c^{k=v}(T_v)\left[\text{Eqn. 4.1.4-4, substituting } T_v \text{ for } T_j\right]}, \text{ Btu/h per W.}$$

$$EER^{k=2}(T_2) = \frac{\dot{Q}_c^{k=2}(T_2)\left[\text{Eqn. 4.1.3-3, substituting } T_2 \text{ for } T_j\right]}{\dot{E}_c^{k=2}(T_2)\left[\text{Eqn. 4.1.3-4, substituting } T_2 \text{ for } T_j\right]}, \text{ Btu/h per W.}$$

4.1.4.3 Unit must operate continuously at maximum (k=2) compressor speed at temperature Tj, $BL(T_j) \geq \dot{Q}_c^{k=2}(T_j)$. Evaluate the Equation 4.1–1 quantities

$$\frac{q_c(T_j)}{N} \text{ and } \frac{e_c(T_j)}{N}$$

as specified in section 4.1.3.4 with the understanding that $\dot{Q}_c^{k=2}(T_j)$ and $\dot{E}_c^{k=2}(T_j)$ correspond to maximum compressor speed operation and are derived from the results of the tests specified in section 3.2.4.

4.2 Heating Seasonal Performance Factor (HSPF) Calculations. Unless an approved alternative rating method is used, as set forth in 10 CFR 430.24(m), Subpart B, HSPF must be calculated as follows: Six generalized climatic regions are depicted in Figure 2 and otherwise defined in Table 17. For each of these regions and for each applicable standardized design heating requirement, evaluate the heating seasonal performance factor using,

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

where,

$e_h(T_j)/N$=
The ratio of the electrical energy consumed by the heat pump during periods of the space heating season when the outdoor temperature fell within the range represented by bin temperature T_j to the total number of hours in the heating season (N), W. For heat pumps having a heat comfort controller, this ratio may also include electrical energy used by resistive elements to maintain a minimum air delivery temperature (see 4.2.5).

$RH(T_j)/N$=
The ratio of the electrical energy used for resistive space heating during periods when the outdoor temperature fell within the range represented by bin temperature T_j to the total number of hours in the heating season (N), W. Except as noted in section 4.2.5, resistive space heating is modeled as being used to meet that portion of the building load that the heat pump does not meet because of insufficient capacity or because the heat pump automatically turns off at the lowest outdoor temperatures. For heat pumps having a heat comfort controller, all or part of the electrical energy used by resistive heaters at a particular bin temperature may be reflected in $e_h(T_j)/N$ (see 4.2.5).

T_j = the outdoor bin temperature, °F. Outdoor temperatures are ''binned'' such that calculations are only performed based one temperature within the bin. Bins of 5 °F are used.

n_j/N=
Fractional bin hours for the heating season; the ratio of the number of hours during the heating season when the outdoor temperature fell within the range represented by bin temperature T_j to the total number of hours in the heating season, dimensionless. Obtain n_j/N values from Table 17.

j = the bin number, dimensionless.

J = for each generalized climatic region, the total number of temperature bins, dimensionless. Referring to Table 17, J is the highest bin number (j) having a nonzero entry for the fractional bin hours for the generalized climatic region of interest.

F_{def} = the demand defrost credit described in section 3.9.2, dimensionless.

$BL(T_j)$ = the building space conditioning load corresponding to an outdoor temperature of T_j; the heating season building load also depends on the generalized climatic region's outdoor design temperature and the design heating requirement, Btu/h.

TABLE 17.—GENERALIZED CLIMATIC REGION INFORMATION

Region Number		I	II	III	IV	V	VI
Heating Load Hours, HLH		750	1250	1750	2250	2750	*2750
Outdoor Design Temperature, T_{OD}		37	27	17	5	−10	30
j	T_j (°F)	Fractional Bin Hours, n_j/N					
1	62	.291	.215	.153	.132	.106	.113
2	57	.239	.189	.142	.111	.092	.206
3	52	.194	.163	.138	.103	.086	.215

TABLE 17.—GENERALIZED CLIMATIC REGION INFORMATION—Continued

4	47	.129	.143	.137	.093	.076	.204
5	42	.081	.112	.135	.100	.078	.141
6	37	.041	.088	.118	.109	.087	.076
7	32	.019	.056	.092	.126	.102	.034
8	27	.005	.024	.047	.087	.094	.008
9	22	.001	.008	.021	.055	.074	.003
10	17	0	.002	.009	.036	.055	0
11	12	0	0	.005	.026	.047	0
12	7	0	0	.002	.013	.038	0
13	2	0	0	.001	.006	.029	0
14	−3	0	0	0	.002	.018	0
15	−8	0	0	0	.001	.010	0
16	−13	0	0	0	0	.005	0
17	−18	0	0	0	0	.002	0
18	−23	0	0	0	0	.001	0

* Pacific Coast Region.

Evaluate the building heating load using

$$BL(T_j) = \frac{(65 - T_j)}{65 - T_{OD}} \cdot C \cdot DHR \qquad (4.2\text{-}2)$$

where,

T_{OD} = the outdoor design temperature, °F. An outdoor design temperature is specified for each generalized climatic region in Table 17.

C = 0.77, a correction factor which tends to improve the agreement between calculated and measured building loads, dimensionless.

DHR = the design heating requirement (see Definition 1.22), Btu/h.

Calculate the minimum and maximum design heating requirements for each generalized climatic region as follows:

$$DHR_{min} = \left\{ \begin{array}{ll} \dot{Q}_h^k(47) \cdot \left[\dfrac{65 - T_{OD}}{60}\right], & \text{for Regions I, II, III, IV, \& VI} \\ \dot{Q}_h^k(47), & \text{for Region V} \end{array} \right\} \begin{array}{l}\text{Rounded to the nearest} \\ \text{standardized DHR} \\ \text{given in Table 18.}\end{array}$$

and

$$DHR_{max} = \left\{ \begin{array}{ll} 2 \cdot \dot{Q}_h^k(47) \cdot \left[\dfrac{65 - T_{OD}}{60}\right], & \text{for Regions I, II, III, IV, \& VI} \\ 2.2 \cdot \dot{Q}_h^k(47), & \text{for Region V} \end{array} \right\} \begin{array}{l}\text{Rounded to the nearest} \\ \text{standardized DHR} \\ \text{given in Table 18.}\end{array}$$

where $\dot{Q}_h^k(47)$ is expressed in units of Btu/h and otherwise defined as follows:

1. For a single-speed heat pump tested as per section 3.6.1, $\dot{Q}_h^k(47) = \dot{Q}_h(47)$, the space heating capacity determined from the H1 Test.

2. For a variable-speed heat pump, a section 3.6.2 single-speed heat pump, or a two-capacity heat pump not covered by item 3, $\dot{Q}_h^k(47) = \dot{Q}_h^{k=2}(47)$, the space heating capacity determined from the $H1_2$ Test.

3. For two-capacity, northern heat pumps (see Definition 1.46), $\dot{Q}^k_h(47) = \dot{Q}^{k=1}_h(47)$, the space heating capacity determined from the $H1_1$ Test.

If the optional $H1_N$ Test is conducted on a variable-speed heat pump, the manufacturer has the option of defining $\dot{Q}^k_h(47)$ as specified above in item 2 or as $\dot{Q}^k_h(47)=\dot{Q}^{k=N}_h(47)$, the space heating capacity determined from the $H1_N$ Test.

For all heat pumps, HSPF accounts for the heating delivered and the energy consumed by auxiliary resistive elements when operating below the balance point. This condition occurs when the building load exceeds the space heating capacity of the heat pump condenser. For HSPF calculations for all heat pumps, see either section 4.2.1, 4.2.2, 4.2.3, or 4.2.4, whichever applies.

For heat pumps with heat comfort controllers (see Definition 1.28), HSPF also accounts for resistive heating contributed when operating above the heat-pump-plus-comfort-controller balance point as a result of maintaining a minimum supply temperature. For heat pumps having a heat comfort controller, see section 4.2.5 for the additional steps required for calculating the HSPF.

TABLE 18.—STANDARDIZED DESIGN HEATING REQUIREMENTS (BTU/H)

5,000	25,000	50,000	90,000
10,000 ...	30,000	60,000	100,000
15,000 ...	35,000	70,000	110,000
20,000 ...	40,000	80,000	130,000

4.2.1 Additional steps for calculating the HSPF of a heat pump having a single-speed compressor that was tested with a fixed-speed indoor fan installed, a constant-air-volume-rate indoor fan installed, or with no indoor fan installed.

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

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

where,

$$X(T_j) = \begin{Bmatrix} BL(T_J)/\dot{Q}_h(T_j) \\ \text{or} \\ 1 \end{Bmatrix}$$

whichever is less; the heating mode load factor for temperature bin j, dimensionless.

$\dot{Q}_h(T_j)$ = the space heating capacity of the heat pump when operating at outdoor temperature T_j, Btu/h.

$\dot{E}_h(T_j)$ = the electrical power consumption of the heat pump when operating at outdoor temperature T_j, W.

$\delta(T_j)$ = the heat pump low temperature cut-out factor, dimensionless.

$PLF_j = 1 - C_D^h \cdot [1 - X(T_j)]$ the part load factor, dimensionless.

Use Equation 4.2–2 to determine $BL(T_j)$. Obtain fractional bin hours for the heating season, n_j/N, from Table 17. If the optional H1C Test described in section 3.6.1 is not conducted, set the heating mode cyclic degradation coefficient, C_D^h, to the default value specified in section 3.8.1. If this optional test is conducted, set C_D^h to the lower of:

a. The value calculated according to section 3.8.1 or

b. The section 3.8.1 default value of 0.25.

Determine the low temperature cut-out factor using

$$\delta(T_j) = \begin{cases} 0, \text{ if } T_j \le T_{off} \text{ or } \dfrac{\dot{Q}_h(T_j)}{3.413 \cdot \dot{E}_h(T_j)} < 1 \\ 1/2, \text{ if } T_{off} < T_j \le T_{on} \text{ and } \dfrac{\dot{Q}_h(T_j)}{3.413 \cdot \dot{E}_h(T_j)} \ge 1 \\ 1, \text{ if } T_j > T_{on} \text{ and } \dfrac{\dot{Q}_h(T_j)}{3.413 \cdot \dot{E}_h(T_j)} \ge 1 \end{cases} \qquad (4.2.1\text{-}3)$$

where,

T_{off} = the outdoor temperature when the compressor is automatically shut off, °F. (If no such temperature exists, T_j is always greater than T_{off} and T_{on}).

T_{on} = the outdoor temperature when the compressor is automatically turned back on, if applicable, following an automatic shut-off, °F.

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

$$\dot{Q}_h(T_j) = \begin{cases} \dot{Q}_h(17) + \dfrac{\left[\dot{Q}_h(47) - \dot{Q}_h(17)\right] \cdot (T_j - 17)}{47 - 17}, \text{ if } T_j \ge 45\ °F \text{ or } T_j \le 17\ °F \\ \dot{Q}_h(17) + \dfrac{\left[\dot{Q}_h(35) - \dot{Q}_h(17)\right] \cdot (T_j - 17)}{35 - 17}, \text{ if } 17\ °F < T_j < 45\ °F \end{cases} \qquad (4.2.1\text{-}4)$$

$$\dot{E}_h(T_j) = \begin{cases} \dot{E}_h(17) + \dfrac{\left[\dot{E}_h(47) - \dot{E}_h(17)\right] \cdot (T_j - 17)}{47 - 17}, \text{ if } T_j \ge 45\ °F \text{ or } T_j \le 17\ °F \\ \dot{E}_h(17) + \dfrac{\left[\dot{E}_h(35) - \dot{E}_h(17)\right] \cdot (T_j - 17)}{35 - 17}, \text{ if } 17\ °F < T_j < 45\ °F \end{cases} \qquad (4.2.1\text{-}5)$$

where $\dot{Q}_h(47)$ and $\dot{E}_h(47)$ are determined from the H1 Test and calculated as specified in section 3.7; $\dot{Q}_h(35)$ and $\dot{E}_h(35)$ are determined from the H2 Test and calculated as specified in section 3.9.1; and $\dot{Q}_h(17)$ and $\dot{E}_h(17)$ are determined from the H3 Test and calculated as specified in section 3.10.

4.2.2 Additional steps for calculating the HSPF of a heat pump having a single-speed compressor and a variable-speed, variable-air-volume-rate indoor fan. The manufacturer must provide information about how the indoor air volume rate or the indoor fan speed varies over the outdoor temperature range of 65 °F to −23 °F. Calculate the quantities

$$\frac{e_h(T_j)}{N} \text{ and } \frac{RH(T_j)}{N}$$

in Equation 4.2–1 as specified in section 4.2.1 with the exception of replacing references to the H1C Test and section 3.6.1 with the $H1C_1$ Test and section 3.6.2. In addition, evaluate the space heating capacity and electrical power consumption of the heat pump $\dot{Q}_h(T_j)$ and $\dot{E}_h(T_j)$ using

$$\dot{Q}_h(T_j) = \dot{Q}_h^{k=1}(T_j) + \frac{\dot{Q}_h^{k=2}(T_j) - \dot{Q}_h^{k=1}(T_j)}{FP_h^{k=2} - FP_h^{k=1}} \cdot \left[FP_h(T_j) - FP_h^{k=1}\right] \qquad (4.2.2\text{-}1)$$

$$\dot{E}_h(T_j) = \dot{E}_h^{k=1}(T_j) + \frac{\dot{E}_h^{k=2}(T_j) - \dot{E}_h^{k=1}(T_j)}{FP_h^{k=2} - FP_h^{k=1}} \cdot \left[FP_h(T_j) - FP_h^{k=1}\right] \qquad (4.2.2\text{-}2)$$

where the space heating capacity and electrical power consumption at both low capacity (k=1) and high capacity (k=2) at outdoor temperature Tj are determined using

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

$$\dot{E}_h^k(T_j) = \begin{cases} E_h^k(17) + \dfrac{\left[\dot{E}_h^k(47) - \dot{E}_h^k(17)\right] \cdot (T_j - 17)}{47 - 17}, \text{ if } T_j \geq 45\ ^\circ\text{F or } T_j \leq 17\ ^\circ\text{F} \\ \dot{E}_h^k(17) + \dfrac{\left[\dot{E}_h^k(35) - \dot{E}_h^k(17)\right] \cdot (T_j - 17)}{35 - 17}, \text{ if } 17\ ^\circ\text{F} < T_j < 45\ ^\circ\text{F} \end{cases} \qquad (4.2.2\text{-}4)$$

For units where indoor fan speed is the primary control variable, $FP_h^{k=1}$ denotes the fan speed used during the required $H1_1$ and $H3_1$ Tests (see Table 10), $FP_h^{k=2}$ denotes the fan speed used during the required $H1_2$, $H2_2$, and $H3_2$ Tests, and $FP_h(T_j)$ denotes the fan speed used by the unit when the outdoor temperature equals T_j. For units where indoor air volume rate is the primary control variable, the three FP_h's are similarly defined only now being expressed in terms of air volume rates rather than fan speeds. Determine $\dot{Q}_h^{k=1}(47)$ and $\dot{E}_h^{k=1}(47)$ from the $H1_1$ Test, and $\dot{Q}_h^{k=2}(47)$ and $\dot{E}_h^{k=2}(47)$ from the $H1_2$ Test. Calculate all four quantities as specified in section 3.7. Determine $\dot{Q}_h^{k=1}(35)$ and $\dot{E}_h^{k=1}(35)$ as specified in section 3.6.2; determine $\dot{Q}_h^{k=2}(35)$ and $\dot{E}_h^{k=2}(35)$ and from the $H2_2$ Test and the calculation specified in section 3.9. Determine $\dot{Q}_h^{k=1}(17)$ and $\dot{E}_h^{k=1}(17$ from the $H3_1$ Test, and $\dot{Q}_h^{k=2}(17)$ and $\dot{E}_h^{k=2}(17)$ from the $H3_2$ Test. Calculate all four quantities as specified in section 3.10.

4.2.3 Additional steps for calculating the HSPF of a heat pump having a two-capacity compressor. The calculation of the Equation 4.2–1 quantities

$$\frac{e_h(T_j)}{N} \text{ and } \frac{RH(T_j)}{N}$$

differs depending upon whether the heat pump would operate at low capacity (section 4.2.3.1), cycle between low and high capacity (Section 4.2.3.2), or operate at high capacity (sections 4.2.3.3 and 4.2.3.4) in responding to the building load. For heat pumps that lock out low capacity operation at low outdoor temperatures, the manufacturer must supply information regarding the cutoff temperature(s) so that the appropriate equations can be selected.

a. Evaluate the space heating capacity and electrical power consumption of the heat pump when operating at low compressor capacity and outdoor temperature T_j using

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

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

b. Evaluate the space heating capacity and electrical power consumption ($\dot{Q}_h^{k=2}(T_j)$ and $\dot{E}_h^{k=2}(T_j)$) of the heat pump when operating at high compressor capacity and outdoor temperature Tj by solving Equations 4.2.2–3 and 4.2.2–4, respectively, for k=2. Determine $\dot{Q}_h^{k=1}(62)$ and $\dot{E}_h^{k=1}(62)$ from the $H0_1$ Test, $\dot{Q}_h^{k=1}(47)$ and $\dot{E}_h^{k=1}(47)$ from the $H1_1$ Test, and $\dot{Q}_h^{k=2}(47)$ and $\dot{E}_h^{k=2}(47)$ from the $H1_2$ Test. Calculate all six quantities as specified in section 3.7. Determine $\dot{Q}_h^{k=2}(35)$ and $\dot{E}_h^{k=2}(35)$ from the $H2_2$ Test and, if required as described in section 3.6.3, determine $\dot{Q}_h^{k=1}(35)$ and $\dot{E}_h^{k=1}(35)$ from the $H2_1$ Test. Calculate the required 35 °F quantities as specified in section 3.9. Determine $\dot{Q}_h^{k=2}(17)$ and $\dot{E}_h^{k=2}(17)$ from the $H3_2$ Test and, if required as described in section 3.6.3, determine $\dot{Q}_h^{k=1}(17)$ and $\dot{E}_h^{k=1}(17)$ from the $H3_1$ Test. Calculate the required 17 °F quantities as specified in section 3.10.

4.2.3.1 Steady-state space heating capacity when operating at low compressor capacity is greater than or equal to the building heating load at temperature T_j, $\dot{Q}_h^{k=1}(T_j) \geq BL(T_j)$.

$$\frac{e_h(T_j)}{N} = \frac{X^{k=1}(T_j)\cdot \dot{E}_h^{k=1}(T_j)\cdot \delta'(T_j)}{PLF_j}\cdot\frac{n_j}{N} \qquad (4.2.3\text{-}1)$$

$$\frac{RH(T_j)}{N} = \frac{BL(T_j)\cdot\left[1 - \delta'(T_j)\right]}{3.413\dfrac{Btu/h}{W}}\cdot\frac{n_j}{N} \qquad (4.2.3\text{-}2)$$

where,

$X^{k=1}(T_j) = BL(T_j) / \dot{Q}_h^{k=1}(T_j)$, the heating mode low capacity load factor for temperature bin *j*, dimensionless.

$PLF_j = 1 - C_D^h \cdot [\,1 - X^{k=1}(T_j)\,]$, the part load factor, dimensionless.

$\delta'(T_j)$ = the low temperature cutoff factor, dimensionless.

If the optional $H0C_1$ Test described in section 3.6.3 is not conducted, set the heating mode cyclic degradation coefficient, C_D^h, to the default value specified in section 3.8.1. If this optional test is conducted, set C_D^h to the lower of:

a. The value calculated according to section 3.8.1; or

b. The section 3.8.1 default value of 0.25.

Determine the low temperature cut-out factor using

$$\delta'(T_j) = \begin{cases} 0, & \text{if } T_j \leq T_{off} \\ 1/2, & \text{if } T_{off} < T_j \leq T_{on} \\ 1, & \text{if } T_j > T_{on} \end{cases} \qquad (4.2.3\text{-}3)$$

where T_{off} and T_{on} are defined in section 4.2.1. Use the calculations given in section 4.2.3.3, and not the above, if:

(a) The heat pump locks out low capacity operation at low outdoor temperatures and

(b) T_j is below this lockout threshold temperature.

4.2.3.2 Heat pump alternates between high (k=2) and low (k=1) compressor capacity to satisfy the building heating load

at a temperature T_j, $\dot{Q}_h^{k=1}(T_j) < BL(T_j) < \dot{Q}_h^{k=2}(T_j)$.
Calculate

$$\frac{RH(T_j)}{N}$$

using Equation 4.2.3–2. Evaluate

$$\frac{e_h(T_j)}{N}$$

using

$$\frac{e_h(T_j)}{N} = \left[X^{k=1}(T_j) \cdot \dot{E}_h^{k=1}(T_j) + X^{k=2}(T_j) \cdot \dot{E}_h^{k=2}(T_j)\right] \cdot \delta'(T_j) \cdot \frac{n_j}{N}$$

where,

$$X^{k=1}(T_j) = \frac{\dot{Q}_h^{k=2}(T_j) - BL(T_j)}{\dot{Q}_h^{k=2}(T_j) - \dot{Q}_h^{k=1}(T_j)}$$

$X^{k=2}(T_j) = 1 - X^{k=1}(T_j)$ the heating mode, high capacity load factor for temperature bin $_j$, dimensionless.

Determine the low temperature cut-out factor, $\delta'(T_j)$, using Equation 4.2.3–3.

4.2.3.3 Heat pump only operates at high (k=2) compressor capacity at temperature T_j and its capacity is greater than the building heating load, $BL(T_j) < \dot{Q}_h^{k=2}(T_j)$. This section applies to units that lock out low compressor capacity operation at low outdoor temperatures. Calculate

$$\frac{RH(T_j)}{N}$$

using Equation 4.2.3–2. Evaluate

$$\frac{e_h(T_j)}{N}$$

using

$$\frac{e_h(T_j)}{N} = \frac{X^{k=2}(T_j) \cdot \dot{E}_h^{k=2}(T_j) \cdot \delta'(T_j)}{PLF_j} \cdot \frac{n_j}{N}$$

where,

$X^{k=2}(T_j) = BL(T_j)/\dot{Q}_h^{k=2}(T_j)$.

$PLF_j = 1 - C_D^h\,[\,1 - X^{k=2}(T_j)\,]$.

When evaluating the above equation for part load factor at high capacity, use the same value of C_D^h as used in the section 4.2.3.1 calculations. Determine the low temperature cut-out factor, $\delta'(T_j)$, using Equation 4.2.3–3.

4.2.3.4 Heat pump must operate continuously at high (k=2) compressor capacity at temperature T_j, $BL(T_j) \geq \dot{Q}_h^{k=2}(T_j)$.

$$\frac{e_h(T_j)}{N} = \dot{E}_h^{k=2}(T_j) \cdot \delta''(T_j) \cdot \frac{n_j}{N}$$

$$\frac{RH(T_j)}{N} = \frac{BL(T_j) - \left[\dot{Q}_h^{k=2}(T_j) \cdot \delta''(T_j)\right]}{3.413\ \frac{Btu/h}{W}} \cdot \frac{n_j}{N}$$

Where

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

4.2.4 Additional steps for calculating the HSPF of a heat pump having a variable-speed compressor. Calculate HSPF using Equation 4.2–1. Evaluate the space heating capacity, $\dot{Q}_h^{k=1}(T_j)$, and electrical power consumption, $\dot{E}_h^{k=1}(T_j)$, of the heat pump when operating at minimum compressor speed and outdoor temperature T_j using

$$\dot{Q}_h^{k=1}(T_j) = \dot{Q}_h^{k=1}(47) + \frac{\dot{Q}_h^{k=1}(62) - \dot{Q}_h^{k=1}(47)}{62 - 47} \cdot (T_j - 47) \qquad (4.2.4\text{-}1)$$

$$\dot{E}_h^{k=1}(T_j) = \dot{E}_h^{k=1}(47) + \frac{\dot{E}_h^{k=1}(62) - \dot{E}_h^{k=1}(47)}{62 - 47} \cdot (T_j - 47) \qquad (4.2.4\text{-}2)$$

where $\dot{Q}_h^{k=1}(62)$ and $\dot{E}_h^{k=1}(62)$ are determined from the $H0_1$ Test, $\dot{Q}_h^{k=1}(47)$ and $\dot{E}_h^{k=1}(47)$ are determined from the $H1_1$ Test, and all four quantities are calculated as specified in section 3.7. Evaluate the space heating capacity, $\dot{Q}_h^{k=2}(T_j)$, and electrical power consumption, $\dot{E}_h^{k=2}(T_j)$, of the heat pump when operating at maximum compressor speed and outdoor temperature T_j by solving Equations 4.2.2–3 and 4.2.2–4, respectively, for k=2. Determine the Equation 4.2.2–3 and 4.2.2–4 quantities $\dot{Q}_h^{k=2}(47)$ and $\dot{E}_h^{k=2}(47)$ from the $H1_2$ Test and the calculations specified in section 3.7. Determine $\dot{Q}_h^{k=2}(35)$ and $\dot{E}_h^{k=2}(35)$ from the $H2_2$ Test and the calculations specified in section 3.9 or, if the $H2_2$ Test is not conducted, by conducting the calculations specified in section 3.6.4. Determine $\dot{Q}_h^{k=2}(17)$ and $\dot{E}_h^{k=2}(17)$ from the $H3_2$ Test and the calculations specified in section 3.10. Calculate the space heating capacity, $\dot{Q}_h^{k=v}(T_j)$, and electrical power consumption, $\dot{E}_h^{k=v}(T_j)$, of the heat pump when operating at outdoor temperature T_j and the intermediate compressor speed used during the section 3.6.4 $H2_V$ Test using

$$\dot{Q}_h^{k=v}(T_j) = \dot{Q}_h^{k=v}(35) + M_Q \cdot (T_j - 35) \qquad (4.2.4\text{-}3)$$

$$\dot{E}_h^{k=v}(T_j) = \dot{E}_h^{k=v}(35) + M_E \cdot (T_j - 35) \qquad (4.2.4\text{-}4)$$

where $\dot{Q}_h^{k=v}(35)$ and $\dot{E}_h^{k=v}(35)$ are determined from the H2$_V$ Test and calculated as specified in section 3.9. Approximate the slopes of the k=v intermediate speed heating capacity and electrical power input curves, M_Q and M_E, as follows:

$$M_Q = \left[\frac{\dot{Q}_h^{k=1}(62) - \dot{Q}_h^{k=1}(47)}{62-47} \cdot (1 - N_Q)\right] + \left[N_Q \cdot \frac{\dot{Q}_h^{k=2}(35) - \dot{Q}_h^{k=2}(17)}{35-17}\right]$$

$$M_E = \left[\frac{\dot{E}_h^{k=1}(62) - \dot{E}_h^{k=1}(47)}{62-47} \cdot (1 - N_E)\right] + \left[N_E \cdot \frac{\dot{E}_h^{k=2}(35) - \dot{E}_h^{k=2}(17)}{35-17}\right]$$

where,

$$N_Q = \frac{\dot{Q}_h^{k=v}(35) - \dot{Q}_h^{k=1}(35)}{\dot{Q}_h^{k=2}(35) - \dot{Q}_h^{k=1}(35)}, \text{ and}$$

$$N_E = \frac{\dot{E}_h^{k=v}(35) - \dot{E}_h^{k=1}(35)}{\dot{E}_h^{k=2}(35) - \dot{E}_h^{k=1}(35)}.$$

Use Equations 4.2.4–1 and 4.2.4–2, respectively, to calculate $\dot{Q}_h^{k=1}(35)$ and $\dot{E}_h^{k=1}(35)$.

The calculation of Equation 4.2–1 quantities

$$\frac{e_h(T_j)}{N} \text{ and } \frac{RH(T_j)}{N}$$

differs depending upon whether the heat pump would operate at minimum speed (section 4.2.4.1), operate at an intermediate speed (section 4.2.4.2), or operate at maximum speed (section 4.2.4.3) in responding to the building load.

4.2.4.1 Steady-state space heating capacity when operating at minimum compressor speed is greater than or equal to the building heating load at temperature T_j, $\dot{Q}_h^{k=1}(T_j \geq BL(T_j)$. Evaluate the Equation 4.2–1 quantities

$$\frac{e_h(T_j)}{N} \text{ and } \frac{RH(T_j)}{N}$$

as specified in section 4.2.3.1. Except now use Equations 4.2.4–1 and 4.2.4–2 to evaluate $\dot{Q}_h^{k=1}(T_j)$ and $\dot{E}_h^{k=1}(T_j)$, respectively, and replace section 4.2.3.1 references to ''low capacity'' and section 3.6.3 with ''minimum speed'' and section 3.6.4. Also, the last sentence of section 4.2.3.1 does not apply.

4.2.4.2 Heat pump operates at an intermediate compressor speed (k=i) in order to match the building heating load at a temperature T_j, $\dot{Q}_h^{k=1}(T_j) < BL(T_j) < \dot{Q}_h^{k=2}(T_j)$. Calculate

$$\frac{RH(T_j)}{N}$$

using Equation 4.2.3–2 while evaluating

$$\frac{e_h(T_j)}{N}$$

using,

$$\frac{e_h(T_j)}{N} = \dot{E}_h^{k=1}(T_j) \cdot \delta'(T_j) \cdot \frac{n_j}{N}$$

where,

$$\dot{E}_h^{k=i}(T_j) = \frac{\dot{Q}_h^{k=i}(T_j)}{3.413 \frac{Btu/h}{W} \cdot COP^{k=i}(T_j)}$$

and $\delta(T_j)$ is evaluated using Equation 4.2.3–3 while,

$\dot{Q}_h^{k=i}(T_j) = BL(T_j)$, the space heating capacity delivered by the unit in matching the building load at temperature (T_j), Btu/h. The matching occurs with the heat pump operating at compressor speed k=i.

$COP^{k=i}(T_j)$ = the steady-state coefficient of performance of the heat pump when operating at compressor speed k=i and temperature T_j, dimensionless.

For each temperature bin where the heat pump operates at an intermediate compressor speed, determine $COP^{k=i}(T_j)$ using,

$COP^{k=i}(T_j) = A + B \,.\, T_j + C \,.\, T_j^2$.

For each heat pump, determine the coefficients A, B, and C by conducting the following calculations once:

$$D = \frac{T_3^2 - T_4^2}{T_{vh}^2 - T_4^2}$$

$$B = \frac{COP^{k=2}(T_4) - COP^{k=1}(T_3) - D \cdot \left[COP^{k=2}(T_4) - COP^{k=v}(T_{vh})\right]}{T_4 - T_3 - D \cdot (T_4 - T_{vh})}$$

where,

T_3 = the outdoor temperature at which the heat pump, when operating at minimum compressor speed, provides a space heating capacity that is equal to the building load ($\dot{Q}_h^{k=1}(T_3) = BL(T_3)$), °F.

Determine T_3 by equating Equations 4.2.4–1 and 4.2–2 and solving for:

$$C = \frac{COP^{k=2}\left(T_4\right) - COP^{k=1}\left(T_3\right) - B \cdot \left(T_4 - T_3\right)}{T_4^2 - T_3^2}$$

$$A = COP^{k=2}\left(T_4\right) - B \cdot T_4 - C \cdot T_4^2.$$

outdoor temperature.

T_{vh} = the outdoor temperature at which the heat pump, when operating at the intermediate compressor speed used during the section 3.6.4 $H2_V$ Test, provides a space heating capacity that is equal to the building load ($\dot{Q}_h^{k=v}(T_{vh}) = BL(T_{vh})$), °F. Determine T_{vh} by equating Equations 4.2.4–3 and 4.2–2 and solving for outdoor temperature.

T_4 = the outdoor temperature at which the heat pump, when operating at maximum compressor speed, provides a space heating capacity that is equal to the building load ($\dot{Q}_h^{k=2}(T_4) = BL(T_4)$), °F. Determine T_4 by equating Equations 4.2.2–3 (k=2) and 4.2–2 and solving for outdoor temperature.

$$COP^{k=1}\left(T_3\right) = \frac{\dot{Q}_h^{k=1}\left(T_3\right)\left[\text{Eqn. 4.2.4-1, substituting } T_3 \text{ for } T_j\right]}{3.413\frac{\text{Btu/h}}{\text{W}} \cdot \dot{E}_h^{k=1}\left(T_3\right)\left[\text{Eqn. 4.2.4-2, substituting } T_3 \text{ for } T_j\right]}$$

$$COP^{k=v}\left(T_{vh}\right) = \frac{\dot{Q}_h^{k=v}\left(T_{vh}\right)\left[\text{Eqn. 4.2.4-3, substituting } T_{vh} \text{ for } T_j\right]}{3.413\frac{\text{Btu/h}}{\text{W}} \cdot \dot{E}_h^{k=v}\left(T_{vh}\right)\left[\text{Eqn. 4.2.4-4, substituting } T_{vh} \text{ for } T_j\right]}$$

$$COP^{k=2}\left(T_4\right) = \frac{\dot{Q}_h^{k=2}\left(T_4\right)\left[\text{Eqn. 4.2.2-3, substituting } T_4 \text{ for } T_j\right]}{3.413\frac{\text{Btu/h}}{\text{W}} \cdot \dot{E}_h^{k=2}\left(T_4\right)\left[\text{Eqn. 4.2.2-4, substituting } T_4 \text{ for } T_j\right]}$$

4.2.4.3 Heat pump must operate continuously at maximum (k=2) compressor speed at temperature T_j, $BL(T_j) \geq \dot{Q}_h^{k=2}(T_j)$. Evaluate the Equation 4.2–1 quantities

$$\frac{e_h\left(T_j\right)}{N} \text{ and } \frac{RH\left(T_j\right)}{N}$$

as specified in section 4.2.3.4 with the understanding that $\dot{Q}_h^{k=2}(T_j)$ and $\dot{E}_h^{k=2}(T_j)$ correspond to maximum compressor speed operation and are derived from the results of the specified section 3.6.4 tests.

4.2.5 Heat pumps having a heat comfort controller. Heat pumps having heat comfort controllers, when set to maintain a typical minimum air delivery temperature, will cause the heat pump condenser to operate less because of a greater contribution from the resistive elements. With a conventional heat pump, resistive heating is only initiated if the heat pump condenser cannot meet the building load (*i.e.*, is delayed until a second stage call from the indoor thermostat). With a heat comfort controller, resistive heating can occur even though the heat pump condenser has adequate capacity to meet the building load (*i.e.*, both on during a first stage call from the indoor thermostat). As a result, the outdoor temperature where the heat pump compressor no longer cycles (*i.e.*, starts to run continuously), will be lower than if the heat pump did not have the heat comfort controller.

4.2.5.1 Heat pump having a heat comfort controller: additional steps for calculating the HSPF of a heat pump having a single-speed compressor that was tested with a fixed-speed indoor fan installed, a constant-air-volume-rate indoor fan installed, or with no indoor fan installed. Calculate the space heating capacity and electrical power of the heat pump without the heat comfort controller being active as specified in section 4.2.1 (Equations 4.2.1–4 and 4.2.1–5) for each outdoor bin temperature, T_j, that is listed in Table 17. Denote these capacities and electrical powers by using the subscript ''hp'' instead of ''h.'' Calculate the mass flow rate (expressed in pounds-mass of dry air per hour) and the specific heat of the indoor air (expressed in $\text{Btu/lbm}_{da} \cdot$ °F) from the results of the H1 Test using:

$$\dot{m}_{da} = \overline{\dot{V}}_s \cdot 0.075\ \frac{1\text{bm}_{da}}{\text{ft}^3} \cdot \frac{60\text{ min}}{\text{hr}} = \frac{\overline{\dot{V}}_{mx}}{v_n' \cdot \left[1 + W_n\right]} \cdot \frac{60\text{ min}}{\text{hr}} = \frac{\overline{\dot{V}}_{mx}}{v_n} \cdot \frac{60\text{ min}}{\text{hr}}$$

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

where $\overline{\dot{V}}_s$, $\overline{\dot{V}}_{mx}$, v'_n (or v_n), and W_n are defined following Equation 3–1. For each outdoor bin temperature listed in Table 17, calculate the nominal temperature of the air leaving the heat pump condenser coil using,

$$T_o\left(T_j\right) = 70\ °F + \frac{\dot{Q}_{hp}\left(T_j\right)}{\dot{m}_{da} \cdot C_{p,da}}.$$

Evaluate $e_h(T_j/N)$, $RH(T_j)/N$, $X(T_j)$, PLF_j, and $\delta(T_j)$ as specified in section 4.2.1. For each bin calculation, use the space heating capacity and electrical power from Case 1 or Case 2, whichever applies.

Case 1. For outdoor bin temperatures where $T_o(T_j)$ is equal to or greater than T_{CC} (the maximum supply temperature determined according to section 3.1.9), determine $\dot{Q}_h(T_j)$ and $\dot{E}_h(T_j)$ as specified in section 4.2.1 (*i.e.*, $\dot{Q}_h(T_j) = \dot{Q}_{hp}(T_j)$ and $\dot{E}_{hp}(T_j) = \dot{E}_{hp}(T_j)$). Note: Even though $T_o(T_j) \geq T_{cc}$, resistive heating may be required; evaluate Equation 4.2.1–2 for all bins.

Case 2. For outdoor bin temperatures where $T_o(T_j) > T_{cc}$, determine $\dot{Q}_h(T_j)$ and $\dot{E}_h(T_j)$ using,

$$\dot{Q}_h\left(T_j\right) = \dot{Q}_{hp}\left(T_j\right) + \dot{Q}_{CC}\left(T_j\right)$$

$$\dot{E}_h\left(T_j\right) = \dot{E}_{hp}\left(T_j\right) + \dot{E}_{CC}\left(T_j\right)$$

where,

$$\dot{Q}_{CC}(T_j) = \dot{m}_{da} \cdot C_{p,da} \cdot \left[T_{CC} - T_o(T_j)\right]$$

$$\dot{E}_{CC}(T_j) = \frac{\dot{Q}_{CC}(T_j)}{3.413 \frac{Btu}{W \cdot h}}$$

Note: Even though $T_o(T_j) < T_{cc}$, additional resistive heating may be required; evaluate Equation 4.2.1–2 for all bins.

4.2.5.2 Heat pump having a heat comfort controller: additional steps for calculating the HSPF of a heat pump having a single-speed compressor and a variable-speed, variable-air-volume-rate indoor fan. Calculate the space heating capacity and electrical power of the heat pump without the heat comfort controller being active as specified in section 4.2.2 (Equations 4.2.2–1 and 4.2.2–2) for each outdoor bin temperature, T_j, that is listed in Table 17. Denote these capacities and electrical powers by using the subscript ''hp'' instead of ''h.'' Calculate the mass flow rate (expressed in pounds-mass of dry air per hour) and the specific heat of the indoor air (expressed in Btu/lbm$_{da}$ · °F) from the results of the H1$_2$ Test using:

$$\dot{m}_{da} = \overline{\dot{V}}_s \cdot 0.075 \frac{1bm_{da}}{ft^3} \cdot \frac{60\ min}{hr} = \frac{\overline{\dot{V}}_{mx}}{v'_n \cdot \left[1 + W_n\right]} \cdot \frac{60\ min}{hr} = \frac{\overline{\dot{V}}_{mx}}{v_n} \cdot \frac{60\ min}{hr}$$

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

where $\overline{\dot{V}}_S$, $\overline{\dot{V}}_{mx}$, v'_n (or v_n), and W_n are defined following Equation 3–1. For each outdoor bin temperature listed in Table 17, calculate the nominal temperature of the air leaving the heat pump condenser coil using,

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

Evaluate $e_h(T_j)/N$, $RH(T_j)/N$, $X(T_j)$, PLF_j, and $\delta(T_j)$ as specified in section 4.2.1 with the exception of replacing references to the H1C Test and section 3.6.1 with the H1C$_1$ Test and section 3.6.2. For each bin calculation, use the space heating capacity and electrical power from Case 1 or Case 2, whichever applies.

Case 1. For outdoor bin temperatures where $T_o(T_j)$ is equal to or greater than T_{CC} (the maximum supply temperature determined according to section 3.1.9), determine $\dot{Q}_h(T_j)$ and $\dot{E}_h(T_j)$ as specified in section 4.2.2 (*i.e.* $\dot{Q}_h(T_j) = \dot{Q}_{hp}(T_j)$ and $\dot{E}_h(T_j) = \dot{E}_{hp}(T_j)$). Note: Even though $T_o(T_j) \geq T_{CC}$, resistive heating may be required; evaluate Equation 4.2.1–2 for all bins.

Case 2. For outdoor bin temperatures where $T_o(T_j) < T_{CC}$, determine $\dot{Q}_h(T_j)$ and $\dot{E}_h(T_j)$ using,

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

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

where,

$\dot{Q}_{CC}(T_j) = \dot{m}_{da} \cdot C_{p,da} \cdot [T_{CC} - T_o(T_j)]$

$$\dot{E}_{CC}(T_j) = \frac{\dot{Q}_{CC}(T_j)}{3.413 \frac{Btu}{W \cdot h}}.$$

Note: Even though $T_o(T_j) < T_{cc}$, additional resistive heating may be required; evaluate Equation

4.2.1–2 for all bins.

4.2.5.3 Heat pumps having a heat comfort controller: additional steps for calculating the HSPF of a heat pump having a two-capacity compressor. Calculate the space heating capacity and electrical power of the heat pump without the heat comfort controller being active as specified in section 4.2.3 for both high and low capacity and at each outdoor bin temperature, T_j, that is listed in Table 17. Denote these capacities and electrical powers by using the subscript ''hp'' instead of ''h.'' For the low capacity case, calculate the mass flow rate (expressed in pounds-mass of dry air per hour) and the specific heat of the indoor air (expressed in Btu/lbm$_{da}$ · °F) from the results of the H1$_1$ Test using:

$$\dot{m}_{da}^{k=1} = \overline{\dot{V}}_s \cdot 0.075 \frac{1bm_{da}}{ft^3} \cdot \frac{60\ min}{hr} = \frac{\overline{\dot{V}}_{mx}}{v'_n \cdot \left[1 + W_n\right]} \cdot \frac{60\ min}{hr} = \frac{\overline{\dot{V}}_{mx}}{v_n} \cdot \frac{60\ min}{hr}$$

$$C_{p,da}^{k=1} = 0.24 + 0.444 \cdot W_n$$

where $\overline{\dot{V}}_s$, $\overline{\dot{V}}_{mx}$, v'_n (or v_n), and W_n are defined following Equation 3–1. For each outdoor bin temperature listed in Table 17, calculate the nominal temperature of the air leaving the heat pump condenser coil when operating at low capacity using,

$$T_o^{k=1}(T_j) = 70\ °F + \frac{\dot{Q}_{hp}^{k=1}(T_j)}{\dot{m}_{da}^{k=1} \cdot C_{p,da}^{k=1}}.$$

Repeat the above calculations to determine the mass flow rate ($\dot{m}_{da}^{k=2}$) and the specific heat of the indoor air ($C_{p,da}^{k=2}$) when operating at high capacity by using the results of the H1$_2$ Test. For each outdoor bin temperature listed in Table 17, calculate the nominal temperature of the air leaving the heat pump condenser coil when operating at high capacity using,

$$T_o^{k=2}(T_j) = 70\ °F + \frac{\dot{Q}_{hp}^{k=2}(T_j)}{\dot{m}_{da}^{k=2} \cdot C_{p,da}^{k=2}}.$$

Evaluate $e_h(T_j)/N$, $RH(T_j)/N$, $X^{k=1}(T_j)$, and/or $X^{k=2}(T_j)$, PLF_j, and $\delta'(T_j)$ or $\delta''(T_j)$ as specified in section 4.2.3.1. 4.2.3.2, 4.2.3.3, or 4.2.3.4, whichever applies, for each temperature bin. To evaluate these quantities, use the low-capacity space heating capacity and the low-capacity electrical power from Case 1 or Case 2, whichever applies; use the high-capacity space heating capacity and the high-capacity electrical power from Case 3 or Case 4, whichever applies.

Case 1. For outdoor bin temperatures where $T_o^{k=1}(T_j)$ is equal to or greater than T_{CC} (the maximum supply temperature determined according to section 3.1.9), determine $\dot{Q}_h^{k=1}(T_j)$ and $\dot{E}_h^{k=1}(T_j)$ as specified in section 4.2.3 (*i.e.*, $\dot{Q}_h^{k=1}(T_j) = \dot{Q}_{hp}^{k=1}(T_j)$ and $\dot{E}_h^{k=1}(T_j) = \dot{E}_{hp}^{k=1}(T_j)$.

Note: Even though $T_o^{k=1}(T_j) \geq T_{CC}$, resistive heating may be required; evaluate $RH(T_j)/N$ for all bins.

Case 2. For outdoor bin temperatures where $T_o^{k=1}(T_j) < T_{CC}$, determine $\dot{Q}_h^{k=1}(T_j)$ and $\dot{E}_h^{k=1}(T_j)$ using,

$\dot{Q}_h^{k=1}(T_j) = \dot{Q}_{hp}^{k=1}(T_j) + \dot{Q}_{CC}^{k=1}(T_j)$

$\dot{E}_h^{k=1}(T_j) = \dot{E}_{hp}^{k=1}(T_j) + \dot{E}_{CC}^{k=1}(T_j)$

where,

$$\dot{Q}_{CC}^{k=1}\left(T_j\right) = \dot{m}_{da}^{k=1} \cdot C_{p,da}^{k=1} \cdot \left[T_{CC} - T_o^{k=1}\left(T_j\right)\right]$$

$$\dot{E}_{CC}^{k=1}\left(T_j\right) = \frac{\dot{Q}_{CC}^{k=1}\left(T_j\right)}{3.413 \frac{Btu}{W \cdot h}}.$$

Note: Even though $T_o^{k=1}(T_j) \geq T_{cc}$, additional resistive heating may be required; evaluate $RH(T_j)/N$ for all bins.

Case 3. For outdoor bin temperatures where $T_o^{k=2}(T_j)$ is equal to or greater than T_{CC}, determine $\dot{Q}_h^{k=2}(T_j)$ and $\dot{E}_h^{k=2}(T_j)$ as specified in section 4.2.3 (*i.e.*, $\dot{Q}_h^{k=2}(T_j) = \dot{Q}_{hp}^{k=2}(T_j)$ and $\dot{E}_h^{k=2}(T_j) = \dot{E}_{hp}^{k=2}(T_j)$). Note: Even though $T_o^{k=2}(T_j) < T_{CC}$, resistive heating may be required; evaluate $RH(T_j)/N$ for all bins.

Case 4. For outdoor bin temperatures where $T_o^{k=2}(T_j) < T_{CC}$, determine $\dot{Q}_h^{k=2}(T_j)$ and $\dot{E}_h^{k=2}(T_j)$ using,

$$\dot{Q}_h^{k=2}\left(T_j\right) = \dot{Q}_{hp}^{k=2}\left(T_j\right) + \dot{Q}_{CC}^{k=2}\left(T_j\right)$$

$$\dot{E}_h^{k=2}\left(T_j\right) = \dot{E}_{hp}^{k=2}\left(T_j\right) + \dot{E}_{CC}^{k=2}\left(T_j\right)$$

where,

$$\dot{Q}_{CC}^{k=2}\left(T_j\right) = \dot{m}_{da}^{k=2} \cdot C_{p,da}^{k=2} \cdot \left[T_{CC} - T_o^{k=2}\left(T_j\right)\right]$$

$$\dot{E}_{CC}^{k=2}\left(T_j\right) = \frac{\dot{Q}_{CC}^{k=2}\left(T_j\right)}{3.413 \frac{Btu}{W \cdot h}}.$$

Note: Even though $T_o^{k=2}(T_j) < T_{cc}$, additional resistive heating may be required; evaluate $RH(T_j)/N$ for all bins.

4.2.5.4 Heat pumps having a heat comfort controller: additional steps for calculating the HSPF of a heat pump having a variable-speed compressor. [Reserved]

4.3 Calculations of the Actual and Representative Regional Annual Performance Factors for Heat Pumps.

4.3.1 Calculation of actual regional annual performance factors (APF_A) for a particular location and for each standardized design heating requirement.

$$APF_A = \frac{CLH_A \cdot \dot{Q}_c^k(95) + HLH_A \cdot DHR \cdot C}{\frac{CLH_A \cdot \dot{Q}_c^k(95)}{SEER} + \frac{HLH_A \cdot DHR \cdot C}{HSPF}}$$

where,

CLH_A = the actual cooling hours for a particular location as determined using the map given in Figure 3, hr.

$\dot{Q}_c^k(95)$ = the space cooling capacity of the unit as determined from the A or A_2 Test, whichever applies, Btu/h.

HLH_A = the actual heating hours for a particular location as determined using the map given in Figure 2, hr.

DHR = the design heating requirement used in determining the HSPF; refer to section 4.2 and Definition 1.22, Btu/h.

C = defined in section 4.2 following Equation 4.2–2, dimensionless.

SEER = the seasonal energy efficiency ratio calculated as specified in section 4.1, Btu/W·h.

HSPF = the heating seasonal performance factor calculated as specified in section 4.2 for the generalized climatic region that includes the particular location of interest (see Figure 2), Btu/W·h. The HSPF should correspond to the actual design heating requirement (DHR), if known. If it does not, it may correspond to one of the standardized design heating requirements referenced in section 4.2.

4.3.2 Calculation of representative regional annual performance factors (APF_R) for each generalized climatic region and for each standardized design heating requirement.

$$APF_R = \frac{CLH_R \cdot \dot{Q}_c^k(95) + HLH_R \cdot DHR \cdot C}{\frac{CLH_R \cdot \dot{Q}_c^k(95)}{SEER} + \frac{HLH_R \cdot DHR \cdot C}{HSPF}}$$

where,

CLH_R = the representative cooling hours for each generalized climatic region, Table 19, hr.

HLH_R = the representative heating hours for each generalized climatic region, Table 19, hr.

HSPF = the heating seasonal performance factor calculated as specified in section 4.2 for the each generalized climatic region and for each standardized design heating requirement within each region, Btu/W.h.

The SEER, $\dot{Q}_c^k(95)$, DHR, and C are the same quantities as defined in section 4.3.1. Figure 2 shows the generalized climatic regions. Table 18 lists standardized design heating requirements.

TABLE 19.—REPRESENTATIVE COOLING AND HEATING LOAD HOURS FOR EACH GENERALIZED CLIMATIC REGION

Region	CLH_R	HLH_R
I	2400	750
II	1800	1250
III	1200	1750
IV	800	2250
V	400	2750
VI	200	2750

4.4. Rounding of SEER, HSPF, and APF for reporting purposes. After calculating SEER according to section 4.1, round it off as specified in subpart B 430.23(m)(3)(i) of Title 10 of the Code of Federal Regulations. Round section 4.2 HSPF values and section 4.3 APF values as per § 430.23(m)(3)(ii) and (iii) of Title 10 of the Code of Federal Regulations.

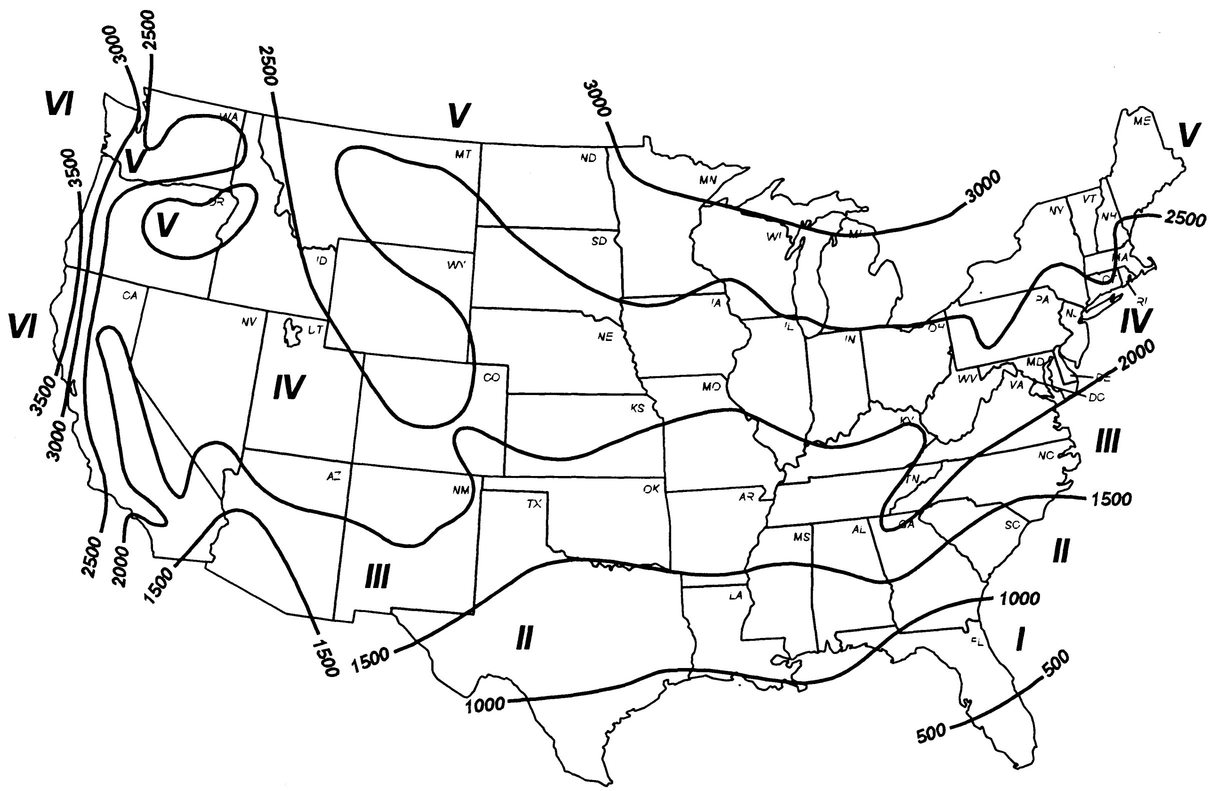

Figure 2 Heating Load Hours (HLH_A) for the United States

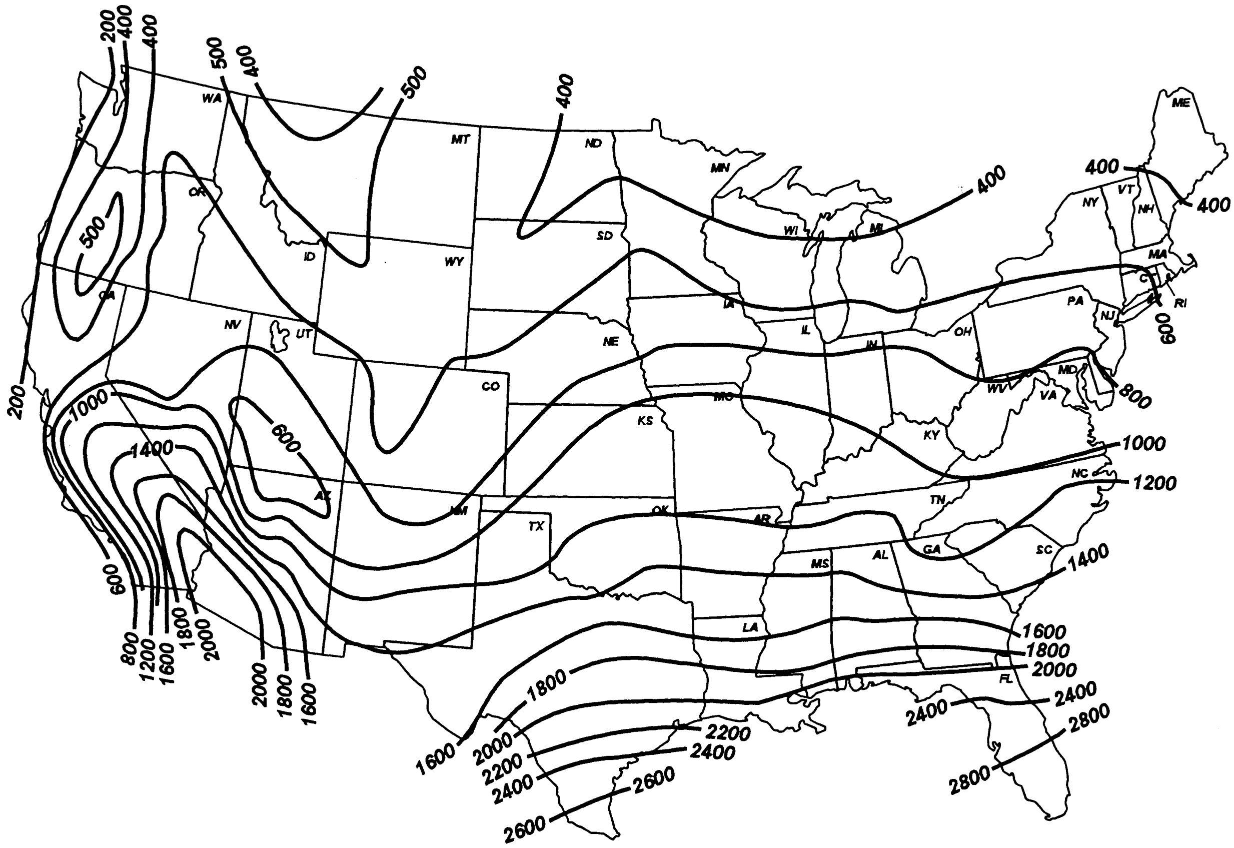

Figure 3 Cooling Load Hours (CLH_A) for the United States

■ 6. Section 430.32 of subpart C is amended by revising the section heading and adding introductory text to paragraph (c) to read as follows:

§ 430.32 Energy conservation standards and effective dates.

* * * * *

(c) *Central air conditioners and heat pumps.* The energy conservation standards defined in terms of the heating seasonal performance factor are based on Region IV, the minimum standardized design heating requirement, and the sampling plan stated in § 430.24(m).

* * * * *

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