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 Procedures for Central Air Conditioners and Heat Pumps

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

ACTION: Proposed rule and public hearing.

SUMMARY: The Department of Energy (DOE) is proposing changes to its regulations on test procedures for central air conditioners and heat pumps. Today's revision of the test procedure is not expected to alter the minimum energy conservation standards currently in effect. The revised test procedure is up-to-date, more complete and better organized than the current version. It should yield more accurate measurements of the energy efficiency of central air conditioners and heat pumps.

DATES: Comments must be received on or before March 23, 2001. DOE is requesting a signed original, a computer disk (WordPerfect 8) and 10 copies of the written comments. The Department will also accept e-mailed comments but you must send a signed original. Oral views, data, and arguments may be presented at the public workshop (hearing) in Washington, DC, beginning at 9 a.m. on February 7, 2001.

The Department must receive requests to speak at the workshop and a copy of your statements no later than 4 p.m., January 9, 2001, and we request that you provide a computer diskette (WordPerfect 8) of each statement at that time. The DOE panel will read the statements in advance of the hearing and requests that speakers limit oral presentations to a summary. Attendees will have an opportunity to ask questions.

ADDRESSES: Please submit written comments, and requests to speak at the public hearing to: Brenda Edwards-Jones, U.S. Department of Energy, Office of Energy Efficiency and Renewable Energy, Hearings and Dockets, Test Procedures for Central Air Conditioners Including Heat Pumps, Docket No. EE-RM–97–440, EE–41, Room 1J–018, Forrestal Building, 1000 Independence Avenue, SW., Washington, DC 20585–0121. You may send email to: *brenda.edwards-jones@ee.doe.gov.* The hearing will be at the U.S. Department of Energy, Forrestal Building, Room 1E–245, 1000 Independence Avenue, SW., Washington, DC. You can find more information concerning public participation in this rulemaking proceeding in section VI, ''Public Comment,'' of this notice.

You may read copies of the transcript of the public hearing and public comments at the Department of Energy Freedom of Information Reading Room, U.S. Department of Energy, Forrestal Building, Room 1E–190, 1000 Independence Avenue, SW., Washington, DC 20585, (202) 586–3142, between the hours of 9 a.m. and 4 p.m., Monday through Friday, except Federal holidays.

FOR FURTHER INFORMATION CONTACT:

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SUPPLEMENTARY INFORMATION: The proposed rule incorporates, by reference, seven test procedures 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 1991), ''Standard Method for Temperature Measurement.''
- Standard 41.2–1987 (Reaffirmed 1992), ''Standard Method for Laboratory Airflow Measurement.''
- Standard 41.6–1994, ''Standard Method for Measurement of Moist Air Properties.''
- Standard 41.9–1988, ''A Standard Calorimeter Test Method for Flow Measurement of a Volatile Refrigerant.''
- Standard 116–1995, ''Methods of Testing for Rating for Seasonal Efficiency of Unitary Air Conditioners and Heat Pumps.''

One test procedure of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers/Air Moving and Conditioning Association, Inc. (ASHRAE/AMCA) is incorporated by reference:

- Standard 51–1999, ''Laboratory Methods of Testing Fans for Rating.''

One test procedure of the Air-Conditioning and Refrigeration Institute (ARI) is incorporated by reference:

- Standard 210/240–1994, ''Unitary Air-Conditioning and Air-Source Heat Pump Equipment.''

You can view copies of these standards at the Department of Energy's Freedom of Information Reading Room at the address stated above. You can also obtain copies of the ASHRAE, ASHRAE/AMCA and ARI Standards from the American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc., 1971 Tullie Circle, NE, Atlanta, GA 30329, http://www.ashrae.org; and the Air-Conditioning and Refrigeration Institute, 4301 North Fairfax Drive, Suite 425, Arlington, VA 22203, http://www.ari.org, respectively.

I. Summary of Proposed Rule

Today's proposed rule concerns the testing aspect for central air-conditioners and central air-conditioning heat pumps. The Department develops these procedures for manufacturers to test products to measure energy efficiency, energy use, or estimated annual operating cost of a product. It will interest manufacturers, but consumers of air conditioners will see no changes due to this revision, which brings the test procedure up-to-date, and makes it more complete and better organized. Nearly all the technical content is preserved and the use of U.S. customary (i.e., inch-pound) units is maintained. Air conditioners and heat pumps that presently meet the NAECA energy conservation standards will still meet these standards when rated using the revised test procedure.

II. Introduction

A. Authority

The Energy Policy and Conservation Act requires the Department of Energy to establish the Energy Conservation Program for Consumer Products. This program sets test procedures, energy consumption and efficiency labeling, and energy conservation standards for many household, consumer products.[1] The Act requires DOE to determine to what extent a proposed test procedure would change the energy efficiency or energy use of a product from the current test procedure. If we determine that a new test procedure would change the efficiency or use of a covered product, we will amend the standard. To determine the new energy conservation standard, we measure the energy efficiency or energy use of a representative sample of covered products that minimally comply with the existing standard. The average efficiency of these representative samples, tested using the amended test procedure, constitutes the amended standard. EPCA, Section 323(e)(2).

B. Background

1. Short and Long-Term Plans

This proposed DOE test procedure is the first step of a planned two-step revision process. The immediate goal is to promulgate a revised test procedure that is up-to-date, more complete and better organized. Nearly all the technical content is preserved and the use of U.S. customary (*i.e.,* inch-pound) units is maintained. One especially important goal of this first step is to have air conditioners and heat pumps that presently meet the NAECA energy conservation standards to still meet these standards when rated using the revised test procedure.

The second step in the planned revision process is to convert the DOE test procedure to using Systeme Internationale (SI) units while maximizing compatibility with pertinent standards of the International Organization for Standardization (ISO). The goal of this second step is a DOE metric test procedure which will also meet the requirements specified by ISO for determining capacities, EER(s) for a ''moderate'' climate, and COP's. For example, DOE plans to directly reference selected ISO indoor and outdoor test conditions. However, the DOE test procedure will impose additional requirements, not found in the ISO test standards, that allow determination of the seasonal performance factors SEER and HSPF. Presently, the pertinent ISO standards are either under revision or are being newly developed so we can not yet fully determine the extent of compatibility between the DOE and ISO testing and rating procedures. DOE, via NIST personnel, is participating in the development of the ISO test standards in an effort to minimize the differences. A proposed DOE metric test procedure will be available for industry review several months after the revision of ISO standards (5151 and 13253 or, possibly a combined standard) is completed.

This two-step test procedure revision will not delay the concurrent revision of the NAECA energy conservation standards, nor will standards revision be delayed because of the planned conversion of the test procedure to SI units. Until a DOE metric test procedure has been promulgated, you will make predictions of seasonal performance using the I–P version of the DOE test procedure, *i.e.,* this revision. This revised test procedure modifies tests for certain configurations, but is not expected to impact the performance measurements. In the coming years, when a DOE metric test procedure is progressing through the rulemaking process, DOE and stakeholders will review the best time line for implementing the metric test procedure and instituting compatible NAECA energy conservation standards. As far as possible, the metric test procedure will retain the current energy efficiency descriptors, SEER and HSPF.

2. Background for Today's Proposed Rulemaking

The first DOE test procedure covering central air conditioners and heat pumps was published in the **Federal Register** on December 27, 1979, and became effective January 17, 1980. 44 FR 76700. The test procedure was modified once, in March 1988. 53 FR 8304 (March 14, 1988). Revisions made in 1988 included expanding coverage to variable-speed air conditioners and heat pumps, addressing split-type non-ducted units, and modifying the method used for crediting heat pumps that provide a demand defrost capability.

Five waivers to the DOE test procedure covering central air conditioners and heat pumps have been granted since the 1988 final rulemaking. Waivers have been granted to two different brands of non-defrost heat pumps, to two brands of combined heat pump-water heating appliances, and for a line of burner-assisted heat pumps. Non-defrost heat pumps do not contain a defrost controller and are designed to shut the compressor off under operating

[1] Public Law 94–163, as amended by the National Energy Conservation Policy Act, Public Law 95–619, the National Appliance Energy Conservation Act of 1987, Public Law 100–12, the National Appliance Energy Conservation Amendments of 1988, Public Law 100–357, and the Energy Policy Act of 1992, Public Law 102–486, Part B of Title III of Energy Policy and Conservation Act, as amended, is referred to in this proposed rule as ''EPCA'' or the ''Act.'' Part B of Title III is codified at 42 U.S.C. 6291–6309.

conditions where frost accumulation on the outdoor coil is likely. Combined appliances use an extra condensing coil to permit the unit to provide domestic water heating in addition to space conditioning. Burner-assisted heat pumps use a gas-fired burner in the outdoor coil while using electricity to power the refrigerant compressor.

In revising this test procedure, we considered whether actions could be taken to eliminate the continued need for any of the granted waivers. Today's proposed rule covers testing and calculation of HSPF for non-defrost, all-electric heat pumps, eliminating the first two of the five waivers discussed in the preceding paragraph. As the market for dual fuel heat pumps, including burner assisted heat pumps, and combined heat pump-water heating appliances grows, we will pursue the development of separate test procedures for these devices, which will eliminate the remaining three waivers.

We completed the first draft of this revised test procedure for central air conditioners and heat pumps in June 1996. The draft test procedure addressed equipment features presently not covered and improved upon the completeness and readability of the document. The June 1996 draft test procedure was distributed to members of the HVAC industry and academia for comment on the proposed changes.

Several parties provided comments on the June 1996 draft test procedures. We determined that more input on several issues would be beneficial, and DOE held a workshop on September 25, 1997. The workshop focused on five areas of concern. The first area was the identification of commercially-available equipment that is not adequately addressed in the existing test procedure. Examples include non-defrost heat pumps, heat pumps that incorporate a heat comfort controller, multi-split non-ducted heat pumps, two-capacity heat pumps that are sized to meet the space cooling load while operating at low capacity, small duct systems, and single-speed heat pumps having a variable-speed indoor fan that is modulated based on outdoor temperature. The second issue was the appropriate way to conduct steady-state and cyclic testing on units having a variable-speed, constant-air-volume-rate indoor blower. The third area of concern dealt with appropriate adjustments in order to credit a demand defrost capability and to account for the effect of barometric pressure. A group of items that pertained to specifics on lab testing procedures composed the fourth topic of discussion. Examples included how to best test packaged units having leakage, whether to limit manufacturer-specified special lab set-up requirements, recommended static pressure tap manifolding, and electrical energy measurement requirements. The fifth issue concerned the development of new defaults for the cyclic degradation coefficients, as an alternative to having to conduct tests to determine the coefficients.

A transcript of the discussions at the September 25 workshop is available for review in the DOE Freedom of Information Reading Room. The section below summarizes comments received throughout the revision process. During the workshop, several items were introduced but left unresolved. In many of these cases, ARI industry members indicated that they would offer more input and, where possible, a consensus response in the months following the workshop.

At the invitation of ARI, NIST participated in a meeting and teleconferences hosted by the ARI Unitary Small Equipment Engineering Committee in September and October of 1997 and February of 1998. For the meetings/teleconferences that followed the September 25, 1997 Workshop, discussions on DOE test procedure issues focused mainly on eleven issues, namely: (1) Small duct systems, (2) non-defrost systems, (3) multiple split heat pumps, (4) variable-speed, constant CFM blowers, (5) heat pumps that incorporate a heat comfort controller, (6) two capacity heat pumps that are sized to meet the design cooling load while operating at low speed, (7) definition for a demand defrost system, (8) effects of barometric pressure, (9) testing of packaged systems with internal leaks, (10) special laboratory setups, and (11) new default values for the cyclic degradation coefficients, C_D (the measure of performance degradation from cycling losses). Written comments were received dated 24 November 1997 from ARI (ARI, No. 6) that addressed these particular areas. ARI formed a task group to provide additional input on three items: #4, #6, and #9. ARI also hoped to provide data and a strawman approach for addressing item #11. These last four items were discussed during a February 1998 teleconference but ARI provided no consensus by the end of February, the cutoff date imposed by DOE.

The 24 November 1997 written comments from ARI are included among the overall comment summary provided below. With regard to unresolved issues associated with ARI items #4, #6, and #9, we implemented changes based on the information gathered to date. Today's rulemaking proposes no changes for the C_D defaults that may be used instead of conducting extra tests. DOE is willing to investigate and consider new C_D defaults based on the hardware features of the air conditioner or heat pump. ARI and its members have thus far provided no test data nor made any recommendations concerning the hardware features (*e.g.,* type of expansion device, with or without a time delay relay on the indoor fan, type of compressor, off-cycle power consumption, refrigerant charge quantity, rated capacity, etc.) that should be included in a statistical analysis to identify the primary factors and the associated correlations.

A draft of this proposed test procedure was posted to the Office of Codes and Standards web site in October 1998. This document was revised during the summer of 1999 to comply with the President's Memorandum of June 1, 1998, ''Plain Language in Government Writing.'' Thereafter, some sections of the proposed test procedure were reorganized and amended in response to comments received during the DOE internal review process.

In the proposed central air conditioner and heat pump standards rule (65 FR 59590, October 5, 2000), the Department discussed issues associated with mandating thermostatic expansion valves, or TXVs, to help maintain equipment performance under improper charge or airflow. In the standards final rule, we decided not to adopt a TXV requirement, but considered pursuing modifications to this test procedure to encourage the use of TXVs. Such modifications will not be part of this rulemaking, but will be considered in a separate process. Related issues that may be discussed in the separate process include the alternate rating method for mixed systems. The alternate rating method is not a part of this revision, which concerns only appendix M to subpart B of 10 CFR part 430. The alternate rating method is discussed in 10 CFR § 430.24(m). In the last revision of this test procedure in 1988, the adoption of a standard rating procedure for untested combinations of split systems was proposed, but the Department decided not to include a standard rating procedure in the test procedure rule. Instead, the Department requested the National Bureau of Standards to develop a rating method available to any manufacturer to use in rating untested combinations. Manufacturers may use this method or any other after obtaining the Department's approval. It may again be time to discuss a standard mixed system rating method included in the test procedure. These issues will be discussed in a workshop to be held in the spring of 2001.

III. Discussion of Comments

Following the September 1997 workshop, we received comments from the ARI Unitary Small Equipment Engineering Committee and individual ARI members, Proctor Engineering Group, and from the Florida Solar Energy Center. We grouped these comments into the following categories corresponding to sections of the test procedure: General, Definitions, Testing Conditions, and Testing Procedures. (ARI, No. 6, PEG, No. 3, FSEC, No. 7)

A. General

1. Non-ducted Split System Air Conditioners and Heat Pumps

Non-ducted units may use one or more indoor coils. When two or more indoor coils are used, they may operate in response to a single or multiple room thermostats. Standards of the International Organization for Standardization (ISO) differentiate non-ducted units as single or multiple room thermostat systems. We refer to equipment having one or more indoor coils all controlled by a single indoor thermostat as mini-split systems. We refer to equipment that uses two or more indoor thermostats to regulate the operation of two or more indoor coils as multi-split systems.

The current DOE test procedure does not differentiate between mini-split and multi-split systems. Both are tested and seasonal calculations are based on all indoor coils operating simultaneously. The zoning capability of multi-split units, though not operating some indoor coils, is not credited.

As part of its 1992 waiver petition, EnviroMaster International (EMI) sought ''to test its three and four zone MC/MH series systems in the manner prescribed in the DOE test for two zone systems.'' 57 FR 53736 (November 12, 1992). The modification noted in the Decision and Order was to change the wording of Section 3.1.7 to the following:

''Subsystems of multizone split-type ductless systems shall be tested as a single system. The system energy efficiency shall be based on the sum of the measured capacities of all of the zones in the system divided by the total input power used by the subsystems compressors, outdoor fans, indoor air handlers, and any additional power used by the system.''

ARI commented on this issue: ''Our members do not believe any change is necessary to the test procedures to address multiple split heat pumps. We are unaware of any unfair treatment of this product in the industry by the current test methods.'' (ARI, No. 6 at 1).

We propose no changes in today's test procedure. The option of testing each zone separately is possible. Such extra testing should provide a more complete description of the unit's capabilities. However, the benefits would have to be weighed against the considerable increase in the testing burden. The Department recommends tabling this issue until a multi-split manufacturer deems that a different and, most likely, more burdensome test approach is needed.

2. Small-duct, High-velocity Systems

Unico originally requested that DOE add a new class (or subclass) of central air conditioners and heat pumps that covered small-duct, high-velocity (SDHV) systems. Unico recommended changes to the test procedure that were coupled with DOE issuing separate NAECA standards for SDHV systems. The main test procedure changes were to impose higher minimum external static pressures and lower maximum air volume rates requirements on SDHV systems. Unico also provided a proposed definition for SDHV systems. (Unico, No. 5 at 2).

Unico noted how small duct systems differ from more conventional systems: external static is typically 1.5 inches of water, air volume rate is usually one-half of a conventional system, duct outlets into the room are typically two inches in diameter, and air velocity entering the room is in the 800 to 2000 feet per minute range. ''We feel this product is different enough that it * * * should be considered for a different class. We have different classes for room air conditioners; we have different classes for packaged terminal units and ductless systems versus ducted systems.'' (Unico, No. 2HH at 42). ARI commented: ''ARI believes that no changes to the existing test procedure are necessary for these products. They are currently tested and rated in accordance with the existing procedure. Furthermore, ARI does not believe a different product class or category should be created for small-duct systems, since that would allow for a potentially separate efficiency standard. They should be held to the same minimum efficiency standards as conventional systems. There is concern that a separate product class could open a loophole in the regulations. Other products might be specifically designed to meet the criteria of the new class, with the only intention being that they would be subjected to a less stringent efficiency standard, while still used in applications for typical equipment.'' (ARI, No. 6 at 1).

Unico later submitted an alternative proposal to DOE. In its alternative proposal, Unico plans to exercise the option of testing its line of SDHV units as coil-only units. In the Unico product line, the blower assembly is sold separately from the indoor coil assembly. The only change in the test procedure needed to implement this alternative approach is to relax the maximum pressure drop allowed when testing coil-only units. Presently, the test procedure states that the pressure drop across the indoor coil assembly must not exceed 0.30 inches of water. Unico requested that the limit be increased, preferably to 0.50 inches of water.

Today's proposed test procedure sets a higher pressure drop limit of 0.5 inches of water when testing coil-only units that meet the definition of a small-duct, high-velocity system. The proposed definition is given in section 1.46. We welcome comments on this action. Possible points for consideration include whether the action is acceptable as proposed or if incorporated in combination with a different default fan power and heat adjustment.

3. Non-defrost (Limited-range) Heat Pumps

We granted the first of two waivers for non-defrost heat pumps to Airlex in 1988. 53 FR 52216 (December 27, 1988). The waiver called for testing at 47 °F and 62 °F in lieu of testing at 35 °F and 17 °F. HSPF was calculated. Airlex, to the knowledge of DOE, has since gone out of business. We granted the second waiver to EMI in November 1992. 57 FR 53736 (November 12, 1992). Unlike Airlex, EMI did not seek to report HSPF and so did not offer proposed modifications to the DOE test procedure. We required that EMI state in its printed materials on its non-defrost products that ''no HSPF value has been measured since the heat pump cannot be operated at temperatures below 35 °F.''

At this time, non-defrost heat pumps appear to be limited to non-ducted, multi-zone, multi-split heat pumps having multiple refrigeration systems where one refrigeration system may be heating while another is cooling. In such systems, having one refrigeration system conduct a defrost while the other refrigeration system(s) is cooling is apparently quite difficult (see below EMI comment). No opposition was voiced at the workshop to a DOE proposal to cover non-defrost heat pumps in the test procedure. We also received the following comments at the workshop:

In response to the question on why EMI can not make its non-ducted, multi-refrigeration system, multi-split heat pumps defrost, EMI stated ''we have put preliminary designs together, but we've never been able to successfully control the defrost cycle while operating all the circuits.'' (EMI, No. 2HH at 23). Trane spoke against the option of creating a new class and a new NAECA HSPF energy standard for non-defrost heat pumps. (Trane, No. 2HH at 24). An ARI representative said that presently a multi-zone multi-split would be tested with all refrigeration systems operating in the same mode. (ARI, No. 2HH at 23). In written comments received following the workshop, ARI stated: ''For the same rationale as with small duct systems, ARI does not believe a separate product class or category is needed for non-defrost heat pumps. We also recommend no change in the current test procedure to accommodate non-defrost systems. As discussed above, they should be held to the same minimum efficiency standards as conventional systems.'' (ARI, No. 6 at 1).

Since the workshop, DOE received information on non-ducted, multi-refrigeration system, multi-split heat pumps made by two manufacturers other than EMI. Both of these competing multi-split products provide a defrost capability. The key differentiating feature is that these units do not provide the option of simultaneous heating and cooling like the EMI product. EMI apparently values this simultaneous cooling/heating capability more than a defrost capability while these other two manufacturers do without the simultaneous mode feature in return for being able to defrost. To date, we have found no product that provides both the simultaneous heating/cooling feature and a reverse defrost cycle capability.

From a test procedure standpoint, several options are available. One option, in line with ARI's comment, is to make no applicable changes to the test procedure and allow the existing EMI (and Airlex) waivers to remain. This option defers the issue until we receive another waiver request for a non-defrost heat pump. A second option is to make additions to the test procedure so that the HSPF of any type of non-defrost heat pump could be evaluated. A third option is to exclude heat pumps that are designed to simultaneously heat and cool ''whether they can defrost or not'' from the scope of the test procedure. The rationale for exclusion would be that such units generally compete with commercial applications where packaged terminal heat pumps and air conditioners are used and so should be tested and rated in a manner comparable to the approach used for packaged terminal equipment (*i.e.,* heating performance descriptor becomes COP at 47 °F and the equipment has no HSPF rating). For this third option, the test procedure could either be changed to cover all other non-defrost heat pumps (even though DOE knows of only the EMI simultaneously heat and cool, non-defrost heat pump) or no changes could be made, again deferring until we receive another waiver petition.

DOE requests comment on the above three and any other options for handling non-defrost heat pumps. To provide an understanding of the test procedure changes required to cover non-defrost heat pumps, we include in today's proposed test procedure (see Sections 3.6.1.1 and 4.2.1.1) the steps required to test and rate most conceivable types of single-speed, non-defrost heat pumps.

4. Heat Pumps That Incorporate a Heat Comfort Controller

Heat comfort controllers modulate the operation of the resistive elements of a heat pump to minimize temperature swings of the heated supply air when operating below the heat pump's balance point. Frequently, they seek to maintain a minimum delivery temperature when operating above the balance point. This latter application can cause the system to use more electrical energy than the heat pump alone would use to meet the building load.

At the 25 September 1997 DOE workshop, the issue was discussed at some length. The workshop members noted that the item can be both an OEM product that is an integral part of the as-shipped heat pump or it can be a field added accessory that is provided by the heat pump manufacturer or, more commonly, by a third party supplier. Also, assuming that the test procedure was modified to cover heat pumps with a heat comfort controller, no workshop invitee spoke in favor of new and separate NAECA standards for such products. The following points were also made at the workshop:

ARI stated if the manufacturer incorporates a heat comfort controller as an OEM feature, it should be covered by the test procedure. (ARI, No. 2HH at 31). Trane stated if the test procedure is modified to cover heat comfort controllers, the rating should be based on operating the controller at its maximum delivery temperature. (Trane, No. 2HH at 60). Proctor Engineering commented: ''Units designed to operate with strip heat above the balance temperature should not receive any special consideration in the test process or the [NAECA] Standard. Allowing special consideration will open the door to lower efficiencies in the field where installation errors already result in excessive strip heat use.'' (PEG, No. 3 at 3). ARI commented: ''We request DOE to develop a rating procedure for heat pumps that incorporate the use of electric resistance heat above the balance point. The procedure should be based on the highest indoor air delivery/supply temperature setting that the control system allows, so that the most conservative rating will be derived. Any heat pump that uses this feature, and still meets the minimum HSPF standard should be permitted. However, the existing ICC Model Energy Code prohibits such systems, because there is no rating method for them.'' (ARI, No. 6 at 2).

Today's proposed test procedure covers heat comfort controllers as applied to most types of single-speed heat pumps. With the heat comfort controller disabled, conduct all the same heating mode tests. Following the normally conducted heating mode test at 47 °F outdoor temperature, conduct an extra abbreviated test with the controller enabled to determine the air delivery temperature when the controller is set to its maximum setting (see Section 3.1.9). We describe proposed steps for calculating the HSPF of a single-speed heat pump having a heat comfort controller in Section 4.2.1.2.

5. Other Commercially-available Equipment that Should Be Covered in the Test Procedure

One focus of the 25 September 1997 DOE workshop was to identify commercially-available equipment that is not covered by the DOE test procedure. For the majority of equipment discussed at the workshop, we provide separate discussions elsewhere in this summary. Equipment types that were discussed and thought not to be a commercial product included: (1) Triple-capacity heat pumps and (2) units that use a two-capacity (two-stage) compressor and a variable-speed indoor fan that is modulated at each fixed stage of compressor operation.

B. Definitions

1. Revise Definition 1.20 ''Demand-defrost Control System''

ARI commented: ''We recommend that DOE expand the current ARI Standard 210/240 definition of a demand defrost system to include sampling intervals of a minimum of 10 minutes and not to have the definition pertain to time adaptive systems.'' (ARI, No. 6 at 2)

DOE's goal is to improve upon the existing definition provided in ARI Standard 210/240–94, Section A1.11, and in particular, to stop allowance of the (maximum) 3 percent HSPF credit to units that truly do not offer a demand defrost capability. We provide a proposed definition for a ''demand-defrost control system'' that seeks to be consistent with ARI's comment as Definition 1.20.

C. Testing Conditions

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

ARI commented: ''In order to provide better repeatability when testing packaged systems, which may be susceptible to internal air leakage, ARI believes it may be necessary to specify an outdoor dew point temperature when units are located in the outdoor chamber (ambient). A task group has been formed to investigate this issue and we will provide our recommendations to DOE as soon as they are available.'' (ARI, No. 6 at 2).

DOE's understanding of the impact of a leak that could result in optimistic results is as follows:

(1) Leak of outdoor air to a location upstream of the indoor coil but downstream of the test facility inlet wet bulb temperature (dew point, relative humidity) sensor. If the outdoor dew point is lower than the indoor dew point, the measured latent capacity will be higher than the true latent capacity, while the measured sensible capacity will be lower than the actual sensible capacity. The effect on total capacity will depend on dry bulb temperature of the outdoor air (82 °F or 95 °F) and the depression of the outdoor dew point relative to the indoor dew point.

(2) Leak of outdoor air to a location downstream of the indoor coil. Results are the same as (1) except that the depression of the outdoor dew point would have to be greater to overcome the negative effect on sensible capacity. The measured air volume rate on the indoor side would be higher than the actual rate at the coil and would thus increase the perceived sensible and latent capacity.

DOE's understanding of the other factors that are related to this issue are as follows. First, psychometric rooms have difficulty achieving and maintaining outdoor wet bulb temperatures in the mid 70's °F and higher during the A and B Tests. If the internal leakage is significant, obtaining a 6 percent energy balance would be difficult to achieve. Although potentially frustrating for a third party tester, the lack of an energy balance should provide impetus for the manufacturer to reduce the leakage. You can avoid the difficulty in maintaining the outdoor wet bulb temperature during the C and D dry coil tests by meeting the requirements of achieving a dry indoor coil, and by using the equation for determining sensible cooling capacity, as opposed to total cooling capacity.

In an effort to avoid potential cases where the leakage causes an optimistic result while still providing an energy balance of 6 percent or less, DOE recommends operating at an outdoor dew point temperature that is the same as the indoor dew point temperature during wet-coil tests where the unit does not reject condensate to the outdoor coil. DOE proposes a test tolerance of ±3.0 °F in the agreement of the average outdoor dew point temperature with the average indoor dew point temperature. In nominal terms, the target outdoor wet bulb temperatures will be 71.7 °F and 67.7 °F for the A and B Tests, respectively.

During heating mode tests, leaks could cause problems if the Outdoor Air Enthalpy Method is used to provide a secondary check of capacity. The proposed test procedure includes a recommendation for regulating the indoor side wet bulb temperature in an effort to minimize the difference between the indoor and outdoor-side dew point temperatures.

2. Section 2.2.5. Additional Refrigerant Charging Requirements

ARI stated: ''We believe the test procedure, as currently written, reasonably addresses the issue of special laboratory setups when conducting tests, as prescribed in manufacturer's installation instructions. Therefore, we do not recommend any change with respect to this issue.'' (ARI, No. 6 at 3).

Presently, any installation step is acceptable so long as it is specified in the manufacturer's installation instructions, including remarks that only apply if conducting laboratory testing. As discussed at the 25 September 1997 DOE workshop, the difficult issue is where to draw the line. Some special setups are justified and/or required when lab testing. DOE is only excluding special lab set-ups for refrigerant charging. With assistance from ARI and third-party laboratories, DOE will monitor test setup requirements to determine if manufacturers are specifying installation instructions inconsistent with the majority of lab installations or otherwise contrary to field practices. Furthermore, DOE is seeking assistance in establishing installation guidelines for items such as pre-washing of coils (*e.g.,* what cleaning agent to use, basic steps that specify the extent of the cleaning), run-in times on compressors, conditions where components (*e.g.,* crankcase heaters) are or are not electrically connected, exclusion of lab-only (or 25 feet only) lineset specifications, etc. These guidelines will be incorporated into future revisions of the test procedure to assist in obtaining consistency in the testing.

In today's proposed test procedure, the title of Section 2.2.5 changes from ''Exclusion of special setup requirements if stated in the manufacturer published installation manual'' to ''Additional refrigerant charging requirements.'' The section is included for two reasons. The first is to disallow the specification of two refrigerant charging criteria, one that applies for lab testing and one that applies for a field installation. The fact that a lab setting provides better quality control is not sufficient for permitting lab testing using a different charging criteria. The second reason for today's Section 2.2.5 is to avoid discrepancies and delays when third party testing is conducted. The third party testing facility should not have to consult with the manufacturer as to how the unit is to be charged. In the case of a certification failure, the issue of whether the testing facility charged the unit correctly should only be based on whether the manufacturer's charging criteria, as specified in the unit's installation instructions, were followed.

D. Testing Procedures

1. Section 3.1.4. Indoor Air Volume Rates for a Variable-Speed, Constant CFM Blower

ARI stated: ''ARI is aware of the need to consider more explicit procedures for testing units with variable speed blowers. Therefore, we have organized a task group to develop a prescribed test method for testing units with variable speed blowers, and we will pass our recommendations on to DOE as soon as they are available.'' (ARI, No. 6 at 1)

Today's proposed test procedure contains several changes from the existing test procedure to address testing of units having a variable-speed, constant CFM blower. For all tests, the exhaust fan of the air flow measuring apparatus is regulated to obtain an external static pressure that is as close to, while not being less than, the minimum external static pressure specified in the test procedure (see 3.1.4.1.1(b), 3.1.4.4.1, 3.1.4.4.2 and 3.1.4.4.3(b)). (The air flow measuring apparatus, by comparison, is not regulated to obtain the specified air volume rate, as is done when testing units having other than a constant-air-volume-rate indoor fan.) For some units, one or more tests may have to be conducted at an external static pressure that is higher than the required minimum value because of instability problems encountered when trying to reduce the external static pressure to the specified minimum. In such cases, steps are outlined for correcting the test results if the difference between the as-tested and the specified minimum external static pressure is 0.03 inches of water or more. An example of the proposed correction method is provided in the last paragraph of Sections 3.3 and 3.7. For systems that operate at multiple air volume rates, the fan laws are used to approximate the target external static pressure for tests conducted at other than the air volume rate used during the A_2 and/or $H1_2$ test.

The proposed test procedure includes a check of the agreement between the lab-measured and manufacturer-certified air volume rates. Today's proposed test procedure calls for the two values to agree within 8 percent (see 3.1.4.1.1(b), 3.1.4.2, 3.1.4.4.2, and 3.1.4.4.3(b)). This percentage is proposed based on manufacturer's comments on the variability of the variable-speed motors relative to estimates of the impact on rated performance caused by an 8 percent deviation. Using the heat pump computer modeling program HPSIM, DOE finds that an 8 percent deviation in SCFM is expected to have a negligible impact on both capacity and EER at the B Test condition while still keeping the maximum impact on capacity at the A Test condition in the 2 percent range. DOE asks that manufacturers provide feedback on the proposed 8 percent tolerance as well as findings from lab testing and computer modeling on the impact on capacity and EER of airflow changes in the 5 to 10 percent range.

Cyclic tests on units having a constant CFM blower may be conducted with or without the indoor fan enabled. If the cyclic test is conducted with the blower disabled, steps for correcting for the power draw of the blower are specified (see 3.5 and 3.5.1).

2. Section 3.1.4.1. Cooling Air Volume Rate

This issue is of interest to the ISO working group that is revising its air conditioner and heat pump test standards. The adoption of a maximum air flow limit has thus far been opposed by the majority of the ISO working group member countries. The following comments were made at the DOE workshop. A Trane representative noted that the 37.5 SCFM per 1000 Btu/h (450 SCFM per ton) maximum air flow requirement is long-standing and is of value because it (1) sets a de facto maximum sensible heat ratio and (2) keeps the air flow in a range that avoids water being blown off the wetted evaporator. (Trane, No. 2HH at 193). A representative of York International suggested reevaluating the basis for the 37.5 SCFM per 1000 Btu/h maximum while considering both full load and part load capacity conditions. (York, No. 2HH at 197).

For today's proposed revision, no change is made in the maximum air volume rate limit. DOE sees such a limit as providing a hedge against promoting efficiency gain at the expense of compromised latent capacity, especially for coil-only units. The limit also helps in having the A and B Tests conducted with a fully wetted coil that, in turn, makes the capacity fluctuations less and the collection of 30 minutes of steady-state data more readily obtainable. DOE encourages and would participate in investigations on whether this limit should be other than its present value of 37.5 SCFM per 1000 Btu/h, or whether an alternative mechanism, such as a limit on sensible heat ratio, should be considered.

Discussion of this issue is timely because no such maximum air volume limit is presently included in ISO Standards 13253 and 5151 for ducted and non-ducted air conditioners and heat pumps. A U.S. proposal to adopt the metric-equivalent of the 37.5 SCFM per 1000 Btu/h limit was voted down by the ISO working group that is presently revising ISO Standards 5151 and 13253. The vast majority of other member countries on the working group perceive air volume rate as a design parameter that should not be impacted by a rating standard. The ISO standards provide capacity test conditions that correspond to a hot, dry climate where latent capacity is not a concern. ISO also provides capacity test conditions for a cool climate. ISO working group members from countries that will rate at this cool climate condition argue that high air volume rates are needed in order to assure that the air delivery temperature is not objectionably cool. Finally, with the exception of the U.S., most countries represented on the ISO working group are predominantly concerned with non-ducted products and calorimeter testing where indoor air volume rate is not typically measured.

The goal when converting the DOE test procedure to a metric format is to make it ISO compatible. Most ducted units sold in the U.S. today are rated at an air volume rate that is less than the 37.5 SCFM per 1000 Btu/h upper limit. This fact suggests that maximum efficiency is achieved at air volume rates lower than 37.5 SCFM per 1000 Btu/h. Thus, having a upper limit may not be important enough to warrant a deviation from ISO. Either way, now is the time to discuss this issue since the revision of the ISO Standard 13253 is still underway. However, it seems unlikely that ISO will adopt an upper limit on air volume rate.

3. Section 3.1.4.1.1. External Static Pressure

Proctor Engineering Group recommended the following changes to make the test specification conform better to measurements of installed systems. When testing units having an indoor fan, ''the minimum static pressure should be revised to:

- 0.50 inches of water column for all systems, or
- The maximum allowable external static pressure specified by the manufacturer, whichever is less.''

When rating fanless units, ''the default Btu/hr (watt draw of the indoor fan motor) should be revised to 2000 Btu/hr per 1000 cfm (586 Watts per 1000 cfm).'' For comparison, the external static pressure and fan heat/power defaults presently used in the existing DOE test procedure are 0.1, 0.15, and 0.2 inches of water, with the assigned value being a function of the unit's rated capacity. The presently referenced fan heat/power default adjustment is 1250 Btu/h per 1000 SCFM (365 Watts per 1000 SCFM). Proctor Engineering Group supported its proposed changes by providing results from field measurements on 28 new systems in new construction in Phoenix, Arizona. (PEG, No. 3 at 3).

The Florida Solar Energy Center sent a report on field monitoring work which indicated that ''the standard assumption of an external static pressure of 0.2 inches of water column (IWC) for the air handler fan was far lower than the typical values encountered in the field. The average we measured in 14 evaluated installations was 0.54 IWC (range was 0.27 to 0.91 IWC).'' The commenter goes on to state his strong belief that ''the ARI test condition should be modified to 0.5 IWC to better reflect the actual performance that will be achieved by the air conditioners operating under realistic conditions. Because of this change, the watt draw of the fan motor (and heat released into the supply air stream) should also be revised to reflect the increase in fan power from this change.'' (FSEC, No. 7 at 1)

Because of concern that such changes would impact the SEER and HSPF of units that have ratings at or near the

NAECA minimum standard levels, DOE does not plan to change the static pressure requirements in this revision of the test procedure. Instead, DOE will continue dialogue with the working group that is revising ISO Standard 13253. When the revision of ISO 13253 is completed, DOE will determine the suitability of incorporating part or all of this test procedure in the DOE metric test procedure. ISO Standard 13253 is presently under revision with the present draft containing the following requirements for the minimum external static requirements. (The Inch-Pound equivalent values are not part of the proposed 13253 table but are included here to aid the reader.)

Minimum static pressures

Standard capacity ratings (kW) [kBtu/h]	Minimum external static pressure (Pa) [inches of H_2O]
0 to <8 [0 to <27.3]	25 [0.10]
8 to <12 [27.3 to <41.0]	37 [0.15]
12 to <20 [41.0 to <68.3]	50 [0.20]
20 to <30 [68.3 to <102.4]	62 [0.25]
30 to <45 [102.4 to <153.6]	75 [0.30]
45 to <82 [153.6 to <279.9]	100 [0.40]
82 to <117 [279.9 to <399.3]	125 [0.50]
117 to <147 [399.3 to <501.7]	150 [0.60]
[Above 147 Above 501.7]	175 [0.70]

The numbers up to 20 kW are consistent with the values presently cited in the existing and in this proposed revision of the DOE test procedure.

As for fanless units, the draft revision of ISO 13253 contains a thermodynamically-based equation (volume flow rate x total pressure drop divided by fan static efficiency x fan motor efficiency) to estimate default fan heat/power adjustments. Total pressure drop is taken as the sum of the following:

(1) The lab-measured pressure drop across the indoor, fanless unit

(2) The applicable minimum external static pressure listed in the above table

(3) An estimate for the pressure drop across a typical blower cabinet (=50 Pa).

The minimum external static pressure requirements thus impact both the rating for fanless and blower coil units. For residential size equipment, ISO Standard 13253R uses the following empirical fits to determine the fan static (*SE*) and fan motor efficiencies (*MEr*).

$$SE = 0.1881 * Ln(P_e + P_c + 50) - 0.4700$$
$$MEr = 0.060 * Ln[Q * (P_e + P_c + 50) / SE] + 0.123$$

Where Q is the measured air volume rate of standard air (m^3/s), P_e is the minimum external static pressure (Pa), and P_c is the internal static pressure drop of the indoor coil cabinet assembly measured during the cooling capacity test (Pa).

Any proposal to raise the minimum external static pressure requirements and possibly tweak the ISO approach for estimating fan heat/power adjustments will first have to be agreed upon by the U.S. delegates on the ISO working group. If the proposal is endorsed by the U.S. delegation, then the delegation must submit the proposed change for the consideration of the full working group. NIST, as a member of the U.S. delegation, has raised the issue for discussion among the U.S. delegation. At this point, the U.S. delegation does not have plans for recommending changes to ISO 13253 in this area.

4. Sections 3.2.3 and 3.6.3. Testing a Two-capacity Compressor System

ARI stated: ‘‘ARI agrees with DOE that the test procedure should be modified to accommodate more appropriate testing of multiple capacity heat pumps that are sized to meet the cooling load at fan speeds lower than the maximum. We have established a task group to investigate this issue, and will provide our recommendations to DOE as soon as they are available.’’ (ARI, No. 6 at 2).

The proposed test procedure covers two-capacity heat pumps that are designed to operate exclusively at low capacity in meeting the space cooling load while using both low and high capacities when space heating. SEER of the unit is evaluated in the same way as specified for a single-speed air conditioner. HSPF is evaluated using the same algorithm as specified for a ‘‘normal’’ two-capacity heat pump except that the building loads for the heating temperature bins are based on the heat pump's heating capacity when tested at *low capacity* and 47 °F outdoor dry bulb temperature. Previously, the building loads were tied to the heat pump's heating capacity at 47 °F and *high* compressor capacity. The change will drive the balance point of the heat pump down. The issue on this particular subject is whether the heat pump must have a lockout feature to prevent cooling at high capacity or is it sufficient that the rating is applicable so long as the heat pump is sized to operate at low capacity at design cooling conditions? The advantages of the lockout would be to (better) assure that high compressor capacity would not be used when cooling and a particular unit would only have one unique NAECA-required SEER and HSPF rating. Without the lockout feature, the unit would have two SEER and HSPF ratings. A lockout feature is required in accordance with today's proposed test procedure but DOE welcomes further discussion on this issue.

5. Section 3.3. Capacity Adjustments for Barometric Effects

ARI commented: ‘‘ARI is aware that barometric pressure can have an affect on test results. However, we believe DOE should allow ASHRAE to finish its analysis of this issue before making changes to the test procedure. The test procedure should be revised to reference ASHRAE Standard 37–1988, with the exception of the section that pertains to corrections made to capacities based on measured barometric pressures, since it is known this section contains an error and is being revised.’’ (ARI, No. 6 at 2).

Today's proposed test procedure reflects the recommendation made by ARI.

6. Sections 3.5.3 and 3.8.1. Cyclic Degradation Coefficients

In the existing DOE test procedure, the default values provided for cooling and heating cyclic degradation coefficients, C^c_D and C^h_D, are both 0.25. On the cooling side, the two optional tests are conducted on the majority of units because the experimentally-determined C^c_D is lower than 0.25. NIST, DOE and ARI members have discussed developing new defaults. The goal is to obtain more representative defaults resulting in less C_D testing while still crediting features that enhance cyclic performance. The manufacturers and ARI, as part of their certification program, have experimentally determined the C_D of many units. DOE believes the available data could be used to evaluate a set of new defaults that depend on the hardware components of the air conditioner or heat pump. The compiling of the data, however, is a formidable and thus far uncompleted task.

ARI originally commented: ''ARI endorses the concept of providing alternate degradation coefficients (C_D) for systems using specific components known to reduce the typical 0.25 default value. This could significantly reduce test burden by decreasing the need for the cumbersome cyclic test. ARI will continue to work with NIST on this effort, and provide whatever data our members authorize, to help determine appropriate alternatives.'' (ARI, No. 6 at 3). More recently, ARI members have reconsidered the merits of seeking new C_D defaults. A final decision from ARI is pending.

DOE encourages ARI to provide data and recommendations needed to begin the investigation into better C_D defaults. If better C_D defaults are identified, DOE will initiate steps to implement defaults that are lower than the existing values of 0.25. Such lower defaults could only positively impact the SEER and HSPF of a unit and so would not require adjustments to the existing NAECA energy conservation standards. Defaults that are higher than the existing 0.25 values, which could only negatively impact SEER and HSPF, would most likely become effective the same time as new NAECA energy conservation standards. We will not delay efforts to move today's proposed rulemaking into a final rulemaking by the pursuit of better C_D defaults. If we identify better defaults in sufficient time before the issue of the final rule, so we can obtain public comments on the proposed C_D values, then we will incorporate the better defaults into the final rulemaking. If better defaults are identified after the final rulemaking, DOE will initiate a new rulemaking process where changes and comments are limited to the issue of new C_D defaults.

E. Calculations of Seasonal Performance Descriptors

1. Sections 4.1.4 and 4.2.4. Variable-speed Bin Calculations

In the existing DOE air conditioner and heat pump test procedure, a quadratic fit is used to approximate the change in EER and COP as a function of the outdoor bin temperature. Prior to the 25 September 1997 DOE workshop, consideration had been given to using an alternative fit, a linear over linear rational function of the form $Y=(A_0+A_1 \cdot X)/(1+B_1 \cdot X)$. The rationale function was considered because it maintains a monotonic shape in all cases whereas a quadratic fit can have an inflection point between the points that it is fitting. For the purposes of interpolating the EER or COP of a variable-speed, all-electric heat pump or air conditioner, both fits are expected to give comparable results because the points being fitted have historically been close to linear. For the one variable-speed heat pump considered by NIST, for example, the two fits resulted in SEER and HSPF changes of 0.06% and 0.14%, respectively.

At the 25 September 1997 DOE workshop, the issue was discussed. Trane spoke against adopting the rational function on the basis that no practical problems had arisen with using the quadratic fit over the approximately 15 years that it has been used. (Trane, No. 2HH at 149–150).

Today's proposed test procedure maintains the use of the quadratic fit.

IV. Summary of Proposed Modifications to the DOE Air Conditioner and Heat Pump Test Procedure

In addition to the modifications cited in Section III, the proposed test procedure also incorporates the following changes.

A. Update and Add References for ASHRAE and ARI Standards

The existing test procedure references ASHRAE Standards 37–78 and 41.1 (no year) and ARI Standards 210–79, 240–77, and 320–76. The proposed revised version references ARI Standard 210/240–94 and ASHRAE Standards 23–93, 37–88, 41.1–86 (RA 91), 41.2–87 (RA 92), 41.6–94, 41.9–88, 51–99, and 116–95.

B. Air Volume Rates

ARI Standard 240–77 was previously referenced. Now, rather than referencing ARI Standard 210/240–94, we have added sections within this proposed test procedure. The main reason for no longer referencing ARI Standard 210/240 is that it does not cover variable-speed, constant CFM blowers and does not directly address two-capacity and variable-speed systems. It is preferable to have the overall issue of air volume rates covered in one place rather than in two. The main objective is to agree on air flow rate specifications. If ARI Standard 210/240 is revised to cover these systems, DOE may again reference the ARI Standard.

ASHRAE Standard 37–78 (or 37–88) is no longer referenced for the equation calculating the air volume rate of standard air. The factor $1+W_n$ is missing from the denominator of the equation given in Standard 37–88. This change has been adopted by the committee working to revise Standard 37.

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

C. Cyclic Testing

Industry practice and the method described in ASHRAE Standard 116 was adopted. Section 5.1 of the current Appendix M implies that the air volume rate is to be measured during cyclic tests. Standard test laboratory practice is to try to obtain the same velocity pressure or nozzle static pressure drop that was obtained during the comparable steady-state test. The air volume rate used in the cyclic test calculations is assumed to be the same air volume rate measured during the comparable steady-state test. This change is reflected in this proposed test procedure.

Concerning split-type non-ducted (ductless) systems, Section 4.1.1.5 of Appendix M states that ''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 3 minutes prior to compressor cut-on and for 3 minutes after compressor cutoff during the final OFF/ON interval. This proposed test procedure 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 3-minute fan-only period. Space cooling capacity is integrated from compressor ON to indoor fan OFF. As with the present test procedure, fan energy for the three minutes after compressor cutoff is added to the integrated cooling capacity.

The present 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. Much of this information is given in Appendix B of ARI Standard 210/240–94. Needed information is incorporated within the text of the proposed test procedure rather than making specific references to each pertinent section of Appendix B of the ARI Standard.

For dry coil tests, the proposed test procedure adopts ARI Standard 210/

240–94 Appendix B language with regard to the requirement that the drain pan be plugged and that the pan should be completely dry.

This notice of proposed rulemaking clarifies 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.

Existing Section 4.1.3.1, which reads ''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,'' has been removed from the proposed test procedure. This requirement is automatically met given the 0.5 °F test condition tolerance associated with each test.

For units having a variable-speed indoor fan, the manufacturer will have the option of conducting the cyclic tests with the indoor fan enabled or disabled, the latter being the default option if an attempt at testing with the fan enabled is unsuccessful. Specifically, if testing with the indoor fan operating and it automatically reverses, shuts down, or operates at an uncharacteristically high external static pressure, then a pull-thru method, where the fan is disabled, must be used. Although allowing the option of testing with the fan disabled is needed because of the potential fighting between the unit's fan and the exhaust fan of the air flow measuring apparatus, DOE seeks data from cyclic tests where the fan operates versus tests where the fan is disabled and the pull-thru method is used.

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. 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. Still, the proposed method of testing will benefit these units. DOE has only been able to obtain data from one unit where two different ramp profiles were compared to the results from imposing step responses in air flow. In one case C_D went from 0.05 (''truth'': ramp) to 0.02 (approximation: step change) while in the second case the values were 0.025 and 0.00. DOE seeks additional data showing the difference between the ramp and step responses during cyclic tests.

D. Fanless (Coil-only) Units

Section 4.1 of the existing Appendix M calls for corrections to capacity and power based on CFM. Section 4.2 of existing Appendix M calls for corrections to capacity and power based on SCFM. ITS uses SCFM in all cases. Thus, the proposed test procedure adopts the practice of only specifying the corrections in terms of SCFM.

The proposed test procedure also adopts the ARI Standard 210/240–94 Appendix B requirement that a specific enclosure be constructed (1 inch ductboard) when testing a coil only unit that does not employ an enclosure.

E. Frost Accumulation Test

The proposed test procedure adopts the ASHRAE Standard 116–95 and ARI 210/240–94 convention of specifying the outdoor wet bulb temperature (33 °F) in place of the presently specified dew point temperature (30 °F).

F. Test Tolerance Tables

The current Appendix M contains tables covering all tests except steady-state cooling mode tests, for which Table III in ASHRAE Standard 37–78 is referenced. Table III of ASHRAE Standard 37–78 has been added to the proposed test procedure since all the other tables are included in Appendix M.

The test tolerance tables have been improved. For example, although a test condition tolerance for external resistance to air flow is provided in the current test procedure, it is not applicable for ducted units. Such a test condition tolerance is, however, now applicable to non-ducted units. Also, a test condition tolerance has been added for electrical supply voltage (previously, only a test operating tolerance was specified). Because ASHRAE Standard 37–78 does not cover cooling mode dry coil tests, a test condition tolerance on the indoor inlet wet bulb temperature is not applicable. Test tolerances given on the outdoor outlet dry and wet bulb temperatures are now noted as only being applicable when the Outdoor Air Enthalpy Method is used to provide the secondary capacity measurement.

For the Frost Accumulation Test, the intervals considered to be heating versus defrosting have been modified slightly. Specifically, in the existing test procedure in Section 4.2.3.3, the first 5 minutes after a defrost termination was included in the defrost interval. In the proposed test procedure, the time interval has been increased to 10 minutes. 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. Given that these tolerances should be less stringent that those required of a steady-state test, the proposed test procedure adopts the values given in ASHRAE Standard 37: 1.5° F and 0.5° F.

G. Pretest Intervals

Statements given in the DOE proposed test procedure regarding operation prior to recording data have been modified. These changes are as follows.

1. Wet Coil Tests

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

Proposed: ''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

Existing: ''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).

Proposed: 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

Existing: '' * * * test 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).

Proposed: ''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

Existing: ''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).

Proposed: ''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

Existing: '' * * * and 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).

Proposed: Same as for the dry coil cooling mode cyclic test (see above).

6. Frost Accumulation Test

Existing: ''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).

Proposed: ''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

Existing: ''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).

Proposed: Same as for the Maximum and High Temperature Heating mode tests (see above) with the following additions. ''After satisfying the Section 3.7 requirements for the pretest interval, but before you begin collecting 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 existing test procedure 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 coverage of them is excluded from today's proposed test procedure. However, coverage was added to address units that lock out low capacity operation at low (heating) or high (cooling) outdoor temperatures. For this new case, a step was added which reverts to a single capacity calculation. The proposed test procedure uses the C_D determined based on cycling at low capacity (or the 0.25 default) in all cases. The Department welcomes comments on any control strategy used by two-capacity units that are not adequately covered in today's proposed test procedure.

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

The proposed test procedure requires two extra steady-state tests for the cooling mode (see Table 4) and two extra steady-state tests for the heating mode (see Table 10). An extra Frost Accumulation test is optional.

3. Specification of the Air Volume Rate for Tests at Low Capacity

In the existing test procedure, the air volume rate to be used when testing a two capacity system while operating at low capacity is not explicitly addressed. The proposed test procedure requires the use of the fan laws, as is now done for variable-speed systems, to determine the air volume rate when testing a unit having an indoor fan. For fanless units, the air volume rate used when conducting tests at low capacity (i.e., 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.

DOE believes that a lower limit is needed given the finite capabilities of the typical multi-speed furnace blower that is used in field installations. The 75 percent minimum is based on very limited data collected by NIST. The subject has been discussed by industry members at such forums as ASHRAE meetings but no formal consensus has yet been reached for the specified percentage. Data and comments are requested, especially with regard to the specified value of the lower limit.

I. Triple-split Systems

The DOE test procedure refers to ASHRAE Standard 37 on the issue of equipment installation and test set up procedures. ASHRAE Standard 37, in turn, states that you must use 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 under test within the indoor chamber. The present requirement is burdensome and DOE knows of no one who uses it when testing triple-splits. Furthermore, the heat loss from the indoor compressor section should be reflected in an adjusted output capacity and not by a raised entering air temperature. The amount of heat dissipated to the ambient by the indoor compressor section of such units is usually minimized as a result of the enclosure of the third section being insulated (mainly in an effort to reduce the operating noise). Based on limited information gained to date, the amount of heat lost from the indoor compressor section is on the order of 2 percent or less of the unit's space conditioning capacity.

The proposed test procedure instructs that triple-split systems are not to be tested using the calorimeter air-enthalpy method arrangement (see note in Section 2.6). At this juncture, no algorithm or method for assigning/determining the heat loss from the indoor compressor section is included. If triple-split systems become more popular and if information becomes available indicating the heat loss from the indoor compressor section exceeds 2 percent of the total, 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. Until a better alternative is identified, defrosts initiated during the ''preliminary'' test and the ''official'' test will be manually induced. The manufacturer will be required to provide information as to how long the unit would optimally frost before initiating a defrost. The manufacturer will have to provide information on how to induce a defrost cycle at the appropriate elapsed time. The controls of the unit, however, will still control the duration of the defrost cycle, once initiated.

K. Test Unit Installation

For the most part, equipment installation requirements will continue to be performed according to the manufacturer's installation instructions. However, the proposed test procedure adopts the lab and field practice of insulating the low pressure line(s) of a split system. Also, Section 2.2.5 restricts the use of special refrigerant charging criteria for lab testing.

L. Test Apparatus and Measurement/Sampling Frequency

1. Inlet Plenum for Blower Coils

In the current DOE test procedure, no inlet plenum is required when testing blower coil units. The proposed test procedure recommends that an inlet plenum be installed if space permits. (Lab ceiling height on vertical installation is a limitation.) The test procedure recommends using an inlet plenum that is constructed according to the design specified for fanless units. See Section 2.4.2.

2. Manifolded Static Pressure Taps

The triple-T configuration was found in 1976 to be the preferred method for manifolding static pressure taps (''The design of piezometer rings'' by K. A. Blake, Journal of Fluid Mechanics, Vol. 78, part 2, pp. 415–428). The triple-T configuration and the more widely used complete ring, four-to-one manifolding configuration are presently part of the draft revision of ASHRAE Standard 37. This revised test procedure recommends use of either of these two manifolding methods, which are shown in Figure 1. The broken ring, four-to-one manifolding configuration may be used but is not recommended.

3. Temperature Measurement Intervals

The proposed test procedure specifies that dry-bulb temperature measurements are to be measured at the intervals specified in ASHRAE Standard 41.1–86 (RA91). Wet bulb temperature, dew point temperature, or relative humidity are to be measured at the minimum sampling interval specified in Definition 1.14.

4. Temperature Measurement Accuracies

The proposed test procedure defers entirely to ASHRAE Standard 41.1–86 (RA 91) for accuracy and precision requirements.

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

The proposed test procedure adopts the ARI Standard 210/240–94 Appendix B requirements that a temperature spread of 1.5 °F or less be obtained, and that the outlet temperature grid be composed of a minimum of 9 sensors (while recommending 16). Also, the proposed test procedure recommends redundant sensors to determine the change in dry bulb temperature across the indoor coil.

6. Duct Loss Correction

The proposed test procedure adds a correction for the heat transfer between the test room and an outlet duct sandwiched between the coil and the outlet temperature grid. 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 test procedure explicitly permits alternatives to using wet bulb temperature sensors. To ease instrumentation selection, required instrument accuracies are provided for dew point hygrometers and relative humidity meters.

8. Voltmeter Accuracy

The required accuracy of voltage measurements has been changed from ±2% to ±1%.

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. Published literature [1–7] supports avoiding the use of induction type meters for measuring such non-sinusoidal power and instead recommends using a meter that is capable of sampling up to the 50th harmonic. This point is included in Section 2.8 of today's test procedure as a recommendation when testing a heat pump or air conditioner having a variable-speed compressor. (In terms of a meter sampling frequency, a 50th harmonic requirement corresponds to a minimum sampling frequency between 3 and 30 kHz, depending upon which technical recommendation you wish to cite.)

The majority of the technical references listed below report the performance of specific meters with specific waveforms, some of which should be representative of those found in presently-marketed residential-size air conditioners and heat pumps. In addition to induction watthour meters, a disconcerting result reported in the noted references is that the use of a non-induction meter that can measure up to the 50th harmonic does not insure an accurate measurement but only improves your chances.

References:

1. P.S. Filipski and R. Arseneau, ''Behavior of Wattmeters and Watthour Meters Under Distorted Waveform Conditions,'' IEEE tutorial course, Nonsinusoidal Situations: Effects on the Performance of Meters and Definitions of Power, IEEE, Piscataway, NJ, pp. 13–22, 1990.
2. A. Domijan, Jr., E. Embriz-Santander, A.J. Gilani, G. Lamer, C. Stiles, and C.W. Williams, Jr., ''Watthour Meter Accuracy Under Controlled Unbalanced Harmonic Voltage and Current Conditions,'' IEEE Transactions Power Delivery, Vol. II, No. 1, pp. 64–78, Jan. 1996.
3. A. Domijan, D. Czarkowski, A. Abu-aisheh, and E. Embriz-Santander, ''Measurements of Electrical Power Inputs to Variable Speed Motors and Their Solid State Power Converters—Phase II,'' ASHRAE Research Project 770, Final Report, November 30, 1995.
4. D. Czarkowski and A. Domijan, Jr., ''Performance of Electrical Power Meters and Analyzers in Adjustable-Speed Drive Applications,'' American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) Transactions 1997, Vol. 103, Part 1.
5. A.J. Baldwin, N.G. Planer, D.E. Nordell, N. Hohan, ''Evaluation of Electrical Interference to the Induction Watthour Meter,'' EPRI EL–2315, Research Project 1738, Final Report, April 1982.
6. Institute of Electrical and Electronics Engineers (IEEE), Inc., ''IEEE Standard 519–1992, IEEE Recommended Practices and Requirements for Harmonic Control in Electrical Power Systems,'' New York, New York.
7. A. Domijan, and E. Embriz-Santander, ''Measurements of Electrical Power Inputs to Variable Speed Motors and Their Solid State Power Converters,'' American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) Transactions 1993, Vol. 99, Part I, pp. 241–258.

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

In the existing test procedure, variable-speed systems that operate at higher speeds when heating than when cooling are covered. In today's proposed revision (as noted above in III.D.4) this allowance has been extrapolated to coverage of two-capacity heat pumps that only operate at low capacity during the cooling season while using both low and high capacities when heating. And, in taking a generic approach, today's test procedure covers any case where the heat pump uses different fan speeds or air volume rates for cooling versus when heating. (See Section 3.1.4.4.2)

N. Secondary Test Requirements.

When using the Outdoor Air Enthalpy test method, a preliminary test is conducted to compensate, if necessary, for any performance impact caused by the outdoor air-side test apparatus. In accordance with the existing test procedure, a preliminary test is conducted prior to all steady-state tests (i.e., those tests where a secondary measurement of capacity is required). In today's revision, relaxing this requirement is proposed. 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.

O. HSPF Calculations

The last paragraph of Sections 5.2.1 and 5.2.2 of the existing test procedure are not similarly placed in the proposed 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 issue of how many HSPF calculations are required has been, and will remain, an item that is covered elsewhere: In 10 CFR part 430, subpart B, § 430.23(m)(3)(ii). In the proposed test procedure, this section along with a short restatement of its contents are included 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, no reference is made to obtaining HSPF or operating cost via interpolation.

V. Procedural Requirements

A. Review Under the National Environmental Policy Act of 1969

In this notice, the Department proposes amendments to the test procedures for central air conditioners and heat pumps. We have reviewed the proposed rule under the National Environmental Policy Act of 1969 (NEPA), 42 U.S.C. 4321 et seq., the regulations of the Council on Environmental Quality, 40 CFR parts 1500–1508, DOE regulations for compliance with NEPA, 10 CFR part 1021, and the Secretarial Policy on the National Environmental Policy Act (June 1994). The Department has determined that this rulemaking is covered under the Categorical Exclusion found at paragraph A.6 of appendix A to subpart D, 10 CFR part 1021, which applies to rulemakings that are strictly procedural. This proposed rule is a procedural rulemaking and its implementation will not affect the quality or distribution of energy usage and therefore will not result in any environmental impacts. Accordingly, neither an environmental assessment nor an environmental impact statement is required.

B. Regulatory Review

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

C. Regulatory Flexibility Review

The proposed rule has been reviewed under the Regulatory Flexibility Act, (42 U.S.C. 601–612), which requires preparation of a regulatory flexibility analysis for any regulation that will have a significant economic impact on a substantial number of small businesses and other small entities. The proposed rule affects manufacturers of central air conditioners and heat pumps. The test procedures would not have a significant economic impact, but rather, would provide common testing methods. This revision of the test procedure will not require a significant investment for new testing equipment. DOE accordingly certifies that the proposed rule would not, if promulgated, have a significant economic impact on a substantial number of small entities and that preparation of a regulatory flexibility analysis is not required.

D. ''Takings'' Assessment Review

DOE has determined pursuant to Executive Order 12630 (52 FR 8859, March 18, 1988) that this proposed regulation, if adopted, would not result in any takings which might require compensation under the Fifth Amendment to the United States Constitution.

E. Federalism Review

Executive Order 13132 (64 FR 43255, August 10, 1999) requires agencies to develop an accountable process to ensure meaningful and timely input by State and local officials in the development of regulatory policies that have ''federalism implications.'' Policies that have federalism implications are defined in the Executive Order to include regulations that have ''substantial direct effects 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.'' 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 determined that it 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 the Executive Order.

F. Paperwork Reduction Act Review

This proposed rule contains no new collections of information under the Paperwork Reduction Act, 44 U.S.C. 3501 *et seq.*

G. Review Under Unfunded Mandates Reform Act of 1995

Section 202 of the Unfunded Mandates Reform Act of 1995 (''Unfunded Mandates Act'') requires that the Department prepare an impact statement before promulgating a rule that includes a Federal mandate that may result in expenditure by state, local, and tribal governments, in the aggregate, or by the private sector, of $100 million or more in any one year. The impact statement must include: (i) Identification of the Federal law under which the rule is promulgated; (ii) a qualitative and quantitative assessment of anticipated costs and benefits of the Federal mandate and an analysis of the extent to which such costs to state, local, and tribal governments may be paid with Federal financial assistance; (iii) if feasible, estimates of the future compliance costs and of any disproportionate budgetary effects the mandate has on particular regions, communities, non-Federal units of government, or sectors of the economy; (iv) if feasible, estimates of the effect on the national economy; and (v) a description of the Department's prior consultation with elected representatives of state, local, and tribal governments and a summary and evaluation of the comments and concerns presented.

The Department has determined that the action proposed today does not include a Federal mandate that may result in estimated costs of $100 million or more to state, local or to tribal governments in the aggregate or to the private sector. Therefore, the requirements of sections 203 and 204 of the Unfunded Mandates Act do not apply to this action.

H. Review Under Executive Order 12988, ''Civil Justice Reform''

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 executive 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. With regard to the review required by Section 3(a), section 3(b) of the Executive Order 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) provide 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 the Executive Order requires Executive agencies to review regulations in light of applicable standards 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 reviewed today's proposed rulemaking under the standards of section 3 of the Executive Order and determined that, to the extent permitted by law, it meets the requirements of those standards.

I. Review Under the Treasury and General Government Appropriations Act, 1999

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

J. Plain Language Review

The President's Memorandum on ''Plain Language in Government Writing,'' 63 FR 31885 (June 10, 1998) directs each federal agency to write all published rulemaking documents in plain language. The Memorandum includes general guidance on what constitutes ''plain language.'' Plain language requirements will vary from one document to another, depending on the intended audience, but all plain language documents should be logically organized and clearly written.

We have tried to make this proposed rule easy to understand. We are also requesting suggestions on how to improve its readability further.

VI. Public Comment Procedures

A. Written Comment Procedures

The Department invites interested persons to participate in the proposed rulemaking by submitting data, comments, or information with respect to the proposed issues set forth in today's proposed rule to Ms. Brenda Edwards-Jones, at the address indicated at the beginning of this notice. We will consider all submittals received by the date specified at the beginning of this notice in developing the final rule.

According to 10 CFR 1004.11, any person submitting information that he or she believes to be confidential and exempt by law from public disclosure should submit one complete copy of the document and ten (10) copies, if possible, from which the information believed to be confidential has been deleted. The Department of Energy will make its own determination with regard to the confidential status of the information and treat it according to its determination.

Factors of interest to the Department when evaluating requests to treat as confidential information that has been submitted include: (1) A description of the items; (2) an indication as to whether and why such items are customarily treated as confidential within the industry; (3) whether the information is generally known by or available from other sources; (4) whether the information has previously been made available to others without obligation concerning its confidentiality; (5) an explanation of the competitive injury to the submitting person which would result from public disclosure; (6) an indication as to when such information might lose its confidential character due to the passage of time; and (7) why disclosure of the information would be contrary to the public interest.

B. Issues for Public Comment

The Department of Energy is interested in receiving comments and data concerning these test procedures. Also, the Department welcomes comments on improvements or alternatives to these approaches. In particular, DOE is interested in gathering comments on the following:

1. Non-defrost (limited-range) heat pumps

Which of the three options described in Section III.A.3 should be invoked?

2. Testing units having a constant-air-volume-rate indoor fan

Are the proposed changes described in Section III.D.1 acceptable? In particular, does the proposed 8 percent tolerance on indoor air volume rate provide a fair balance between assuring repeatable results while not being too restrictive given the variation in blower motor performance?

3. Cyclic testing of units having a variable-speed indoor fan (that may or may not provide a constant air volume rate)

For units that ramp the indoor fan speed when cycling on and/or off, data are sought of the type referenced in the last paragraph of Section IV.C (i.e., data that quantifies the effect on C_D from using a ramped air volume rate versus forcing the air volume rate to have a step profile). Also, as described in the second-to-last paragraph of Section IV.C, data from cyclic tests conducted with the indoor fan enabled and disabled are sought.

4. Two-capacity heat pumps that are designed to meet the seasonal cooling load while operating at low capacity

As discussed in the last paragraph of Section III.D.4, should the heat pump be required to have controls that lock out high capacity operation when cooling?

5. Lower limit on the air volume rate used when testing a fanless, two-capacity unit at low compressor capacity

As discussed in Section IV.H.3, data and comments are requested regarding the assigned limit for the air volume rate when testing a fanless, two-capacity unit at low compressor capacity.

Related to this issue is whether the manufacturer should be required to supply, with the unit, the hardware needed to allow the use of two fan speeds on the furnace blower that the unit would be used with in the field. Conceivably, if such hardware was not provided, the test procedure could call for using the same air volume rate for all tests, regardless of whether the compressor is operating at high or low capacity. On the other extreme, do any manufacturers provide hardware that allows a multi-speed furnace blower to operate as a variable-speed blower? Or, are there safeguards that will result in all or the vast majority of fanless, two-capacity units to be applied with furnaces having variable-speed blowers? If so, then the lower limit noted in the previous paragraph may not be applicable.

6. Fan defaults for fanless (i.e., coil-only) two-capacity units

In the existing test procedure, the fan heat/power default that is applied when rating fanless units is 1250 Btu/h per 1000 SCFM (365 watts per 1000 SCFM). When testing two-capacity fanless units, this adjustment is applied when evaluating space conditioning capacities and electrical power usages for both high and low compressor capacity operation. Do blower curves for multi-speed indoor fans support the use of the same default for both low and high capacity?

7. Differentiation among two-capacity air conditioners and heat pumps

Is there a need to differentiate between two-capacity units that can transition between high and low compressor capacities on-the-fly versus units that must shut off the compressor for some finite time interval when transitioning? Both the existing test procedure and today's proposed revision do not offer a means for providing such differentiation. To begin to do so would require information on how EER and COP are affected as they change from the value associated with steady operation at one compressor capacity until steady operation is obtained at the other compressor capacity following the transition. DOE seeks comments and data that would help to determine whether the test procedure needs to account for low/high compressor transitioning performance.

8. Testing single-packaged units

Today's proposed test procedure includes new test requirements when testing certain types of single-packaged units. The proposed additions are summarized in III.C.1. As presently proposed, the changes are limited to cooling mode tests where all or part of the indoor section is located in the outdoor test room and to heating mode tests where all or part of the outdoor section is located in the indoor test room. Comments are sought on the general proposal and on whether the approaches should be invoked when testing all packaged units.

9. Multi-capacity units

Are there any multi-capacity units that operate at less than maximum speed or high capacity at the lowest outdoor temperatures (prior to cycling off the compressor, if applicable)? Possibly such a strategy is needed to insure component reliability. Such a contingency is not covered in the existing or proposed test procedure.

10. Cyclic degradation coefficients

Comments are sought on the proposed actions discussed above in Section III.D.6 for working towards new C_D defaults.

11. NAECA energy conservation standards

Changes introduced in today's proposed test procedure are not expected to cause a minimally-compliant unit to now become non-compliant. If a particular proposed change is found to negatively affect minimally compliant units, then DOE would like to know.

12. Small-duct, high-velocity systems

Comments are sought on the proposed actions discussed in Section III.A.2.

C. Public Workshop

1. Procedures for Submitting Requests to Speak

You will find the time and place of the public workshop listed at the beginning of this notice of proposed rulemaking. The Department invites any person who has an interest in today's notice of proposed rulemaking, or who is a representative of a group or class of persons that has an interest in these proposed issues, to make a request for an opportunity to make an oral presentation. If you would like to attend the public workshop, please notify Ms. Brenda Edwards-Jones at (202) 586–2945. You may hand deliver requests to speak to the address indicated at the beginning of this notice between the hours of 8:00 a.m. and 4:00 p.m., Monday through Friday, except Federal holidays, or send them by mail or e-mail to brenda.edwards-jones@ee.doe.gov.

The person making the request should state why he or she, either individually or as a representative of a group or class of persons, is an appropriate spokesperson, briefly describe the nature of the interest in the rulemaking, and provide a telephone number for contact.

The Department requests each person wishing to speak to submit an advance copy of his or her statement at least 10 days prior to the date of this workshop as indicated at the beginning of this notice. The Department, at its discretion, may permit any person wishing to speak who cannot meet this requirement to participate if that person has made alternative arrangements with the Office of Building Research and Standards in advance. The letter making a request to give an oral presentation must ask for such alternative arrangements.

2. Conduct of Workshop

The workshop (hearing) will be conducted in an informal, conference style. The Department may use a professional facilitator to facilitate discussion, and a court reporter will be present to record the transcript of the meeting. We will present summaries of comments received before the workshop, allow time for presentations by workshop participants, and encourage all interested parties to share their views on issues affecting this rulemaking. Following the workshop, we will provide an additional comment period, during which interested parties will have an opportunity to comment on the proceedings at the workshop, as well as on any aspect of the rulemaking proceeding.

The Department will arrange for a transcript of the workshop and will make the entire record of this rulemaking, including the transcript, available for inspection in the Department's Freedom of Information Reading Room. Any person may purchase a copy of the transcript from the transcribing reporter.

List of Subjects in 10 CFR Part 430

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

Issued in Washington, DC., on December 19, 2000.

Dan W. Reicher,

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 proposed to be 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. By adding paragraphs (b)(5)3. through (b)(5)10.;

b. by adding paragraph (b)(7).

The additions specified above read as follows:

§ 430.22 Reference Sources.

* * * * *

(b) * * *

(5) * * *

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

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

5. American Society of Heating, Refrigerating, and Air-Conditioning Engineers Standard 41.1–1986 (Reaffirmed 1991), ''Standard Method for Temperature Measurement.''

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

7. American Society of Heating, Refrigerating, and Air-Conditioning Engineers Standard 41.6–1994, ''Method for Measurement of Moist Air Properties.''

8. American Society of Heating, Refrigerating, and Air-Conditioning Engineers Standard 41.9–1988, ''A Standard Calorimeter Test Method for Flow Measurement of a Volatile Refrigerant.''

9. American Society of Heating, Refrigerating, and Air-Conditioning Engineers/Air Moving and Conditioning Association, Inc. Standard 51–1999, ''Laboratory Methods of Testing Fans for Rating.''

10. 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.''

* * * * *

(7) Air-Conditioning and Refrigeration Institute (ARI), 4301 North Fairfax Drive, Suite 425, Arlington, Virginia 22203, (703) 524–8800, ARI Standard 210/240–1994, ''Unitary Air-Conditioning and Air-Source Heat Pump Equipment.''

* * * * *

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

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

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.
3.6.3 Tests for a heat pump having a two-capacity compressor
3.6.4 Tests for a heat pump having a variable-speed compressor.
3.7 Test procedures for steady-state Maximum Temperature and High Temperature heating mode tests (the H0, $H0_1$, H1, $H1_2$, $H1_1$, and $H1_N$ Tests).
3.8 Test procedures for the optional cyclic heating mode tests (the $H0C_1$, H1C, and $H1C_1$ Tests).
3.8.1 Heating mode cyclic degradation coefficient calculation.
3.9 Test procedures for Frost Accumulation heating mode tests
3.9.1 Average space heating capacity and electrical power calculations
3.9.2 Demand defrost credit
3.10 Test procedures for steady-state Low Temperature heating mode tests
3.11 Additional requirements for the secondary test methods
3.11.1 If using the Outdoor Air Enthalpy Method as the secondary test method
3.11.1.1 If a preliminary test precedes the official test
3.11.1.2 If a preliminary test does not precede the official test
3.11.1.3 Official test
3.11.2 If using the Compressor Calibration Method as the secondary test method
3.11.3 If using the Refrigerant Enthalpy Method as the secondary test method
3.12 Rounding of space conditioning capacities for reporting purposes.

4. Calculations of Seasonal Performance Descriptors

4.1 Seasonal Energy Efficiency Ratio (SEER) Calculations
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
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 2.1.2.2.1 where indoor fan capacity modulation correlates with the outdoor dry bulb temperature
4.1.2.2 Units covered by Section 2.1.2.2.2 where indoor fan capacity modulation is used to adjust the sensible to total cooling capacity ratio
4.1.3 SEER calculations for an air conditioner or heat pump having a two-capacity compressor.
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
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
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
4.1.3.4 Unit must operate continuously at high (k=2) compressor capacity at temperature T_j
4.1.4 SEER calculations for an air conditioner or heat pump having a variable-speed compressor
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
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
4.1.4.3 Unit must operate continuously at maximum (k=2) compressor speed at temperature 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
4.2.1.1 Space heating capacity and the electrical power consumption calculations for a non-defrost heat pump
4.2.1.2 Space heating capacity and the electrical power consumption calculations for a heat pump having a heat comfort controller
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
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
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
4.2.3.4 Heat pump must operate continuously at high (k=2) compressor capacity at temperature 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
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
4.2.4.3 Heat pump must operate continuously at maximum (k=2) compressor speed at temperature T_j
4.3 Calculations
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. Section 430.23(m)(3)(iii) 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 210/240–94* means the test standard ‘‘Unitary Air-Conditioning and Air-Source Heat Pump Equipment’’ published in 1994 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 91)* means the test standard ‘‘Standard Method for Temperature Measurement’’ published in 1986 and reaffirmed in 1991 by ASHRAE.

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

1.9 *ASHRAE Standard 41.9–88* means the test standard ‘‘A Standard Calorimeter Test Method for Flow Measurement of a Volatile Refrigerant’’ published in 1988 by ASHRAE.

1.10 *ASHRAE Standard 51–99* means the test standard ‘‘Laboratory Methods of Testing Fans for Rating’’ published in 1999 by ASHRAE and the Air Movement and Control Association, Inc.

1.11 *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.

1.12 *CFR* means Code of Federal Regulations.

1.13 *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.

1.14 *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 91). If such dry bulb temperatures are used only for test room control, sample at regular intervals that are 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.

1.15 *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.

1.16 *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.

1.17 *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.

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

1.19 *Degradation coefficient (C_D)* means a parameter used in calculating the part load factor. The degradation coefficient for cooling is denoted by $C^c{}_D$. The degradation coefficient for heating is denoted by $C^h{}_D$.

1.20 *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, 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.

1.21 *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.

1.22 *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.

1.23 *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.

1.24 *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.

1.25 *Heating load factor (HLF)* means the ratio having as it 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 + OFF) interval.

1.26 *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. A method for testing and rating heat pumps having a heat comfort controller is presently limited to heat pumps that meet the equipment criteria of Section 3.6.1.

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. 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. Unless an approved alternative rating method is used, as set forth in 10 CFR part 430, subpart B, § 430.24(m), HSPF must be calculated according to this appendix. Repeat the calculations for each of the six generalized U.S. climatic regions listed in this appendix. For each region, evaluate an HSPF for each standardized design heating requirement that applies. (See 10 CFR part 430 subpart B, § 430.23(m)(3)(ii).) The HSPF used to evaluate compliance with the Energy Conservation Standards (see 10 CFR part 430, subpart C, § 430.32(c)) is based on Region IV, the minimum standardized design heating requirement, and the sampling plan stated in 10 CFR part 430, subpart B, § 430.24(m).

1.28 *Mini-split air conditioners and heat pumps* means non-ducted 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.

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

1.30 *Non-defrost heat pumps* means equipment that is incapable of defrosting the outdoor coil. The equipment ceases to operate the refrigeration system at outdoor temperatures that are conducive to frost accumulation. A method for testing and rating non-defrost heat pumps is presently limited to heat pumps that meet the equipment criteria of Section 3.6.1.

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 space 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. Unless using an approved alternative rating method, as set forth in 10 CFR part 430, subpart B, § 430.24(m), SEER must be calculated according to Section 4.1 of this appendix. [See 10 CFR part 430, subpart B, 430.23(m)(3)(i).] This Section 4.1 SEER 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 part 430, subpart C, § 430.32(c).)

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

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

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

1.36 *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

1.37 *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.

1.38 *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.

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

1.40 *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.

1.41 *Time adaptive defrost control system* is a demand-defrost control system (see Definition 1.20) that measures the length of the prior defrost period(s) and uses that information to automatically determine when to initiate the next defrost cycle.

1.42 *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 1.20).

1.43 *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.

1.44 *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, and

(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, and

(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.).

High capacity means:

(1) Operating at high compressor speed,

(2) Operating the higher capacity compressor,

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

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

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

1.46 *Small-duct system* means equipment that contains a blower and indoor coil combination that produces at least 1.5 inches of external static across the indoor unit when operated at the certified air volume rate. When applied in the field, small-duct systems use branch ducts having less than 6.0 square inches of free area.

1.47 *ASHRAE Standard 41.6–94* means the test standard ''Method for Measurement of Moist Air Properties'' published in 1994 by ASHRAE.

2. Testing Conditions

a. 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.

b. 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 your equipment.

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 four (special) equipment features described by the Table footnote 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 you know the secondary method for determining the unit's cooling and/or heating capacity, 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.

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

d. 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 from 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 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;
N = non-defrost heat pump

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	3.1.41.2	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	3.6.1	3.6.1.1
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	HP HH	
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																						N
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;
N = non-defrost heat pump

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	3.6.1.2	3.6.2	3.6.3	3.6.4	3.7 to 3.8.1	3.9 to 3.10	3.11	3.11.1 to 3.11.3	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†	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;
N = non-defrost heat pump

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

† Does not apply to non-defrost heat pumps

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	4.2.1.1	4.2.1.2	4.2.2	4.2.3 to 4.2.3.4	4.2.4 to 4.2.4.3	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								N	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;
- N = non-defrost heat pump

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, DOE recommends using 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, DOE recommends using a heater(s) having a capacity that is close to the sensible cooling capacity of the test unit's evaporator. 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. With respect to interconnecting tubing used when testing split systems, however, follow the requirements given in Section 5.1.3.5 of ARI Standard 210/240–94. When testing triple-split systems (see Definition 1.43), use the tubing length specified in Section 5.1.3.5 of ARI Standard 210/240–94 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. 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 1.26).

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 Table 2, note 3 states (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·ft^2·°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 1.41), 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 as 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 operating simultaneously. To ease the testing burden of cyclic tests, consider creating a single control circuit that allows simultaneous cycling of all compressor systems. In this 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, you may need 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 you use the Outdoor Air Enthalpy test method 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, means the manufacturer's installation instructions that come packaged with the unit. For third party testing, for example, do not consult the manufacturer about how to charge 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.

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 paragraph b. of this section.

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 that are specified in paragraph b. of this section.

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 resistance (R-value) of at least 19 hr·ft^2·°F/Btu.

2.4.1 *Outlet plenum for the indoor coil.* 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 provides recommended options for the manifold configuration. See Figures 7 and 8 of ASHRAE Standard 37–88 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.

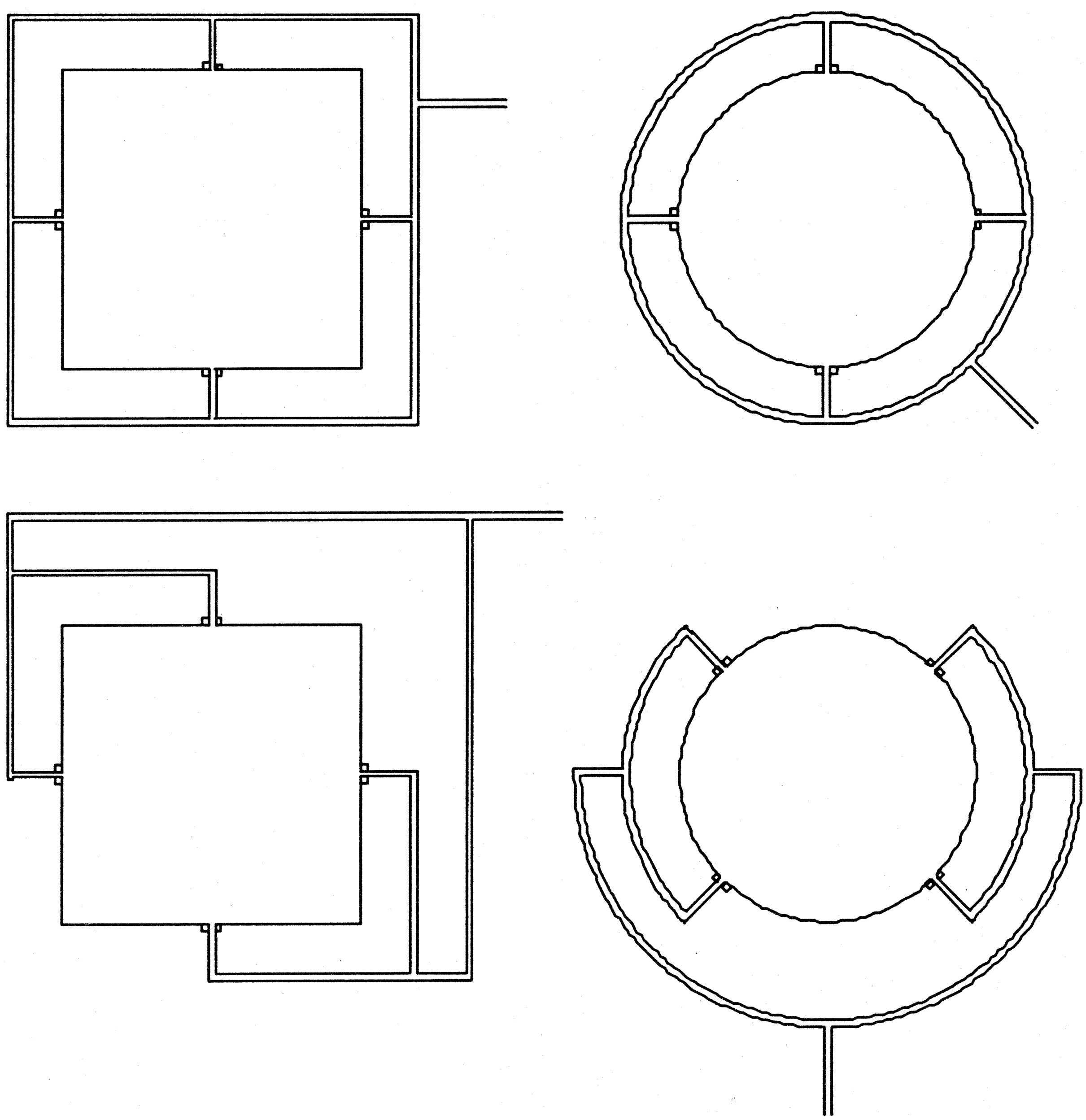

Figure 1. Recommended configuration for manifolding static pressure taps.

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. See Figure 8 of ASHRAE Standard 37–88 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), DOE recommends installing an inlet plenum if sufficient space exists within the test room. If using an inlet plenum, add four static pressure taps and a manifold that connects them together. DOE recommends constructing the inlet plenum and locating the static pressure taps as shown in Figure 8 of ASHRAE Standard 37–88. 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 91) 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. You may use the air sampling device and the remotely located temperature sensor(s) to determine the entering air dry bulb temperature during any test. You may use the air sampling device and the remotely located leaving air dry bulb temperature sensor(s) 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, DOE recommends installing an outlet air damper box when testing heat pumps, both ducted and non-ducted, that cycle off the indoor fan during defrost cycles. 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 the following:

(1) The inlet plenum (Section 2.5.1.1 units); or

(2) To 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 you do not use the Section 2.4.2 inlet plenum, but you are using a grid of dry bulb temperature sensors, locate the grid approximately 6 inches 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 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 you are not using a grid of sensors, 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 describes the recommended method for fabricating static pressure taps. Also refer to Figure 2A of ASHRAE Standard 51–99. 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 you are not using an inlet plenum or inlet damper box, 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. DOE recommends taping 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 passed 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 *Additional recommendations.* DOE recommends installing a mixing device(s) upstream of the outlet air, dry-bulb temperature grid (but downstream of the outlet plenum static pressure taps). Also, consider using a perforated screen located between the mixing device and the dry-bulb temperature grid. DOE recommends using a screen having 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 91) and Section 5.2.2 of ASHRAE Standard 41.2–87 (RA 92).

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 91). The transient testing requirements cited in Section 4.3 of ASHRAE Standard 41.1–86 (RA 91) apply if conducting a cyclic or Frost Accumulation test.

b. Distribute the sensors of a dry-bulb temperature grid over the entire flow area. DOE recommends using 16 temperature sensors within each temperature grid. The required minimum is 9 sensors per 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.

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 91). As specified in ASHRAE Standard 41.1, 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. 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 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 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. Refer to Figure 12 of ASHRAE Standard 51–99 or Figure 14 of ASHRAE Standard 41.2–87 (RA 92) 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, and Figures B1, B2, and B4 of ARI Standard 210/240–94 for illustrative examples of how the test apparatus may be applied within a complete laboratory set-up. Instead of following one of these examples, you may use an alternative set-up 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 when testing triple-split units.)

2.7 *Electrical voltage supply.* Perform all tests at the voltage specified in Section 5.1.3.2 of ARI Standard 210/240–94 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. Instead, consider using a watt-hour measuring system that is capable of measuring up to the 50th harmonic.

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. For all steady-state tests, in addition, conduct a second, independent measurement of capacity. For split systems, use one of the following secondary measurement methods: Outdoor Air Enthalpy Method, Compressor Calibration Method, or Refrigerant Enthalpy Method. Use either the Outdoor Air Enthalpy Method or the Compressor Calibration Method as the secondary measurement when testing a single packaged unit.

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 that 5 °F, total capacity from separate calibration tests conducted under identical operating conditions. Install instrumentation; measure refrigerant properties; adjust the refrigerant charge according to Section 7.4.2 of ASHRAE Standard 37–88. 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.

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. Refer to Section 7.6.2 of ASHRAE Standard 37–88 for the requirements for using the method, 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.

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), add instrumentation to permit measurement of the indoor test room dry-bulb temperature.

b. If you are not using the Outdoor Air Enthalpy Method, add instrumentation to measure the dry-bulb temperature and the water vapor content of the air entering the outdoor coil. DOE recommends measuring water vapor content by using an air sampling device to divert air to a remotely located sensor(s). If used, construct and apply the air sampling device as per Section 6 of ASHRAE Standard 41.1–86 (RA 91). You may use the air sampling device to also divert air to a sensor that measures outdoor-side entering dry bulb temperature. However, DOE recommends positioning dry bulb temperature sensors around the exterior of the entire outdoor coil and using them to determine an average entering dry bulb temperature. In such cases, use individually monitored sensors to identify any significant temperature distribution. Take steps (e.g., add or re-position a lab circulating fan), as needed, to minimize the magnitude of the temperature distribution. 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 91). 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.

3. Testing Procedures

3.1 *General Requirements.* If during the testing process you make an equipment set-up adjustment 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, maintain the airflow nozzle(s)' 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 test 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, E_V, F_1, G_1, H0, $H0_1$, H1, $H1_2$, $H1_1$, HI_N, H3, $H3_2$, 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 (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 5.1.3.4 of ARI Standard 210/240–94 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. For the A or A_2 Test (exclusively), the average air volume rate from the 30-minute data collection interval (see Section 3.3) and the manufacturer-provided Cooling Certified Air Volume Rate must differ by 8 percent or less.

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 a specified maximum. The maximum value is 0.30 inches of water for all units except small-duct, high-velocity systems (see 1.46) for which the limit is 0.50 inches of water. If the maximum value 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,

$$\text{Cooling Minimum Air Vol. Rate} = \text{Cooling Certified Air Vol. Rate} \cdot \frac{\text{Cooling Minimum Fan Speed}}{A_2 \text{ Test Fan Speed}},$$

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—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,

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

where $\Delta P_{st, A2}$ is the applicable Table 2 minimum external static pressure that was targeted during the A_2 (and B_2) Test. Only for the first test, the average measured air volume rate and the manufacturer-specified Cooling Minimum Air Volume Rate must differ by 8 percent or less.

c. For ducted two-capacity units that are tested without an indoor fan installed, the Cooling Minimum Air Volume Rate is the higher of the rate specified by the manufacturer or 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,

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

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

c. 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,

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

where $\Delta P_{st, A2}$ is the applicable Table 2 minimum external static pressure that was targeted during the A_2 (and B_2) Test.

d. 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 heat pumps that lock out high 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,

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

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

c. 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,

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

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. For the first test that uses the Heating Certified Air Volume Rate, the average measured air volume rate and the manufacturer-specified Heating Certified Air Volume Rate, both expressed in SCFM, must differ by 8 percent or less.

d. When testing ducted, two-capacity heat pumps that lock out high capacity operation when cooling, use the appropriate approach of the above two cases for units that are tested with an indoor fan installed. For coil-only (fanless) heat pumps that lock out high capacity cooling, 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.* This section applies when testing ducted two-capacity heat pumps that lock out high capacity operation when cooling. 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, 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. For the H1 or $H1_2$ Test (exclusively), the average air volume rate from the 30-minute data collection interval (see Section 3.7) and the manufacturer-provided Heating Certified Air Volume Rate must differ by 8 percent or less.

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 a specified maximum. The maximum value is 0.30 inches of water for all units except small-duct, high-velocity systems (see 1.46) for which the limit is 0.50 inches of water. If the maximum value 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,

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

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—the $H0_1$, $H1_1$, $H2_1$, and $H3_1$, 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,

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

where is $\Delta P_{st,H1_2}$ is the minimum external static pressure that was targeted during the $H1_2$ Test. Only for the first test, the average measured air volume rate and the manufacturer-specified Heating Minimum Air Volume Rate must differ by 8 percent or less.

c. When testing ducted, two-capacity heat pumps that lock out high capacity operation when cooling, use the appropriate approach of the above two cases for units that are tested with a indoor fan installed.

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 heat pumps that are tested without an indoor fan installed, and that lock out high capacity operation when cooling, 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,

$$\text{Heating Intermediate Air Volume Rate} = \text{Heating Certified Air Volume Rate} \cdot \frac{\text{H2}_V \text{ Test Fan Speed}}{\text{H1}_2 \text{ Test Fan Speed}}.$$

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

c. 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,

$$\text{H2}_V \text{ Test } \Delta P_{st} = \Delta P_{st,H1_2} \cdot \left[\frac{\text{Heating Intermediate Air Volume Rate}}{\text{Heating Certified Air Volume Rate}}\right]^2,$$

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

d. 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.’’

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), 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. 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:

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

where,

$\bar{\dot{V}}_s =$

air volume rate of standard (dry) air, $(\text{ft}^3/\text{min})_{da}$

$\bar{\dot{V}}_{mx} =$

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

$v'_n =$

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

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 or verify 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 or verify 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 EV Test if you expect 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 be 1.5 °F or less. 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 that follows the H1 or, if conducted, the H1C 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^c_D. If the two optional tests are not conducted, assign C^c_D 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. DOE recommends using an indoor air wet-bulb temperature of 57 °F or less.
[4] Maintain the airflow nozzle(s) static pressure difference or velocity pressure during the ON period at the same pressure difference or velocity pressure as measured during the 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^c_D. If the two optional tests are not conducted, assign C^c_D 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 IS CONTROLLED AS SPECIFIED IN 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. DOE recommends using an indoor air wet-bulb temperature of 57 °F or less.
[5] Maintain the airflow nozzle(s) static pressure difference or velocity pressure during the ON period at the same pressure difference or velocity pressure as measured during the C_1 Test.

3.2.3 *Tests for a unit having a two-capacity compressor.* a. (See Definition 1.44.) 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^c_D. If the two optional tests are not conducted, assign C^c_D 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. Two-capacity units that operate exclusively, via a lockout feature, at low compressor capacity when space cooling must be tested as a single speed system (see Section 3.2.1 and Table 3). If a two-capacity unit 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)	[4] 80	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. DOE recommends using an indoor air wet-bulb temperature of 57 °F or less.
[5] Maintain the airflow nozzle(s) static pressure difference or velocity pressure during the ON period at the same pressure 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_I Test—required (steady, wet coil)	80	67	82	[1]65	Minimum	Cooling Minimum [4]
F_I Test—required (steady, wet coil)	80	67	67	[1]53.5	Minimum	Cooling Minimum [4]
G_I Test[5]—optional (steady, dry coil)	80	(5)	67	....	Minimum	Cooling Minimum [4]
I_I 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. DOE recommends using an indoor air wet bulb temperature of 57 °F or less.
[6] Maintain the airflow nozzle(s) static pressure difference or velocity pressure during the ON period at the same pressure difference or velocity pressure as measured during the G_I Test.

3.3 *Test procedures for steady-state wet coil cooling mode tests* (the *A*, A_2, A_I, *B*, B_2, B_I, E_V, and F_I 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 1.14):

(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.

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

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

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

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

Where

$$\bar{\dot{V}}_s$$

is the average measured indoor air volume rate expressed in units of cubic feet per minute of standard air (SCFM).

TABLE 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 1.40.
[2] See Definition 1.39.
[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 Section 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:

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

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

3.4 Test procedures for the optional steady-state dry coil cooling mode tests (the C, C_I, 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 $\dot{Q}^{k}_{ss,dry}$(T) and $\dot{E}_{ss,dry}$(T). In preparing for the Section 3.5 cyclic test, record the average indoor-side air volume rate, $\dot{V}$, specific heat $C_{p,a}$ of the air, (expressed on dry air basis), specific volume of the air at the nozzle(s), v_n , humidity ratio at the nozzle(s),W_n , and either pressure difference or velocity pressure for the flow nozzle(s). 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_I, and I_I 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 switchover 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 you complete the test. 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(s) 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 a pull-thru approach 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 a external static pressure that is 0.1 inches of water or more higher than the value measured during the prior steady-state test.

e. For the pull-thru approach, disable the indoor fan and use the exhaust fan of the airflow measuring apparatus to generate the specified flow nozzle(s) 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. 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. DOE recommends obtaining repeatable results for two or more data collection intervals before terminating the test. If available, use electric resistance heaters (see Section 2.1) to minimize the variation in the inlet air temperature. 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.

f. 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.

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, you will terminate data collection used to determine the electrical energy before you terminate 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 1.40.
[2] See Definition 1.39.
[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.

g. 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,

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

where $\dot{V}$, $C_{p,a}$, 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,

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

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^c_D. If the two optional tests are not conducted, assign C^c_D the default value of 0.25. Evaluate C^c_D using the above results and those from the Section 3.4 dry coil steady-state test.0

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

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

$$CLF = \frac{q_{cyc,\,dry}}{\dot{Q}_{ss,\,cyc} \cdot \Delta\tau_{cyc,\,dry}},$$

the cooling load factor, dimensionless.

Round the calculated value for C^c_D to the nearest 0.01. If C^c_D 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 nozzle(s) static pressure difference or velocity pressure during the ON period at the same pressure difference or velocity pressure as measured during the *H1* Test.

3.6.1.1 Non-defrost heat pump. For non-defrost heat pumps (see Definition 1.30) that cease compressor operation at outdoor dry-bulb temperatures less than 37 °F, do not conduct the *H2* and *H3* Tests. Instead, conduct a Maximum Temperature (*H0*) Test using the Table 9 Heating Certified Air Volume Rate and the indoor and outdoor coil air inlet conditions specified for the $H0_1$ Test in Table 11.

3.6.1.2 Heat pump having a heat comfort controller. Test any heat pump that has a heat comfort controller (see Definition 1.26) according to Section 3.6.1 and Table 9 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.

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 one 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 you do not conduct the optional $H2_1$ Test, 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,

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

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

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 nozzle(s) 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 1.44).

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. You need to know the heat pump's capacity and electrical power at low compressor capacity for outdoor temperatures of 37 °F and less 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]
$H1_1$ Test (required, steady)	70	60(max)	47	43	Low	Heating minimum [2]
$H2_2$ Test (required)	70	60(max)	35	33	High	Heating certified [3]
$H2_1$ Test [4] (required)	70	60(max)	35	33	High	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 nozzle(s) static pressure difference or velocity pressure during the ON period at the same pressure difference or velocity pressure as measured during the *H01* 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. 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 you do not conduct the $H2_2$ Test, use the following equations to approximate the capacity and electrical power at the $H2_2$ test conditions:

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

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

Determine the quantities $\dot{Q}_h^{k=2}(47)$ and $\dot{E}_h^{k=2}(47)$ from the $H1_2$ Test and evaluate them according to Section 3.7. Determine the quantities $\dot{Q}_h^{k=2}(17)$ 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 you conduct the $H1_N$ Test, operate the heat pump's compressor at the same speed as 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 [2]
$H1_N$ 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 nozzle(s) static pressure difference or velocity pressure during the ON period at the same pressure difference or velocity pressure as measured during the H01 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.

3.7 *Test procedures for steady-state Maximum Temperature and High Temperature heating mode tests* (the *H0*, $H0_1$, *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 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 you obtain a 30-minute period (e.g., four consecutive 10-minute samples) 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	(3) 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 1.40.
[2] See Definition 1.39.
[3] Only applies when the Outdoor Air Enthalpy Method is used.
[4] Only applies when testing non-ducted units.

Calculate indoor-side total heating capacity as specified in Section 7.3.4.1 of ASHRAE Standard 37–88. 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 and $\dot{Q}_h{}^k(T)$ 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 *k*=N to denote results from the optional $H1_N$ Test, if conducted.

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

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

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

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

where

$$\bar{\dot{V}}_s$$

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

If conducting the optional cyclic heating mode test, which is described in Section 3.8, record the average indoor-side air volume rate,

$$\bar{\dot{V}},$$

specific heat of the air $C_{p,a}$ (expressed on dry air basis), specific volume of the air at the nozzle(s), v_n (or v_n), humidity ratio at the nozzle(s), W_n, and either pressure difference or velocity pressure for the flow nozzle(s). If either or both of the below criteria apply, determine the average, steady-state, electrical power consumption of the indoor fan motor ($\dot{E}_{fan,1}$).

a. 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

b. 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 ($\Delta P_{\min}$) 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 ‘‘b’’ criteria applies, conduct the following four steps after determining $\dot{E}_{fan,1}$ (which corresponds to ΔP_1).

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

2. 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.

3. Approximate the average power consumption of the indoor fan motor if the 30-minute test had been conducted at $\Delta P_{\min}$ using linear extrapolation:

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

4. 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,

$$\bar{\dot{V}},$$

$C_{p,a}v_b$ (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,

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

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

$$(\bar{\dot{V}}_s)$$

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. For single-speed heat pumps that defrosted before completing the Section 3.7 *H1* (or $H1_1$) steady-state test, DOE recommends initiating a defrost cycle before cycling the heat pump OFF and ON according to Section 3.5. Do not restrict air movement through the indoor coil if the heat pump cycles off its indoor fan during the defrost cycle. If conducting a defrost cycle, operate the single-speed heat pump for at least 10 minutes after defrost termination. After that, begin cycling the heat pump immediately or delay until you have re-established the specified test conditions. Pay attention to preventing defrosts after beginning the cycling process. However, if a defrost is automatically or manually initiated once the OFF/ON cycling begins, switch to operating the heat pump continuously until 10 minutes after defrost termination. After that, 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 was 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.

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

where,

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

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

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

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

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

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_h^D to the nearest 0.01. If C^h_D 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 [3]	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	2.0 [4]
Electrical voltage [5], % of rdg	2.0	1.5

[1] See Definition 1.40.
[2] See Definition 1.39.
[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 1.41), 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. You should use the outlet damper box described in Section 2.5.4.1, if installed, 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, you must 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: When heating, except for the first 10 minutes after the termination of a defrost cycle (Sub-interval H) and when defrosting, plus these same first 10 minutes after defrost termination (Sub-interval D). 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 (act to) 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) at equal intervals that span 10 minutes or less. Record the electrical energy consumed, expressed in watt-hours, from defrost termination to defrost termination, $e^k_{DEF}(35)$, 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 H [3]	Sub-interval D [4]	
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	....	[6] 0.02
Electrical voltage, % of rdg	2.0	....	1.5

[1] See Definition 1.40.
[2] See Definition 1.39.
[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. Evaluate average space heating capacity, $\dot{Q}_h^k(35)$, when expressed in units of Btu per hour, using:

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

where,

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

$C_{p,a} = 0.24 + 0.444 \cdot W_n$, 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 / $lbm_{da} \cdot$ °F.

V_n = specific volume of the air-water vapor mixture at the nozzle, ft^3 / 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.

Evaluate average electrical power, $\dot{E}^k_h(35)$, 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 $\dot{Q}^k_h(35)$ by,

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

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

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

where $\dot{V}_s$ 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). For 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 Section 3.1.4.4 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 30-minute interval used for determining capacity.

2. After the 30-minute interval 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 30-minute tests been conducted at ΔP_{min} using linear extrapolation:

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

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

3.9.2 *Demand defrost credit.* 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 1.20). 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.

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 you begin collecting 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, you should 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, you should 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.* 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. 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 you conduct a preliminary test, 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. Calculate heating capacity based on outdoor air enthalpy measurements as specified in Section 7.3.4.2 of the same ASHRAE Standard. You may adjust outdoor side capacities according to Section 7.3.3.3 of ASHRAE Standard 37–88 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.

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, ASHRAE Standard 41.9–88, and Section 7.5 of ASHRAE Standard 37–88.

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.

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. 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.* 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.

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.* For equipment covered under Sections 4.1.2, 4.1.3, and 4.1.4, evaluate the seasonal energy efficiency ratio,

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

where,

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

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

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

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

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

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

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

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

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:

$$SEER = PLF(0.5) \cdot EER_B$$

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.

$PLF(0.5) = 1 - 0.5 \cdot C^c_D$, 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, $\dot{Q}_D^c$, to the default value specified in Section 3.5.3. If these optional tests are conducted, set $\dot{Q}^c_d$ 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 5.4-1. Evaluate the quantity

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

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

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

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.

$$\frac{n_j}{N} =$$

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.

a. For the space cooling season, assign

$$\frac{n_j}{N}$$

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,

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

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

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

in Equation 4.1 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.

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

a. The value calculated as per Section 3.5.3 or

b. The Section 3.5.3 default value of 0.25.

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

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

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.

The parameters $FP_c^{k=1}$, $FP_c^{k=2}$, 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 $\dot{E}_c^{k=2}(95)$.

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 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

4.1.2.2 *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

$$\frac{q_c(T_j)}{N} \text{ and } \frac{e_c(T_j)}{N}$$

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,

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

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

$$\frac{n_j}{N},$$

Obtain the fractional bin hours for the cooling season,

$$\frac{n_j}{N},$$

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.

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,

$$\frac{n_j}{N},$$

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,

$$\frac{n_j}{N},$$

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,

$$\frac{n_j}{N},$$

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 F_1 Test, and all four quantities are calculated as specified in Section 3.3. Evaluate the space cooling capacity, $\dot{Q}_c^{k=2}(T_j)$, and electrical power consumption, $\dot{E}_c^{k=2}(T_j)$, of the test unit when operating at 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, $\dot{Q}_c^{k=1}(T_j) \geq BL(T_j)$.

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

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

where,

$X^{k=1}(T_j) = BL(T_j) / \dot{Q}_c^{k=1}(T_j)$, the cooling mode 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.

$$\frac{n_j}{N} =$$

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,

$$\frac{n_j}{N},$$

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, $\dot{Q}_c^{k=1}(T_j) < BL(T_j) < \dot{Q}_c^{k=1}(T_j)$.

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

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

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,

$$\frac{n_j}{N},$$

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:

$$D = \frac{T_2^2 - T_1^2}{T_v^2 - T_1^2}$$

$$C = \frac{EER^{k=1}(T_1) - EER^{k=2}(T_2) - B \cdot (T_1 - T_2)}{A = EER^{k=2}(T_2) - T_1^2 - T_2^2 - C \cdot T_2^2}$$

$$B = \frac{EER^{k=1}(T_1) - EER^{k=2}(T_2) - D \cdot \left[EER^{k=1}(T_1) - EER^{k=v}(T_v)\right]}{T_1 - T_2 - D \cdot (T_1 - T_v)}$$

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_l 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 [$\dot{Q}_c^{k=2}(T_2) = BL(T_v)$], °F. Determine T_2 by equating Equations 4.1.3–3 and 4.1–2 and solving for outdoor temperature.

$$EER^{k=1}(T_1) = \frac{\dot{Q}_c^{k=1}(T_1) \quad [\text{Eqn. 4.1.4-1, substituting } T_1 \text{ for } T_j]}{\dot{E}_c^{k=1}(T_1) \quad [\text{Eqn. 4.1.4-2, substituting } T_1 \text{ for } T_j]}, \text{ Btu/h per W.}$$

$$EER^{k=v}(T_v) = \frac{\dot{Q}_c^{k=v}(T_v) \quad [\text{Eqn. 4.1.4-3, substituting } T_v \text{ for } T_j]}{\dot{E}_c^{k=v}(T_v) \quad [\text{Eqn. 4.1.4-4, substituting } T_v \text{ for } T_j]}, \text{ Btu/h per W.}$$

$$EER^{k=2}(T_2) = \frac{\dot{Q}_c^{k=2}(T_2) \quad [\text{Eqn. 4.1.3-3, substituting } T_2 \text{ for } T_j]}{\dot{E}_c^{k=2}(T_2) \quad [\text{Eqn. 4.1.3-4, substituting } T_2 \text{ for } T_j]}, \text{ Btu/h per W.}$$

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

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

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.

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

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 4.2.1.2, 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.

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.

$$\frac{n_j}{N} =$$

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

$$\frac{n_j}{N}$$

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 Tj for a given generalized climatic region and 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
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	−130	0	0	0	.005	0	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,

$$BL\left(T_J\right) = \frac{\left(65 - T_J\right)}{65 - T_{OD}} \cdot C \cdot DHR \quad 4.2\text{-}2$$

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 1.21), 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

$$DHR_{max} = \left\{ \begin{array}{ll} 2 \cdot \dot{Q}_h^k(47) \cdot \left[\dfrac{65 - T_{OD}}{60}\right], & \text{for Regions I, II, III, IV, \& VI} \\ 2 \cdot 2 \cdot \dot{Q}_h^k(47), & \text{for Region V} \end{array} \right\} \begin{array}{l}\text{Rounded to the nearest} \\ \text{standardized DHR} \\ \text{given in Table 18.}\end{array}$$

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, $\dot{Q}_h^k(47) = \dot{Q}_h^k(47)$, 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 $H1_N$ Test.

3. For two-capacity heat pumps that are designed to operate exclusively, via an equipment lockout feature, at low compressor capacity when space cooling while using both high and low capacities when space heating, $\dot{Q}_h^k(47) = \dot{Q}_h^k(47)$, the space heating capacity determined from the $H1_n$ Test.

If the optional $H1_N$ Test is conducted on a variable-speed heat pump, the manufacturer has the option of defining $\dot{Q}_h^k(47)$ as specified above in item 2 or as $\dot{Q}_h^k(47) = \dot{Q}_h^{k=N}(47)$, the space heating capacity determined from the $H1_N$ Test.

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,

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

whichever is less; the heating mode load factor for temperature bin_j, dimensionless.

$\dot{Q}_h(T_j)$ = the space hearing 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.

$PFL_j = 1 - C_{D>}^h \cdot [1 - X(T_j)]$, 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.

For non-defrost heat pumps covered under Section 3.6.1.1, determine its space heating capacity, $\dot{Q}_h(T_j)$, and the electrical power consumption, $\dot{E}_h(T_j)$, as specified in Section 4.2.1.1. For heat pumps having a heat comfort controller that are covered under Section 3.6.1.2, determine $\dot{Q}_h(T_j)$ and $\dot{E}_h(T_j)$ as specified in Section 4.2.1.2. For all other heat pumps covered under Section 4.2.1 (and Section 3.6.1), calculate $\dot{Q}_h(T_j)$ and $\dot{E}_h(T_j)$ using,

$$\dot{Q}_h(T_j) = \begin{cases} \dot{Q}_h(17) + \dfrac{\left[\dot{Q}_h(47) - \dot{Q}_h(17)\right] \cdot (T_j - 17)}{47 - 17}, \text{ if } T_j \ge 45\text{ °F or } T_j \le 17\text{ °F} \\ \dot{Q}_h(17) + \dfrac{\left[\dot{Q}_h(35) - \dot{Q}_h(17)\right] \cdot (T_j - 17)}{35 - 17}, \text{ if } T_j \ge 17\text{ °F} < T_j < 45\text{ °F} \end{cases} \qquad (4.2.1\text{-}4)$$

$$\dot{E}_h(T_j) = \begin{cases} \dot{E}_h(17) + \dfrac{\left[\dot{E}_h(47) - \dot{E}_h(17)\right] \cdot (T_j - 17)}{47 - 17}, \text{ if } T_j \ge 45\text{°F or } T_j \le 17\text{°F} \\ \dot{E}_h(17) + \dfrac{\left[\dot{E}_h(35) - \dot{E}_h(17)\right] \cdot (T_j - 17)}{35 - 17}, \text{ if } 17\text{°F} < T_j < 45\text{°F} \end{cases} \qquad (4.2.1\text{-}5)$$

where
$\dot{Q}_h(47)$ and $\dot{E}_h(47)$ are determined from the *H1* Test and calculated and 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.

4.2.1.1 *Space heating capacity and the electrical power consumption calculations for a non-defrost heat pump.* Calculate the space heating capacity, $\dot{Q}_h(T_j)$, and the electrical power consumption, $\dot{E}_h(T_j)$, for a non-defrost heat pump covered under Section 3.6.1.1 using,

$$\dot{Q}_h(T_j) = \dot{Q}_h(47) + \left[\dot{Q}_h(62) - \dot{Q}_h(47)\right] \cdot \frac{(T_j - 47)}{62 - 47}$$

$$\dot{E}_h(T_j) = \dot{E}_h(47) + \left[\dot{E}_h(62) - \dot{E}_h(47)\right] \cdot \frac{(T_j - 47)}{62 - 47}$$

where
$\dot{Q}_h(62)$ and $\dot{E}_h(62)$ are determined from the *H0* Test, $\dot{Q}_h(47)$ and $\dot{E}_h(47)$ are determined from the H1 Test, and all four quantities are calculated as specified in Section 3.7. The low temperature cut-out factor, (T_j), must be greater than or equal to 37 °F, in accordance with Section 3.6.1.1.

4.2.1.2 *Space heating capacity and the electrical power consumption calculations for a heat pump having a heat comfort controller.* 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 Btu/lbm$_{da}$ · °F) from the results of the *H1* Test using:

$$\dot{m}_{da} = \overline{\dot{V}_s} \cdot 0.075 \frac{\text{lbm}_{da}}{\text{ft}^3} \cdot \frac{60\ \text{min}}{\text{hr}} = \frac{\overline{\dot{V}_{mx}}}{v_n' \cdot [1 + W_n]} \cdot \frac{60\ \text{min}}{\text{hr}} = \frac{\dot{V}_{mx}}{v_n} \cdot \frac{60\ \text{min}}{\text{hr}}$$

$$C_{p,da} = 0.24 + 0.444 \cdot W_n$$

where,

$$\overline{\dot{V}_s}, \overline{\dot{V}_{mx}}, v_n' (\text{or } v_n), \text{ and } W_n$$

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,

$$T_o(T_j) = 70\ °\text{F} + \frac{\dot{Q}_{hp}(T_j)}{\dot{m}_{da} \cdot C_{p,da}}.$$

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 $\dot{E}_h(T_j) = E_{hp}(T_j)$].

For outdoor bin temperatures where $T_o(T_j) < T_{CC}$, $\dot{Q}_h(Tj) = \dot{Q}_{hp}(T_j) + \dot{Q}_{CC}(T_j)$

$$\dot{E}_h(T_j) = \dot{E}_{hp}(T_j) + \dot{E}_{CC}(T_j)$$

where,

$$\dot{Q}_{CC}(T_j) = \dot{m}_{da} \cdot C_{p,da} \cdot \left[T_{CC} - T_o(T_j)\right]$$

$$\dot{E}_{CC}(T_j) = \frac{\dot{Q}_{CC}(T_j)}{3.413 \frac{\text{Btu}}{\text{W} \cdot \text{h}}}.$$

Calculate the *HSPF* of a heat pump having a heat comfort controller as specified in Section 4.2.1 with the exception of using the space heating capacity and electrical power given above [$\dot{Q}_h(T_j)$ and $\dot{E}_h(T_j)$] for the calculations at each outdoor bin temperature.

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

$$\frac{e_h(T_j)}{N} \text{ and } \frac{RH_h(T_j)}{N}$$

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,

$$\dot{Q}_h(T_j) = \dot{Q}_h^{k=1}(T_j) + \frac{\dot{Q}_h^{k=2}(T_j) - \dot{Q}_h^{k=1}(T_j)}{FP_h^{k=2} - FP_h^{k=1}} \cdot \left[FP_h(T_j) - FP_h^{k=1}\right]$$

$$\dot{E}_h(T_j) = \dot{E}_h^{k=1}(T_j) + \frac{\dot{E}_h^{k=2}(T_j) - \dot{E}_h^{k=1}(T_j)}{FP_h^{k=2} - FP_h^{k=1}} \cdot \left[FP_h(T_j) - FP_h^{k=1}\right]$$

where the space heating capacity and electrical power consumption at both low capacity (k=1) and high capacity (k=2) at outdoor temperature T_j are determined using,

$$\dot{Q}_h^k(T_j) = \begin{cases} \dot{Q}_h^k(17) + \dfrac{\left[\dot{Q}_h^k(47) - \dot{Q}_h^k(17)\right] \cdot (T_j - 17)}{47 - 17}, & \text{if } T_j \geq 45\ ^\circ\text{F or } T_j \leq 17\ ^\circ\text{F} \\ \dot{Q}_h^k(17) + \dfrac{\left[\dot{Q}_h^k(35) - \dot{Q}_h^k(17)\right] \cdot (T_j - 17)}{35 - 17}, & \text{if } 17\ ^\circ\text{F} < T_j < 45\ ^\circ\text{F} \end{cases} \qquad (4.2.2\text{-}1)$$

$$\dot{E}_h^k(T_j) = \begin{cases} \dot{E}_h^k(17) + \dfrac{\left[\dot{E}_h^k(47) - \dot{E}_h^k(17)\right] \cdot (T_j - 17)}{47 - 17}, & \text{if } T_j \geq 45\ ^\circ\text{F or } T_j \leq 17\ ^\circ\text{F} \\ \dot{E}_h^k(17) + \dfrac{\left[\dot{E}_h^k(35) - \dot{E}_h^k(17)\right] \cdot (T_j - 17)}{35 - 17}, & \text{if } 17\ ^\circ\text{F} < T_j < 45\ ^\circ\text{F} \end{cases} \qquad (4.2.2\text{-}2)$$

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) 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

$$\frac{e_h(T_j)}{N}$$

and

$$\frac{RH_c(T_j)}{N}$$

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 high and/or low capacity operation at low outdoor temperatures, the manufacturer must supply information regarding the cutoff temperature(s) so that you can select the appropriate equations.

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,

$$\dot{Q}_h^{k=1}(T_j) = \begin{cases} \dot{Q}_h^{k=1}(47) + \dfrac{\left[\dot{Q}_h^{k=1}(62) - \dot{Q}_h^{k=1}(47)\right] \cdot (T_j - 47)}{62 - 47}, & \text{if } T_j \geq 40\ ^\circ\text{F} \\ \dot{Q}_h^{k=1}(17) + \dfrac{\left[\dot{Q}_h^{k=1}(35) - \dot{Q}_h^{k=1}(17)\right] \cdot (T_j - 17)}{35 - 17}, & \text{if } 17\ ^\circ\text{F} \leq T_j \leq 40\ ^\circ\text{F} \\ Q_h^{k=1}(17) + \dfrac{\left[\dot{Q}_h^{k=1}(47) - \dot{Q}_h^{k=1}(17)\right] \cdot (T_j - 17)}{47 - 17}, & \text{if } T_j < 17\ ^\circ\text{F} \end{cases}$$

$$\dot{E}_h^{k=1}(T_j) = \begin{cases} \dot{E}_h^{k=1}(47) + \dfrac{\left[\dot{E}_h^{k=1}(62) - \dot{E}_h^{k=1}(47)\right] \cdot (T_j - 47)}{62 - 47}, & \text{if } T_j \geq 40\ ^\circ\text{F} \\ \dot{E}_h^{k=1}(17) + \dfrac{\left[\dot{E}_h^{k=1}(35) - \dot{E}_h^{k=1}(17)\right] \cdot (T_j - 17)}{35 - 17}, & \text{if } 17\ ^\circ\text{F} \leq T_j < 40\ ^\circ\text{F} \\ \dot{E}_h^{k=1}(17) + \dfrac{\left[\dot{E}_h^{k=1}(47) - \dot{E}_h^{k=1}(17)\right] \cdot (T_j - 17)}{47 - 17}, & \text{if } T_j < 17\ ^\circ\text{F} \end{cases}$$

b. Evaluate the space heating capacity and electrical power consumption [$\dot{Q}_h^{k=2}(T_j)$ and $\dot{E}_h^{k=2}(T_j)$] of the heat pump when operating at high compressor capacity and outdoor temperature T_j by solving Equations 4.2.2–1 and 4.2.2–2, 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 $\dot{Q}_h^{k=2}$ (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) and 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, $\dot{Q}_h^{k=1}(T_j) \geq BL(T_j)$.

$$\frac{e_h(T_j)}{N} = \frac{X^{k=1}(T_j) \cdot \dot{E}_h^{k=1}(T_J) \cdot \delta'(T_j)}{PLF_j} \cdot \frac{n_j}{N} \qquad (4.2.3\text{-}1)$$

$$\frac{RH(T_j)}{N} = \frac{BL(T_j) \cdot \left[1 - \delta'(T_j)\right]}{3.413 \frac{Btu/h}{W}} \cdot \frac{n_j}{N} \qquad (4.2.3\text{-}2)$$

where,

$X^{k=1} = BL(T_j/\dot{Q}_h^{k=1}(T_j)$, 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.

δ'(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,

$$\delta'\left(T_j\right) = \begin{cases} 0, & \text{if } T_j \leq T_{off} \\ 1/2, & \text{if } T_{off} < T_j \leq T_{on} \\ 1, & \text{if } T_j > T_{on} \end{cases} \qquad (4.2.3\text{-}3)$$

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:

(1) the heat pump locks out low capacity operation at low outdoor temperatures and

(2) 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 $\frac{RH\left(T_j\right)}{N}$ using Equation 4.2.3-2. Evaluate $\frac{e_h\left(T_j\right)}{N}$ using,

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

where,

$$X^{k=1}\left(T_j\right) = \frac{\dot{Q}_h^{k=2}\left(T_j\right) - BL\left(T_j\right)}{\dot{Q}_h^{k=2}\left(T_j\right) - \dot{Q}_h^{k=1}\left(T_j\right)}.$$

$X^{k=2}(T_j)$ 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, δ(T), using Equation 4.2.3–3.

4.2.3.3 Heat pump only operates at high (k=2) compressor capacity at temperature Tj 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

$$\frac{RH\left(T_j\right)}{N}$$

using Equation 4.2.3–2. Evaluate

$$\frac{e_h\left(T_j\right)}{N}$$

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,

$X^{k=2}(T_j) = BL(T_j) / \dot{Q}_h^{k=2}(T_j)$.

$PLF_j = 1 - C_D^h \cdot [1 - X^{k=2}(T_j)]$.

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, $BL(T_j)$.*

$$\frac{e_h(T_j)}{N} = \dot{E}_h^{k=2}(T_j) \cdot \delta''(T_j) \cdot \frac{n_j}{N}$$

$$\frac{RH(T_j)}{N} = \frac{BL(T_j) - \left[\dot{Q}_h^{k=2}(T_j) \cdot \delta''(T_j)\right]}{3.413 \frac{\text{Btu/h}}{\text{W}}} \cdot \frac{n_j}{N}$$

where,

$$\delta''(T_j) = \begin{cases} 0, \text{ if } T_j \le T_{off} \text{ or } \dfrac{\dot{Q}_h^{k=2}(T_j)}{3.413 \cdot \dot{E}_h^{k=2}(T_j)} < 1 \\ 1/2, \text{ if } T_{off} < T_j \le T_{on} \text{ and } \dfrac{\dot{Q}_h^{k=2}(T_j)}{3.413 \cdot \dot{E}_h^{k=2}(T_j)} \ge 1 \\ 1, \text{ if } T_j > T_{on} \text{ and } \dfrac{\dot{Q}_h^{k=2}(T_j)}{3.413 \cdot \dot{E}_h^{k=2}(T_j)} \ge 1 \end{cases}$$

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 4.2.2–1 and 4.2.2–2, respectively, for k=2. Determine the Equation 4.2.2–1 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, $\dot{Q}_h^{k=i}(T_j)$, and electrical power consumption, $\dot{E}_h^{k=1}(T_j)$, 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:

$$M_Q = \left[\frac{\dot{Q}_h^{k=1}(62) - \dot{Q}_h^{k=1}(47)}{62 - 47} \cdot \left(1 - N_Q\right)\right] + \left[N_Q \cdot \frac{\dot{Q}_h^{k=2}(35) - \dot{Q}_h^{k=2}(17)}{35 - 17}\right]$$

$$M_E = \left[\frac{\dot{E}_h^{k=1}(62) - \dot{E}_h^{k=1}(47)}{62 - 47} \cdot \left(1 - N_E\right)\right] + \left[N_E \cdot \frac{\dot{E}_h^{k=2}(35) - \dot{E}_h^{k=2}(17)}{35 - 17}\right]$$

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

$$\frac{e_h\left(T_j\right)}{N} \text{ and } \frac{RH\left(T_j\right)}{N}$$

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

$$\frac{e_h\left(T_j\right)}{N} \text{ and } \frac{RH\left(T_j\right)}{N}$$

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,

$$\frac{e_h(T_j)}{N} = \dot{E}_h^{k=i}(T_j) \cdot \delta'(T_j) \cdot \frac{n_j}{N}$$

where,

$$\dot{E}_h^{k=i}(T_j) = \frac{\dot{Q}_h^{k=i}(T_j)}{3.413 \frac{Btu/h}{W} \cdot COP_h^{k=i}(T_j)}$$

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 $k=i$.

$COP_h^{k=i}(T_j)$ = 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 $COP_h^{k=i}(T_j)$ using,

$$COP_h^{k=i}(T_j) = A + B \cdot T_j + C \cdot T_j^2.$$

For each heat pump, determine the coefficients A, B, and C by conducting the following calculations once:

$$D = \frac{T_3^2 - T_4^3}{T_{vh}^2 - T_4^2}$$

$$B = \frac{COP^{k=2}\left(T_4\right) - COP^{k=1}\left(T_3\right) - D \cdot \left[COP^{k=2}\left(T_4\right) - COP^{k=v}\left(T_{vh}\right)\right]}{T_4 - T_3 - D \cdot \left(T_4 - T_{vh}\right)}$$

$$C = \frac{COP^{k=2}\left(T_4\right) - COP^{k=1}\left(T_3\right) - B \cdot \left(T_4 - T_3\right)}{T_4^2 - T_3^2}$$

$$A = COP^{k=2}\left(T_4\right) - B \cdot T_4 - C \cdot T_4^2.$$

where,

T_3 = the outdoor temperature at which the heat pump, when operating at minimum compressor speed, provides a space heating capacity that is equal to the building load [$\dot{Q}_h^{k=1}(T_3) = BL(T_3)$], °F. Determine T_3 by equating Equations 4.2.4–1 and 4.2–2 and solving for outdoor temperature.

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 by 4.2.2–1 ($k=2$) and 4.2–2 and solving for outdoor temperature.

$$COP^{k=1}(T_3) = \frac{\dot{Q}_h^{k=1}(T_3) \quad [\text{Eqn. 4.2.4-1, substituting } T_3 \text{ for } T_j]}{3.413 \frac{\text{Btu/h}}{\text{W}} \cdot \dot{E}_h^{k=1}(T_3) \quad [\text{Eqn. 4.2.4-2, substituting } T_3 \text{ for } T_j]}$$

$$COP^{k=v}(T_{vh}) = \frac{\dot{Q}_h^{k=v}(T_{vh}) \quad [\text{Eqn. 4.2.4-3, substituting } T_{vh} \text{ for } T_j]}{3.413 \frac{\text{Btu/h}}{\text{W}} \cdot \dot{E}_h^{k=v}(T_{vh}) \quad [\text{Eqn. 4.2.4-4, substituting } T_{vh} \text{ for } T_j]}$$

$$COP^{k=2}(T_4) = \frac{\dot{Q}_h^{k=2}(T_4) \quad [\text{Eqn. 4.2.2-1, substituting } T_4 \text{ for } T_j]}{3.413 \frac{\text{Btu/h}}{\text{W}} \cdot \dot{E}_h^{k=2}(T_4) \quad [\text{Eqn. 4.2.2-2, substituting } T_4 \text{ for } T_j]}.$$

4.2.4.3 *Heat pump must operate continuously at maximum (k=2) compressor speed at temperature T_j, $BL(T_j) \geq \dot{Q}_h^{k=2}(T_j)$.* Evaluate the Equation 4.2–1 quantities

$$\frac{e_h(T_j)}{N} \text{ and } \frac{RH(T_j)}{N}$$

as specified in Section 4.2.3.4 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.

4.3 *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 1.21, 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 preferably correspond to the actual design heating requirement (*DHR*) if known. But 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 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 the Code of Federal Regulations. Round Section 4.2 *HSPF* values and Section 4.3 *APF* values as per paragraphs (ii) and (iii), respectively, of Subpart B, § 430.23(m)(3) of the Code of Federal Regulations.

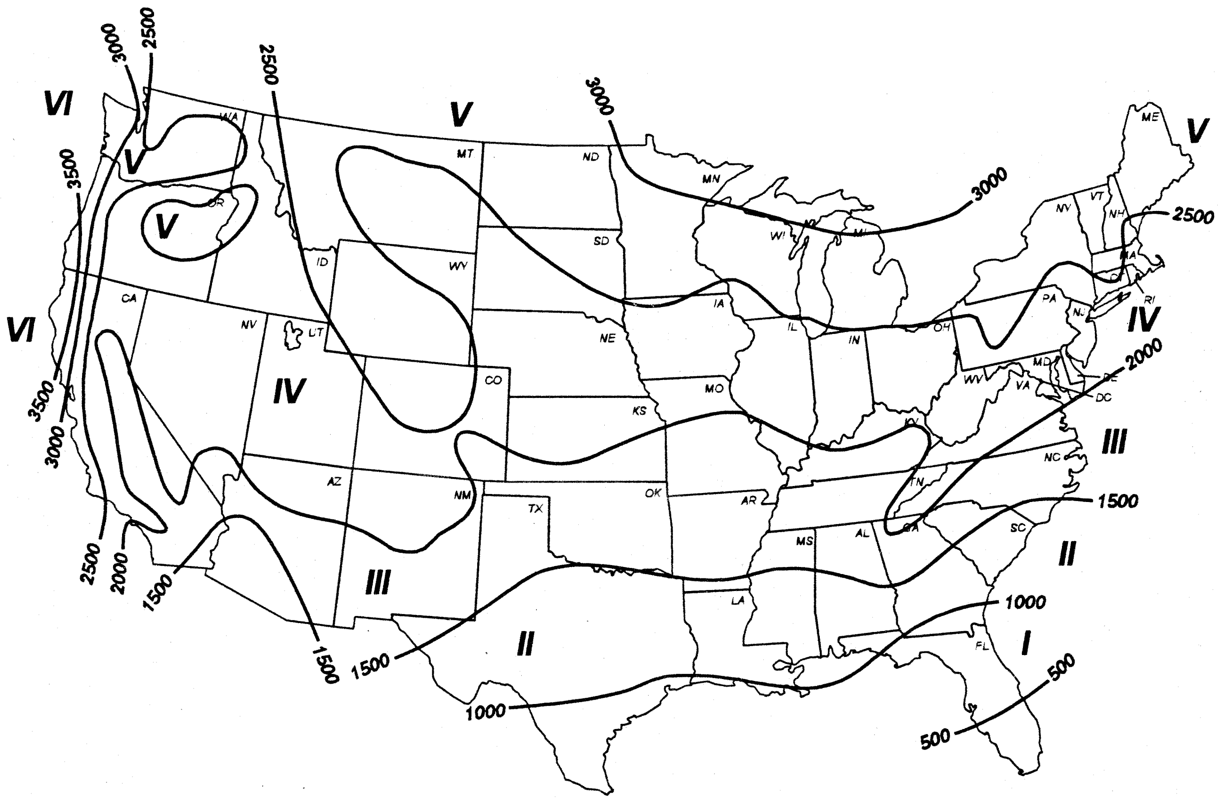

Figure 2 Heating Load Hours (HLH_A) for the United States

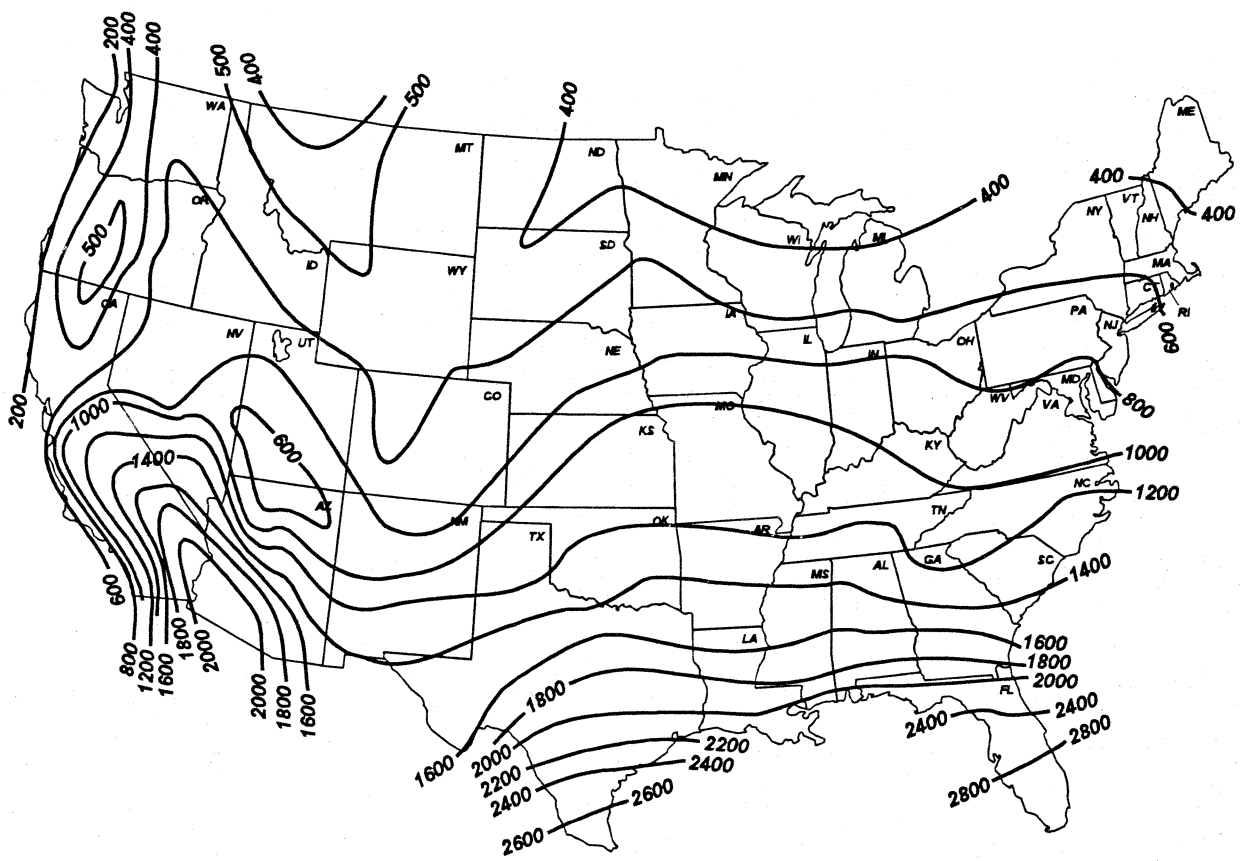

Figure 3 Cooling Load Hours (CLH_A) for the United States

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