2003
STANDARD for

UNITARY AIR-CONDITIONING AND AIR-SOURCE HEAT PUMP EQUIPMENT

AIR-CONDITIONING &
REFRIGERATION
INSTITUTE

Standard 210/240

4100 N. FAIRFAX DR.,STE. 200 • ARLINGTON, VIRGINIA 22203

IMPORTANT

SAFETY DISCLAIMER

ARI 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.

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

ARI CERTIFICATION PROGRAM PROVISIONS

Scope of the Certification Program

The Certification Program includes all Unitary Air-Conditioning and Air-Source Unitary Heat Pump equipment rated below 65,000 Btu/h [19,000 W] at ARI Standard Rating Conditions (Cooling).

Certified Ratings

The following Certification Program ratings are verified by test:

Unitary Air-Conditioners

A. Air-cooled under 65,000 Btu/h [19,000 W]
   1. ARI Standard Rating Cooling Capacity, Btu/h [W]
   2. Seasonal Energy Efficiency Ratio, SEER, Btu/(W·h)

B. Water-cooled and evaporative-cooled under 65,000 Btu/h [19,000 W]
   1. ARI Standard Rating Cooling Capacity, Btu/h [W]
   2. Energy Efficiency Ratio, EER, Btu/(W·h)

Air-Source Unitary Heat Pumps

Air-cooled under 65,000 Btu/h [19,000 W]
1. ARI Standard Rating Cooling Capacity, Btu/h [W]
2. Seasonal Energy Efficiency Ratio, SEER, Btu/(W·h)
3. High Temperature Heating Standard Rating Capacity, Btu/h [W]
4. Region IV Heating Seasonal Performance Factor, HSPF, Minimum Design Heating Requirement, Btu/(W·h)

Conformance to the requirements of the Maximum Operating Conditions Test, Voltage Tolerance Test, Low-Temperature Operation Test (Cooling), Insulation Effectiveness Test (Cooling), and Condensate Disposal Test (Cooling), as outlined in Section 8, are also verified by test.

Note:

This standard supersedes ARI Standard 210/240-94.

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UNITARY AIR-CONDITIONING AND AIR-SOURCE HEAT PUMP EQUIPMENT

Section 1. Purpose

1.1 *Purpose.* The purpose of this standard is to establish, for Unitary Air-Conditioners and Air-Source Unitary Heat Pumps: definitions; classifications; 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, engineers, 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 factory-made Unitary Air-Conditioners and Air-Source Unitary Heat Pumps as defined in Section 3.

2.1.1 *Energy Source.* This standard applies only to electrically operated, vapor compression refrigeration systems.

2.2 *Exclusions.* This standard does not apply to the rating and testing of individual assemblies, such as condensing units or coils, for separate use.

2.2.1 This standard does not apply to heat operated air-conditioning/heat pump equipment, or to packaged terminal air-conditioners/heat pumps, or to room air-conditioners/heat pumps.

2.2.2 This standard does not apply to Unitary Air-Conditioners as defined in ARI Standard 340/360 with capacities of 65,000 Btu/h [19,000 W] or greater.

2.2.3 This standard does not apply to Air-Source Unitary Heat Pumps as defined in ARI Standard 340/360 with cooling capacities of 65,000 Btu/h [19,000 W] or greater, or to water-source heat pumps, to ground water-source heat pumps, and to ground source closed-loop heat pumps.

2.2.4 This standard does not include water heating heat pumps.

2.2.5 This standard does not apply to rating units equipped with desuperheater/water heating devices in operation.

Section 3. Definitions

All terms in this document shall follow the standard industry definitions in the current edition of *ASHRAE Terminology of Heating, Ventilation, Air- Conditioning and Refrigeration*, unless otherwise defined in this section.

Note: See Appendix C for definitions that apply to the testing and calculation procedures required by Appendix C.

3.1 *Air-Source Unitary Heat Pump.* One or more factory-made assemblies which normally include an indoor conditioning coil(s), compressor(s), and outdoor coil(s), including means to provide a heating function. When such equipment is provided in more than one assembly, the separated assemblies shall be designed to be used together, and the requirements of rating outlined in the standard are based upon the use of matched assemblies.

3.1.1 *Functions.* They shall provide the function of air heating with controlled temperature, and may include the functions of air-cooling, air-circulating, air-cleaning, dehumidifying or humidifying.

3.2 *Degradation Coefficient (C_D).* The measure of the efficiency loss due to the cycling of the units as determined in Appendices C and D.

3.3 *Design Heating Requirement (DHR).* This is the amount of heating required to maintain a given indoor temperature at a particular outdoor design temperature.

3.4 *Energy Efficiency Ratio (EER).* A ratio of the cooling capacity in Btu/h to the power input value in watts at any given set of Rating Conditions expressed in Btu/(W·h).

3.4.1 *Standard Energy Efficiency Ratio.* A ratio of the capacity to power input value obtained at Standard Rating Conditions.

3.5 *Heating Seasonal Performance Factor (HSPF).* The total heating output of a heat pump, including supplementary electric heat necessary to achieve building heating requirements during its normal annual usage period for heating divided by the total electric power during the same period, as determined in Appendices C (Section C4.2) and D, expressed in Btu/(W·h).

3.6 *Integrated Part-Load Value (IPLV).* A single number part-load efficiency figure of merit calculated per the method described in this standard.

3.7 *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 capacity and type (identification) produced by the same manufacturer. As used herein, 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.

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

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

3.8 *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.

3.8.1 *Standard Rating Conditions.* Rating Conditions used as the basis of comparison for performance characteristics.

3.9 *Seasonal Energy Efficiency Ratio (SEER).* The total cooling of a central air-conditioner during its normal usage period for cooling (not to exceed 12 months) divided by the total electric energy input during the same period as determined in Appendices C (Section C4.1) and D, expressed in Btu/(W·h).

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

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

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

3.11 *Standard Air.* Air weighing 0.075 lb/ft^3 [1.2 kg/m^3] which approximates dry air at 70°F [21°C] and at a barometric pressure of 29.92 in Hg [101.3 kPa].

3.12 *Unitary Air-Conditioner.* One or more factory-made assemblies which normally include an evaporator or cooling coil(s), compressor(s), and condenser(s). Where such equipment is provided in more than one assembly, the separated assemblies are to be designed to be used together, and the requirements of rating outlined in this standard are based upon the use of these assemblies in operation together.

3.12.1 *Functions.* Either alone or in combination with a heating plant, the functions are to provide air-circulation, air-cleaning, cooling with controlled temperature and dehumidification, and may optionally include the function of heating and/or humidifying.

Section 4. Classifications

Equipment covered within the scope of this standard shall be classified as shown in Tables 1 and 2.

Section 5. Test Requirements

All Standard Ratings shall be verified by tests conducted in accordance with ANSI/ASHRAE Standard 37 and with the test methods and procedures as described in this standard and its appendices.

Air-cooled units shall be tested in accordance with Appendices C and D. Water-cooled and evaporative-cooled units shall be tested in accordance with ANSI/ASHRAE Standard 37.

Section 6. Rating Requirements

6.1 *Standard Ratings.* Standard Ratings shall be established at the Standard Rating Conditions specified in 6.1.3.

Air-cooled units shall be rated at conditions specified in Table 3 or Table 4.

Water-cooled and evaporative-cooled units shall be rated at conditions specified in Table 5.

Standard Ratings relating to cooling or heating capacities shall be net values, including the effects of circulating-fan heat, but not including supplementary heat. Power input shall be the total power input to the compressor(s) and fan(s), plus controls and other items required as part of the system for normal operation.

Standard Ratings of units which do not have indoor air-circulating fans furnished as part of the model, i.e., split systems with indoor coil alone, shall be established by subtracting from the total cooling capacity 1,250 Btu/h per 1,000 cfm [775 W/m^3/s], and by adding the same amount to the heating capacity. Total power input for both heating and cooling shall be increased by 365 W per 1,000 cfm [226 W/m^3/s] of indoor air circulated.

Standard Ratings of water-cooled units shall include a total allowance for cooling tower fan motor and circulating water pump motor power inputs to be added in the amount of 10.0 W per 1,000 Btu/h [34.1 W per 1,000 W] cooling capacity.

6.1.1 *Values of Standard Capacity Ratings.* These ratings shall be expressed only in terms of Btu/h [W] as shown:

Capacity Ratings, Btu/h [W]	Multiples, Btu/h [W]
< 20,000 [5,900]	100 [30]
≥ 20,000 and < 38,000 [5,900 up to 11,000]	200 [60]
≥ 38,000 and < 65,000 [11,000 up to 19,000]	500 [150]

6.1.2 *Values of Measures of Energy Efficiency.* Standard measures of energy efficiency, whenever published, shall be expressed in multiples of the nearest 0.05 Btu/(W·h) for EER, SEER and HSPF. and in multiples of 0.1 for IPLV.

6.1.3 *Standard Rating Tests.* Tables 3, 4 and 5 indicate the test and test conditions which are required to determine values of standard capacity ratings and values of measures of energy efficiency.

6.1.3.1 *Assigned Degradation Factor.* In lieu of conducting C and D tests or the heating cycling test (as shown in Table 3), an assigned value of 0.25 may be used for either the cooling or heating Degradation Coefficient, C_D, or both. For units with two compressor speeds, two compressors or cylinder unloading, if the assigned C_D is used for one cooling mode, it must be used for both cooling modes. If the assigned C_D is used for one heating mode, it must be used for both heating modes.

6.1.3.2 *Electrical Conditions.* Standard Rating tests shall be performed at the nameplate rated voltage(s) and frequency.

For air-cooled equipment which is rated with 208-230V 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.

6.1.3.3 *Indoor-Coil Airflow Rate.* All Standard Ratings shall be determined at an indoor-coil airflow rate as outlined below. All airflow rates shall be expressed in terms of Standard Air.

a. Equipment with indoor fans intended for use with field installed duct systems shall be rated at the indoor-coil airflow rate (not to exceed 37.5 SCFM per 1,000 Btu/h [0.06 m^3/s per 1,000 W] of rated capacity) delivered when operating against the minimum external pressure specified in 6.1.3.6 or at a lower indoor-coil airflow rate if so specified by the manufacturer.

b. Equipment with indoor fans not intended for use with field installed duct systems (free discharge) shall be rated at the indoor-coil airflow rate delivered when operating at 0 in H_2O [0 Pa] external pressure as specified by the manufacturer.

c. Equipment which does not incorporate an indoor fan, but is rated in combination with a device employing a fan shall be rated as described under 6.1.3.3 a. For equipment of this class which is rated for general use to be applied to a variety of heating units, the indoor-coil airflow rate shall be specified by the manufacturer in Standard Ratings, not to exceed 37.5 SCFM/1,000 Btu/h [0.06 m^3/s per 1,000 W] of rated capacity or the airflow rate obtained through the indoor coil assembly when the pressure drop across the indoor coil assembly and the recommended enclosures and attachment means is not greater than 0.30 in H_2O [75 Pa], whichever is less.

Table 1. Classification of Unitary Air-Conditioners

Types of Unitary Air-Conditioners		
Designation	ARI Type [1,2]	Arrangement
Single Package	SP-A SP-E SP-W	FAN COMP EVAP COND
Refrigeration Chassis	RCH-A RCH-E RCH-W	COMP EVAP COND
Year-Round Single Package	SPY-A SPY-E SPY-W	FAN HEAT COMP EVAP COND
Remote Condenser	RC-A RC-E RC-W	FAN EVAP COND COMP
Year-Round Remote Condenser	RCY-A RCY-E RCY-W	FAN EVAP COND HEAT COMP
Condensing Unit, Coil Alone	RCU-A-C RCU-E-C RCU-W-C	EVAP COND COMP
Condensing Unit, Coil And Blower	RCU-A-CB RCU-E-CB RCU-W-CB	FAN COND EVAP COMP
Year-Round Condensing Unit, Coil and Blower	RCUY-A-CB RCUY-E-CB RCUY-W-CB	FAN EVAP COND HEAT COMP

Notes:

1 A suffix of "-O" following any of the above classifications indicates equipment not intended for use with field-installed duct systems (6.1.3.3b).

2 A suffix of "-A" indicates air-cooled condenser, "-E" indicates evaporative-cooled condenser and "-W" indicates water-cooled condenser.

Table 2. Classification of Air-Source Unitary Heat Pumps

Types of Air-Source Unitary Heat Pumps

Designation	ARI Type[1]		Arrangement
	Heating and Cooling	Heating Only	
Single Package	HSP-A	HOSP-A	FAN | COMP INDOOR COIL | OUTDOOR COIL
Remote Outdoor Coil	HRC-A-CB	HORC-A-CB	FAN INDOOR COIL COMP OUTDOOR COIL
Remote Outdoor Coil With No Indoor Fan	HRC-A-C	HORC-A-C	INDOOR COIL COMP OUTDOOR COIL
Split System	HRCU-A-CB	HORCU-A-CB	FAN INDOOR COIL COMP OUTDOOR COIL
Split System With No Indoor Fan	HRCU-A-C	HORCU-A-C	INDOOR COIL COMP OUTDOOR COIL

Note:

[1] A suffix of "-O" following any of the above classifications indicates equipment not intended for use with field-installed duct systems (6.1.3.3 b.).

Table 3. Conditions for Standard Rating Tests and Operating Requirement Tests for Air-cooled Equipment Using Appendix C

	TEST	INDOOR UNIT				OUTDOOR UNIT			
		Air Entering				Air Entering			
		Dry-Bulb °F	Dry-Bulb °C	Wet-Bulb °F	Wet-Bulb °C	Dry-Bulb °F	Dry-Bulb °C	Wet-Bulb °F	Wet-Bulb °C
COOLING	Standard Rating Conditions "A" Cooling Steady State [1]	80.0	26.7	67.0	19.4	95.0	35.0	75.0 [2]	23.9
COOLING	"B" Cooling Steady State	80.0	26.7	67.0	19.4	82.0	27.8	65.0 [2]	18.3
COOLING	"C" Cooling Steady State Dry Coil	80.0	26.7	57.0 [5]	13.9	82.0	27.8	65.0 [2]	18.3
COOLING	"D" Cooling Cyclic Dry Coil	80.0	26.7	57.0 [5]	13.9	82.0	27.8	65.0 [2]	18.3
COOLING	Low Temperature Operation Cooling	67.0	19.4	57.0	13.9	67.0	19.4	57.0 [2]	13.9
COOLING	Insulation Efficiency	80.0	26.7	75.0	23.9	80.0	26.7	75.0 [2]	23.9
COOLING	Condensate Disposal	80.0	26.7	75.0	23.9	80.0	26.7	75.0 [2]	23.9
COOLING	Maximum Operating Conditions	80.0	26.7	67.0	19.4	115.0	46.1	75.0 [2]	23.9
HEATING	Standard Rating Conditions High Temperature Heating [3] Steady State	70.0	21.1	60.0 (max)	15.6	47.0	8.3	43.0	6.1
HEATING	High Temperature Heating Cyclic	70.0	21.1	60.0 (max)	15.6	47.0	8.3	43.0	6.1
HEATING	High Temperature Heating[4] Steady State	70.0	21.1	60.0 (max)	15.6	62.0	16.7	56.5	13.6
HEATING	Low Temperature Heating Steady State	70.0	21.1	60.0 (max)	15.6	17.0	-8.3	15.0	-9.4
HEATING	Frost Accumulation	70.0	21.1	60.0 (max)	15.6	35.0	1.7	33.0	0.6
HEATING	Maximum Operating Conditions	80.0	26.7	-	-	75.0	23.9	65.0	18.3

Notes:
1 Same conditions used for Voltage Tolerance Tests.
2 The wet-bulb temperature condition is not required when testing air-cooled condensers which do not evaporate condensate.
3 Same conditions used for Voltage Tolerance Tests (Heating-only units).
4 For two speed, two compressor or units with compressor unloading capability.
5 Wet-bulb temperature sufficiently low that no condensate forms on evaporator.

Table 4. Conditions for Standard Rating Tests for Air-cooled Variable Speed Equipment Meeting the Requirements of Appendix C

TEST	INDOOR COIL AIR ENTERING				OUTDOOR COIL AIR ENTERING			
	Dry-Bulb		Wet-Bulb		Dry Bulb		Wet Bulb	
	°F	°C	°F	°C	°F	°C	°F	°C
"A" Cooling Steady State At Maximum (k=2) Compressor Speed	80.0	26.7	67.0	19.4	95.0	35.0	75.0 [1]	23.9
"B-2" Cooling Steady State At Maximum (k=2) Compressor Speed	80.0	26.7	67.0	19.4	82.0	27.8	65.0 [1]	18.3
"B-1" Cooling Steady State At Minimum (k=1) Compressor Speed	80.0	26.7	67.0	19.4	82.0	27.8	65.0 [1]	18.3
Low Ambient Cooling Steady State At Minimum (k=1) Compressor Speed	80.0	26.7	67.0	19.4	67.0	19.4	53.5 [1]	11.9
Dry Coil Cooling Steady State At Minimum (k=1) Compressor Speed	80.0	26.7	57.0 [4]	13.9	67.0	19.4	53.5 [1]	11.9
Cyclic Cooling Dry Coil [4] At Minimum (k=1) Compressor Speed	80.0	26.7	57.0 [4]	13.9	67.0	19.4	53.5 [1]	11.9
Intermediate Cooling Steady State At Intermediate (k=i) Compressor Speed	80.0	26.7	67.0	19.4	87.0	30.6	69.0[1]	20.6
Standard Rating-Heating At Nominal [2] (k=n) Compressor Speed	70.0	21.1	60.0 (max)	15.6	47.0	8.3	43.0	6.1
Max Temperature Heating At Minimum (k=1)Compressor Speed	70.0	21.1	60.0	15.6	62.0	16.7	56.5	13.6
Cyclic Heating [4] At Minimum (k=1) Compressor Speed	70.0	21.1	60.0	15.6	62.0	16.7	56.5	13.6
High Temperature Heating At Maximum (k=2) Compressor Speed	70.0	21.1	60.0	15.6	47.0	8.3	43.0	6.1
High Temperature Heating At Minimum (k=1) Compressor Speed	70.0	21.1	60.0	15.6	47.0	8.3	43.0	6.1
Frost Accumulation [3] At Maximum (k=2) and/or Intermediate (k=i) Compressor Speed	70.0	21.1	60.0	15.6	35.0	1.7	33.0	0.6
Low Temperature Heating At Maximum (k=2) Compressor Speed	70.0	21.1	60.0	15.6	17.0	-8.3	15.0	-9.4

All tests are performed at the outdoor fan speed and indoor blower speed intended for normal operation.
k = Compressor speed
Notes:
1 Not maintained if no condensate rejected to outdoor coil.
2 Optional test used to determine the DHR. The nominal speed is the lesser of the cooling and heating maximum speeds.
3 Optional equations may be used in lieu of the maximum speed test. The intermediate speed is the same as the cooling intermediate speed.
4 Wet-bulb temperature sufficiently low that no condensate forms on evaporator.

Table 5. Conditions for Standard Rating Tests and Operating Requirement Tests for Water-cooled and Evaporative-cooled Equipment Using ASHRAE Standard 37

TEST		INDOOR SECTION				OUTDOOR SECTION									
		Air Entering				Evaporative-cooled						Water-cooled			
						Air Entering						Condenser Inlet		Condenser Outlet	
		Dry-Bulb		Wet-Bulb		Dry-Bulb		Wet-Bulb		Make-up Water [3]					
		°F	°C	°F	°C	°F	°C	°F	°C	°F	°C	°F	°C	°F	°C
COOLING	Standard Rating Conditions Cooling [1]	80.0	26.7	67.0	19.4	95.0	35.0	75.0	23.9	85.0	29.4	85.0	29.4	95.0	35.0
COOLING	Low Temperature Operating Cooling	67.0	19.4	57.0	13.9	67.0	19.4	57.0	13.9	67.0	19.4	-	-	70.0	21.1
COOLING	Insulation Efficiency	80.0	26.7	75.0	23.9	80.0	26.7	75.0	23.9	85.0	29.4	-	-	80.0	26.7
COOLING	Condensate Disposal	80.0	26.7	75.0	23.9	80.0	26.7	75.0	23.9	85.0	29.4	-	-	80.0	26.7
COOLING	Maximum Operating Conditions	80.0	26.7	67.0	19.4	100.0	37.8	80.0	26.7	90.0	32.2	90.0	32.2	100.0	37.8
COOLING	Part-Load Conditions (IPLV)	80.0	26.7	67.0	19.4	80.0	26.7	67.0	19.4	77.0	25.0	75.0[2]	23.9	-	-

Notes:
1 Same conditions used for Voltage Tolerance Tests
2 Water flow rate as determined from Standard Rating Conditions.
3 Water in basin shall not overflow.

Indoor-coil airflow rates and pressures as referred to herein apply to the airflow rate experienced when the unit is cooling and dehumidifying under the conditions specified in this section. This airflow rate, except as noted in 6.1.3.3 b and 8.4, shall be employed in all other tests prescribed herein without regard to resulting external static pressure. Heating only units shall use the airflow rate experienced when the unit is operating under the High Temperature Heating Standard Rating Conditions Test.

6.1.3.4 *Outdoor-Coil Airflow Rate.* All Standard Ratings shall be determined at the outdoor-coil airflow rate specified by the manufacturer where the fan drive is adjustable. Where the fan drive is non-adjustable, they shall be determined at the outdoor-coil airflow rate inherent in the equipment when operated with all of the resistance elements associated with inlets, louvers, and any ductwork and attachments considered by the manufacturer as normal installation practice. Once established, the outdoor coil air circuit of the equipment shall remain unchanged throughout all tests prescribed herein.

6.1.3.5 *Requirements for Separated Assemblies.* All Standard Ratings for equipment in which the outdoor section is separated from the indoor section, as in Types RC, RCY, RCU, RCUY, HRC, HORC, HRCU and HORCU (shown in Section 4), shall be determined with at least 25 ft [7.6 m] of interconnection tubing on each line of the size recommended by the manufacturer. Such equipment in which the interconnection tubing is furnished as an integral part of the machine not recommended for cutting to length shall be tested with the complete length of tubing furnished, or with 25 ft [7.6 m] of tubing, whichever is greater. At least 10 ft [3.0 m] of the interconnection tubing shall be exposed to the outside conditions. The line sizes, insulation, and details of installation shall be in accordance with the manufacturer's published recommendation.

6.1.3.6 *Minimum External Pressure.* Indoor air-moving equipment intended for use with field installed duct systems shall be designed to operate against, and tested at not less than, the minimum external pressure shown in Table 6 when delivering the rated capacity and airflow rate specified in 6.1.3.3.

Indoor air-moving equipment not intended for use with field installed duct systems (free discharge) shall be tested at 0 in H_2O [0 Pa] external pressure.

Table 6. Minimum External Pressure

Standard Capacity Ratings [1]		Minimum External Resistance	
MBtu/h	kW	in H_2O	Pa
≤ 28	≤ 8.2	0.10	25
> 28 and ≤ 42	> 8.2 and ≤ 12.4	0.15	37
> 42 and < 65	> 12.4 and ≤ 19.0	0.20	50

[1] Cooling capacity for units with cooling function; High Temperature Heating Capacity for heating-only units

Interpreting this requirement, it is understood that the most restrictive filters, supplementary heating coils, and other equipment specified as part of the unit be in place and that the net external pressure specified above is available.

6.1.3.7 *Moisture Removal Determination.* Indoor air moisture removed shall be determined at Standard Rating Conditions (cooling) for units tested in accordance with both ANSI/ASHRAE Standard 37 and Appendix C. The expression of the removal rate shall be based upon the net cooling capacity, including an allowance of 1,250 Btu/h per 1,000 cfm [775 $W/m^2/s$] fan heat for blowerless equipment.

6.2 *Part-Load Rating.* Only systems which are capable of capacity reduction shall be rated at 100% and at each step of capacity reduction provided by the refrigeration system(s) as published by the manufacturer. These rating points shall be used to calculate the IPLV (see 6.2.2).

6.2.1 *Part-Load Rating Conditions.* Test conditions for part-load ratings shall be per Table 5.

Any water flow required for system function shall be at water flow rates established at (full load) Standard Rating Conditions.

Capacity reduction means may be adjusted to obtain the specified step of unloading. No manual adjustment of indoor and outdoor airflow rates from those of the Standard Rating Conditions shall be made. However, automatic adjustment of airflow rates by system function is permissible.

6.2.2 *Integrated Part-Load Value (IPLV).* For equipment covered by this standard, the IPLV shall be calculated as follows:

a. Determine the capacity and EER at the conditions specified in Table 5

b. Determine the Part-Load Factor (PLF) from Figure 1, "Part-Load Factor Curve," at each rating point (see Appendix E)

c. Use the following equation to calculate IPLV:

$$
\begin{aligned}
IPLV = {} & \left(PLF_1 - PLF_2\right) \times \frac{\left(EER_1 + EER_2\right)}{2} \\
& + \left(PLF_2 - PLF_3\right) \times \frac{\left(EER_2 + EER_3\right)}{2} + \cdots\cdots \\
& + \left(PLF_{n-1} - PLF_n\right) \times \frac{\left(EER_{n-1} + EER_n\right)}{2} \\
& + \left(PLF_n\right) \times \left(EER_n\right)
\end{aligned}
$$

where:

PLF	=	Part-load factor determined from Figure 1
n	=	Total number of capacity steps
Superscript 1	=	100% capacity and EER at part-load Rating Conditions
Subscript 2, 3 etc.	=	Specific capacity and EER at part-load steps per 6.2

6.3 *Application Ratings.* Ratings at conditions of temperature or airflow rate other than those specified in

6.1.3 and 6.2.1 may be published as Application Ratings, and shall be based on data determined by the methods prescribed in 6.1. Application Ratings in the defrost region shall include net capacity and COP based upon a complete defrost cycle.

6.4 *Publication of Ratings.* Wherever Application Ratings are published or printed, they shall include, or be accompanied by the Standard Ratings plus the IPLV (where applicable), clearly designated as such, including a statement of the conditions at which the ratings apply.

6.4.1 *Capacity Designation.* The capacity designation used in published specifications, literature or advertising, controlled by the manufacturer, for equipment rated under this standard, shall be expressed only in Btu/h [W] at the Standard Rating Conditions specified in 6.1.3 plus part-load Rating Conditions specified in 6.2.1 and in the terms described in 6.1.1 and 6.1.2. Horsepower, tons or other units shall not be used as capacity designation.

6.5 *Tolerances.* To comply with this standard, measured test results shall not be less than 95% of Published Ratings for performance ratios and capacities.

(Note: Products covered by the National Appliance Energy Conservation Act (NAECA) shall be rated in accordance with 10 CFR 430, Section 24 m (1) (i) and (ii).)

Section 7. Minimum Data Requirements for Published Ratings

7.1 *Minimum Data Requirements for Published Ratings.* As a minimum, Published Ratings shall consist of the following information:

a. For Unitary Air-Conditoners (air-cooled)

1. ARI Standard Rating cooling capacity
2. Seasonal Energy Efficiency Ratio, SEER

b. For Unitary Air-Conditioners (water-cooled and evaporative-cooled)

1. ARI Standard Rating cooling capacity
2. Energy Efficiency Ratio, EER

c. For all Air-Source Unitary Heat Pumps

1. ARI Standard Rating cooling capacity
2. Seasonal Energy Efficiency Ratio, SEER
3. High temperature heating Standard Rating capacity
4. Region IV Heating Seasonal Performance Factor, HSPF, minimum design heating requirement

7.2 *Latent Capacity Designation*. The moisture removal designation shall be published in the manufacturer's specifications and literature. The value shall be expressed consistently in either gross or net in one or more of the following forms:

a. Sensible capacity/total capacity ratio and total capacity
b. Latent capacity and total capacity
c. Sensible capacity and total capacity

7.3 *Rating Claims*. All claims to ratings within the scope of this standard shall include the statement "Rated in accordance with ARI Standard 210/240". All claims to ratings outside the scope of this standard shall include the statement: "Outside the scope of ARI Standard 210/240". Wherever Application Ratings are published or printed, they shall include a statement of the conditions at which the ratings apply.

Section 8. Operating Requirements

8.1 *Operating Requirements.* Unitary equipment shall comply with the provisions of this section such that any production unit will meet the requirements detailed herein.

8.2 *Maximum Operating Conditions Test.* Unitary equipment shall pass the following maximum operating conditions test with an indoor-coil airflow rate as determined under 6.1.3.3.

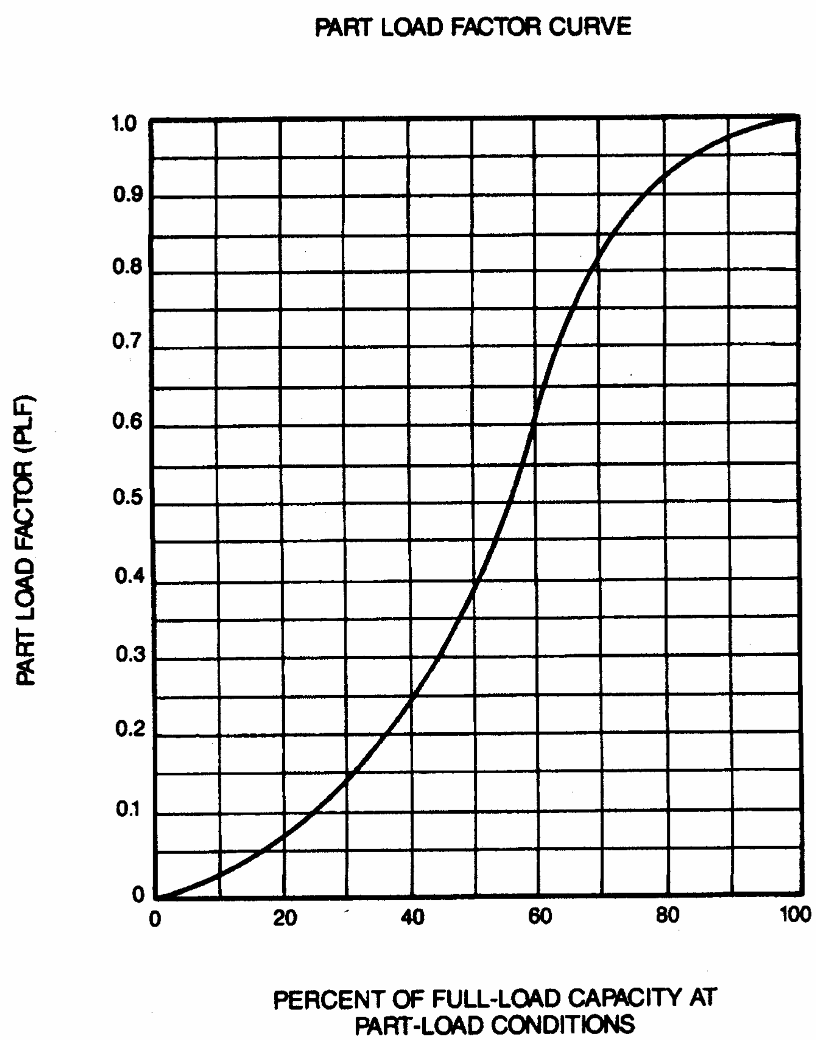
8.2.1 *Temperature Conditions.* Temperature conditions shall be maintained as shown in Tables 3, 4 or 5.

8.2.2 *Voltages.* The test shall be run at the Range A minimum utilization voltage from ARI Standard 110, Table 1, based upon the unit's nameplate rated voltage(s). This voltage shall be supplied at the unit's service connection and at rated frequency.

8.2.3 *Procedure.* The equipment shall be operated for one hour at the temperature conditions and voltage specified.

8.2.4 *Requirements.* The equipment shall operate continuously without interruption for any reason for one hour.

8.2.4.1 Units with water-cooled condensers shall be capable of operation under these maximum conditions at a water-



Note: The curve is based on following equation:

$$PLF = A0 + (A1 \times Q) + (A2 \times Q^2) + (A3 \times Q^3) + (A4 \times Q^4) + (A5 \times Q^5) + (A6 \times Q^6)$$

where: PLF = Part-Load Factor
Q = Percent of full-load capacity at part-load rating conditions.
$A0 = -0.12773917 \times 10^{-6}$
$A1 = -0.27648713 \times 10^{-3}$
$A2 = 0.50672449 \times 10^{-3}$
$A3 = -0.25966636 \times 10^{-4}$
$A4 = 0.69875354 \times 10^{-6}$
$A5 = -0.76859712 \times 10^{-8}$
$A6 = 0.28918272 \times 10^{-10}$

Figure 1. Part-Load Factor Curve

pressure drop not to exceed 15.0 psi [103 kPa], measured across the unit.

8.3 *Voltage Tolerance Test.* Unitary equipment shall pass the following voltage tolerance test with a cooling coil airflow rate as determined under 6.1.3.3.

8.3.1 *Temperature Conditions.* Temperature conditions shall be maintained at the standard cooling (and/or standard heating, as required) steady state conditions as shown in Table 3, Table 4 or Table 5.

8.3.2 *Voltages.*

8.3.2.1 Tests shall be run at the Range B minimum and maximum utilization voltages from ARI Standard 110, Table 1, based upon the unit's nameplate rated voltage(s). These voltages shall be supplied at the unit's service connection and at rated frequency. A lower minimum or a higher maximum voltage shall be used, if listed on the nameplate.

8.3.2.2 The power supplied to single phase equipment shall be adjusted just prior to the shut-down period (8.3.3.2) so that the resulting voltage at the unit's service connection is 86% of nameplate rated voltage when the compressor motor is on locked-rotor. (For 200V or 208V nameplate rated equipment the restart voltage shall be set at 180V when the compressor motor is on locked rotor). Open circuit voltage for three-phase equipment shall not be greater than 90% of nameplate rated voltage.

8.3.2.3 Within one minute after the equipment has resumed continuous operation (8.3.4.3), the voltage shall be restored to the values specified in 8.3.2.1.

8.3.3 *Procedure.*

8.3.3.1 The equipment shall be operated for one hour at the temperature conditions and voltage(s) specified.

8.3.3.2 All power to the equipment shall be shut off for a period sufficient to cause the compressor to stop (not to exceed five seconds) and then restored.

8.3.4 *Requirements.*

8.3.4.1 During both tests, the equipment shall operate without failure of any of its parts.

8.3.4.2 The equipment shall operate continuously without interruption for any reason for the one hour period preceding the power interruption.

8.3.4.3 The unit shall resume continuous operation within two hours of restoration of power and shall then operate continuously for one half hour. Operation and resetting of safety devices prior to establishment of continuous operation is permitted.

8.4 *Low-Temperature Operation Test (Cooling) (Not required for heating-only units).* Unitary equipment shall pass the following low-temperature operation test when operating with initial airflow rates as determined in 6.1.3.3 and 6.1.3.4 and with controls and dampers set to produce the maximum tendency to frost or ice the evaporator, provided such settings are not contrary to the manufacturer's instructions to the user.

8.4.1 *Temperature Conditions.* Temperature Conditions shall be maintained as shown in Table 3 or Table 5.

8.4.2 *Procedure.* The test shall be continuous with the unit on the cooling cycle, for not less than four hours after establishment of the specified temperature conditions. The unit will be permitted to start and stop under control of an automatic limit device, if provided.

8.4.3 *Requirements.*

8.4.3.1 During the entire test, the equipment shall operate without damage or failure of any of its parts.

8.4.3.2 During the entire test, the air quantity shall not drop more than 25% from that determined under the Standard Rating test.

8.4.3.3 During the test and during the defrosting period after the completion of the test, all ice or meltage must be caught and removed by the drain provisions.

8.5 *Insulation Effectiveness Test (Cooling) (not required for heating-only units).* Unitary equipment shall pass the following insulation effectiveness test when operating with airflow rates as determined in 6.1.3.3 and 6.1.3.4 with controls, fans, dampers, and grilles set to produce the maximum tendency to sweat, provided such settings are not contrary to the manufacturer's instructions to the user.

8.5.1 *Temperature Conditions.* Temperature conditions shall be maintained as shown in Table 3 or Table 5.

8.5.2 *Procedure.* After establishment of the specified temperature conditions, the unit shall be operated continuously for a period of four hours.

8.5.3 *Requirements.* During the test, no condensed water shall drop, run, or blow off from the unit casing.

8.6 *Condensate Disposal Test (Cooling)* (not required for heating-only units).* Unitary equipment which rejects condensate to the condenser air shall pass the following condensate disposal test when operating with airflow rates as determined in 6.1.3.3 and 6.1.3.4 and with controls and dampers set to produce condensate at the maximum rate, provided such settings are not contrary to the manufacturer's instructions to the user.

* This test may be run concurrently with the Insulation Effectiveness Test (8.5).

8.6.1 *Temperature Conditions.* Temperature conditions shall be maintained as shown in Table 3 or Table 5.

8.6.2 *Procedure.* After establishment of the specified temperature conditions, the equipment shall be started with its condensate collection pan filled to the overflowing point and shall be operated continuously for four hours after the condensate level has reached equilibrium.

8.6.3 *Requirements.* During the test, there shall be no dripping, running-off, or blowing-off of moisture from the unit casing.

8.7 *Tolerances.* The conditions for the tests outlined in Section 8 are average values subject to tolerances of ± 1.0°F [± 0.6°C] for air wet-bulb and dry-bulb temperatures, ± 1.0% of the reading for voltages.

Section 9. Marking and Nameplate Data

9.1 *Marking and Nameplate Data.* As a minimum, the nameplate shall display the manufacturer's name, model designation, and electrical characteristics.

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

Section 10. Conformance Conditions

10.1 *Conformance.* While conformance with this standard is voluntary, conformance shall not be claimed or implied for products or equipment within its *Purpose* (Section 1) and *Scope* (Section 2) unless such claims meet all of the requirements of this standard.

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 ANSI/ASHRAE Standard 37-1988, *Methods of Testing for Rating Unitary Air-Conditioning and Heat Pump Equipment*, 1988, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., 1791 Tullie Circle N.E., Atlanta, GA 30329, U.S.A.

A1.2 ANSI/ASHRAE Standard 41.1-1986 (RA 2001), *Standard Method for Temperature Measurement,* 2001, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., 1791 Tullie Circle, N.E., Atlanta, GA 30329, U.S.A.

A1.3 ARI Standard 110-2002, *Air-Conditioning and Refrigerating Equipment Nameplate Voltages*, Air-Conditioning and Refrigeration Institute, 2002, 4100 North Fairfax Drive, Suite 200, Arlington, VA 22203, U.S.A.

A1.4 ARI Standard 340/360-2000, *Commercial and Industrial Unitary Air-Conditioning and Heat Pump Equipment*, 2000, Air-Conditioning and Refrigeration Institute, 4100 North Fairfax Drive, Suite 200, Arlington, VA 22203, U.S.A.

A1.5 ASHRAE *Terminology of Heating, Ventilation, Air-Conditioning and Refrigeration,* Second Edition, 1991, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., 1791 Tullie Circle, N.E., Atlanta, GA 30329, U.S.A.

A1.6 IEC Standard Publication 60038, *IEC Standard Voltages,* 1983, International Electrotechnical Commission, 3, rue de Varembe, P.O. Box 131, 1211 Geneva 20, Switzerland.

APPENDIX B. REFERENCES - INFORMATIVE

None.

APPENDIX C. *UNIFORM TEST METHOD FOR MEASURING THE ENERGY CONSUMPTION OF CENTRAL AIR-CONDITIONERS - NORMATIVE

(Note: All items in this Appendix previously labeled "A" are now to be referred to as "C")

Foreword

This appendix to ARI Standard 210/240-94 is derived from the appropriate combining and editing by ARI of "Uniform Test Method for Measuring the Energy Consumption of Central Air-Conditioners" Appendix M to Subpart B, pages 76707 through 76723, Federal Register, Vol. 44, No. 249, Thursday, December 27, 1979 and "Part 430—Energy Conservation Program for Consumer Products," pages 8311 through 8319 (omitting page 8312 and parts of pages 8311 and 8313), Federal Register, Vol. 53 No. 49, Monday, March 14, 1988. Note the prefix A has been added to all section numbers in this appendix for clarity (to avoid confusion with section numbers in the standard).

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

A1. *Definitions*

A1.1 "Annual performance factor" means the total heating and cooling done by a heat pump in a particular region in one year divided by the total electric power used in one year.

A1.2 "ARI" means Air-Conditioning and Refrigeration Institute.

A1.3 "ARI Standard 210/240-94" means the test standard published in 1994 by the ARI and titled "Unitary Air-Conditioning and Air-Source Heat Pump Equipment".

A1.4 "ARI Standard 320-93" means the test standard published in 1993 by ARI and titled "Water-Source Heat Pumps".

A1.5 "ASHRAE" means the American Society of Heating, Refrigeration and Air-Conditioning Engineers, Inc.

A1.6 "ASHRAE Standard 37-88 means the test standard published by ASHRAE in 1988 and titled "Methods of Testing for Rating Unitary Air-Conditioning and Heat Pump Equipment."

A1.7 "Continuously recorded" means a method of recording measurements in intervals no greater than 5 seconds.

A1.8 "Cooling load factor (CLF)" means the ratio of the total cooling done in a complete cycle of a specified time period, consisting of an "on" time and "off" time, to the steady-state cooling done over the same period at constant ambient conditions.

A1.9 "Cyclic Test" means a test where the indoor and outdoor conditions are held constant, but the unit is manually turned "on" and "off" for specific time periods to simulate part-load operation.

A1.10 "Degradation coefficient (C_D)" means the measure of the efficiency loss due to the cycling of the unit.

A1.11 "Demand-defrost control system" means a system which is designed to perform the defrost function on the outdoor coil of the heat pump only when a predetermined degradation of performance is measured.

Note: The following examples of explanations are offered (reference James A. Smith, Department of Energy, letter of November 23, 1981 to Robert Newell, Rheem A/C Div.):

"Examples which comply are:

1. differential air pressure sensors,
2. differential temperature (coil to ambient air),
3. feedback systems that measure length of defrost period and adjust defrost frequency accordingly,
4. systems that measure outdoor fan power, and/or current,
5. optical sensors.

A demand defrost system must be able to monitor at least one parameter which always varies with the amount of frost accumulated on the outdoor coil of the heat pump. When this parameter reaches a certain value, the system initiates a defrost. The termination of the defrost can be accomplished by measuring any parameter that can be used to sense the elimination of frost from the coil.

Systems that vary defrost intervals according to outdoor dry-bulb temperature are not demand defrost systems. This is because knowledge of dry-bulb temperature only predicts the occurrence of frost. A demand defrost system must function in response to a parameter which varies directly with frosting.

When a demand defrost system is used in conjunction with a time-initiated defrost system, the combination will not be considered a demand system if time initiated defrosts occur more frequently than every 6 hours of compressor operating time."

A1.12 "Design heating requirement (DHR)" is the amount of heating required to maintain a given indoor temperature at a particular outdoor design temperature.

A1.13 "Dry-coil test" means a test conducted at a wet-bulb temperature and a dry-bulb temperature such that moisture will not condense on the evaporator coil of the unit.

A1.14 "Heating seasonal performance factor (HSPF)" means the total heating output of a heat pump during its normal annual usage period for heating divided by the total electric power input during the same period.

A1.15 "Heating load factor (HLF)" means the ratio of the total heating done in a complete cycle of a specified time period, consisting of an "on" time "off" time, to the steady state heating done over the same period at constant ambient conditions.

A1.16 "Latent cooling" means the amount of cooling in Btu's necessary to remove water vapor from the air passing over the indoor coil by condensation during a period of time.

A1.17 "Part-load factor (PLF)" means the ratio of the cyclic energy efficiency ratio to the steady-state energy efficiency ratio at identical ambient conditions.

A1.18 "Seasonal energy efficiency ratio (SEER)" means the total cooling of a central air conditioner in Btu's during its normal annual usage period for cooling divided by the total electric power input in watt-hours during the same period.

A1.19 "Sensible cooling" means the amount of cooling in Btu's performed by a unit over a period of time, excluding latent cooling.

A1.20 "Single package unit" means any central air conditioner in which all the major assemblies are enclosed in one cabinet.

A1.21 "Split system" means any central air conditioner in which one or more of the major assemblies are separate from the others.

A1.22 "Steady-state test" means a test in which all indoor and outdoor conditions are held constant and the unit is in non-changing operating mode.

A1.23 "Temperature bin" means a 5 F increment over a dry-bulb temperature range of 65 F through 104 F for the cooling cycle and − 25 F through 64 F for the heating cycle.

A1.24 "Time-temperature defrost control system" means a system which automatically provides the defrost function at a predetermined time interval whenever the outdoor temperature drops below a level where frosting will occur.

A1.25 "Test condition tolerance" means the maximum permissible variation of the average of the test observations from the standard or desired test condition as provided in A6.1.1, A6.2.2, and A6.2.3.

A1.26 "Test operating tolerance" means the maximum permissible difference between the maximum and the minimum instrument observation during a test as provided in A6.1.1, A6.2.1, A6.2.2 and A6.2.3.

A1.27 "Wet-coil test" means a test conducted at a wet-bulb temperature and a dry-bulb temperature such that moisture will condense on the test unit evaporator coil.

A1.28 "Central air conditioner" (DOE Covered) means a product, other than a packaged terminal air conditioner powered by single phase electric current, which is air-cooled, rated below 65,000 Btuh, not contained within the same cabinet as a furnace, the rated capacity of which is above 225,000 Btuh, and is a heat pump or a cooling only unit.

A1.29 "Heat pump" (DOE Covered) means a product, other than a packaged terminal heat pump, which consists of one or more assemblies, powered by single phase electric current, rated below 65,000 Btuh, utilizing an indoor conditioning coil, compressor, and refrigerant-to-outdoor air heat exchanger to provide air heating, and may also provide air cooling, dehumidifying, humidifying circulating, and air cleaning.

A1.30 "Coil family" means a group of coils with the same basic design features that affect the heat exchanger performance. These features are the basic configuration, i.e., A-shape, V-shape, slanted or flat top, the heat transfer surfaces on refrigerant and air sides (flat tubes vs. grooved tubes, fin shapes), the tube and fin materials, and the coil circuitry. When a group of coils has all these features in common, it constitutes a "coil family."

A2. *Testing Required*

A2.1 *Testing required for air source cooling only units.* Two steady state wet coil tests are required to be performed test A and test B. Test A is to be conducted as an outdoor dry bulb temperature of 95 F and test B at 82 F. Test C and D are optional tests to be conducted when cyclic performance parameters are to be measured in order to determine the degradation coefficient, C_D. Test C is a steady state dry coil test conducted at an outdoor dry bulb temperature of 82 F. Test D is a cyclic test also conducted at an outdoor dry bulb temperature of 82 F. In lieu of conducting tests C and D, an assigned value of 0.25 may be used for the degradation coefficient, C_D.

A2.1.1 *Testing required for units with single speed compressors and single speed condenser fans.* Test A and test B shall be performed according to the test procedures outlined in A4.1 of this Appendix. In addition, the cyclic performance shall be evaluated by conducting test C and D according to the requirements outlined in A4.1.

A2.1.2 *Testing required for units with single speed compressors and multiple-speed condenser fans.* The test requirements for multiple-speed condenser fan units shall be the same as described in section A2.1.1 for single speed condensor fan units.

A2.1.3 *Testing required for units with two-speed compressors, two compressors, or cylinder unloading.* The test requirements for two-speed compressor units, two compressor units, or units with cylinder unloading are the same as described in A2.1.1 except that test A and test B shall be performed at each compressor speed or at each compressor capacity.

A2.1.4 *Testing required for units with two-speed compressors, two compressors, or cylinder unloading capable of varying the sensible to total (S/T) capacity ratio.* When a unit employing a two-speed compressor, two compressors, or cylinder unloading provides a method of varying the ratio of the sensible cooling capacity to the total cooling capacity, (S/T), the test requirements are the same as for two-speed compressor units as described in A2.1.3.

A2.1.5 *Testing required for units with triple-capacity compressors.* (Reserved)

A2.1.6 *Testing required for units with variable-speed compressors.* The tests for variable-speed equipment consist of five (5) wet coil tests and two (2) dry coil tests. Two of the wet coil tests, A and B, are conducted at the maximum speed. Two wet coil tests, B_2 and low temperature test, are conducted at the minimum speed. The fifth wet coil test is conducted at an intermediate speed. Dry coil tests, C and D, are conducted at the minimum speed if the coefficient of degradation (C_D) value of 0.25 is not adopted. The test conditions and procedures for the above are outlined in sections A3.1 and A4.1.

A2.1.7 *Testing required for split-type ductless systems.* The tests for split-type ductless systems are determined by the type of compressor installed in the outdoor unit. For the appropriate tests refer to sections A2.1.1, A2.1.2, A2.1.3, A2.1.4, A2.1.5, or A2.1.6.

A2.2 *Testing required for air source heating only units.*

A2.2.1 *Testing required for units with single speed compressors.* Units with single speed compressors shall be subjected respectively to the High Temperature Test at 47 F described in section A3.2.1.1, the Cylic Test as described in section A3.2.1.2, the Frost Accumulation Test as described in section A3.2.1.3, and the Low Temperature Test as described in section A3.2.1.4.

A2.2.2 *Testing required for units with two-speed compressors, two compressors, or cylinder unloading.* With the unit operating: at high compressor speed (two-speed compressor), with both compressors in operation (two-compressors), or at the maximum capacity (cylinder unloading); the following tests are required to be performed on all units; the High Temperature Test at 47 F, the Frost Accumulation Test, and the Low Temperature Test. An additional test (cyclic at 47 F) is required, with the unit operating at the high compressor speed (two-speed compressor), with both compressors in operation (two compressors), or at the maximum capacity (cylinder unloading); if the normal mode of operation requires cycling "on" and "off" of the compressor(s) at high speed or maximum capacity.

With the unit operating: at the low compressor speed (two-speed compressor), with the single compressor which normally operates at low loads (two compressors), or at the low compressor capacity (cylinder unloading); the following tests are required to be performed on all units: the High Temperature Test at 47 F, the High Temperature Test at 62 F, and the Cyclic Test. Additional tests, (Frost Accummulation Test and Low Temperature Test) are required, with the unit operating: on low compressor speed (two-speed compressor), with the single compressor which normally operates at low loads (two compressors) or at the low compressor capacity cylinder unloading), if the unit's low speed, one compressor or low capacity performance at and below 40 F is needed to calculate its seasonal performance.

A2.2.3 *Testing required for units with triple-capacity compressors.* (Reserved)

A2.2.4 *Testing required for units with variable-speed compressors.* There are seven basic tests and one optional test for variable-speed units. Three tests (high temperature test, low temperature test, and frost accumulation test) are performed at the maximum speed. Three tests (two high temperatures and one cyclic test) are performed with the unit operating at minimum speed. A second frost accumulation test is performed at an intermediate speed. The intermediate speed is the same as in the cooling mode.

In lieu of the maximum speed frost accumulation test, two equations are provided in section A4.2. In lieu of the cyclic test an assigned value of 0.25 may be used for the coefficient of degradation C_D. The optional test is a nominal capacity test applicable to units which have a heating mode maximum speed greater than the cooling mode maximum speed. The conditions and procedures for the above tests are decribed in sections A3.2 and A4.2 respectively.

A2.2.5 *Testing required for split-type ductless system.* The type of compressor installed in the outdoor unit determines the testing required, refer to previous sections A2.2.1, A2.2.2, A2.2.3, or A2.2.4. The conditions and procedures will be modified as indicated for the various types as stated in sections A3.2 and A4.2 respectively.

A2.3 *Testing required for air source units which provide both heating and cooling.* The requirements for units which provide both heating and cooling shall be the same as the requirements in Section A2.1 and A2.2.

A3. *Testing conditions*

A3.1 *Testing conditions for air source cooling only units.* The test room requirement and equipment installation procedures are the same as those specified in section 8.1 and 8.6 of ASHRAE Standard 37-88. Units designed for both horizontal and vertical installation shall be tested in the orientation in which they are most frequently installed. All tests shall be performed at the normal residential voltage and frequency for which the equipment is designed (either 115 or 230 volts and 60 hertz), the test installation shall be designed such that there will be no air flow through the cooling coil due to natural or forced convection while the indoor fan is "off". This shall be accomplished by installing dampers upstream and downstream of the test unit to block the off period air flow. Values of capacity for rating purposes are to be rounded off to the nearest 100 Btuh for capacities less than 20,000 Btuh, to the nearest 200 Btuh for capacities between 20,000 and 37,999 Btuh, and to the nearest 500 Btuh for capacities between 38,000 and 64,999 Btuh.

The following conditions listed in ARI Standard 210/240-94 shall apply to all tests performed in Section A3.1: 5.1.3.3 "Cooling Coil Air Quantity," 5.1.3.5 "Requirements for Separated Assemblies."

A3.1.1 *Testing conditions for units with single speed compressors and single speed condenser fans.*

A3.1.1.1 *Steady state wet-coil performance tests (Test A and Test B).* Test A and test B shall be performed with the air entering the indoor side of the unit having a dry-bulb temperature of 80 F and a wet-bulb temperature of 67 F. The dry-bulb temperature of the air entering the outdoor side of the unit shall be 95 F in test A and 82 F in test B. The temperature of the air surrounding the outdoor side of the unit in each test shall be the same as the outdoor entering air temperature except for units or sections thereof intended to be installed only indoors, in which case the dry-bulb temperature surrounding that indoor side of the unit shall be 80 F. For those units which reject condensate to the condenser, located in the outdoor side of the unit, the outdoor wet-bulb temperature surrounding the outdoor side of the unit shall be 75 F in test A and 65 F in test B.

A3.1.1.2 *Steady state dry coil performance test (Test C) and cyclic dry coil performance test (Test D).* Test C and test D shall be performed with the air entering the indoor side of the unit having a dry-bulb temperature of 80 F and a wet-bulb temperature which does not result in formation of condensate on the indoor coil. (It is recommended that an indoor wet-bulb temperature of 57 F or less be used.) The dry-bulb temperature of the air entering the outdoor portion of the unit shall be 82 F. The outdoor portion of the unit shall be subject to the same conditions as the requirements for conducting test B as stated previously in section A3.1.1.1. Test C shall be conducted with the unit operating steadily. Test D shall be conducted by cycling the unit "on" and "off" by manual or automatic operation of the normal control circuit of the unit. The unit shall cycle with the compressor "on" for 6 minutes and "off" for 24 minutes. The indoor fan shall also cycle "on" and "off", the duration of the indoor fan "on" and "off" periods being governed by the automatic controls which the manufacturer normally supplies with the unit. The results of tests C and D shall be used to calculate a degredation coefficient, C_D by the procedures outlined in A5.1.

A3.1.2 *Testing conditions for units with single speed compressors and multiple-speed condenser fans.* The condenser fan speed to be used in test A shall be that speed which normally occurs at an outdoor dry-bulb temperature 95 F, and for test B, the fan speed shall be that which normally occurs at an outdoor dry-bulb temperature of 82 F. If elected to be performed, tests C and D shall be conducted at the same condenser fan speed as in test B.

A3.1.3 *Testing conditions for units with two-speed compressors, two compressors, or cylinder unloading.* The condenser fan speed used in conducting test A at each compressor speed shall be that which normall occurs at an outdoor dry-bulb temperature of 95 F. For test B, the condenser fan speed at each compressor speed shall be that which normally occurs at an outdoor dry-bulb temperature of 82 F. If elected to be performed, tests C and D shall be conducted at the low compressor speed with the same condenser fan speed as used in test B. For those two-speed units in which the normal mode of operation involves cycling the compressor "on" and "off" at high speed, tests C and D shall also be performed with the compressor operating at high speed and at a condenser fan speed that normally occurs at test A ambient conditions. Units consisting of two compressors are subject to the same requirements as those units containing two-speed compressors, except that when operated at high speed, both compressors shall be operating and when operating at low speed, only the compressor which normally operates at an outdoor dry-bulb temperature of 82 F shall be operating.

In lieu of conducting tests C and D, an assigned value of 0.25 may be used for the degradation coefficient, C_D, at each compressor speed. If the assigned degradation coefficient is used for one compressor speed it must also be used for the other compressor speed.

In the case of units with cylinder unloading, the loaded and the unloaded conditions correspond to high and low compressor speed on two-speed units respectively.

A3.1.4 *Testing conditions for units with two-speed compressors, two compressors, or cylinder unloading capable of varying the sensible to total (S/T) capacity ratio.* The mode of operation selected for controlling the S/T ratio in the performance of test A and test B at each compressor speed shall be such that it does not result in an operating configuration which is not typical of a normal residential installation. If elected to be performed, tests C and D shall be conducted at low compressor speed (single compressor operating) with the same S/T control mode as used in test B when performed at the low compressor speed. Likewise, tests C and D shall also be conducted at high compressor speed (two compressors operating) and with the same S/T control mode as in test A when performed at the high compressor speed.

In the case of units with cylinder unloading, the loaded and unloaded conditions correspond to high and low compressor speed on two-speed units respectively.

A3.1.5 *Testing conditions for units with triple-capacity compressors.* (Reserved)

A3.1.6 *Additional testing conditions for cooling-only units with variable-speed compressors.* For cooling-only units and air-source heat pumps with variable-speed compressors, the air flow rate at fan speeds less than the maximum fan speed shall be determined by using the fan law for a fixed resistance system. The air flow rate is given by the ratio of the actual fan speed to the maximum fan speed multiplied by the air flow rate at the maximum fan speed. Minimum static pressure requirements only apply when the fan is running at the maximum speed.

A3.1.6.1 *Testing conditions for steady state wet coil tests.* Tests A and B shall be performed at the maximum speed at conditions specified in section A3.1.1. Test B_2 and the low temperature test are performed at the minimum speed with outdoor dry bulb temperatures of 82 F and 67 F respectively. The intermediate speed wet coil test is performed at the outdoor dry bulb temperature of 87 F. For units which reject condensate the outdoor wet bulb temperature shall be maintained at 75 F for Test A, 65 F for Tests B and B_2, 53.5 F for the low temperature test and 69 F for the intermediate test. The indoor conditions for all wet coil tests are the same as those given in section A3.1.1.

A3.1.6.2 *Test conditions for dry coil tests.* Dry coil Tests C and D are conducted at an outdoor dry bulb temperature of 82 F. For units which reject condensate the outdoor wet bulb temperature shall be maintained at 53.5 F. The indoor dry bulb temperature shall be 80 F and the wet bulb temperature shall be sufficiently low so no condensation occurs on the evaporator (It is recommended that an indoor wet bulb temperature of 57 F or less be used).

A3.1.7 *Split-type ductless systems.* Test conditions shall be the same as those specified for the same single outdoor unit compressor type, assuming it was matched with a single indoor coil.

A3.1.7.1 *Interconnection.* For split-type ductless systems, all standard rating tests shall be performed with a minimum length of 25 feet of interconnecting tubing between each indoor fan-coil unit and the common outdoor unit. Such equipment in which the interconnection tubing is furnished as an integral part of the machine not recommended for cutting to length shall be tested with complete length of tubing furnished, or with 25 feet of tubing, whichever is greater. At least 10 feet of the interconnection tubing shall be exposed to the outside conditions. The line sizes, insulation and details of installation shall be in accordance with the manufacturer's published recommendation.

A3.1.7.2 *Control testing conditions for split-type ductless systems.* For split-type ductless systems, a single control circuit shall be substituted for any multiple thermostats in order to maintain a uniform cycling rate during test D and the high temperature heating cyclic test. During the steady-state tests, all thermostats shall be shunted resiting in all indoor fan-coil units being in operation.

A3.1.7.3 *Split-type ductless systems with multiple coils or multiple discharge outlets shall have short plenums attached to each outlet.* Each plenum shall discharge into a single common duct section, the duct section in turn discharging into the air measuring device (or a suitable dampering device when direct air measurement is not employed). Each plenum shall have an adjustable restrictor located in the plane where the plenums enter the common duct section for the purpose of equalizing the static pressures in each plenum. The length of the plenum is a minimum of 2.5 × $(A \times B)^{.5}$, A = width and B = height of duct or outlet. Static pressure readings are taken at a distance of 2 × $(A \times B)^{.5}$ from the outlet.

A3.2 *Testing conditions for air source heating only units.* The equipment under test shall be installed according to the requirements of Section 8.6 of ASHRAE Standard 37-88 and Section 5.1.3.5 of ARI Standard 210/240-94. Test chamber requirements are the same as given in Section 8.1 of ASHRAE Standard 37-88. Units designed for both horizontal and vertical installation shall be tested in the orientation in which they are most often installed. All tests shall be performed at the normal residential voltage and frequency for which the equipment is designed (either 115 or 230 volts and 60 hertz). Values of capacity for rating purposes are to be rounded off to the nearest 100 Btuh for capacities less than 20,000 Btuh; to the nearest 200 Btuh for capacities between 20,000 and 37,999 Btuh; and to the nearest 500 Btuh for capacities between 38,000 and 64,999 Btuh.

A3.2.1 *Testing conditions for units with single speed compressors.*

A3.2.1.1 *High temperature test conditions.* The High Temperature Test at 47 F shall be conducted at an outdoor dry-bulb temperature of 47 F and an outdoor wet-bulb temperature of 43 F. The High Temperature Test at 62 F shall be conducted at an outdoor dry-bulb temperature of 62 F and an outdoor wet-bulb temperature of 56.5 F. For both tests the dry-bulb air temperature entering and surrounding the indoor portion of the unit shall be 70 F and a maximum wet-bulb temperature of 60 F. The duration of the tests shall be for a minimum of ½ hour.

A3.2.1.2 *Cycling test conditions.* The Cycling Test at 47 F shall be conducted at the same dry-bulb and wet-bulb temperature as the High Temperature Test at 47 F as described in A3.2.1.1. During the Cycling Test, the indoor fan shall cycle "on" and "off", as the compressor cycles "on" and "off", except that the indoor fan cycling times may be delayed due to controls that are normally installed with the unit. The compressor cycling times shall be 6 minutes "On" and 24 minutes "off." The test installation shall be designed such that there will be no airflow through the indoor unit due to natural or forced convection while the indoor fan is "off." This shall be accomplished by installing dampers upstream and downstream of the test unit to block the off period airflow.

A3.2.1.3 *Frost accumulation test conditions.* The dry-bulb temperature and the resultant dew-point temperature of the air entering the outdoor portion of the unit shall be 35 F and 30 F respectively. The indoor dry-bulb temperature shall be 70 F and the maximum indoor wet-bulb temperature shall be 60 F. The Frost Accumulation Test requires that the unit undergo a defrost prior to the actual test. The test then begins at defrost termination and ends at the next defrost termination. Defrost termination occurs when the controls normally installed within the unit are actuated to cause it to change defrost operation to normal heating operation. During the test, auxiliary resistance heaters shall not be employed during either the heating or defrost portion of the test.

A3.2.1.4 *Low temperature test conditions.* The Low Temperature Test shall be conducted at a dry-bulb temperature entering the outdoor portion of the unit of 17 F and a wet-bulb temperature of 15 F. The air entering the indoor portion of the unit shall have a dry-bulb temperature of 70 F and a maximum wet-bulb temperature of 60 F.

A3.2.1.5 *Additional testing conditions.* All tests shall be conducted at the indoor-side air quantities specified in Section 5.1.3.3, 5.1.3.6 and Table 6 of ARI Standard 210/240-94. The following conditions listed in ARI Standard 210/240-94 shall apply to all tests performed in Section A3.2; 5.1.3.4 "Outdoor-Side Air Quantity"; 5.1.3.5 "Requirements for Separated Assemblies." In all tests, the specified dry-bulb temperature entering the outdoor portion of the unit also applies to the air temperature surrounding the outdoor portion of the unit. Similarly, models where portions are intended to be installed indoors shall have the air temperature surrounding that portion of the unit the same as the indoor air temperature.

A3.2.2 *Testing conditions for units with two-speed compressors, two compressors or cylinder unloading.* The testing conditions for two-speed compressors, two compressors, or cylinder unloading shall be the same as those for single speed units as described in A3.2.1.

A3.2.3 *Testing conditions for units with triple-capacity compressors.* (Reserved)

A3.2.4 *Testing conditions for units with variable-speed compressors.* The testing condition for variable-speed compressors shall be the same as those for single speed units as described in section A3.2.1 with the following exceptions: the cyclic test is performed with an outdoor dry bulb temperature of 62 F and a wet bulb temperature of 56.5 F. The optional, nominal capacity test shall be performed at the conditions specified for the 47 F high temperature test.

A3.2.5 *Testing conditions for split-type ductless systems.* The testing conditions for split-type ductless systems shall be based on the type of compressor installed in the single outdoor unit. The heating mode shall have the same piping and control requirements as in A3.1.7.

A3.3 *Testing conditions for air source units which provide both heating and cooling.* The testing conditions for units which provide both heating and cooling shall be the same as the requirements in Section A3.1. and A3.2.

4.0 *Testing procedures.* Measure all electrical inputs as described in the procedures below. All electrical measurements during all "on" and "off" periods shall include auxiliary power or energy (controls, transformers, crankcase heaters, etc.) delivered to the unit.

A4.1 *Testing procedures for air source cooling only units.* All steady-state wet- and dry-coil performance tests on single package units shall simultaneously employ the Air-Enthalpy Method (Section 7.3 of ASHRAE Standard 37-88) on the indoor side and one other method consisting of either the Air-Enthalpy Method or the Compressor Calibration Method (Section 7.4 of ASHRAE Standard 37-88) on the outdoor side. All steady-state wet- and dry-coil performance tests on split systems shall simultaneously employ the Air-Enthalpy Method or the Compressor Calibration Method on the indoor side and the Air-Enthalpy Method, the Compressor Calibration Method or the Refrigerant Flow Method (Section 7.6.2 of ASHRAE Standard 37-88) on the outside. All cyclic dry-coil performance tests shall employ the Air-Enthalpy Method, indoor side only. The values calculated from the two test methods must agree within 6 percent in order to constitute a valid test. Only the results from the Air-Enthalpy Method on the indoor side shall be used in the calculations in Section 5.1. Units shall be installed and tested in such a manner that when operated under steady-state conditions, the cooling coil and condenser coil air flows meet the requirements of Sections 5.1.3.4, 5.1.3.5, and 5.1.3.6 of ARI Standard 210/240-94.

A4.1.1 *Test operating procedures.*

A4.1.1.1 *Steady-state wet-coil performance tests (Test A and Test B).* Steady-state wet-coil performance tests (A and B) shall be conducted in accordance with the conditions described in sections A3.1.1.1, A3.1.2, A3.1.3, A3.1.4, and A3.1.5 of this appendix and the test procedures described for cooling tests in Section 8 of ASHRAE standard 37-88 and evaluated in accordance with the cooling-related requirements of Section 10 of the ASHRAE Standard 37-88. The test room reconditioning apparatus and the equipment under test shall be operated until equilibrium conditions are attained.

A4.1.1.2 *Steady-state and cyclic and dry-coil performance tests (Test C and D).* The steady-state and cyclic dry-coil tests (C and D) shall be conducted as described below in accordance with the conditions described in sections A3.1.1.2, A3.1.2, A3.1.3, A3.1.4, and A3.1.5 and A3.1.6.2. The results shall be evaluated in accordance with the cooling related requirements of Section 10.1.5, 10.1.6 and 10.1.7, of ASHRAE Standard 37-88. The test room reconditioning apparatus and the equipment under test shall be operated until equilibrium conditions are attained, but not for less than one hour before data for test C are recorded. For all equipment test methods including the Compressor Calibration Method, test C shall be performed with data recorded at 10-minute intervals until four consecutive sets of readings are attained with the tolerance prescribed in Section 9.2 of ASHRAE Standard 37-88. When the Air-Enthalpy Method is used on the outdoor side for test C, the requirements of this section shall apply to both the preliminary test and the regular equipment test; the requirements of Section 8.5 of ASHRAE Standard 37-88 shall also apply. Immediately after test C is completed the test unit shall be manually cycled "off" and "on" using the time periods from A3.1.1 of this Appendix until steadily repeating ambient conditions are again achieved in both the indoor and outdoor test chambers, but for not less than 2 complete "off"/"on" cycles. Without a break in the cycling pattern, the unit shall be run through an additional "off"/"on" cycle during which the test data required in A5.1 shall be recorded. During this last cycle, which is referred to as the test cycle, the indoor and outdoor test room ambient conditions shall remain within the tolerances specified in A4.1.3 during the cyclic dry-coil tests, all air moving equipment on the condenser side shall cycle "on" and "off" when the compressor cycles "on" and "off". The indoor air moving equipment shall also cycle "off" as governed by any automatic controls normally installed with the unit. This last requirement applies to units having an indoor fan time delay. Units not supplied with an indoor fan time delay shall have the indoor air moving equipment cycle "on" and "off" as the compressor cycles "on" and "off."

Cooling cyclic tests for variable-speed units shall be conducted by cycling the compressor 12 minutes "on" and 48 minutes "off". The capacity shall be measured for the integration time (θ), which is the compressor "on" time of 12 minutes or the "on" time as extended by fan delay, if so equipped. The electrical energy shall be measured for the total integration time (θ_{cyc}) of 60 minutes. In lieu of conducting C and D tests, an assigned value of 0.25 shall be used for the degradation coefficient for cooling, C_D.

A4.1.1.3 *Testing procedures for triple-capacity compressors.* (Reserved)

A4.1.1.4 *Intermediate cooling steady-state test for units with variable-speed compressors.* For units with variable-speed compressors, an intermediate cooling steady-state test shall be conducted in which the unit shall be operated at a constant, intermediate compressor speed (k = i) in which the dry-bulb and wet-bulb temperatures of the air entering the indoor coil are 80 F_{DB} and 67 F_{WB} and the outdoor coil are 87 $_{DB}$ and 69 F_{WB}. The tolerances for the dry-bulb and wet-bulb temperatures of the air entering the indoor and outdoor coils shall be the test operating tolerance and test condition tolerance specified in A6.1.1. The intermediate compressor speed shall be the minimum compressor speed plus one-third the difference between the maximum and minimum speeds of the cooling mode. (Inter. speed = min. speed + ⅓ max. speed − min. speed.) A tolerance of plus five percent or the next higher inverter frequency step from that calculated is allowed.

A4.1.1.5 *Testing procedures for split-type ductless systems.* Cyclic tests of ductless units will be conducted without dampers. The data cycle shall be preceded by a minimum of two cycles in which the indoor fan cycles on and off with the compressor. For the data cycle the indoor fan will operate three minutes prior to compressor cut-on and remain on for three minutes after compressor cut-off. The integration time for capacity and power shall be from compressor cut-on time to indoor fan cut-off time. The fan power for three minutes after compressor cut-off shall be added to the integrated cooling capacity.

A4.1.2 *Test instrmentation.* The steady-state and cyclic performance tests shall have the same requirements pertaining to instrumentation and data as those specified in Section 5 and Table 5 of ASHRAE Standard 37-88. For the cyclic dry-coil performance tests, the dry-bulb temperature of the air entering and leaving the cooling coil, or the difference between these two dry-bulb temperatures, shall be continuously recorded with instrumentation accurate to within ±0.3 F of indicated value and have a response time of 2.5 seconds or less. Response time is the time required for the instrumentation to obtain 63 percent of the final steady-state temperature difference when subjected to a step change in temperature difference of 15 F or more. Electrical measurement devices (watt-hour meters) used during all tests shall be accurate to within ±0.5 percent of indicated value.

A4.1.3 *Test tolerances.* All steady-state wet- and dry-coil performance tests shall be performed within the applicable operating and test condition tolerances specified in Section 9.2 and Table 4 of ASHRAE Standard 37-88.

A4.1.3.1 The indoor and outdoor average dry-bulb temperature for the cyclic dry coil test D shall both be within 1.0 F of the indoor and outdoor average dry bulb temperature for the steady-state dry coil test C, respectively.

4.1.3.2 The test condition and test operating tolerances for conducting test D are stated in A6.1.1. Variation in the test conditions greater than the tolerances prescribed in A6.1.1 shall invalidate the test. It is suggested that an electric resistance heater having a heating capacity approximately equal to the sum of the cooling capacity and compressor and condenser fan power should be installed in the outdoor test room and cycled "off" and "on" as the unit cycles "on" and "off" respectively to improve control in the outdoor test room. Similarly, an electric resistance heater having a heating capacity approximately equal to the cooling capacity of the unit could be installed in the indoor test room, and cycled "on" and "off" as the test unit cycles "on" and "off" to improve indoor room control.

A4.2 *Testing procedures for air source heating only units.*

A4.2.1 *Test operating procedures.* All High Temperature Tests, the Cyclic Test, the Frost Accumulation Test, and the low Temperature test shall have the performance evaluated by the Air-Enthalpy Method on the indoor side. In addition, the High Temperature Test and the Low Temperature Test shall have a simultaneous test method (as described in A4.1) used as a check. The values calculated from the two methods must agree within 6 percent in order to constitute a valid test. Only the results from the Air-Enthalpy Method on the indoor side shall be used in the calculations in section A5.2.

A4.2.1.1 *Test procedure for high temperature test.* When the outdoor Air-Enthalpy Method is used, the outdoor chamber must not interfere with the normal air circulating pattern during the preliminary test. It is necessary to determine and adjust for system resistance when the outdoor air measuring apparatus is attached to the outdoor portion of the unit. The test room apparatus and test units must be operated for at least one hour with at least ½ hour at equilibrium and at the specified test conditions prior to starting the test. The High Temperature Test shall then be conducted for a minimum of ½ hour with intermittent data being recorded at 10-minute intervals. For all units, especially those having controls which periodically cause the unit to operate in defrost mode, attention should be given to prevent defrost during the High Temperature Test. Units which have undergone a defrost should operate in the heating mode for at least 10-minutes after defrost termination proir to the start of the test. When the outdoor Air-Enthalpy Method is used as a second test then a preliminary test must be conducted for a minimum of 30 minutes with 4 or more sets of data recorded at 10 minute intervals, all remaining requirements of Section 8.5 in the ASHRAE Standard 37-88 shall then apply in conducting the preliminary test for the outdoor air enthalpy method. For some units, at the ambient condition of the test, frost may accumulate on the outdoor coil. If the supply air temperature or the difference between the supply air temperature and the indoor air entering temperature has decreased by more than 1.5 F at the end of the test, the unit shall be defrosted and the test restarted. Only the results of this second High Temperature Test shall be used in the heating seasonal performance calculation in section A5.2. Prior to beginning the High Temperature Test, a unit shall operate in the heating mode for at least 10 minutes after defrost termination to establish equilibrium conditions for the unit and the room reconditioning apparatus. The High Temperature Test may only begin when the test unit and room conditions are within the test condition tolerances specified in Section A6.2.1.

A4.2.1.2 *Test procedures for the cyclic test.* The cyclic test shall follow the High Temperature Test and by cycled "on" and "off" as specified in A3.2.1.2 until steadily repeating ambient conditions are achieved for both the indoor and outdoor test chambers, but for not less than 2 complete "off"/"on" cycles. Without a break in the cycling pattern, the unit shall be operated through an additional "off"/"on" cycle, during which the required test data shall be recorded. During the last cycle, which is referred to as the test cycle, the indoor and outdoor test room ambient conditions shall remain within the tolerances specified in section A6.2.2. If, prior to the High Temperature Test, the unit underwent a defrost cycle to rid the outdoor coil of any accumulated frost, then prior to cycling the unit "off" and "on" it should be made to undergo a defrost. After defrost is completed and before starting the cycling process, the unit shall be operated continuously in the heating mode for at least 10 minutes to assure that equilibrium conditions have again been established for the unit and the room conditioning apparatus. Cycling the unit may begin when the test unit and room conditions are within the High Temperature Test condition tolerances specified in section A6.2.1. Attention should be given to prevent defrost after the cycling process has begun.

The cycle times for variable-speed units is the same as the cyclic time in the cooling mode as specified in section A4.1.1.2. Cyclic tests of split-type ductless units will be conducted without dampers, and the data cycle shall be preceded by a minimum of two cycles in which the indoor fan cycles on and off with the compressor. During the data cycle for the split type ductless units, the indoor fan will operate three minutes prior to compressor "cut-on" and remain on for three minutes after compressor "cut-off". The integration time for capacity and power will be from compressor "cut-on' time to indoor fan "cut-off" time. The fan power for the three minutes after compressor "cut-off" shall be subtracted from the integrated heating capacity. For split-type ductless systems which turn the indoor fan off during defrost, the indoor supply duct shall not be blocked.

4.2.1.3 *Test procedures for the frost accumulation test.* The defrost controls shall be set at the normal settings which most typify those encountered in Region IV as described in section A6.2.4 and A6.2.5. The test room reconditioning equipment and the unit under test shall be operated for at least ½ hour prior to the start of a "preliminary" test period. The preliminary test period and the test period itself are to be conducted within the test tolerances given in section A4.2.3.3. In some cases, the preliminary defrost cycle may be manually induced, however, it is important that the normally operating controls govern the defrost termination in all cases. For units containing defrost controls which are likely to cause defrost at intervals less than one hour when the unit is operating at the required test conditions, the preliminary test period shall start at the termination of a defrost cycle which automatically occurs and shall end at the termination of the next automatically occurring defrost cycle. For units containing defrost controls which are likely to cause defrost at intervals exceeding one hour when operating at the required test condition, the preliminary test period consists of "heating-only" preliminary operation for at least one hour, after which a defrost may be manually or automatically induced. The test period then begins at the termination of this defrost cycle and ends at the termination of the next automatically occurring defrost cycle. If the unit has not undergone a defrost after 12 hours, then the tests shall be concluded and the results calculated for this 12-hour period. For units which turn the indoor fan off during defrost the indoor supply duct shall be blocked during all defrost cycles to prevent natural or forced convection through the indoor unit. During defrost, resistance heaters normally installed with the unit shall be prevented from operating.

For units with variable-speed compressors, the frost accumulation test at the intermediate speed shall be conducted such that the unit will operate at a constant, intermediate compressor speed (k = i) as determined in section A4.1.1.4. The following two equations may be used in lieu of the frost accumulation test for variable-speed.

$$\text{(a)}\quad Q_{def}^{k=2}(35) = 0.90 \times [Q_{ss}^{k=2}(17) + (Q_{ss}^{k=2}(47) - Q_{ss}^{k=2}(17)) \times (35 - 17)/(47 - 17)]$$

$$\text{(b)}\quad E_{def}^{k=2}(35) = 0.985 \times [E_{ss}^{k=2}(17) + (E_{ss}^{k=2}(47) - E_{ss}^{k=2}(17)) \times (35\text{-}17)/(47\text{-}17)]$$

A4.2.1.4 *Test procedures for the low temperature test.* Where applicable, the High Temperature Test preparation and performance requirements shall also be used in the Low Temperature Test. The test room reconditioning equipment shall first be operated in a steady-state manner for at least one-half hour at equilibrium and at the specified test conditions. The unit shall then undergo a defrost, either automatic or manually induced. It is important that the unit terminate the defrost sequence by the action of its own defrost controls. The defrost controls are to remain at the same setting as specified in A4.2.1.3. At a time no earlier than 10 minutes after defrost termination, the test shall start. Test duration is one-half hour. For all units, defrost should be prevented during the one-half test period.

A4.2.2 *Test instrumentation.*

A4.2.2.1 *Test instrumentation for the high temperature test.* The indoor air flow rate shall be determined as described in Section 7.8 through 7.8.3 of ASHRAE Standard 37-88. This requires the construction of an air receiving chamber and discharge chamber separated by partition in which one or more nozzles are located. The receiving chamber is connected to the indoor air discharge side of the test specimen through a short plenum. The exhaust side of the air flow rate measuring device contains an exhaust fan with some means to vary its capacity to obtain the desired external resistance to air flow rate. The exhaust side is then left open to the test room or is ducted through a conditioning apparatus and then back to the test specimen inlet. The static pressure across the nozzles, the velocity pressure, and the static pressure measurements at the nozzle throat shall be measured with manometers which will result in errors which are no greater than ±1.0 percent of indicated value and having minimum scale divisions not exceeding 2.0 percent of the reading. Static pressure and temperature measurements must be taken at the nozzle throat in order to obtain density of the air. The areas of the nozzles shall be determined by measuring their diameter with an error no greater than ±0.2 percent in four places approximately 45 degrees apart around the nozzle in each of two places through the nozzle throat, one at the outlets and the others in the straight section near the radius. The energy usage of the compressor, indoor and outdoor fan, and all other equipment components shall be measured with a watt-hour meter which is accurate to within ±0.5 percent of the quantity measured. Measurements of the air temperature entering and leaving the indoor coil or the difference between these two shall be made in accordance with the requirements of ASHRAE Standard 41.1-86 "Standard Method for Temperature Measurement." These temperatures shall be continuously recorded with instrumentation having a total system accuracy with ±0.3 F of indicated value and a response time of 2.5 seconds or less, upstream of the static pressure tap on the inlet and downstream of the static pressure taps on the outlet. The indoor and outdoor dry-bulb temperatures shall be continuously recorded with instrumentation which will result in an error no greater than ±0.3 F of indicated value. The outdoor wet-bulb temperature shall be continuously recorded. Static pressure measurements in the ducts and across the unit shall be made in accordance with Section 6.4 of ASHRAE Standard 37-88 using equipment which will result in an error no greater than ±0.01 inch of water. Static pressure measurements shall be made and recorded at 5 minute intervals. All other data not continuously recorded shall be recorded at 10 minute intervals.

A4.2.2.2 *Test instrumentations for the cycling test.* The air flow rate during the on-period of the Cyclic Test shall be the same agreement within ±1. percent as the air flow rate measured during the previously conducted High Temperature Test. All other instrumentation requirements are identical to A4.2.2.1.

A4.2.2.3 *Test instrumentation for the frost accumulation test.* The air flow rate for the Frost Accumulation Test shall be the same as described in A4.2.2.1. The indoor-side dry-bulb temperature and outdoor-side dry-bulb temperature shall be continuously recorded with instrumentation having a total system accuracy within ±0.3 F of indicated value. The outdoor dew point temperature shall be determined with an error no greater than ±0.5 F of indicated value using continuously recording instrumentation. All other data shall be recorded at 10 minute intervals during the heating cycle. Defrost initiation, termination and complete test cycle time (from defrost termination to defrost termination) shall be recorded. Defrost initiation is defined as the actuation (either automatically or manually) of the controls normally installed with the unit which cause it to alter its normal heating operation in order to eliminate possible accumulations of frost on the outdoor coil. Defrost termination occurs when the controls normally within the unit are actuated to change from defrost operation to normal heating operation. Provisions should be made so that instrumentation is capable of recording the cooling done during defrost as well as the total electrical energy usage during defrost. These data and the continuously recorded data need be the only data obtained during defrost.

4.2.2.4 *Test instrumentation for the low temperature test.* Instrumentation for the Low Temperature Test is identical to that of the High Temperature Test described in section A4.2.2.1.

A4.2.3 *Test tolerances.*

A4.2.3.1 *Test tolerances for the high temperature test.* All tests shall be conducted within the tolerances specified in Section A6.2.1. Variation greater than those given shall invalidate the test. The heating capacity results by the indoor Air Enthalpy Method shall agree within 6 percent of the value determined by any other simultaneously conducted capacity test in order for the test to be valid.

A4.2.3.2 *Test tolerances for the cyclic test.* The test condition tolerances and test operating tolerances for the on-period portion of the test cycle are specified in Section A6.2.2. Variations exceeding any specified test tolerances shall invalidate the test results.

A4.2.3.3 *Test tolerances for the frost accumulation test.* Test condition and test operating tolerances for Frost Accumulation Tests are specified in Section A6.2.3. Test operating tolerances during heating applies when the unit is in the heating mode, except for the first 5 minutes after the termination of a defrost cycle. Test operating tolerance during defrost applies during a defrost cycle and during the first 5 minutes after defrost termination when the unit is operating in the heating mode. In determining whether the test condition tolerances are met, only the heating portion of the test period shall be used n calculating the average values. Variations exceeding the tolerances presented in Section A6.2.3 shall invalidate the test.

A4.2.3.4 *Test tolerances for the low temperature test.* During the test period for the Low Temperature Test, the operating conditions shall be within the tolerances specified in Section A6.2.1.

A4.3 *Testing procedures for air source units which provide both heating and cooling.* The testing procedures for units which provide both heating and cooling shall be the same as those specified in Sections A4.1 and A4.2. Also during the off-period of the dry-coil cooling test (test D), the switch-over valve shall remain in the cooling mode, unless the controls normally supplied with the unit are designed to reverse it, in which case the controls shall operate the valve. During the off-period of the cyclic test at 47 F, the switch-over valve shall remain in the heating mode, unless the controls normally supplied with the unit are designed to reverse it, in which case the controls shall operate the valve.

A5.0 *Calculations for performance factors.*

A5.1 *Calculations of seasonal energy efficiency ratios (SEER) in air-source units.* The testing data and results required to calculate the seasonal energy efficiency ratio (SEER) in Btu's per watt-hour shall include the following:

(i) Cooling capacities (Btuh) from tests A and B and, if applicable, the cooling capacity (Btuh) from test C and the total cooling done from test D (Btu's).

Q_{ss}^* (95F)
Q_{ss}^* (82F)
Q_{ss}, dry
Q_{cyc}, dry

(ii) Electrical power input to all components and controls (watts) from tests A, B, and, if applicable, the electrical power input to all components and controls (watts) from test C and the electrical usage (watt-hour) from test D.

E_{ss}^* (95F)
E_{ss}^* (82F)
E_{ss}^*, dry
E_{cyc}, dry

(iii) Indoor air flow rate (SCFM) and external resistance to indoor air flow (inches of water W.C.).

(iv) Air temperatures (F)

Outdoor dry bulb
Outdoor wet bulb
Indoor dry bulb
Indoor wet bulb

Where the cooling capacities Q_{ss}^* (95F), from test A, Q_{ss}^* (82), from test B, and Q_{ss}, dry, from test C, are calculated using the equation specified in section 7.3.3 of ASHRAE Standard 37-88. The total cooling done, Q_{cyc}, dry from test D, is calculated using equation (1) below.

Units which do not have indoor air circulating fans furnished as part of the model shall have their measured total cooling capacities adjusted by subtracting 1250 Btuh per 1,000 CFM of measured indoor air flow and adding to the total steady-state electrical power input 365 watts per 1,000 CFM of measured indoor air flow.

Energy efficiency ratios from tests A, B, and C, EER_A, EER_B, EER_C, dry respectively, are each calculated as the ratio of the total cooling capacity in Btuh to the total electrical power input in watts.

Units which do not have indoor air circulating fans furnished as part of the model shall adjust their total cooling done and energy used in one complete cycle for the effect of circulating indoor air equipment power. The vaue to be used for the circulating indoor air equipment power shall be 1250 Btuh per 1,000 CFM of circulating indoor air. The energy usage required in one comlete cycle required for indoor air circulation is the product of the circulating indoor air equipment power and the duration of time in one cycle that the circulating indoor air equipment is on. The total cooling done shall then be the measured cooling in one complete cycle minus the energy usage required for indoor air circulation in one complete cycle. The total electrical energy usage shall be the sum of the energy usage required for indoor air circulation in one complete cycle and the energy used by the remaining equipment components (compressor(s), outdoor fan, crankcase heater, transformer(s), etc.) in one complete test cycle.

Energy efficiency ratio from tests D, EER_{cyc}, dry is calculated as the ratio of the total cooling done in Btu's to the total electrical energy usage in watt-hours.

The results of the cyclic and steady-state dry-coil performance tests shall be used in the following (4) equations:

$$(1)\qquad Q_{cyc,\,dry}=\frac{60\times\dot{V}\times C_{pa}\times\Gamma}{[V_n'\times(1+W_n)]}$$

where

$Q_{cyc,\,dry}$ = Total cooling over a cycle consisting of one compressor "off" period and one compressor "on" period (Btu's).
$\dot{V}$ = Indoor air flow rate (cfm) at the dry-bulb temperature, humidity ratio, and pressure existing in the region of measurement.
C_{pa} = Specific heat at constant pressure of air-water mixture per pound of dry air, (Btu/lb - F)
V_n' = Specific volume of air-water mixture at the same dry-bulb temperature, humidity ratio, and pressure used in the determination of the indoor air flow rate (ft^3/lb).
W_n = Humidity ratio (lb/lb).

and Γ (hr.-F), which is described by the equation:

(2)

$$\Gamma=\int_{(\text{time indoor fan on})}^{(\text{time indoor fan off})}[T_{a1}(t)-T_{a2}(t)]\,dt$$

where

$T_{a1}(t)$ = Dry bulb temperature of air entering the indoor coil (F) at time (t).
$T_{a2}(t)$ = Dry-bulb temperature of air leaving the indoor coil (F) at time (t).

$$(3)\qquad CLF=\frac{Q_{cyc,\,dry}}{Q_{ss\,dry}\times\tau}$$

where

CLF = cooling load factor.
Q_{ss} dry = Total steady-state cooling capacity from test C (Btuh)
τ = Duration of time (hours) for one complete cycle consisting of one compressor "on" time and one compressor "off" time.

The preceding equations are then used in the following equation to calculate a degradation coefficient C_D rounded to the nearest .01.

$$(4)\qquad C_D=\frac{1-\dfrac{EER_{cyc,\,dry}}{EER_{ss,\,dry}}}{1-CLF}$$

where

$EER_{ss,\,dry}$ = Energy efficiency ratio from test C, (Btu/watt-hr).

A5.1.1 Method for calculating a SEER for units with single-speed compressor and single-speed condenser fans. The seasonal energy efficiency ratio for units employing single-speed compressors and single-speed condenser fans shall be based on the performance of test B and a method outlined in A2.1.1 and A3.1.1 to account for the cyclic performance.

The seasonal energy efficiency ratio in Btu's/watt-hour shall be determined by the equation:

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

where

EER_B = Energy efficiency ratio determined from test B as outlined in 2.1.1.
$PLF(0.5)$ = Part-load performance factor when cooling load factor = 0.5 as determined from the equation:

$$PLF(0.5)=1-0.5\times C_D$$

where

C_D = Is the degradation coefficient described in 2.1.1 or as calculated in equation (4) above.

A5.1.2 Method for calculating a SEER for units with single-speed compressors and multi-speed condenser fans. The seasonal energy efficiency ratio (SEER) for units employing single-speed compressors and multi-speed condenser fans shall be based on the energy efficiency ratio obtained for test B and the method outlined in A2.1.2 to account for the performance under cyclic conditions. The energy efficiency ratio for test B is obtained with the unit operating with the condenser fan speed which normally occurs at test B ambient conditions.

The seasonal energy efficiency ratio in Btu's/watt-hour shall be determined by the equation:

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

where

EER_B = energy efficiency ratio determined from test B in 2.8
$PLF(0.5)$ = Part-load performance factor as determined from the equation:

$$PLF(0.5)=1-0.5\times C_D$$

where

C_D = The degradation coefficient described in 2.1.2 or as calculated in equation (4) above.

A5.1.3 Method for calculating a SEER for units with two speed compressors or two compressors, or cylinder unloading. The calculation procedure described in this section shall be based on the performance of test A and B at each of the compressor speeds for two-speed compressor units, subject to the conditions on condenser fan speed described in A3.1.3.

Units operating with two compressors shall have the SEER calculated in the same manner as two-speed compressor units. The superscripted index $k=1$ (and the term "low-speed") designates the compressor that normally operates at an outdoor dry-bulb temperature of 82 F ad $k=2$ (and the term "high speed") denotes operation with both compressors.

In order to evaluate the steady-state capacity $Q_{ss}^{k}(T_j)$, and power input, $E_{ss}^{k}(T_j)$, at temperature T_j for each compressor speed, $k=1$, $k=2$, the results of tests A and B from A5.1 shall be used in the following equation:

$$Q_{ss}^{k}(T_j)=Q_{ss}^{k}(95\text{ F})+\frac{Q_{ss}^{k}(82\text{ F})-Q_{ss}^{k}(95\text{ F})}{95-82}[33-(5\times j)]$$

where

$Q_{ss}^{k}(95\text{ F})$ = steady-state capacity measured from test A as outlined in A2.1.3.
$Q_{ss}^{k}(82\text{ F})$ = Steady-state capacity measured from test B as outlined in A2.1.3.

$$E_{ss}^{k}(T_j)=E_{ss}^{k}(95\text{ F})+\frac{E_{ss}^{k}(82\text{ F})-E_{ss}^{k}(95\text{ F})}{95-82}[33-(5\times j)]$$

when

$E_{ss}^{k}(95\text{ F})$ = Electrical power input measured using test A as outlined in A2.1.3.

$E_{ss}^{k}(82\text{ F})$ = Electrical power input measured from using test A as outlined in A2.1.3.

The building cooling load BL (T_j) for the four cases described in section A5.1.3.1 through A5.1.3.4 shall be obtained from the following equation:

$$BL(T_j)=\frac{(5\times j)-3}{95-65}\times\frac{Q_{ss}^{k=2}(95\text{ F})}{1.1}$$

where

$Q_{ss}^{k=2}(95\text{ F})$ = Steady-state capacity measured from test A in A2.9 at the high compressor speed.

The value of the degradation coefficient $C_D^{k=1}$ for low compressor speed cycling and $C_D^{k=2}$ for high speed on/off compressor cycling is determined as described in section A2.1.3, or as calculated above in equation (1).

A5.1.3.1 Units operating at low compressor speed ($k=1$) for which the steady-state cooling capacity, $Q_{ss}^{k=1}(T_j)$, is greater than or equal to the building cooling load, BL (T_j), evaluate the following equations:

$$(1)\qquad X^{k=1}=\frac{BL(T_j)}{Q_{ss}^{k=1}(T_j)}$$

where

$X^{k=1}$ = Load factor.

$BL(T_j)$ = Building cooling load (Btuh) at temperature (T_j) from section A5.1.3.

$Q_{ss}^{k=1}(T_j)$ = Steady-state cooling capacity (Btuh) at temperatures (T_j) from section A5.1.3.

$$(2)\qquad \frac{Q(T_j)}{N}=X^{k=1}\times Q_{ss}^{k=1}(T_j)\times\frac{n_j}{N}$$

where

$\frac{Q(T_j)}{N}$ = ratio of total cooling (Btu) in temperature bin j to the number of temperature bin hours.

$\frac{n_j}{N}$ is the fractional number of hours in temperature bin j from A6.1.2.

$$(3)\qquad \frac{E(T_j)}{N}=\frac{X^{k=1}\times E_{ss}^{k=1}(T_j)}{PLF^{k=1}}\times\frac{n_j}{N}$$

where

$\frac{E(T_j)}{N}$ = ratio of Energy usage (watt-hr.) in temperature bin j to the number of temperature bin hours.

$$PLF^{k=1}=1-C_D^{k=1}(1-X^{k=1}).$$

Where C_D, the degradation coefficient as described in section A2.1.3 or as calculated above in equation (1).

A5.1.3.2 When a unit must alternate between high ($k=2$) and low ($k=1$) compressor speeds to satisfy the building cooling load at a temperature T_j, evaluate the following equations:

$$(1)\qquad X^{k=1}=\frac{Q_{ss}^{k=2}(T_j)-BL(T_j)}{Q_{ss}^{k=2}(T_j)-Q_{ss}^{k=1}(T_j)}$$

$$(2)\qquad X^{k=2}=1-X^{k=1}$$

$$(3)\qquad \frac{Q(T_j)}{N}=[X^{k=1}\times Q_{ss}^{k=1}(T_j)+X^{k=2}\times Q_{ss}^{k=2}(T_j)]\times\frac{n_j}{N}$$

$$(4)\qquad \frac{E(T_j)}{N}=[X^{k=1}\times E_{ss}^{k=1}(T_j)+X^{k=2}\times E_{ss}^{k=2}(T_j)]\times\frac{n_j}{N}$$

A5.1.3.3 When a unit must cycle on and off at high compressor speed ($k=2$) in order to satisfy the building cooling load at a temperature T_j, evaluate the equations provided in section A5.1.3.1 replacing ($k=1$) data with ($k=2$) data.

A5.1.3.4 When a unit operates continuously at high compressor speed ($k=2$) at an outdoor temperature T_j evaluate the following equations:

$$(1)\qquad \frac{Q(T_j)}{N}=Q_{ss}^{k=2}(T_j)\times\frac{n_j}{N}$$

$$(2)\qquad \frac{E(T_j)}{N}=E_{ss}^{k=2}(T_j)\times\frac{n_j}{N}$$

A5.1.3.5 Calculate the SEER in Btu's/watt-hr. using the values for the terms

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

and

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

as determined at each temperature bin according to the applicable conditions described in sections A5.1.3.1 through A3.1.3.4 as follows:

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

A5.1.4 Method for calculating a SEER for units with two speed compressor, two compressor or cylinder unloading capable of varying the sensible to total capacity (S/T) ratio. Multi-speed compressor and two-speed compressor units capable of varying the sensible to total capacity ratio (S/T) shall have the seasonal energy efficiency ratio determined as described in section A5.1.3. For such units, the mode of operation selected to determine the steady-state capacities $Q_{ss}^k(93)$, $Q_{ss}^k(82)$, $E_{ss}^k(93)$, $E_{ss}^k(82)$, and power inputs at each compressor speed $k = 1$, $k = 2$, for tests A and B is outlined in section A2.10.

A5.1.5 *Seasonal energy efficiency ratio for air-source units with triple-capacity compressors.* (Reserved)

A5.1.6 *Seasonal energy efficiency ratio for air-source units with variable-speed compressors.* For air-source units with variable-speed compressors, the seasonal energy efficiency ratio (SEER), shall be defined as follows:

$$\text{SEER} = \frac{\sum_{j=1}^{8} \frac{Q(T_j)}{N}}{\sum_{j=1}^{8} \frac{E(T_j)}{N}}$$

where the number of hours in the j^{th} temperature bin (n_j)/N is defined in Table A6.1.2.

The SEER shall be determined by evaluating three cases of the compressor operation. Case I is the same as specified in A5.1.3.1 with the exception that the quantities $Q_{ss}^{k=1}(T_j)$ and $E_{ss}^{k=1}(T_j)$ shall be calculated by the following equations:

$$Q_{ss}^{k=1}(T_j) = Q_{ss}^{k=1}(82\ \text{F}) + \frac{Q_{ss}^{k=1}(67\ \text{F}) - Q_{ss}^{k=1}(82\ \text{F})}{82 - 67} \times (82 - T_j)$$

$$E_{ss}^{k=1}(T_j) = E_{ss}^{k=1}(82\ \text{F}) + \frac{E_{ss}^{k=1}(67\ \text{F}) - E_{ss}^{k=1}(82\ \text{F})}{82 - 67} \times (82 - T_j)$$

Case II is when the compressor operates at any intermediate $(k = v)$ speed between the maximum $(k = 2)$ and minimum $(k = 1)$ speeds to satisfy the building cooling load. Evaluate the following equations:

$$Q_{ss}^{k=v}(T_j) = \text{BL}\ (T_j)$$

$$E_{ss}^{k=v}(T_j) = \frac{Q_{ss}^{k=v}(T_j)}{EER_{ss}^{k=v}(T_j)}$$

$$\frac{Q(T_j)}{N} = Q_{ss}^{k=v}(T_j) \times \frac{n_j}{N}$$

$$\frac{E(T_j)}{N} = E_{ss}^{k=v}(T_j) \times \frac{n_j}{N}$$

where

$E_{ss}^{k=v}(T_j)$ the electrical power input required by the unit to deliver capacity matching the building load at temperature T_j.

where

$Q_{ss}^{k=v}(T_j)$ = the capacity delivered by the unit matching the building load at temperature T_j.

$EER_{ss}^{k=v}(T_j)$ = the steady-state energy efficiency ratio at temperature T_j and an intermediate speed at which the unit capacity matches the building load.

Before the steady-state intermediate speed energy efficiency ratio, $EER_{ss}^{k=v}(T_j)$, can be calculated, the unit performance has to be evaluated at the compressor speed $(k = i)$ at which the intermediate speed test was conducted. The capacity of the unit at any temperature T_j when the compressor operates at the intermediate speed $(k = i)$ may be determined by:

$$Q_{ss}^{k=i}(T_j) = Q_{ss}^{k=i}(87) + M_Q\ (T_j - 87)$$

Where:

$Q_{ss}^{k=1}(87)$ = the capacity of the unit at 87 F determined by the intermediate cooling steady-state test.

M_Q = slope of the capacity curve for the intermediate compressor speed $(k = i)$

$$M_Q = \frac{Q_{ss}^{k=1}(82) - Q_{ss}^{k=1}(67)}{82 - 67} \times (1 - N_Q) + N_Q \times \frac{Q_{ss}^{k=2}(95) - Q_{ss}^{k=2}(82)}{95 - 82}$$

$$N_Q = \frac{Q_{ss}^{k=i}(87) - Q_{ss}^{k=1}(87)}{Q_{ss}^{k=2}(87) - Q_{ss}^{k=1}(87)}$$

Once the equation for $Q_{ss}^{k=i}(T_j)$ has been determined, the temperature where $Q_{ss}^{k=i} = BL(T_j)$ can be found. This temperature is designated as (T_{vc}). The electrical power input for the unit operating at the intermediate compressor speed $(k = i)$ and the temperature (T_{vc}) is determined by:

$$E_{ss}^{k=i}(T_{vc}) = E_{ss}^{k=i}(87) + M_E\ (T_{vc} - 87)$$

where:

$E_{ss}^{k=1}(87)$ = the electrical power input of the unit at 87 F determined by the intermediate cooling steady-state test

M_E = scope of the electrical power input curve for the intermediate compressor speed $(k = 1)$

$$M_E = \frac{E_{ss}^{k=1}(82) - E_{ss}^{k=1}(67)}{82 - 67} \times 1 - N_E) + N_E \frac{E_{ss}^{k=2}(95) - E_{ss}^{k=2}(82)}{95 - 82}$$

$$N_E = \frac{E_{ss}^{k=i}(87) - E_{ss}^{k=1}(87)}{E_{ss}^{k=2}(87) - E_{ss}^{k=1}(87)}$$

The energy efficiency ratio of the unit, $\text{EER}_{ss}(T_{vc})$, at the intermediate speed $(k = i)$ and temperature T_{vc} can be calculated by the equation:

$$EER_{ss}^{k=i}(T_{vc}) = \frac{Q_{ss}^{k=i}(T_{vc})}{E_{ss}^{k=i}(T_{vc})}$$

Similarly, energy efficiency ratios at temperature T_1 and T_2 can be calculated by the equations:

$$EER_{ss}^{k=1}(T_1) = \frac{Q_{ss}^{k=1}(T_1)}{E_{ss}^{k=1}(T_1)}$$

$$EER_{ss}^{k=2}(T_2) = \frac{Q_{ss}^{k=2}(T_2)}{E_{ss}^{k=2}(T_2)}$$

where:

T_1 = temperature at which the unit, operating at the minimum compressor speed, delivers capacity equal to the building load $(Q_{ss}^{k=1}(T_1) = BL(T_3))$, found by equating the capacity equation $[(Q_{ss}^{k=1}(T_j)]$ and building load equation $[BL\ (T_j)]$ in section A5.1.3 and solving for temperature.

T_2 = temperature at which the unit, operating at the maximum compressor speed, delivers capacity equal to the building load $(Q_{ss}^{k=2}(T_2) = BL\ (T_2))$, found by equating the capacity equation $[(Q_{ss}^{k=2}(T_j)]$ and the building equation $[BL(T_j)]$ in section A5.1.3 and solving for temperature.

$EER_{ss}^{k=1}(T_1)$ = the steady state energy efficiency ratio at the minimum compressor speed at temperature T_1.

$EER_{ss}^{k=2}(T_2)$ = the steady state energy efficiency ratio at the maximum compressor speed at temperature T_2.

$E_{ss}^{k=1}(T_1)$ = the electrical power input at the minimum compressor speed at temperature T_1, calculated by the eqution in section A5.1.3.

$E_{ss}^{k=2}(T_2)$ = the electrical power input at the maximum compressor speed at temperature T_2, calculated by the equation in section A5.1.3.

The energy efficiency ratio, $EER_{ss}^{k=v}(T_j)$, shall be calculated by the following equation:

$$EER_{ss}^{k=v}(T_j) = A + B \times T_j + C \times T_j^2$$

where coefficients A, B, and C shall be evaluated using the following calculation steps:

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

$$B = \frac{EER_{ss}^{k=1}(T_1) - EER_{ss}^{k=2}(T_2) - D\ (EER_{ss}^{k=1}(T_1) - EER_{ss}^{k=i}(T_{vc}))}{T_1 - T_2 - D \times (T_1 - T_{vc})}$$

$$C = \frac{EER_{ss}^{k=1}(T_1) - EER_{ss}^{k=2}(T_2) - B \times (T_1 - T_2)}{T_1^2 - T_2^2}$$

$$A = EER_{ss}^{k=2}(T_2) - B \times T_2 - C \times T_2^2$$

Case III is the same as specified in A5.1.3.4. The quantities $Q_{ss}^{k=1}(T_j)$ and $E_{ss}^{k=2}(T_j)$ and $E_{ss}^{k=2}(T_j)$ shall be calculated by the equations prescribed in A5.1.3.

A5.1.7 *Seasonal energy efficiency ratio for split-type ductless systems.* For split-type ductless systems, *SEER* shall be defined as specified in section A5.1.1 for each combination set of indoor coils to be used with a common outdoor unit.

A5.2 *Calculation of Heating Seasonal Performance Factors (HSPF) for Air-Source Units.*

The testing data and results required to calculate the heating seasonal performance factor (HSPF), in Btu's per watt-hr, shall include the following:

(i) Heating capacities (Btuh) from the indoor air enthalpy method for the High Temperature Tests, and the total heating done (Btu's) for the cyclic and frost accumulation tests.

$\dot{Q}_{ss}(47)$ or $\dot{Q}_{ss}(62)$.
$\dot{Q}_{ss}(17)$.
$\dot{Q}_{cyc}(47)$.
$Q_{DEF}(35)$.

(ii) Electrical power input to all components (watts) for the steady state tests, and the electrical usage (watt-hours) for the cyclic and frost accumulation tests

$E_{ss}(47)$ or $E_{ss}(62)$.
$E_{ss}(17)$.
$E_{cyc}(47)$.
$E_{DEF}(35)$.

(iii) Indoor air flow rate (SCFM) and external resistance to indoor air flow (inches of water W.C.)

(iv) Air temperature (F)
Outdoor dry bulb
Outdoor wet bulb or dew point
Indoor dry bulb and
Indoor wet bulb.

(v) Data as specified in Table 5 of ASHRAE Standard 37-88

Where the heating capacities $Q_{ss}(47)$, $Q_{ss}(62)$ and $Q_{ss}(17)$ and the indoor air flow rate are calculated using the equations specified in section 7.3.4 and 7.8.3 of ASHRAE standard 37-88. The total heating done, $Q_{cyc}(47)$ and $Q(_{DEF}35)$ are calculated using the equations below.

Units not having an indoor fan as part of the model tested shall add 1250 Btuh per 1,000 SCFM of indoor air handled to the measured capacity to obtain the total heating capacity, $Q_{ss}(17)$, $Q_{ss}(47)$, or $Q_{ss}(62)$, and add 365 watts per 1,000 SCFM of indoor air handled to the measured power to obtain the total power input, $E_{ss}(17)$, $E_{ss}(47)$, or $E_{ss}(62)$, to the unit.

The coefficients of performance (COP) for the High Temperature Tests $COP_{ss}(62)$ or $COP_{ss}(47)$, and Low Temperature Test, $COP_{ss}(17)$, are calculated as the ratio of the heating capacity in Btuh to the product of 3.413 and the power inputs to the indoor fan in watts and the power inputs to the remaining equipment components (including all controls) in watts.

Units which do not have indoor air circulating fans furnished as part of the model shall have their total heating done ($Q_{cyc}(47)$) and energy used $E_{cyc}(47)$ in one complete cycle adjusted for the effect of circulating indoor air equipment power. For units tested without an indoor fan as part of the model, $Q_{cyc}(47)$ shall be increased by a quantity of heat equal to the product of 1250 Btuh per 1,000 SCFM, the length of the on-period of the test cycle in hours, and the flow rate of indoor air circulated in units of 1,000 SCFM. The total energy usage, $E_{cyc}(47)$, shall be the sum of the energy usage required for air circulation during the test cycle and the energy used by the remaining equipment components (including all controls) during the test cycle. Units not having an indoor fan as part of the model tested, shall set the energy required for indoor air circulation equal to the quantity given by the product of 365 watts per 1000 SCFM, the length of the on-period of the test cycle in hours, and the rate of indoor air circulated in units of 1000 SCFM.

The cyclic coefficient of performance, $COP_{cyc}(47)$ is calculated as the ratio of the total heating done ($Q_{cyc}(47)$) in Btu's to the product of 3.413 Btu/watt-hour and the total energy usage ($E_{cyc}(47)$) in watt hours.

The net heating capacity, $\dot{Q}_{DEF}(35)$ Btuh, is the total net heating done over the test period (including any credit for the indoor fan heat) divided by the total length of the test period, in hours.

For air-source units that are equipped with "demand defrost control systems", the value for HSPF, as determined above shall be multiplied by an enhancement factor F_{def} to compensate for improved performance not measured in the Frost Accumulation Test. The factor F_{def} depends on the number of defrost cycles in a 12-hour period and should be calculated as follows:

$$F_{def} = 1 + 0.03 \times (1 - (T_{test} - 90)/(T_{max} - 90))$$

where:

F_{def} = demand defrost credit (used as a multiplier to HSPF)
T_{test} = time between defrost terminations in minutes or 90, (whichever is greater)
T_{max} = maximum time between defrosts allowed by controls, (in minutes or 720 (whichever is less))

For units tested without indoor fans, the value determined for $Q_{DEF}(35)$ below shall be increased by a quantity of heat equal to the product of 1250 Btuh per 1000 SCFM, the length of time in hours during the Frost Accumulation Test that there were indoor air circulating, and the average flow rate of indoor air circulated in units of 1000 SCFM.

The total energy usage, $E_{DEF}(35)$ shall be the sum of the energy usage required for indoor-air circulation during the test period and the energy used by the remaining equipment components during the test period. Units not having an indoor fan as part of the model tested, shall set the energy required for indoor air circulation equal to the quantity given by the product of 365 watts per 1000 SCFM, the length of time in hours during the Frost Accumulation Test that there was indoor air circulating, and the average flow rate of indoor air circulated in units of 1000 SCFM.

The actual heating done during the Cyclic Test, $Q_{cyc}(47)$, shall be determined using the following equation:

$$(1) \qquad Q_{cyc}(47) = \frac{60 \times \dot{V} \times C_{pa} \times \Gamma}{[V_n^1 \times (1 + W_n)]}$$

where

$\dot{V}$ = the flow rate during the on-period calculated in accordance with section 7.8.3 of ASHRAE Standard 37-88 in CFM.
C_{pa} = Specific heat at constant pressure of air-water mixture per pound of dry air. (Btu/lb-F).
V_n^1 = Specific volume of air-water mixture at the same dry-bulb temperature, humidity ratio, and pressure used in the determination of the indoor air flow rate (ft³/lb).
W_n = Humidity ratio (lb/lb).

and Γ (hr-F), which is described by the equation:

$$\Gamma = \int_{\text{(time indoor fan on)}}^{\text{(time indoor fan off)}} [T_{a2}(t) - T_{a1}(t)]dt$$

where

$T_{a1}(t)$ = Dry-bulb temperature of air entering the indoor coil (F) at time (t).
$T_{a2}(t)$ = Dry-bulb temperature of air leaving the indoor coil (F) at time (t).

The net heating, Q_{DEF} (35) in Btu's done during the test period shall be determined using the following equation:

$$(2) \qquad Q_{DEF}(35) = \frac{60 \times \bar{\dot{V}} \times C_{pa} \times \Gamma}{[V_n^1 \times (1 + W_n)]}$$

where

$\bar{\dot{V}}$ = the average of the air flow rate calculated at four or more time intervals throughout the heating phase of the test using the equation in section 7.8.3 of ASHRAE Standard 37-88.
C_{pa} = Specific heat at constant pressure of air-water mixture per pound of dry air. (Btu/lb-F)
V_n^1 = Specific volume of air-water mixture at the same dry-bulb temperature, humidity ratio, and pressure used in the determination of the indoor air flow rate (ft³/lb).
W_n = Humidity ratio (lb/lb).

and Γ (hr.-F), which is described by the equation:

$$\Gamma = \int_{\text{(time of defrost termination}}^{\text{(time of next defrost termination)}} \times [T_{a2}(t) - T_{a1}(t)]dt$$

where

$T_a^1(t)$ = Dry-bulb temperature of air entering the indoor coil (F) at time (t).
$T_a^2(t)$ = Dry-bulb temperature of air leaving the indoor coil (F) at time (t).

The cyclic degradation coefficient shall be calculated as follows:

$$(3) \qquad C_D = \frac{1 - \dfrac{COP_{cyc}(47)}{COP_{ss}(47)}}{1 - HLF}$$

where

C_D = the cyclic degradation coefficient rounded to the nearest .01
$COP_{cyc}(47)$ as defined above
$COP_{ss}(47)$ as defined above
HLF is the heating load factor calculated as follows:

$$HLF = \frac{Q_{cyc}(47)}{\dot{Q}_{ss}(47) \times \tau}$$

where

$Q_{cyc}(47)$ as defined above
$\dot{Q}_{ss}(47)$ as defined above
τ = Duration of time (hours) for one complete cycle consisting of one compressor "on" time and one compressor "off" time.

A5.2.1 Calculation of the heating seasonal performance factor (HSPF) for air-source heat pumps with single speed compressors.

For each climate region listed in section A6.2.4, and for design heating requirements equal to both the standardized minimum and maximum design heating requirements defined below, calculate the HSPF defined as:

*HSPF

$$= \frac{\sum_j \frac{n_j}{N} BL(T_j)}{\left[\sum_j \frac{n_j}{N} \frac{X(T_j)}{PLF(X)} \delta(T_j) \dot{E}(T_j) + \sum_j \frac{RH(T_j)}{N}\right]},$$

*These items have been corrected in accordance with a letter from the Department of Energy, dated December 7, 1980.

where

j = 1,2,3, . . . , n corresponds to the j^{th} temperature bin.

n = total number of non-zero temperature bins in the climatic region.

T_j = (67 − 5j) is the representative temperature of the j^{th} bin. (F)

Σ = indicated the quantity following the symbol is to be summed over all temperature bins.

$\frac{RH(T_j)}{N}$ = supplementary resistance heat term at temperature T_j required in those cases where the heat pump automatically turns off ($T_j < T_{off}$) or when it is needed to meet the balance of the building heating requirements. (watts).

$\frac{n_j}{N}$ = is the number of hours in the j^{th} temperature bin divided by $N \equiv \sum_j n_j$ and is referred to as the "fractional hours in j^{th} temperature bin".

3.413 = is a conversion factor which converts watt hours to Btu.

$BL(T_j)$ = building load at temperature T_j, Btuh

$\delta(T_j)$ = heat pump low temperature cut-out factor.

$X(T_j)$ = heat pump heating load factor.

$PLF(X)$ = heat pump part load factor.

The quantities $BL(T_j)$, $\delta(T_j)$, $X(T_j)$, $PLF(X)$ and

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

are defined by the following equations:

$$BL(T_j) = \left(\frac{65 - T_j}{65 - T_{OD}}\right)(C)(DHR)$$

where

(C) = 0.77 is a correction factor which tends to improve the agreement between calculated and measured building loads

(DHR) = the minimum and maximum design heating requirement which the heat pump is likely to encounter when installed in a residence, rounded off to the nearest standardized DHR in section A6.26 in Btuh

where

(minimum design heating requirement)

$$= \begin{cases} \dot{Q}_{ss}(47)\,\dfrac{(65 - T_{OD})}{60}, & \text{for regions I, II, III, IV, and VI} \\ \dot{Q}_{ss}(47), & \text{for region V} \end{cases}$$

and

maximum design heating requirement

$$= \begin{cases} 2\dot{Q}_{ss}(47)\,\dfrac{(65 - T_{OD})}{60}, & \text{for regions I, II, III, IV, and VI} \\ 2.2\dot{Q}_{ss}(47), & \text{for region V} \end{cases}$$

where

$\dot{Q}_{ss}(47)$ is the heat pump capacity measured during the High Temperature Test at 47 F

T_{OD} is the outdoor design temperature given in section A6.2.4

$$\delta(T_j) = \begin{cases} 0; & T_j \le T_{OFF} \text{ or } \dfrac{\dot{Q}(T_j)}{(3.413)(\dot{E}(T_j))} < 1 \\ \dfrac{1}{2}; & T_{OFF} < T_j \le T_{ON} \text{ and } \dfrac{\dot{Q}(T_j)}{(3.413)(\dot{E}(T_j))} \ge 1 \\ 1; & T_j > T_{ON} \text{ and } \dfrac{\dot{Q}(T_j)}{(3.413)(\dot{E}(T_j))} \ge 1 \end{cases}$$

$$X(T_j) = \begin{cases} \dfrac{BL(T_j)}{\dot{Q}(T_j)}; & \dot{Q}(T_j) \ge BL(T_j) \\ 1; & \dot{Q}(T_j) \le BL(T_j) \end{cases}$$

$$PLF(X) = 1 - C_D(1 - X(T_j))$$

$$\frac{RH(T_j)}{N} = \frac{[BL(T_j) - \dot{Q}(T_j)X(T_j)\delta(T_j)]\,\frac{n_j}{N}}{3.413}$$

where

T_{OFF} = the outdoor temperature that the compressor is automatically shut off at (If no such temperature exists, T_j is always greater than T_{OFF} and T_{ON}.)

T_{ON} = the outdoor temperature that the compressor is automatically turned on (if applicable) if designed for low temperature automatic shut-off.

C_D = degradation factor determined described in section A5.2 and A2.2.1.

In using the above equation to calculate HSPF, the heat pump capacity in Btuh, $\dot{Q}$, and the power in watts, $\dot{E}$, shall be obtained as follows:

$$\dot{Q}(T_j) = \begin{cases} \dot{Q}_{ss}(17) + \dfrac{(\dot{Q}_{ss}(47) - \dot{Q}_{ss}(17)) \times (T_j - 17)}{30}, & T_j \ge 45\text{F or } T_j \le 17\text{F} \\ \dot{Q}_{ss}(17) + \dfrac{(\dot{Q}_{DEF}(35) - \dot{Q}_{ss}(17)) \times (T_j - 17)}{18}, & 17\text{F} < T_j < 45\text{F} \end{cases}$$

$$\dot{E}(T_j) = \begin{cases} \dot{E}_{ss}(17) + \dfrac{(\dot{E}_{ss}(47) - \dot{E}_{ss}(17)) \times (T_j - 17)}{30}, & T_j \ge 45\text{F or } T_j \le 17\text{F} \\ \dot{E}_{ss}(17) + \dfrac{(\dot{E}_{DEF}(35) - \dot{E}_{ss}(17)) \times (T_j - 17)}{18}, & 17\text{F} < T_j < 45\text{F} \end{cases}$$

where $\dot{Q}_{ss}(47)$ and $\dot{E}_{ss}(47)$ and $\dot{Q}_{DEF}(35)$ and $\dot{E}_{ss}(47)$ and $\dot{Q}_{ss}(17)$ and $\dot{E}_{ss}(17)$ are the capacities (in Btuh) and powers (in watts), measured during the High Temperature Test, the Frost Accumulation test, and the Low Temperature Test, respectively.

For each of the six regions specified in section A6.2.5, calculate the heating seasonal performance factors and seasonal operating costs corresponding to the standardized maximum and minimum design heating requirements and for all other standardized design heating requirements (see section A6.2.6) between the maximum and the minimum.

A5.2.2 Calculation of the heating seasonal performance factor (HSPF) for air source heat pumps with a two-speed compressor, two compressors, or cylinder unloading.

For each climatic region listed in section A6.2.4, and for design heating requirements [illegible] the standardized minimum and maximum design heating requirements defined below, calculate the HSPF defined as:

$$HSPF = \frac{\sum_j \frac{n_j}{N} BL(T_j)}{\left[\sum_j \frac{E(T_j)}{N} + \sum_j \frac{RH(T_j)}{N}\right]}$$

where

Σ as defined in A5.2.1

$\frac{n_j}{N}$ as defined in A5.2.1

T_j as defined in A5.2.1

and $BL(T_j)$ is the building load at temperature T_j, in Btuh, calculated by:

$$BL(T_j) = \left(\frac{65 - T_j}{65 - T_{OD}}\right) \times (C) \times (DHR)$$

where

(C) = 0.77 is a correction factor which tends to improve the agreement between calculated and measured building loads

(DHR) = the minimum and maximum design heating requirement which the heat pump is likely to encounter when installed in a residence, rounded off to the nearest standardized DHR in section A6.2.6 in Btuh

where

(minimum design heating requirement)

$$= \begin{cases} \dot{Q}_{ss}^{k=2}(47)\,\dfrac{(65 - T_{OD})}{60}, & \text{for regions I, II, III, IV, and VI} \\ \dot{Q}_{ss}^{k=2}(47), & \text{for region V} \end{cases}$$

and

(maximum design heating requirement)

$$= \begin{cases} 2\dot{Q}_{ss}^{k=2}(47)\,\dfrac{(65 - T_{OD})}{60}, & \text{for regions I, II, III, IV and VI} \\ 2.2\dot{Q}_{ss}^{k=2}(47) & \text{for region V} \end{cases}$$

where

$Q_{ss}^{(k=2)}(47)$ is the heat pump capacity measured during the high temperature performance test at 47 F, with the unit operating at the high compressor speed or with both compressors in operation, in Btuh

T_{OD} is the outdoor design temperature given in section A6.2.4 in degrees F.

NOTE: The superscript (k=1) and (k=2) refer to the heat pump operating at low speed or single compressor operation and high speed or two compressor operation respectively.

$\frac{E(T_j)}{N}$ is the heat pump electrical energy usage in the j^{th} temperature bin divided by the total number of bin hours and is evaluated according to the four possible cases of heat pump operation denoted below in watts.

$\frac{RH(T_j)}{N}$ as defined in A5.2.1 and is evaluated according to the four possible cases of heat pump operation denoted below (in watts)

Case I.—Units operating at low compressor speed or with a single compressor, i.e., k=1, for which the building heating load, $BL(T_j)$ is less than or equal to the heating capacity, $\dot{Q}^{k=1}(T_j)$.

$$BL(T_j) \le \dot{Q}^{k=1}(T_j)$$

$$\frac{E(T_j)}{N} = \frac{\dot{E}^{k=1}(T_j)X^{k=1}(T_j)\delta'(T_j)}{PLF^{k=1}}\,\frac{n_j}{N}$$

$$\frac{RH(T_j)}{N}=\frac{\frac{n_j}{N}\,BL(T_j)[1-\delta'(T_j)]}{3.413}$$

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

$$PLF^{k=1}=1-C_D^{k=1}(1-X_j^{k=1})$$

$$\delta'(T_j)=\begin{cases}0; & T_j\le T_{OFF}\\ \frac{1}{2}; & T_{OFF}<T_j\le T_{ON}\\ 1; & T_j>T_{ON}\end{cases}$$

Case II.—Units alternating between high speed or two compressor operation ($k=2$) and low speed or single compressor operation ($k=1$) to satisfy the building heating load at temperature T_j.

$$\dot{Q}^{k=1}(T_j)<BL(T_j)<\dot{Q}^{k=2}(T_j)$$

$$\frac{E(T_j)}{N}=[\dot{E}^{k=1}(T_j)X^{k=1}(T_j)+\dot{E}^{k=2}(T_j)X^{k=2}(T_j)]\delta'(T_j)\,\frac{n_j}{N}$$

$$\frac{RH(T_j)}{N}=\frac{\frac{n_j}{N}\,BL(T_j)[1-\delta'(T_j)]}{3.413}$$

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

$$X^{k=2}(T_j)=1-X^{k=1}(T_j)$$

$$\delta'(T_j)\begin{cases}0; & T_j\le T_{OFF}\\ \frac{1}{2}; & T_{OFF}<T_j\le T_{ON}\\ 1; & T_j>T_{ON}\end{cases}$$

Case III.—Units cycling on and off at high compressor speed or cycling both compressors on and off together ($k=2$) in order to satisfy the building heating load at temperature T_j.

$$\dot{Q}^{k=1}(T_j)<BL(T_j)<\dot{Q}^{k=2}(T_j)$$

$$\frac{E(T_j)}{N}=\frac{\dot{E}^{k=2}(T_j)X^{k=2}(T_j)\delta''(T_j)\frac{n_j}{N}}{PLF^{k=2}}$$

$$\frac{RH(T_j)}{N}=\frac{\frac{n_j}{N}BL(T_j)[1-\delta''(T_j)]}{3.413}$$

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

$$PLF^{k=2}=1-C_D^{k=2}(1-X^{k=2}(T_j))$$

$$\delta''(T_j)=\begin{cases}0; & T_j\le T_{OFF}\\ \frac{1}{2}; & T_{OFF}<T_j\le T_{ON}\\ 1; & T_j>T_{ON}\end{cases}$$

Case IV.—Units operating continuously at high compressor speed or with both compressors in continuous operation ($k=2$) in order to satisfy the building heating load at temperature T_j.

$$BL(T_j)\ge\dot{Q}^{k=2}(T_j)$$

$$\frac{E(T_j)}{N}=\dot{E}^{k=2}(T_j)X^{k=2}(T_j)\delta''(T_j)\,\frac{n_j}{N}$$

$$\frac{RH(T_j)}{N}=\frac{[BL(T_j)-\dot{Q}^{k=2}(T_j)X^{k=2}(T_j)\delta''(T_j)]\,\frac{n_j}{N}}{3.413}$$

$$X^{k=2}(T_j)=1.0$$

$$\delta''(T_j)=\begin{cases}0; \; T_j\le T_{OFF} \text{ or } \dfrac{\dot{Q}^{k=2}(T_j)}{(3.413)(\dot{E}^{k=2}(T_j))}<1\\ \frac{1}{2}; \; T_{OFF}<T_j\le T_{ON} \text{ and } \dfrac{\dot{Q}^{k=2}(T_j)}{(3.413)(\dot{E}^{k=2}(T_j))}\ge1\\ 1; \; T_j\ge T_{ON} \text{ and } \dfrac{\dot{Q}^{k=2}(T_j)}{(3.413)(\dot{E}^{k=2}(T_j))}\ge1\end{cases}$$

Where in each of the above cases

$X(T_j)$ = heat pump heating load factor.
PLF = heat pump part load factor (not required for cases II and IV).
$\delta(T_j)$ = heat pump low temperature cut-out factor.
T_{off} as defined in A5.2.1
T_{on} as defined in A5.2.1
$C_D^{(k=2)}$ = the part load degradation factor described in section A2.2.1 and A5.2 for the unit cycling at high compressor speed or with both compressors simultaneously cycling (if applicable).
$C_D^{(k=1)}$ = the part load degradation factor described in section A2.2.1 and A5.2 for the unit cycling at low compressor speed or with the single compressor that normally operates at low heating loads.

$$\dot{Q}^{k=1}(T_j)=\begin{cases}\dot{Q}_H^{k=1}(47)+\dfrac{(\dot{Q}_H^{k=1}(62)-\dot{Q}_H^{k=1}(47))\times(T_j-47)}{15}; & T_j\ge40\text{ F}\\ \dot{Q}_H^{k=1}(17)+\dfrac{(\dot{Q}_{DEF}^{k=1}(35)-\dot{Q}_H^{k=1}(17))\times(T_j-17)}{18}; & 17\text{ F}\le T_j<40\text{ F}\\ \dot{Q}_H^{k=1}(17)+\dfrac{(\dot{Q}_H^{k=1}(47)-\dot{Q}_H^{k=1}(17))\times(T_j-17)}{30}; & T_j<17\text{ F}\end{cases}$$

$$\dot{Q}^{k=2}(T_j)=\begin{cases}\dot{Q}_H^{k=2}(17)+\dfrac{(\dot{Q}_H^{k=2}(47)-\dot{Q}_H^{k=2}(17))\times(T_j-17)}{30}; & T_j\ge45\text{ F or }T_j\le17\text{ F}\\ \dot{Q}_H^{k=2}(17)+\dfrac{(\dot{Q}_{DEF}^{k=2}(35)-\dot{Q}_H^{k=2}(17))\times(T_j-17)}{18}; & 17\text{ F}<T_j<45\text{ F}\end{cases}$$

$$\dot{E}^{k=1}(T_j)=\begin{cases}\dot{E}_H^{k=1}(47)+\dfrac{(\dot{E}_H^{k=1}(62)-\dot{E}_H^{k=1}(47))\times(T_j-47)}{15}; & T_j\ge40\text{ F}\\ \dot{E}_H^{k=1}(17)+\dfrac{(\dot{E}_{DEF}^{k=1}(35)-\dot{E}_H^{k=1}(17))\times(T_j-17)}{18}; & 17\text{ F}\le T_j<40\text{ F}\\ \dot{E}_H^{k=1}(17)+\dfrac{(\dot{E}_H^{k=1}(47)-\dot{E}_H^{k=1}(17))\times(T_j-17)}{30}; & T_j<17\text{ F}\end{cases}$$

$$\dot{E}^{k=2}(T_j)=\begin{cases}\dot{E}_H^{k=2}(17)+\dfrac{(\dot{E}_H^{k=2}(47)-\dot{E}_H^{k=2}(17))\times(T_j-17)}{30}; & T_j\ge45\,\cancel{\text{F or }T_j\le17}\text{ F}\\ \dot{E}_H^{k=2}(17)+\dfrac{(\dot{E}_{DEF}^{k=2}(35)-\dot{E}_H^{k=2}(17))\times(T_j-17)}{18}; & 17\text{ F}<T_j<45\text{ F}\end{cases}$$

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For each of the six regions specified in section A6.2.5, calculate the heating seasonal performance factors and seasonal operating costs corresponding to the standardized maximum and minimum design heating requirements and for all other standardized design heating requirements (see section A6.2.6) between the maximum and the minimum.

A5.2.3 *Heating seasonal performance factor for air-source units with triple-capacity compressors.* (Reserved)

A5.2.4 *Heating seasonal performance factor for units with variable-speed compressors.* For units with variable-speed compressors, the heating seasonal performance factor (HSPF) is defined by the following equation:

$$HSPF=\frac{\sum_j\frac{n_j}{N}BL(T_j)}{\left(\sum_j\frac{E(T_j)}{N}+\sum_j\frac{RH(T_j)}{N}\right)}$$

$$\frac{RH(T_j)}{N}=\frac{\frac{n_j}{N}\,BL(T_j)[1-\delta'(T_j)]}{3.413}$$

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

$$PLF^{k=1}=1-C_D^{k=1}(1-X_j^{k=1})$$

$$\delta'(T_j)=\begin{cases}0; & T_j\le T_{OFF}\\ \frac{1}{2}; & T_{OFF}<T_j\le T_{ON}\\ 1; & T_j>T_{ON}\end{cases}$$

Case II.—Units alternating between high speed or two compressor operation ($k=2$) and low speed or single compressor operation ($k=1$) to satisfy the building heating load at temperature T_j.

$$\dot{Q}^{k=1}(T_j)<BL(T_j)<\dot{Q}^{k=2}(T_j)$$

$$\frac{E(T_j)}{N}=[\dot{E}^{k=1}(T_j)X^{k=1}(T_j)+\dot{E}^{k=2}(T_j)X^{k=2}(T_j)]\delta'(T_j)\,\frac{n_j}{N}$$

$$\frac{RH(T_j)}{N}=\frac{\frac{n_j}{N}\,BL(T_j)[1-\delta'(T_j)]}{3.413}$$

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

$$X^{k=2}(T_j)=1-X^{k=1}(T_j)$$

$$\delta'(T_j)\begin{cases}0; & T_j\le T_{OFF}\\ \frac{1}{2}; & T_{OFF}<T_j\le T_{ON}\\ 1; & T_j>T_{ON}\end{cases}$$

Case III.—Units cycling on and off at high compressor speed or cycling both compressors on and off together ($k=2$) in order to satisfy the building heating load at temperature T_j.

$$\dot{Q}^{k=1}(T_j)<BL(T_j)<\dot{Q}^{k=2}(T_j)$$

$$\frac{E(T_j)}{N}=\frac{\dot{E}^{k=2}(T_j)X^{k=2}(T_j)\delta''(T_j)\frac{n_j}{N}}{PLF^{k=2}}$$

$$\frac{RH(T_j)}{N}=\frac{\frac{n_j}{N}BL(T_j)[1-\delta''(T_j)]}{3.413}$$

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

$$PLF^{k=2}=1-C_D^{k=2}(1-X^{k=2}(T_j))$$

$$\delta''(T_j)=\begin{cases}0; & T_j\le T_{OFF}\\ \frac{1}{2}; & T_{OFF}<T_j\le T_{ON}\\ 1; & T_j>T_{ON}\end{cases}$$

Case IV.—Units operating continuously at high compressor speed or with both compressors in continuous operation ($k=2$) in order to satisfy the building heating load at temperature T_j.

$$BL(T_j)\ge\dot{Q}^{k=2}(T_j)$$

$$\frac{E(T_j)}{N}=\dot{E}^{k=2}(T_j)X^{k=2}(T_j)\delta''(T_j)\,\frac{n_j}{N}$$

$$\frac{RH(T_j)}{N}=\frac{[BL(T_j)-\dot{Q}^{k=2}(T_j)X^{k=2}(T_j)\delta''(T_j)]\,\frac{n_j}{N}}{3.413}$$

$$X^{k=2}(T_j)=1.0$$

$$\delta''(T_j)=\begin{cases}0; & T_j\le T_{OFF}\ \text{or}\ \dfrac{\dot{Q}^{k=2}(T_j)}{(3.413)(\dot{E}^{k=2}(T_j))}<1\\ \frac{1}{2}; & T_{OFF}<T_j\le T_{ON}\ \text{and}\ \dfrac{\dot{Q}^{k=2}(T_j)}{(3.413)(\dot{E}^{k=2}(T_j))}\ge1\\ 1; & T_j\ge T_{ON}\ \text{and}\ \dfrac{\dot{Q}^{k=2}(T_j)}{(3.413)(\dot{E}^{k=2}(T_j))}\ge1\end{cases}$$

Where in each of the above cases

$X(T_j)$ = heat pump heating load factor.

PLF = heat pump part load factor (not required for cases II and IV).

$\delta(T_j)$ = heat pump low temperature cut-out factor.

T_{off} as defined in A5.2.1

T_{on} as defined in A5.2.1

$C_D^{(k=2)}$ = the part load degradation factor described in section A2.2.1 and A5.2 for the unit cycling at high compressor speed or with both compressors simultaneously cycling (if applicable).

$C_D^{(k=1)}$ = the part load degradation factor described in section A2.2.1 and A5.2 for the unit cycling at low compressor speed or with the single compressor that normally operates at low heating loads.

$$\dot{Q}^{k=1}(T_j)=\begin{cases}\dot{Q}_h^{k=1}(47)+\dfrac{(\dot{Q}_h^{k=1}(62)-\dot{Q}_h^{k=1}(47))\times(T_j-47)}{15}; & T_j\ge40\text{ F}\\ \dot{Q}_h^{k=1}(17)+\dfrac{(\dot{Q}_{DEF}^{k=1}(35)-\dot{Q}_h^{k=1}(17))\times(T_j-17)}{18}; & 17\text{ F}\le T_j<40\text{ F}\\ \dot{Q}_h^{k=1}(17)+\dfrac{(\dot{Q}_h^{k=1}(47)-\dot{Q}_h^{k=1}(17))\times(T_j-17)}{30}; & T_j<17\text{ F}\end{cases}$$

$$\dot{Q}^{k=2}(T_j)=\begin{cases}\dot{Q}_h^{k=2}(17)+\dfrac{(\dot{Q}_h^{k=2}(47)-\dot{Q}_h^{k=2}(17))\times(T_j-17)}{30}; & T_j\ge45\text{ F or }T_j\le17\text{ F}\\ \dot{Q}_h^{k=2}(17)+\dfrac{(\dot{Q}_{DEF}^{k=2}(35)-\dot{Q}_h^{k=2}(17))\times(T_j-17)}{18}; & 17\text{ F}<T_j<45\text{ F}\end{cases}$$

$$\dot{E}^{k=1}(T_j)=\begin{cases}\dot{E}_h^{k=1}(47)+\dfrac{(\dot{E}_h^{k=1}(62)-\dot{E}_h^{k=1}(47))\times(T_j-47)}{15}; & T_j\ge40\text{ F}\\ \dot{E}_h^{k=1}(17)+\dfrac{(\dot{E}_{DEF}^{k=1}(35)-\dot{E}_h^{k=1}(17))\times(T_j-17)}{18}; & 17\text{ F}\le T_j<40\text{ F}\\ \dot{E}_h^{k=1}(17)+\dfrac{(\dot{E}_h^{k=1}(47)-\dot{E}_h^{k=1}(17))\times(T_j-17)}{30}; & T_j<17\text{ F}\end{cases}$$

$$\dot{E}^{k=2}(T_j)=\begin{cases}\dot{E}_h^{k=2}(17)+\dfrac{(\dot{E}_h^{k=2}(47)-\dot{E}_h^{k=2}(17))\times(T_j-17)}{30}; & T_j\ge45\text{ F or }T_j\le17\text{ F}\\ \dot{E}_h^{k=2}(17)+\dfrac{(\dot{E}_{DEF}^{k=2}(35)-\dot{E}_h^{k=2}(17))\times(T_j-17)}{18}; & 17\text{ F}<T_j<45\text{ F}\end{cases}$$

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For each of the six regions specified in section A6.2.5, calculate the heating seasonal performance factors and seasonal operating costs corresponding to the standardized maximum and minimum design heating requirements and for all other standardized design heating requirements (see section A6.2.6) between the maximum and the minimum.

A5.2.3 *Heating seasonal performance factor for air-source units with triple-capacity compressors.* (Reserved)

A5.2.4 *Heating seasonal performance factor for units with variable-speed compressors.* For units with variable-speed compressors, the heating seasonal performance factor (HSPF) is defined by the following equation:

$$HSPF=\frac{\sum_j\frac{n_j}{N}BL(T_j)}{\left(\sum_j\frac{E(T_j)}{N}+\sum_j\frac{RH(T_j)}{N}\right)}$$

A5.3.2 Calculation of representative regional annual performance factors (APF_R) for each region and for each standardized design heating requirement.

$$(APF_R) = \frac{(CLH_R)(\dot{Q}_{ss}^{k}(95F)) + (HLH_R)(DHR)(C)}{\frac{(CLH_R)(\dot{Q}_{ss}^{k}(95F))}{SEER} + \frac{(HLH_R)(DHR)(C)}{HSPF}}$$

$(Q_{ss}^{k}(95F))$ is defined in A5.1.
(DHR) is defined in A5.2.2.
(C) is defined in A5.2.2.

where

(CLH_R) is the representative cooling load hours for each heating load hours region, as determined in section A6.3.
(HLH_R) is the representtive heating load hours for each region as determined in section A6.2.5.
$(SEER)$ is the seasonal energy efficiency ratio as determined in section A5.1.
$(HSPF)$ is the heating seasonal performance factor as determined in section A5.2 for each region and for each standardized design heating requirement within each region.

where the regions are listed in section A6.2.5 and, the standardized design heating requirements within the regions are determined in sections A5.2 and A6.2.6.

A5.4 *Calculations of Seasonal Energy Efficiency Ratio for Water Source Units Which Provide Both Heating and Cooling.* (Reserved)

A5.5 *Calculation of Heating Seasonal Performance Factor for Water Source Units Which Provide Both Heating and Cooling.* (Reserved)

A5.6 *Calculation of Annual Performance Factor For Water Source Units Which Provide Both Heating and Cooling.* (Reserved)

A6.0 Reference Material

A6.1 Cooling reference material.

A6.1.1 -Test operating and test condition tolerance for cyclic dry-coil tests

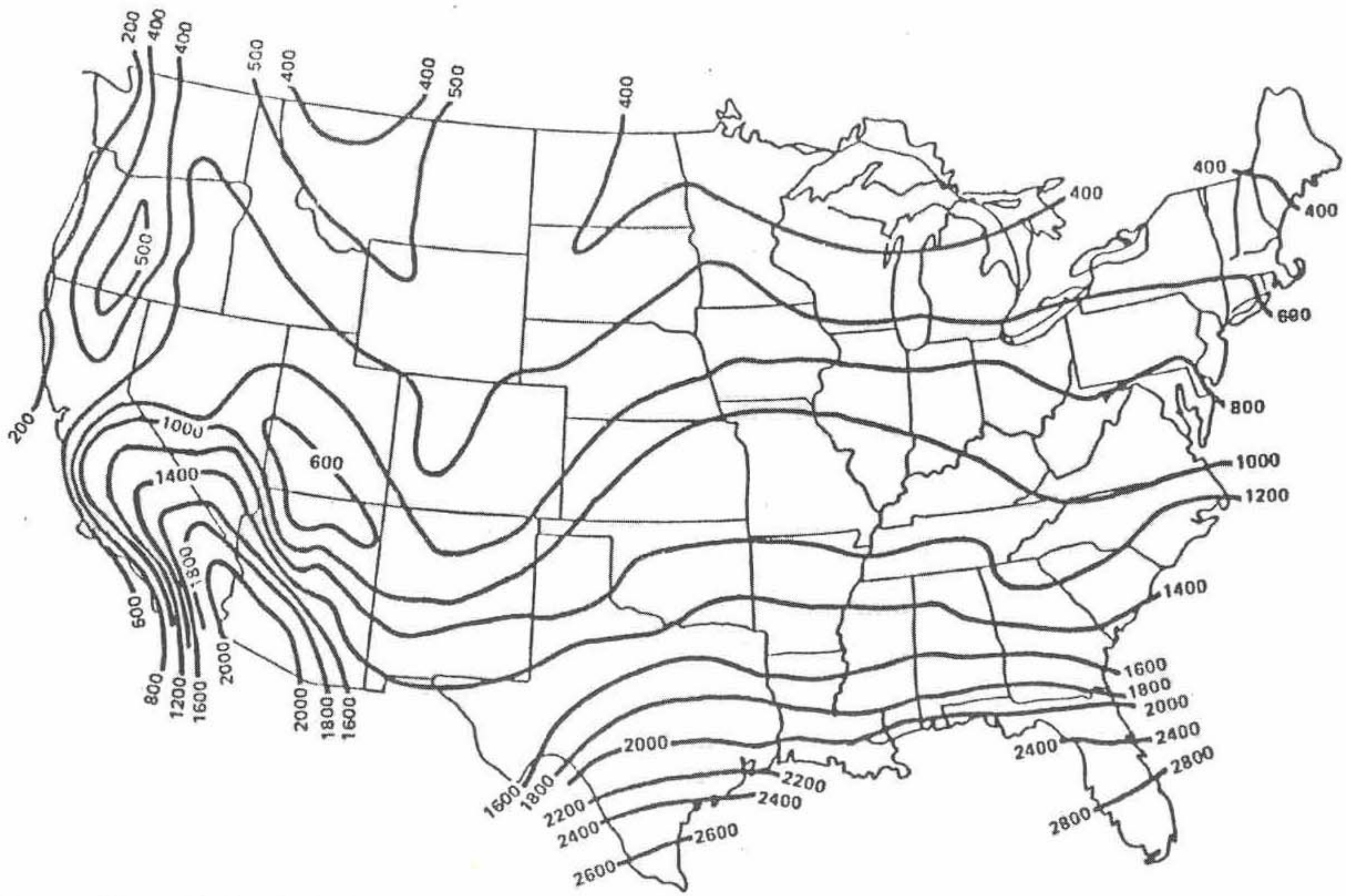
Readings, remarks	Test operating tolerance (total observed range)	Test condition tolerance (variation of average from specified test condition)
Outdoor dry-bulb air temperature, Fahrenheit:		
Entering....................	2.0	0.5
Indoor dry-bulb air temperature, Fahrenheit:		
Entering....................	2.0	0.5
Indoor wet-bulb air temperature, Fahrenheit:		
Entering....................	(*)	(*)
After the 1st 30 Sec after compressor startup.		
External resistance to airflow, inches water....	0.05	0.02
Nozzle pressure drops, percent of reading........	2.0	....
Electrical voltage inputs to the test unit, percent.	2.0	....

* Shall at no time exceed that value of the wet-bulb temperature which results in the production of condensate by the indoor coil at the dry-bulb temperature existing for the air entering the indoor portion of the unit.

A6.1.2 -Distribution of fractional hours in temperature bins to be used for calculation of the SEER for 2-speed compressor and 2-compressor units

Bin temperature range (degrees Fahrenheit)	Representative temperature bin for (degrees Fahrenheit)	Fraction of total temperature bin hours n_j/N
Bin No., j:		
1..... 65 to 69...	67	.214
2..... 70 to 74...	72	.231
3..... 75 to 79...	77	.216
4..... 80 to 84...	82	.161
5..... 85 to 89...	87	.104
6..... 90 to 94...	92	.052
7..... 95 to 99...	97	.018
8..... 100 to 104.	102	.004

A6.1.3 Distribution of Actual Cooling Load Hours (CLH_A) Throughout the United States



		Cooling	Load	Hours
Alaska	— 0	"	"	"
Canal Zone	— 6,000	"	"	"
Guam	— 6,600	"	"	"
Hawaii	— 2,300	"	"	"
Puerto Rico	— 6,000	"	"	"
Samoa	— 6,600	"	"	"
Virgin Island	— 6,000	"	"	"

A6.2 Heating Reference Material

A6.2.1 Test operating and test condition tolerance for

Steady-State High Temperature Test [at 47 F (8.3°C) or 62 F (16.7°C)]

and

Low Temperature Test [at 17 F (−8.3°C)]

	Test Operating* Tolerance	Test Condition** Tolerance
Indoor dry-bulb, F		
Entering	2.0	0.5
Leaving	2.0	---
Indoor wet-bulb, F		
Entering	1.0	---
Leaving	1.0	---
Outdoor dry-bulb, F		
Entering	2.0	0.5
Leaving	2.0	---
Outdoor wet-bulb, F		
Entering	1.0	0.3
Leaving	1.0	---
External resistance to air flow, inches of water	.05	.02
Electrical voltage, %	2.0	---

* Test Operating Tolerance is the maximum permissable variation of any measurement. When expressed as a percentage, the maximum allowable variation is the specified percentage of the average value.

** Test Condition Tolerance is the maximum permissable variation of the average value of the measurement from the standard or desired test condition.

A6.2.2 Test operating and test condition tolerance for the on-period portion of cyclic performance tests

	Test Operating Tolerances* (Applies after the 1st 30 sec. after compressor start-up)	Test Condition Tolerance**
Indoor dry-bulb, F		
Entering	2.0	0.5
Leaving	---	---
Indoor wet-bulb, F		
Entering	1.0	---
Leaving	---	---
Outdoor dry-bulb, F		
Entering	2.0	0.5
Leaving	---	---
Outdoor wet-bulb, F		
Entering	2.0	1.0
Leaving	---	---
External resistance to air-flow, inches of water	.05	.02
Electrical voltage, %	2.0	---

* Test Operating Tolerance is the maximum permissable variation of any measurement. When expressed as a percentage, the maximum allowable variation is the specified percentage of the average value.

** Test Condition Tolerance is the maximum permissable variation of the average value of the measurement from the standard or desired test condition.

A6.2.3 Test operating and test condition tolerance for frost accumulation tests

	Testing Operating Tolerance[1] During Heating	Testing Operating Tolerance[1] During Defrost	Test Condition Tolerance[2] (Heating Portion Only)
Indoor dry-bulb, F			
Entering	2.0	4.0[3]	0.5
Leaving	---	---	---
Indoor wet-bulb, F			
Entering	1.0	---	---
Leaving	---	---	---
Outdoor dry-bulb, F			
Entering	2.0	10.0	1.0
Leaving	---	---	---
Outdoor dew-point, F			
Entering	1.5	---	0.7
Leaving	---	---	---
External resistance to air-flow, inches of water	.05	---	.02
Electrical voltage, %	2.0	---	---

1 Test Operating Tolerance is the maximum permissable variation of any measurement. When expressed as a percentage, the maximum allowable variation is the specified percentage of the average value. Test Operating Tolerance During Heating applies when the heat pump is in the heating mode, except for the first 5 minutes after termination of a defrost cycle. Test Operating Tolerance During Defrost applies during a defrost cycle and during the first 5 minutes after the termination of a defrost cycle when the heat pump is operating in the heating mode.

2 Test Condition Tolerance is the maximum permissable variation of the average value of the measurement from the standard or desired test condition. Test Condition Tolerance applies only when the heat pump is operating in the heating mode.

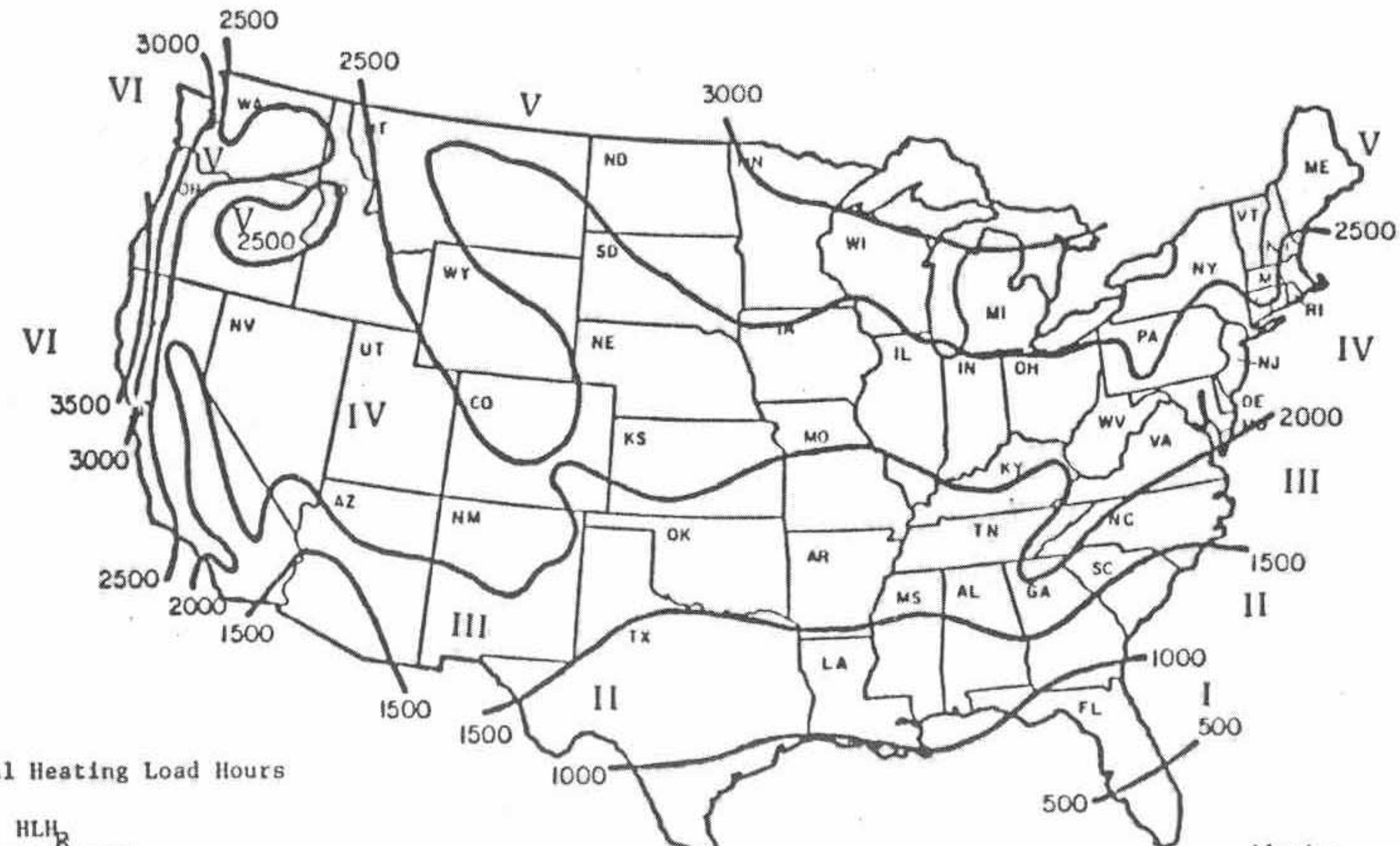
3 Not applicable during defrost if the indoor fan is off.

A6.2.4 Distribution of fractional hours in temperature bins, heating load hours and outdoor design temperature for the different climatic regions

Region		I	II	III	IV	V	VI
Heating Load Hours, HLH		750	1250	1750	2250	2750	2750*
Outdoor Design Temperature, T_{OD} for the region		37	27	17	5	-10	30
Fractional Hours:							
Bin #	T_j (F)						
j = 1	62	.291	.215	.153	.132	.106	.113
2	57	.239	.189	.142	.111	.092	.205
3	52	.194	.163	.138	.103	.086	.215
4	47	.129	.143	.137	.093	.076	.204
5	42	.081	.112	.135	.100	.078	.141
6	37	.041	.088	.118	.109	.087	.076
7	32	.019	.056	.092	.126	.102	.034
8	27	.005	.024	.047	.087	.094	.008
9	22	.001	.008	.021	.055	.074	.003
10	17	0	.002	.009	.036	.055	0
11	12	0	0	.005	.026	.047	0
12	7	0	0	.002	.013	.038	0
13	2	0	0	.001	.006	.029	0
14	-3	0	0	0	.002	.018	0
15	-8	0	0	0	.001	.010	0
16	-13	0	0	0	0	.005	0
17	-18	0	0	0	0	.002	0
18	-23	0	0	0	0	.001	0

*Pacific Coast Region

A6.2.5 Actual heating load hours (HLH_A) and regional heating load hours (HLH_R) for the Unites States



Regional Heating Load Hours

Region	HLH_R
I	750
II	1250
III	1750
IV	2250
V	2750
VI	2750

This map is reasonably accurate for most parts of the United States but is necessarily highly generalized, and consequently not too accurate in mountainous regions, particularly in the Rockies

Alaska - 3500 HLH
Hawaii and Territories - 0 HLH

A6.2.6 Standard Design Heating Requirements (Btuh)

5,000	25,000	50,000	90,000
10,000	30,000	60,000	100,000
15,000	35,000	70,000	110,000
20,000	40,000	80,000	130,000

A6.3 Representative Cooling Load Hours (CLH_R) for Each Heating Load Hours Region

Region	CLH_R	HLH_R
I	2400	750
II	1800	1250
III	1200	1750
IV	800	2250
V	400	2750
VI	200	2750

A6.4 Ground Water Temperature Map (Reserved)

[FR Doc. 79-38293 Filed 12-17-79; 8:45 am]

BILLING CODE 6450-01-C

APPENDIX D. PRESCRIPTIVE METHODOLOGY FOR THE CYCLIC TESTING OF DUCTED SYSTEMS REQUIRED BY C4.1 AND C4.2 - NORMATIVE

For the purpose of uniformity in the cyclic test requirements of Appendix C, the following test apparatus and conditions shall be met:

D1 The test apparatus is a physical arrangement of dampers, damper boxes, mixers, thermopile and ducts all properly sealed and insulated. See Figures D1 through D4 for typical test apparatus. The arrangement and size(s) of the components may be altered to meet the physical requirements of the unit to be tested.

D2 Dampers and their boxes shall be located outside of the ANSI/ASHRAE Standard 37 pressure measurement locations in the inlet air and outlet air ducts.

D3 The entire test apparatus shall not have a leakage rate which exceeds 20 cfm [0.01 m^3/s] when a negative pressure of 1.0 in H_2O [0.25 kPa] is maintained at the apparatus exit air location.

D4 The apparatus shall be insulated to have "U" value not to exceed 0.04 Btu/(h·ft^2·°F) total.

D5 The air mixer and a 40% maximum open area perforated screen shall be located in the outlet air portion of the apparatus upstream of the outlet damper. The mixer(s) shall be as described in ANSI/ASHRAE Standard 41.1. The mixing device shall achieve a maximum temperature spread of 1.5°F [0.8 °C] across the device. An inlet air mixer is not required.

D6 The temperature difference between inlet air and outlet air shall be measured by a thermopile. The thermopile shall be constructed of 24 gauge thermocouple wire with 16 junctions at each end. At each junction point the wire insulation shall be stripped for a length of 1.0 in [25 mm]. The junction of the wires shall have no more than two bonded turns.

D7 The dampers shall be capable of being completely opened or completely closed within a time period not to exceed 10 seconds for each action. Airflow through the equipment being tested should stop within 3 seconds after the airflow measuring device is de-energized. The air pressure difference (ΔP) at the nozzle shall be within 2% of steady state ΔP within 15 seconds from the time the air measuring device is re-energized.

D8 Test set up, temperature and electrical measurements must be identical for "C" and "U" tests in order to obtain minimum error in C_D. Electrical measurements shall be taken with an integrating type meter per ANSI/ASHRAE Standard 37 having an accuracy for all ranges experienced during the cyclic test.

D9 Prior to taking test data, the unit shall be operated at least one hour after achieving dry coil conditions. The drain pan shall be drained and the drain opening plugged. The drain pan shall be completely dry in order to maximize repeatability and reproducibility of test results.

D10 For coil only units not employing an enclosure, the coil shall be tested with an enclosure constructed of 1.0 in [25 mm] fiberglass ductboard with a density of 6 lb/ft^3 [100 kg/m^3] or an equivalent "R" value. For units with enclosures or cabinets, no extra insulating or sealing shall be employed.

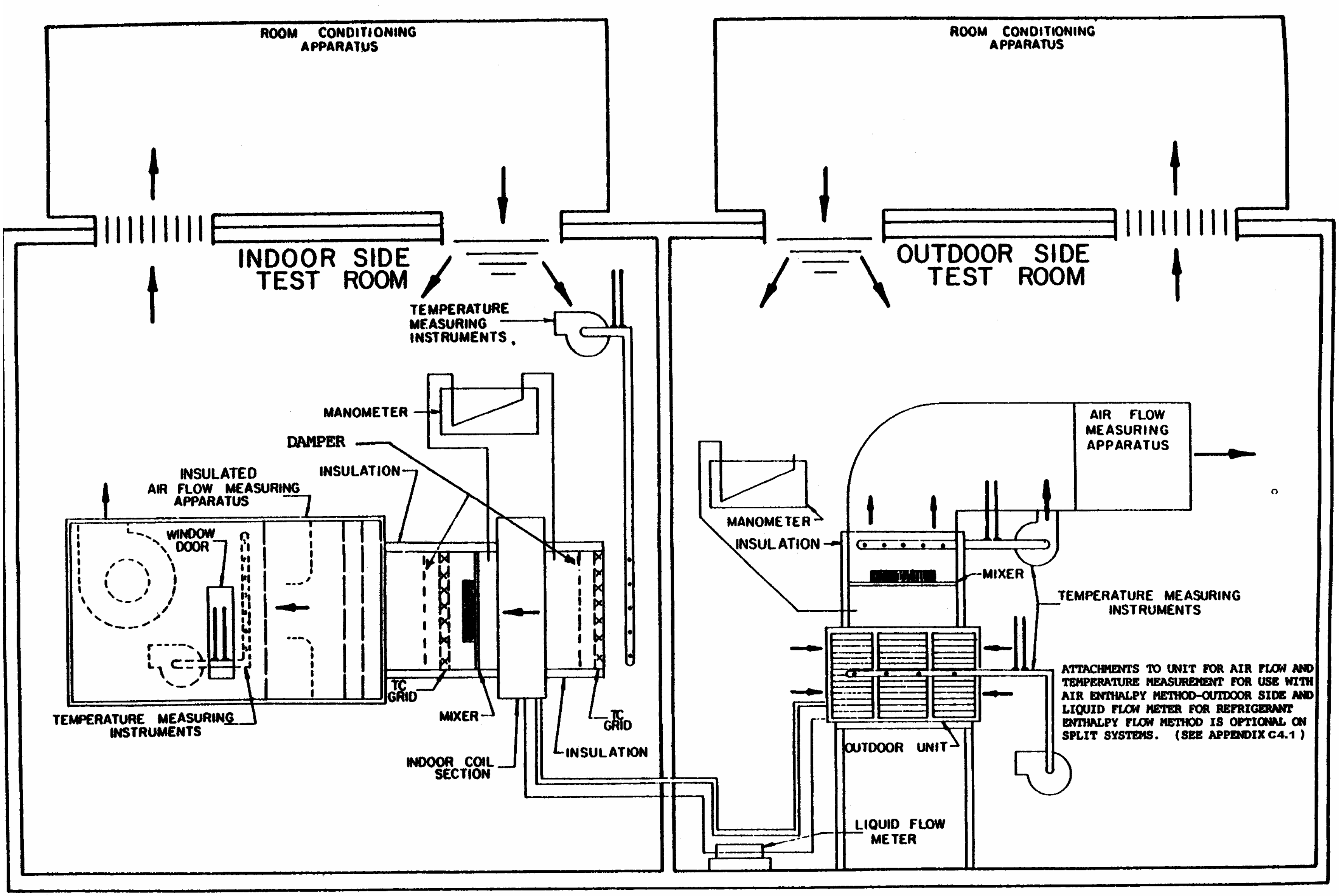


Figure D1. Tunnel Air Enthalpy Test Method Arrangement

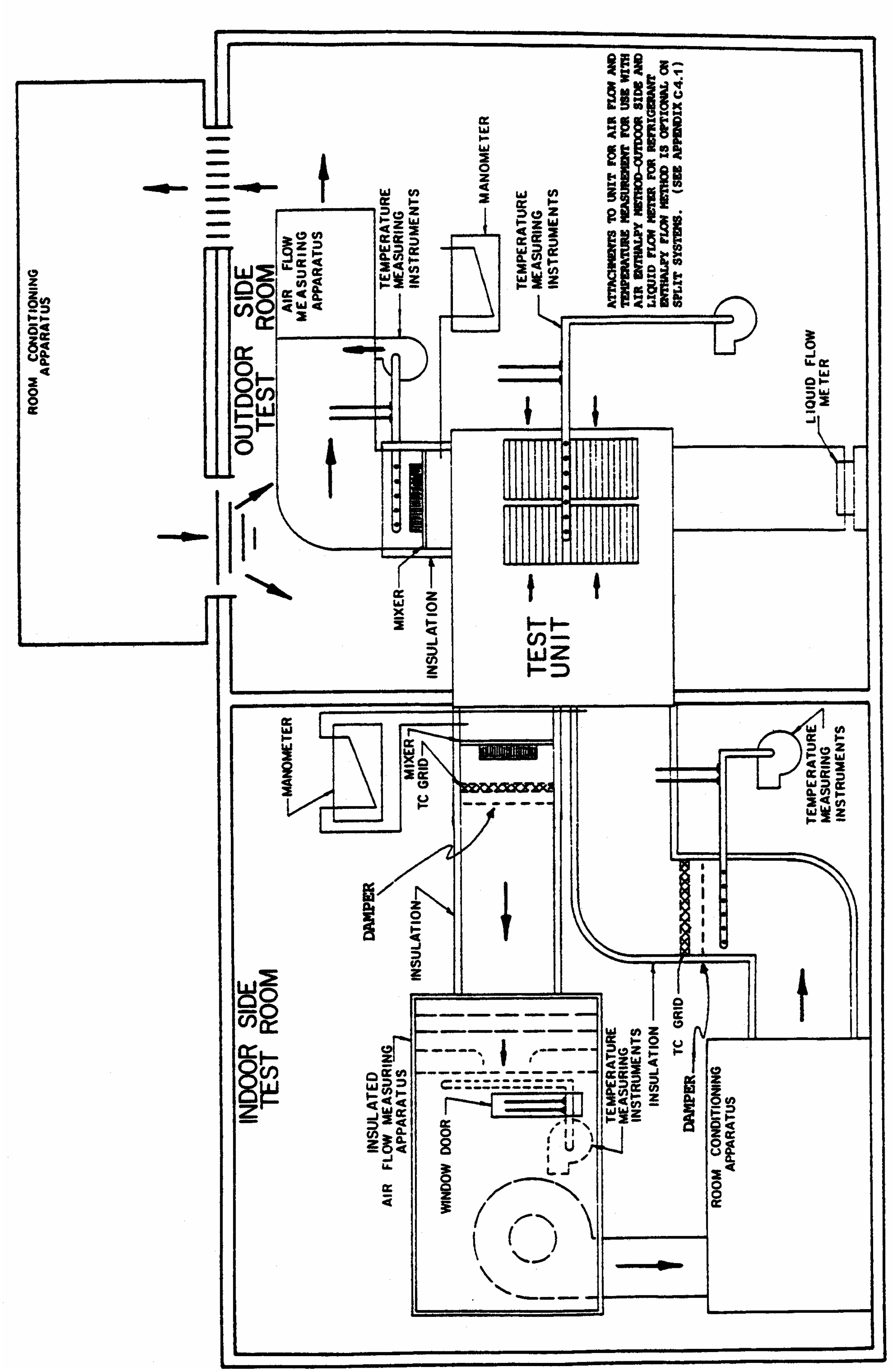


Figure D2. Loop Air Enthalpy Test Method Arrangement

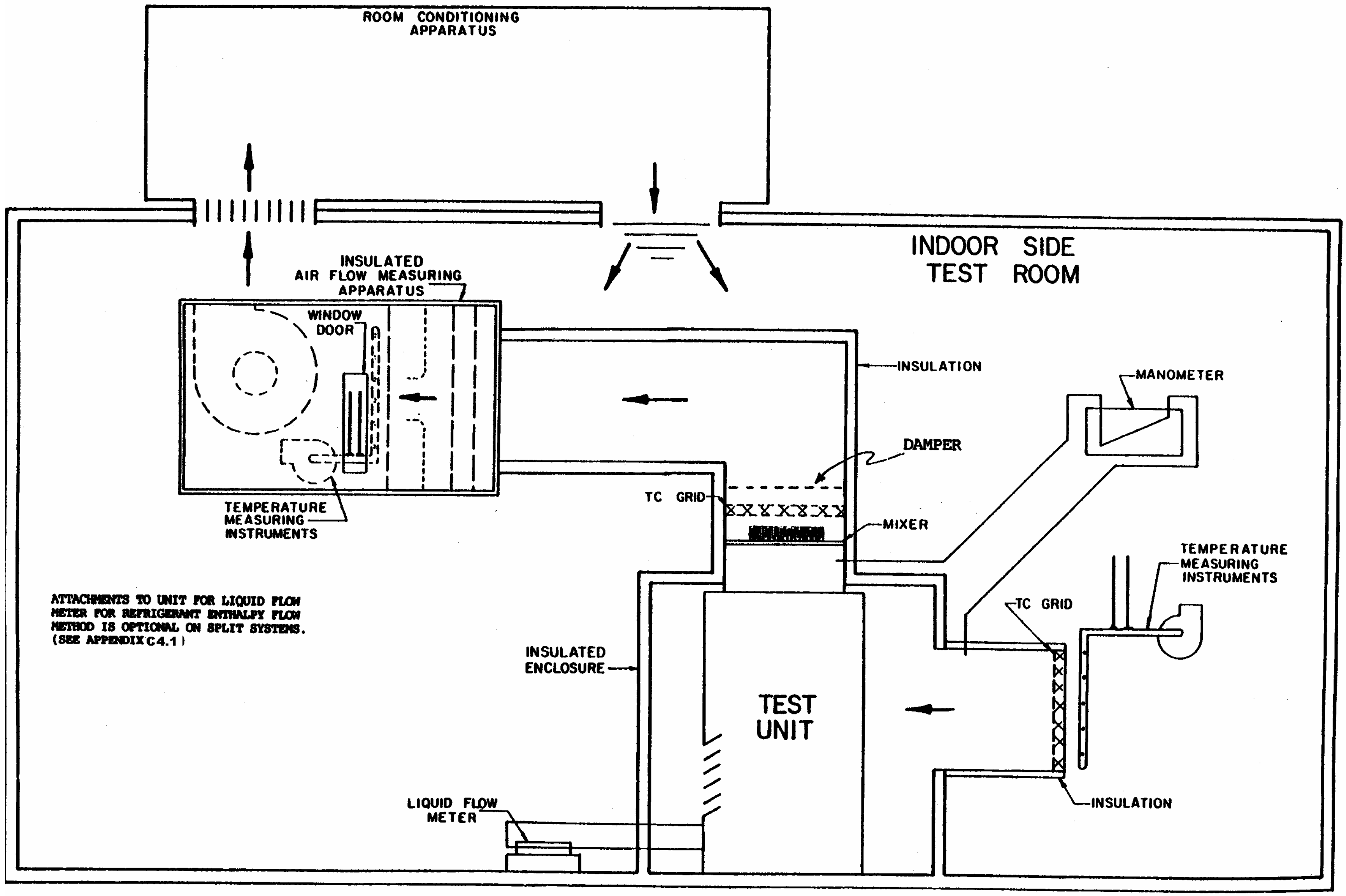


Figure D3. Calorimeter Air Enthalpy Test Method Arrangement

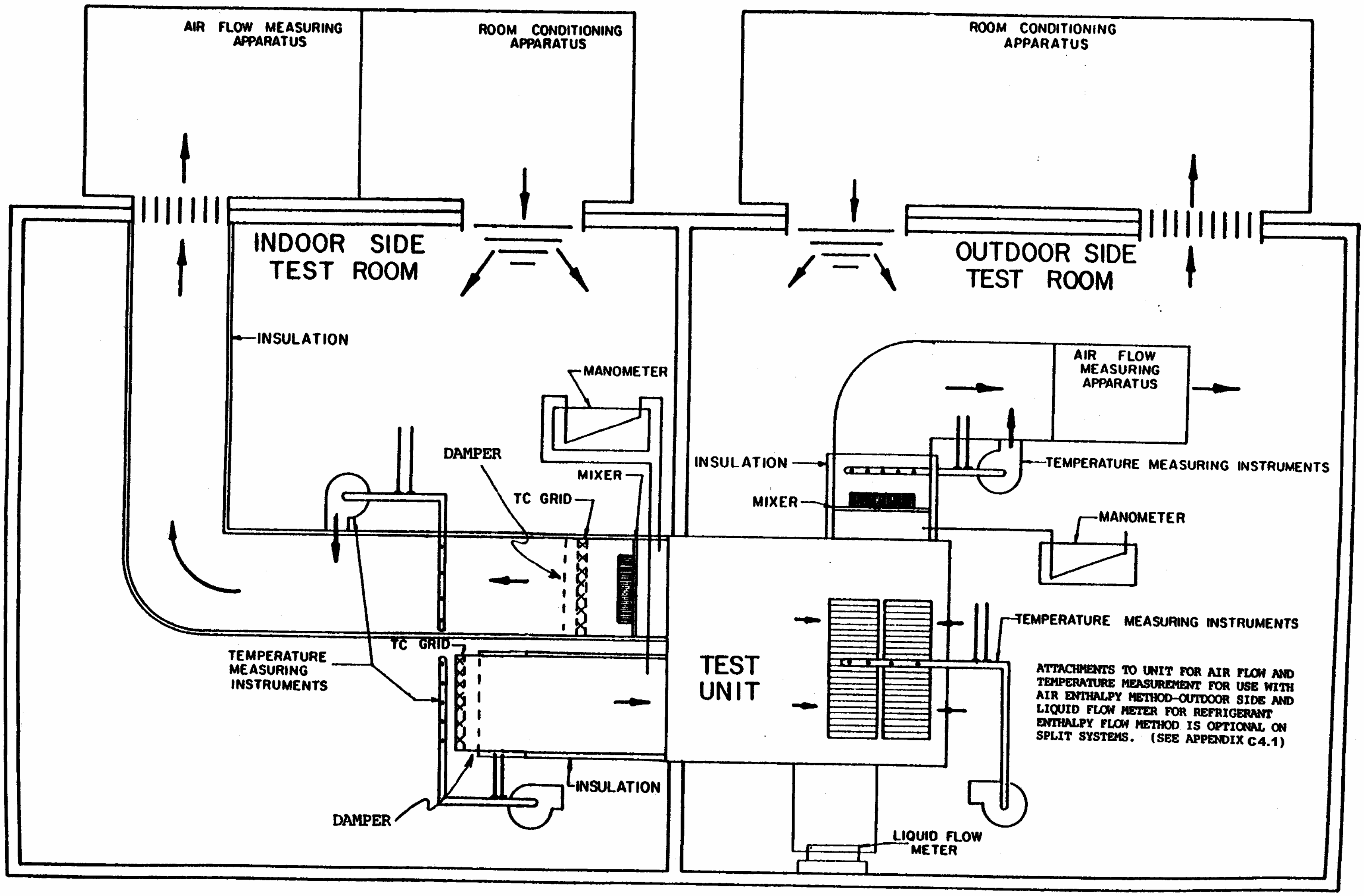


Figure D4. Room Air Enthalpy Test Method Arrangement

APPENDIX E. EXAMPLE OF CALCULATING INTEGRATED PART-LOAD VALUES (IPLV) - NORMATIVE

E1 *Purpose and Scope.*

E1.1 *Purpose.* This appendix shows example calculations for determining Integrated Part-Load Values (IPLV).

E1.2 *Scope.* This appendix is for equipment covered by this standard.

E2 *General Equation and Definitions of Terms.*

$$IPLV = (PLF_1 - PLF_2)\left(\frac{EER_1 + EER_2}{2}\right) \quad \text{E1}$$

$$+ (PLF_2 - PLF_3)\left(\frac{EER_2 + EER_3}{2}\right)$$

$$+ \ldots\ldots\ldots$$

$$(PLF_{n-1} - PLF_n)\left(\frac{EER_{n-1} + EER_n}{2}\right)$$

$$+ (PLF_n)(EER_n)$$

where:

PLF	=	Part-load factor determined from Figure E1
n	=	Total number of capacity steps
Subscript 1	=	100% capacity and EER at part- load Rating Conditions
Subscript 2, 3, etc.	=	Specific capacity and EER at part-load steps per 6.2 of this standard

E3 *Calculation Example for a Four Capacity Step System.*

E3.1 Assume equipment has four capacity steps as follows:

1 100% (full load)
2 75% of full load
3 50% of full load
4 25% of full load

E3.2 Obtain part-load factors from Figure E1.

E3.3 Obtain EER at each capacity step per 6.2 of this standard.

E3.4 Calculate IPLV using the general equation with:

$n = 4$
$PLF_1 = 1.0 \quad EER_1 = 8.9$
$PLF_2 = 0.9 \quad EER_2 = 7.7$
$PLF_3 = 0.4 \quad EER_3 = 7.1$
$PLF_4 = 0.1 \quad EER_4 = 5.0$

Enter the above values in Equation E1:

$$IPLV = (1.0 - 0.9)\left(\frac{8.9 + 7.7}{2}\right)$$

$$+ (0.9 - 0.4)\left(\frac{7.7 + 7.1}{2}\right)$$

$$+ (0.4 - 0.1)\left(\frac{7.1 + 5.0}{2}\right)$$

$$+ 0.1 \times 5.0$$

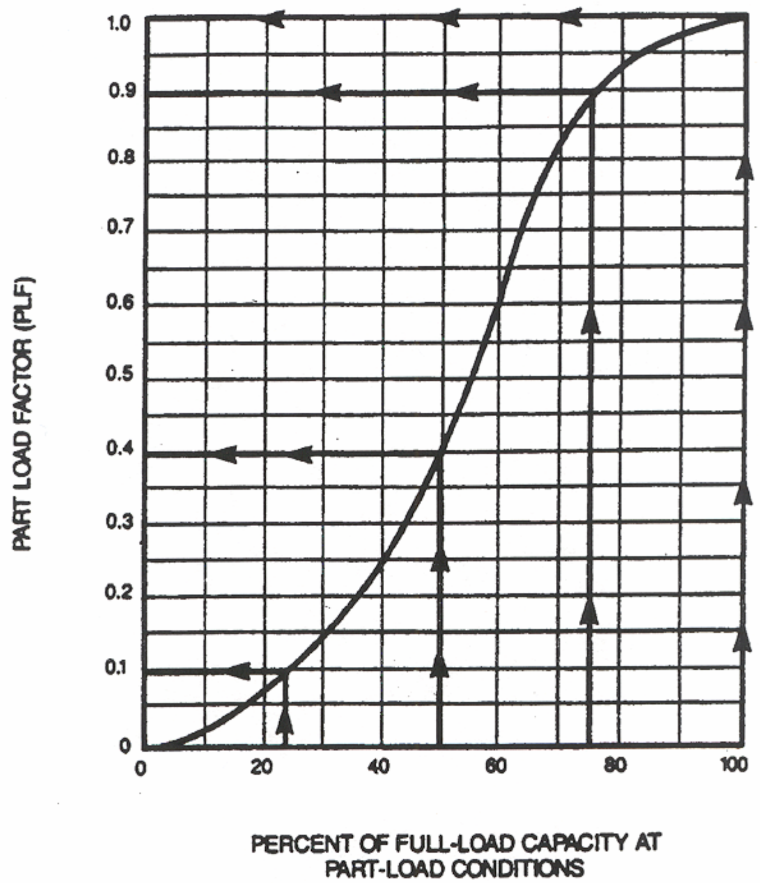
$$= (0.1 \times 8.3) + (0.5 \times 7.4)$$

$$+ (0.3 \times 6.0) + 0.5$$

$$= 0.83 + 3.70 + 1.80 + 0.5$$

$$IPLV = 6.8\ \text{Btu/(W·h)}$$

To further illustrate the calculation process, see the example in Table E1.



Note: The curve is based on following equation:

$$PLF = A0 + (A1 \times Q) + (A2 \times Q^2) + (A3 \times Q^3) + (A4 \times Q^4) + (A5 \times Q^5) + (A6 \times Q^6)$$

where: PLF = Part-Load Factor

Q = Percent of full-load capacity at part-load rating conditions.

$A0 = -0.12773917 \times 10^{-6}$
$A1 = -0.27648713 \times 10^{-3}$
$A2 = 0.50672449 \times 10^{-3}$
$A3 = -0.25966636 \times 10^{-4}$
$A4 = 0.69875354 \times 10^{-6}$
$A5 = -0.76859712 \times 10^{-8}$
$A6 = 0.28918272 \times 10^{-10}$

Figure E1. Part -Load Factor Example

Using information from E3:

Table E1. Example IPLV Calculation

Capacity Step	% Full Load Cap.[2]	PLF[3]	Mfrs. Part-Load EER	Avg. Part-Load EER	PLF Diff.	Avg. EER x PLF Diff. =	Weighted Avg.
1	100%	1.0	8.9 [2]		(1.0 - 0.9) = 0.1	8.3 x 0.1 =	0.83
			=	8.3			
2	75%	0.9	7.7			7.4 x 0.5 =	3.70
			=	7.4	(0.9 - 0.4) = 0.5		
3	50%	0.4	7.1			6.0 x 0.3 =	1.80
			=	6.0			
4	25%	0.1	5.0		(0.4 - 0.1) = 0.3	5.0 [1] x 0.1 =	0.50
			=	5.0 [1]			
	0%	0.0		-----		Single number IPLV	6.83 [4]
					(0.1 - 0.0) = 0.1		

Notes:

[1] For the range between 0% capacity and the last capacity step, use EER of the last capacity step for the average EER.
[2] The 100% capacity and EER are to be determined at the part-load Rating Conditions.
[3] Part-load factor from Figure E1.
[4] Rounded to 6.8 Btu/(W·h).