AIR-CONDITIONING, HEATING, & REFRIGERATION INSTITUTE
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2111 Wilson Boulevard Suite 500 Arlington VA 22201-3001 USA
Phone 703 524 8800 | Fax 703 562 1942
www.ahrinet.org

March 24, 2014

Ms. Brenda Edwards
U.S. Department of Energy
Building Technologies Program, Mailstop EE-2J
1000 Independence Avenue SW
Washington, DC 20585

Re: Energy Conservation for Certain Industrial Equipment: Alternative Efficiency Determination Methods and Test Procedures for Walk-In Coolers and Walk-In Freezers *[Docket Number EERE–2011–BT–TP–0024]*

Dear Ms. Edwards:

These comments are submitted by the Air-Conditioning, Heating, and Refrigeration Institute (AHRI) in response to the U.S. Department of Energy's (DOE) supplemental notice of proposed rulemaking (SNOPR) on the alternative efficiency determination methods (AEDMs) and test procedures for walk-in coolers and walk-in freezers appearing in the *Federal Register* on February 20, 2014.

AHRI is the trade association representing manufacturers of heating, cooling, water heating, and commercial refrigeration equipment. More than 300 members strong, AHRI is an internationally recognized advocate for the industry, and develops standards for and certifies the performance of many of the products manufactured by our members. In North America, the annual output of the HVACR industry is worth more than $20 billion. In the United States alone, our members employ approximately 130,000 people, and support some 800,000 dealers, contractors, and technicians.

As discussed in detail below, AHRI has some concerns with the regulatory approach of the proposed rule. In the following comments, AHRI proposes several modifications to ensure that the rulemaking is appropriate for this equipment. AHRI is also pleased to announce that planned revisions to AHRI Standard 1250-2009, *Performance Rating of Walk-In Coolers and Freezers* are complete and the standard is on track to be published by mid-April 2014. These revisions address and incorporate all proposed modifications outlined in the SNOPR. Therefore, we sincerely hope that DOE will be able to adopt the standard, in its entirety, as the new revised test procedure for unit coolers and condensers sold separately, as well as for matched systems.

Summary of updates to 1250-2009 and timeline to publication

The following modifications to the test procedure for WICF refrigeration components have been accounted for in the revision to AHRI 1250:

- Clarifications to the defrost test procedure;
- An alternative method for calculating the defrost energy and heat load of a system with electric defrost in lieu of a frosted coil test;
- A method for calculating defrost energy and heat load of a system with hot gas defrost;
- Change to the minimum fan speed and duty cycle during the off-cycle evaporator fan test;
- Removal of the refrigerant oil and refrigerant composition analysis testing requirements;
- Clarifications and changes to the temperature measurement requirements, intended to reduce testing burden;
- Addition of a test condition tolerance for electrical power frequency and removal the test condition tolerance for temperature of air leaving the unit;
- Quantification of the requirements for insulating refrigerant lines;
- Clarification of piping length requirement;
- Changes to the list of tests for unit coolers in table 15 to achieve consistency with another similar test method; and
- Clarification of voltage imbalance for three-phase power.

The March 24, 2014 draft of AHRI 1250 is attached as Exhibit-B.

AHRI appreciates the opportunity to comment on issues in which the DOE has expressed interest.

Issues on which DOE seeks comments – AEDMs

The Department proposes to allow WICF refrigeration system manufacturers to use AEDMs to rate and certify their basic models by using the projected energy efficiency derived from these simulation models in lieu of testing.

1. DOE requests comment on its proposal to align AEDM validation requirements for WICF refrigeration equipment to the validation requirements for commercial HVAC, refrigeration, and WH equipment.

   AHRI Comments: Generally members support DOE's proposal to align AEDM validation requirements for WICF refrigeration equipment to the validation requirements for commercial HVAC, refrigeration, and WH equipment; however, concern has been raised on one issue: The WICF Basic Model definition may need to be amended to account for correct application of complex and unique equipment with respect to AEDMs. This issue is examined in depth later in this letter.

2. DOE requests comment on the following tolerances for WICF AEDMs. For energy consumption metrics, the AEDM result for a model must be equal to or greater than 95 percent of the tested results for that same model. For energy efficiency metrics, the AEDM results for a model must be less than or equal to 105 percent of the tested results for that same model.

   AHRI Comments: AHRI supports DOEs proposal to align the AEDM tolerances for energy consumption and energy efficiency metrics with those of other commercial equipment. Further, AHRI also agrees with the DOEs proposal to make a sampling plan for walk-ins consistent with other commercial equipment regulated under EPCA.

3. DOE seeks comment regarding the proposed requirement imposed on the manufacturer to re-certify any basic model with test data, including test data provided by DOE, in the case of a model failing to meet its AEDM rating.

   AHRI Comments: AHRI requests that the same requirements developed for commercial HVAC, refrigeration and water heating equipment (AEDM final rulemaking published on December 31, 2013; 78FR79579) apply to walk-in coolers and freezers. This should include provisions on re-certification of basic models as well as on witness testing. AHRI also hopes DOE will take into consideration comments submitted in response to the Certification of Commercial HVAC, WH and Refrigeration Equipment, Docket No. EERE-2013-BT-NOC-0023, on February 13, 2014, specifically those regarding engineered-to-order (ETO) products.

4. DOE requests comment on its proposal to not require re-validation of an AEDM upon every change in a federal conservation standard or test procedure, but retain discretion to evaluate each case individually and require re-validation on a case-by-case basis in the NOPR upon issuance of a final standard rule or test procedure.

   AHRI Comments: Re-validation should only be required when a change in the test procedure is significant enough to result in a product having a different rated energy consumption or efficiency.

5. DOE requests comment on whether 90 days is an appropriate amount of time to complete the re-validation, re-rating and re-certification steps for cases where they are necessary for AEDMs.

   AHRI Comments: This question contradicts DOE's own conclusions on page 9828 of the Federal Register which states that the Department is not proposing a time limit to re-validate AEDMs. We agree with DOE that a time limit to re-validate an AEDM should not be imposed. This is consistent with the position taken by the Department in the final AEDM rulemaking for commercial HVAC, refrigeration and water heating equipment published on December 31, 2013.

Issues on which DOE seeks comments – Testing and rating unit coolers and condensing units sold separately

This SNOPR puts forth an alternative method for testing and rating the WICF refrigeration system for unit coolers and condensing units that are sold separately.

6. DOE requests comment on its proposal to allow unit coolers and condensing units to be rated separately, and particularly the nominal values described in Table III.6.

   AHRI Comments: The requirements for rating systems and components may unfairly disadvantage companies who manufacture both unit coolers and condensers unless further guidance is provided. There are two cases where the unit cooler and condensing unit could be covered under the matched rating requirement. The first case is a packaged system which can only be tested and evaluated as a matched system since the condensing unit and unit cooler portion cannot be tested as separate components. The second case is where a manufacturer markets several options for split-system matched pairs[1]. To address the concern for certification requirements, AHRI proposes clarification on components which can be sold separately, or as a pair, as follows with additions underlined, deletions shown ~~struck through~~ and all changes in red text:

   > (1) A manufacturer that only produces unit coolers or a manufacturer that produces both unit coolers and condensing units that are marketed and sold as separate basic models for use in a WICF refrigeration system would use the test method described above to establish the WICF refrigeration system rating for each unit cooler (system performance would be established by testing the unit cooler as though it is to be connected to a multiplex system (i.e., using the "Walk-in Unit Cooler Match to Parallel Rack System" test method in AHRI 1250, section 7.9))--then, the ~~unit cooler~~ manufacturer would certify the compliance of those basic models with the WICF refrigeration system standard; (2) a manufacturer that only produces condensing units or a manufacturer that produces both unit coolers and condensing units that are marketed and sold as separate basic models would use the test method described above to establish the WICF refrigeration system rating for each condensing unit (system performance would be established by testing each condensing unit and combining it with the unit cooler nominal values (as proposed in this SNOPR))--then, the ~~condensing unit~~ manufacturer would certify compliance of those basic models with the WICF refrigeration system standard; or (3) a manufacturer that produces both unit cooler basic models and condensing unit basic models that are marketed and sold as a matched system would use the test method in AHRI 1250 to test the unit cooler and the condensing unit as a matched system to get a WICF

[1] MOHAVE hot gas refrigeration technical guide manufactured by BOHN
http://www.cpsohio.com/app/load/manual1.aspx?id=BOH0051

refrigeration system rating for each matched system it produces and then certify compliance.

To extend the proposal to address the rating of walk-in cooler and walk in freezer refrigeration systems components sold separately, AHRI recommends the following addition, shown underlined, red, to 42 U.S.C. §431.304:

(12) *Rating of walk-in cooler and freezer refrigeration system components sold separately*

(i) A unit cooler, if sold separately, shall be rated using the method for testing a unit cooler connected to a multiplex condensing system.

(ii) A condensing unit, if sold separately, shall be rated using the following nominal values:

…

(iii) Only fixed capacity condensing units may be certified in this manner. Multiple-capacity condensing units must be rated and certified as part of a matched system.

(iiii) Any unit cooler that is rated using the method for testing a unit cooler connected to a multiplex condensing system or any fixed capacity condensing unit that is rated using the nominal values shall not be required to be rated again if such equipment is sold both separately as well as together as part of a refrigeration system. Unless required by Section (12)(iii), no condensing unit or unit cooler must be rated as part of a matched refrigeration system unless they are exclusively marketed and sold as a matched system.

Issues on which DOE seeks comments – Modifications to the test procedure for WICF refrigeration components

7. DOE seeks comment on its nominal values for calculating electric defrost power and heat load in the absence of a full defrost test or for an individual condensing unit. DOE also seeks comment on its nominal values for calculating hot gas defrost power and heat load. The nominal values may be found in sections III. B. 1. and III. B. 2.

- **AHRI Comments:** AHRI agrees to proposed changes and nominal values with respect to the test method; however, there is concern on the effect these rating strategies will have on minimum levels and would like them to be reevaluated. We appreciate the consideration of suggested revisions to AHRI 1250 made in comments on the energy conservation NOPR, Docket No. EERE-2008-BT-STD-0015, published in the Federal Register on September 11, 2013; however, several fundamental concepts were not incorporated and now need to be accounted for to ensure the economic and technical justifications. AHRI suggested that condensing units and unit coolers sold separately should have a separate metric than the AWEF metric appropriate for matched split-systems and packaged systems. While this strategy would allow for a direct comparison between like products, it would not allow for a comparison of components to systems. The DOE's strategy to pair tested components with nominal values for a match does allow for a component to be rated with the AWEF metric. AHRI strongly recommends that DOE conduct a thorough analysis, to be publicly reviewed, to alleviate the concern that efficiency levels proposed in the NOPR may not be appropriate for refrigeration components rated separately and to allow for full evaluation of the impact of changes to the test procedure. The short timeframe for comments to this SNOPR coupled with the simultaneous release of several DOE rulemakings affecting the same manufacturer base have limited resources necessary for a full industry analysis of the impact. While AHRI supports the use of AEDMs for this equipment, the evaluation of the applicability of all HVAC provisions to WICFs was not afforded sufficient time for complete analysis. For example, the Commercial Certification Working Group of the Appliance Standards and Rulemaking Federal Advisory Committee (ASRAC) was afforded the opportunity to recommend amended basic model definitions for commercial products in the scope of the rulemaking and to provide guidance for implementing the verification procedures for each type of equipment. The Commercial Certification Working Group met for several months to establish these recommendations. While most of the recommendations can be implemented for WICFs, this equipment is complex and unique. AHRI requests additional time is allowed to review the basic model definition for WICFs with respect to AEDMs. Please refer to Exhibit-A for specific list of changes to 1250-2009.

8. DOE requests comment on its proposed amendments and clarifications to the test procedure; specifically (but not limited to) its modifications to the off-cycle evaporator fan test (section III. B. 3.), temperature measurement (section III. B. 5.), refrigerant line insulation (section III. B. 7.), and composition analysis (section III. B. 8.).

   AHRI Comments: Generally members support DOE's proposed amendments and clarifications to the test procedure. Refer to attachment for a summary of changes to AHRI 1250 in response to SNOPR comments.

   - Refrigerant Oil Testing (Section III. B. 4.) While DOE only proposed eliminating the requirement to measure the ratio of oil to refrigerant in the liquid refrigerant passing from the condenser to the unit cooler per ANSI-AHRI Standard 41.4 for condensing units with on-board oil-separators, AHRI recommends extending the requirement to all condensing units which will align 1250 with similar standards. Single compressor condensing units typically do not have on-board oil-separators and that testing refrigerants for those units would be extremely expensive and unnecessary. Section C3.4.6 of AHRI Standard 1250 has been removed in its entirety. Please refer to Exhibit-A for specific list of changes to 1250-2009.

<u>Issues on which DOE seeks comments – Modifications to procedure for measuring the insulation R-value of WICF panels</u>

9. DOE asks whether the proposed requirement to remove facers or protective skins from panels before measuring thermal resistance is appropriate.

   AHRI Comments: If DOE is proposing a maximum thickness of 1” “taken from the center of a panel”, then removal of the facers is a given. The original sample (typically a 4” thick panel) should be supplied intact, with facers and protective skins in-place to be removed by the test lab during the cutting operation to extract the final 1” thick test sample.

10. DOE asks whether the proposed requirement that a test sample for panel thermal resistance measurement be 1 inch in thickness and from the center of a WICF panel is appropriate.

   AHRI Comments: AHRI agrees with the proposed requirement that a test sample for panel thermal resistance measurement be 1 inch in thickness and from the center of a WICF panel is appropriate

11. DOE asks whether the tolerances specified for flatness (+/-0.03) and parallelism (.030 inches) for WICF panels before measuring thermal resistance are appropriate and sufficient.

   AHRI Comments: These tolerances are impractical for the purposes of the proposed test, are inconsistent with normal WICF panel manufacturers’ standard processes and are likely not within the capabilities of most current panel manufacturing processes. AHRI recommends that the sample tolerance specification for the panel be removed or not referenced.

12. DOE asks whether a tolerance of ±1 degree Fahrenheit for mean temperature during thermal resistance measurement is appropriate and sufficient.

   AHRI Comments: AHRI agrees that a tolerance of ±1 degree Fahrenheit for mean temperature during thermal resistance measurement is appropriate and sufficient.

13. DOE asks whether a 48-hour period after cutting the WICF panel for measuring thermal resistance is appropriate and sufficient.

   AHRI Comments: AHRI agrees that a 48-hour period after cutting the WICF panel for measuring thermal resistance is appropriate and sufficient.

<u>Issues on which DOE seeks comments – Modifications to procedure for measuring the insulation R-value of WICF panels</u>

14. DOE requests comment on its proposal to remove the test procedures in 10 CFR 431, Appendix A to Subpart R that reference ASTM C1363-05 and DIN EN 13164/13165 and their accompanying calculation procedures, leaving only ASTM C518-04 testing in 10 CFR 431.304 for establishing the thermal resistance of WICF panels

    AHRI Comments: AHRI supports DOE's proposal to remove the existing performance-based test procedures for WICF floor and non-floor panels and further recommends that DOE translate the proposed remaining test standard ASTM C518-04 to prescriptive requirements which would eliminate unnecessary testing requirements. Further, non-display doors should also have the option of meeting R-value standards as they are panel components.

15. DOE asks whether the surface heat transfer coefficients prescribed by NFRC 100[E0A1] are appropriate.

    AHRI Comments: AHRI has no issues with the proposed clarification.

<u>Blast Coolers, Blast Freezers, and Ripening Rooms</u>

AHRI would like to reiterate that the test procedures and efficiency standards in the NOPR do not apply to blast coolers and freezers. In the presentation slides provided at the October 9, 2013, public meeting, DOE defines walk-in coolers and freezers as "an enclosed storage space refrigerated to temperatures…" However, blast coolers and freezers are used for processing products and never for storage. Blast coolers and freezers and products specially designed as ripening rooms (i.e. banana or apple ripening rooms) do not operate in the same method as walk-in coolers or freezers and are defined by manufacturers as "food processing equipment."

Neither AHRI nor DOE have data to show whether food processing equipment have lower efficiencies than walk-in coolers and freezers. However, the energy consuming characteristics (as noted in the NOPR) are certainly different and DOE should not assume that the test procedures or efficiency levels for walk-in coolers and freezers should also apply to food processing equipment. Blast coolers and freezers should be held to different efficiency standards or excluded from the scope of this rulemaking.

There are also products used for commercial baking to proof and retard dough which do not operate in a manner similar to typical WICFs. The Proofer and Retarder/Proofer cabinets proof racks of dough products under controlled temperatures and humidity prior to baking. The retarders operate like a refrigerator by keeping the interior cool, which also provides an excellent environment for storing dough prior to proofing. These processes take place at humidity between 70% to 90%, depending on the product, and can take up to several days. These products should be excluded from the scope of this rulemaking.

AHRI appreciates the opportunity to provide these comments. If you have any questions regarding this submission, please do not hesitate to contact me.

Sincerely,

Laura Petrillo-Groh, PE
Engineering Manager, Regulatory Affairs
Direct: (703) 600-0335
Email: LPetrillo-Groh@ahrinet.org

Exhibits:

A. Summary of changes to AHRI Standard 1250-2009 in response to items 7 and 8
B. March 24, 2014 draft of AHRI Standard 1250

Exhibit A - Summary of Changes to 1250

Comment #	SNOPR Reference in FR Vol. 79, No. 34	FR Page #	Pre-publication Page #	DOE Comment	1250 WG Response	Change(s) to 1250-2009 (Excerpts from standard *italicized* ; additions underlined; deletions in ~~strikeout~~)	1250 Reference
6	Rating of Refrigeration Components (Section III. B. 1.)	9830	46	DOE requests comment on its proposal to allow unit coolers and condensing units to be rated separately, and	Refer to Answers to Comment 7 and 8		
7	Defrost Test (Section III. B. 2.)	9831	48	Allow manufacturers to choose between full defrost test and alternate methodology in SNOPR	Agree.	Modifications to Section C11. Defrost Test (freezer only) to add calculation methodology and nominal values for electric defrost	Sections altered or added: C11.1 obtain DF_d in W-hr; C11.2 frosted coil condition (calculation methodology and optional test); Eq. C17 obtain N_{DF} for systems w/o adaptive defrost; C11.3 time b/w dry coil defrost for adaptive defrost optional test and calculation; Equs added to C11.4
7	Defrost Test (Section III. B. 2.)	9832	52	DF = 0 for UC connected to a multiplex condensing system; For UC utilizing hot gas defrost and connected to a dedicated condensing system, $DF = 0.5*Q_{DF}$ (converted to W)	Agree.	Added text and equation	Eq. C14
7	Rating of Refrigeration Components (Section III. B. 1.)	9830	46	For UC sold separately: Approach to allow manufacturers to test UC with a parallel rack system and rate with AWEF metric	Agree.		Section 7.9
7	Rating of Refrigeration Components (Section III. B. 1.)	9830	46	For condensing unit sold separately: Approach to allow manufacturers to test condenser using mix/match test method	Agree.	Nominal values for testing condensing units sold separately added to C13 (Table III.6 from SNOPR incoporated for reference UC values); Note that the name of the test has changed from mix/match to "Method of testing condensing units for walk-in cooler and freezer systems where condenser is sold separately."	Section C13
8	Temperature Measurement (Section III. B. 5.)	9832	54	Increase refrigerant temperature measurement accuracy requirement from 0.2 °F to 0.5°F	Agree.	Increased minimum accuracy of temerature measurement to 0.5°F	Section 4, Table 1. Instrument Accuracy
8	Temperature Measurement (Section III. B. 5.)	9832	54	Required tolerance for test temperature be maintained at ±0.5°F for measurements at the inlet and outlet of the UC , and ±1.0°F for a other temperature measurements	Agree.	Removed all Leaving temperature Test Operating Tolerances from Table 2 and incoporated DOE's recommended tolerances in the SNOPR	*Section 4, Table 2. Test Operating and Test Condition Tolerances for Steady-State Test*
8	Temperature Measurement (Section III. B. 5.)	9832	55	Allow temperature measurement test to be conducted using sheathed sensors immersed in the flowing refrigerant temerature downtream of th UC	Agree.	*Refrigerant temperatures entering and leaving the Unit Cooler shall be measured by sheathed temperature sensors immersed in flowing refrigerant or by a temperature measuring instrument placed in a thermometer well and inserted into the refrigerant stream.*	Section C3.1.6
8	Temperature Measurement (Section III. B. 5.)	9832	55	No refrigerant temperature measurements other than those upstream and downstream of the UC would require a thermometer well or sheathed sensor.	Agree.	No additional changes required.	N/A
8	Refrigerant Oil Testing (Section III. B. 4.)	9832	54	Condensing units with on-board oil-filters woud not be required to perform the measurement of the ratio of oil to refrigerant in the liquid refrigerant passing from the condneser to the UC er ANSI/ASHRAE 41.1	Agree.	*The ratio of oil to refrigerant shall be less than 1 % by weight. ~~Unless~~ If the system ~~does not~~ contain s refrigerant oil and does not have on-board oil separators, tests for oil concentration shall be made a minimum of once per Test Series per ANSI/ASHRAE Standard 41.4.*	Section C3.4.6
8	Off-cycle Evaporator Fan Test (Section III. B. 3.)	9832	53	Ammend the test procedure such that "stir cycle" controls shall be adjusted so that the greater of a 50% duty cycle of the manufacturer default is used for measuring off-cyce fan energy	Agree.	Increased duty cycle from 25% to 50% in off-cycle evaporator fan test	Section C10
8	Off-cycle Evaporator Fan Test (Section III. B. 3.)	9832	53	Variable speed controls shall be adjusted so that the greater of 50% fan speed of the manufacturer's default fan speed shall be used for measuring off-cycle fan energy	Agree.	Increased fan speed from 25% to 50% in off-cycle evaporator fan test	Section C10
8	Refrigerant Line Insulation (Section III. B. 7.)	9833	56	(1) Insulation on pipe lines b/w the UC and condensing unit be set up as recommended by the manufacturer in installation literature or, if there is no recommendation, insulation shall be equivalent to a 1/2" thick insulation with a material having an R-Value of at least 3.7/in; (2) flow meters would not need to be insulated but must not contact the floor	Agree.	Provisions added	Section C3.8
8	Composition Analysis (Section III. B. 8.)	9833	57	Delete the requirement to test a sample of the superheated vapor refrigerant be extracted while the system is still running	Agree.	Deleted requirement	Section C3.3.6 has been deleted

EXHIBIT B

Draft AHRI Standard 1250-2014, Standard for Performance Rating of Walk-In Coolers and Freezers

The attached draft standard is current as of March 24, 2014 and is expected to be approved for publication on April 2, 2014. AHRI intends to forward the approved version to the Department of Energy without delay.

AHRI Standard 1250P (I-P)

DRAFT

2014 Standard for Performance Rating of Walk-In Coolers and Freezers

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Air-Conditioning, Heating, and Refrigeration Institute

2111 Wilson Boulevard, Suite 500
Arlington, VA 22201, USA
www.ahrinet.org

PH 703.524.8800
FX 703.528.3816

IMPORTANT

SAFETY DISCLAIMER

AHRI does not set safety standards and does not certify or guarantee the safety of any products, components or systems designed, tested, rated, installed or operated in accordance with this standard/guideline. It is strongly recommended that products be designed, constructed, assembled, installed and operated in accordance with nationally recognized safety standards and code requirements appropriate for products covered by this standard/guideline.

AHRI uses its best efforts to develop standards/guidelines employing state-of-the-art and accepted industry practices. AHRI does not certify or guarantee that any tests conducted under its standards/guidelines will be non-hazardous or free from risk.

Note:

This standard supersedes AHRI Standard 1250 (I-P)-2009
For SI ratings, Refer to AHRI Standard 1251 (SI) – 2014

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PERFORMANCE RATING OF WALK-IN COOLERS AND FREEZERS

Section 1. Purpose

1.1 *Purpose*. The purpose of this standard is to establish, for walk-in coolers and freezers: definitions; test requirements; rating requirements; minimum data requirements for Published Ratings; operating requirements; marking and nameplate data and conformance conditions.

1.1.1 *Intent*. This standard is intended for the guidance of the industry, including manufacturers, designers, installers, contractors and users.

1.1.2 *Review and Amendment*. This standard is subject to review and amendment as technology advances.

Section 2. Scope

2.1 *Scope*. This standard applies to mechanical refrigeration equipment consisting of an integrated single package refrigeration unit, or separate unit cooler and condensing unit sections, where the condensing section can be located either outdoor or indoor. Controls may be integral, or can be provided by a separate party as long as performance is tested and certified with the listed mechanical equipment accordingly.

2.2 *Exclusions.* This standard does not apply to:

a. Enclosures used for telecommunications switch gear or other equipment requiring cooling
b. Enclosures designed for medical, scientific or research purposes
c. Performance testing and efficiency characterization of large parallel rack refrigeration systems (condensing unit)

Section 3. Definitions

All terms in this document will follow the standard industry definitions in the *ASHRAE Wikipedia* website (http://wiki.ashrae.org/index.php/ASHRAEwiki) unless otherwise defined in this section.

3.2 *Annual Walk-in Energy Factor (AWEF).* A ratio of the total heat, not including the heat generated by the operation of refrigeration systems, removed, in Btu, from a walk-in box during one year period of usage for refrigeration to the total energy input of refrigeration systems, in watt-hours, during the same period.

3.3 *Energy Efficiency Ratio, (EER)*. A ratio of the Refrigeration Capacity in Btu/h to the power input values in watts at any given set of Rating Conditions expressed in Btu/W·h.

3.4 *Forced-Circulation Free-Delivery Unit Coolers (Unit Coolers).* A factory-made assembly, including means for forced air circulation and elements by which heat is transferred from air to refrigerant without any element external to the cooler imposing air resistance. These may also be referred to as Air Coolers, Cooling Units, Air Units or Evaporators.

3.5 *Glide.* The absolute value of the difference between the starting and ending temperatures of a phase-change process (condensation or evaporation) for a zeotropic refrigerant exclusive of any liquid subcooling or vapor superheating.

3.6 *Gross Refrigeration Capacity.* The heat absorbed by the refrigerant, Btu/h. This is the sum of the Net Refrigeration Capacity and the heat equivalent of the energy required to operate the Unit Cooler. This includes both sensible and latent cooling.

3.7 *High Box Load ($\dot{BLH}$).* Walk-in box load during a high load period.

3.8 *High Load Period.* A period of the day corresponding to frequent door openings, product loading events, and other design load factors,

3.9 *Load Factor.* A ratio of the total walk-in system heat load to the steady-state net refrigeration capacity.

3.10 *Low Box Load (BĿL).* Walk-in box load during a low load period.

3.11 *Low Load Period.* Any period of the day other than a high load period.

3.12 *Net Refrigeration Capacity.* The refrigeration capacity available for space and product cooling, Btu/h. It is equal to the Gross Refrigeration Capacity less the heat equivalent of energy required to operate the Unit Cooler (e.g: evaporator fans, defrost)

3.13 *Positive Displacement Condensing Unit*. A specific combination of refrigeration system components for a given refrigerant, consisting of one or more electric motor driven positive displacement compressors, condensers, and accessories as provided by the manufacturer.

3.14 *Published Rating.* A statement of the assigned values of those performance characteristics, under stated rating conditions, by which a unit may be chosen to fit its application. These values apply to all units of like nominal size and type (identification) produced by the same manufacturer. The term Published Rating includes the rating of all performance characteristics shown on the unit or published in specifications, advertising or other literature controlled by the manufacturer, at stated Rating Conditions.

3.14.1 *Application Rating*. A rating based on tests performed at application Rating Conditions, (other than Standard Rating Conditions).

3.14.2 *Standard Rating*. A rating based on tests performed at Standard Rating Conditions.

3.15 *Rated Fan Power.*

3.15.1 For single phase motors, total fan motor input power, W.

3.15.2 For polyphase motors, individual fan motor output power, W.

3.16 *Rating Conditions.* Any set of operating conditions under which a single level of performance results and which causes only that level of performance to occur.

3.16.1 *Standard Ratings Conditions.* Rating conditions used as the basis of comparison for performance characteristics.

3.17 *Refrigeration Capacity*. The capacity associated with the increase in total enthalpy between the liquid refrigerant entering the expansion valve and superheated return gas multiplied by the mass flow rate of the refrigerant.

3.18 *"Shall" or "Should".* "Shall" or "should" shall be interpreted as follows:

3.18.1 *Shall*. Where "shall" or "shall not" is used for a provision specified, that provision is mandatory if compliance with the standard is claimed.

3.18.2 *Should.* "Should," is used to indicate provisions which are not mandatory, but which are desirable as good practice.

3.19 *Refrigerant Saturation Temperature.* Refrigerant temperature at the Unit Cooler inlet or outlet determined either by measuring the temperature at the outlet of the two-phase refrigerant flow, for a Liquid Overfeed Unit Cooler, or by measuring refrigerant pressure and determining the corresponding temperature from reference thermodynamic tables or equations for the refrigerant, ºF. For zeotropic refrigerants, the corresponding temperature to a measured pressure is the refrigerant Dew Point.

3.20 *Steady-State Conditions*. An operating state of a system, including its surroundings, in which the extent of change with time is within the required limits (refer to Table 2).

3.21 *Temperature Difference (TD)*. The difference between the dry-bulb temperature of the air entering the Unit Cooler and the Refrigerant Saturation Temperature at the unit cooler outlet, ºF.

3.22 *Test Measurement.* The reading of a specific test instrument at a specific point in time. The Test Measurement may be averaged with other measurements of the same parameter at the same time to determine a Test Reading or averaged over the duration of the test to determine the value for the Test Run. Refer to Table C1 for test reading minimum time rate, number of test readings and minimum test duration.

3.23 *Test Reading.* The recording of one full set of the Test Measurements required to assess the performance of the test unit.

3.24 *Total Walk-in System Heat Load.* Total heat load to the walk-in system including walk-in box load and the heat load to the box contributed by the operation of the refrigeration system.

3.25 *Volatile Refrigerant.* A refrigerant which changes from liquid to vapor in the process of absorbing heat.

3.26 *Walk-in Box Load.* Heat load to the walk-in box resulting from conduction, infiltration and internal heat gains from equipment that is not related to the refrigeration system, such as lights and anti-sweat heaters, etc.

3.27 *Walk-in System High Load.* Total Walk-in System Heat Load during a High Load Period.

3.28 *Walk-in System Low Load.* Total Walk-in System Heat Load during a Low Load Period.

Section 4. Test Requirements

4.1 *Instruments.* All measuring instruments shall be selected to meet or exceed the accuracy criteria listed in Table 1 for each type of measurement. All temperature measurement shall be made in accordance with Table 2, Test Tolerance. Precision instruments and automated electronic data acquisition equipment shall be used to measure and record temperature, pressure and refrigerant flow rate test parameters. All measuring instruments and instrument systems (e.g. data acquisition coupled to temperature, pressure, or flow sensors) shall be calibrated by comparison to primary or secondary standards with calibrations traceable to National Institute of Standards and Technology (NIST) measurements, other recognized national laboratories, or derived from accepted values of natural physical constants. All test instruments shall be calibrated annually, whenever damaged, or when the accuracy is called into question.

Table 1. Instrumentation Accuracy

Measurement	Medium	Minimum Accuracy
Temperature	Air dry-bulb	± 0.5°F
	Air wet-bulb	
	Refrigerant liquid	
	Refrigerant vapor	± 0.5 °F
	Air Dew Point	± 0.5 °F
	Others	± 1.0 °F
Relative humidity[1]	Air	± 3 % points rh
Pressure	Refrigerant	Pressure corresponding to ± 0.2 °F of saturation temperature
	Air	±0.05 inches of mercury
Flow	Refrigerant	1 % of reading
	Liquids	1 % of reading
Electrical	Motor kilowatts/amperes/voltage	1 % of reading
	Auxiliary kilowatt input (e.g. heater)	
Speed	Motor / fan shaft	1 % of reading

Table 1. Instrumentation Accuracy		
Measurement	Medium	Minimum Accuracy
Specific Gravity	Brine	1 % of reading
Time	Hours / minutes / seconds	0.5 % of time interval

Note:

(1) Relative humidity and air dew point measurements are intended to confirm the dry coil condition.

Table 2. Test Operating and Test Condition Tolerances for Steady-State Test

	Test Operating Tolerance (1)	Test Condition Tolerance (2)
Indoor dry-bulb, °F		
Entering temperature	3.6	0.5
Indoor wet-bulb, °F		
Entering temperature	3.6	0.5
Outdoor dry-bulb, °F		
Entering temperature	3.6	1.0
Outdoor wet-bulb, °F		
Entering temperature	3.6	1.0
Electrical voltage, % of reading.	2.0	
Electrical Frequency, % of reading	1.0	

Notes:

(1) Test Operating Tolerance is the maximum permissible range of any measurement. When expressed as a percentage, the maximum allowable variation is the specified percentage of the average value,

(2) Test Condition Tolerance is the maximum permissible variation of the average value of the measurement from the specified test condition.

4.2 *Method of test/test procedure.* The test procedure for walk-in cooler and freezer systems that have matched unit coolers and condensing units, and the procedures of testing condensing units and unit coolers individually for mix-matched systems are described in Appendix C of this standard.

4.3 *Test conditions.* Walk-in systems, condensing units and unit coolers shall be tested under the standard rating conditions defined in Section 5.

Section 5. Rating Requirements

5.1 *Standard Ratings*. Standard Ratings shall be established at the Standard Rating Conditions in the following listed tables and shall include its associated power input and Energy Efficiency Ratio (EER). When tested with a specified motor, the associated compressor speed (external drive compressors only) shall also be included as part of the rating. The power required to operate all included accessories such as condenser fans, water pumps, controls, and similar accessories shall be accounted for in the power input and Energy Efficiency Ratio. When external accessories such as water pumps, remote fans, and similar accessories are required for the operation of the unit but not included with the unit, the manufacturer shall clearly state that the rated power input and Energy Efficiency Ratio do not account for additional power required by these external accessories. If a water-cooled condenser is used, the cooling water flow rate and pressure drop shall be specified as part of the rating. The condensing unit entering air wet bulb temperature only applies to evaporative condensers.

Table 3. Fixed Capacity Matched Refrigerator System, Condensing Unit Located Indoor

Test Description	Unit Cooler Air Entering Dry-Bulb, ˚F	Unit Cooler Air Entering Relative Humidity, %	Condenser Air Entering Dry-Bulb, ˚F	Condenser Air Entering Wet-Bulb, ˚F	Compressor Capacity	Test Objective
Off Cycle Fan Power	35	<50	-	-	Compressor Off	Measure fan input wattage during compressor off cycle
Refrigeration Capacity	35	<50	90	75	Compressor On	Determine Net Refrigeration Capacity of Unit Cooler, input power, and EER at Rating Condition

Table 4. Fixed Capacity Matched Refrigerator System, Condensing Unit Located Outdoor						
Test Description	**Unit Cooler Air Entering Dry-Bulb, ˚F**	**Unit Cooler Air Entering Relative Humidity, %**	**Condenser Air Entering Dry-bulb, ˚F**	**Condenser Air Entering Wet-Bulb, ˚ F**	**Compressor Capacity**	**Test Objective**
Off Cycle Fan Power	35	<50	-	-	Compressor Off	Measure fan input wattage during compressor off cycle
Refrigeration Capacity A	35	<50	95	75	Compressor On	Determine Net Refrigeration Capacity of Unit Cooler , input power, and EER at Rating Condition
Refrigeration Capacity B	35	<50	59	54	Compressor On	Determine Net Refrigeration Capacity of Unit Cooler and system input power at moderate condition
Refrigeration Capacity C	35	<50	35	34	Compressor On	Determine Net Refrigeration Capacity of Unit Cooler and system input power at cold condition

Table 5. Two Capacity Matched Refrigerator System, Condensing Unit Located Outdoor						
Test Description	**Unit Cooler Air Entering Dry-Bulb, ˚F**	**Unit Cooler Air Entering Relative Humidity, %**	**Condenser Air Entering Dry-Bulb, ˚F**	**Condenser Air Entering Wet-Bulb, ˚F**	**Compressor Capacity**	**Test Objective**
Off Cycle Fan Power	35	<50	-	-	Compressor Off	Measure fan input wattage during compressor off cycle
Refrigeration Capacity A Low Speed	35	<50	95	75	Minimum Capacity	Determine Net Refrigeration Capacity of Unit Cooler and input power at Rating Condition and minimum compressor capacity
Refrigeration Capacity A High Speed	35	<50	95	75	Maximum Capacity	Determine Net Refrigeration Capacity of Unit Cooler , input power, and EER at Rating Condition and maximum compressor

Table 5. Two Capacity Matched Refrigerator System, Condensing Unit Located Outdoor

Test Description	**Unit Cooler Air Entering Dry-Bulb, ˚F**	**Unit Cooler Air Entering Relative Humidity, %**	**Condenser Air Entering Dry-Bulb, ˚F**	**Condenser Air Entering Wet-Bulb, ˚F**	**Compressor Capacity**	**Test Objective**
						capacity
Refrigeration Capacity B Low Speed	35	<50	59	54	Minimum Capacity	Determine Net Refrigeration Capacity of Unit Cooler and system input power at moderate condition and minimum compressor capacity
Refrigeration Capacity B High Speed	35	<50	59	54	Maximum Capacity	Determine Net Refrigeration Capacity of Unit Cooler and system input power at moderate condition and maximum compressor capacity
Refrigeration Capacity C Low Speed	35	<50	35	34	Minimum Capacity	Determine Net Refrigeration Capacity of Unit Cooler and system input power at cold condition and minimum compressor capacity
Refrigeration Capacity C High Speed	35	<50	35	34	Maximum Capacity	Determine Net Refrigeration Capacity of Unit Cooler and system input power at cold condition and maximum compressor capacity

Table 6. Variable Capacity Matched Refrigerator System, Condensing Unit Located Outdoor

Test Description	**Unit Cooler Air Entering Dry-Bulb, ˚F**	**Unit Cooler Air Entering Relative Humidity, %**	**Condenser Air Entering Dry-Bulb, ˚F**	**Condenser Air Entering Wet-Bulb, ˚F**	**Compressor Capacity**	**Test Objective**
Off Cycle Fan Power	35	<50	-	-	Compressor Off	Measure fan input wattage during compressor off cycle
Refrigeration Capacity A Low Speed	35	<50	95	75	Minimum Capacity	Determine Net Refrigeration Capacity of Unit Cooler and

Table 6. Variable Capacity Matched Refrigerator System, Condensing Unit Located Outdoor						
Test Description	**Unit Cooler Air Entering Dry-Bulb, ˚F**	**Unit Cooler Air Entering Relative Humidity, %**	**Condenser Air Entering Dry-Bulb, ˚F**	**Condenser Air Entering Wet-Bulb, ˚F**	**Compressor Capacity**	**Test Objective**
						system input power at Rating Condition and minimum compressor capacity
Refrigeration Capacity A Variable Speed	35	<50	95	75	Intermediate Capacity*	Determine Net Refrigeration Capacity of Unit Cooler and system input power at Rating Condition and intermediate compressor capacity
Refrigeration Capacity A High Speed	35	<50	95	75	Maximum Capacity	Determine Net Refrigeration Capacity of Unit Cooler , system input power, and EER at Rating Condition and maximum compressor capacity
Refrigeration Capacity B Low Speed	35	<50	59	54	Minimum Capacity	Determine Net Refrigeration Capacity of Unit Cooler and system input power at moderate condition and minimum compressor capacity
Refrigeration Capacity B Variable Speed	35	<50	59	54	Intermediate Capacity*	Determine Net Refrigeration Capacity of Unit Cooler and system input power at moderate condition and intermediate compressor capacity
Refrigeration Capacity B High Speed	35	<50	59	54	Maximum Capacity	Determine Net Refrigeration Capacity of Unit Cooler and system input power at moderate condition and maximum compressor capacity
Refrigeration Capacity C Low Speed	35	<50	35	34	Minimum Capacity	Determine Net Refrigeration Capacity of Unit Cooler and system input power at cold condition and minimum compressor capacity
Refrigeration Capacity C Variable	35	<50	35	34	Intermediate Capacity*	Determine Net Refrigeration Capacity of Unit Cooler and

Table 6. Variable Capacity Matched Refrigerator System, Condensing Unit Located Outdoor

Test Description	Unit Cooler Air Entering Dry-Bulb, ˚F	Unit Cooler Air Entering Relative Humidity, %	Condenser Air Entering Dry-Bulb, ˚F	Condenser Air Entering Wet-Bulb, ˚F	Compressor Capacity	Test Objective
Speed						system input power at cold condition and intermediate compressor capacity
Refrigeration Capacity C High Speed	35	<50	35	34	Maximum Capacity	Determine Net Refrigeration Capacity of Unit Cooler and system input power at cold condition and maximum compressor capacity

*For the intermediate capacity test, the compressor capacity shall be set to 40% of its maximum capacity if possible. Otherwise, it shall be set to the capacity that is the closest to the 40% of its maximum capacity.

Table 7. Fixed Capacity Matched Freezer System, Condensing Unit Located Indoor

Test Description	Unit Cooler Air Entering Dry-Bulb, ˚F	Unit Cooler Air Entering Relative Humidity, %	Condenser Air Entering Dry-Bulb, ˚F	Condenser Air Entering Wet-Bulb, ˚F	Compressor Capacity	Test Objective
Off Cycle Fan Power	-10	<50	-	-	Compressor Off	Measure fan input wattage during compressor off cycle
Refrigeration Capacity	-10	<50	90	75	Compressor On	Determine Net Refrigeration Capacity of Unit Cooler, system input power, and EER at Rating Condition
Defrost Frost Load	-10	various	90	75	System Dependent	Test according to Appendix C Section C11.

Table 8. Fixed Capacity Matched Freezer System, Condensing Unit Located Outdoor

Test Description	Unit Cooler Air Entering Dry-Bulb, ˚F	Unit Cooler Air Entering Relative Humidity, %	Condenser Air Entering Dry-Bulb , ˚F	Condenser Air Entering Wet-Bulb, ˚F	Compressor Capacity	Test Objective
Off Cycle Fan Power	-10	<50	-	-	Compressor Off	Measure fan input wattage during compressor off cycle

Table 8. Fixed Capacity Matched Freezer System, Condensing Unit Located Outdoor

Test Description	**Unit Cooler Air Entering Dry-Bulb, ˚F**	**Unit Cooler Air Entering Relative Humidity, %**	**Condenser Air Entering Dry-Bulb , ˚F**	**Condenser Air Entering Wet-Bulb, ˚F**	**Compressor Capacity**	**Test Objective**
Refrigeration Capacity A	-10	<50	95	75	Compressor On	Determine Net Refrigeration Capacity of Unit Cooler, system input power, and EER at Rating Condition
Refrigeration Capacity B	-10	<50	59	54	Compressor On	Determine Net Refrigeration Capacity of Unit Cooler and system input power at moderate condition
Refrigeration Capacity C	-10	<50	35	34	Compressor On	Determine Net Refrigeration Capacity of Unit Cooler and system input power at cold condition
Defrost Frost Load	-10	various	95	75	System Dependent	Test according to Appendix C Section C11.

Table 9. Two Capacity Matched Freezer System, Condensing Unit Located Outdoor

Test Description	**Unit Cooler Air Entering Dry-Bulb, ˚F**	**Unit Cooler Air Entering Relative Humidity, %**	**Condenser Air Entering Dry-Bulb, ˚F**	**Condenser Air Entering Wet-Bulb, ˚F**	**Compressor Capacity**	**Test Objective**
Off Cycle Fan Power	-10	<50	-	-	Compressor Off	Measure fan input wattage during compressor off cycle
Refrigeration Capacity A Low Speed	-10	<50	95	75	Minimum Capacity	Determine Net Refrigeration Capacity of Unit Cooler and input power at Rating Condition and minimum compressor capacity

Table 9. Two Capacity Matched Freezer System, Condensing Unit Located Outdoor

Test Description	Unit Cooler Air Entering Dry-Bulb, ˚F	Unit Cooler Air Entering Relative Humidity, %	Condenser Air Entering Dry-Bulb, ˚F	Condenser Air Entering Wet-Bulb, ˚F	Compressor Capacity	Test Objective
Refrigeration Capacity A High Speed	-10	<50	95	75	Maximum Capacity	Determine Net Refrigeration Capacity of Unit Cooler , input power, and EER at Rating Condition and maximum compressor capacity
Refrigeration Capacity B Low Speed	-10	<50	59	54	Minimum Capacity	Determine Net Refrigeration Capacity of Unit Cooler and system input power at moderate condition and minimum compressor capacity
Refrigeration Capacity B High Speed	-10	<50	59	54	Maximum Capacity	Determine Net Refrigeration Capacity of Unit Cooler and system input power at moderate condition and maximum compressor capacity
Refrigeration Capacity C Low Speed	-10	<50	35	34	Minimum Capacity	Determine Net Refrigeration Capacity of Unit Cooler and system input power at cold condition and minimum compressor capacity
Refrigeration Capacity C High Speed	-10	<50	35	34	Maximum Capacity	Determine Net Refrigeration Capacity of Unit Cooler and system input power at cold condition and maximum compressor capacity

Table 9. Two Capacity Matched Freezer System, Condensing Unit Located Outdoor						
Test Description	**Unit Cooler Air Entering Dry-Bulb, ˚F**	**Unit Cooler Air Entering Relative Humidity, %**	**Condenser Air Entering Dry-Bulb, ˚F**	**Condenser Air Entering Wet-Bulb, ˚F**	**Compressor Capacity**	**Test Objective**
Defrost Frost Load	-10	various	95	75	System Dependent	Test according to Appendix C Section C11.

Table 10. Variable Capacity Matched Freezer System, Condensing Unit Located Outdoor						
Test Description	**Unit Cooler Air Entering Dry-Bulb , ˚F**	**Unit Cooler Air Entering Relative Humidity, %**	**Condenser Air Entering Dry-Bulb, ˚F**	**Condenser Air Entering Wet-Bulb, ˚F**	**Compressor Capacity**	**Test Objective**
Off Cycle Fan Power	-10	<50	-	-	Compressor Off	Measure fan input wattage during compressor off cycle
Refrigeration Capacity A Low Speed	-10	<50	95	75	Minimum Capacity	Determine Net Refrigeration Capacity of Unit Cooler and input power at Rating Condition and minimum compressor capacity
Refrigeration Capacity A Variable Speed	-10	<50	95	75	Intermediate Capacity*	Determine Net Refrigeration Capacity of Unit Cooler and input power at Rating Condition and intermediate compressor capacity
Refrigeration Capacity A High Speed	-10	<50	95	75	Maximum Capacity	Determine Net Refrigeration Capacity of Unit Cooler , input power, and EER at Rating Condition and maximum compressor capacity

Table 10. Variable Capacity Matched Freezer System, Condensing Unit Located Outdoor

Test Description	Unit Cooler Air Entering Dry-Bulb , ˚F	Unit Cooler Air Entering Relative Humidity, %	Condenser Air Entering Dry-Bulb, ˚F	Condenser Air Entering Wet-Bulb, ˚F	Compressor Capacity	Test Objective
Refrigeration Capacity B Low Speed	-10	<50	59	54	Minimum Capacity	Determine Net Refrigeration Capacity of Unit Cooler and system input power at moderate condition and minimum compressor capacity
Refrigeration Capacity B Variable Speed	-10	<50	59	54	Intermediate Capacity*	Determine Net Refrigeration Capacity of Unit Cooler and system input power at moderate condition and intermediate compressor capacity
Refrigeration Capacity B High Speed	-10	<50	59	54	Maximum Capacity	Determine Net Refrigeration Capacity of Unit Cooler and system input power at moderate condition and maximum compressor capacity
Refrigeration Capacity C Low Speed	-10	<50	35	34	Minimum Capacity	Determine Net Refrigeration Capacity of Unit Cooler and system input power at cold condition and minimum compressor capacity

Table 10. Variable Capacity Matched Freezer System, Condensing Unit Located Outdoor

Test Description	Unit Cooler Air Entering Dry-Bulb , ˚F	Unit Cooler Air Entering Relative Humidity, %	Condenser Air Entering Dry-Bulb, ˚F	Condenser Air Entering Wet-Bulb, ˚F	Compressor Capacity	Test Objective
Refrigeration Capacity C Variable Speed	-10	<50	35	34	Intermediate Capacity*	Determine Net Refrigeration Capacity of Unit Cooler and system input power at cold condition and intermediate compressor capacity
Refrigeration Capacity C High Speed	-10	<50	35	34	Maximum Capacity	Determine Net Refrigeration Capacity of Unit Cooler and system input power at cold condition and maximum compressor capacity
Defrost Frost Load	-10	various	95	75	System Dependent	Test according to Appendix C Section C11.

*For the intermediate capacity test, the compressor capacity shall be set to 50% of its maximum capacity if possible. Otherwise, it shall be set to the capacity that is the closest to the 50% of its maximum capacity

Table 11. Fixed Capacity Refrigerator Condensing Unit, Condensing Unit Located Indoor*

Test Description	Suction Dewpoint, ˚F	Return Gas, ˚F	Condenser Air Entering Dry-Bulb, ˚F	Condenser Air Entering Wet-Bulb, ˚F	Compressor Capacity	Test Objective
Refrigeration Capacity Compressor On	23	41**	90	75***	Compressor On	Determine gross refrigeration capacity and input power of condensing unit at Rating Condition

*Subcooling to be set according to equipment specification and reported as part of standard rating
** Measured at the condensing unit inlet location.
*** Required only for evaporative condensing units

Table 12. Fixed Capacity Refrigerator Condensing Unit, Condensing Unit Located Outdoor*

Test Description	Suction Dewpoint, ˚F	Return Gas, ˚F	Condenser Air Entering Dry-Bulb, ˚F	Condenser Air Entering Wet-Bulb, ˚F	Compressor Capacity	Test Objective
Refrigeration Capacity, Ambient Condition A	23	41**	95	75***	Compressor On	Determine gross refrigeration capacity and input power of condensing unit at Rating Condition
Refrigeration Capacity Ambient Condition B	23	41**	59	54***	Compressor On	Determine gross refrigeration capacity and input power of condensing unit at moderate condition
Refrigeration Capacity Ambient Condition C	23	41**	35	34***	Compressor On	Determine gross refrigeration capacity and input power of condensing unit at cold condition

*Subcooling to be set according to equipment specification and reported as part of standard rating
** Measured at the condensing unit inlet location.
*** Required only for evaporative condensing units

Table 13. Fixed Capacity Freezer Condensing Unit, Condensing Unit Located Indoor*

Test Description	Suction Dewpoint, ˚F	Return Gas, ˚F	Condenser Air Entering Dry-Bulb , ˚F	Condenser Air Entering Wet-Bulb, ˚F	Compressor Capacity	Test Objective
Refrigeration Capacity, Compressor On	-22	5**	90	75***	Compressor On	Determine gross refrigeration capacity and input power of condensing unit at Rating Condition

*Subcooling to be set according to equipment specification and reported as part of standard rating
** Measured at the condensing unit inlet location.
*** Required only for evaporative condensing units

Table 14. Fixed Capacity Freezer Condensing Unit, Condensing Unit Located Outdoor*

Test Description	Suction Dewpoint, ˚F	Suction Gas, ˚F	Condenser Air Entering Dry-Bulb, ˚F	Condenser Air Entering Wet-Bulb, ˚F	Compressor Capacity	Test Objective
Refrigeration Capacity Ambient Condition A	-22	5**	95	75***	Compressor On	Determine gross refrigeration capacity and input power of condensing unit at Rating Condition
Refrigeration Capacity, Ambient Condition B	-22	5**	59	54***	Compressor On	Determine gross refrigeration capacity and input power of condensing unit at moderate condition
Refrigeration Capacity, Ambient Condition C	-22	5**	35	34***	Compressor On	Determine gross refrigeration capacity and input power of condensing unit at cold condition

*Subcooling to be set according to equipment specification and reported as part of standard rating
** Measured at the condensing unit inlet location.
*** Required only for evaporative condensing units

Table 15. Refrigerator Unit Cooler

Test Description	Unit Cooler Air Entering Dry-Bulb, ˚F	Unit Cooler Air Entering Relative Humidity, %	Saturated Suction Temp, ˚F	Liquid Inlet Saturation Temperature	Liquid Inlet Subcooling, ˚F	Compressor Capacity	Test Objective
Off Cycle Fan Power	35	<50	-		-	Compressor Off	Measure fan input power during compressor off cycle
Refrigeration Capacity	35	<50	25	105	9	Compressor On	Determine Net Refrigeration Capacity of Unit Cooler

Note: Superheat to be set according to equipment specification in equipment or installation manual, if no superheat specification is given a default superheat value of 6.5°F shall be used. The superheat setting used in the test shall be reported as part of standard rating

Table 16. Freezer Unit Cooler

Test Description	Unit Cooler Air Entering Dry-Bulb, ˚F	Unit Cooler Air Entering Relative Humidity, %	Saturated Suction Temp, ˚F	Liquid Inlet Saturation Temperature	Liquid Inlet Subcooling, ˚F	Compressor Capacity	Test Objective
Off Cycle Fan Power	-10	<50	-		-	Compressor Off	Measure fan input wattage during compressor off cycle
Refrigeration Capacity	-10	<50	-20	105	9	Compressor On	Determine Net Refrigeration Capacity of Unit Cooler
Defrost	-10	various	-			Compressor Off	Test according to Appendix C Section C11.
	Note: Superheat to be set according to equipment specification in equipment or installation manual, if no superheat specification is given a default superheat value of 6.5°F shall be used. The superheat setting used in the test shall be and reported as part of standard rating						

5.2 *Application Ratings*. Application Ratings shall consist of a Capacity Rating plus the associated power input and Energy Efficiency Ratio (EER). When tested with a specified motor, the associated compressor speed (external drive compressors only), shall also be included as part of the rating. The power required to operate all included accessories such as condenser fans, water pumps, controls, and similar accessories shall be accounted for in the power input and Energy Efficiency Ratio.

When external accessories such as water pumps, remote fans, and similar accessories are required for the operation of the unit but are not included with the unit, the manufacturer shall clearly state that the rated power input and Energy Efficiency Ratio do not account for additional power required by these external accessories.

Application Ratings shall be reported at rated voltage, phase, and frequency.

5.3 *Tolerances*. To comply with this standard, measured test results shall not be less than 95% of Published Ratings for capacity and energy efficiency. Power input shall be no more than 105% of the rated values.

5.4 *Electric Conditions.* Standard Rating tests shall be performed at the nameplate rated voltage(s) and frequency. For air-cooled equipment which is rated with 208-230 V dual nameplate voltages, Standard Rating tests shall be performed at 230 V. For all other dual nameplate voltage equipment covered by this standard, the Standard Rating tests shall be performed at both voltages or at the lower of the two voltages if only a single Standard Rating is to be published.

Section 6. Calculation for Walk-in Box Load

6.1 *General Description.* The walk-in box load is comprised of a high load period ($B\dot{L}H$) of the day corresponding to frequent door openings, product loading events, and other design load factors, and a low load period of the day ($B\dot{L}L$) corresponding to the minimum load resulting from conduction, internal heat gains from equipment that is not related to the refrigeration system, and infiltration when the door is closed. Both the $B\dot{L}H$ and $B\dot{L}L$ are defined as a linear relationship with outdoor ambient temperature. This relationship accounts for the influence of outdoor ambient on the conduction and infiltration loads for a "typical" walk-in box. At the refrigeration system design point of 95 ˚F for outdoor condensing unit/coil or 90 ˚F for indoor condensing unit, the ratio of $B\dot{L}H$ to steady state refrigeration system capacity is 0.70 for refrigerators and 0.80 for freezers; the ratio of $B\dot{L}L$ to steady state refrigeration system capacity is 0.10 for refrigerators and 0.40 for freezers. The high load period for $B\dot{L}H$ is 8 hours per day, and the low load period for $B\dot{L}L$ is 16 hours per day.

6.2 *Refrigerator Load Equations*

6.2.1 *Indoor Condensing Unit.* The walk-in box and the condensing unit are both located within a conditioned space. The walk-in box load during a high load period is calculated by

$$B\dot{L}H = 0.7 \cdot \dot{q}_{ss}(90^{\circ}F)$$

In which, the box load equals to 70% of the refrigeration system steady state net capacity at the design point of 90˚F. The Net Refrigeration Capacity is to be measured directly from the test by following the procedure defined in the Section 4 of this standard.

The box load during a low load period equals to 10% of the refrigeration system steady state net capacity at the design point of 90˚F, and can be calculated by

$$B\dot{L}L = 0.1 \cdot \dot{q}_{ss}(90^{\circ}F)$$

6.2.2 *Outdoor Condensing Unit.* The walk-in box load at different bin temperatures (t_j) during high and low load periods are calculated by

$$B\dot{L}H(t_j) = 0.65 \cdot \dot{q}_{ss}(95^{\circ}F) + 0.05 \cdot \frac{\dot{q}_{ss}(95^{\circ}F) \cdot (t_j - 35)}{60}$$

$$B\dot{L}L(t_j) = 0.03 \cdot \dot{q}_{ss}(95^{\circ}F) + 0.07 \cdot \frac{\dot{q}_{ss}(95^{\circ}F) \cdot (t_j - 35)}{60}$$

6.3 *Freezer Load Equations*

6.3.1 *Indoor Condensing Unit.* The walk-in box and the condensing unit are both located within a conditioned space. The walk-in box load during a high load period is calculated as.

$$B\dot{L}H = 0.8 \cdot \dot{q}_{ss}(90^{\circ}F)$$

In which, the box load equals to 80% of the refrigeration system steady state net capacity at the design point of 90˚F. The net refrigeration capacity is to be measured directly from the test by following the procedure defined in the Section 4 of this standard.

The box load during a low load period equals to 40% of the refrigeration system steady state net capacity at the design point of 90˚F, and can be calculated by,

$$B\dot{L}L = 0.4 \cdot \dot{q}_{ss}(90^{\circ}F)$$

6.3.2 *Outdoor Condensing Unit.* The walk-in box load during high and low load periods are calculated by:

$$B\dot{L}H\left(t_j\right) = 0.55 \cdot \dot{q}_{ss}(95^{\circ}F) + 0.25 \cdot \frac{\dot{q}_{ss}(95^{\circ}F) \cdot \left(t_j + 10\right)}{105}$$

$$B\dot{L}L\left(t_j\right) = 0.15 \cdot \dot{q}_{ss}(95^{\circ}F) + 0.25 \cdot \frac{\dot{q}_{ss}(95^{\circ}F) \cdot \left(t_j + 10\right)}{105}$$

Section 7. Calculation for Annual Walk-in Energy Factor

7.1 *General Description.* The calculation procedure described in this section is based on the data performance obtained from the tests under the standard rating conditions defined in Section 5 for single-capacity, two-capacity and variable capacity systems. The calculation method depends on outlining system capacity and power profiles over different temperature bins using laboratory test results. The annual walk-in energy factor, AWEF, is calculated by weighting system performance at individual bins with bin hours (number of hours for a given temperature occurs over the year), that is defined in Appendix D.

7.2 The total walk-in system heat load include the walk-in box load ($B\dot{L}H$ and $B\dot{L}L$), defined in Section 6, and the heat load contributed by the operation of the refrigeration system (i.e. evaporator fan power and defrost). The total walk-in system heat load is also comprised of a high load period ($W\dot{L}H$) and a low load period ($W\dot{L}L$), corresponding to the walk-in box loads $B\dot{L}H$ and $B\dot{L}L$. The refrigeration system operates 1/3 of operating hours under high load period, and 2/3 of operating time during low load period as defined in Section 6.1.

7.3 Load factor is defined as the ratio of the total walk-in system heat load to the system net refrigeration capacity. The load factors during high and low load periods at each bin temperature can be calculated by

$$LFH\left(t_j\right) = \frac{W\dot{L}H\left(t_j\right)}{\dot{q}_{ss}\left(t_j\right)}$$

$$LFL\left(t_j\right) = \frac{W\dot{L}L\left(t_j\right)}{\dot{q}_{ss}\left(t_j\right)}$$

7.4 *Walk-in Unit with Single Capacity Compressor.*

7.4.1 The operation of units with single capacity compressors is illustrated in Figure 7-1. The total walk-in system heat loads at each bin temperature during high and low load periods for the walk-in unit with single capacity compressor are calculated by the following equations. The terms of defrost power contributing to the box load, $\dot{Q}_{DF}$, and to the system power consumption, $D\dot{F}$, in these equations shall only be applied to the walk-in freezer systems, and shall be set to zero during the calculation for the walk-in refrigerator systems.

$$W\dot{L}H\left(t_j\right) = B\dot{L}H\left(t_j\right) + 3.412 \cdot E\dot{F}_{comp,off}\left(1 - LFH\left(t_j\right)\right) + \dot{Q}_{DF}$$

$$W\dot{L}L\left(t_j\right) = B\dot{L}L\left(t_j\right) + 3.412 \cdot E\dot{F}_{comp,off}\left(1 - LFL\left(t_j\right)\right) + \dot{Q}_{DF}$$

Where

$$LFH(t_j)=\frac{W\dot{L}H(t_j)}{\dot{q}_{ss}(t_j)}=\frac{B\dot{L}H(t_j)+3.412\cdot \dot{EF}_{comp,off}+\dot{Q}_{DF}}{\dot{q}_{ss}(t_j)+3.412\cdot \dot{EF}_{comp,off}}$$

$$LFL(t_j)=\frac{W\dot{L}L(t_j)}{\dot{q}_{ss}(t_j)}=\frac{B\dot{L}L(t_j)+3.412\cdot \dot{EF}_{comp,off}+\dot{Q}_{DF}}{\dot{q}_{ss}(t_j)+3.412\cdot \dot{EF}_{comp,off}}$$

The annual walk-in energy factor, AWEF, is determined by

$$AWEF=\sum_{j=1}^{n} BL(t_j)\Big/\sum_{j=1}^{n} E(t_j)$$

The term $BL(t_j)$ and $E(t_j)$, summed over temperature bins, are evaluated at each temperature bin, and calculated by:

$$BL(t_j)=\left[0.33\cdot B\dot{L}H(t_j)+0.67\cdot B\dot{L}L(t_j)\right]\cdot n_j$$

$$E(t_j)=\left\{\begin{matrix} 0.33\cdot\left[\dot{E}_{ss}(t_j)\cdot LFH(t_j)+\dot{EF}_{comp,off}\left(1-LFH(t_j)\right)\right]+0.67\cdot \\ \left[\dot{E}_{ss}(t_j)\cdot LFL(t_j)+\dot{EF}_{comp,off}\left(1-LFL(t_j)\right)\right]+\dot{DF} \end{matrix}\right\}\cdot n_j$$

In the calculation above, the refrigeration system operates 1/3 of operating hours under high load period, and 2/3 of operating time during low load period as defined in Section 7.2.

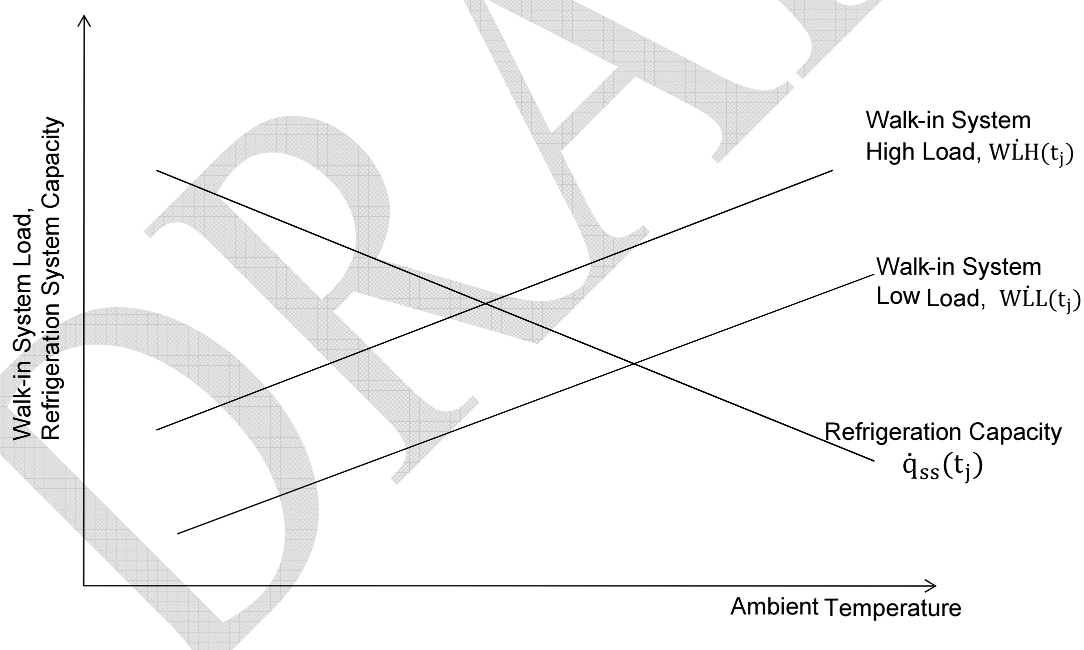

Figure 7-1: Schematic of the operation for units with single capacity compressor

7.4.2 The system steady state net refrigeration capacity and power consumption at a specific temperature bin shall use the measured values directly from the steady state tests if the bin temperature coincides with the designated rating conditions.

When a bin temperature does not coincide with the designated rating condition, use the follow equations to calculate the net capacity and the power consumption.

If $t_j \le 59°F$

$$\dot{q}_{ss}(t_j) = \dot{q}_{ss}(35°F) + \frac{\left(\dot{q}_{ss}(59°F) - \dot{q}_{ss}(35°F)\right)}{59-35}(t_j - 35)$$

$$\dot{E}_{ss}(t_j) = \dot{E}_{ss}(35°F) + \frac{\left(\dot{E}_{ss}(59°F) - \dot{E}_{ss}(35°F)\right)}{59-35}(t_j - 35)$$

If $t_j > 59°F$

$$\dot{q}_{ss}(t_j) = \dot{q}_{ss}(59°F) + \frac{\left(\dot{q}_{ss}(95°F) - \dot{q}_{ss}(59°F)\right)}{95-59}(t_j - 59)$$

$$\dot{E}_{ss}(t_j) = \dot{E}_{ss}(59°F) + \frac{\left(\dot{E}_{ss}(95°F) - \dot{E}_{ss}(59°F)\right)}{95-59}(t_j - 59)$$

7.5 *Walk-in Unit with Two-Capacity Compressor*

7.5.1 Two-capacity compressor means a walk-in unit that has one of the following:

(1) A two-speed compressor,
(2) Two compressors where only one compressor ever operates at a time,
(3) Two compressors where one compressor (Compressor #1) operates at low loads and both compressors (Compressors #1 and #2) operate at high loads but Compressor #2 never operates alone, or
(4) A compressor that is capable of cylinder or scroll unloading.

For such systems, low capacity means:

(1) Operating at low compressor speed,
(2) Operating the lower capacity compressor,
(3) Operating Compressor #1, or
(4) Operating with the compressor unloaded (*e.g.*, operating one piston of a two-piston reciprocating compressor, using a fixed fractional volume of the full scroll, etc.).

For such systems, high capacity means:

(1) Operating at high compressor speed,
(2) Operating the higher capacity compressor,
(3) Operating Compressors #1 and #2, or
(4) Operating with the compressor loaded (*e.g.*, operating both pistons of a two-piston reciprocating compressor, using the full volume of the scroll).

The unit shall be tested at the designated test conditions for both high and low capacities to evaluate the steady state capacities and power consumptions.

7.5.2 For two-capacity compressor units, the annual walk-in energy factor, AWEF, is calculated by

$$AWEF = \sum_{j=1}^{n} BL(t_j) \Big/ \sum_{j=1}^{n} E(t_j)$$

The term $BL(t_j)$ and $E(t_j)$, summed over temperature bins, are evaluated at each temperature bin according to four possible cases shown in Figure 7-2 and described as follows. These four cases can be identified in terms of three outdoor temperatures, t_{IH}, t_{IL} and t_{IIH}, which are also shown in Figure 7-2. The outdoor temperature t_{IH} is the temperature at which the total walk-in system heat load equals system net capacity when the compressor operates at low capacity (k=1) during the high load period. The outdoor temperature t_{IL} is the temperature at which the total walk-in system heat load equals system net capacity when the compressor operates at low capacity (k=1) during the low load period. The outdoor temperature t_{IIH} is the temperature at which the total walk-in system heat load equals system net capacity when the compressor operates at high capacity (k=2) during the high load period.

The system steady state net refrigeration capacity and power consumption at a specific temperature bin shall use the measured values directly from the steady state tests if the bin temperature coincides with the designated rating conditions, otherwise use the following equations to calculate the net capacities and the power consumptions for low capacity operation. For high capacity operation, use the same equations, but replace the superscript k=1 by k=2.

If $t_j \leq 59^{\circ}F$

$$\dot{q}_{ss}^{k=1}(t_j) = \dot{q}_{ss}^{k=1}(35^{\circ}F) + \frac{\left(\dot{q}_{ss}^{k=1}(59^{\circ}F) - \dot{q}_{ss}^{k=1}(35^{\circ}F)\right)}{59-35}(t_j - 35)$$

$$\dot{E}_{ss}^{k=1}(t_j) = \dot{E}_{ss}^{k=1}(35^{\circ}F) + \frac{\left(\dot{E}_{ss}^{k=1}(59^{\circ}F) - \dot{E}_{ss}^{k=1}(35^{\circ}F)\right)}{59-35}(t_j - 35)$$

If $t_j > 59^{\circ}F$

$$\dot{q}_{ss}^{k=1}(t_j) = \dot{q}_{ss}^{k=1}(59^{\circ}F) + \frac{\left(\dot{q}_{ss}^{k=1}(95^{\circ}F) - \dot{q}_{ss}^{k=1}(59^{\circ}F)\right)}{95-59}(t_j - 59)$$

$$\dot{E}_{ss}^{k=1}(t_j) = \dot{E}_{ss}^{k=1}(59^{\circ}F) + \frac{\left(\dot{E}_{ss}^{k=1}(95^{\circ}F) - \dot{E}_{ss}^{k=1}(59^{\circ}F)\right)}{95-59}(t_j - 59)$$

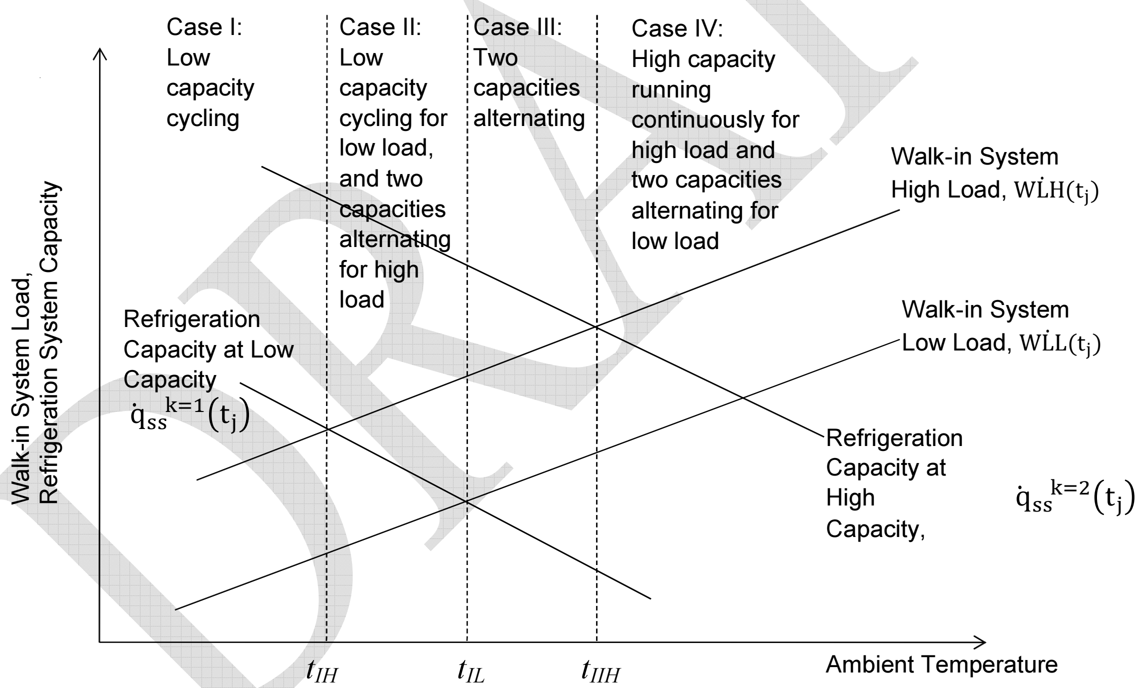

Figure 7-2: Schematic of the various modes of operation for units with two capacity compressors

7.5.2.1 *Case I. Low capacity cycling during both low and high load periods ($t_j < t_{IH}$).* Units operate only at low compressor capacity, and cycle on and off to meet the total walk-in system heat load during both low and high load periods. In this case, units operate identically to single capacity units. The calculation of terms $BL(t_j)$ and $E(t_j)$ shall follow the single capacity compressor procedure described in Section 7.4.

7.5.2.2 *Case II. Low capacity cycling during low load period and two capacities alternating during high load period ($t_{IH} < t_j < t_{IL}$).* During a low load period, units operate at low compressor capacity, and cycle on and off to meet the total walk-in system load. During a high load period, units alternate between high (k=2) and low (k=1) compressor capacities to satisfy the total walk-in system heat load at temperature t_j. In such a case, the compressor operates continuously during high load period. The terms of defrost power contributing to the box load, $\dot{Q}_{DF}$, and to the system power consumption, $\dot{DF}$, in these equations shall only be applied to the walk-in freezer systems, and shall be set to zero during the calculation for the walk-in refrigerator systems.

$$\dot{WLH}(t_j)=\dot{BLH}(t_j)+\dot{Q}_{DF}$$

$$\dot{WLL}(t_j)=\dot{BLL}(t_j)+3.412\cdot\dot{EF}_{comp,off}\left(1-LFL^{k=1}(t_j)\right)+\dot{Q}_{DF}$$

$$LFH^{k=1}(t_j)=\frac{\dot{q}_{ss}^{k=2}(t_j)-\dot{WLH}(t_j)}{\dot{q}_{ss}^{k=2}(t_j)-\dot{q}_{ss}^{k=1}(t_j)}$$

$$LFH^{k=2}(t_j)=1-LFH^{k=1}(t_j)$$

$$LFL^{k=1}(t_j)=\frac{\dot{WLL}(t_j)}{\dot{q}_{ss}^{k=1}(t_j)}=\frac{\dot{BLL}(t_j)+3.412\cdot\dot{EF}_{comp,off}+\dot{Q}_{DF}}{\dot{q}_{ss}^{k=1}(t_j)+3.412\cdot\dot{EF}_{comp,off}}$$

$$BL(t_j)=\left[0.33\cdot\dot{BLH}(t_j)+0.67\cdot\dot{BLL}(t_j)\right]\cdot n_j$$

$$E(t_j)=\left\{\begin{array}{l}0.33\cdot\left(\dot{E}_{ss}^{k=2}(t_j)\cdot LFH^{k=2}(t_j)+\dot{E}_{ss}^{k=1}\cdot LFH^{k=1}(t_j)\right)+0.67\cdot\\ \left[\dot{E}_{ss}^{k=1}(t_j)\cdot LFL^{k=1}(t_j)+\dot{EF}_{comp,off}(1-LFL^{k=1}(t_j))\right]+\dot{DF}\end{array}\right\}\cdot n_j$$

7.5.2.3 *Case III. Two capacities alternating during both low and high load periods* ($t_{IL}<t_j<t_{IIH}$). Units alternate between high (k=2) and low (k=1) compressor capacities to satisfy the total walk-in system load at temperature t_j. In such a case, the compressor operates continuously. The terms of defrost power contributing to the box load, $\dot{Q}_{DF}$, and to the system power consumption, $\dot{DF}$, in these equations shall only be applied to the walk-in freezer systems, and shall be set to zero during the calculation for the walk-in refrigerator systems.

$$\dot{WLH}(t_j)=\dot{BLH}(t_j)+\dot{Q}_{DF}$$

$$\dot{WLL}(t_j)=\dot{BLL}(t_j)+\dot{Q}_{DF}$$

$$LFH^{k=1}(t_j)=\frac{\dot{q}_{ss}^{k=2}(t_j)-\dot{WLH}(t_j)}{\dot{q}_{ss}^{k=2}(t_j)-\dot{q}_{ss}^{k=1}(t_j)}$$

$$LFH^{k=2}(t_j)=1-LFH^{k=1}(t_j)$$

$$LFL^{k=1}(t_j)=\frac{\dot{q}_{ss}^{k=2}(t_j)-\dot{WLL}(t_j)}{\dot{q}_{ss}^{k=2}(t_j)-\dot{q}_{ss}^{k=1}(t_j)}$$

$$LFL^{k=2}(t_j)=1-LFL^{k=1}(t_j)$$

$$BL(t_j)=\left[0.33\cdot\dot{BLH}(t_j)+0.67\cdot\dot{BLL}(t_j)\right]\cdot n_j$$

$$E(t_j)=[0.33\cdot\left(\dot{E}_{ss}^{k=2}(t_j)\cdot LFH^{k=2}(t_j)+\dot{E}_{ss}^{k=1}(t_j)\cdot LFH^{k=1}(t_j)\right)+0.67\cdot \left(\dot{E}_{ss}^{k=2}(t_j)\cdot LFL^{k=2}(t_j)+\dot{E}_{ss}^{k=1}(t_j)\cdot LFL^{k=1}(t_j)\right)+\dot{D}F]\cdot n_j$$

7.5.2.4 *Case IV. High capacity running continuously during high load period and two capacities alternating during low load period ($t_{IIH}<t_j$).* During a low load period, units alternate between high (k=2) and low (k=1) compressor capacities to satisfy the total walk-in system load at temperature t_j. During a high load period, units operate at high (k=2) compressor capacity continuously. The terms of defrost power contributing to the box load, $\dot{Q}_{DF}$, and to the system power consumption, $\dot{D}F$, in these equations shall only be applied to the walk-in freezer systems, and shall be set to zero during the calculation for the walk-in refrigerator systems.

$$W\dot{L}H(t_j)=B\dot{L}H(t_j)+\dot{Q}_{DF}$$

$$W\dot{L}L(t_j)=B\dot{L}L(t_j)+\dot{Q}_{DF}$$

$$LFH^{k=2}(t_j)=1$$

$$LFL^{k=1}(t_j)=\frac{\dot{q}_{ss}^{k=2}(t_j)-W\dot{L}L(t_j)}{\dot{q}_{ss}^{k=2}(t_j)-\dot{q}_{ss}^{k=1}(t_j)}$$

$$LFL^{k=2}(t_j)=1-LFL^{k=1}(t_j)$$

$$BL(t_j)=\left[0.33\cdot B\dot{L}H(t_j)+0.67\cdot B\dot{L}L(t_j)\right]\cdot n_j$$

$$E(t_j)=\begin{bmatrix}0.33\cdot\dot{E}_{ss}^{k=2}(t_j)\cdot LFH^{k=2}(t_j)+0.67\cdot\\ \left(\dot{E}_{ss}^{k=2}(t_j)\cdot LFL^{k=2}(t_j)+\dot{E}_{ss}^{k=1}(t_j)\cdot LFL^{k=1}(t_j)\right)+\dot{D}F\end{bmatrix}\cdot n_j$$

7.6 *Walk-in Unit with Variable Capacity Compressor*

7.6.1 The annual walk-in energy factor, AWEF, for the walk-in units with variable capacity compressors is determined by:

$$AWEF=\sum_{j=1}^{n}BL(t_j)/\sum_{j=1}^{n}E(t_j)$$

The term $BL(t_j)$ and $E(t_j)$, summed over temperature bins, are evaluated at each temperature bin according to four possible cases shown in Figure 7-3 and described as follows. These four cases can be identified in terms of three outdoor temperatures, t_{IH}, t_{IL} and t_{IIH}, which are also shown in Figure 7-3. The outdoor temperature t_{IH} is the temperature at which the total walk-in system heat load equals system net capacity when the compressor operates at its minimum capacity (k=1) during the high load period. The outdoor temperature t_{IL} is the temperature at which the total walk-in system heat load equals system net capacity when the compressor operates at its minimum capacity (k=1) during the low load period. The outdoor temperature t_{IIH} is the temperature at which the total walk-in system heat load equals system net capacity when the compressor operates at its maximum capacity (k=2) during the high load period.

The system steady state net refrigeration capacity and power consumption at a specific temperature bin shall use the measured values directly from the steady state tests if the bin temperature coincides with the designated rating conditions, otherwise use the following equations to calculate the net capacities and the power consumptions for minimum capacity operation. For intermediate and maximum capacities operation, use the same equations, but replace the superscript k=1 by k=i and k=2, respectively.

If $t_j \leq 59°F$

$$\dot{q}_{ss}^{k=1}(t_j)=\dot{q}_{ss}^{k=1}(35°F)+\frac{\left(\dot{q}_{ss}^{k=1}(59°F)-\dot{q}_{ss}^{k=1}(35°F)\right)}{59-35}(t_j-35)$$

$$\dot{E}_{ss}^{k=1}(t_j)=\dot{E}_{ss}^{k=1}(35°F)+\frac{\left(\dot{E}_{ss}^{k=1}(59°F)-\dot{E}_{ss}^{k=1}(35°F)\right)}{59-35}(t_j-35)$$

If $t_j > 59°F$

$$\dot{q}_{ss}^{k=1}(t_j)=\dot{q}_{ss}^{k=1}(59°F)+\frac{\left(\dot{q}_{ss}^{k=1}(95°F)-\dot{q}_{ss}^{k=1}(59°F)\right)}{95-59}(t_j-59)$$

$$\dot{E}_{ss}^{k=1}(t_j)=\dot{E}_{ss}^{k=1}(59°F)+\frac{\left(\dot{E}_{ss}^{k=1}(95°F)-\dot{E}_{ss}^{k=1}(59°F)\right)}{95-59}(t_j-59)$$

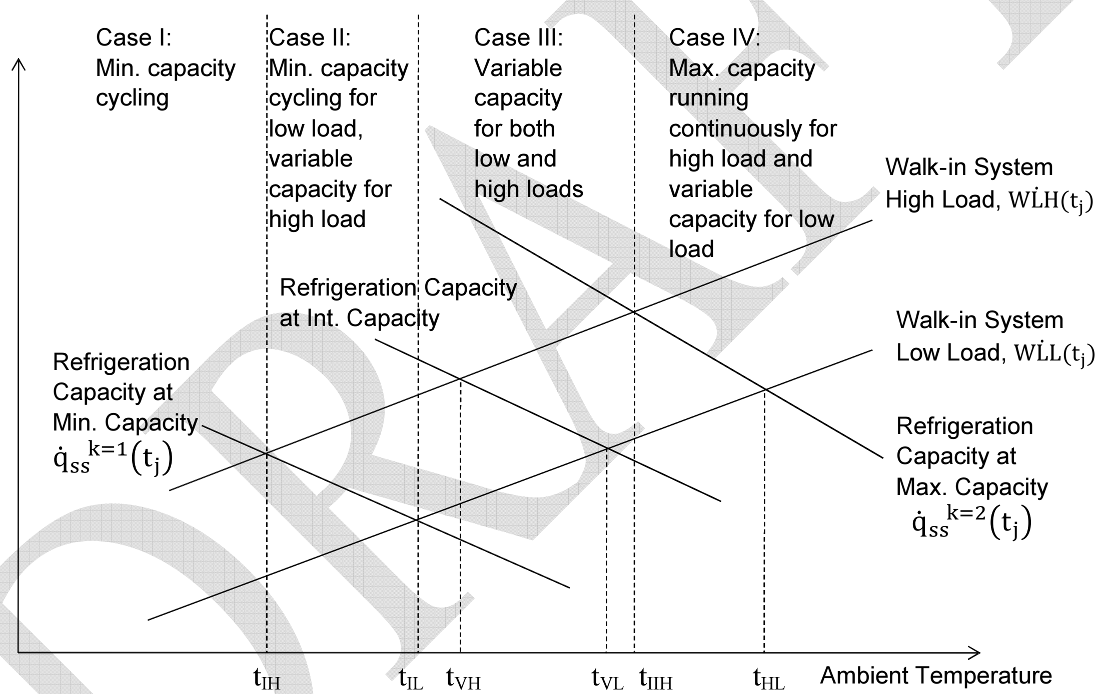

Figure 7-3: Schematic of the various modes of operation for units with variable capacity compressors

7.6.1.1 ***Case I. Minimum capacity cycling during both low and high load periods ($t_j < t_{IH}$).*** Units operate at the minimum capacity, and cycle on and off to meet the total walk-in system load during both low and high load periods. The terms of defrost power contributing to the box load, $\dot{Q}_{DF}$, and to the system power consumption, $\dot{DF}$, in these equations shall only be applied to the walk-in freezer systems, and shall be set to zero during the calculation for the walk-in refrigerator systems.

$$\dot{W}LH(t_j)=B\dot{L}H(t_j)+3.412\cdot\dot{EF}_{comp,off}\left(1-LFH(t_j)\right)+\dot{Q}_{DF}$$

$$\dot{W}LL(t_j)=B\dot{L}L(t_j)+3.412\cdot\dot{EF}_{comp,off}\left(1-LFL(t_j)\right)+\dot{Q}_{DF}$$

$$LFH(t_j)=\frac{\dot{W}LH(t_j)}{\dot{q}_{ss}^{k=1}(t_j)}=\frac{B\dot{L}H(t_j)+3.412\cdot\dot{EF}_{comp,off}+\dot{Q}_{DF}}{\dot{q}_{ss}^{k=1}(t_j)+3.412\cdot\dot{EF}_{comp,off}}$$

$$LFL(t_j)=\frac{\dot{W}LL(t_j)}{\dot{q}_{ss}^{k=1}(t_j)}=\frac{B\dot{L}L(t_j)+3.412\cdot\dot{EF}_{comp,off}+\dot{Q}_{DF}}{\dot{q}_{ss}^{k=1}(t_j)+3.412\cdot\dot{EF}_{comp,off}}$$

$$BL(t_j)=\left[0.33\cdot B\dot{L}H(t_j)+0.67\cdot B\dot{L}L(t_j)\right]\cdot n_j$$

$$E(t_j)=\left\{\begin{matrix}0.33\cdot\left[\dot{E}_{ss}^{k=1}(t_j)\cdot LFH(t_j)+\dot{EF}_{comp,off}\left(1-LFH(t_j)\right)\right]+0.67\cdot\\ \left[\dot{E}_{ss}^{k=1}(t_j)\cdot LFL(t_j)+\dot{EF}_{comp,off}\left(1-LFL(t_j)\right)\right]+\dot{DF}\end{matrix}\right\}\cdot n_j$$

7.6.1.2 *Case II. Minimum capacity cycling during low load period and intermediate capacity operating continuously during high load period ($t_{IH}<t_j<t_{IL}$).* During a low-load period, units operate at minimum capacity, and cycle on and off to meet the total walk-in system load. During a high-load period, units operate at variable capacity (k=v). In such a case, the compressor varies the capacity between its minimum and maximum capacities, and continuously operates to match the total walk-in system load at temperature t_j. The terms of defrost power contributing to the box load, $\dot{Q}_{DF}$, and to the system power consumption, $\dot{DF}$, in these equations shall only be applied to the walk-in freezer systems, and shall be set to zero during the calculation for the walk-in refrigerator systems.

$$\dot{W}LL(t_j)=B\dot{L}L(t_j)+3.412\cdot\dot{EF}_{comp,off}\left(1-LFL^{k=1}(t_j)\right)+\dot{Q}_{DF}$$

$$\dot{W}LH(t_j)=B\dot{L}H(t_j)+\dot{Q}_{DF}$$

$$LFL^{k=1}(t_j)=\frac{B\dot{L}L(t_j)+3.412\cdot\dot{EF}_{comp,off}+\dot{Q}_{DF}}{\dot{q}_{ss}^{k=1}(t_j)+3.412\cdot\dot{EF}_{comp,off}}$$

$$\dot{q}_{SS,H}^{k=v}(t_j)=\dot{W}LH(t_j)$$

$$\dot{E}_{SS,H}^{k=v}(t_j)=\frac{\dot{q}_{SS,H}^{k=v}(t_j)}{EER_{SS,H}^{k=v}(t_j)}$$

$$EER_{SS,H}^{k=v}(t_j)=a+b\cdot t_j+c\cdot t_j^2$$

To determine the coefficients a, b and c, it is required to evaluate the unit EER at three different compressor capacities: the minimum capacity (k=1), the maximum capacity (k=2), and the capacity (k=i) at which the intermediate-capacity test was conducted. The following is a procedure for evaluation of the coefficients a, b and c.

$$a=EER_{SS}^{k=2}(t_{IIH})-b\cdot t_{IIH}-c\cdot t_{IIH}^2$$

$$b=\frac{EER_{SS}^{k=1}(t_{IH})-EER_{SS}^{k=2}(t_{IIH})-d\cdot[EER_{SS}^{k=1}(t_{IH})-EER_{SS}^{k=i}(t_{VH})]}{t_{IH}-t_{IIH}-d\cdot[t_{IH}-t_{IIH}]}$$

$$c=\frac{EER_{SS}^{k=1}(t_{IH})-EER_{SS}^{k=2}(t_{IIH})-b\cdot(t_{IH}-t_{IIH})}{t_{IH}^2-t_{IIH}^2}$$

$$d=\frac{t_{IIH}{}^2-t_{IH}{}^2}{t_{VH}{}^2-t_{IH}{}^2}$$

Where

$$EER_{ss}^{k=1}(t_{IH})=\frac{\dot{q}_{ss}^{k=1}(t_{IH})}{\dot{E}_{ss}^{k=1}(t_{IH})}$$

$$EER_{ss}^{k=2}(t_{IIH})=\frac{\dot{q}_{ss}^{k=2}(t_{IIH})}{\dot{E}_{ss}^{k=2}(t_{IIH})}$$

$$EER_{ss}^{k=i}(t_{VH})=\frac{\dot{q}_{ss}^{k=i}(t_{VH})}{\dot{E}_{ss}^{k=i}(t_{VH})}$$

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The outdoor temperature t_{VH} is the temperature at which the total walk-in system heat load equals system net capacity when the compressor operates at its intermediate capacity (k=i) during the high load period.

$$BL(t_j)=[0.33\cdot B\dot{L}H(t_j)+0.67\cdot B\dot{L}L(t_j)]\cdot n_j$$

$$E(t_j)=\{0.33\cdot\dot{E}_{SS,H}{}^{k=v}(t_j)+0.67\cdot[\dot{E}_{SS}{}^{k=1}(t_j)\cdot LFL^{k=1}(t_j)+E\dot{F}_{comp,off}(1-LFL^{k=1}(t_j))]+D\dot{F}\}\cdot n_j$$

7.6.1.3 ***Case III. Intermediate capacity running continuously during both low and high load periods*** **($t_{IL} < t_j < t_{IIH}$).** Units operate at variable compressor capacities (k=v) during both low and high load periods. The compressor varies the capacity between its minimum and maximum capacities, and continuously operate to match the total walk-in system load at temperature t_j. The terms of defrost power contributing to the box load, $\dot{Q}_{DF}$, and to the system power consumption, $D\dot{F}$, in these equations shall only be applied to the walk-in freezer systems, and shall be set to zero during the calculation for the walk-in refrigerator systems.

$$W\dot{L}H(t_j)=B\dot{L}H(t_j)+\dot{Q}_{DF}$$

$$W\dot{L}L(t_j)=B\dot{L}L(t_j)+\dot{Q}_{DF}$$

$$\dot{q}_{SS,H}{}^{k=v}(t_j)=W\dot{L}H(t_j)$$

$$\dot{q}_{SS,L}{}^{k=v}(t_j)=W\dot{L}L(t_j)$$

$$\dot{E}_{SS,H}{}^{k=v}(t_j)=\frac{\dot{q}_{SS,H}{}^{k=v}(t_j)}{EER_{SS,H}{}^{k=v}(t_j)}$$

$$\dot{E}_{SS,L}{}^{k=v}(t_j)=\frac{\dot{q}_{SS,L}{}^{k=v}(t_j)}{EER_{SS,L}{}^{k=v}(t_j)}$$

$$EER_{SS,L}{}^{k=v}(t_j)=a+b\cdot t_j+c\cdot t_j{}^2$$

Where,

$$a = EER_{SS}^{k=2}(t_{IIL}) - b \cdot t_{IIL} - c \cdot t_{IIL}^2$$

$$b = \frac{EER_{SS}^{k=1}(t_{IL}) - EER_{SS}^{k=2}(t_{IIL}) - d \cdot [EER_{SS}^{k=1}(t_{IL}) - EER_{SS}^{k=i}(t_{VL})]}{t_{IL} - t_{IIL} - d \cdot [t_{IL} - t_{IIL}]}$$

$$c = \frac{EER_{SS}^{k=1}(t_{IL}) - EER_{SS}^{k=2}(t_{IIL}) - b \cdot [(t_{IL}) - (t_{IIL})}{t_{IL}^2 - t_{IIL}^2}$$

$$d = \frac{t_{IIL}^2 - t_{IL}^2}{t_{VL}^2 - t_{IL}^2}$$

In which,

$$EER_{ss}^{k=1}(t_{IL}) = \frac{\dot{q}_{ss}^{k=1}(t_{IL})}{\dot{E}_{ss}^{k=1}(t_{IL})}$$

$$EER_{ss}^{k=2}(t_{IIL}) = \frac{\dot{q}_{ss}^{k=2}(t_{IIL})}{\dot{E}_{ss}^{k=2}(t_{IIL})}$$

$$EER_{ss}^{k=i}(t_{VL}) = \frac{\dot{q}_{ss}^{k=i}(t_{VL})}{\dot{E}_{ss}^{k=i}(t_{VL})}$$

The outdoor temperature t_{VL} is the temperature at which the total walk-in system heat load equals system net capacity when the compressor operates at its intermediate capacity (k=i) during the low load period.

$$BL(t_j) = \left[0.33 \cdot B\dot{L}H(t_j) + 0.67 \cdot B\dot{L}L(t_j)\right] \cdot n_j$$

$$E(t_j) = \left[0.33 \cdot \dot{E}_{SS,H}{}^{k=v}(t_j) + 0.67 \cdot \dot{E}_{SS,L}{}^{k=v}(t_j) + \dot{DF}\right] \cdot n_j$$

7.6.1.4 *Case IV. High capacity running continuously during high load period and intermediate capacity running continuously during low load period ($t_{IIH} < t_j$).* During a low-load period, units operate at variable compressor capacities (k=v). The compressor varies the capacity between its minimum and maximum capacities, and continuously operate to match the total walk-in system load at temperature t_j. During a high-load period, units operate at maximum (k=2) compressor capacity continuously. The terms of defrost power contributing to the box load, $\dot{Q}_{DF}$, and to the system power consumption, $\dot{DF}$, in these equations shall only be applied to the walk-in freezer systems, and shall be set to zero during the calculation for the walk-in refrigerator systems.

$$W\dot{L}H(t_j) = B\dot{L}H(t_j) + \dot{Q}_{DF}$$

$$W\dot{L}L(t_j) = B\dot{L}L(t_j) + \dot{Q}_{DF}$$

$$BL(t_j) = \left[0.33 \cdot B\dot{L}H(t_j) + 0.67 \cdot B\dot{L}L(t_j)\right] \cdot n_j$$

$$E(t_j) = \left[0.33 \cdot \dot{E}_{SS}{}^{k=2}(t_j) + 0.67 \cdot \dot{E}_{SS,L}{}^{k=v}(t_j) + \dot{DF}\right] \cdot n_j$$

7.7 *Walk-in box and condensing unit located in conditioned space.* In such a case, the walk-in system load and the refrigeration system performance are independent to the outdoor ambient conditions. The AWEF is calculated by the following equations. The terms of defrost power contributing to the box load, $\dot{Q}_{DF}$, and to the system power consumption, $\dot{DF}$, in these equations shall only be applied to the walk-in freezer systems, and shall be set to zero during the calculation for the walk-in refrigerator systems.

$$\dot{WLH}=\dot{BLH}+3.412\cdot\dot{EF}_{comp,off}\left(1\text{-}LFH\right)+\dot{Q}_{DF}$$

$$\dot{WLL}=\dot{BLL}+3.412\cdot\dot{EF}_{comp,off}\left(1\text{-}LFL\right)+\dot{Q}_{DF}$$

Where $\dot{BLH}$ and $\dot{BLL}$ for refrigerator and freezer systems are defined in Section 6.2.1 and 6.3.1 of this standard, respectively; and the load factors (LFH and LFL) are calculated as follows.

$$LFH=\frac{\dot{WLH}}{\dot{q}_{ss}(90^{\circ}F)}=\frac{\dot{BLH}+3.412\cdot\dot{EF}_{comp,off}+\dot{Q}_{DF}}{\dot{q}_{ss}(90^{\circ}F)+3.412\cdot\dot{EF}_{comp,off}}$$

$$LFL=\frac{\dot{WLL}}{\dot{q}_{ss}(90^{\circ}F)}=\frac{\dot{BLL}+3.412\cdot\dot{EF}_{comp,off}+\dot{Q}_{DF}}{\dot{q}_{ss}(90^{\circ}F)+3.412\cdot\dot{EF}_{comp,off}}$$

The annual walk-in energy factor, AWEF, is determined by

$$AWEF=\frac{0.33\cdot\dot{BLH}+0.67\cdot\dot{BLL}}{0.33\cdot[\dot{E}_{ss}(90^{\circ}F)\cdot LFH+\dot{EF}_{comp,off}\left(1\text{-}LFH\right)]+0.67\cdot[\dot{E}_{ss}(90^{\circ}F)\cdot LFL+\dot{EF}_{comp,off}\left(1\text{-}LFL\right)]+\dot{DF}}$$

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Figure 7-4: Diagram of the linear fit based rating procedure

7.9 *Walk-in Unit Cooler (applied to all Unit Coolers rated separately)* .

7.9.1 The following table (Table 17) from AHRI Standard 1200 defines the power required by the rack system to handle the walk-in unit cooler load:

Table 17. EER for Remote Commercial Refrigerated Display Merchandisers and Storage Cabinets

Medium Temperature		Low Temperature	
Adjusted Dew Point ºF	EER	Adjusted Dew Point ºF	EER
0.0	9.25	-36.0	5.48
1.0	9.37	-35.0	5.56
2.0	9.50	-34.0	5.64
3.0	9.63	-33.0	5.73
4.0	9.76	-32.0	5.81
5.0	9.87	-31.0	5.9
6.0	10.03	-30.0	5.98
7.0	10.19	-29.0	6.06
8.0	10.36	-28.0	6.15
9.0	10.52	-27.0	6.24
10.0	10.69	-26.0	6.33
11.0	10.87	-25.0	6.41
12.0	11.05	-24.0	6.5
13.0	11.22	-23.0	6.6

Table 17. EER for Remote Commercial Refrigerated Display Merchandisers and Storage Cabinets

Medium Temperature		Low Temperature	
Adjusted Dew Point ºF	EER	Adjusted Dew Point ºF	EER
14.0	11.4	-22.0	6.7
15.0	11.58	-21.0	6.78
16.0	11.79	-20.0	6.88
17.0	11.99	-19.0	6.98
18.0	12.19	-18.0	7.08
19.0	12.39	-17.0	7.19
20.0	12.59	-16.0	7.29
21.0	12.85	-15.0	7.39
22.0	13.04	-14.0	7.49
23.0	13.27	-13.0	7.60
24.0	13.49	-12.0	7.70
25.0	13.72	-11.0	7.81
26.0	13.95	-10.0	7.92
27.0	14.18	-9.0	8.03
28.0	14.47	-8.0	8.14
29.0	14.73	-7.0	8.25
30.0	14.98	-6.0	8.36
31.0	15.27	-5.0	8.48
32.0	15.56	-4.0	8.59
33.0	15.84	-3.0	8.71
34.0	16.13	-2.0	8.83
35.0	16.42	-1.0	8.95

Note:

1. EER values at Medium and Low Temperature Applications are based on a typical reciprocating compressor.
2. Linear interpolation shall be used to calculate EER values for temperatures not shown in Table 17.

The Adjusted Dewpoint Value for a refrigerator application shall be 23 ˚F and for a freezer application it shall be -22 ˚F, unless the unit cooler is rated at a suction dewpoint other than 25 ˚F for a refrigerator or -20 ˚F for a freezer, in which case the Adjusted Dewpoint Value shall be 2 ˚F less than the unit cooler rating suction dewpoint.

7.9.2 *Unit cooler with fixed evaporator fan speed*

7.9.2.1 The net capacity, $\dot{q}_{mix,evap}$ is determined from the test data for the unit cooler at the 25 ˚F suction dewpoint for a refrigerator and the -20 ˚F suction dewpoint for a freezer. The power consumption of the system is calculated by.

$$\dot{E}_{mix,rack} = \frac{\dot{q}_{mix,evap} + 3.412 \cdot \dot{EF}_{comp,on}}{EER(\text{Adjusted Dewpoint Value})} + \dot{EF}_{comp,on}$$

In which, EER can be determined from Table 17 with corresponding adjusted dewpoint value.

7.9.2.2 The walk-in refrigerator system box load for the system during high and low load periods can be calculated by

$$B\dot{L}H = 0.7 \cdot \dot{q}_{mix,evap}$$

$$B\dot{L}L = 0.1 \cdot \dot{q}_{mix,evap}$$

7.9.2.3 The walk-in freezer system box load for the system during high and low load periods can be calculated by

$$\dot{BLH} = 0.8 \cdot \dot{q}_{mix,evap}$$

$$\dot{BLL} = 0.4 \cdot \dot{q}_{mix,evap}$$

7.9.2.4 The AWEF of the system is calculated by the following equations. The terms of defrost power contributing to the box load, $\dot{Q}_{DF}$, and to the system power consumption, $\dot{DF}$, in these equations shall only be applied to the walk-in freezer systems, and shall be set to zero during the calculation for the walk-in refrigerator systems.

$$\dot{WLH} = \dot{BLH} + 3.412 \cdot \dot{EF}_{comp,off}\left(1\text{-}LFH\right) + \dot{Q}_{DF}$$

$$\dot{WLL} = \dot{BLL} + 3.412 \cdot \dot{EF}_{comp,off}\left(1\text{-}LFL\right) + \dot{Q}_{DF}$$

Where $(BLH)^{\cdot}$ and $\dot{BLL}$ for refrigerator and freezer systems are defined in Section 7.9.2.2 and 7.9.2.3 of this standard, respectively; and the load factors (LFH and LFL) are calculated as follows.

$$LFH = \frac{\dot{WLH}}{\dot{q}_{mix,evap}} = \frac{\dot{BLH} + 3.412 \cdot \dot{EF}_{comp,off} + \dot{Q}_{DF}}{\dot{q}_{mix,evap} + 3.412 \cdot \dot{EF}_{comp,off}}$$

$$LFL = \frac{\dot{WLL}}{\dot{q}_{mix,evap}} = \frac{\dot{BLL} + 3.412 \cdot \dot{EF}_{comp,off} + \dot{Q}_{DF}}{\dot{q}_{mix,evap} + 3.412 \cdot \dot{EF}_{comp,off}}$$

The annual walk-in energy factor, AWEF, is determined by

$$AWEF = \frac{0.33 \cdot \dot{BLH} + 0.67 \cdot \dot{BLL}}{0.33 \cdot [\dot{E}_{mix,rack} \cdot LFH + \dot{EF}_{comp,off}\left(1\text{-}LFH\right)] + 0.67 \cdot [\dot{E}_{mix,rack} \cdot LFL + \dot{EF}_{comp,off}\left(1\text{-}LFL\right)] + \dot{DF}}$$

7.9.3 *Unit Cooler with Variable Speed Evaporator Fans*. For unit coolers with variable speed evaporator fans that modulate fan speed in response to load, the fan shall be operated under its minimum, maximum and intermediate speed that equals to the average of the maximum and minimum speeds, respectively during the unit cooler test. These unit coolers are designed for use with variable capacity refrigerant systems.

7.9.3.1 The evaporator net capacities, fan operating speed and the fan power consumptions under the three fan speeds shall be determined from the test data for the unit cooler at the 25 ˚F suction dewpoint for a refrigerator and the -20 ˚F suction dewpoint for a freezer, and correlated by the following equations.

$$s\left(\dot{q}_{mix,evap}\right) = k_7 + k_8 \cdot \dot{q}_{mix,evap} + k_9 \cdot \dot{q}^2_{mix,evap}$$

$$\dot{EF}_{comp,on}(s) = k_{10} + k_{11} \cdot s + k_{12} \cdot s^2$$

Where s stands for fan operating speed, and is a function of the coil net capacity; k_7~k_{12} are coefficients derived from evaporator coil test points.

7.9.3.2 The walk-in refrigerator system box load for the system during high and low load periods can be calculated by

$$\dot{BLH} = 0.7 \cdot \dot{q}_{mix,evap,max}$$

$$\dot{BLL} = 0.1 \cdot \dot{q}_{mix,evap,\,max}$$

Where $\dot{q}_{mix,evap,max}$ is the net coil capacity under the maximum fan speed test point.

7.9.3.3 The walk-in freezer system box load for the system during high and low load periods can be calculated by

$$\dot{BLH} = 0.8 \cdot \dot{q}_{mix,evap,max}$$

$$\dot{BLL} = 0.4 \cdot \dot{q}_{mix,evap,max}$$

Where $\dot{q}_{mix,evap,max}$ is the net coil capacity under the maximum fan speed test point.

7.9.3.4 The total walk-in system load during high and low load periods can be calculated by

$$\dot{WLH} = \dot{BLH} + \dot{Q}_{DF}$$

$$\dot{WLL} = \dot{BLL} + 3.412 \cdot \dot{EF}_{comp,off}\left(1\text{-}LFL\right) + \dot{Q}_{DF}, \text{ when } \dot{WLL} < \dot{q}_{mix,evap,min}$$

$$\dot{WLL} = \dot{BLL} + \dot{Q}_{DF}, \text{ when } \dot{WLL} \geq \dot{q}_{mix,evap,min}$$

Where $\dot{BLH}$ and $\dot{BLL}$ for refrigerator and freezer systems are defined in Section 7.9.3.2 and 7.9.3.3 of this standard, respectively; and the load factor during low load period is calculated by

$$LFL = \frac{\dot{WLL}}{\dot{q}_{mix,evap,min}} = \frac{\dot{BLL} + 3.412 \cdot \dot{EF}_{comp,off} + \dot{Q}_{DF}}{\dot{q}_{mix,evap,min} + 3.412 \cdot \dot{EF}_{comp,off}}$$

Where $\dot{q}_{mix,evap,min}$ is the net coil capacity under the minimum fan speed test point. The terms of defrost power contributing to the box load, $\dot{Q}_{DF}$, in these equations shall only be applied to the walk-in freezer systems, and shall be set to zero during the calculation for the walk-in refrigerator systems.

7.9.3.5 The power consumption of the system during the high and low load periods are calculated by.

$$\dot{E}_{mix,rack,\,H} = \frac{\dot{WLH} + 3.412 \cdot \dot{EF}_{comp,on}(s_H)}{EER(\text{Adjusted Dewpoint Value})} + \dot{EF}_{comp,on}(s_H)$$

Where, the evaporator fan speed during the high load period, s_H, results in a coil capacity which matches $\dot{WLH}$, the combined box and defrost load during the high load period.

$$\dot{E}_{mix,rack,\,L} = \frac{\dot{WLL} + 3.412 \cdot \dot{EF}_{comp,on}(s_L)}{EER(\text{Adjusted Dewpoint Value})} + \dot{EF}_{comp,on}(s_L), \text{ when } \dot{WLL} \geq \dot{q}_{mix,evap,min}$$

Where, the evaporator fan speed during the low load period, s_L, results in a coil capacity at that speed, which matches $\dot{WLL}$, the combined box and defrost load during the low load period.

$$\dot{E}_{mix,rack,\,L} = \frac{\dot{q}_{mix,evap,min} + 3.412 \cdot \dot{EF}_{comp,on}(s_{min})}{EER(\text{Adjusted Dewpoint Value})} + \dot{EF}_{comp,on}(s_{min}), \text{ when } \dot{WLL} < \dot{q}_{mix,evap,min}$$

Where, fan speed during the low load period, matches the minimum tested fan speed, s_{min}, because at the minimum fan speed the coil capacity exceeds $\dot{WLL}$, the combined box and defrost load during the low load period.

In the above equations, EER can be determined from Table 17; s_{min} is the minimum operating speed of the evaporator fan; s_H and s_L are fan operating speeds under the high and low load periods, respectively, and determined by

$$s_H = k_7 + k_8 \cdot \dot{WLH} + k_9 \cdot \dot{WLH}^2$$

$$s_L = k_7 + k_8 \cdot \dot{WLL} + k_9 \cdot \dot{WLL}^2$$

7.9.3.6 The system annual walk-in energy factor, AWEF, is determined by the following equations.

If $W\dot{L}L \geq \dot{q}_{mix,evap,min}$, then

$$AWEF = \frac{0.33 \cdot B\dot{L}H + 0.67 \cdot B\dot{L}L}{0.33 \cdot \dot{E}_{mix,rack,\ H} + 0.67 \cdot \dot{E}_{mix,rack,\ L} + \dot{DF}}$$

If $W\dot{L}L < \dot{q}_{mix,evap,min}$, then

$$AWEF = \frac{0.33 \cdot B\dot{L}H + 0.67 \cdot B\dot{L}L}{0.33 \cdot \dot{E}_{mix,rack,\ H} + 0.67 \cdot [\dot{E}_{mix,rack,L} \cdot LFL + \dot{EF}_{comp,off}(1 - LFL)] + \dot{DF}}$$

The terms of defrost power contributing to the system power consumption, $\dot{DF}$, in the above equations shall only be applied to the walk-in freezer systems, and shall be set to zero during the calculation for the walk-in refrigerator systems.

7.10 *Remote Fixed Capacity Condensing Units Serving Walk-ins*

Table 18. Unit Cooler Nominal Values for Condensing Unit Energy Calculations		
Description	**Cooler**	**Freezer**
Saturated Suction Temperature (°F)	25	-20
On-cycle evaporator fan power, per Btu/h of gross capacity at ambient condition (W-h/Btu)	0.016	0.016
Off-cycle evaporator fan power (W)	0.2 x on-cycle evaporator fan power	
Electric defrost energy per cycle, per Btu/h of gross capacity (W-h/cycle per Btu/h)	0	0.12
Number of cycles per day	N/A	4
Daily electric defrost contribution (Btu)	0.95 x daily defrost energy use x 3.413	

7.10.1 Indoor *Condensing Units* Serving Walk-in Refrigerators. The condensing unit shall be tested at the Capacity A, Suction A test conditions in Table 11. Electrical energy consumption by the condensing unit shall be measured during on-cycle periods $E_{CU,\ on,}$.

7.10.1.1 When the condensing unit is on, its steady state gross capacity, Q_{ref}, is reduced by the heat content of on-cycle evaporator fan power, $EF_{comp,on}$, to yield its net capacity, q_{net}.

The reference evaporator has an on-cycle evaporator fan power, in Watts, as a function of condensing unit gross capacity:

$$EF_{comp,on} = 0.016 \cdot Qref$$

$$q_{net} = Q_{ref} - 3.412 \cdot EF_{comp,on} = Q_{ref}\,[1 - 3.412 \cdot 0.016]$$

$$q_{net} = Q_{ref} \cdot 0.9454$$

7.10.1.2 For purpose of this calculation, one third of the time the walk-in box load is assumed to be high, BLH, at 70% of the refrigeration system steady state net capacity, and two thirds of the time the walk-in box load is assumed to be low, BLH, at 10% of the refrigeration system steady state net capacity.

$$B\dot{L}H = 0.7 \cdot \dot{q}_{ss}(90) = 0.7 \cdot Q_{ref} \cdot 0.945 = 0.662 \cdot \dot{Q}_{gross}(90)$$

$$\dot{BLL} = 0.1 \cdot \dot{q}_{net} = 0.1 \cdot Q_{ref} \cdot 0.945 = 0.0945 \cdot Q_{ref}$$

7.10.1.3 The total walk-in system heat load at high load periods and low load periods, WLH and WLL respectively, includes the box heat load and the heat load from the fan energy when the condensing unit compressor is off. This in turn can be used to calculate the fraction of time that the compressor is on during high load and low load periods, LFH and LFL respectively.

$$\dot{WLH} = \dot{BLH} + 3.412 \cdot \dot{EF}_{comp,off}(1 - LFH)$$

$$\dot{WLL} = \dot{BLL} + 3.412 \cdot \dot{EF}_{comp,off}(1 - LFL)$$

Where $EF_{comp,off}$ is assumed to consume 20% of the energy as $EF_{comp,on}$; and the load factors (LFH and LFL) are calculated as follows.

$$EF_{comp,off} = 0.2 \cdot EF_{comp,on} = 0.2 \cdot 0.016 \cdot Q_{ref} = 0.0032 \cdot Q_{ref}$$

$$LFH = \frac{\dot{WLH}}{\dot{q}_{net}} = \frac{\dot{BLH} + 3.412 \cdot \dot{EF}_{comp,off}}{\dot{q}_{net} + 3.412 \cdot \dot{EF}_{comp,off}}$$

$$LFH = \frac{0.7 \cdot \dot{q}_{net} + 3.412 \cdot 0.0032 \cdot Q_{ref}}{\dot{q}_{net} + 3.412 \cdot 0.0032 \cdot Q_{ref}} = \frac{(0.7)(0.945) \cdot Q_{ref} + 0.0109 \cdot Q_{ref}}{0.945 \cdot Q_{ref} + 0.0109 \cdot Q_{ref}} = 0.703$$

$$LFL = \frac{\dot{WLL}}{\dot{q}_{net}} = \frac{0.1 \cdot \dot{q}_{net} + 3.412 \cdot \dot{EF}_{comp,off}}{\dot{q}_{net} + 3.412 \cdot \dot{EF}_{comp,off}}$$

$$LFL = \frac{0.1 \cdot \dot{q}_{net} + 3.412 \cdot 0.0032 \cdot Q_{ref}}{\dot{q}_{net} + 3.412 \cdot 0.0032 \cdot Q_{ref}} = \frac{(0.1)(0.945) \cdot Q_{ref} + 0.0109 \cdot Q_{ref}}{0.945 \cdot Q_{ref} + 0.0109 \cdot Q_{ref}} = 0.110$$

7.10.1.4 The annual walk-in energy factor, AWEF, is determined by a calculation that is a function of two measurements: steady state gross capacity, Q_{ref}, and steady state electrical consumption by the condensing unit measured during on-cycle periods $E_{CU, on}$,

$$AWEF = \frac{0.33 \cdot \dot{BLH} + 0.67 \cdot \dot{BLL}}{\begin{array}{c}0.33 \cdot [(\dot{E}_{CU,on} + \dot{EF}_{comp,on}) \cdot LFH + (\dot{EF}_{comp,off})(1 - LFH)] + \\ 0.67 \cdot [(\dot{E}_{CU,on} + \dot{EF}_{comp,on}) \cdot LFL + (\dot{EF}_{comp,off})(1 - LFL)]\end{array}}$$

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$$AWEF = \frac{(0.33)(0.662) \cdot Q_{ref} + (0.67)(0.0945) \cdot Q_{ref}}{\begin{array}{c}0.33 \cdot [(\dot{E}_{CU,on} + 0.016 \cdot Q_{ref}) \cdot 0.703 + (0.0032 \cdot Q_{ref})(1 - 0.703)] + \\ 0.67 \cdot [(\dot{E}_{CU,on} + 0.016 \cdot Q_{ref}) \cdot 0.110 + (0.0032 \cdot Q_{ref})(1 - 0.110)]\end{array}}$$

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$$AWEF = \frac{0.282 \cdot Q_{ref}}{0.306 \cdot \dot{E}_{CU,on} + 0.00711 \cdot Q_{ref}}$$

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7.10.2 Outdoor *Condensing Units* Serving Walk-in Refrigerators. The condensing unit shall be tested at the Capacity A, Suction A; Capacity B, Suction A and Capacity C, Suction A test conditions in Table 12. Electrical energy consumption by the condensing unit shall be measured at the three ambient outdoor temperature conditions listed in Table 12 during on-cycle periods $E_{CU,on}(t)$, The steady state gross capacity, $Q_{ref}(t)$, shall also be measured at the three ambient outdoor temperature conditions listed in Table 12 during on-cycle periods.

7.10.2.1 When the condensing unit is on, its steady state gross capacity, $Q_{ref}(t)$, is reduced by the heat content of on-cycle evaporator fan power, $EF_{comp,on}$, to yield its steady state net capacity, $q_{ss}(t)$.

The reference evaporator has an on-cycle evaporator fan power, in Watts, as a function of condensing unit gross capacity:

$$EF_{comp,on} = 0.016 \cdot Q_{ref}(95°F)$$

$$q_{ss}(t_j) = Q_{ref}(t_j) – 3.412 \cdot EF_{comp,on} = Q_{ref}(t_j) – 3.412 \cdot 0.016 \cdot Q_{ref}(95°F)$$

$$q_{ss}(t_j)= Q_{ref}(t_j) – 0.0546 \cdot Q_{ref}(95°F)$$

The steady state net capacity at 95°F, is used to calculate the walk-in box load and can be simplified

$$q_{ss}(95°F) = Q_{ref}(95°F) – 3.412 \cdot EF_{comp,on} = Q_{ref}(95°F) [1 – 3.412 \cdot 0.016] = Q_{ref}(95°F) \cdot 0.9454$$

7.10.2.2 For purpose of this calculation, one third of the time the walk-in box load is assumed to be high, BLH, and two thirds of the time the walk-in box load is assumed to be low, BLL. These box load terms, BLH and BLL are a function of the refrigeration system steady state net capacity at 95°F and outdoor air temperature, t_j.

$$B\dot{L}H(t_j) = 0.65 \cdot \dot{q}_{ss}(95°F) + 0.05 \cdot \frac{\dot{q}_{ss}(95°F)\cdot(t_j-35)}{60} = 0.9454 \cdot \left[0.65 \cdot Q_{ref}(95°F) + 0.05 \cdot \frac{\dot{Q}_{ref}(95°F)\cdot(t_j-35)}{60}\right]$$ 149

$$B\dot{L}H(t_j) = \left[0.6145 + 0.000788 \cdot (t_j - 35)\right] \cdot Q_{ref}(95°F)$$

$$B\dot{L}L(t_j) = 0.03 \cdot \dot{q}_{ss}(95°F) + 0.07 \cdot \frac{\dot{q}_{ss}(95°F)\cdot(t_j-35)}{60} = 0.9454 \cdot \left[0.03 \cdot \dot{Q}_{ref}(95°F) + 0.07 \cdot \frac{\dot{Q}_{ref}(95°F)\cdot(t_j-35)}{60}\right]$$ 151

$$B\dot{L}L(t_j) = \left[0.0284 + 0.0011 \cdot (t_j - 35)\right] \cdot Q_{ref}(95°F)$$

7.10.2.3 The total walk-in system heat load at high load periods and low load periods, WLH and WLL respectively, includes the box heat load and the heat load from the fan energy when the condensing unit compressor is off. This in turn can be used to calculate the fraction of time that the compressor is on during high load and low load periods, LFH and LFL respectively.

$$\dot{W}LH\left(t_j\right) = B\dot{L}H\left(t_j\right) + 3.412 \cdot \dot{EF}_{comp,off}\left(1 - LFH\left(t_j\right)\right)$$

$$\dot{W}LL\left(t_j\right) = B\dot{L}L\left(t_j\right) + 3.412 \cdot \dot{EF}_{comp,off}\left(1 - LFL\left(t_j\right)\right)$$

Where

$EF_{comp,off}$ is assumed to consume 20% of the energy as $EF_{comp,on}$; and the load factors (LFH and LFL) are calculated as follows.

$EF_{comp,off} = 0.2 \text{ x } EF_{comp,on} = 0.2 \text{ x } 0.016 \text{ x } Q_{ref}(95) = 0.0032 \text{ x } Q_{ref}(95)$

$$LFH\left(t_j\right) = \frac{\dot{W}LH(t_j)}{\dot{q}_{ss}(t_j)} = \frac{B\dot{L}H\left(t_j\right)+3.412\cdot\dot{EF}_{comp,off}}{\dot{q}_{ss}\left(t_j\right)+3.412\cdot\dot{EF}_{comp,off}}$$

$$LFH\left(t_j\right) = \frac{0.9454\cdot\left[0.65\cdot Q_{ref}(95^{\circ}F)+0.05\cdot\frac{\dot{Q}_{ref}(95^{\circ}F)\cdot\left(t_j-35\right)}{60}\right]+(3.412)(0.0032)\cdot\dot{Q}_{ref}(95^{\circ}F)}{\dot{Q}_{ref}\left(t_j\right)-0.0546\cdot\dot{Q}_{ref}(95^{\circ}F)+(3.412)(0.0032)\cdot\dot{Q}_{ref}(95^{\circ}F)}$$

$$LFH\left(t_j\right) = \frac{\left[0.625\cdot+0.000788\cdot\left(t_j-35\right)\right]\cdot\dot{Q}_{ref}(95^{\circ}F)}{\dot{Q}_{ref}\left(t_j\right)-0.0437\cdot\dot{Q}_{ref}(95^{\circ}F)}$$

$$LFL\left(t_j\right) = \frac{\dot{W}LL(t_j)}{\dot{q}_{ss}(t_j)} = \frac{B\dot{L}L\left(t_j\right)+3.412\cdot\dot{EF}_{comp,off}}{\dot{q}_{ss}\left(t_j\right)+3.412\cdot\dot{EF}_{comp,off}}$$

$$LFL\left(t_j\right) = \frac{0.9454\cdot\left[0.03\cdot Q_{ref}(95^{\circ}F)+0.07\cdot\frac{\dot{Q}_{ref}(95^{\circ}F)\cdot\left(t_j-35\right)}{60}\right]+(3.412)(0.0032)\cdot\dot{Q}_{ref}(95^{\circ}F)}{\dot{Q}_{ref}\left(t_j\right)-0.0546\cdot\dot{Q}_{ref}(95^{\circ}F)+(3.412)(0.0032)\cdot\dot{Q}_{ref}(95^{\circ}F)}$$

$$LFL\left(t_j\right) = \frac{\left[0.0393\cdot+0.0011\cdot\left(t_j-35\right)\right]\cdot\dot{Q}_{ref}(95^{\circ}F)}{\dot{Q}_{ref}\left(t_j\right)-0.0437\cdot\dot{Q}_{ref}(95^{\circ}F)}$$

Where $Q_{ref}(t_j)$ is calculated from the measured condensing unit gross capacity at the ambient test conditions of 35°F, 59°F and 95°F as follows:

If $t_j \leq 59^{\circ}F$

$$\dot{Q}_{ref}\left(t_j\right) = \dot{Q}_{ref}(35^{\circ}F) + \frac{\left(\dot{Q}_{ref}(59^{\circ}F)-\dot{Q}_{ref}(35^{\circ}F)\right)}{59-35}\left(t_j - 35\right)$$

If $t_j > 59^{\circ}F$

$$\dot{Q}_{ref}\left(t_j\right) = \dot{q}_{ss}(59^{\circ}F) + \frac{\left(\dot{Q}_{ref}(95^{\circ}F)-\dot{Q}_{ref}(59^{\circ}F)\right)}{95-59}\left(t_j - 59\right)$$

7.10.2.4 The annual walk-in energy factor, AWEF, is determined by a calculation that is a function of two measurements: steady state gross capacity, Q_{ref}, steady state electrical consumption by the condensing unit measured during on-cycle periods $E_{CU,\ on,}$. These calculations are weighted by the number of bin hours, n_j, from Table D-1 in Appendix D, for each of the 20 bin temperatures, t_j, in table D-1.

$$AWEF = \frac{\sum_{j=1}^{20}\left[0.33\cdot \dot{BLH}(t_j)+0.67\cdot \dot{BLL}(t_j)\right]\cdot n_j}{\sum_{j=1}^{20}\left[\begin{array}{c}0.33\cdot[(\dot{E}_{CU,on}+\dot{EF}_{comp,on})\cdot LFH(t_j)+(\dot{EF}_{comp,off})(1-LFH(t_j))]+\\ 0.67\cdot[(\dot{E}_{CU,on}+\dot{EF}_{comp,on})\cdot LFL(t_j)+(\dot{EF}_{comp,off})(1-LFL(t_j))]\end{array}\right]\cdot n_j}$$

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$$AWEF = \frac{\sum_{j=1}^{20}\left\{0.33\cdot[0.6145+0.000788\cdot(t_j-35)]+0.67\cdot[0.0284+0.0011\cdot(t_j-35)]\right\}\cdot Q_{ref}(95^{\circ}F)\cdot n_j}{\sum_{j=1}^{20}\left[\begin{array}{c}0.33\cdot\left\{\left[\dot{E}_{CU,on}+0.016\cdot Q_{ref}(95^{\circ}F)\right]\cdot LFH(t_j)+\left[0.0032\cdot Q_{ref}(95^{\circ}F)\right]\cdot[1-LFH(t_j)]\right\}+\\ 0.67\cdot\left\{\left[\dot{E}_{CU,on}+0.016\cdot Q_{ref}(95^{\circ}F)\right]\cdot LFL(t_j)+\left[0.0032\cdot Q_{ref}(95^{\circ}F)\right]\cdot[1-LFL(t_j)]\right\}\end{array}\right]\cdot n_j}$$

$$AWEF = \frac{\sum_{j=1}^{20}[0.222+0.000999\cdot(t_j-35)]\cdot Q_{ref}(95^{\circ}F)\cdot n_j}{\sum_{j=1}^{20}\left[\begin{array}{c}Q_{ref}(95^{\circ}F)\cdot[+0.0032+0.00422\cdot LFH(t_j)+0.00858\cdot LFH(t_j)]+\\ \dot{E}_{CU,on}(t_j)\cdot[0.33\cdot LFH(t_j)+0.67\cdot LFL(t_j)]\end{array}\right]\cdot n_j}$$

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If $t_j \le 59^{\circ}F$

$$\dot{E}_{CU,on}(t_j) = \dot{E}_{CU,on}(35^{\circ}F) + \frac{\left(\dot{E}_{CU,on}(59^{\circ}F)-\dot{E}_{CU,on}(35^{\circ}F)\right)}{59-35}(t_j-35)$$

If $t_j > 59^{\circ}F$

$$\dot{E}_{CU,on}(t_j) = \dot{E}_{CU,on}(59^{\circ}F) + \frac{\left(\dot{E}_{CU,on}(95^{\circ}F)-\dot{E}_{CU,on}(59^{\circ}F)\right)}{95-59}(t_j-59)$$

7.10.3 Indoor *Condensing Units* Serving Walk-in Freezers. The condensing unit shall be tested at the Capacity A, Suction A test conditions in Table 13Electrical energy consumption by the condensing unit shall be measured during on-cycle periods $E_{CU,on,}$.

7.10.3.1 When the condensing unit is on, its steady state gross capacity, Q_{ref}, is reduced by the heat content of on-cycle evaporator fan power, $EF_{comp,on}$, to yield its net capacity, q_{net}.

The reference evaporator has an on-cycle evaporator fan power, in Watts, as a function of condensing unit gross capacity:

$EF_{comp,on}$ = 0.016 x Qref

$q_{net} = Q_{ref} - 3.412 \text{ x } EF_{comp,on} = Q_{ref}\,[1 - 3.412 \text{ x } 0.016]$

$q_{net} = Q_{ref} \text{ x } 0.9454$

7.10.3.2 For purpose of this calculation, one third of the time the walk-in box load is assumed to be high, BLH, at 80% of the refrigeration system steady state net capacity, and two thirds of the time the walk-in box load is assumed to be low, BLH, at 40% of the refrigeration system steady state net capacity.

$$\dot{BLH} = 0.8 \cdot \dot{q}_{\text{net}} = 0.8 \cdot Q_{ref} \cdot 0.945 = 0.756 \cdot Q_{ref}$$

$$\dot{BLL} = 0.4 \cdot \dot{q}_{\text{net}} = 0.4 \cdot Q_{ref} \cdot 0.945 = 0.378 \cdot Q_{ref}$$

7.10.3.3 The total walk-in system heat load at high load periods and low load periods, WLH and WLL respectively, includes the box heat load and the heat load from the fan energy when the condensing unit compressor is off. This in turn can be used to calculate the fraction of time that the compressor is on during high load and low load periods, LFH and LFL respectively.

$$\dot{WLH} = \dot{BLH} + 3.412 \cdot \dot{EF}_{\text{comp,off}}(1 - \text{LFH}) + Q_{DF}$$

$$\dot{WLL} = \dot{BLL} + 3.412 \cdot \dot{EF}_{\text{comp,off}}(1 - \text{LFL}) + Q_{DF}$$

Where,
$EF_{comp,off}$ is assumed to consume 20% of the energy as $EF_{comp,on}$;

$EF_{comp,off} = 0.2 \text{ x } EF_{comp,on} = 0.2 \text{ x } 0.016 \text{ x } Q_{ref} = 0.0032 \text{ x } Q_{ref}$

Defrost energy for the reference unit cooler is assumed to be 0.12 W-h per cycle for each Btu/h of gross capacity, and defrost is expected to occur 4 times per 24 hour period.
The daily defrost energy contributions to total walk-in system heat loads, Q_{DF}, is 0.95 x electrical defrost load x 3.412
The hourly defrost load is the daily defrost load divided by 24.

DF = 0.12 W-h x Qref x 4 / 24 = 0.02 x Qref

$Q_{DF} = DF \text{ x } 3.412 \text{ x } 0.95 = 0.0648 \text{ x } Q_{ref}$

The load factors (LFH and LFL) are calculated as follows.

$$\text{LFH} = \frac{\dot{WLH}}{\dot{q}_{\text{net}}} = \frac{\dot{BLH} + 3.412 \cdot \dot{EF}_{\text{comp,off}} + Q_{DF}}{\dot{q}_{\text{net}} + 3.412 \cdot \dot{EF}_{\text{comp,off}}}$$

$$\text{LFH} = \frac{0.8 \cdot \dot{q}_{\text{net}} + 3.412 \cdot 0.0032 \cdot Q_{ref}}{\dot{q}_{\text{net}} + 3.412 \cdot 0.0032 \cdot Q_{ref}} = \frac{(0.8)(0.945) \cdot Q_{ref} + 0.0109 \cdot Q_{ref} + 0.0649 \cdot Q_{ref}}{0.945 \cdot Q_{ref} + 0.0109 \cdot Q_{ref}} = 0.870$$

$$\text{LFL} = \frac{\dot{WLL}}{\dot{q}_{\text{net}}} = \frac{0.4 \cdot \dot{q}_{\text{net}} + 3.412 \cdot \dot{EF}_{\text{comp,off}} + Q_{DF}}{\dot{q}_{\text{net}} + 3.412 \cdot \dot{EF}_{\text{comp,off}}}$$

$$\text{LFL} = \frac{0.4 \cdot \dot{q}_{\text{net}} + 3.412 \cdot 0.0032 \cdot Q_{ref}}{\dot{q}_{\text{net}} + 3.412 \cdot 0.0032 \cdot Q_{ref}} = \frac{(0.4)(0.945) \cdot Q_{ref} + 0.0109 \cdot Q_{ref} + 0.0649 \cdot Q_{ref}}{0.945 \cdot Q_{ref} + 0.0109 \cdot Q_{ref}} = 0.475$$

7.10.3.4 The annual walk-in energy factor, AWEF, is determined by a calculation that is a function of three measurements: steady state gross capacity, Q_{ref}, steady state electrical consumption by the condensing unit measured during on-cycle periods $E_{CU,\ on,}$ and electrical consumption by the condensing unit during off-cycle periods, $E_{CU,\ off}$.

$$AWEF = \frac{0.33 \cdot \dot{BLH} + 0.67 \cdot \dot{BLL}}{0.33 \cdot [(\dot{E}_{CU,on} + \dot{EF}_{comp,on}) \cdot LFH + (\dot{EF}_{comp,off})(1 - LFH)] + 0.67 \cdot [(\dot{E}_{CU,on} + \dot{EF}_{comp,on}) \cdot LFL + (\dot{EF}_{comp,off})(1 - LFL)] + \dot{DF}} \quad (184)$$

$$AWEF = \frac{(0.33)(0.756) \cdot Q_{ref} + (0.67)(0.378) \cdot Q_{ref}}{0.33 \cdot [(\dot{E}_{CU,on} + 0.016 \cdot Q_{ref}) \cdot 0.870 + (0.0032 \cdot Q_{ref})(1 - 0.870)] + 0.67 \cdot [(\dot{E}_{CU,on} + 0.016 \cdot Q_{ref}) \cdot 0.475 + (0.0032 \cdot Q_{ref})(1 - 0.475)] + 0.02 \cdot Q_{ref}} \quad (185)$$

$$AWEF = \frac{0.502 \cdot Q_{ref}}{0.605 \cdot \dot{E}_{CU,on} + 0.0309 \cdot Q_{ref}} \quad (186)$$

7.10.4 Outdoor *Condensing Units* Serving Walk-in Freezers. The condensing unit shall be tested at the Capacity A, Suction A; Capacity B, Suction A and Capacity C, Suction A test conditions in Table 14. Electrical energy consumption by the condensing unit shall be measured at the three ambient outdoor temperature conditions listed in Table 14 during on-cycle periods $E_{CU,on}$(t The steady state gross capacity, $Q_{ref}(t)$, shall also be measured at the three ambient outdoor temperature conditions listed in Table 14 during on-cycle periods.

7.10.4.1 When the condensing unit is on, its steady state gross capacity, $Q_{ref}(t)$, is reduced by the heat content of on-cycle evaporator fan power, $EF_{comp,on}$, to yield its steady state net capacity, $q_{ss}(t)$.

The reference evaporator has an on-cycle evaporator fan power, in Watts, as a function of condensing unit gross capacity:

$$EF_{comp,on} = 0.016 \text{ x } Q_{ref}(95°F)$$

$$q_{ss}(t_j) = Q_{ref}(t_j) - 3.412 \text{ x } EF_{comp,on} = Q_{ref}(t_j) - 3.412 \text{ x } 0.016 \text{ x } Q_{ref}(95°F)$$

$$q_{ss}(t_j) = Q_{ref}(t_j) - 0.0546 \text{ x } Q_{ref}(95°F)$$

The steady state net capacity at 95°F, is used to calculate the walk-in box load and can be simplified

$$q_{ss}(95°F) = Q_{ref}(95°F) - 3.412 \text{ x } EF_{comp,on} = Q_{ref}(95°F)\ [1 - 3.412 \text{ x } 0.016] = Q_{ref}(95°F) \text{ x } 0.9454$$

7.10.4.2 For purpose of this calculation, one third of the time the walk-in box load is assumed to be high, BLH, and two thirds of the time the walk-in box load is assumed to be low, BLL. These box load terms, BLH and BLL are a function of the refrigeration system steady state net capacity at 95°F and outdoor air temperature, t_j.

$$\dot{BLH}(t_j) = 0.55 \cdot \dot{q}_{ss}(95°F) + 0.25 \cdot \frac{\dot{q}_{ss}(95°F)\cdot(t_j+10)}{105} = 0.9454 \cdot \left[0.55 \cdot Q_{ref}(95°F) + 0.25 \cdot \frac{\dot{Q}_{ref}(95°F)\cdot(t_j+10)}{105}\right]$$ 194

$$\dot{BLH}(t_j) = \left[0.52 + 0.00225 \cdot (t_j + 10)\right] \cdot Q_{ref}(95°F)$$

$$\dot{BLL}(t_j) = 0.15 \cdot \dot{q}_{ss}(95°F) + 0.25 \cdot \frac{\dot{q}_{ss}(95°F)\cdot(t_j+10)}{105} = 0.9454 \cdot \left[0.15 \cdot \dot{Q}_{ref}(95°F) + 0.25 \cdot \frac{\dot{Q}_{ref}(95°F)\cdot(t_j+10)}{105}\right]$$ 196

$$\dot{BLL}(t_j) = \left[0.142 + 0.00225 \cdot (t_j + 10)\right] \cdot Q_{ref}(95°F)$$

7.10.4.3 The total walk-in system heat load at high load periods and low load periods, WLH and WLL respectively, includes the box heat load and the heat load from the fan energy when the condensing unit compressor is off. This in turn can be used to calculate the fraction of time that the compressor is on during high load and low load periods, LFH and LFL respectively.

$$\dot{WLH}\,(t_j) = \dot{BLH}(t_j) + 3.412 \cdot \dot{EF}_{comp,off}\left(1 - LFH(t_j)\right) + Q_{DF}$$

$$\dot{WLL}(t_j) = \dot{BLL}(t_j) + 3.412 \cdot \dot{EF}_{comp,off}\left(1 - LFL(t_j)\right) + Q_{DF}$$

Where

$EF_{comp,off}$ is assumed to consume 20% of the energy as $EF_{comp,on}$; and the load factors (LFH and LFL) are calculated as follows.

$EF_{comp,off} = 0.2 \text{ x } EF_{comp,on} = 0.2 \text{ x } 0.016 \text{ x } Q_{ref}(95) = 0.0032 \text{ x } Q_{ref}(95)$

Defrost energy for the reference unit cooler is assumed to be 0.12 W-h per cycle for each Btu/h of gross capacity, and defrost is expected to occur 4 times per 24 hour period.
The daily defrost energy contributions to total walk-in system heat loads, Q_{DF}, is 0.95 x electrical defrost load x 3.412.
The hourly defrost load is the daily defrost load divided by 24.

DF = 0.12 W-h x Qref(95°F) x 4 / 24 = 0.02 x Qref(95°F)

Q_{DF} = DF x 3.412 x 0.95 = 0.0648 x Q_{ref}(95°F)

The load factors (LFH and LFL) are calculated as follows.

$$LFH = \frac{\dot{WLH}}{\dot{q}_{net}} = \frac{\dot{BLH} + 3.412\cdot\dot{EF}_{comp,off} + Q_{DF}}{\dot{q}_{net} + 3.412\cdot\dot{EF}_{comp,off}}$$

$$LFH(t_j) = \frac{\dot{WLH}(t_j)}{\dot{q}_{ss}(t_j)} = \frac{\dot{BLH}(t_j) + 3.412\cdot\dot{EF}_{comp,off} + Q_{DF}}{\dot{q}_{ss}(t_j) + 3.412\cdot\dot{EF}_{comp,off}}$$

$$LFH(t_j) = \frac{0.9454\cdot\left[0.55\cdot Q_{ref}(95^{\circ}F)+0.25\cdot\frac{\dot{Q}_{ref}(95^{\circ}F)\cdot(t_j+10)}{105}\right]+(3.412)(0.0032)\cdot\dot{Q}_{ref}(95^{\circ}F)+0.0648\cdot\dot{Q}_{ref}(95^{\circ}F)}{\left[\dot{Q}_{ref}(t_j)-0.0546\cdot\dot{Q}_{ref}(95^{\circ}F)\right]+(3.412)(0.0032)\cdot\dot{Q}_{ref}(95^{\circ}F)}$$

$$LFH(t_j) = \frac{\left[0.5957\cdot+0.002251\cdot(t_j+10)\right]\cdot\dot{Q}_{ref}(95^{\circ}F)}{\dot{Q}_{ref}(t_j)-0.0437\cdot\dot{Q}_{ref}(95^{\circ}F)}$$

$$LFL(t_j) = \frac{\dot{W}LL(t_j)}{\dot{q}_{ss}(t_j)} = \frac{B\dot{L}L(t_j)+3.412\cdot\dot{EF}_{comp,off}+Q_{DF}}{\dot{q}_{ss}(t_j)+3.412\cdot\dot{EF}_{comp,off}}$$

$$LFL(t_j) = \frac{0.9454\cdot\left[0.15\cdot Q_{ref}(95^{\circ}F)+0.25\cdot\frac{\dot{Q}_{ref}(95^{\circ}F)\cdot(t_j+10)}{105}\right]+(3.412)(0.0032)\cdot\dot{Q}_{ref}(95^{\circ}F)+0.0648\cdot\dot{Q}_{ref}(95^{\circ}F)}{\dot{Q}_{ref}(t_j)-0.0546\cdot\dot{Q}_{ref}(95^{\circ}F)+(3.412)(0.0032)\cdot\dot{Q}_{ref}(95^{\circ}F)}$$

$$LFL(t_j) = \frac{\left[0.218\cdot+0.00225\cdot(t_j+10)\right]\cdot\dot{Q}_{ref}(95^{\circ}F)}{\dot{Q}_{ref}(t_j)-0.0437\cdot\dot{Q}_{ref}(95^{\circ}F)}$$

Where $Q_{ref}(t_j)$ is calculated from the measured condensing unit gross capacity at the ambient test conditions of 35°F, 59°F and 95°F as follows.

If $t_j \leq 59^{\circ}F$

$$\dot{Q}_{ref}(t_j) = \dot{Q}_{ref}(35^{\circ}F) + \frac{\left(\dot{Q}_{ref}(59^{\circ}F)-\dot{Q}_{ref}(35^{\circ}F)\right)}{59-35}(t_j - 35)$$

If $t_j > 59^{\circ}F$

$$\dot{Q}_{ref}(t_j) = \dot{q}_{ss}(59^{\circ}F) + \frac{\left(\dot{Q}_{ref}(95^{\circ}F)-\dot{Q}_{ref}(59^{\circ}F)\right)}{95-59}(t_j - 59)$$

7.10.4.4 The annual walk-in energy factor, AWEF, is determined by a calculation that is a function of three measurements: steady state gross capacity, Q_{ref}, steady state electrical consumption by the condensing unit measured during on-cycle periods $E_{CU,\ on,}$ and electrical consumption by the condensing unit during off-cycle periods, $E_{CU,\ off}$. These calculations are weighted by the number of bin hours, n_j, from Table D-1 in Appendix D, for each of the 20 bin temperatures, t_j, in table D-1.

$$AWEF = \frac{\sum_{j=1}^{20}\left[0.33\cdot BL\dot{H}(t_j)+0.67\cdot BL\dot{L}(t_j)\right]\cdot n_j}{\sum_{j=1}^{20}\left[\begin{array}{l}0.33\cdot[(\dot{E}_{CU,on}+\dot{EF}_{comp,on})\cdot LFH(t_j)+(\dot{EF}_{comp,off})(1-LFH(t_j))]+\\ 0.67\cdot[(\dot{E}_{CU,on}+\dot{EF}_{comp,on})\cdot LFL(t_j)+(\dot{EF}_{comp,off})(1-LFL(t_j))]\end{array}\right]\cdot n_j}$$

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$$\mathrm{AWEF} = \frac{\sum_{j=1}^{20}\left\{0.33\cdot\left[0.52+0.00225\cdot(t_j+10)\right]+0.67\cdot\left[0.142+0.00225\cdot(t_j+10)\right]\right\}\cdot Q_{ref}(95^\circ F)\cdot n_j}{\sum_{j=1}^{20}\begin{bmatrix}0.33\cdot\left\{\left[\dot{\mathrm{E}}_{\mathrm{CU,on}}+0.016\cdot Q_{ref}(95^\circ F)\right]\cdot \mathrm{LFH}(t_j)+\left[0.0032\cdot Q_{ref}(95^\circ F)\right]\cdot\left[1-\mathrm{LFH}(t_j)\right]\right\}+\\ 0.67\cdot\left\{\left[\dot{\mathrm{E}}_{\mathrm{CU,on}}+0.016\cdot Q_{ref}(95^\circ F)\right]\cdot \mathrm{LFL}(t_j)+\left[0.0032\cdot Q_{ref}(95^\circ F)\right]\cdot\left[1-\mathrm{LFL}(t_j)\right]\right\}\end{bmatrix}\cdot n_j}$$

$$\mathrm{AWEF} = \frac{\sum_{j=1}^{20}\left[0.267+0.00225\cdot(t_j+10)\right]\cdot Q_{ref}(95^\circ F)\cdot n_j}{\sum_{j=1}^{20}\begin{bmatrix}Q_{ref}(95^\circ F)\cdot\left[+0.0032+0.00422\cdot \mathrm{LFH}(t_j)+0.00858\cdot \mathrm{LFH}(t_j)\right]+\\ \dot{\mathrm{E}}_{\mathrm{CU,on}}(t_j)\cdot\left[0.33\cdot \mathrm{LFH}(t_j)+0.67\cdot \mathrm{LFL}(t_j)\right]\end{bmatrix}\cdot n_j}$$

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Where $E_{CU,on}(t_j)$ and $E_{CU,off}(t_j)$ is calculated from the measured condensing unit gross capacity at the ambient test conditions of 35°F, 59°F and 95°F as follows.

If $t_j \leq 59^\circ\mathrm{F}$

$$\dot{\mathrm{E}}_{\mathrm{CU,on}}(t_j) = \dot{\mathrm{E}}_{\mathrm{CU,on}}(35^\circ\mathrm{F}) + \frac{\left(\dot{\mathrm{E}}_{\mathrm{CU,on}}(59^\circ\mathrm{F})-\dot{\mathrm{E}}_{\mathrm{CU,on}}(35^\circ\mathrm{F})\right)}{59-35}(t_j-35)$$

$$\dot{\mathrm{E}}_{\mathrm{CU,off}}(t_j) = \dot{\mathrm{E}}_{\mathrm{CU,off}}(35^\circ\mathrm{F}) + \frac{\left(\dot{\mathrm{E}}_{\mathrm{CU,off}}(59^\circ\mathrm{F})-\dot{\mathrm{E}}_{\mathrm{CU,off}}(35^\circ\mathrm{F})\right)}{59-35}(t_j-35)$$

If $t_j > 59^\circ\mathrm{F}$

$$\dot{\mathrm{E}}_{\mathrm{CU,on}}(t_j) = \dot{\mathrm{E}}_{\mathrm{CU,on}}(59^\circ\mathrm{F}) + \frac{\left(\dot{\mathrm{E}}_{\mathrm{CU,on}}(95^\circ\mathrm{F})-\dot{\mathrm{E}}_{\mathrm{CU,on}}(59^\circ\mathrm{F})\right)}{95-59}(t_j-59)$$

$$\dot{\mathrm{E}}_{\mathrm{CU,off}}(t_j) = \dot{\mathrm{E}}_{\mathrm{CU,off}}(59^\circ\mathrm{F}) + \frac{\left(\dot{\mathrm{E}}_{\mathrm{CU,off}}(95^\circ\mathrm{F})-\dot{\mathrm{E}}_{\mathrm{CU,off}}(59^\circ\mathrm{F})\right)}{95-59}(t_j-59)$$

Section 8. Symbols and Subscripts

8.1 *Symbols and Subscripts.* The symbols and subscripts used in this standard are as follows:

AWEF:	Annual Walk-in Energy Factor, Btu/W·h
$\mathrm{BL}(t_j)$:	Heat removed from walk-in box that does not include the heat generated by the operation of refrigeration systems, W·h
$\dot{\mathrm{BLH}}(t_j)$:	Non-equipment related walk-in box load during high load period, Btu/h
$\dot{\mathrm{BLL}}(t_j)$:	Non-equipment related walk-in box load during low load period, Btu/h
c_{pi}C:	Specific heat of ice, Btu/lb·°F
c_{pw}:	Specific heat of water, Btu/lb·°F
DF:	Daily average defrost energy required for the refrigeration system, W·h
$DF_{f:}$	Energy input for a defrost cycle for frost coil condition, W·h
$DF_{d:}$	Energy input for a defrost cycle for dry coil condition, W·h
$\dot{DF}$:	Defrost power consumption, W

EER Energy Efficiency Ratio, Btu/W·h
$E(t_j)$: System energy consumption at t_j , W·h
$\dot{E}_c$: Total power consumption of the heater and auxiliary equipment of the calibrated box, W
$\dot{E}_{cu,on}$: Power consumption of the condensing unit for stand alone test, W
$\dot{E}_{mix,rack}$: Power consumption of the rack system for mix-match system, W
$\dot{E}_{mix,rack,H}$: Power consumption of the rack system for mix-match system during a high load period, W
$\dot{E}_{mix,rack,L}$: Power consumption of the rack system for mix-match system during a low load period, W
$\dot{E}_{ss}(t_j)$: System steady state power consumption at t_j, including power consumptions of compressor(s), both condenser and evaporator fans, W
$\dot{EF}_{comp,off}$: Evaporator fan power consumption during compressor off period, W
$\dot{EF}_{comp,on}$: Evaporator fan power consumption during compressor on period, W
FS: Fan speed (s), rpm
H: Refrigerant enthalpy, Btu/lb
H_{fus} Latent heat of fusion, Btu/lb
i: Intermediate compressor capacity case in which the compressor was tested at the designated testing condition.
j: Bin Numberk: Case Number, (1: low capacity or minimum capacity; 2: high capacity or maximum capacity; i: intermediate capacity; v: variable capacity)
k_7~k_{12}: coefficients derived from evaporator coil test points
K_{cb} : Heat leakage coefficient of calibrated box, Btu/h·˚F
LFH: Load factor during high load period
LFL: Load factor during low load period
$\dot{m}_{ref}$: Refrigerant mass flow rate, lb/h
$\dot{m}_{ref,1}$: Refrigerant mass flow rate measured at subcooled refrigerant liquid line (1st independent measurement), lb/h
$\dot{m}_{ref,2}$: Refrigerant mass flow rate measured at subcooled or superheated refrigerant vapor line (2nd independent measurement), lb/h
m_w: Weight of the drained water from defrost, lb
n_j: Bin hours, hr
N: Number of motors
N_{DF}: Number of defrost per day
P_b: Barometric pressure, in Hg
$q(t_j)$: Heat removed from the walk-in box at t_j , Btu
$\dot{q}_{ss}(t_j)$: System steady state net refrigeration capacity at t_j, Btu/h
$\dot{q}_{mix,cd}$: Condensing unit capacity for mix-match system, Btu/h
$\dot{q}_{mix,evap}$: Evaporator coil net capacity for mix-match system, Btu/h
$\dot{q}_{mix,evap,max}$: net coil capacity under the maximum fan speed test point, Btu/h
$\dot{Q}_{air}$: Air-side gross refrigeration capacity, Btu/h
Q_{DF}: Daily contribution of load attributed to defrost, Btu
$\dot{Q}_{DF}$: Defrost power consumption contributed to the box load, Btu/h

$Qg\dot{r}oss$

$\dot{Q}_{mix,evap}$: Coil gross capacity for mix-match system, Btu/h
$\dot{Q}_t$: Gross total refrigeration capacity, Btu/h
$\dot{Q}_{ref}$: Refrigerant-side gross capacity, Btu/h
$\dot{Q}_{ref,1}$: Refrigerant-side gross capacity calculated based on the first independent measurement, Btu/h
$\dot{Q}_{ref,2}$: Refrigerant-side gross capacity calculated based on the second independent measurement, Btu/h
s_H: Evaporator fan speed resulting in a coil capacity which matches walk-in system load during a high load period, rpm
s_L: Evaporator fan speed resulting in a coil capacity which matches walk-in system load during a low load period, rpm
s_{min}: Minimum evaporator fan speed, rpm
T_{cb} : Average dry-bulb temperature of the air within the calibrated box, ˚F

T_{en}:	Average dry-bulb temperature of the air within the temperature controlled enclosure, ˚F
T_{evap}:	Evaporating temperature, °F
T_{db}:	Dry-bulb temperature of air at inlet, °F
T_{wb}:	Wet-bulb temperature of air at inlet, °F
T_{w}:	Temperature of the drained water from defrost, ˚F
t_j:	Bin temperature, ˚F
t_{IH}:	The outdoor temperature at which the total walk-in system heat load equals system net capacity when the compressor operates at low or minimum capacity (k=1) during the high load period, ˚F
t_{IIH}:	The outdoor temperature at which the total walk-in system heat load equals system net capacity when the compressor operates at high or maximum capacity (k=2) during the high load period, ˚F
t_{IL}:	The outdoor temperature at which the total walk-in system heat load equals system net capacity when the compressor operates at low or minimum capacity (k=1) during the low load period, ˚F
t_{IIL}:	The outdoor temperature at which the total walk-in system heat load equals system net capacity when the compressor operates at high or maximum capacity (k=2) during the low load period, ˚F
t_{VH}:	The outdoor temperature at which the unit, when operating at the intermediate capacity that is tested under the designated condition, provides a refrigeration capacity that is equal to the total walk-in system heat load during high load period, ˚F
t_{VL}:	The outdoor temperature at which the unit, when operating at the intermediate capacity that is tested under the designated condition, provides a refrigeration capacity that is equal to the total walk-in system heat load during low load period, ˚F
V:	Voltage of each phase, V
$\dot{V}_{air}$:	Air volumetric flow rate, cfm
$W\dot{L}H(t_j)$:	Total walk-in system heat load during high load period, Btu/h
$W\dot{L}L(t_j)$:	Total walk-in system heat load during low load period, Btu/h

Subscript

H:	High load period
in:	Inlet
L:	Low load period
out:	Outlet
ss:	Steady state
v:	Variable compressor capacity case in which the compressor was operated at any capacity between the max and min capacities.

Section 9. Minimum Data Requirements for Published Ratings

9.1 *Minimum Data Requirements for Published Ratings.* As a minimum, Published Ratings shall include all Standard Ratings. All claims to ratings within the scope of this standard shall include the statement "Rated in accordance with AHRI Standard 1250 (I-P)". All claims to ratings outside the scope of this standard shall include the statement "Outside the scope of AHRI Standard 1250 (I-P)". Wherever Application Ratings are published or printed, they shall include a statement of the conditions at which the ratings apply.

Section 10. Marking and Nameplate Data

10.1 *Marking and Nameplate Data.* As a minimum, the manufacturer name or trade-name; model number; refrigerant(s); current, A; voltage, V; frequency, Hz; and phase shall be shown in a conspicuous place on the unit.

Nameplate voltages for 60 Hertz systems shall include one or more of the equipment nameplate voltage ratings shown in Table 1 of AHRI Standard 110. Nameplate voltages for 50 Hertz systems shall include one or more of the utilization voltages shown in Table 1 of IEC Standard 60038.

Section 11. Conformance Conditions

11.1 *Conformance*. While conformance with this standard is voluntary, conformance shall not be claimed or implied for products or equipment within the standard's *Purpose* (Section 1) and *Scope* (Section 2) unless such product claims meet all of the requirements of the standard and all of the testing and rating requirements are measured and reported in complete compliance with the standard. Any product that has not met all the requirements of the standard shall not reference, state, or acknowledge the standard in any written, oral, or electronic communication.

APPENDIX A. REFERENCES – NORMATIVE

A1. Listed here are all standards, handbooks, and other publications essential to the formation and implementation of the standard. All references in this appendix are considered as part of this standard.

A1.1 AHRI Standard 210/240-2008 (formerly ARI Standard 210/240-2008), *Unitary Air-Conditioning and Air-Source Heat Pump Equipment,* 2008, Air-Conditioning, Heating and Refrigeration Institute, 2111 Wilson Boulevard, Suite 500, Arlington, VA 22201, U.S.A.

A1.2 AHRI Standard 420-2008 (formerly ARI Standard 420-2008), *Performance Rating of Forced-Circulation Free-Delivery Unit Coolers for Refrigeration,* 2008, Air-Conditioning, Heating and Refrigeration Institute, 2111 Wilson Boulevard, Suite 500, Arlington, VA 22201, U.S.A.

A1.3 AHRI Standard 520-2004 (formerly ARI Standard 520-2004), *Performance Rating of Positive Displacement Condensing Units,* 2004, Air-Conditioning, Heating and Refrigeration Institute, 2111 Wilson Boulevard, Suite 500, Arlington, VA 22201, U.S.A.

A1.4 AHRI Standard 1200-2008 (formerly ARI Standard 1200-2008), *Performance Rating of Commercial Refrigerated Display Merchandisers and Storage Cabinets,* 2008, Air-Conditioning, Heating and Refrigeration Institute, 2111 Wilson Boulevard, Suite 500, Arlington, VA 22201, U.S.A.

A1.5 ANSI/ASHRAE 23 -2005, *Methods of Testing for Rating Positive Displacement Refrigerant Compressors and Condensing Units*, 2005, American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc., 1791 Tullie CircleN.E., Atlanta, GA 30329, U.S.A.

A1.6 ANSI/ASHRAE 116-1995, *Methods of Testing For Rating Seasonal Efficiency of Unitary Air Conditioners and Heat Pumps*, 1995, American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc., 1791 Tullie CircleN.E., Atlanta, GA 30329, U.S.A.

A1.7 ANSI/ASHRAE Standard 41.4-1996 (RA 2006), *Standard Method for Measurement of Proportion of Lubricant in Liquid Refrigerant*, 2006, American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc., 1791 Tullie Circle, N.E., Atlanta, GA 30329, U.S.A.

A1.8 ASHRAE Standard 37-2005, *Methods of Testing for Rating Unitary Air-Conditioning and Heat Pump Equipment*, 2005, American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc., 1791 Tullie Circle N.E., Atlanta, GA 30329, U.S.A.

A1.9 ANSI/ASHRAE Standard 41.1-1986 (RA 2006), *Standard Method For Temperature Measurement*, 2006, American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc., 1791 Tullie Circle, N.E., Atlanta, GA 30329, U.S.A.

A1.10 ANSI/ASHRAE Standard 41.3-1989, *Standard Method For Pressure Measurement*, 1989, American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc., 1791 Tullie Circle, N.E., Atlanta, GA 30329, U.S.A.

A1.11 ANSI/ASHRAE Standard 41.10-2008, *Standard Methods for Volatile-Refrigerant Mass Flow Measurements Using Flowmeters,* 2008, American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc., 1791 Tullie Circle, N.E., Atlanta, GA 30329, U.S.A.

A1.12 ANSI/ASHRAE Standard 41.2.1987, Standard methods for laboratory air-flow measurement, 1987, American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc., 1791 Tullie Circle, N.E., Atlanta, GA 30329, U.S.A.
A1.13 TYM3WeatherDataforKansasCity,Missouri,http://rredc.nrel.gov/solar/old_data/nsrdb/1991-2005/tmy3/by_state_and_city.html#O

A1.14 *ASHRAEwiki, Terminology,* http://wiki.ashrae.org/index.php/ASHRAEwiki, 2014, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., 1791 Tullie Circle, N.E., Atlanta, GA 30329, U.S.A.

APPENDIX B. REFERENCES – INFORMATIVE

None.

APPENDIX C. METHODS OF TESTING WALK-IN COOLER AND FREEZER SYSTEMS – NORMATIVE

C1. *Purpose*. The purpose of this appendix is to provide a method of testing for walk-in cooler and freezer systems that have either matched or mix-matched unit coolers and condensing units

C2. *Scope.* These methods of testing apply to walk-in cooler and freezer systems that have either matched or mix-matched factory-made, forced circulation, free-delivery unit coolers and factory-made electric motor driven, single and variable capacity positive displacement condensing units.

C2.1 *Exclusions.* These methods of testing do not apply to:

C2.1.1 Air-conditioning units used primarily for comfort cooling for which testing methods are given in other standards.

C2.1.2 Unit Coolers installed in or connected to ductwork

C2.1.3 Parallel rack refrigeration systems

C2.1.4 Field testing of Unit Coolers

C3. *Measurements.* All the measurements associated with the test shall be in accordance with AHRI Standard 1250 Table 1 for required instrumentation accuracy.

C3.1 *Temperature Measurements*

C3.1.1 Temperature measurements shall be made in accordance with ANSI/ASHRAE Standard 41.1.

C3.1.2 Air wet-bulb and dry-bulb temperatures entering the Unit Cooler shall be measured based on the airflow area at the point of measurement. One measuring station is required for each 2.0 ft^2 of the first 10.0 ft^2 of airflow area and one additional measuring station is required for each 4.0 ft^2 of airflow area above 10.0 ft^2. A minimum of two stations shall be used and the number of measuring stations shall be rounded up to the next whole number.

C3.1.3 The airflow area shall be divided into the required number of equal area rectangles with aspect ratios no greater than 2 to 1. Additional measuring stations may be necessary to meet this requirement. The measuring stations shall be located at the geometric center of each rectangle.

C3.1.4 The maximum allowable deviation between any two air temperature measurement stations shall be 2.0 °F.

C3.1.5 If sampling tubes are used, each tube opening may be considered a temperature measuring station provided the openings are uniformly spaced along the tube, the airflow rates entering each port are relatively uniform (± 15%) and the arrangement of tubes complies with the location requirements of C3.1.3. Additionally, a one time temperature traverse shall be made over the measurement surface, prior to the test to assess the temperature variation and ensure it complies with the allowable deviation specified in C3.1.4. (Refer to ANSI/ASHRAE Standard 41.1 for more information and diagrams).

C3.1.6 Refrigerant temperatures entering and leaving the Unit Cooler shall be measured by sheathed temperature sensors immersed in flowing refrigerant or by a temperature measuring instrument placed in a thermometer well and inserted into the refrigerant stream. These wells shall be filled with non-solidifying, thermal conducting liquid or paste to ensure the temperature sensing instrument is exposed to a representative temperature. The entering temperature of the refrigerant shall be measured within six pipe diameters upstream of the control device.

C3.2 *Pressure Measurements.* Connections for pressure measurements shall be smooth and flush within the pipe wall and shall be located not less than six pipe diameters downstream from fittings, bends, or obstructions. (Refer to ANSI/ASHRAE Standard 41.3 for more information and diagrams).

C3.3 *Refrigerant Properties Measurement*

C3.3.1 With the equipment operating at the desired test conditions, the temperature and pressure of the refrigerant leaving the unit cooler, entering the expansion device, and entering and leaving the compressor shall be measured. For cases where the calibrated box method is also conducted, data used to calculate capacity according to the refrigerant enthalpy method and the calibrated box method shall be collected over the same intervals.

C3.3.2 On equipment not sensitive to refrigerant charge, pressure measuring instruments may be tapped into the refrigerant lines provided that they do not affect the total charge by more than 0.5%.

C3.3.3 On equipment sensitive to refrigerant charge, a preliminary test is required prior to connecting any pressure gauges or beginning the first official test. In preparation for this preliminary test, temperature sensors shall be attached to the equipment's evaporator and condenser coils. The sensors shall be located at points that are not affected by vapor superheat or liquid subcooling. Placement near the midpoint of the coil, at a return bend, is recommended. The preliminary test shall be conducted with the requirement that the temperatures of the on-coil sensors be included with the regularly recorded data. After the preliminary test is completed, the refrigerant shall be removed from the equipment, and the needed pressure gauges shall be installed. The equipment shall be evacuated and recharged with refrigerant. The test shall then be repeated. Once steady-state operation is achieved, refrigerant shall be added or removed until, as compared to the average values from the preliminary test, the following conditions are achieved: (1) each on-coil temperature sensor indicates a reading that is within ±0.5°F, (2) the temperatures of the refrigerant entering and leaving the compressor are within ±4°F, and (3) the refrigerant temperature entering the expansion device is within ±1°F. Once these conditions have been achieved over an interval of at least ten minutes, refrigerant charging equipment shall be removed and the first of the official tests shall be initiated.

C3.3.4 No instrumentation shall be removed, replaced, or otherwise disturbed during any portion of a complete capacity test.

C3.3.5 Temperatures and pressures of the refrigerant vapor entering and leaving the compressor shall be measured at approximately 10 inches from the compressor shell.

C3.4 *Refrigerant Flow Measurement*

C3.4.1 Refrigerant flow meters shall be installed and the flow rate of Volatile Refrigerants shall be measured in accordance with ANSI/ASHRAE Standard 41.10.

C3.4.2 The refrigerant flow rate shall be measured with an integrating type flow meter connected in the liquid line upstream of the refrigerant control device. This meter shall be sized so that its pressure drop does not exceed the vapor pressure change that a 4°F saturation temperature change would produce. Refrigerant flow meter is only allowed to be installed at the superheated vapor line as second independent measurement when the refrigerant enthalpy method is used. In such a case, refrigerant vapor must be superheated both upstream and downstream of the meter to ensure the vapor remains single phase.

C3.4.3 Flow meters shall be installed with at least ten pipe diameters upstream and five diameters downstream of the meter, in straight pipe free of valves and fittings, or in accordance with the manufacturer's instructions.

C3.4.4 A direct reading mass-flow-rate measuring device, such as a Coriolis meter, is the preferred instrument for measuring the refrigerant flow rate. Other flow meters that demonstrate the capability to measure flow rate with the specified accuracy are also acceptable.

C3.4.5 Temperature and pressure measuring instruments and a sight glass shall be installed immediately downstream of the meter to determine if the refrigerant liquid is adequately subcooled. Subcooling of 3°F and the absence of any vapor bubbles in the liquid are considered adequate. It is recommended that the meter be installed at the bottom of a vertical downward loop in the liquid line to take advantage of the static head of liquid thus provided.

C3.5 *Unit Cooler Fan Power Measurement.* The following shall be measured and recorded during a compressor-off cycle fan power test.

$\dot{EF}_{comp,off}$	Total electrical power input to fan motor(s) of Unit Cooler, W
FS	Fan speed (s), rpm
N	Number of motors
P_b	Barometric pressure, in.Hg
T_{db}	Dry-bulb temperature of air at inlet, °F
T_{wb}	Wet-bulb temperature of air at inlet, °F
V	Voltage of each phase, V

For a given motor winding configuration, the total power input shall be measured at the highest nameplated voltage. For three-phase power, voltage imbalance shall be no more than 2 % from phase to phase.

C3.6 *Recording and Measurement Intervals*. For steady state testing, data shall be recorded after the unit to be tested approached its steady state conditions for at least 30 minutes at the specified test conditions defined in Section 5.1. The unit shall be maintaining its steady state throughout the entire recording period. Measurement intervals shall be in accordance with Table C1.

C3.6.1 The steady state operation is defined as follows. The variations of the air-side temperatures are within ±2˚F of the average values. The saturated refrigerant temperatures corresponding to the measured refrigerant-side pressures have maximum variations of ±3˚F of the average values. The refrigerant mass flow rates' fluctuations are within 2% of the readings.

Table C1. Test Readings[1]

Test Parameter	Minimum Data Collection Rate, Test Readings per Hour	Minimum Number of Test Readings per Test Run [3]
Temperature	30	15
Pressure	30	15
Refrigerant mass flow rate	30	15
Test room barometric pressure	1	1
Fan speed(s)	1	1
Total electrical power input to fan motor(s)	3	2
Total electrical power input to heater and auxiliary equipment [2]	3	2

Notes:
1. Once the system approaches steady state condition, data shall be recorded.
2. For calibrated box only (Method 2)
3. Duration of recording data shall be a minimum of 30 minutes

C4. *Walk-in system General Data.* Refer to AHRI Standard 420 and ANSI/AHRI Standard 520 for the information that shall be recorded, where applicable, regarding the system physic data and test information.

C5. *Methods of Testing for walk-in cooler and freezer systems that have matched unit coolers and condensing units or stand alone unit coolers*. The testing of the walk-in cooler and freezer systems include steady state test, defrost test and off-cycle fan power test.

C5.1 The Gross Total Refrigeration Capacity of Unit Coolers from steady state test shall be determined by either one of the following methods.

C5.1.1 *Method 1, DX Dual Instrumentation (Refrigerant Enthalpy Method).* The Refrigeration Capacity shall be determined by measuring the enthalpy change and the mass flow rate of the refrigerant across the Unit Cooler using two independent measuring systems.

C5.1.2 *Method 2, DX Calibrated Box.* The Refrigeration Capacity shall be determined concurrently by measuring the enthalpy change and the mass flow rate of the refrigerant across the Unit Cooler and the heat input to the calibrated box.

C5.2 Upon the completion of the steady state test, an off-cycle evaporator fan power test shall be conduct to measure the evaporator fan power consumption during a compressor-off period in accordance with C10 of this standard.

C5.3 Upon the completion of the steady state test for walk-in freezer systems, a mandatory defrost test shall be conducted to establish the energy input for a defrost cycle. An optional defrost test to establish the energy input for a defrost cycle and the time between defrost intervals for a frosted coil condition and an additional optional test to establish credit for an adaptive or demand defrost system may be elected after the mandatory defrost test.

C6. *Test Chambers Requirements.*

C6.1 The Unit Cooler and the Condensing Unit shall be installed in separate environment chambers with sufficient size to avoid airflow restrictions or recirculation such that:

a. No obstacle is positioned within a distance of $2\sqrt{AB}$ from the discharge of the Unit Cooler and the condensing unit, where A and B are the air inlet dimensions, in, per fan section of the Unit Cooler and the condensing unit.

b. All other distances correspond to the minimum requirements of the installation instructions provided by the manufacturer.

c. The minimum volume, ft^3, of the test chamber shall be 20 % of the airflow rate, ft^3/min produced by the Unit Cooler together with all auxiliary air moving devices that operate simultaneously with the Unit Cooler.

C6.2 Both environmental chambers shall be equipped with essential air handling units and controllers to process and maintain the enclosed air to any required test conditions.

C7. *General Test Conditions and Data Recording.*

C7.1 Refer to the standard rating conditions for a particular application listed in Section 5 of this standard. Test acceptance criteria listed in Table 2 in section 4 of this standard apply to both methods of test.

C7.2 Data that need to be recorded during the test are listed in Table C2.

Table C2. Data To be Recorded

	Units	Refrigerant Enthalpy Method	Calibrated Box Method
Date		X	X
Observer(s)		X	X
Barometric pressure	In.Hg	X	X
Times		X	X
Power input to condensing unit	W	X	X
Power input to unit cooler fan(s)	W	X	X
Applied Voltage to condensing unit	Volts	X	X
Applied Voltage to unit cooler fan	Volts	X	X
Total electrical power input to heater and auxiliary equipment	W		X
Frequency	Hz	X	X
Fan speed(s) if adjustable	Rpm	X	X
Air inlet relative humidity		X	X
Average dry-bulb temperature of air within the calibrated box	˚F		X
Average dry-bulb temperature of air within the temperature controlled enclosure	˚F		X
Dry-bulb temperatures of air entering unit cooler and condensing unit	˚F	X	X
Wet-bulb temperatures of air entering unit cooler and condensing unit	˚F	X	X

Table C2. Data To be Recorded

	Units	Refrigerant Enthalpy Method	Calibrated Box Method
Dry-bulb temperatures of air leaving unit cooler and condensing unit	˚F	X	
Wet-bulb temperatures of air leaving unit cooler and condensing unit	˚F	X	
Condensing pressure or temperature	psi/˚F	X	X
Evaporator pressure or temperature	psi/˚F	X	X
Pressure of subcooled refrigerant liquid entering the expansion valve	psi	X	X
Pressure of superheated refrigerant vapor leaving the Unit Cooler	psi	X	X
Pressure of refrigerant vapor at compressor suction	psi	X	X
Pressure of refrigerant vapor at compressor discharge	psi	X	X
Temperature of subcooled refrigerant liquid entering the expansion	˚F	X	X
Temperature of superheated refrigerant vapor leaving the Unit Cooler	˚F	X	X
Temperature of refrigerant vapor at compressor suction	˚F	X	X
Temperature of refrigerant vapor at compressor discharge	˚F	X	X
Mass flow rate of subcooled refrigerant liquid through M1	lb/h	X	X
Mass flow rate of subcooled refrigerant liquid through M2 or superheated refrigerant vapor through M2ALT	lb/h	X	X
		X	X
		X	X

FIGURE C1
METHOD 1: DX - DUAL INSTRUMENTATION

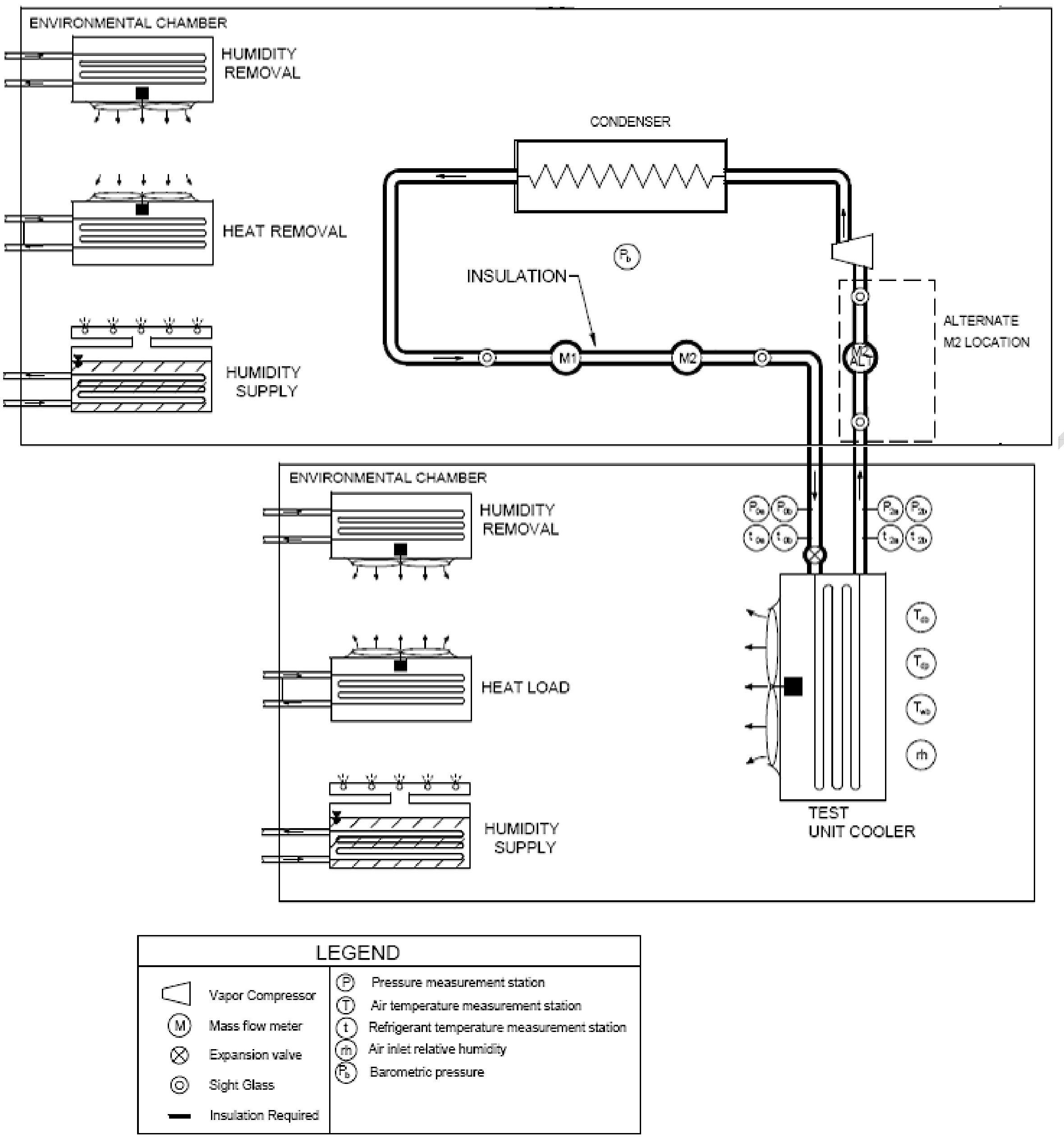
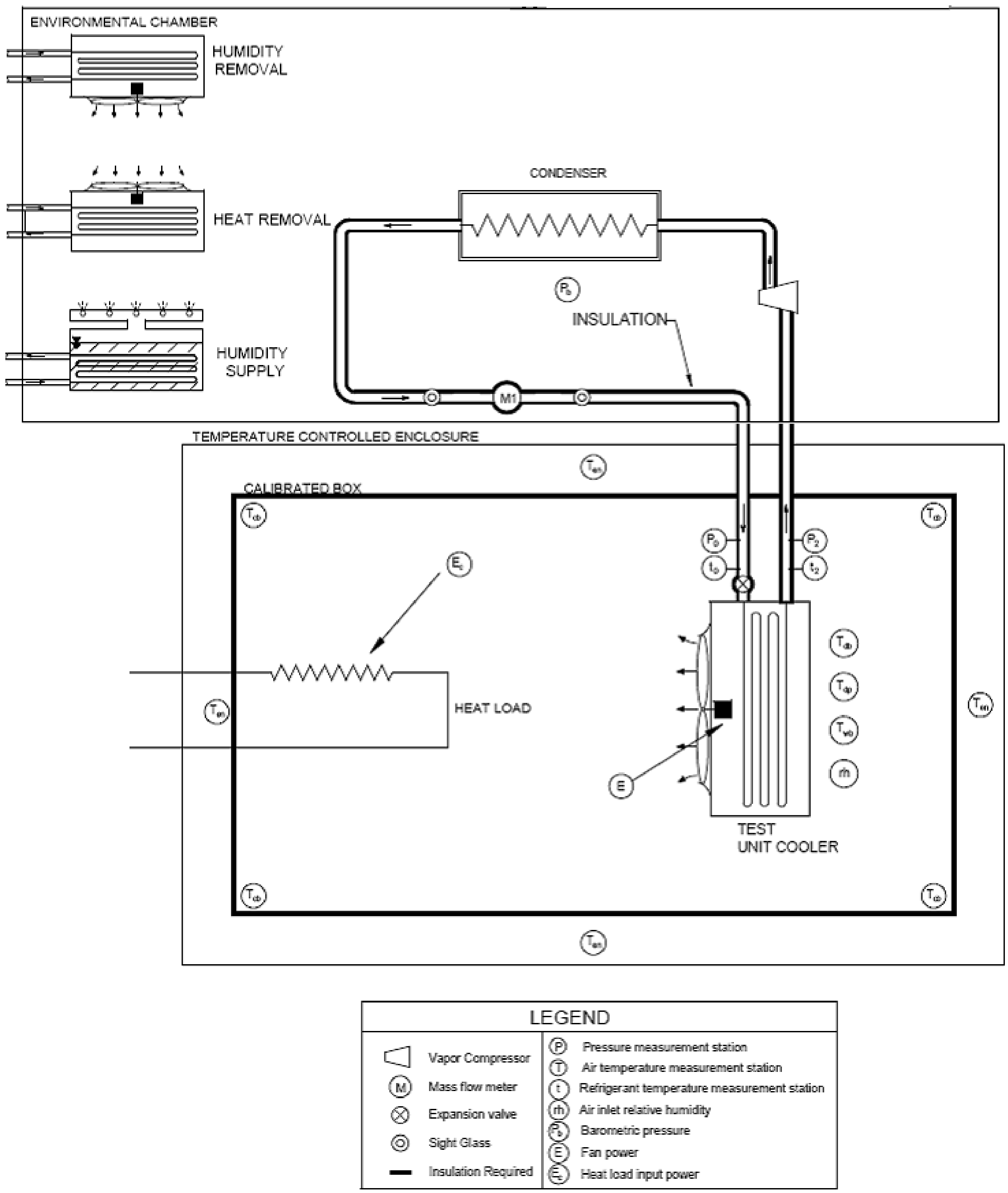


FIGURE C2

METHOD 2: DX - CALIBRATED BOX



C8. *DX Dual Instrumentation Test Procedure (Method 1: Refrigerant Enthalpy Method)*

C8.1 *General Description.* In this method, capacity is determined from the refrigerant enthalpy change and flow rate. Enthalpy changes are determined from measurements of entering and leaving pressures and temperatures of the refrigerant, and the flow rate is determined by a suitable flow meter in the liquid line. For cases where calibrated box method is also conducted, data used to calculate capacity as described in the refrigerant enthalpy method and the calibrated box method shall be collected over the same intervals. This method may be used for tests of equipment in which the refrigerant charge is not critical and where normal installation procedures involve the field connection of refrigerant lines. This method shall not be used for tests in which the refrigerant liquid leaving the flow meter is subcooled less than 3°F or for tests in which any instantaneous measurement of the superheat of the vapor leaving the evaporator coil is less than 5°F. If supplementary cooling in the liquid line is artificially introduced to ensure enough subcooling, the added cooling capacity shall be measured and deducted from the gross refrigeration capacity calculated in C8.5.2.

C8.2 *Measurements.* Refer to Section C3 for requirements of air-side and refrigerant-side measurements

C8.3 *Test Setup and Procedure*. Refer to Section C6, C7 and Figure C1 for specific test setup. The condensing unit and the unit cooler shall be connected by pipes with manufacturer' specified size. The pipe lines shall be insulated with a minimum total thermal resistance equivalent to ½" thick insulation having a flat-surface R-Value of 3.7 ft2-°F-hr/Btu per inch or greater . Flow meters need not be insulated but must not be in contact with the floor. The lengths of each of the connected liquid line and suction line shall be 25 feet, not including the requisite flow meters. Of this length, no more than 15 feet shall be in the conditioned space. In the case that there are multiple branches of piping, the maximum length of piping applies to each branch individually as opposed to the total length of the piping.

C8.4 *Data to be Measured and Recorded.* Refer to Table C2 in Section C7.2 for the required data that need to measured and recorded.

C8.5 *Refrigeration capacity Calculation.*

C8.5.1 The refrigerant-side gross capacities by independent measurement are calculated by

$$\dot{Q}_{ref,1}=\dot{m}_{ref,1}\cdot\left(h_{out}-h_{in}\right) \qquad \text{C1}$$

$$\dot{Q}_{ref,2}=\dot{m}_{ref,2}\cdot(h_{out}-h_{in}) \qquad \text{C2}$$

C8.5.2 Gross refrigeration capacity is calculated by

$$\dot{Q}_{ref}=\frac{\dot{Q}_{ref,1}+\dot{Q}_{ref,2}}{2} \qquad \text{C3}$$

C8.5.3 Allowable Cooling Capacity heat balance

$$-5\%\leq\frac{\dot{Q}_{ref,1}-\dot{Q}_{ref,2}}{\dot{Q}_{ref}}\times100\%\leq5\% \qquad \text{C4}$$

C8.5.4 The net refrigeration capacity is calculated by

$$\dot{q}_{ss}=\dot{Q}_{ref}-3.412\cdot\dot{EF}_{comp,on} \qquad \text{C5}$$

C9. *DX Calibrated Box Test Procedure (Method 2)*

C9.1 *Measurements.* Refer to Section C3 for requirements of air-side and refrigerant-side measurements.

C9.2 *Test Setup and Procedure*. Refer to Section C6, C7 and Figure C2 for specific test setup. The condensing unit and the unit cooler shall be connected by pipes with manufacturer' specified size. The pipe lines shall be well insulated. The lengths of the connected liquid line and suction line shall be 26 feet, respectively.

C9.2.1 *Apparatus Setup for Calibrated Box Calibration and Test*

C9.2.1.1 The calibrated box shall be installed in a temperature controlled enclosure in which the temperature can be maintained at a constant level.

C9.2.1.2 The temperature controlled enclosure shall be of a size that will provide clearances of not less than 18 in at all sides, top and bottom, except that clearance of any one surface may be reduced to not less than 5.5 in.

C9.2.1.3 In no case shall the heat leakage of the calibrated box exceed 30 % of the Gross Total Cooling Effect of the Unit Cooler under test. The ability to maintain a low temperature in the temperature controlled enclosure will reduce the heat leakage into the calibrated box and may extend its application range.

C9.2.1.4 Refrigerant lines within the calibrated box shall be well insulated to avoid appreciable heat loss or gain.

C9.2.1.5 Instruments for measuring the temperature around the outside of the calibrated box shall be located at the center of each wall, ceiling, and floor at a distance of 6 in from the calibrated box. Exception: in the case where a clearance around the outside of the calibrated box, as indicated above, is reduced to less than 18 in, the number of temperature measuring devices on the outside of that surface shall be increased to six, which shall be treated as a single temperature to be averaged with the temperature of each of the other five surfaces. When the clearance is reduced below 12 in (one surface only), the temperature measuring instruments shall be located midway between the outer wall of the calibrated box and the adjacent wall. The six temperature measuring instruments shall be located at the center of six rectangular sections of equal area.

C9.2.1.6 Heating means inside the calibrated box shall be shielded or installed in a manner to avoid radiation to the Unit Cooler, the temperature measuring instruments, and to the walls of the box. The heating means shall be constructed to avoid stratification of temperature, and suitable means shall be provided for distributing the temperature uniformly.

C9.2.1.7 The average air dry-bulb temperature in the calibrated box during Unit Cooler tests and calibrated box heat leakage tests shall be the average of eight temperatures measured at the corners of the box at a distance of 2 in to 4 in from the walls. The instruments shall be shielded from any cold or warm surfaces except that they shall not be shielded from the adjacent walls of the box. The Unit Cooler under test shall be mounted such that the temperature instruments are not in the direct air stream from the discharge of the Unit Cooler.

C9.2.2 *Calibration of the Calibrated Box.* A calibration test shall be made for the maximum and the minimum forced air movements expected in the use of the calibrated box. The calibration heat leakage shall be plotted as a straight line function of these two air quantities and the curve shall be used as calibration for the box.

C9.2.2.1 The heat input shall be adjusted to maintain an average box temperature not less than 25.0 ºF above the test enclosure temperature.

C9.2.2.2 The average dry-bulb temperature inside the calibrated box shall not vary more than 1.0 ºF over the course of the calibration test.

C9.2.2.3 A calibration test shall be the average of eleven consecutive hourly readings when the box has reached a steady-state temperature condition.

C9.2.2.4 The box temperature shall be the average of all readings after a steady-state temperature condition has been reached.

C9.2.2.5 The calibrated box has reached a steady-state temperature condition when:

1. The average box temperature is not less than 25 ºF above the test enclosure temperature.
2. Temperature variations do not exceed 5.0 ºF between temperature measuring stations.
3. Temperatures do not vary by more than 2 ºF at any one temperature- measuring station.

C9.3 *Data to be Measured and Recorded.* Refer to Table C2 in Section C7.2 for the required data that need to measured and recorded.

C9.4 *Refrigeration capacity Calculation.*

C9.4.1 The heat leakage coefficient of the calibrated box is calculated by

$$K_{cb}=\frac{3.412\cdot\dot{E}_c}{T_{en}-T_{cb}} \qquad \text{C6}$$

C9.4.2 For each Dry Rating Condition, calculate the air-side Gross Total Refrigeration Capacity:

$$\dot{Q}_{air}=K_{cb}\cdot\left(T_{en}-T_{cb}\right)+3.412\cdot(\dot{E}_c+\dot{EF}_{comp,on}) \qquad \text{C7}$$

C9.4.3 For each Dry Rating Condition, calculate the refrigerant-side Gross Total Refrigeration Capacity:

$$\dot{Q}_{ref}=\dot{m}_{ref}\cdot(h_{out}-h_{in}) \qquad \text{C8}$$

C9.4.4 Gross Total Refrigeration Capacity:

$$\dot{Q}_t=\frac{\dot{Q}_{air}+\dot{Q}_{ref}}{2} \qquad \text{C9}$$

C9.4.5 Allowable Refrigeration Capacity heat balance

$$-5\%\leq\frac{\dot{Q}_{air}-\dot{Q}_{ref}}{\dot{Q}_t}\times100\%\leq5\% \qquad \text{C10}$$

C10. *Off-cycle evaporator fan test.* Upon the completion of the steady state test for walk-in systems, the compressors of the walk-in systems shall be turned off. The unit coolers fans' power consumption shall be measured in accordance with the requirements in Section C 3.5. Off-cycle fan power shall be equal to on-cycle fan power unless evaporator fans are controlled by a qualifying control. Qualifying evaporator fan controls shall have a user adjustable method of destratifying air during the off-cycle including but not limited to: adjustable fan speed control or periodic "stir cycles." Controls shall be adjusted so that the greater of a 50% duty cycle or the manufacturer default is used for measuring off-cycle fan energy. For variable speed controls, the greater of 50% fan speed or the manufacturer's default fan speed shall be used for measuring off-cycle fan energy. When a cyclic control is used, at least three full "stir cycles" are measured.

C11. *Defrost test (freezer only).* The defrost test consists of a mandatory test to establish the energy input for a defrost cycle for a dry coil condition, an optional test to establish the energy input for a defrost cycle and the time between defrost intervals for a frosted coil condition, and an additional optional test to establish credit for an adaptive or demand defrost system. Refer to the standard rating conditions for defrost test conditions listed in Section 5 of this standard.

C11.1 *Dry coil condition (mandatory test).* During the test, no adjustments are to be made to the defrost settings. Following a defrost, the unit cooler shall be operated at the dry coil conditions specified in Section 5 until stable, and then a defrost shall be initiated either through manual override or by the automatic controls. The energy input and duration of the dry coil condition (DF_d in W.h) shall be measured from the time the refrigeration system stops until it restarts again.

C11.2 *Frosted coil condition (calculation methodology and optional test).*

C11.2.1 In lieu of testing, the frosted coil energy input and duration of the frosted coil condition (DF_f in W.h) shall be the product of 1.05 times the energy input of the dry coil condition (DF_d in W.h) obtained from the test conducted in C11.1.

C11.2.2 *Optional test.* Upon completion of the test conducted in C11.1,the room conditions shall then be set to the frost load conditions specified below, the defrost frequency set to either the recommendation in the installation instructions or if not specified, set to 4 defrosts per day (6 hour interval) and the unit operated until the unit self-initiates a defrost cycle. The energy input and duration of the frosted coil condition (DF_f in W.h) shall be measured from the time the refrigeration system stops until it restarts again. The drain water shall be collected, weighed (m_w) and the temperature (T_w) recorded. This would be used to reduce the defrost energy contribution to the box load calculation. The total energy would still be used in the power side of the AWEF calculation. C11.2.2.3 *Frost Load Condition.* The frost load shall occur through the infiltration of air at 75.2°F dry-bulb / 64.4°F wet-bulb into the walk-in freezer. Infiltration shall arise from the introduction of a constant rate of air flow at the above stated conditions, into the box during the test period. The flow rate shall be determined by the following equation, and be measured in accordance with ASHRAE Standard 41.2.

$\dot{V}_{air} = k_{13} \cdot \dot{q}_{ss}\,(95°F) + k_{14}$ for the case that the condensing units is located outdoor, or C11

$\dot{V}_{air} = k_{13} \cdot \dot{q}_{ss}\,(90°F) + k_{14}$ for the case that condensing unit is located within a conditioned space. C12

Where

$\dot{V}_{air}$ – infiltration air volumetric flow rate, cfm
k_{13} = 0.0001, cfm.h/btu
k_{14} = 3.49, cfm

C11.2.4 The number of defrosts per day (N_{DF}) shall be equal to the defrost frequency recommended in the installation instructions for the unit; if no defrost frequency is specified, the number of defrosts per day shall be set to 4.

C11.3 Adaptive Defrost *Optional test.*

C11.3.1 Method one: If the system has an adaptive or demand defrost system, an optional test can be run at the dry coil conditions to establish the maximum time interval allowed between dry coil defrosts and at the frosted coil condition to establish the maximum time interval allowed between frosted coil defrosts. The defrost frequency, if specified, shall be set to the recommendation in the installation instructions. The unit shall be operated until one of the following occurs: 1) the unit self-initiates a defrost cycle, 2) 12 hours of elapsed run time is reached, or 3) the unit cooler leaving vapor pressure decreases the equivalent of 5°F of saturation temperature drop to indicate a sufficiently blocked or frosted coil. The measured time between successive defrosts for dry coil condition shall be averaged with the time between successive defrosts for the frosted load condition, and this average time interval used to calculate the number of defrost per day (N_{DF}).

C11.3.2 Method two: If the system has an adaptive or demand defrost system, the number of defrosts per day (N_{DF}) may be calculated by taking the average of 1 and the number of defrosts per day that would occur under frosted load conditions.

C11.4 *Defrost adequacy test.* The test shall verify that any defrost setting and arrangement is adequate to melt all frost and ice from the coil and drain it out of the Walk-In. At the conclusion of the frosted load defrost test, the unit cooler shall continue to operate at the same stabilized condition for a period of not less than two additional frosted load defrost cycles, or 24 hours, whichever comes first. Upon conclusion of this test, all drain pans, fans and coils shall be checked for residual ice or frost that might continue to accumulate over time. If ice or frost is found, then an additional 48-hour test period shall be performed without changing the test or defrost settings. At the conclusion of the 48-hour test period, another check for residual ice or frost shall be conducted. If the accumulation has stabilized and not increased, then the test data are acceptable. If the ice or frost accumulation has increased then the test is unacceptable for inclusion in the test performance data and this occurrence shall be reported. For unit coolers utilizing electric defrost, the average of the dry coil and frost load defrost energy shall be used for calculating the value of DF, the daily average defrost energy required for the refrigeration system.

$$DF = \frac{DF_d + DF_f}{2} \cdot N_{DF} \qquad C13$$

For unit coolers utilizing hot gas defrost and connected to a multiplex system, DF is equal to zero. For unit coolers utilizing hot gas defrost and connected to a dedicated condensing system, DF is calculated as follows.

$$DF=0.5\cdot\frac{Q_{DF}}{3.412}\cdot N_{DF} \qquad C14$$

For unit coolers utilizing electric defrost the daily contribution of the load attributed to defrost shall be calculated using an average of the dry coil and net frost load defrost energy.

$$Q_{DF}=\frac{3.412\cdot DF_d+\{3.412\cdot DF_f-m_w[c_{pi}(32+10)+H_{fus}+c_{pw}(T_w-32)]\}}{2}\cdot N_{DF} \qquad C15$$

If the optional frost load defrost test is not performed for units utilizing electric defrost, Q_{DF} shall be calculated as follows.

$$Q_{DF}=0.95\cdot 3.412\cdot DF \qquad C16$$

For unit coolers utilizing hot gas defrost the daily contribution of the load attributed to defrost shall be calculated utilizing the unit cooler's capacity. The number of defrosts per day for this calculation shall be set to the number recommended in the installation instructions for the unit (or if no instructions, shall be set to 4) for units without adaptive defrost and 2.5 for units with adaptive defrost.

$$Q_{DF}=0.18\cdot Q_{ref}\cdot N_{DF} \qquad C17$$

Where

Q_{ref} = Gross refrigeration capacity as measured at the high ambient condition, Btu/h

Consequently, the defrost power consumption contributed to the total system power consumption and to the box load are calculated by the following equations respectively.

$$\dot{DF}=\frac{DF}{24} \qquad C18$$

$$\dot{Q}_{DF}=\frac{Q_{DF}}{24} \qquad C19$$

C12. *Method of testing condensing units for walk-in cooler and freezer systems where condenser is sold separately.* The purpose of this section is to provide a testing method for stand-alone condensing units that provide sufficient data to allow for Standard Rating performance and AWEF determination for a reference unit cooler. The reference unit cooler shall have the following values:

Table C3. Unit Cooler Nominal Values for Condensing Unit Energy Calculations

Description	Cooler	Freezer
Saturated Suction Temperature (°F)	25	-20
On-cycle evaporator fan power, per Btu/h of gross capacity at ambient condition (W-h/Btu)	0.016	0.016
Off-cycle evaporator fan power (W)	0.2 x on-cycle evaporator fan power	
Electric defrost energy per cycle, per Btu/h of gross capacity (W-h/cycle per Btu/h)	0	0.12
Number of cycles per day	N/A	4
Daily electric defrost contribution (Btu)	0.95 x daily defrost energy use x 3.413	

The suction condition test points in Table 11, 12, 13 and 14 in Section 5.1 of this standard must be run and the results reported, depending upon applications. The AWEF shall be calculated as described in Section 7.10. The condensing units shall have proper refrigerant charge to meet the subcooling requirement according to the equipment specification during the test period. The specified amount of subcooling shall be reported as a part of the test results. Refer to ASHRAE Standard 23 for the test methods, requirements and procedures..

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C13. *Method of testing unit coolers for walk-in cooler and freezer systems for use in mix-match system ratings for equipment sold together as a system.* The purpose of this section is to provide a testing method for stand-alone unit coolers that provide sufficient data to allow for standard rating performance and AWEF determination for mix-match combinations with available walk-in condensing units. This rating only applies for products that are sold together.

All A and B saturation condition test points in Table 15 and 16 in Section 5.1 of this standard must be run and the results reported, depending upon applications. The refrigerant superheat at the unit coolers outlet shall meet the requirement according to the equipment specification during the test period. The specified amount of superheat shall be reported as a part of the test results. Refer AHRI 420 for the test methods, requirements and procedures.

C14. *Method of testing unit coolers for walk-in cooler and freezer systems where the unit cooler is sold separately.* The purpose of this section is to provide a testing method for stand-alone unit coolers that provide sufficient data to allow for standard rating performance and AWEF determination when combined with an unknown refrigeration system. These unit coolers are sold separately and may be combined with a remote condensing unit or a Parallel Rack System. The AWEF shall be calculated as described in Section 7.9.

All saturation condition test points in Table 15 and 16 of Section 5.1 in this standard must be run and the results reported, depending upon applications.

Appendix D. Weather Data in Region IV – Normative

D.1 The temperature bins and corresponding bin hours applied in the AWEF shall be based on the TMY-3 weather data of Kansans City, Missouri, which corresponds closely to the the 'use cycle' climate parameters prescribed in other DOE appliance standards (10 CFR 430.23). The temperature and bin hours are listed in Table D1.

Table D1. Bin temperatures and corresponding Bin hours for AWEF Calculation

Bin Temperature [˚F]	Bin hours [h]
100.4	9
95	74
89.6	257
84.2	416
78.8	630
73.4	898
68	737
62.6	943
57.2	628
51.8	590
46.4	677
41	576
35.6	646
30.2	534

24.8	322
19.4	305
14	246
8.6	189
3.2	78
-2.2	5