AHRI Standard 550/590 (I-P)

2011 Standard for

Performance Rating Of Water-Chilling and Heat Pump Water-Heating Packages Using the Vapor Compression Cycle

Air-Conditioning, Heating, and Refrigeration Institute

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

PH 703.524.8800
FX 703.562.1942

IMPORTANT

SAFETY DISCLAIMER

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

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

AHRI CERTIFICATION PROGRAM PROVISIONS

The scope of the Certification Program is defined below. This scope is current as of the publication date of the standard. Revisions to the scope of the certification program can be found on AHRI website www.ahrinet.org. The scope of the Certification Program should not be confused with the scope of the standard as the standard covers products that are not covered by a certification program.

Included in Certification Program:

50 Hz[a] and 60 Hz Air-Cooled Chiller (ACCL) Product Inclusions

- Chillers between 0 and 200 $tons_R$[b] manufactured prior to July 2011
- Chillers between 0 and 400 $tons_R$[b] manufactured between July 2011 and July 2013
- Chillers between 0 and 600 $tons_R$[b] manufactured after July 2013
- Units selected for use within the range of Application Rating Conditions as per AHRI Standard 550/590 (I-P)
- Hermetic or open type, electric motor driven
- Up to 600 volts
- All compressor types
- Units intended for use with glycol or other secondary coolant for freeze protection with a leaving chilled fluid temperature above 32.0°F are certified when tested with water at Standard Rating Conditions

Note a: 50 Hz products selectively certified as per Section 1.4 of the Air-Cooled Water Chilling Packages Using Vapor Compression Cycle Operations Manual
Note b: The cooling capacity in $tons_R$ at full-load AHRI Standard Rating Conditions per Table 1 of AHRI Standard 550/590 (I-P).

60 Hz Water-Cooled Chiller (WCCL) Product Inclusions

- All compressor types;
- Chillers rated between 0 and 2,500 $tons_R$[c] manufactured prior to January 2012
- Chillers rated between 0 and 3,000 $tons_R$[c] manufactured after January 2012
- Hermetic or open type electric motor driven
- Units selected for use within the range of Application Rating Conditions as per AHRI Standard 550/590 (I-P)
- Voltages up to 11,000 volts
- Voltages up to 15,000 volts after June 15, 2011
- Positive Displacement Units intended for use with glycol or other secondary coolant for freeze protection with a leaving chilled fluid temperature above 32.0°F are certified when tested with water at Standard Rating Conditions

Note c: Rated capacity, $tons_R$, for Positive Displacement chillers is the net cooling capacity at full-load AHRI Standard Rating Conditions per Table 1 of AHRI Standard 550/590 (I-P). Rated capacity, $tons_R$, for centrifugal chillers is the net cooling capacity at full-load AHRI Application Rating Conditions within the range permitted in Table 2 of AHRI Standard 550/590 (I-P).

Price $10.00 (M) $20.00 (NM)
Printed in U.S.A.

50 Hz WCCL Product Inclusions

- Centrifugal & screw compressor chillers
- Chillers rated between 200 and 2500 $tons_R$[d]
- Hermetic & open type, electric motor driven
- Units selected for use within the range of Application Rating Conditions as per AHRI Standard 550/590 (I-P)
- Voltages up to 11,000 volts
- Voltages up to 15,000 volts manufactured after June 15, 2011
- Positive Displacement Units intended for use with glycol or other secondary coolant for freeze protection with a leaving chilled fluid temperature above 32°F are certified when tested with water at Standard Rating Conditions

  Note d: Rated capacity, $tons_R$, for Positive Displacement chillers is the net cooling capacity at full-load AHRI Standard Rating Conditions per Table 1 of AHRI Standard 550/590. Rated capacity, $tons_R$, for centrifugal chillers is the Net *Refrigerating* Capacity at full-load Application Rating Conditions within the range permitted in Table 2 of AHRI Standard 550/590 (I-P).

Excluded from the Certification Program:

50 Hz and 60 Hz ACCL Product Exclusions

- Condenserless chillers
- Evaporatively cooled chillers
- Chillers above 200 $tons_R$ manufactured prior to July 2011
- Chillers above 400 $tons_R$ manufactured prior to July 2013
- Chillers above 600 $tons_R$
- Chillers with voltages above 600 volts
- Glycol and other secondary coolants are excluded when leaving chiller fluid temperature is below 32.0°F
- Custom units as defined in the section specific Operations Manual
- Field trial units as defined in the section specific Operations Manual
- Heat recovery & heat pump ratings are not certified, however manufacturers may elect to certify these chillers in the cooling mode and with the heat recovery option turned off
- Units for use outside of Application Rating Conditions
- Chillers that are not electrically driven, or that use open type compressors not supplied with motors by the manufacturer
- 50 Hz Air-Cooled units that the manufacturer elects not to certify

60 Hz WCCL Product Exclusions

- Condenserless chillers
- Evaporatively cooled chillers
- Chillers above 2500 $tons_R$ manufactured prior to January 2012
- Chillers above 3000 $tons_R$
- Chillers with voltages above 11,000 volts prior to June 15, 2011
- Chillers with voltages above 15,000 volts
- Chillers that are not electrically driven
- Chillers with motors not supplied with the unit by the manufacturer
- Glycol and other secondary coolants are excluded when leaving chiller fluid temperature is below 32.0°F
- Custom units as defined in the section specific Operations Manual
- Field trial units as defined in the section specific Operations Manual
- Units for use outside of Application Rating Conditions
- Heat recovery & heat pump ratings are not certified; however, manufacturers may elect to certify these chillers in the cooling mode and with the heat recovery option turned off

50 Hz WCCL Product Exclusions

- Condenserless chillers
- Evaporatively cooled chillers
- Reciprocating and scroll Water-Chilling Packages
- Chillers below 200 $tons_R$
- Chillers above 2,500 $tons_R$ manufactured prior to January 2012
- Chillers above 3,000 $tons_R$
- Chillers with voltages above 11,000 volts prior to June 15, 2011
- Chillers with voltages above 15,000 volts
- Chillers that are not electrically driven
- Chillers with motors not supplied with the unit by the manufacturer
- Glycol and other secondary coolants are excluded when leaving chiller fluid temperature is below 32.0°F
- Custom units as defined in the section specific Operations Manual
- Field trial units as defined in the section specific Operations Manual
- Units for use outside of Application Rating Conditions
- Heat recovery & heat pump ratings are not certified, however manufacturers may elect to certify these chillers in the cooling mode and with the heat recovery option turned off

Certified Ratings

The Water-Cooled and Air-Cooled Certification Program ratings verified by test are:

Operating Conditions		Water-Cooled	Air-Cooled
Standard Rating Conditions[1]	Full Load	• Capacity[3] • Energy Efficiency • Water Pressure Drop	• Capacity[3] • Energy Efficiency • Water Pressure Drop
	Part Load	• IPLV[4] Energy Efficiency	• IPLV[4] Energy Efficiency
Application Rating Conditions [2]	Full Load	• Capacity[3] • Energy Efficiency • Water Pressure Drop	• Capacity[3] • Energy Efficiency • Water Pressure Drop
	Part Load	• NPLV[5] Energy Efficiency	• *Not Applicable*

Notes:

1. Standard Rating Conditions per AHRI Standard 550/590 Section 5.2
2. Application Rating Conditions per AHRI Standard 550/590 Section 5.3
3. Certified Capacity is the net Refrigerating Capacity per AHRI Standard 550/590 Section 3.3
4. Integrated Part-Load Value (IPLV) per AHRI Standard 550/590 Section 5.4
5. Non-Standard Part-Load Value (NPLV) per AHRI Standard 550/590 Section 5.4

With the following units of measure:

- Net Capacity, $tons_R$
- Energy Efficiency, as applicable:
  - Power Input per Capacity, kW/ton_R; or
  - Energy Efficiency Ratio (EER), Btu/(W·h); or
  - Coefficient of Performance (COP), watts/watt
- Evaporator and/or condenser Water Pressure Drop, ft H_2O

Note:

This standard supersedes AHRI Standard 550/590-2003 and is effective 1 January 2012
For SI ratings, see AHRI Standard 551/591 (SI)-2011.
The requirements of Appendix G shall be effective on 1 January 2013 and optional prior to that date.
Accompanying this standard is an Excel Spreadsheet for the Computation of the Pressure Drop Adjustment Factors (http://www.ahrinet.org/search+standards.aspx).

TABLE OF CONTENTS

SECTION PAGE

Section 1. Purpose 1

Section 2. Scope 1

Section 3. Definitions 1

Section 4. Test Requirements 4

Section 5. Rating Requirements 4

Section 6. Minimum Data Requirements for Published Ratings 22

Section 7. Conversions and Calculations 27

Section 8. Marking and Nameplate Data 27

Section 9. Conformance Conditions 28

TABLES

Table 1. Standard Rating Conditions 6

Table 2. Application Rating Conditions 8

Table 3. Part-Load Conditions for Rating 10

Table 4. Chiller Performance – IPLV 13

Table 5. Chiller Performance – NPLV 14

Table 6. Chiller Performance – Interpolated Data 14

Table 7. Actual and Adjusted Performance for Example 4 16

Table 8. Performance Data for Example 5 17

Table 9. Actual and Adjusted Performance for Example 6 18

Table 10. Definition of Tolerances 19

Table 11. Published Values 25

Table 12. Conversion Factors 27

FIGURES

Figure 1. Part Load Condenser Temperature for IPLV Test Points 12

Figure 2. Rating Point Interpolation 15

Figure 3. Allowable Tolerance **(Tol_1)** Curves for Full and Part Load Points................................20

Figure 4. IPLV and NPLV Tolerance **(Tol_2)** Curve 20

APPENDICES

Appendix A. References – Normative 29

Appendix B. References – Informative 30

Appendix C. Method of Testing Water-Chilling and Water-Heating Packages Using the Vapor Compression Cycle – Normative..31

Appendix D. Derivation of Integrated Part-Load Value (IPLV) – Informative 45

Appendix E. Chiller Condenser Entering Air Temperature Measurement – Normative 53

Appendix F. Barometric Pressure Adjustment – Normative..59

Appendix G. Water Side Pressure Drop Correction Procedure – Normative.......................................61

Appendix H. Heating Capacity Test Procedure – Normative 63

TABLES FOR APPENDICES

Table C1. Accuracy Requirements for Test Instrumentation..34

Table D1. Group 1 Air-Cooled IPLV Data and Calculation..50

Table D2. Group 1 Water-Cooled IPLV Data and Calculation...51

Table D3. Group 1 – 4 IPLV Summary 52

Table E1. Temperature Measurement Requirements...53

Table E2. Criteria for Air Distribution and Control of Air Temperature 54

Table F1. Terms..60

Table F2. Correction Factor (CF) Coefficients..60

Table G1. K Factors for Elbow Arrangements..62

Table H1. Test Tolerances..67

FIGURES FOR APPENDICES

Figure D1. Ton_R-Hour Distribution Categories 47

Figure D2. Bin Groupings – Ton_R-Hours 48

Figure D3. Group 1 Ton_R-Hour Distribution Categories 48

Figure D4. Group 2 Ton_R-Hour Distribution Categories 49

Figure E1. Typical Air Sampling Tree 55

Figure E2. Aspirating Psychrometer 56

Figure E3. Determination of Measurement Rectangles and Required Number of Air Sampler Trees 57

Figure E4. Typical Test Setup Configurations 58

Figure G1. Calibration Term for Included Angle for Expansion/Contraction Fittings 62

PERFORMANCE RATING OF WATER-CHILLING AND HEAT PUMP WATER-HEATING PACKAGES USING THE VAPOR COMPRESSION CYCLE

Section 1. Purpose

1.1 *Purpose*. The purpose of this standard is to establish for Water-Chilling and Water-Heating Packages using the vapor compression cycle: definitions; test requirements; rating requirements; minimum data requirements for Published Ratings; 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, efficiency regulators, 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 vapor compression refrigeration Water-Chilling and Water-Heating Packages including one or more hermetic or open drive compressors. These Water-Chilling and Water-Heating Packages include:

- Water-Cooled, Air-Cooled, or Evaporatively-Cooled Condensers
- Water-Cooled heat reclaim condensers
- Air-to-water heat pump
- Water-to-water heat pumps with a capacity greater or equal to 135,000 Btu/h. Water-to-water heat pumps with a capacity less than 135,000 Btu/h are covered by the latest edition of AHRI Standard 320

Note that this standard covers products that may not currently be covered under a certification program.

Section 3. Definitions

All terms in this document 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.

3.1 *Auxiliary Power*. Power provided to devices that are not integral to the operation of the vapor compression cycle such as, but not limited to: oil pumps, refrigerant pumps, control power, fans and heaters.

3.2 *Bubble Point.* Refrigerant liquid saturation temperature at a specified pressure.

3.3 *Capacity*. A measurable physical quantity that characterizes the water side heat flow rate, Btu/h or $tons_R$. Capacity is defined as the mass flow rate of the water multiplied by the difference in enthalpy of water entering and leaving the heat exchanger, Btu/h or $tons_R$. For this standard, the enthalpy change is approximated as the sensible heat transfer using specific heat and temperature difference, and in some calculations also the energy associated with water-side pressure losses.

> **3.3.1** Gross *Heating Capacity*. The capacity of the Water Cooled Condenser as measured by the heat transfer from the refrigerant in the condenser. This value includes both the sensible heat transfer and the pressure drop effects of the water flow through the condenser. This value is used to calculate the test heat balance. (Refer to Equations C12a and C12b).
>
> **3.3.2** Gross *Refrigerating Capacity*. The capacity of the water-cooled evaporator as measured by the heat transfer to the refrigerant in the evaporator. This value includes both the sensible heat transfer and the pressure drop effects of the water flow through the evaporator. This value is used to calculate the test heat balance. (Refer to Equation C11).

3.3.3 *Net Heating Capacity.* The capacity of the heating condenser available for useful heating of the thermal load external to the Water-Heating Package and is calculated using only the sensible heat transfer. (Refer to Equations 7a and 7b).

3.3.4 Net *Refrigerating Capacity*. The capacity of the evaporator available for cooling of the thermal load external to the Water-Chilling Package and is calculated using only the sensible heat transfer. (Refer to Equation 6).

3.4 *Compressor Saturated Discharge Temperature.* For single component and azeotrope refrigerants, it is the saturated temperature corresponding to the refrigerant pressure at the compressor discharge. For zeotropic refrigerants, it is the arithmetic average of the Dew Point and Bubble Point temperatures corresponding to refrigerant pressure at the compressor discharge. It is usually taken at or immediately downstream of the compressor discharge service valve (in either case on the downstream side of the valve seat), where discharge valves are used.

3.5 *Condenser*. A refrigeration system component which condenses refrigerant vapor. Desuperheating and sub-cooling of the refrigerant may occur as well.

3.5.1 *Air-Cooled Condenser.* A component which condenses refrigerant vapor by rejecting heat to air mechanically circulated over its heat transfer surface causing a rise in the air temperature.

3.5.2 *Evaporatively-Cooled Condenser.* A component which condenses refrigerant vapor by rejecting heat to a water and air mixture mechanically circulated over its heat transfer surface, causing evaporation of the water and an increase in the enthalpy of the air.

3.5.3 *Water-Cooled Condenser.* A component which utilizes refrigerant-to-water heat transfer means, causing the refrigerant to condense and the water to be heated.

3.5.4 *Water-Cooled Heat Reclaim Condenser.* A component which utilizes refrigerant-to-water heat transfer means, causing the refrigerant to condense and the water to be heated. This Condenser may be a separate condenser, the same as, or a portion of the Water-Cooled Condenser.

3.6 *Dew Point.* Refrigerant vapor saturation temperature at a specified pressure.

3.7 *Energy Efficiency.*

3.7.1 *Cooling Energy Efficiency.*

3.7.1.1 *Cooling Coefficient of Performance (COP_R).* A ratio of the Net Refrigerating Capacity in watts to the power input values in watts at any given set of Rating Conditions expressed in watts/watt. (Refer to Equation 1)

3.7.1.2 *Energy Efficiency Ratio (EER).* A ratio of the Net Refrigerating Capacity in Btu/h to the power input value in watts at any given set of Rating Conditions expressed in Btu/(h · W). (Refer to Equation 2)

3.7.1.3 *Power Input per Capacity.* A ratio of the power input, W_{INPUT}, supplied to the unit in kilowatts [kW], to the Net Refrigerating Capacity at any given set of Rating Conditions, expressed in kilowatts per ton_R of Refrigeration [kW/ton_R]. (Refer to Equation 3)

3.7.2 *Heating Energy Efficiency.*

3.7.2.1 *Heating Coefficient of Performance (COP_H).* A ratio of the Net Heating Capacity in watts to the power input values in watts at any given set of Rating Conditions expressed in watts/watt. (Refer to Equation 4).

3.7.2.2 *Heat Reclaim Coefficient of Performance (COP_{HR}).* COP_{HR} applies to units that are operating in a manner that uses either all or only a portion of heat generated during chiller operation, q_{hrc}, to heat the occupied space, while the remaining heat, q_{cd}, if any, is rejected to the outdoor ambient. COP_{HR} takes into account the beneficial cooling capacity, q_{ev}, as well as the Heat Recovery capacity, q_{hrc} (Refer to Equation 5).

3.8 *Fouling Factor*. The thermal resistance due to fouling accumulated on the water side or air side heat transfer surface.

3.8.1 *Fouling Factor Allowance (FFA)*. Provision for anticipated water side or air side fouling during use expressed in $h \cdot ft^2 \cdot {}^{o}F/Btu$.

3.9 *Liquid Refrigerant Temperature*. The temperature of the refrigerant liquid leaving the condenser but prior to the expansion device.

3.10 *Part-Load Value (PLV).* A single number figure of merit expressing part-load efficiency for equipment on the basis of weighted operation at various partial load capacities for the equipment. (Refer to Appendix D for information regarding the use of IPLV and NPLV.)

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

3.10.2 *Non-Standard Part-Load Value (NPLV)*. A single number part-load efficiency figure of merit calculated per the method described in this standard referenced to conditions other than IPLV conditions. (i.e. For units with Water-Cooled Condensers that are not designed to operate at Standard Rating Conditions.)

3.11 *Percent Load (%Load).* The part-load net capacity divided by the full-load rated net capacity at the full-load rating conditions, stated in decimal format. (e.g.100% = 1.0).

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

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

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

3.13 *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.

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

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

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

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

3.15 *Total Power Input*. Power input of all components of the unit.

3.16 *Total Heat Rejection*. Heat rejected through the condenser including heat utilized for heat recovery (q_{cd}+q_{hrc}).

3.17 *Water-Chilling or Water-Heating Package.* A factory-made and prefabricated assembly (not necessarily shipped as one package) of one or more compressors, condensers and evaporators, with interconnections and accessories designed for the purpose of cooling or heating water. It is a machine specifically designed to make use of a vapor compression refrigeration cycle to remove heat from water and reject the heat to a cooling medium, usually air or water. The refrigerant condenser may or may not be an integral part of the package.

3.17.1 *Heat Reclaim Water-Chilling Package*. A factory-made package, designed for the purpose of chilling water and containing a condenser for reclaiming heat. Where such equipment is provided in more than one assembly, the separate assemblies are to be designed to be used together, and the requirements of rating outlined in this standard are based upon the use of matched assemblies. It is a package specifically designed to make use of the refrigerant cycle to remove heat from the water source and to reject the heat to another fluid for heating use. Any excess heat may be rejected to another medium, usually air or water.

3.17.2 *Heat Pump Water-Chilling Package*. A factory-made package, designed for the purpose of heating water. Where such equipment is provided in more than one assembly, the separate assemblies are to be designed to be used together, and the requirements of rating outlined in this standard are based upon the use of matched assemblies. It is a package specifically designed to make use of the refrigerant cycle to remove heat from an air or water source and to reject the heat to water for heating use. This unit can include valves to allow for reverse-cycle (cooling) operation.

3.17.3 *Modular Chiller Package*. A modular chiller is a package that is made up of multiple water-chilling units that can function individually or as a single unit.

3.18 *Water Pressure Drop*. A measured value of the reduction in water pressure associated with the flow through a water-type heat exchanger. This value is expressed as a rating in ft H_2O.

Section 4. Test Requirements

4.1 *Test Requirements*. Ratings shall be established at the Rating Conditions specified in Section 5. Ratings shall be verified by tests conducted in accordance with the test method and procedures described in Appendix C.

Section 5. Rating Requirements

5.1 *Standard Rating Metrics*.

5.1.1 *Cooling Energy Efficiency*. The general forms of the Cooling Energy Efficiency terms are listed as equations 1 through 3. These terms are calculated at both design point and at part load conditions. They also may be modified by adjustments for barometric pressure as shown in Appendix F or by a part load degradation factor as detailed in Equation 15.

5.1.1.1 The Cooling Coefficient of Performance (COP_R) [W/W] shall be calculated as follows:

$$COP_R = \frac{q_{ev}}{K1 \cdot W_{INPUT}} \quad (1)$$

Where:

$K1$ = 3.41214, Btu/W·h
q_{ev} = Net Refrigerating Capacity, Btu/h
W_{INPUT} = Total Power Input, W

5.1.1.2 The Energy Efficiency Ratio (EER) [Btu/W·h] shall be calculated as follows:

$$EER = \frac{q_{ev}}{W_{INPUT}} \quad (2)$$

5.1.1.3 The Power Input per Capacity [kW/ton_R] shall be calculated as follows:

$$\text{Power Input Per Capacity} = \frac{K2 \cdot W_{INPUT}}{q_{ev}} \quad (3)$$

Where:

$K2$ = 12000, Btu/ton_R·h

5.1.2 *Heating Energy Efficiency*

5.1.2.1 The Heating Coefficient of Performance (COP_H) [W/W] shall be calculated as follows:

$$COP_H = \frac{q_{cd}}{K1 \cdot W_{INPUT}} \qquad 4$$

Where:

q_{cd} = Net Heating Capacity, Btu/h

5.1.2.2 The Heat Reclaim Coefficient of Performance (COP_{HR}) [W/W] shall be calculated as follows:

$$COP_{HR} = \frac{q_{ev} + q_{hrc}}{K1 \cdot W_{INPUT}} \qquad 5$$

Where:

q_{hrc} = Heat generated during chiller operation, Btu/h

5.1.3 *Net Refrigerating Capacity*. The Net Refrigerating Capacity, [Btu/h], for the evaporator shall use the water temperatures, water flow rate and water properties at the evaporator entering and leaving conditions and be calculated as follows:

$$q_{ev} = m_w \cdot c_p \cdot (t_e - t_l) \qquad 6$$

Where:

c_p = Specific heat at the average of entering and leaving water temperatures, Btu/lbm,°F
m_w = Mass flow rate, lbm/h
t_e = Entering water temperature, °F
t_l = Leaving water temperature, °F
=

5.1.4 *Net Heating Capacity*. The Net Heating Capacity, [Btu/h], for either a standard or heat recovery condenser shall use the water temperatures, water flow rate, and water properties at the entering and leaving conditions and be calculated as follows:

$$q_{cd} = m_w \cdot c_p \cdot (t_l - t_e) \qquad 7a$$

$$q_{hrc} = m_w \cdot c_p \cdot (t_l - t_e) \qquad 7b$$

5.1.5 *Water Pressure Drop*. For the Water Pressure Drop calculations, refer to Appendices C and G.

7 **5.2** *Standard Ratings and Conditions.* Standard Ratings for all Water-Chilling Packages shall be established at the Standard Rating Conditions. These packages shall be rated for cooling, heat reclaim, or heating performance at conditions specified in Table 1. Standard Ratings shall include a water-side Fouling Factor Allowance as specified in the notes section of Table 1. Modular Chiller Packages consisting of multiple units and rated as a single package must be tested as rated.

Table 1. Standard Rating Conditions

Operating Category	Conditions	Cooling Mode Evaporator[2]			Cooling Mode Heat Rejection Heat Exchanger													
					Tower (Fluid Conditions)[3]			Heat/Reclaim (Fluid Conditions)[4]		Evaporatively Cooled Entering Temperature[5]		Air-Cooled (AC) Entering Temperature[6, 8]		Without Condenser			
														Air-Cooled Refrigerant Temp.		Water & Evaporatively Cooled Refrigerant Temp.	
		Entering Temp. °F[9]	Leaving Temp. °F	Flow rate gpm/ton_R	Entering Temp. °F	Leaving Temp °F	Flow rate gpm/ton_R	Entering Temp. °F	Leaving Temp °F	Dry-Bulb °F	Wet-Bulb °F	Dry-Bulb °F	Wet-Bulb °F	SDT[12] °F	LIQ[13] °F	SDT[12] °F	LIQ[13] °F
All Cooling	Std	54.0 [9]	44.0	2.4	85.0	94.3[10]	3.0	--	--	95.0	75.0	95.0	--	125.0	105.0	105.0	98.0
AC Heat Pump High Heating[7]	Low	--	105.0	Note[1]	--		--	--	--	--	--	47.0	43.0	--	--	--	--
	Med	--	120.0	Note[1]			--	--	--	--	--	47.0	43.0	--	--	--	--
	High	--	140.0	Note[1]	--		--	--	--	--	--	47.0	43.0	--	--	--	--
AC Heat Pump Low Heating[7]	Low	--	105.0	Note[1]	--		--	--	--	--	--	17.0	15.0	--	--	--	--
	Med	--	120.0	Note[1]	--		--	--	--	--	--	17.0	15.0	--	--	--	--
	High	--	140.0	Note[1]	--		--	--	--	--	--	17.0	15.0	--	--	--	--
Water Cooled Heating	Low	--	44.0	2.4	--		--	95.0	105.0	--	--	--	--	--	--	--	--
	Med	--	44.0	2.4	--		--	105.0	120.0	--	--	--	--	--	--	--	--
	High	--	44.0	2.4	--		--	120.0	140.0	--	--	--	--	--	--	--	--
	Boost	75.0	65.0	--	--		--	120.0	140.0	--	--	--	--	--	--	--	--
Heat Reclaim	Low	--	44.0	2.4	75.0		Cooling[11]										
	Med	--	44.0	2.4	75.0		Cooling[11]	105.0	120.0	40.0	38.0	40.0	38.0	--	--	--	--

Notes

1. The water flow rate used for the heating tests of reverse cycle air to water heat pumps shall be the flow rate determined during the cooling test.
2. The rating Fouling Factor for the cooling mode evaporator or the heating condenser for AC reversible cycles shall be 0.000100 $h \cdot ft^2 \cdot °F/Btu$.
3. The rating Fouling Factor for tower heat exchangers shall be 0.000250 $h \cdot ft^2 \cdot °F/Btu$.
4. The rating Fouling Factor for heating and heat reclaim heat exchangers shall be 0.000100 $h \cdot ft^2 \cdot °F/Btu$ for closed loop and 0.000250 $h \cdot ft^2 \cdot °F/Btu$ for open loop systems.

5. Evaporatively cooled condensers shall be rated with a Fouling Factor of zero (0.000) $h \cdot ft^2 \cdot {}^{o}F/Btu$.
6. Air-Cooled Condensers shall be rated with a Fouling Factor of zero (0.000) $h \cdot ft^2 \cdot {}^{o}F/Btu$.
7. Assumes a reversible cycle where the cooling mode evaporator becomes the condenser circuit in the heating mode.
8. Air-Cooled unit ratings are at standard barometric condition (sea level). Measured data will be corrected to a Barometric Pressure of 14.696 psia per Appendix F.
9. The entering temperature shown is for reference at Standard Rating Conditions only. The entering temperature is determined by the evaporator leaving water temperature and flow rate at the rated capacity.
10. The leaving temperature shown is for reference at Standard Rating Condition only. The leaving temperature is determined by the entering water temperature, the flow rate at the rated capacity and the efficiency of the chiller.
11. Flow rate established for the cooling mode.
12. Saturated Discharge Temperature (SDT).
13. Liquid Refrigerant Temperature (LIQ).

5.3 *Application Rating Conditions.* Application Ratings should include the range of Rating Conditions listed in Table 2 or be within the operating limits of the equipment. For guidance to the industry, designing to large Fouling Factors significantly impacts the performance of the chiller. It is best to maintain heat transfer surfaces by cleaning or maintaining proper water treatment to avoid highly fouled conditions and the associated efficiency loss. From a test perspective, highly fouled conditions are simulated with clean tubes by testing at decreased evaporator water temperatures and increased condenser water temperatures. High Fouling Factors can increase or decrease these temperatures to conditions outside test loop or equipment capabilities. For this test standard the application range for the water side fouling shall be between clean (0.000) and 0.001000 h·ft^2·°F/Btu. Fouling factors above these values are outside of the scope of this standard and shall be noted as such.

<table>
<tr><th colspan="7">Table 2. Application Rating Conditions</th></tr>
<tr><th></th><th colspan="3">Evaporator</th><th colspan="3">Condenser</th></tr>
<tr><td rowspan="9">Cooling</td><th colspan="3">Water Cooled</th><th colspan="3">Water Cooled</th></tr>
<tr><td>Leaving Temperature[1], °F</td><td>Temperature Difference Across Heat Exchanger, °F</td><td>Fouling Factor Allowance, h·ft^2·°F/Btu</td><td>Entering Temperature[2], °F</td><td>Flow Rate, gpm/ton$_R$</td><td>Fouling Factor Allowance, h·ft^2·°F/Btu</td></tr>
<tr><td rowspan="7">36.0 to 60.0</td><td rowspan="7">5.0 to 20.0</td><td rowspan="7">0.000 to 0.001000</td><td>55.0 to 105.0</td><td>1.0 to 6.0</td><td>0.000 to 0.001000</td></tr>
<tr><th colspan="3">Air-Cooled</th></tr>
<tr><td colspan="3">Entering Air Dry Bulb[3], °F</td></tr>
<tr><td colspan="3">55.0 to 125.0</td></tr>
<tr><th colspan="3">Evaporatively Cooled</th></tr>
<tr><td colspan="3">Entering Air Wet Bulb[4], °F</td></tr>
<tr><td colspan="3">50.0 to 80.0</td></tr>
<tr><td rowspan="6">Heating</td><th colspan="3">Water Source Evaporator</th><th colspan="3">Water Cooled Condenser</th></tr>
<tr><td colspan="2">Entering Water Temperature[1], °F</td><td>Fouling Factor Allowance, h·ft^2·°F/Btu</td><td>Leaving Water Temperature[2], °F</td><td>Temperature Difference Across Heat Exchanger, °F</td><td>Fouling Factor Allowance, h·ft^2·°F/Btu</td></tr>
<tr><td colspan="2">40.0 to 80.0</td><td>0.000 to 0.001000</td><td rowspan="4">105.0 to 160.0</td><td rowspan="4">5.0 to 20.0</td><td rowspan="4">0.000 to 0.001000</td></tr>
<tr><th colspan="3">Air Source Evaporator</th></tr>
<tr><td colspan="3">Entering Air Temperature, °F</td></tr>
<tr><td colspan="3">15.0 to 60.0</td></tr>
<tr><td colspan="7">Notes
1. Evaporator water temperatures shall be published in rating increments of no more than 4.0°F.
2. Condenser water temperatures shall be published in rating increments of no more than 5.0°F.
3. Entering air temperatures shall be published in rating increments of no more than 10.0°F.
4. Air wet bulb temperatures shall be published in rating increments of no more than 2.5°F.</td></tr>
</table>

5.4 *Part-Load Rating For Cooling Only.* Water-Chilling Packages shall be rated at 100%, 75%, 50%, and 25% load relative to the full-load rating Net Refrigerating Capacity at the conditions defined in Table 3. For chillers capable of operating in multiple modes (cooling, heating, and /or heat reclaim), part-load ratings are only required for cooling mode operation. Part-load ratings are not required for heating mode operation or cooling operation with active heat reclaim operation.

Part-load rating points shall be presented in one or more of the following four ways:

a. IPLV. Based on the conditions defined in Table 3.
b. NPLV. Based on the conditions defined in Table 3.
c. Individual Part-Load Data Point(s) Suitable for Calculating IPLV or NPLV as defined in Table 3.
d. Within the application rating limits of Table 2, other part-load points that do not meet the requirements of footnotes (3) and (4) in Table 3 (i.e. variable water flow rates or other entering condenser water temperatures). Neither IPLV nor NPLV shall be calculated for such points.

5.4.1 *Determination of Part-Load Performance*. For Water-Chilling Packages covered by this standard, the IPLV or NPLV shall be calculated as follows:

a. Determine the part-load energy efficiency at 100%, 75%, 50%, and 25% load points at the conditions specified in Table 3.

b. Use the following equation to calculate the IPLV or NPLV for units rated with COP_R and EER.

$$\text{IPLV or NPLV} = 0.01A + 0.42B + 0.45C + 0.12D \qquad 8$$

For COP_R and EER:

Where: A = COP_R or EER at 100% load
B = COP_R or EER at 75% load
C = COP_R or EER at 50% load
D = COP_R or EER at 25% load

c. Use the following equation to calculate the IPLV or NPLV for units rated with kW/ton_R:

$$\text{IPLV or NPLV} = \frac{1}{\frac{0.01}{A}+\frac{0.42}{B}+\frac{0.45}{C}+\frac{0.12}{D}} \qquad 9$$

Where: A = Power Input per Capacity, kW/ton_R at 100% load
B = Power Input per Capacity, kW/ton_R at 75% load
C = Power Input per Capacity, kW/ton_R at 50% load
D = Power Input per Capacity, kW/ton_R at 25% load

5.4.1.1 For a derivation of Equations 8 and 9, and an example of an IPLV or NPLV calculation, see Appendix D. The weighting factors have been based on the weighted average of the most common building types and operations using average weather in 29 U.S. cities, with and without airside economizers.

Table 3. Part-Load Conditions for Rating

	IPLV[5]	NPLV
Evaporator (All Types)		
All loads LWT, ºF[2]	44.0	Selected LWT
Flow Rate, gpm/ton$_R$[3]	2.4	Selected flow rate
FFA, h·ft^2·ºF/Btu	0.000100	As Specified
Water-Cooled Condenser[1,2]		
100% load EWT, ºF[2]	85.0	Selected EWT
75% load EWT, ºF	75.0	Note[4]
50% load EWT, ºF	65.0	Note[4]
25% load EWT, ºF	65.0	Note[4]
Flow rate, gpm/ton$_R$[3]	3.0	Selected flow rate
FFA, h·ft^2·ºF/Btu	0.000250	As Specified
Air-Cooled Condenser[1]		
100% load EDB, ºF	95.0	No Rating Requirements
75% load EDB, ºF	80.0	
50% load EDB, ºF	65.0	
25% load EDB, ºF	55.0	
FFA, h·ft^2·ºF/Btu	0.000	
Evaporatively-Cooled Condenser[1]		
100% load EWB, ºF	75.00	No Rating Requirements
75% load EWB, ºF	68.75	
50% load EWB, ºF	62.50	
25% load EWB, ºF	56.25	
FFA, h·ft^2·ºF/Btu	0.000	
Air-Cooled Without Condenser		
100% load SDT, ºF	125.00	No Rating Requirements
75% load SDT, ºF	107.50	
50% load SDT, ºF	90.00	
25% load SDT, ºF	72.50	
FFA, h·ft^2·ºF/Btu	0.000	
Water-Cooled or Evaporatively-Cooled Without Condenser		
100% load SDT, ºF	105.0	No Rating Requirements
75% load SDT, ºF	95.0	
50% load SDT, ºF	85.0	
25% load SDT, ºF	75.0	
FFA, h·ft^2·ºF/Btu	0.000	

Notes:

1. If the unit manufacturer's recommended minimum temperatures are greater than those specified in Table 3, then those may be used in lieu of the specified temperatures.
2. Corrected for Fouling Factor Allowance by using the calculation method described in C6.3.
3. The flow rates are to be held constant at full-load values for all part-load conditions.
4. For part-load entering condenser water temperatures, the temperature should vary linearly from the selected EWT at 100% load to 65.0 ºF at 50% loads, and fixed at 65.0°F for 50% to 0% loads.
5. Reference Equations 10 through 14 for calculation of temperatures at any point that is not listed.
   5.1- Saturated discharge temperature (SDT).
   5.2- Leaving water temperature (LWT).
   5.3- Entering water temperature (EWT).
   5.4- Entering air dry-bulb temperature (EDB).
   5.5- Entering air wet-bulb temperature (EWB).

5.4.1.2 The IPLV or NPLV rating requires that the unit efficiency be determined at 100%, 75%, 50% and 25% at the conditions as specified in Table 3. If the unit, due to its capacity control logic cannot be operated at 75%, 50%, or 25% capacity then the unit shall be operated at other load points and the 75%, 50%, or 25% capacity efficiencies shall be determined by plotting the efficiency versus the % load using straight line segments to connect the actual performance points. The 75%, 50%, or 25% load efficiencies shall then be determined from the curve. Extrapolation of data shall not be used. An actual chiller capacity point, equal to, or less than the required rating point, must be used to plot the data. For example, if the minimum actual capacity is 33% then the curve can be used to determine the 50% capacity point, but not the 25% capacity point. For test points that are not run at the 75%, 50%, and 25% rating points, the condenser temperature for determination of IPLV shall be based on the measured part-load percentage for the actual test point using the Equations 10 through 14. For example for an air-cooled chiller test point run at 83% capacity, the entering air temperature for the test shall be 84.8 ºF (60·0.83 + 35).

Entering air dry-bulb temperature (EDB) [°F] for an Air-Cooled Condenser at IPLV part load conditions (refer to Figure 1):

$$\mathrm{EDB} = \begin{vmatrix} 60 \cdot \%\ \mathrm{Load} + 35 & \text{for Load } > 33\% \\ 55 & \text{for Load } \leq 33\% \end{vmatrix} \qquad 10$$

Note: In the case of an air-cooled chiller, the Load term used to calculate the EDB temperature is based on the adjusted capacity after using the barometric correction.

Entering water temperature (EWT) [°F] for a Water-Cooled Condenser at IPLV part load conditions (refer to Figure 1):

$$\mathrm{EWT} = \begin{vmatrix} 40 \cdot \%\ \mathrm{Load} + 45 & \text{for Load } > 50\% \\ 65 & \text{for Load } \leq 50\% \end{vmatrix} \qquad 11$$

Entering air wet-bulb temperature (EWB) [°F] for an Evaporatively-Cooled Condenser at IPLV part load conditions (refer to Figure 1):

$$\mathrm{EWB} = 25 \cdot \%\mathrm{Load} + 50 \qquad 12$$

Saturated discharge temperature (SDT) [°F] for an air-cooled unit without condenser at IPLV part load conditions (refer to Figure 1):

$$\mathrm{AC\ SDT} = 70 \cdot \%\mathrm{Load} + 55 \qquad 13$$

Saturated discharge temperature (SDT) [°F] for a water-cooled (WC) or evaporatively-cooled (EC) unit without condenser at IPLV part load conditions (refer to Figure 1):

$$\mathrm{WC\ \&\ EC\ SDT} = 40 \cdot \%\mathrm{Load} + 65 \qquad 14$$

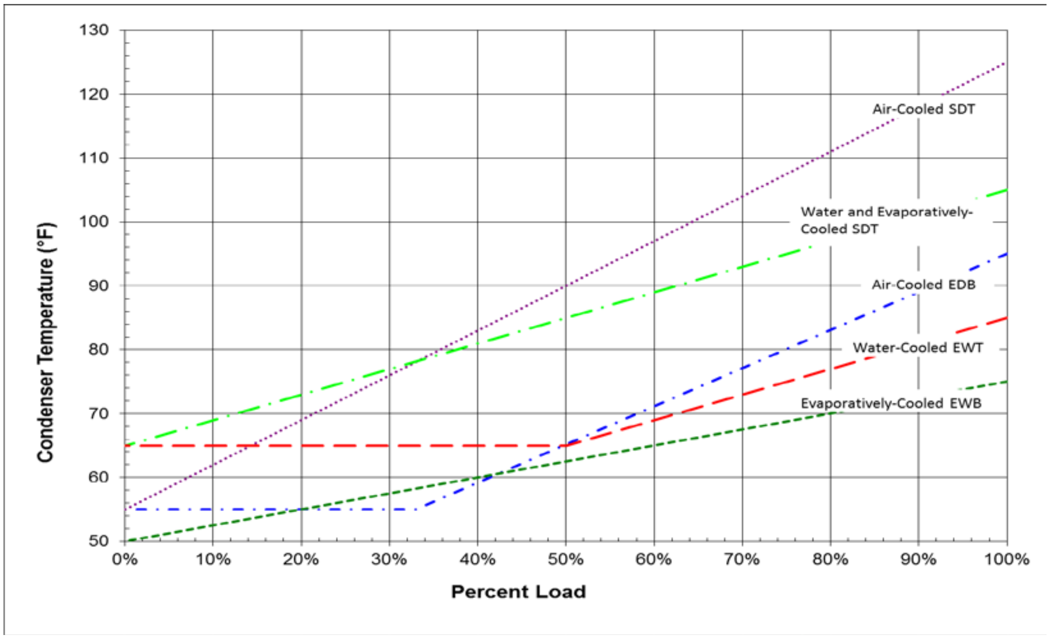

Figure 1. Part Load Condenser Temperature for IPLV Test Points

If a unit cannot be unloaded to the 25%, 50%, or 75% capacity point, then the unit shall be run at the minimum step of unloading at the condenser entering water or air temperature based on Table 3 for 25%, 50% or 75% capacity points as required. The efficiency shall then be determined by using one of the following three equations:

$$EER_{CD} = \frac{\mathrm{EER_{Test}}}{\mathrm{C_D}} \qquad 15a$$

$$\mathrm{COP_{R,CD}} = \frac{\mathrm{COP_{Test}}}{\mathrm{C_D}} \qquad 15b$$

$$\left(\frac{\mathrm{kW}}{\mathrm{ton_R}}\right)_{CD} = \left(\frac{\mathrm{kW}}{\mathrm{ton_R}}\right)_{Test} \cdot \mathrm{C_D} \qquad 15c$$

Where:

$\mathrm{COP_{Test}}$ = Efficiency at test conditions
$\mathrm{EER_{Test}}$ = Efficiency at test conditions
$\mathrm{kW/ton_{RTest}}$ = Efficiency at test conditions

$\mathrm{EER_{Test}}$, $\mathrm{COP_{Test}}$, $\mathrm{kW/ton_{RTest}}$ is the efficiency at the test conditions (after barometric pressure adjustment as per Appendix F, as applicable) and $\mathrm{C_D}$ is a degradation factor to account for cycling of the compressor for capacities less than the minimum step of capacity.

$\mathrm{C_D}$ shall be calculated using the following equation:

$$\mathrm{C_D} = (-0.13 \cdot \mathrm{LF}) + 1.13 \qquad 16$$

Where LF is the load factor calculated using the following equation:

$$\mathrm{LF} = \frac{(\%\mathrm{Load})\ (q_{\mathrm{ev\ 100\%}})}{(q_{\mathrm{ev\ min\%Load}})} \qquad 17$$

Where:

%Load is one of the standard rating points, i.e. 75%, 50%, or 25%
$q_{ev\ 100\%}$ is the rated unit net capacity at design conditions.
$q_{evmin\%Load}$ is the measured or calculated unit net capacity at the minimum step of capacity.

Part-Load unit capacity is the measured or calculated unit capacity from which Standard Rating points are determined using the method above.

5.4.1.3 *Sample Calculations.* The following are examples of the IPLV calculations:

Example 1

The chiller is a water-cooled centrifugal chiller that has proportional capacity control and can be tested at each of the four rating points of 100%, 75%, 50% and 25% as defined in Table 3. The chiller has a full-load rated capacity of 500 $tons_R$ and a full-load efficiency of 0.600 kW/ton_R. Table 4 shows the test results needed to compute IPLV.

Table 4. Chiller Performance – IPLV

% of full Load rated ton_R	Condenser EWT (ºF)	Capacity Target (ton_R)	Measured Net Refrigerating Capacity (ton_R)	Total Input Power (kW)	Efficiency[2] (kW/ton_R)	Deviation from capacity target (ton_R)	Percent difference from target based on full-load capacity (%)
100%	85.0	500.0	505.0	303.0	0.600	5	(5/500) or 1.0%
75%	75.0	375.0	381.0	174.1	0.457	6	(6/500) or 1.2% [1]
50%	65.0	250.0	245.0	85.5	0.349	-5	(-5/500) or -1.0% [1]
25%	65.0	125.0	130.0	56.7	0.436	5	(5/500) or 1.0% [1]

1. Because the chiller can be run within the capacity tolerances associated with the target loads required to calculate the IPLV, the above data can be used directly to calculate the IPLV (refer to Table 10 for test tolerances).
2. Because the Power Input per Capacity rating is in kW/ton_R use Equation 9.

$$\text{IPLV} = \frac{1}{\frac{0.01}{0.600} + \frac{0.42}{0.457} + \frac{0.45}{0.349} + \frac{0.12}{0.436}} = 0.400$$

Example 2

The chiller is a water-cooled centrifugal chiller that has proportional capacity control and can be tested at each of the four rating points of 100%, 75%, 50% and 25% as defined in Table 3. The chiller has a full-load rated capacity of 800 $tons_R$ and a full-load efficiency of 0.632 $kW/ton_{R.}$ The full load design conditions for the evaporator have a 42 ºF leaving water temperature at 2.0 gpm/ton water flow rate. The condenser conditions at full load design are 89 ºF entering water temperature with 3.0 gpm/ton water flow rate. Table 5 shows the test results needed to compute NPLV.

Table 5. Chiller Performance – NPLV							
% of full Load rated ton_R	Condenser EWT (ºF)	Capacity Target (ton_R)	Measured Capacity (ton_R)	Input Power (kW)	Efficiency[2] (kW/ton_R)	Deviation from capacity target (ton_R)	Percent difference from target based on full-load capacity (%)
100%	89.0	800.0	803.7	507.9	0.632	3.7	(3.7/800)= 0.5%
75%	77.0	600.0	608.2	316.9	0.521	8.2	(8.2/800)= 1.0% [1]
50%	65.0	400.0	398.5	183.3	0.460	-1.5	(-1.5/800)= -0.2% [1]
25%	65.0	200.0	211.2	125.2	0.593	11.2	(11.2/800)= 1.4% [1]

1. Because the chiller can be run within the capacity tolerances associated with the target loads required to calculate NPLV, the above data can be used directly to calculate the NPLV (refer to Table 10 for test tolerances).
2. Because the Power Input per Capacity rating is in kW/ton_R use Equation 9.

$$\text{NPLV} = \frac{1}{\frac{0.01}{0.632}+\frac{0.42}{0.521}+\frac{0.45}{0.460}+\frac{0.12}{0.593}} = 0.499$$

Example 3

The chiller is an air-cooled chiller rated at 150 $tons_R$. The full-load measured capacity is 148.2 $tons_R$ with an EER of 10.440. After barometric adjustment to sea level conditions, the capacity is 148.4 $tons_R$ with a full-load EER of 10.480. The unit has 10 stages of capacity control and can unload down to a minimum of 15% of rated load. Only the following 7 stages of capacity control shall be used for the computation of rating point data (Table 6). The degradation factor does not have to be used and the four IPLV rating efficiency levels can be obtained using interpolation (Figure 2). The barometric pressure was measured at 14.42 psia during the test. The following unit performance data is needed for IPLV computation:

Table 6. Chiller Performance – Interpolated Data										
Test Point	Measured EDB (ºF)	Measured Capacity (ton_R)	Measured Power (kW)	Efficiency (EER)	Capacity correction factor (App F)	Efficiency correction factor (App F)	Corrected Capacity (ton_R)	Corrected Efficiency (EER)	% of Rated Load	Target EDB (ºF)
1	94.8	148.2	170.3	10.440	1.0017	1.0039	148.4	10.480	99.0%	95.0
2	84.8	124.5	125.8	11.880	1.0014	1.0032	124.7	11.918	83.1%	84.9
3	77.2	105.7	93.8	13.521	1.0012	1.0027	105.8	13.558	70.6%	77.3
4	67.7	82.4	66.8	14.813	1.0009	1.0021	82.5	14.844	55.0%	68.0
5	60.1	62.8	49.5	15.213	1.0007	1.0016	62.8	15.238	41.9%	60.1
6	55.2	45.2	36.2	15.002	1.0005	1.0012	45.2	15.019	30.1%	55.0
7	55.3	22.5	19.0	14.212	1.0003	1.0006	22.5	14.220	15.0%	55.0

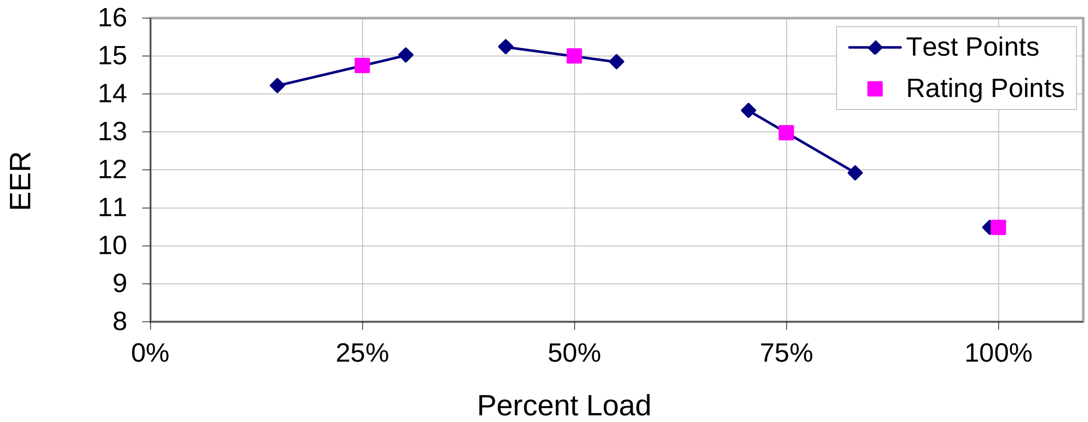

Figure 2. Rating Point Interpolation

Note that the chiller cannot run at the required rating points of 75%, 50% and 25%, but there are stages of capacity to either side of the 75%, 50%, and 25% rating points that allow for interpolation. The capacity stages closest to the rating points are used (Figure 2). Due to the fact that the chiller cannot run at the desired rating points, use Equation 10 to determine the target entering dry-bulb temperature (EDB). Use these target outdoor air temperatures when evaluating tolerance criteria in Table E2.

$$\mathrm{EER}_{25\%} = \left(\frac{25\% - 15.0\%}{30.1\% - 15.0\%}\right)(15.019 - 14.220) + 14.220 = 14.748$$

In a similar fashion, the 50% and 75% efficiency values are determined. The following performance is then used for the IPLV calculation.

Rating points	Efficiency (EER)
100.0%	10.480
75.0%	12.977
50.0%	14.994
25.0%	14.748

The IPLV can then be calculated using the efficiencies determined from the interpolation for the 75%, 50% and 25% rating points. Note: because the ratings are in EER, Equation 8 is used.

$$\mathrm{IPLV} = (0.01 \cdot 10.480) + (0.42 \cdot 12.977) + (0.45 \cdot 14.994) + (0.12 \cdot 14.748) = 14.072$$

Example 4

For this example we have an air-cooled chiller rated at 110 $\mathrm{tons_R}$. The full-load measured capacity is 110.0 $\mathrm{tons_R}$ with an EER of 9.558. After barometric adjustment to sea level conditions, the capacity is 110.5 $\mathrm{tons_R}$ with a full-load EER of 9.650. The unit has 3 stages of capacity with the last stage of capacity greater than the required 25% rating point. The degradation C_D factor will have to be used. The barometric pressure measured during the test was 14.20 psia. The actual and adjusted performance of the chiller is shown in Table 7.

Table 7. Actual and Adjusted Performance for Example 4										
Test Point	**Measured EDB (ºF)**	**Measured Capacity ($\mathrm{ton_R}$)**	**Measured Power (kW)**	**Efficiency (EER)**	**Capacity correction factor (App F)**	**Efficiency correction factor (App F)**	**Corrected Capacity ($\mathrm{ton_R}$)**	**Corrected Efficiency (EER)**	**% of Rated Load**	**Target EDB (ºF)**
1	95.0	110.0	138.1	9.558	1.0042	1.0096	110.5	9.650	100.4%	95.0
2	78.3	79.3	77.7	12.247	1.0030	1.0069	79.5	12.332	72.3%	78.4
3	59.8	45.4	39.0	13.969	1.0017	1.0040	45.5	14.024	41.3%	59.8
4[1]	55.0	46.5	40.9	13.643	1.0018	1.0041	46.6	13.699	42.3%	55.0
1. Denotes a case where a degradation factor (C_D) shall be used.										

Because the chiller cannot unload below 42.3%, interpolation cannot be used to determine the 25% rating point. An additional rating point needs to be determined at the lowest stage of capacity running at the entering dry bulb temperature of 55 ºF required for the 25% rating point.

First you will have to obtain the ratings for the 75% and 50% rating points by using interpolation.

For the 75% rating point you will have to interpolate between the 100.4% and 72.3% data points.

$$\mathrm{EER}_{75\%} = \left(\frac{75\% - 72.3\%}{100.4\% - 72.3\%}\right)(9.650 - 12.332) + 12.332 = 12.074$$

For the 50% rating point you then have to interpolate between the 72.3% and 41.3% data points.

$$\mathrm{EER}_{50\%} = \left(\frac{50\% - 41.3\%}{72.3\% - 41.3\%}\right) \cdot (12.332 - 14.024) + 14.024 = 13.549$$

For the 25% rating point the C_D factor will have to be used as there is not a lower capacity point to allow for interpolation:

$$\mathrm{LF} = \frac{0.25 \cdot 110}{46.6} = .590$$

$$C_D = (-0.13 \cdot 0.590) + 1.13 = 1.053$$

$$\mathrm{EER}_{25\%,\mathrm{CD}} = \frac{13.699}{1.053} = 13.009$$

The following performance is then used for the IPLV calculation.

Load point	Efficiency (EER)
100.0%	9.650
75.0%	12.074
50.0%	13.549
25.0%	13.009

The IPLV can then be calculated using the efficiencies determined from the interpolation for the 75%, 50% and 25% rating points. Note: because the ratings are in EER, Equation 8 is used.

$$\mathrm{IPLV} = (0.01 \cdot 9.650) + (0.42 \cdot 12.074) + (0.45 \cdot 13.549) + (0.12 \cdot 13.009) = 12.826$$

Example 5

For this example, the chiller is a water-cooled 15 ton_R positive displacement chiller with a full-load efficiency of 0.780 kW/ton_R. It only has 1 stage of capacity so the C_D degradation factor shall be used to generate the rating data for the 75%, 50%, and 25% rating points. The units can only run at full-load, thus additional rating information shall be obtained with the unit running at the 75 ºF entering condenser water temperature for the 75% rating point and at 65 ºF condenser entering water for the 50% and 25% rating points. The condenser water temperature is 65 ºF for both the 50% and 25% rating points, thus only 3 total test points are required to generate the IPLV rating data. The chiller has the following test information in Table 8.

Table 8. Performance Data for Example 5

Test Point	Condenser EWT (ºF)	Measured Capacity (ton_R)	Input Power (kW)	Efficiency (kW/ton_R)	Load
1	85.0	15.3	11.93	0.780	102.0%
2 [1]	75.0	17.3	10.6	0.617	115.0%
3 [1]	65.0	19.8	11.3	0.568	132.3%

1. Denotes a case where a degradation factor (C_D) shall be used.

For the 75% rating point the C_D factor will have to be used as there is not a lower capacity point to allow for interpolation.

$$\mathrm{LF} = \frac{0.75 \cdot 15}{17.3} = 0.652$$

$$\mathrm{C_D} = (\text{-}0.13 \cdot 0.652) + 1.13 = 1.045$$

$$\mathrm{kW/ton_{R75\%,CD}} = 1.045 \cdot 0.617 = 0.645$$

For the 50% rating point the C_D factor will have to be used as there is not a lower capacity point to allow for interpolation.

$$\mathrm{LF} = \frac{0.50 \cdot 15}{19.8} = 0.378$$

$$\mathrm{C_D} = (\text{-}0.13 \cdot 0.378) + 1.13 = 1.080$$

$$\mathrm{kW/ton_{R50\%,CD}} = 1.080 \cdot 0.568 = 0.613$$

For the 25% rating point the C_D factor will have to be used as there is not a lower capacity point to allow for interpolation.

$$\mathrm{LF} = \frac{0.25 \cdot 15}{19.8} = 0.189$$

$$\mathrm{C_D} = (\text{-}0.13 \cdot 0.189) + 1.13 = 1.105$$

$$\mathrm{kW/ton_{R25\%,CD}} = 1.105 \cdot 0.568 = 0.628$$

The following performance is then used for the IPLV calculation.

Load point	Efficiency (kW/ton$_R$)
100%	0.780
75%	0.645
50%	0.613
25%	0.628

The IPLV can then be calculated using the efficiencies determined from the interpolation for the 75%, 50% and 25% rating points. Note: because the ratings are in kW/ton$_R$, Equation 9 is used.

$$\text{IPLV} = \frac{1}{\frac{0.01}{0.780} + \frac{0.42}{0.645} + \frac{0.45}{0.613} + \frac{0.12}{0.628}} = 0.629$$

Example 6

For this example we have an air-cooled chiller with continuous unloading rated at 200 tons$_R$. The full-load measured capacity is 197.2 tons$_R$ with an EER of 9.718. After barometric adjustment to sea level conditions, the capacity is 199.2 tons$_R$ with a full-load EER of 9.938. The measured and adjusted performance, for both full and part-load test points, are shown in Table 9. The barometric pressure measured during the test was 13.50 psia.

Table 9. Actual and Adjusted Performance for Example 6

Test Point	Measured EDB (ºF)	Measured Capacity (ton$_R$)	Measured Power (kW)	Efficiency (EER)	Capacity correction factor (App F)	Efficiency correction factor (App F)	Corrected Capacity (ton$_R$)	Corrected Efficiency (EER)	% of Rated Load	Target EDB (ºF)
1	94.8	197.2	243.5	9.718	1.0100	1.0226	199.2	9.938	99.6%	95.0
2	79.5	149.1	146.0	12.252	1.0074	1.0169	150.2	12.459	75.1%	80.0
3	65.2	100.2	87.0	13.815	1.0050	1.0113	100.7	13.972	50.3%	65.0
4	55.1	51.2	46.5	13.226	1.0025	1.0058	51.3	13.303	25.7%	55.0

Note: Because the chiller can be run within the capacity tolerances associated with the target loads required to calculate the IPLV, the above data can be used directly to calculate the IPLV.

The IPLV can be calculated using the efficiencies determined from the 100%, 75%, 50% and 25% rating points. Note: because the ratings are in EER, Equation 8 is used.

$$\text{IPLV} = (0.01 \cdot 9.938) + (0.42 \cdot 12.459) + (0.45 \cdot 13.972) + (0.12 \cdot 13.303) = 13.216$$

5.5 *Fouling Factor Allowances*. When ratings are published, they shall include those with Fouling Factors as specified in Table 1. Additional ratings, or means of determining those ratings, at other Fouling Factor Allowances may also be published if the Fouling Factor is within the ranges defined in Section 5.3 and Table 2.

5.5.1 *Method of Establishing Clean and Fouled Ratings from Laboratory Test Data.*

5.5.1.1 A series of tests shall be run in accordance with the method outlined in Appendix C to establish the performance of the unit.

5.5.1.2 Evaporator water-side and condenser water-side or air-side heat transfer surfaces shall be considered clean during testing. Tests conditions will reflect Fouling Factors of zero (0.000) $h \cdot ft^2 \cdot {}^\circ F/Btu$.

5.5.1.3 To determine the capacity of the Water-Chilling Package at the rated fouling conditions, the procedure defined in Section C6.3 shall be used to determine an adjustment for the evaporator and or condenser water temperatures.

5.6 *Tolerances.*

5.6.1 *Allowable Tolerances.* The allowable test tolerance on net capacity, full and part load efficiency, and heat balance shall be determined from Table 10.

To comply with this standard, published or reported values shall be in accordance with Table 10.

<table>
<tr><th colspan="4">Table 10. Definition of Tolerances</th></tr>
<tr><th colspan="2"></th><th>Limits</th><th>Related Tolerance Equations[2,3,4]</th></tr>
<tr><td rowspan="2">Capacity</td><td>Cooling or Heating Capacity for units with continuous unloading[1]</td><td>Full Load minimum: 100% - Tol_1
Full Load maximum: 102%
Part Load test capacity shall be within 2% of the full-load rated $tons_R$ at the target part-load capacity</td><td rowspan="7">$$Tol_1 = 0.105 - (0.07 \cdot \%Load) + \left(\frac{0.15}{\Delta T_{FL} \cdot \%Load}\right) \quad 18$$
ΔT_{FL}= Difference between entering and leaving chilled water temperature at full-load, °F

See Figure 3 for graphical representation of the Tol_1 tolerance.</td></tr>
<tr><td>Cooling or Heating Capacity for units with discrete capacity steps</td><td>Full Load shall be at the maximum stage of capacity
Part Load test points shall be taken as close as practical to the specified part-load rating points as stated in Table 3</td></tr>
<tr><td colspan="2">Water cooled heat balance (HB)</td><td>$-Tol_1 \leq HB \leq +Tol_1$</td></tr>
<tr><td rowspan="6">Efficiency</td><td>EER</td><td>Minimum of:
(100%- Tol_1)·(rated EER)</td></tr>
<tr><td>kW/ton_R</td><td>Maximum of:
(100%+ Tol_1)·(rated kW/ton_R)</td></tr>
<tr><td>COP</td><td>Minimum of:
(100%- Tol_1)·(rated COP)</td></tr>
<tr><td>IPLV/NPLV (EER)</td><td>Minimum of:
(100%- Tol_2)·(rated EER)</td><td rowspan="3">$$Tol_2 = 0.065 + \left(\frac{0.35}{\Delta T_{FL}}\right) \quad 19$$
See Figure 4 for graphical representation of the Tol_2 tolerance.</td></tr>
<tr><td>IPLV/NPLV (kW/ton_R)</td><td>Maximum of:
(100%+ Tol_2)·(rated kW/ton_R)</td></tr>
<tr><td>IPLV/NPLV (COP_R)</td><td>Minimum of:
(100%- Tol_2)·(rated COP_R)</td></tr>
<tr><td colspan="2">Water Pressure Drop</td><td>Maximum of:
(1.15)·(rated pressure drop at rated flow rate)
or
rated pressure drop plus 2 feet of H_2O, whichever is greater</td><td></td></tr>
<tr><td colspan="4">Notes:
1. The target set point condenser entering temperatures (Figure 1) for continuous unloading units will be determined at the target part load test point.
2. For air-cooled units, all tolerances are computed for values after the barometric adjustment is taken into account.
3. %Load and Tol_1 are in decimal form.
4. Tol_2 is in decimal form.</td></tr>
</table>

The following figure is a graphical representation of the related tolerance equation for capacity, efficiency, and heat balance as noted in Table 10.

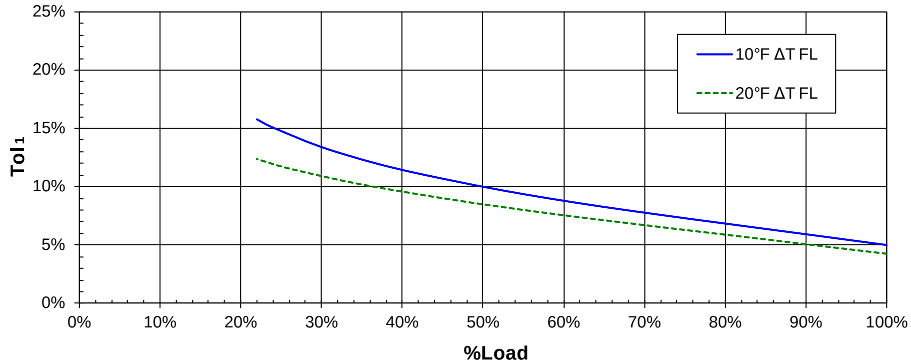

Figure 3. Allowable Tolerance (Tol_1) Curves for Full and Part Load Points

The following figure is a graphical representation of the related tolerance equation for IPLV and NPLV as noted in Table 10. The PLV line shown can represent either IPLV or NPLV depending on use.

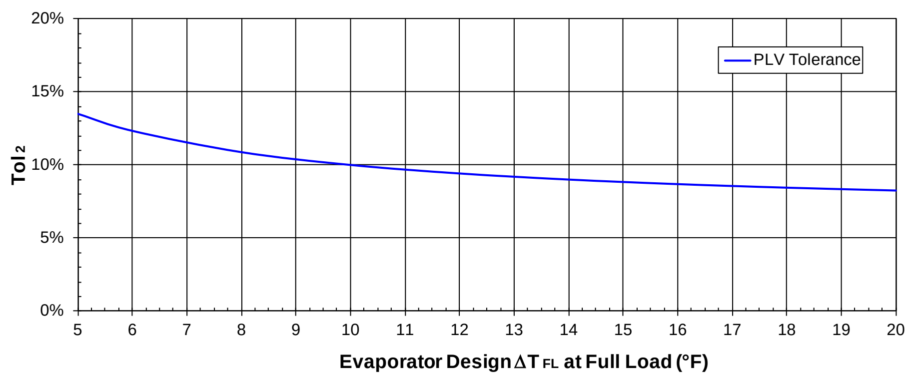

Figure 4. IPLV and NPLV Tolerance (Tol_2) Curve

5.6.2 *Full-Load Tolerance Examples.*

Full-Load Example in EER

Rated Full-Load Performance:

Rated Capacity = 100 ton_R
Rated Power = 111 kW
Cooling ΔT_{FL} = 10°F

$$\text{EER} = \frac{100\ \text{ton}_\text{R} \cdot 12{,}000\,\frac{\text{Btu}}{\text{h}\cdot\text{ton}_\text{R}}}{111\ \text{kW} \cdot 1{,}000\ \text{W/kW}} = 10.81\ \frac{\text{Btu}}{\text{W}\cdot\text{h}}$$

$$\text{Allowable Test Tolerance} = \text{Tol}_1 = 0.105 - (0.07 \cdot 1.00) + \left(\frac{0.15}{10 \cdot 1.00}\right) = 0.05 = 5.0\%$$

$$\text{Min. Allowable Tolerance} = 100\% - \text{Tol}_1 = 100\% - 5\% = 95\%$$

$$\text{Min. Allowable Capacity}(\text{ton}_\text{R}) = 95\% \cdot 100 = 95$$

$$\text{Min. Allowable EER}\left(\frac{\text{Btu}}{\text{W} \cdot \text{h}}\right) = 95\% \cdot 10.81 = 10.27$$

Full-Load Example in kW/ton$_R$

Rated Full-Load Performance:

Rated Capacity = 100 ton$_\text{R}$
Rated Power = 70 kW
Cooling ΔT_{FL} = 10ºF

$$\text{kW/ton}_\text{R} = \frac{70\ \text{kW}}{100\ \text{ton}_\text{R}} = 0.700\ \frac{\text{kW}}{\text{ton}_\text{R}}$$

$$\text{Allowable Test Tolerance} = \text{Tol}_1 = 0.105 - (0.07 \cdot 1.00) + \left(\frac{0.15}{10 \cdot 1.00}\right) = 0.05 = 5.0\%$$

$$\text{Min. Allowable Tolerance} = 100\% - \text{Tol}_1 = 100\% - 5\% = 95\%$$

$$\text{Min. Allowable Capacity} = 95\% \cdot 100\ \text{ton}_\text{R} = 95\ \text{ton}_\text{R}$$

$$\text{Max. Allowable Tolerance} = 100\% + \text{Tol}_1 = 100\% + 5\% = 105\%$$

$$\text{Max. Allowable kW/ton}_\text{R} = 105\% \cdot 0.70\ \text{kW/ton}_\text{R} = 0.735\ \text{kW/ton}_\text{R}$$

Full-Load Example in COP (Heat Pump)

Rated Full-Load Performance:

Rated Heating Capacity = 1,500,000 Btu/h
Rated Power = 70 kW
Condenser ΔT_{FL} = 10ºF

$$\text{Heating COP} = \frac{1{,}500{,}000\ \text{Btu/h}}{70\ \text{kW} \cdot 3{,}412.14\ \text{Btu/h} \cdot \text{kW}} = 6.28$$

$$\text{Allowable Test Tolerance} = \text{Tol}_1 = 0.105 - (0.07 \cdot 1.) + \left(\frac{0.15}{10 \cdot 1.00}\right) = 0.05 = 5.0\%$$

$$\text{Min. Allowable Tolerance} = 100\% - \text{Tol}_1 = 100\% - 5\% = 95\%$$

$$\text{Min. Allowable Capacity} = 95\% \cdot 1{,}500{,}000\ \text{Btu/h} = 1{,}425{,}000\ \text{Btu/h}$$

$$\text{Min. Allowable COP} = 95\% \cdot 6.28 = 5.97$$

5.6.3 *Part-Load*. The tolerance on part-load EER shall be the tolerance as determined from 5.6.1.

Part-Load Example in EER

Rated Part-Load Performance:

Power at 69.5% Rated Capacity = 59.6 kW
69.5% Rated Capacity = 69.5 $tons_R$
Cooling ΔT_{FL} = 10°F

$$\text{EER} = \frac{69.5\ \text{ton}_\text{R} \cdot 12{,}000\,\frac{\text{Btu}}{\text{h}\cdot\text{ton}_\text{R}}}{59.6\ \text{kW} \cdot 1{,}000\ \text{W/kW}} = 14.0\ \frac{\text{Btu}}{\text{W}\cdot\text{h}}$$

$$\text{Allowable Test Tolerance} = \text{Tol}_1 = 0.105 - (0.07 \cdot 0.695) + \left(\frac{0.15}{10 \cdot 0.695}\right) = 0.078 = 7.8\%$$

$$\text{Minimum Allowable Tolerance} = 100\% - \text{Tol}_1 = 100\% - 7.8\% = 92.2\%$$

$$\text{Minimum Allowable EER} = 92.2\% \cdot 14.0\ \frac{\text{Btu}}{\text{W}\cdot\text{h}} = 12.91\ \frac{\text{Btu}}{\text{W}\cdot\text{h}}$$

Part-Load Example in kW/ton_R

Rated Part-Load Performance:

Power at 50% Rated Capacity = 35 kW
50% Rated Capacity = 50 $tons_R$
Cooling ΔT_{FL} = 10 °F

$$\text{kW/ton} = \frac{35\ \text{kW}}{50\ \text{tons}_\text{R}} = 0.70\ \text{kW/ton}_\text{R}$$

$$\text{Allowable Test Tolerance} = \text{Tol}_1 = 0.105 - (0.07 \cdot 0.50) + \left(\frac{0.15}{10 \cdot 0.50}\right) = 0.10 = 10\%$$

$$\text{Maximum Allowable Tolerance} = 100\% + \text{Tol}_1 = 100\% + 10\% = 110\%$$

$$\text{Maximum Allowable kW/ ton}_\text{R} = 110\% \cdot 0.70 = 0.770\ \text{kW/ton}_\text{R}$$

Section 6. Minimum Data Requirements for Published Ratings

6.1 *Minimum Data Requirements for Published Ratings.* As a minimum, Published Ratings shall include all Standard Ratings. All claims to ratings within the scope of this standard shall include the statement "Rated in accordance with AHRI Standard 550/590 (I-P)." All claims to ratings outside the scope of the standard shall include the statement "Outside the scope of AHRI Standard 550/590 (I-P)." Wherever Application Ratings are published or printed, they shall include a statement of the conditions at which the ratings apply.

6.2 *Published Ratings.* Published Ratings shall state all of the standard operating conditions and shall include the following.

6.2.1 *General.*

6.2.1.1 Refrigerant designation in accordance with ANSI/ASHRAE Standard 34

6.2.1.2 Model number designations providing identification of the Water-Chilling Packages to which the ratings shall apply

6.2.1.3 Net Refrigerating Capacity (Equation 6), or Net Heating Capacity (Equation 7a), Btu/h or $tons_R$

6.2.1.4 Total Power Input to chiller, kW

6.2.1.4.1 Excluding power input to integrated water pumps, when present (refer to Section C3.1.5.6)

6.2.1.5 Energy Efficiency, expressed as EER, COP_R, COP_H, COP_{HR} or kW/ton_R.

It is important to note that pump energy associated with pressure drop through the chiller heat exchangers is not included in the chiller input power. This is done because any adjustment to the chiller performance would confuse the overall system analysis for capacity and efficiency. It is therefore important for any system analysis to account for the cooling loads associated with the system pump energy and to include the pump power into the overall equations for system efficiency.

6.2.1.6 Evaporator Fouling Factor, $h \cdot ft^2 \cdot {}^{\circ}F/Btu$, as stated in Table 1

6.2.1.7 Chilled water entering and leaving temperatures, ºF, as stated in Table 1, or leaving water temperature and temperature difference, ºF

6.2.1.8a Units with an integral pump: Evaporator heat exchanger Water Pressure Drop, ft H_2O

6.2.1.8b Units without an integral pump: Chilled Water Pressure Drop (customer inlet to customer outlet), ft H_2O

Note: Due to industry standard practice, Water Pressure Drop is reported in head, ft H_2O; however test data is acquired in pressure, psid, for use in calculations.

6.2.1.9 Chilled water flow rate, gpm, at entering heat exchanger conditions

6.2.1.10 Nominal voltage, V, and frequency, Hz, for which ratings are valid. For units with a dual nameplate voltage rating, testing shall be performed at the lower of the two voltages

6.2.1.11 Components that utilize Auxiliary Power shall be listed

6.2.1.12 Part load weighted efficiency metric IPLV/NPLV, expressed as EER, COP_R, or kW/ton_R

6.2.2 *Water-Cooled Condenser Packages.*

6.2.2.1 Condenser Water Pressure Drop (inlet to outlet), ft H_2O

6.2.2.2 Any two of the following:

6.2.2.2.1 Entering condenser water temperature, ºF

6.2.2.2.2 Leaving condenser water temperature, ºF

6.2.2.2.3 Water temperature rise through the condenser, ºF

6.2.2.3 Condenser water flow rate, gpm at entering heat exchanger conditions.

6.2.2.4 Condenser Fouling Factor, h·ft^2·ºF/Btu, as stated in Table 1.

6.2.3 *Air-Cooled Condenser Packages.*

6.2.3.1 Entering air dry-bulb temperature, ºF, as stated in Table 1

6.2.3.2 Power input to fan(s), kW

6.2.4 *Evaporatively-Cooled Condenser Packages.*

6.2.4.1 Entering air wet-bulb temperature, ºF, as stated in Table 1

6.2.4.2 Power input to fan(s), kW

6.2.4.3 Condenser spray pump power consumption, kW

6.2.4.4 Statement of condenser Fouling Factor Allowance on heat exchanger, h·ft^2·ºF/Btu

6.2.5 *Packages without Condenser (for use with Remote Condensers).*

6.2.5.1 Compressor saturated discharge temperature (SDT) (refer to definition 3.4), ºF, as stated in Table 1

6.2.5.2 Liquid Refrigerant Temperature (LIQ) entering chiller package, ºF, as stated in Table 1

6.2.5.3 Condenser heat rejection capacity requirements, Btu/h

6.2.6 *Heat Reclaim Condenser(s).*

6.2.6.1 Heat Reclaim net capacity, MBtu/h

6.2.6.2 Water Pressure Drop (inlet to outlet), ft H_2O

6.2.6.3 Entering and leaving heat reclaim condenser water temperatures, ºF, as stated in Table 1

6.2.6.4 Heat reclaim condenser water flow rate, gpm at entering heat exchanger conditions

6.2.6.5 Fouling Factor, h·ft^2·ºF/Btu, as stated in Table 1

6.2.7 *Water-to-Water Heat Pumps*

6.2.7.1 Heating Capacity, MBtu/h

6.2.7.2 Water Pressure Drop (inlet to outlet), ft H_2O

6.2.7.3 Entering and leaving condenser water temperatures, ºF, as stated in Table 1

6.2.7.4 Condenser water flow rate, gpm at entering heat exchanger conditions

6.2.7.5 Fouling Factor, h·ft^2·ºF/Btu, as stated in Table 1

6.2.7.6 Any two of the following:

6.2.7.6.1 Entering evaporator water temperature, ºF

6.2.7.6.2 Leaving evaporator water temperature, ºF

6.2.7.6.3 Water temperature difference through the evaporator, ºF

6.2.8 *Air-to-Water Heat Pumps*

6.2.8.1 Heating Capacity, MBtu/h

6.2.8.2 Water Pressure Drop (inlet to outlet), ft H_2O

6.2.8.3 Entering and leaving condenser water temperatures, ºF, as stated in Table 1

6.2.8.4 Condenser water flow rate, gpm at entering heat exchanger conditions

6.2.8.5 Fouling Factor, $h \cdot ft^2 \cdot {}^{\circ}F/Btu$, as stated in Table 1

6.2.8.6 Entering air dry-bulb temperature, ºF, as stated in Tables 1 and 2

6.2.8.7 Entering air wet-bulb temperature, ºF, as stated in Table 1

6.2.8.8 Power input to fan(s), kW

6.3 *Summary Table of Data to be published*

Table 11 provides a summary of Section 6 items. In case of discrepancy, the text version shall be followed.

<table>
<tr><th colspan="12">Table 11. Published Values</th></tr>
<tr><th>Published Values</th><th>Units</th><th>Water-Cooled Chiller (Cooling)</th><th>Water-Cooled Heat Reclaim Chiller</th><th>Evaporatively Cooled Chiller</th><th>Air-Cooled Chiller</th><th>Condenserless Chiller</th><th>Air-Cooled HP (Cooling)</th><th>Air-Cooled HP (Heating)</th><th>Air Cooled Heat Reclaim Chiller</th><th>Water to Water HP (Cooling)</th><th>Water to Water HP (Heating)</th></tr>
<tr><td colspan="12">General</td></tr>
<tr><td>Voltage</td><td>V</td><td>■</td><td>■</td><td>■</td><td>■</td><td>■</td><td>■</td><td>■</td><td>■</td><td>■</td><td>■</td></tr>
<tr><td>Frequency</td><td>Hz</td><td>■</td><td>■</td><td>■</td><td>■</td><td>■</td><td>■</td><td>■</td><td>■</td><td>■</td><td>■</td></tr>
<tr><td>Refrigerant Designation</td><td></td><td>■</td><td>■</td><td>■</td><td>■</td><td>■</td><td>■</td><td>■</td><td>■</td><td>■</td><td>■</td></tr>
<tr><td>Model Number</td><td></td><td>■</td><td>■</td><td>■</td><td>■</td><td>■</td><td>■</td><td>■</td><td>■</td><td>■</td><td>■</td></tr>
<tr><td colspan="12">Net Capacity</td></tr>
<tr><td>Refrigeration Capacity</td><td>$tons_R$</td><td>■</td><td>■</td><td>■</td><td>■</td><td>■</td><td>■</td><td></td><td>■</td><td>■</td><td>■</td></tr>
<tr><td>Heat Rejection Capacity</td><td>Btu/h</td><td>■</td><td>■</td><td></td><td></td><td>■</td><td></td><td>■</td><td>■</td><td>■</td><td>■</td></tr>
<tr><td>Heat Reclaim Capacity</td><td>Btu/h</td><td></td><td>■</td><td></td><td></td><td></td><td></td><td></td><td>■</td><td></td><td></td></tr>
<tr><td colspan="12">Efficiency</td></tr>
<tr><td>Cooling EER</td><td>Btu/W·h</td><td rowspan="3">■</td><td rowspan="3">■</td><td>■</td><td>■</td><td>■</td><td>■</td><td></td><td rowspan="2">■</td><td rowspan="3">■</td><td rowspan="3">■</td></tr>
<tr><td>Cooling COP</td><td>W/W</td><td></td><td></td><td></td><td></td><td></td></tr>
<tr><td>Cooling $kW/tons_R$</td><td>$kW/tons_R$</td><td></td><td></td><td></td><td></td><td></td><td></td></tr>
<tr><td>Heating COP</td><td>W/W</td><td></td><td></td><td></td><td></td><td></td><td></td><td>■</td><td></td><td></td><td>■</td></tr>
<tr><td>Heat Reclaim COP</td><td>W/W</td><td></td><td>■</td><td></td><td></td><td></td><td></td><td></td><td>■</td><td></td><td></td></tr>
<tr><td rowspan="3">IPLV/NPLV</td><td>Btu/W·h</td><td rowspan="3">■</td><td rowspan="3"></td><td rowspan="3">■</td><td rowspan="3">■</td><td rowspan="3">■</td><td rowspan="3">■</td><td rowspan="3"></td><td rowspan="3"></td><td rowspan="3">■</td><td rowspan="3"></td></tr>
<tr><td>W/W</td></tr>
<tr><td>$kW/tons_R$</td></tr>
<tr><td colspan="12">Power</td></tr>
<tr><td>Total Power</td><td>kW</td><td>■</td><td>■</td><td>■</td><td>■</td><td>■</td><td>■</td><td>■</td><td>■</td><td>■</td><td>■</td></tr>
<tr><td>Condenser Spray Pump Power</td><td>kW</td><td></td><td></td><td>■</td><td></td><td></td><td></td><td></td><td></td><td></td><td></td></tr>
</table>

Table 11. Published Values

Published Values	Units	Water-Cooled Chiller (Cooling)	Water-Cooled Heat Reclaim Chiller	Evaporatively Cooled Chiller	Air-Cooled Chiller	Condenserless Chiller	Air-Cooled HP (Cooling)	Air-Cooled HP (Heating)	Air Cooled Heat Reclaim Chiller	Water to Water HP (Cooling)	Water to Water HP (Heating)
Fan Power	kW			■	■		■	■	■		
Cooling Mode Evaporator											
Entering Water[1]	ºF	■	■	■	■	■	■	■	■	■	■
Leaving Water[1]	ºF	■	■	■	■	■	■	■	■	■	■
Flow	gpm	■	■	■	■	■	■	■	■	■	■
Pressure Drop	ft H_2O	■	■	■	■	■	■	■	■	■	■
Fouling Factor	h·ft^2·ºF/Btu	■	■	■	■	■	■	■	■	■	■
Cooling Mode Heat Rejection Exchanger											
Tower Condenser											
Entering Water[1]	ºF	■	■								
Leaving Water[1]	ºF	■	■								
Flow	gpm	■	■								
Pressure Drop	ft H_2O	■	■								
Fouling Factor	h·ft^2·ºF/Btu	■	■								
Heat Reclaim Condenser											
Entering Water[1]	ºF		■						■		
Leaving Water[1]	ºF		■						■		
Flow	gpm		■						■		
Pressure Drop	ft H_2O		■						■		
Fouling Factor	h·ft^2·ºF/Btu		■						■		
Dry-bulb air	ºF								■		
Heat Rejection Condenser											
Entering Water[1]	ºF									■	■
Leaving Water[1]	ºF									■	■
Flow	gpm									■	■
Pressure Drop	ft H_2O									■	■
Fouling Factor	h·ft^2·ºF/Btu									■	■
Evaporatively Cooled											
Dry-bulb	ºF			■							
Wet-bulb	ºF			■							
Air Cooled											
Dry-bulb	ºF				■		■	■	■		
Wet-bulb	ºF							■			
Without Condenser											
Saturated Discharge	ºF					■					
Liquid Temperature or Subcooling	ºF					■					

1. An alternate to providing entering and leaving water temperatures is to provide one of these along with the temperature difference across the heat exchanger

Section 7. Conversions and Calculations

7.1 *Conversions.* For units that require conversion the following factors shall be utilized:

Table 12. Conversion Factors		
To Convert From	**To**	**Multiply By**
1 ft H_2O (at 60ºF)[1]	psi	0.43310
inch Hg	psia	0.49115
kilowatt (kW)[2]	Btu/h	3412.14
ton of refrigeration	Btu/h	12000
ton of refrigeration[2]	kilowatt (kW)	3.51685

1. For water side pressure drop, the conversion from water column “ft H_2O” to “psi” is per ASHRAE Fundamentals Handbook 2009. Note that 60ºF is used as the reference temperature for the density of water in the manometer.

2. The British thermal unit (Btu) used in this standard is the International Table Btu. The Fifth International Conference on the Properties of Steam (London, July 1956) defined the calorie (International Table) as 4.1868 J. Therefore, the exact conversion factor for the Btu (International Table) is 1.055 055 852 62 kJ.

7.2 *Water Side Properties Calculation Methods.* The following calculation methods shall be utilized:

7.2.1 Water density, ρ, (lbm/ft^3) =

$$(-7.4704 \cdot 10^{-10} \cdot t^4) + (5.2643 \cdot 10^{-7} \cdot t^3) - (1.8846 \cdot 10^{-4} \cdot t^2) + (1.2164 \cdot 10^{-2} \cdot t) + 62.227 \qquad 20$$

7.2.2 Specific Heat, c_p, (Btu/lbm·ºF) =

$$(-4.0739 \cdot 10^{-13} \cdot t^5) + (3.1031 \cdot 10^{-10} \cdot t^4) - (9.2501 \cdot 10^{-8} \cdot t^3) + (1.4071 \cdot 10^{-5} \cdot t^2) - (1.0677 \cdot 10^{-3} \cdot t) + 1.0295 \qquad 21$$

Where:

t = water temperature, ºF

Refer to the equation using the density and specific heat terms for the value of the water temperature to be used.

Note: Specific heat and density are curve fit at 100 psia from data generated by NIST Refprop v9.0 using a temperature range of 32 to 212 ºF. The 100 psia value used for the water property curve fits was established as a representative value to allow for the calculation of water side properties as a function of temperature only. This eliminates the complexity of measuring and calculating water side properties as a function of both temperature and pressure. This assumption, in conjunction with a formulation for capacity that does not make explicit use of enthalpy values, provides a mechanism for computing heat exchanger capacity for fluids other than pure water where specific heat data are generally known but enthalpy curves are not available.

Section 8. Marking and Nameplate Data

8.1 *Marking and Nameplate Data*. As a minimum, the nameplate shall display the following:

8.1.1 Manufacturer's name and location
8.1.2 Model number designation providing performance-essential identification
8.1.3 Refrigerant designation (in accordance with ANSI/ASHRAE Standard 34)
8.1.4 Voltage, phase and frequency

8.1.5 Serial number

8.2 *Nameplate Voltage.* Where applicable, nameplate voltages for 60 Hertz systems shall include one or more of the equipment nameplate voltage ratings shown in Table 1 of ANSI/AHRI Standard 110. Where applicable, nameplate voltages for 50 Hertz systems shall include one or more of the utilization voltages shown in Table 1 of IEC Standard 60038.

Section 9. Conformance Conditions

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

APPENDIX A. REFERENCES – NORMATIVE

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

A1.1 AHRI Standard 551/591-2011, *Performance Rating of Water-Chilling and Heat Pump Water-Heating Packages Using the Vapor Compression Cycle*, 2011, Air-Conditioning, Heating and Refrigeration Institute, 2111 Wilson Boulevard, Suite 500, Arlington, VA 22201, U.S.A.

A1.2 ANSI/AHRI Standard 110-2002, *Air-Conditioning and Refrigerating Equipment Nameplate Voltages*, 2002, Air-Conditioning and Refrigeration Institute, 2111 Wilson Boulevard, Suite 500, Arlington, VA 22201, U.S.A.

A1.3 ANSI/ASHRAE Standard 34-2010 with Addenda, *Number Designation and Safety Classification of Refrigerants*, 2007, American Society of Heating, Refrigeration, and Air-Conditioning Engineers, Inc., ASHRAE, 25 West 43rd Street, 4th Fl., New York, NY, 10036, U.S.A./1791 Tullie Circle, N.E., Atlanta, Georgia, 30329, U.S.A.

A1.4 ASHRAE *Fundamentals Handbook*, 2009, American Society of Heating, Refrigeration, and Air-Conditioning Engineers, Inc., 2009, ASHRAE, 1791 Tullie Circle, N.E., Atlanta, Georgia, 30329, U.S.A.

A1.5 ASHRAE Standard 30-1995, *Method of Testing Liquid Chilling Packages*, 1995, American Society of Heating, Refrigeration, and Air-Conditioning Engineers, Inc. ASHRAE, 25 West 43rd Street, 4th Fl., New York, NY, 10036, U.S.A./1791 Tullie Circle, N.E., Atlanta, Georgia, 30329, U.S.A.

A1.6 ASHRAE Standard 41.1-86 (RA 2006), *Measurements Guide - Section on Temperature Measurements*, 2001, American Society of Heating, Refrigeration, and Air-Conditioning Engineers, Inc. ASHRAE, 25 West 43rd Street, 4th Fl., New York, NY, 10036, U.S.A./1791 Tullie Circle, N.E., Atlanta, Georgia, 30329, U.S.A.

A1.7 ASHRAE Standard 41.3-1989 (RA2006), Standard Method for Pressure Measurement, 2006, American Society of Heating, Refrigeration, and Air-Conditioning Engineers, Inc. ASHRAE, 25 West 43rd Street, 4th Fl., New York, NY, 10036, U.S.A./1791 Tullie Circle, N.E., Atlanta, Georgia, 30329, U.S.A.

A1.8 ASHRAE *Terminology of Heating Ventilation, Air Conditioning and Refrigeration,* Second Edition, 1991, American Society of Heating, Refrigeration, and Air-Conditioning Engineers, Inc. ASHRAE, 1791 Tullie Circle, N.E., Atlanta, Georgia, 30329, U.S.A.

A1.9 ASME Standard PTC 19.2-2010, *Pressure Measurement*, *Instruments and Apparatus Supplement*, *2010*, American Society of Mechanical Engineers. ASME, Three Park Avenue, New York, NY 10016, U.S.A.

A1.10 ASME Standard MFC-3M-2004, Measurement of Fluid Flow in Pipes Using Orifice, Nozzle, and Venturi, 2004, American Society of Mechanical Engineers. ASME, Three Park Avenue, New York, NY 10016, U.S.A.

A1.11 ASME Standard MFC-6M-1998, Measurement of Fluid Flow in Pipes Using Vortex Flowmeters,1998 (R2005), American Society of Mechanical Engineers. ASME, Three Park Avenue, New York, NY 10016, U.S.A.

A1.12 ASME Standard MFC-11-2006, Measurement of Fluid Flow by Means of Coriolis Mass Flowmeters, 2006, American Society of Mechanical Engineers. ASME, Three Park Avenue, New York, NY 10016, U.S.A.

A1.13 ASME Standard MFC-16-2007, Measurement of Liquid Flow in Closed Conduits With Electromagnetic Flowmeters, 2007, American Society of Mechanical Engineers. ASME, Three Park Avenue, New York, NY 10016, U.S.A.

A1.14 Crane Technical Paper Number 410, 2009 edition.

A1.15 Excel Spreadsheet for the Computation of the Pressure Drop Adjustment Factors per Appendix G. Available as download from the AHRI web site (http://www.ahrinet.org/search+standards.aspx). Air-Conditioning and Refrigeration Institute, 2111 Wilson Boulevard, Suite 500, Arlington, VA 22201, U.S.A.

A1.16 IEC Standard 60038, *IEC Standard Voltages*, 2000, International Electrotechnical Commission, rue de Varembe, P.O. Box 131, 1211 Geneva 20, Switzerland.

A1.17 IEEE 120-1989 (RA97), *Master Test Guide for Electrical Measurements in Power Circuits*, Institute of Electrical and Electronic Engineers, 1997.

A1.18 ISA Standard RP31.1, *Recommended Practice Specification, Installation, and Calibration of Turbine Flowmeters,* 1977, Instrument Society of America, ISA, 67 Alexander Drive, P.O. Box 12277, Research Triangle Park, NC 27709, U.S.A.

APPENDIX B. REFERENCES – INFORMATIVE

B1.1 ANSI/ASHRAE Standard 37-2009, Method of Testing for Ratings Electrically Driven Unitary Air Conditioning and Heat Pump Equipment, 2009 American Society of Heating, Refrigeration, and Air-Conditioning Engineers, Inc., ASHRAE, 25 West 43rd Street, 4th Fl., New York, NY, 10036, U.S.A./1791 Tullie Circle, N.E., Atlanta, Georgia, 30329, U.S.A.

B1.2 ASHRAE Standard 90.1-2010, *Energy Standard for Buildings Except for Low-Rise Residential Buildings, 2010*, American Society of Heating, Refrigeration, and Air-Conditioning Engineers, Inc. ASHRAE, 1791 Tullie Circle, N.E., Atlanta, Georgia, 30329, U.S.A.

B1.3 ASHRAE Standard 140-2001, Standard Method of Test for the Evaluation of Building Energy Analysis Computer Programs, 2001, American Society of Heating, Refrigeration, and Air-Conditioning Engineers, Inc. ASHRAE, 25 West 43rd Street, 4th Fl., New York, NY, 10036, U.S.A./1791 Tullie Circle, N.E., Atlanta, Georgia, 30329, U.S.A.

B1.4 ASHRAE Technical Report, *Develop Design Data of Large Pipe Fittings, 2010,* American Society of Heating, Refrigeration, and Air-Conditioning Engineers, Inc. ASHRAE, 1791 Tullie Circle, N.E., Atlanta, Georgia, 30329, U.S.A

B1.5 ASHRAE *Fundamentals Handbook,* Second Edition, 1991, American Society of Heating, Refrigeration, and Air-Conditioning Engineers, Inc. ASHRAE, 1791 Tullie Circle, N.E., Atlanta, Georgia, 30329, U.S.A.

B1.6 *Commercial Buildings Characteristics 1992;* April 1994, DOE/EIA-0246(92).

B1.6 ISO/IEC Standard 17025 *General Requirements for the Competence of Testing and Calibration Laboratories*, 2005, International Standards Organization. 1 ch. de la Voie-Creuse CP 56, CH-1211 Geneva 20, Switzerland.

B1.7 NIST Reference Fluid Thermodynamic and Transport Properties Database (Refprop) v9.0., 2009.

APPENDIX C. METHOD OF TESTING WATER-CHILLING AND WATER-HEATING PACKAGES USING THE VAPOR COMPRESSION CYCLE – NORMATIVE

C1. *Purpose*. The purpose of this appendix is to prescribe a method of testing for Water-Chilling and Water-Heating Packages using the vapor compression cycle and to verify capacity and power requirements at a specific set of conditions.

Testing shall occur at a laboratory site where instrumentation is in place and load stability can be obtained.

Testing shall not be conducted in field installations to the provisions of this standard. Steady state conditions and requirements for consistent, reliable measurement are difficult to achieve in field installations.

C2. *Definitions*. Definitions for this appendix are identical to those in Section 3 of AHRI Standard 550/590.

C3. *Test Methods.*

C3.1 *Test Method.*

C3.1.1 The test will measure cooling and/or Heating Capacity (both net and gross) and may include heat recovery capacity and energy requirements, at a specific set of conditions.

C3.1.2 A test loop must reach steady-state conditions prior to beginning a test. Steady-state conditions must be maintained during testing. This is confirmed when each of the four sets of test data, taken, at five-minute intervals, are within the tolerances set forth in C6.2.1.

To minimize the effects of transient conditions, individual measurements of test data should be taken as simultaneously as possible. Software may be used to capture data over the course of the 15 minute test but must provide at least four distinct readings at five-minute intervals (i.e. at 0, 5, 10 and 15).

C3.1.3 The test shall include a measurement of the heat added to or removed from the water as it passes through the heat exchanger by determination of the following values (See Section C7 for calculations):

C3.1.3.1 Determine water mass flow rate, lbm/hr

Note:(refer to AHRI Standard 550/590 (I-P) Section 7.2) if a volumetric flow meter was used, the conversion to mass flow shall use the density corresponding to either:

C3.1.3.1.1 The temperature of the water at the location of flow meter; or

C3.1.3.1.2 The water temperature measurement, either entering or leaving, which best represents the temperature at the flow meter.

C3.1.3.2 Measure entering and leaving water temperatures, °F

Note: Units with an optional integrated evaporator or condenser water pump shall be tested in either of the following 2 modes.

C3.1.3.2.1 If the pump is to be operational during the test, the pump shall not be located between the entering and leaving water temperature measurement locations. In this case the unit must be modified to include a temperature measurement station between the pump and the heat exchanger. Care must be taken to ensure proper water mixing for an accurate representation of the bulk fluid temperature.

C3.1.3.2.2 If the pump is not operational during the test, temperature measurements external to the unit shall be used. In this case, the water shall flow freely through the pump with the pump in the off position.

C3.1.3.3 Measure pressure drop across the heat exchanger, psid

C3.1.3.3.1 Static pressure taps shall be located external to the unit per Appendix G. Appendix G specifies the acceptable adjustment factors to be used to adjust the pressure drop measurement for external piping between the static pressure tap and the unit connections.

C3.1.3.3.2 For units containing an integrated water pump, the measured pressure drop shall not include the effects of the pump. For these cases, the pressure drop measurement is to be taken across the heat exchanger only and will not include the pressure rise associated with the pump that is operational or the pressure drop of a non-operational pump or other internal components. A single static pressure tap upstream and downstream of the heat exchanger is acceptable.

C3.1.4 If a heat reclaim condenser is included, the test shall include simultaneous determination of the heat reclaim condenser capacity by obtaining the data as defined in Section C5.1.6 for Water-Cooled Heat Reclaim Condensers. Measurement methods shall follow the same procedures defined in Sections C3.1.3.1 through C3.1.3.3.

C3.1.5 *Electric Drive.* The test shall include the determination of the unit power requirement. For electrical drives, this power shall be determined by measurement of electrical input to the chiller.

C3.1.5.1 For motors supplied by others, the determination of compressor shaft horsepower input shall be outlined in the test procedure.

C3.1.5.2 For units provided with self-contained starters, transformers, or variable speed drives, the unit power requirement shall include the losses due to the starter, transformer, and drive and shall be tested on the line side.

C3.1.5.3 For units with remote mounted or customer supplied starters, the unit power measurement will be comprised of the power supplied at the compressor terminals and the auxiliaries needed to run the unit.

C3.1.5.4 For units supplied with remote mounted variable speed drives, the unit power measurement shall be taken at the line side of a drive that has similar losses and speed control to that supplied to the customer.

C3.1.5.5 For Air-Cooled or Evaporatively-Cooled Condensers, the test shall include the Condenser fan and Condenser spray pump power.

C3.1.5.6 For units containing optional integrated water pumps, the test measurements shall exclude the pump power from the measurement of chiller input power (refer to Section C5.1.1.4. For units tested with the integral pump turned off, the electrical power connection from the pump motor must be physically disconnected from the unit power by means of a contactor or disconnected wiring.

C3.1.6 *Non-Electric Drive.* Where turbine or engine drive is employed, compressor shaft horsepower input shall be determined from steam, gas, or oil consumption, at measured supply and exhaust conditions and prime mover manufacturer's certified performance data.

C3.1.7 *Test Verification*.

C3.1.7.1 For the case of Water-Cooled Condensers, data shall be taken to prepare a heat balance (C6.4.1) to substantiate the validity of the test.

C3.1.7.2 For Air-Cooled and Evaporatively-Cooled Condensers, it is impractical to measure heat rejection in a test; therefore, a heat balance cannot be calculated. To verify test accuracy, concurrent redundant instrumentation method (C6.4.2) shall be used to measure water temperatures, flow rates, and power inputs.

C3.1.7.3 For heat reclaim units with Air-Cooled Condensers or Water-Cooled Condensers, where the capacity is not sufficient to fully condense the refrigerant, the concurrent redundant instrumentation methods (C6.4.2) shall be used.

C3.1.7.4 For heat reclaim units with Water-Cooled Condensers that fully condense the refrigerant, the heat balance methods (C6.4.1) shall be used.

C3.1.8 *Air-Cooled Chiller Testing.* Temperature conditions shall be maintained per Table 1 or Table 2. Set up procedures and tolerances shall comply with the provisions detailed in Appendix E.

C3.1.9 *Air Source Heat Pump Testing.* Temperature conditions shall be maintained per Table 1 or Table 2. Set up procedures and tolerances shall comply with the provisions detailed in Appendix E, with additional requirements detailed in Appendix H.

C3.2 *Condition of Heat Transfer Surfaces.*

C3.2.1 Tests conducted in accordance with this standard may require cleaning (in accordance with manufacturer's instructions) of the heat transfer surfaces. The as tested Fouling Factors shall then be assumed to be zero (0.000) $h \cdot ft^2 \cdot °F/Btu$.

C4 *Instrumentation.*

C4.1 Instruments shall be selected, installed, operated, and maintained according to the requirements of Table C1.

C4.2 All instruments and measurement systems shall be calibrated over a range that exceeds the range of test readings. Data acquisition systems shall be either calibrated as a system, or all individual component calibrations shall be documented in a manner that demonstrates the measurement system meets the accuracy requirements specified in Table C1. Calibrations shall include no less than four (4) points compared to a calibration standard. Calibration standards shall be traceable to NIST or equivalent laboratories that participate in inter-laboratory audits. It is recommended that standards such as ISO 17025 be used by test facilities to improve processes for the development and maintenance of instrument systems to achieve desired accuracy and precision levels.

C4.3 Full scale range for instruments and measurement systems shall be such that readings will be at least 10% of full scale at any test point (i.e. at any Percent Load). A test facility may require multiple sets of instruments to accommodate a range of Water-Chilling or Water-Heating Package sizes.

C4.4 Accuracy of electrical measurements shall include all devices in the measurement system (i.e. power meter or power analyzer, potential transformers, current transformers, data acquisition signals). Electrical measurements include voltage, current, power, and frequency for each phase. Electrical power measurements shall be made at appropriate location(s) to accurately measure the power input at the customer connection point(s) or terminals. The measurement location shall exclude losses from transformers, or other equipment comprising the power supply and shall minimize losses due to cabling from the measurement location to the connection point on the chiller. Liquid chillers that utilize power-altering equipment, such as variable frequency drive or inverter, may require appropriate isolation and precautions to ensure that accurate power measurements are obtained. Liquid chillers that utilize power-altering equipment may require the use of instrumentation that is capable of accurately measuring signals containing high frequency and/or high crest factors. In these cases, the instrumentation used shall have adequate bandwidth and/or crest factor specifications to ensure the electrical power input measurement errors are within the accuracy requirements of Table C1 for the quantity measured.

Table C1. Accuracy Requirements for Test Instrumentation

Measurement	Measurement System Accuracy [3]	Turn Down Ratio [3, 7]	Display Resolution	Selected, Installed, Operated, Maintained in Accordance With
Liquid Temperature	±0.2°F	N/A	≤0.01°F	ANSI/ASHRAE Standard 41.1-1986 (RA 2006)
Air Temperature	±0.2°F	N/A	≤0.1°F	ANSI/ASHRAE Standard 41.1-1986 (RA 2006)
Liquid Mass Flow Rate [4]	±1.0% RDG[1]	10:1	a minimum of 4 significant digits	ASME MFC-3M-2004 (orifice & venturi type) ASME MFC-6M-1998 (vortex type) ASME MFC-11-2006 (coriolis type) ASME MFC-16-2007 (electromagnetic type) ISA Standard RP31.1-1977 (turbine type)
Differential Pressure	±1.0% RDG[1]	10:1	0.1 ft H_2O	ASME Power Test Code PTC 19.2-2010
Electrical Power ≤ 600V > 600 V	Notes [1, 2, 6] ±1.0% FS, ± 2.0% RDG ±1.5% FS, ±2.5% RDG	10:1	a minimum of 4 significant digits	IEEE 120-1989
Barometric Pressure	±0.15 psia	1.5:1	0.01 psia	ASME Power Test Code PTC 19.2-2010
Steam condensate mass flow rate	±1.0% RDG[1]	10:1	a minimum of 4 significant digits	
Steam pressure	±1.0% RDG[1]	10:1	1.0 PSI	
Fuel volumetric flow rate	±1.0% RDG[1]	-	0.2 CFH[5]	
Fuel energy content	-	-	-	Gas quality shall be acquired by contacting the local authority and requesting a gas quality report for calorific value on the day of the test

Notes:

1. Percent of Reading = %RDG, %FS = percent of full scale for the measurement instrument or measurement system.
2. Current Transformers (CT's) and Potential Transformers (PT's) will have a metering class of 0.3 or better.
3. Measurement system accuracy shall apply over the range indicated by the turn down ratio, i.e. from full scale down to a value of full scale divided by the turn down ratio. For some instruments and/or systems this may require exceeding the accuracy requirement at full scale.
4. Accuracy requirement also applies to volumetric type meters.
5. CFH= Cubic Feet per Hour
6. If dual requirements are shown in the table, both requirements shall be met.
7. Turn Down Ratio = the ratio of the maximum to the minimum measurement value in the range over which the measurement system meets the specified accuracy.

C5 *Measurements.*

C5.1 *Data to be Recorded During the Test.*

C5.1.1 *Test Data.* Compressor/ Evaporator (All Condenser Types).

C5.1.1.1 Temperature of water entering evaporator, °F

C5.1.1.2 Temperature of water leaving evaporator, °F

C5.1.1.3 Chilled water flow rates, measure in lbm/hr, report in gpm

C5.1.1.4 Total electrical Power input to Water-Chilling Package, kW, excluding integrated evaporator and condenser water pump power,

or

Steam consumption of turbine, lbm/h

Steam supply pressure, psig

Steam supply temperature, ºF

Steam exhaust pressure, psig or in·Hg vacuum, or

Gas consumption of turbine or engine, therms or ft^3/h, and calorific value, Btu/ft^3,

or

Fuel consumption of diesel or gasoline, gal/h and calorific value, Btu/gal.

C5.1.1.5 Measured and corrected evaporator Water Pressure Drop (inlet to outlet), measure in psi, report in ft H_2O as per Appendix G.

C5.1.1.6 Electrical power input to controls and auxiliary equipment, kW (if not included in C5.1.1.4)

C5.1.2 *Water-Cooled Condenser.*

C5.1.2.1 Temperature of water entering the Condenser, °F

C5.1.2.2 Temperature of water leaving the Condenser, °F

C5.1.2.3 Condenser water flow rate, measure in lbm/hr, report in gpm

C5.1.2.4 Measured and corrected condenser Water Pressure Drop (inlet to outlet), measure in psi, report in ft H_2O as per Appendix G

C5.1.3 *Air-Cooled Condenser.*

C5.1.3.1 Dry-bulb temperature of air entering the Condenser, °F

C5.1.3.2 Condenser fan motor power consumption shall be included in C5.1.1.4 above, kW

C5.1.3.3 Barometric pressure, psia

C5.1.4 *Evaporatively-Cooled Condenser.*

C5.1.4.1 Wet-bulb temperature of air entering the Condenser, °F

C5.1.4.2 Condenser fan motor power consumption shall be included in 5.1.1.4 above, kW

C5.1.4.3 Condenser spray pump power consumption shall be included in 5.1.1.6 above, kW

C5.1.4.4 Barometric pressure, psia

C5.1.5 *Without Condenser.*

C5.1.5.1 Discharge temperature leaving compressor, °F

C5.1.5.2 Discharge pressure leaving compressor, psia

C5.1.5.3 Liquid Refrigerant Temperature entering the expansion device, °F

C5.1.5.4 Liquid pressure entering the expansion device, psia

C5.1.6 *Water-Cooled Heat Reclaim Condenser.*

C5.1.6.1 Temperature of heat reclaim entering condenser water, °F

C5.1.6.2 Temperature of heat reclaim leaving condenser water, °F

C5.1.6.3 Heat reclaim condenser water flow rate, measure in lbm/hr, report in gpm

C5.1.6.4 Measured and corrected heat reclaim condenser Water Pressure Drop (inlet to outlet), measure in psi, report in ft H_2O as per Appendix G

C5.1.7 If evaporator water is used to remove heat from any other source(s) within the package, the temperature and flow measurements of chilled water shall be made at points so that the flow and temperature measurement reflects the Net Refrigerating Capacity.

C5.1.8 If condenser water is used to cool the compressor motor or for some other incidental function within the package, the flow and temperature measurements of condenser water must be made at points, such that the measurement reflects the Gross Heating Capacity.

C5.1.9 *Power Measurements.*

C5.1.9.1 For motor driven compressors where the motor is supplied by the manufacturer, the compressor power input shall be measured as close as practical to the compressor motor terminals. If the Water-Chilling Package is rated with a transformer, frequency conversion device, or motor starter as furnished as part of the compressor circuit, the compressor power input shall be measured at the input terminals of the transformer, frequency converter or motor starter. If the Water-Chilling Package being tested is not equipped with the starter or frequency converter furnished for it, then a starter or frequency converter of similar type, similar losses and similar speed control shall be used for the test.

C5.1.9.2 Power consumption of auxiliaries shall be measured during normal operation of the package and included in total power consumption.

C5.1.9.3 For open-type compressors, where the motor and/or gear set is not supplied by the manufacturer, or for engine or turbine drives, the compressor shaft input shall be determined as stated in C6.4.1.3 or C6.4.1.4 .

C5.1.9.4 For Air-Cooled or Evaporatively-Cooled Condensers, the additional condenser fan and condenser spray pump power consumption shall be measured.

C5.2 *Auxiliary Data to be Recorded for General Information.*

C5.2.1 Nameplate data including make, model, size, serial number and refrigerant designation number, sufficient to completely identify the water chiller. Unit voltage and frequency shall be recorded.

C5.2.2 Compressor driver rotational speed, rpm, for open-type compressors

C5.2.3 Ambient temperature at test site, °F

C5.2.4 Actual voltage, V, and current, A, for each phase of electrical input to chiller package as well as line side measured frequency, Hz

C5.2.5 Motor, engine or turbine nameplate data

C5.2.6 Inlet pressure, inches H_2O, inlet temperature, °F and exhaust pressure, inches H_2O for steam turbine nameplate data

C5.2.7 Fuel gas specification for gas turbine drive, including pressure, inches H_2O

C5.2.8 Heat balance for C6.4

C5.2.9 Date, place, and time of test

C5.2.10 Names of test supervisor and witnessing personnel

C6 *Test Procedure.*

C6.1 *Preparation for Test.*

C6.1.1 The Water-Chilling Package, which has been completely connected in accordance with the manufacturer's instructions and is ready for normal operation, shall be provided with the necessary instrumentation. Air-cooled packages, or air-sourced packages in heating mode, will be set up as specified in Appendix E.

C6.1.2 The test shall not be started until non-condensables have been removed from the system

C6.1.3 At the manufacturer’s option, condenser and evaporator surfaces may be cleaned as provided in Section C3.2.1.

C6.2 *Operations and Limits.*

C6.2.1 Start the system and establish the testing conditions in accordance with the following tolerances and instructions.

C6.2.1.1 *Evaporator (All Condenser Types)*

C6.2.1.1.1 The entering chilled water flow rate, gpm, shall not deviate more than $\pm$5% from that specified.

C6.2.1.1.2 The individual readings of water temperature leaving the evaporator shall not vary from the specified values by more than 0.5°F. Care must be taken to insure that these water temperatures are the average bulk stream temperatures.

C6.2.1.1.3 The leaving chilled water temperature shall be adjusted by an increment calculated per C6.3 corresponding to the specified field Fouling Factor Allowance required for test.

C6.2.1.2 *Water-Cooled Condenser.*

C6.2.1.2.1 The entering water flow rate, gpm, through the Condenser shall not deviate more than ±5% from that specified.

C6.2.1.2.2 The individual readings of water temperatures entering the refrigerant Condenser shall not vary from the specified values by more than 0.5°F. Care must be taken to insure that these water temperatures are the average bulk stream temperatures.

C6.2.1.2.3 The entering condensing water temperature shall be adjusted by an increment calculated per C6.3 corresponding to the specified Fouling Factor Allowance.

C6.2.1.3 *Air-Cooled and Evaporatively-Cooled Condenser.* The setup and operating limits for the units tested shall be in accordance with Appendix E.

C6.2.1.4 *Air Source Evaporator for Hot Water Heating*. The setup and operating limits for the units tested shall be in accordance with Appendix E.

C6.2.1.5 *Chiller Without Condenser.*

C6.2.1.5.1 The saturated discharge temperature shall not vary from the values required for test by more than 0.5°F

C6.2.1.5.1 The Liquid Refrigerant Temperature shall not vary from the specified values by more than 1.0°F

C6.2.1.6 *Miscellaneous.*

C6.2.1.6.1 For electrically driven machines, voltage and frequency at the chiller terminals shall be maintained at the nameplate values within tolerances of ±10% on voltage and ±1% on frequency. For dual nameplate voltage ratings, tests shall be performed at the lower of the two voltages.

C6.2.1.6.2 For steam-turbine driven machines, steam conditions to the turbine, and Condenser pressure or vacuum, shall be maintained at nameplate values.

C6.2.1.6.3 For gas-turbine or gas-engine operating machines, gas pressure to turbine or engine, and exhaust back-pressure at the turbine or engine shall be maintained at nameplate values.

C6.2.1.6.4 In all cases, the governor, if provided, shall be adjusted to maintain rated compressor speed.

C6.3 *Method for Simulating Fouling Factor Allowance at Full Load and Part-Load Conditions.*

C6.3.1 Obtain the log mean temperature difference (LMTD) for the evaporator and/or Condenser using the following equation at the specified Fouling Factor Allowance (ff_{sp}).

$$\mathrm{LMTD} = \frac{\mathrm{R}}{\ln\left(1 + \frac{\mathrm{R}}{\mathrm{S}}\right)} \qquad \mathrm{C1}$$

Where:

R = Water temperature range
= absolute value (t_{wl} - t_{we}), °F

S = Small temperature difference
= absolute value (t_s - t_{wl}), °F

C6.3.2 Derivation of LMTD:

$$\text{LMTD} = \frac{(t_s - t_{we}) - (t_s - t_{w1})}{\ln\left[\frac{t_s - t_{we}}{t_s - t_{w1}}\right]} = \frac{(t_{w1} - t_{we})}{\ln\left[\frac{(t_s - t_{w1}) + (t_{w1} - t_{we})}{t_s - t_{w1}}\right]} \quad \text{C2}$$

The Incremental LMTD (ILMTD) is determined using the following equation:

$$\text{ILMTD} = ff_{sp}\left(\frac{q}{A}\right) \quad \text{C3}$$

C6.3.3 The water temperature needed to simulate the additional fouling, TD_a, can now be calculated:

$$TD_a = S_{sp} - S_c \quad \text{C4a}$$

$$TD_a = S_{sp} - \frac{R}{e^Z - 1} \quad \text{C4b}$$

Where:

$$Z = \frac{R}{\text{LMTD} - \text{ILMTD}} \quad \text{C5}$$

$$S_c = \frac{R}{e^Z - 1} \quad \text{C6}$$

S_{sp} = Small temperature difference as specified, °F
S_c = Small temperature difference as tested in cleaned condition, °F

The water temperature difference, TD_a, is then added to the condenser entering water temperature or subtracted from the evaporator leaving water temperature to simulate the additional Fouling Factor.

C6.3.4 *Special Consideration forMmultiple Refrigerant Circuits.*

For units that have multiple refrigeration circuits for the evaporator or condenser, a unique refrigerant saturation temperature, inlet and outlet water temperatures, and a computed heat exchange quantity may exist for each heat exchanger. In this case an adjustment temperature (TD_a) will need to be computed for each heat exchanger and then combined into a single water temperature adjustment. For series water circuits, the intermediate water temperatures may be calculated when measurement is not practical. For this purpose a weighted average for the TD_as shall be computed as follows:

$$TD_{a,weighted} = \frac{\sum(q_i \cdot TD_{a,i})}{\sum(q_i)} \quad \text{C7}$$

Where:

q_i = Heat transfer rate for each heat exchanger
$TD_{a,i}$= Computed temperature adjustment for each heat exchanger as defined in C6.3.3

For this purpose, the weighted temperature adjustment, $TD_{a,weighted}$, will be added to the condenser entering water temperature or subtracted from the evaporator leaving water temperature to simulate the additional Fouling Factor.

C6.3.5 *Example-Condenser Fouling Inside Tubes.*

Specified Fouling Factor Allowance,

ff_{sp} =0.000250 h·ft^2·°F/Btu
Condenser load, q = 2,880,000 Btu/h
Specified Condenser leaving water temp, t_{wl} = 95°F
Specified Condenser entering water temp, t_{we} = 85 °F
Inside tube surface area, A_i = 550 ft^2 (since fouling is inside tubes in this example)
Specified saturated condensing temperature, t_s = 101 °F

$S_{sp} = t_s - t_{wl} = 101 - 95 = 6$ °F

$R = t_{wl} - t_{we} = 95 - 85 = 10$ °F

$$\mathrm{LMTD} = \frac{R}{\ln(1 + R/S)} = \frac{10}{\ln(1 + 10/6)} = 10.2$$

$$\mathrm{ILMTD} = ff_{sp}\left(\frac{q}{A}\right) = 0.000250\left[\frac{2{,}880{,}000}{550}\right] = 1.31$$

$$TD_a = S_{sp} - \frac{R}{e^Z - 1}$$

Where:

$$Z = \frac{R}{\mathrm{LMTD} - \mathrm{ILMTD}} = \frac{10}{10.2 - 1.31} = 1.125$$

$$TD_a = 6.0 - \frac{10}{e^{1.125} - 1} = 6.0 - 4.8 = 1.2°\mathrm{F}$$

The entering condenser water temperature for testing is then raised 1.2°F to simulate the Fouling Factor Allowance of 0.000250 h·ft^2·°F/Btu. The entering condenser water temperature will be 85+1.2 or 86.2°F.

C6.4 *Test Verification.*

C6.4.1 *Heat Balance-Substantiating Test.*

C6.4.1.1 *Calculation of Heat Balance.* In most cases, heat losses or heat gain caused by radiation, convection, bearing friction, oil coolers, etc., are relatively small and may or may not be considered in the overall heat balance.

In general for water-cooled chillers, the total measured power to the Water-Chilling Package is assumed to equal W_{input} as in C6.4.1.2. In cases where the difference in the total power measured and the compressor work is significant, an analysis that provides a calculated value of W_{input} shall be performed and used in the heat balance equation. Typical examples are shown in C6.4.1.3 through C6.4.1.4.

Gross capacity shall be used for heat balance calculations.

Omitting the effect of the small heat losses and gains mentioned above, the general heat balance equation is as follows:

$$q'_{ev} + (W_{input}) \cdot 3412.14 = q'_{cd} + q'_{hrc} \qquad \text{C8}$$

Where:

W_{input} = compressor work input as defined in C6.4.1.2 through C6.4.1.4.

C6.4.1.2 In a hermetic package, where the motor is cooled by refrigerant, chilled water or condenser water, the motor cooling load will be included in the measured condenser load, hence

W_{input} = electrical power input to the compressor motor, kW

C6.4.1.3 In a package using an open-type compressor with prime mover and external gear drive:

$$W_{input} = q_{prime\ mover} - q_{gear} \qquad C9$$

Where:

W_{input} = Power input to the compressor shaft, kW
$q_{prime\ mover}$ = Power delivered by prime mover, kW
q_{gear} = Friction loss in the gear box, kW

The value of $q_{prime\ mover}$ shall be determined from the power input to prime mover using certified data from the prime mover manufacturer.

The value of q_{gear} shall be determined from certified gear losses provided by the gear manufacturer.

C6.4.1.4 In a package using an open-type compressor with direct drive and the prime mover not furnished by the manufacturer:

W_{input} = Power input to the compressor shaft, kW

For determination of W_{input} for turbine or engine operated machines, the turbine or engine manufacturer's certified power input/output data shall be used.

In the case of motor drive:

W_{input} = Power measured at motor terminals plus power to auxiliaries as in C.5.1.9.

C6.4.1.5 *Percent Heat Balance.* Heat balance, in %, is defined as:

$$HB = \frac{\left(q'_{ev} + (W_{input}) \cdot 3412.14\right) - \left(q'_{cd} + q'_{hrc}\right)}{q'_{cd} + q'_{hrc}} \cdot 100\% \qquad C10$$

For any test of a liquid cooled chiller to be acceptable, the heat balance (%) shall be within the allowable tolerance calculated per Section 5.6 for the applicable conditions.

C6.4.2 *Concurrent Redundant Verification Method for Air-Cooled or Evaporatively-Cooled Condensers or Air-Source Evaporators for Heating Mode.*

C6.4.2.1 *Measurement Verification*: Redundant instrument measurements shall be within the limitations below:

C6.4.2.1.1 Entering water temperature measurements shall not differ by more than 0.2°F

C6.4.2.1.2 Leaving water temperature measurements shall not differ by more than 0.2°F

C6.4.2.1.3 Flow measurements shall not differ by more than 2%

C6.4.2.1.4 Power input measurements shall not differ by more than 2%

C6.4.2.2 *Capacity Calculation Method.* For capacity calculations use the average of the entering water temperature measurements, the average of the leaving water temperature measurements and the average of the flow measurements. For efficiency calculations use the average of the power measurements.

C6.4.2.3 *Example Calculation for Capacity with Concurrent Redundant Verification.*

t_{e1} = 54.10°F, t_{e2} = 53.91°F (difference of 0.19°F is acceptable)
t_{l1} = 44.10°F, t_{l2} = 43.90°F (difference 0.20°F is acceptable)
Chilled water flow rate 1 = 101020 lbm/h, water flow rate 2 = 99080 lbm/h (difference of 1.94% is acceptable)

$t_{e,avg}$ = 54.005°F
$t_{l,avg}$ = 44.000°F
$m_{w,avg}$ = 100050 lbm/h

Properties of water are calculated per Section 7.2 as follows:

c_p = 1.0024 Btu/lbm·°F
using an average entering and leaving temperature of (54.005+44.000)/2=49.0025°F

Net Refrigeration Capacity:
q_{ev} = 100050 lbm/h · 1.0024 Btu/lbm·°F · (54.005°F – 44.000°F)
q_{ev} = 83.62 $tons_R$

C7 *Calculation of Results*

C7.1 *Capacity Equations for Heat Exchangers Using Water as the Heat Transfer Medium.*

C7.1.1 To provide increased accuracy for heat balance calculations, the energy associated with pressure loss across the heat exchanger is included in the equation for gross capacity. This formulation aligns closely with the method of calculating heat transfer capacity based on the change of enthalpy of the water flowing through the heat exchanger. For the evaporator the pressure term is added to the sensible heat change in order to include all energy transferred from the water flow to the working fluid of the refrigeration cycle. For the condenser, or heat rejection heat exchanger, this term is subtracted. The incorporation of the terms associated with pressure loss results in a more accurate representation of the energy rate balance on a control volume surrounding the water flowing through the heat exchanger. Although these pressure influences may have a near negligible effect at full-load standard rating conditions, they have an increasing effect at part-load conditions and high water flow rates where the temperature change of water through the heat exchangers is smaller.

C7.1.2 The Gross (q'_{ev}) and Net (q_{ev}) Refrigerating Capacity of the evaporator, Btu/h, shall be obtained by the following equations:

$$q'_{ev} = m_w \cdot \left[c_p \cdot (t_e - t_1) + \frac{K_3 \cdot \Delta p}{\rho}\right] \qquad \text{C11}$$

Where:

K_3 = 0.18505 = 144 in^2 / ft^2· Btu / 778.17 ft·lbf
ρ = Water density determined at the average of the entering and leaving water temperatures, lbm/ft³
Δp = Water-side pressure drop for the heat exchanger, psid

$$q_{ev} = m_w \cdot c_p \cdot (t_e - t_1) \qquad \text{(per Equation 6)}$$

Capacity, $tons_R$, to be calculated as follows: $\frac{q_{ev}}{12000}$ or $\frac{q'_{ev}}{12000}$

C7.1.3 The Gross (q'_{cd}) and Net (q_{cd}) Heating Capacity of the condenser, or heat reclaim condenser, Btu/h, shall be obtained by the following equations:

$$q'_{cd} = m_w \cdot \left[c_p \cdot (t_1 - t_e) - \frac{K_3 \cdot \Delta p}{\rho}\right] \qquad \text{C12a}$$

$$q'_{hrc} = m_w \cdot \left[c_p \cdot (t_1 - t_e) - \frac{K_3 \cdot \Delta p}{\rho}\right] \qquad \text{C12b}$$

For q_{cd}, refer to Equation 7a.

For q_{hrc}, refer to Equations 7b.

C7.1.4. *Validity of Test.* Calculate the heat balance for each of the four test points (C3.1.2). All four heat balances must be within the tolerance specified in Section 5.6. Then average the data taken from the four test points and calculate capacity and power input per Section C7 using averaged data for reporting purpose.

C8 *Symbols and Subscripts.* The symbols and subscripts used are as follows:

Symbols:

A = Total water side heat transfer surface area, ft^2 for evaporator or condenser
c_p = Specific heat of water is determined at the arithmetic average between the entering and leaving measured water temperatures, Btu/lbm°F. Use the equation specified in Section 7.
cfm = Air flow rate, ft^3/min
e = Base of natural logarithm
ff = Fouling Factor Allowance
h·ft^2·°F/Btu
HB = Heat Balance
m = Mass flow rate, lbm/h
q = Capacity in Btu/h
R = Water temperature range, °F
= Absolute value (t_{wl}- t_{we},), °F
S = Small temperature difference
= Absolute value (t_s-t_{wl},), °F
t = Temperature, °F
t_s = Saturated vapor temperature for single component or azeotrope refrigerants and for zeotropic refrigerants it is the arithmetic average of the Dew Point and Bubble Point temperatures corresponding to refrigerant pressure, °F
Δp = Water side pressure drop, psi
ρ = Water density, lbm/ft^3. Use the equation specified in Section 7.
TD = Temperature difference
W = Power

Subscripts:

a = Additional fouling
c = Clean
cd = Condenser
e = Entering
ev = Evaporator
f = Fouled or fouling
H = Heating
hrc = Heat recovery

i	=	Inside
l	=	Leaving
o	=	Outside
R	=	Cooling or refrigerating
s	=	Saturation
sp	=	Specified
w	=	Water

APPENDIX D. DERIVATION OF INTEGRATED PART-LOAD VALUE (IPLV) – INFORMATIVE

D1 *Purpose.* This appendix is intended to show the derivation of the Integrated Part-Load Value (IPLV).

D2 *Background*. Prior to the publication of ASHRAE 90.1-1988 which included an AHRI proposal for IPLV, the standard rating condition, design efficiency (full-load/design ambient), was the only widely accepted metric used to compare relative chiller efficiencies. A single chiller's design rating condition represents the performance at the simultaneous occurrence of both full-load and design ambient conditions which typically are the ASHRAE 1% weather conditions. The design efficiency contains no information representative of the chiller's operating efficiency at any off-design condition (part-load, reduced ambient).

The IPLV metric was developed to create a numerical rating of a single chiller as simulated by 4 distinct operating conditions, established by taking into account blended climate data to incorporate various load and ambient operating conditions. The intent was to create a metric of part-load/reduced ambient efficiency that, in addition to the design rating, can provide a useful means for regulatory bodies to specify minimum chiller efficiency levels and for Engineering firms to compare chillers of like technology. The IPLV value was not intended to be used to predict the annualized energy consumption of a chiller in any specific application or operating conditions.

There are many issues to consider when estimating the efficiency of chillers in actual use. Neither IPLV nor design rating metrics on their own can predict a building's energy use. Additionally, chiller efficiency is only a single component of many which contribute to the total energy consumption of a chiller plant. It is for this reason that AHRI recommends the use of building energy analysis programs, compliant with ASHRAE Standard 140, that are capable of modeling not only the building construction and weather data but also reflect how the building and chiller plant operate. In this way the building designer and operator will better understand the contributions that the chiller and other chiller plant components make to the total chiller plant energy use. Modeling software can also be a useful tool for evaluating different operating sequences for the purpose of obtaining the lowest possible energy usage of the entire chiller plant. To use these tools, a complete operating model of the chiller, over the intended load and operating conditions, should be used.

In summary, it is best to use a comprehensive analysis that reflects the actual weather data, building load characteristics, operational hours, economizer capabilities and energy drawn by auxiliaries such as pumps and cooling towers, when calculating the chiller and system efficiency. The intended use of the IPLV (NPLV) rating is to compare the performance of similar technologies, enabling a side-by-side relative comparison, and to provide a second certifiable rating point that can be referenced by energy codes. A single metric, such as design efficiency or IPLV shall not be used to quantify energy savings.

D3 *Equation and Definition of Terms*.

D3.1 The energy efficiency of a chiller is commonly expressed in one of the three following ratios.

D3.1.1 Coefficient of Performance, COP_R

D3.1.2 Energy Efficiency Ratio, EER for cooling only

D3.1.3 Total Power Input per Capacity, kW/ton_R

These three alternative ratios are related as follows:

$COP_R = 0.293\ EER$, $EER = 3.412\ COP_R$
$kW/ton_R = 12/EER$, $EER = 12/(kW/ton_R)$
$kW/ton_R = 3.516/COP_R$ $COP_R = 3.516/(kW/ton_R)$

The following equation is used when an efficiency is expressed as EER [Btu/(W·h)] or COP_R [W/W]:

$$IPLV = 0.01A + 0.42B + 0.45C + 0.12D \qquad \text{D1a}$$

Where, at operating conditions per Tables D-1 and D-3:

A = EER or COP_R at 100% capacity
B = EER or COP_R at 75% capacity
C = EER or COP_R at 50% capacity
D = EER or COP_R at 25% capacity

The following equation is used when the efficiency is expressed in Total Power Input per Capacity, kW/ton_R:

$$IPLV = \frac{1}{\frac{0.01}{A} + \frac{0.42}{B} + \frac{0.45}{C} + \frac{0.12}{D}} \quad D1b$$

Where, at operating conditions per Tables D-1 and D-3:

A = kW/ton_R at 100% capacity
B = kW/ton_R at 75% capacity
C = kW/ton_R at 50% capacity
D = kW/ton_R at 25% capacity

The IPLV or NPLV rating requires that the unit efficiency be determined at 100%, 75%, 50% and 25% at the conditions as specified in Table 3. If the unit, due to its capacity control logic cannot be operated at 75%, 50%, or 25% capacity then the unit can be operated at other load points and the 75%, 50%, or 25% capacity efficiencies should be determined by plotting the efficiency versus the % load using straight line segments to connect the actual performance points. The 75%, 50%, or 25% load efficiencies can then be determined from the curve. Extrapolation of data shall not be used. An actual chiller capacity point equal to or less than the required rating point must be used to plot the data. For example, if the minimum actual capacity is 33% then the curve can be used to determine the 50% capacity point, but not the 25% capacity point.

If a unit cannot be unloaded to the 25%, 50%, or 75% capacity point, then the unit should be run at the minimum step of unloading at the condenser entering water or air temperature based on Table D3 for the 25%, 50% or 75% capacity points as required. The efficiency shall then be determined by using the following equation:

$$EER_{CD} = \frac{EER_{Test}}{C_D} \quad D2$$

Where C_D is a degradation factor to account for cycling of the compressor for capacities less than the minimum step of capacity. C_D should be calculated using the following equation:

$$C_D = (-0.13 \cdot LF) + 1.13 \quad D3$$

The load factor LF should be calculated using the following equation:

$$LF = \frac{\%Load \cdot (\text{Full Load unit capacity})}{(\text{Part-Load unit capacity})} \quad D4$$

Where:

%Load is the standard rating point i.e. 75%, 50% and 25%.

Part-Load unit capacity is the measured or calculated unit capacity from which standard rating points are determined using the method above.

D3.2 *Equation Constants.* The constants 0.01, 0.42, 0.45 and 0.12 (refer to Equations D1a and D1b) are based on the weighted average of the most common building types, and operating hours, using average USA weather data. To reduce the number of data points, the ASHRAE based bin data was reduced to a design bin and three bin groupings as illustrated in Figure D1.

D3.3 *Equation Derivation.* The ASHRAE Temperature Bin Method was used to create four separate NPLV/IPLV formulas to represent the following building operation categories:

Group 1 - 24 hrs/day, 7 days/wk, 0°F and above
Group 2 - 24 hrs/day, 7 days/wk, 55°F and above
Group 3 - 12 hrs/day, 5 days/wk, 0°F and above
Group 4 - 12 hrs/day, 5 days/wk, 55°F and above

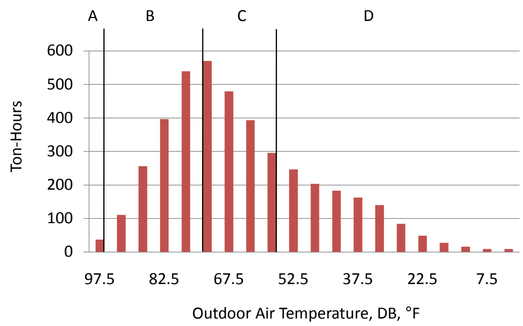

Figure D1. Ton_R-Hour Distribution Categories

The following assumptions were used:

D3.3.1 Modified ASHRAE Temperature Bin Method for energy calculations was used.

D3.3.2 Weather data was a weighted average of 29 cities across the U.S.A, specifically targeted because they represented areas where 80% of all chiller sales occurred over a 25 year period (1967-1992).

D3.3.3 Building types were a weighted average of all types (with chiller plants only) based on a DOE study of buildings in 1992 [DOE/EIA-0246(92)].

D3.3.4 Operational hours were a weighted average of various operations (with chiller plants only) taken from the DOE study of 1992 and a BOMA study (1995 BEE Report).

D3.3.5 A weighted average of buildings (with chiller plants only) with and without some form of economizer, based upon data from the DOE and BOMA reports, was included.

D3.3.6 The bulk of the load profile used in the last derivation of the equation was again used, which assumed that 38% of the buildings' load was average internal load (average of occupied vs. unoccupied internal load). It varies linearly with outdoor ambient and mean Condenser wet-bulb (MCWB) down to 50°F DB, then flattens out below that to a minimum of 20% load.

D3.3.7 Point A was predetermined to be the design point of 100% load and 85°F ECWT/95°F EDB for IPLV/NPLV. Other points were determined by distributional analysis of ton_R-hours, MCWBs and EDBs. ECWTs were based upon actual MCWBs plus an 8°F tower approach.

The individual equations that represent each operational type were then averaged in accordance with weightings obtained from the DOE and BOMA studies.

The load line was combined with the weather data hours (Figure D2) to create ton$_R$-hours (Figure D3) for the temperature bin distributions. See graphs below:

Figure D2. Bin Groupings –Ton$_R$ Hours

A more detailed derivation of the Group 1 equation is presented here to illustrate the method. Groups 2, 3, and 4 are done similarly, but not shown here. In the chart below, note that the categories are distributed as follows:

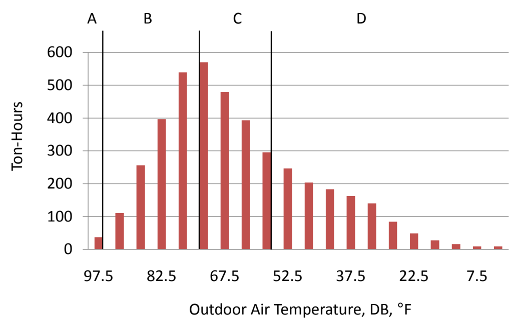

Figure D3. Group 1 Ton$_R$-Hour Distribution Categories

Point A = 1 bin for Design Bin
Point B = 4 bins for Peak Bin
Point C = 4 bins for Low Bin
Point D = all bins below 55°F for Min Bin

See Table D1 for Air Cooled and Table D2 for water-cooled calculations. The result is average weightings, ECWT's (or EDB's), and % Loads.

The next step would be to begin again with Group 2 Ton Hour distribution as below. Note Group 2 is Group 1, but with 100% Economizer at 55°F.

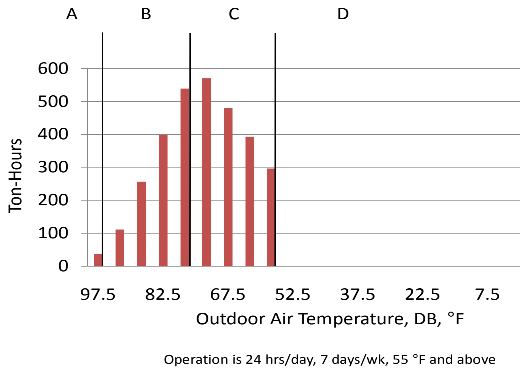

Figure D4. Group 2 Ton$_R$-Hour Distribution Categories

After creating similar tables as in Tables D1 and D2 for Groups 2, 3, and 4, the resulting Group IPLV/ NPLV equations are in Table D3.

The next step is to determine the % of each group which exists in buildings with central chiller plants, so that one final equation can be created from the four. From the DOE and BOMA studies, using goal seeking analysis, it was determined that:

Group 1 - 24.0%
Group 2 - 12.2%
Group 3 - 32.3%
Group 4 - 31.5%

This calculates to the following new equation:
IPLV equation (kW/ton$_R$):

$$\text{IPLV} = \frac{1}{\frac{0.014}{A} + \frac{0.416}{B} + \frac{0.446}{C} + \frac{0.124}{D}} \qquad \text{D5}$$

A = kW/ton$_R$ @ 100% Load and 85°F ECWT or 95°F EDB
B = kW/ton$_R$@ 76.1% Load and 75.6°F ECWT or 82.1°F EDB
C = kW/ton$_R$@ 50.9% Load and 65.6°F ECWT or 65.8°F EDB
D = kW/ton$_R$@ 32.2% Load and 47.5°F ECWT or 39.5°F EDB

Rounding off and rationalizing:

$$\text{IPLV} = \frac{1}{\frac{0.01}{A} + \frac{0.42}{B} + \frac{0.45}{C} + \frac{0.12}{D}} \qquad \text{D6}$$

A = kW/ton$_R$@ 100% Load and 85°F ECWT or 95°F EDB
B = kWton$_R$@ 75% Load and 75°F ECWT or 80°F EDB
C = kW/ton$_R$ @ 50% Load and 65°F ECWT or 65°F EDB
D = kW/ton$_R$@ 25% Load and 65°F ECWT or 55°F EDB

After rounding off and applying the rationale of where the manufacturers' and the current test facilities capabilities lie, the final Equation D1b is shown in Section D3.1.

Table D1. Group 1 Air-Cooled IPLV Data and Calculation

													C/S	Chiller
							Min Bin		Low Bin		Peak Bin		Design Bin	
Outside Temp (°F)	**Average DB (°F)**	**OA DB (°F)**	**Total Hours (h)**	**DBH (°F·h)**	**Total ton_R-h**	**Cooling Load %**	**DBH (°F·h)**	**ton_R-(h)**	**DBH (°F·h)**	**ton_R-(h)**	**DBH (°F·h)**	**ton_R-(h)**	**DBH (°F·h)**	**ton_R-(h)**
95-99	97.5	97.5	37	3608	37	100%	0	0	0	0	0	0	3608	37
90-94	92.5	92.5	120	11100	111	92%	0	0	0	0	11100	111	0	0
85-89	87.5	87.5	303	26513	256	85%	0	0	0	0	26513	256	0	0
80-84	82.5	82.5	517	42653	397	77%	0	0	0	0	42653	397	0	0
75-79	77.5	77.5	780	60450	539	69%	0	0	0	0	60450	539	0	0
70-74	72.5	72.5	929	67353	570	61%	0	0	67353	570	0	0	0	0
65-69	67.5	67.5	894	60345	479	54%	0	0	60345	479	0	0	0	0
60-64	62.5	62.5	856	53500	393	46%	0	0	53500	393	0	0	0	0
55-59	57.5	57.5	777	44678	296	38%	0	0	44678	296	0	0	0	0
50-54	52.5	52.5	678	35595	247	36%	35595	247	0	0	0	0	0	0
45-49	47.5	47.5	586	27835	204	35%	27835	204	0	0	0	0	0	0
40-44	42.5	42.5	550	23375	183	33%	23375	183	0	0	0	0	0	0
35-39	37.5	37.5	518	19425	163	32%	19425	163	0	0	0	0	0	0
30-34	32.5	32.5	467	15178	140	30%	15178	140	0	0	0	0	0	0
25-29	27.5	27.5	299	8223	84	28%	8223	84	0	0	0	0	0	0
20-24	22.5	22.5	183	4118	49	27%	4118	49	0	0	0	0	0	0
15-19	17.5	17.5	111	1943	28	25%	1943	28	0	0	0	0	0	0
10-14	12.5	12.5	68	850	16	23%	850	16	0	0	0	0	0	0
05-09	7.5	7.5	40	300	9	22%	300	9	0	0	0	0	0	0
00-04	2.5	2.5	47	118	9	20%	118	9	0	0	0	0	0	0
Total	57.9	57.9	8670	507155	4210	DBH Total	136958	1132	225628	1738	140715	1303	3608	37
						Weighting:		26.9%		41.3%		30.9%		0.9%
						EDB °F:		38.6		65.4		81.8		95.0
						Load:		31.9%		50.3%		75.7%		100%
						Points		D		C		B		A

Table D2. Group 1 Water-Cooled IPLV Data and Calculation

														C/S	Chiller
								Min Bin		Low Bin		Peak Bin		Design Bin	
Outside Temp (°F)	**Average DB (°F)**	**MC WB (sy)**	**CWH (°F·h)**	**Total Hours (h)**	**CWH (°F·h)**	**Total ton_R-(h)**	**Cooling Load %**	**CWH (°F·h)**	**ton_R-(h)**	**CWH (°F·h)**	**ton_R-(h)**	**CWH (°F·h)**	**ton_R-(h)**	**CWH (°F·h)**	**ton_R-(h)**
95-99	97.5	72	80	37	2960	37	100%	0	0	0	0	0	0	2960	37
90-94	92.5	71	79	120	9480	111	92%	0	0	0	0	9480	111	0	0
85-89	87.5	69	77	303	23331	256	85%	0	0	0	0	23331	256	0	0
80-84	82.5	68	76	517	39292	397	77%	0	0	0	0	39292	397	0	0
75-79	77.5	66	74	780	57720	539	69%	0	0	0	0	57720	539	0	0
70-74	72.5	63	71	929	65959	570	61%	0	0	65959	570	0	0	0	0
65-69	67.5	59	67	894	59898	479	54%	0	0	59898	479	0	0	0	0
60-64	62.5	55	63	856	53928	393	46%	0	0	53928	393	0	0	0	0
55-59	57.5	50	59	777	45843	296	38%	0	0	45843	296	0	0	0	0
50-54	52.5	45	55	678	37290	247	36%	37290	247	0	0	0	0	0	0
45-49	47.5	41	52	586	30472	204	35%	30472	204	0	0	0	0	0	0
40-44	42.5	37	49	550	26950	183	33%	26950	183	0	0	0	0	0	0
35-39	37.5	32	45	518	23310	163	32%	23310	163	0	0	0	0	0	0
30-34	32.5	27	41	467	19147	140	30%	19147	140	0	0	0	0	0	0
25-29	27.5	22	40	299	11960	84	28%	11960	84	0	0	0	0	0	0
20-24	22.5	17	40	183	7320	49	27%	7320	49	0	0	0	0	0	0
15-19	17.5	13	40	111	4440	28	25%	4440	28	0	0	0	0	0	0
10-14	12.5	8	40	68	2720	16	23%	2720	16	0	0	0	0	0	0
05-09	7.5	4	40	40	1600	9	22%	1600	9	0	0	0	0	0	0
00-04	2.5	1	40	47	1880	9	20%	1880	9	0	0	0	0	0	0
Total	57.9	49.3	60.0	8670	525500	4210	CWH Total	167089	1132	225628	1738	129823	1303	2960	37
							Weighting:		26.9%		41.3%		30.9%		0.9%
							ECWT °F:		47.1		65.3		81.8		85.0
							Load:		31.9%		50.3%		75.7%		100 %
							Points:		D		C		B		A

Table D3. Group 1 – 4 IPLV Summary									
Group 1	**% Load**	**ECWT, ºF**	**EDB, ºF**	**Weight**	**Group 2**	**% Load**	**ECWT, ºF**	**EDB, ºF**	**Weight**
A	100.0%	85.0	95.0	0.95%	A	100.0%	85.0	95.0	1.2%
B	75.7%	75.5	81.8	30.9%	B	75.7%	75.5	81.8	42.3%
C	50.3%	65.3	65.4	41.3%	C	50.3%	65.3	65.4	56.5%
D	31.9%	47.1	38.6	26.9%	D	N/A	N/A	N/A	0.0%
IPLV =	$\frac{1}{.009/A + .309/B + .413/C + .269/D}$				IPLV =	$\frac{1}{.012/A + .423/B + .565/C + 0.0/D}$			
Group 3	% Load	ECWT, ºF	EDB, ºF	Weight	Group 4	% Load	ECWT, ºF	EDB, ºF	Weight
A	100.0%	85.0	95.0	1.5%	A	100.0%	85.0	95.0	1.8%
B	75.7%	75.6	82.2	40.9%	B	76.4%	75.6	82.2	50.1%
C	50.3%	65.8	66.0	39.2%	C	51.3%	65.8	66.0	48.1%
D	31.9%	47.7	40.0	18.4%	D	N/A	N/A	N/A	0.0%
IPLV =	$\frac{1}{.015/A + .409/B + .392/C + .184/D}$				IPLV =	$\frac{1}{.018/A + .501/B + .481/C + 0.0/D}$			

APPENDIX E. CHILLER CONDENSER ENTERING AIR TEMPERATURE MEASUREMENT – NORMATIVE

Note: This appendix includes modifications to the test stand setup and instrumentation to be compliant with the AHRI certification program. As such, additional provisions are made for instrumentation and facility review by the auditing laboratory.

E1 *Purpose.* The purpose of this appendix is to prescribe a method for measurement of the air temperature entering the air-cooled or Evaporatively-Cooled Condenser section of an Air-Cooled Water-Chilling Package. The appendix also defines the requirements for controlling the air stratification and what is considered acceptable for a test. Measurement of the air temperatures are needed to establish that the conditions are within the allowable tolerances of this Standard. For air-cooled chillers operating in the cooling mode, only the dry-bulb temperature is required. For evaporatively-cooled and heat pump chilled water packages operating in the heating mode, both the dry-bulb and wet-bulb temperatures are required for the test.

E2 *Definitions.*

E2.1 *Air Sampling Tree*. The air sampling tree is an air sampling tube assembly that draws air through sampling tubes in a manner to provide a uniform sampling of air entering the Air-Cooled Condenser coil. See section E4 for design details

E2.2 *Aspirating Psychrometer*. A piece of equipment with a monitored airflow section that draws a uniform airflow velocity through the measurement section and has probes for measurement of air temperature and humidity. See section E5 for design details.

E3 *General Requirements.* Temperature measurements shall be made in accordance with ANSI/ASHRAE Standard 41.1. Where there are differences between this document and ANSI/ASHRAE Standard 41.1, this document shall prevail.

Temperature measurements shall be made with an instrument or instrument system, including read-out devices, meeting or exceeding the following accuracy and precision requirements detailed in Table E1:

Table E1. Temperature Measurement Requirements

Measurement	Accuracy	Display Resolution
Dry-Bulb and Wet-Bulb Temperatures[2]	≤ ±0.2°F	≤ 0.1°F
Thermopile Temperature[1]	≤ ±1.0°F	≤ 0.1°F

Notes:
1. To meet this requirement, thermocouple wire must have special limits of error and all thermocouple junctions in a thermopile must be made from the same spool of wire; thermopile junctions are wired in parallel.
2. The accuracy specified is for the temperature indicating device and does not reflect the operation of the aspirating psychrometer.

To ensure adequate air distribution, thorough mixing, and uniform air temperature, it is important that the room and test setup is properly designed and operated. The room conditioning equipment airflow should be set such that recirculation of condenser discharged air is avoided. To check for the recirculation of condenser discharged air back into the condenser coil(s) the following method shall be used: Multiple individual reading thermocouples (at least one per sampling tree location) will be installed around the unit air discharge perimeter so that they are below the plane of condenser fan exhaust and just above the top of the condenser coil(s). These thermocouples may not indicate a temperature difference greater than 5.0°F from the average inlet air. Air distribution at the test facility point of supply to the unit shall be reviewed and may require remediation prior to beginning testing. Mixing fans can be used to ensure adequate air distribution in the test room. If used, mixing fans must be oriented such that they are pointed away from the air intake so that the mixing fan exhaust direction is at an angle of 90°-270° to the air entrance to the condenser air inlet. Particular attention should be given to prevent recirculation of condenser fan exhaust air back through the unit.

A valid test shall meet the criteria for adequate air distribution and control of air temperature as shown in Table E2.

Table E2. Criteria for Air Distribution and Control of Air Temperature

Item	Purpose	Maximum Variation
		°F
Dry-Bulb Temperature		
Deviation from the mean air dry-bulb temperature to the air dry-bulb temperature at any individual temperature measurement station [1]	Uniform temperature distribution	±2.00 (≤200 $tons_R$)
		±3.00 (>200 $tons_R$)
Difference between dry-bulb temperature measured with air sampler thermopile and with aspirating psychrometer	Uniform temperature distribution	±1.50
Difference between mean dry-bulb air temperature and the specified target test value [2]	Test condition tolerance, for control of air temperature	±1.00
Mean dry-bulb air temperature variation over time (from the first to the last of the data sets)	Test operating tolerance, total observed range of variation over data collection time	±1.50
Wet-bulb Temperature [3]		
Deviation from the mean wet-bulb temperature and the individual temperature measurement stations	Uniform humidity distribution	±1.00
Difference between mean wet-bulb air wet bulb temperature and the specified target test value [2]	Test condition tolerance, for control of air temperature	±1.00
Mean wet-bulb air temperature variation over time	Test operating tolerance, total observed range of variation over data collection time (from the first to the last of the data sets)	±1.00

Notes
1. Each measurement station represents an average value as measured by a single Aspirating Psychrometer.
2. The mean dry-bulb temperature is the mean of all measurement stations.
3. The wet-bulb temperature measurement is only required for evaporatively-cooled units and heat pump chillers operating in the heating mode.

E4 *Air Sampling Tree Requirements.* The air sampling tree is intended to draw a uniform sample of the airflow entering the Air-Cooled Condenser section. A typical configuration for the sampling tree is shown in Figure E1 for a tree with overall dimensions of 4 feet by 4 feet sample. Other sizes and rectangular shapes can be used, and should be scaled accordingly as long as the aspect ratio (width to height) of no greater than 2 to 1 is maintained. It shall be constructed of stainless steel, plastic or other suitable, durable materials. It shall have a main flow trunk tube with a series of branch tubes connected to the trunk tube. It must have from 10 to 20 branch tubes. The branch tubes shall have appropriately spaced holes, sized to provide equal airflow through all the holes by increasing the hole size as you move further from the trunk tube to account for the static pressure regain effect in the branch and trunk tubes. The number of sampling holes shall be greater than 50. The average minimum velocity through the sampling tree holes shall be 2.5 ft/sec as determined by evaluating the sum of the open area of the holes as compared to the flow area in the aspirating psychrometer. The assembly shall have a tubular connection to allow a flexible tube to be connected to the sampling tree and to the aspirating psychrometer.

The sampling tree shall also be equipped with a thermocouple thermopile grid to measure the average temperature of the airflow over the sampling tree. The thermopile shall have at least 16 junction points per sampling tree, evenly spaced across the sampling tree, and connected in a parallel wiring circuit. On smaller units with only two sampling trees it is acceptable to individually measure the 16 thermocouple points as a determination of room stratification. The air sampling trees shall be placed within 6-12 inches of the unit to minimize the risk of damage to the unit while ensuring that the air sampling tubes are measuring the air going into the unit rather than the room air around the unit.

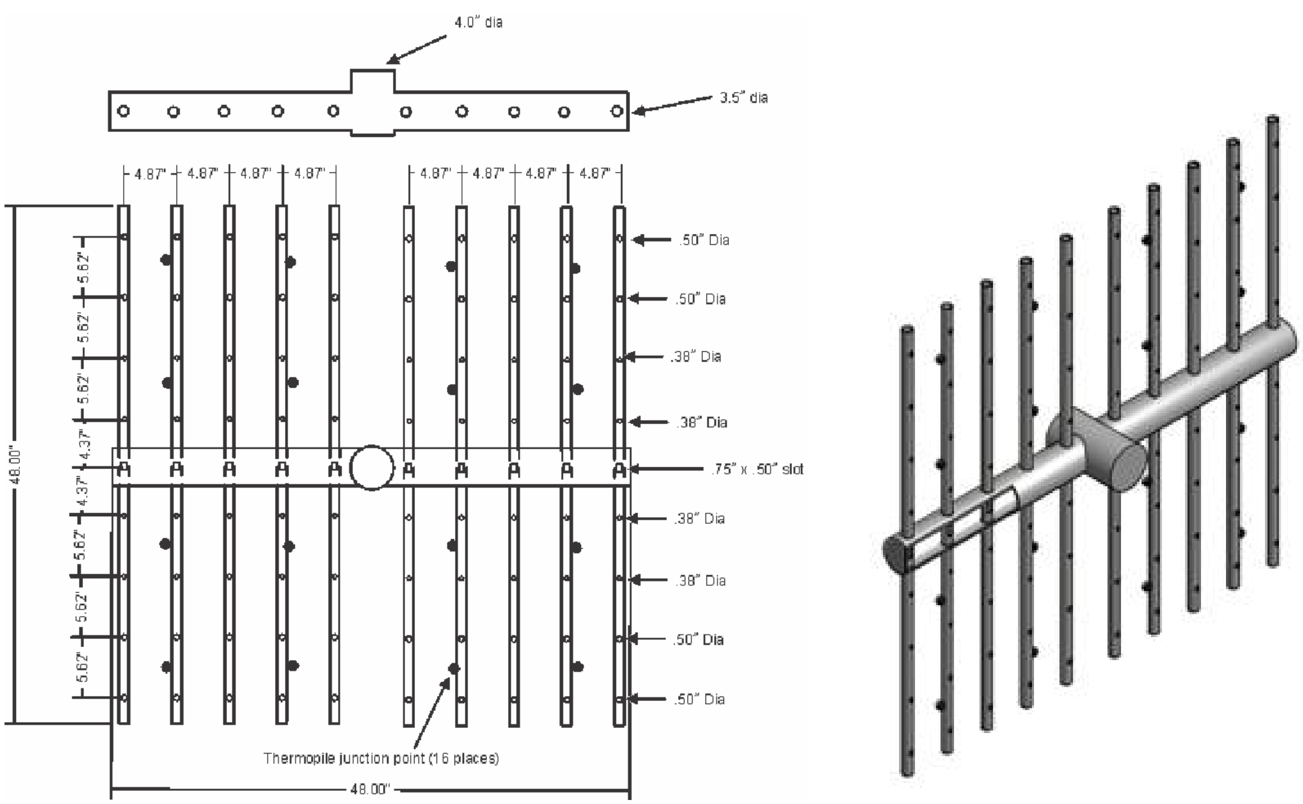

Figure E1. Typical Air Sampling Tree

Note: The .75" by .50" slots referenced in Figure E1 are cut into the branches of the sampling tree and are located inside of the trunk of the sampling tree. They are placed to allow air to be pulled into the main trunk from each of the branches.

E5 *Aspirating Psychrometer.* The aspirating psychrometer consists of a flow section and a fan to draw air through the flow section and measures an average value of the sampled air stream. The flow section shall be equipped with two dry-bulb temperature probe connections, one of which will be used for the facility temperature measurement and one of which shall be available to confirm this measurement using an additional or a third-party's temperature sensor probe. For applications where the humidity is also required, for testing of evaporatively cooled units or heat pump chillers in heating mode, the flow section shall be equipped with two wet-bulb temperature probe connection zone of which will be used for the facility wet-bulb measurement and one of which shall be available to confirm the wet-bulb measurement using an additional or a third-party's wet-bulb sensor probe. The psychrometer shall include a fan that either can be adjusted manually or automatically to maintain average velocity across the sensors. A typical configuration for the aspirating psychrometer is shown in Figure E2.

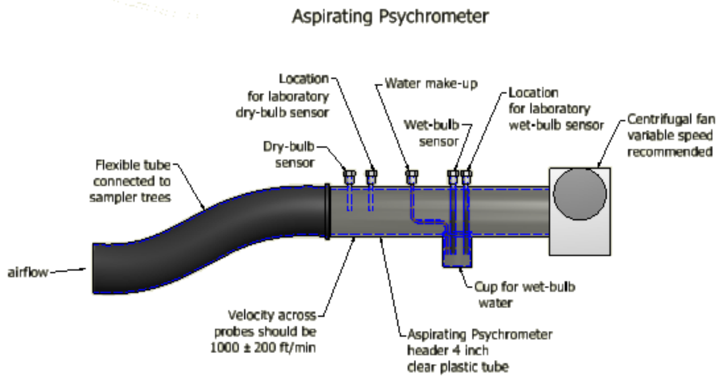

Figure E2. Aspirating Psychrometer

E6 *Test Setup Description.* Air wet-bulb and/or dry-bulb temperature shall be measured at multiple locations entering the condenser, based on the airflow nominal face area at the point of measurement. Multiple temperature measurements will be used to determine acceptable air distribution and the mean air temperature.

The use of air sampling trees as a measuring station reduces the time required to setup a test and allows an additional or third party sensor(s) for redundant dry-bulb and wet-bulb temperatures. Only the dry-bulb sensors need to be used for Air-Cooled Condensers, but wet-bulb temperature shall be used with evaporatively cooled and heat pump chillers running in the heating mode.

The nominal face area may extend beyond the condenser coil depending on coil configuration and orientation, and must include all regions through which air enters the unit. The nominal face area of the airflow shall be divided into a number of equal area sampling rectangles with aspect ratios no greater than 2 to 1. Each rectangular area shall have one air sampler tree.

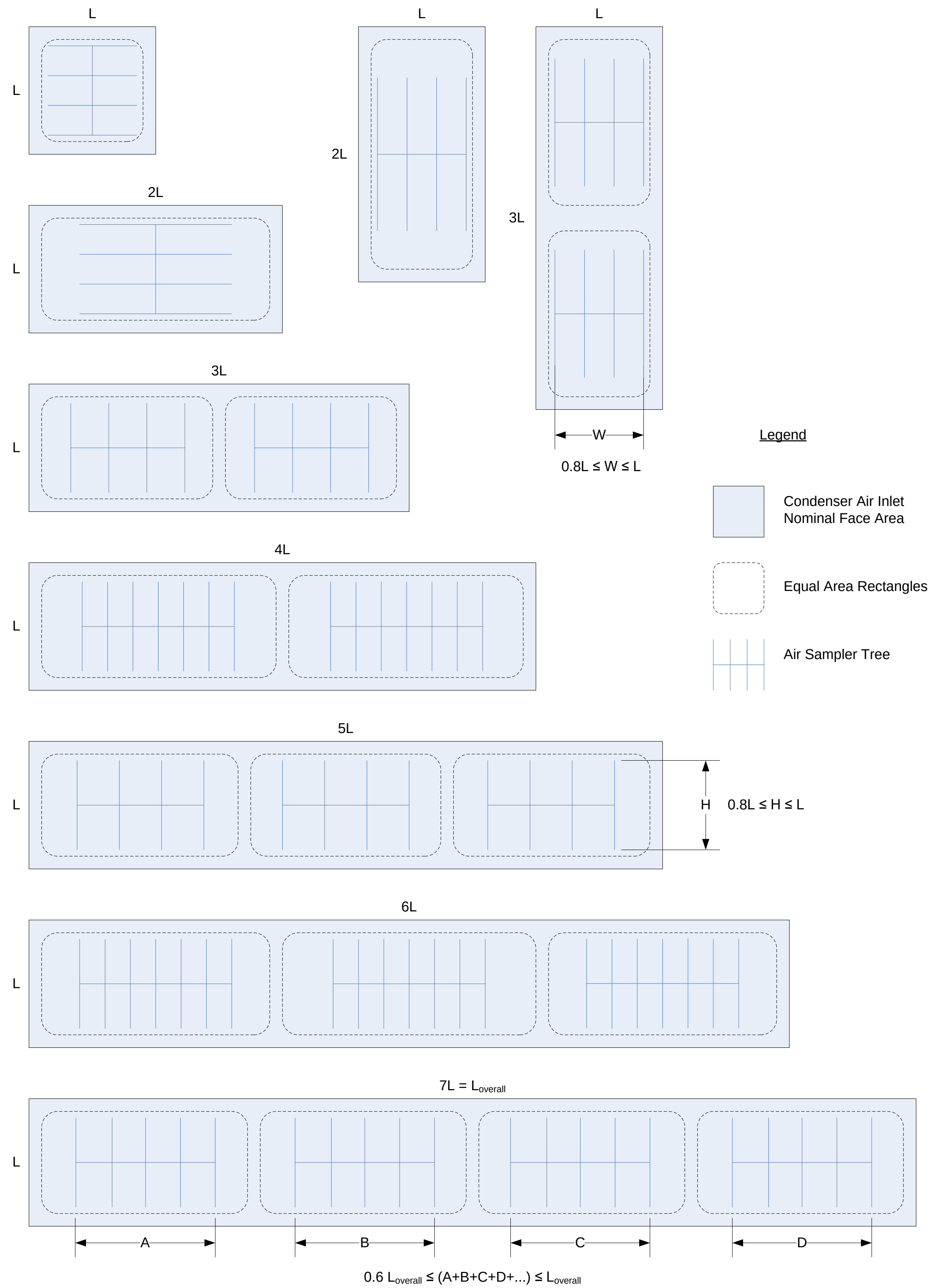

Figure E3. Determination of Measurement Rectangles and Required Number of Air Sampler Trees

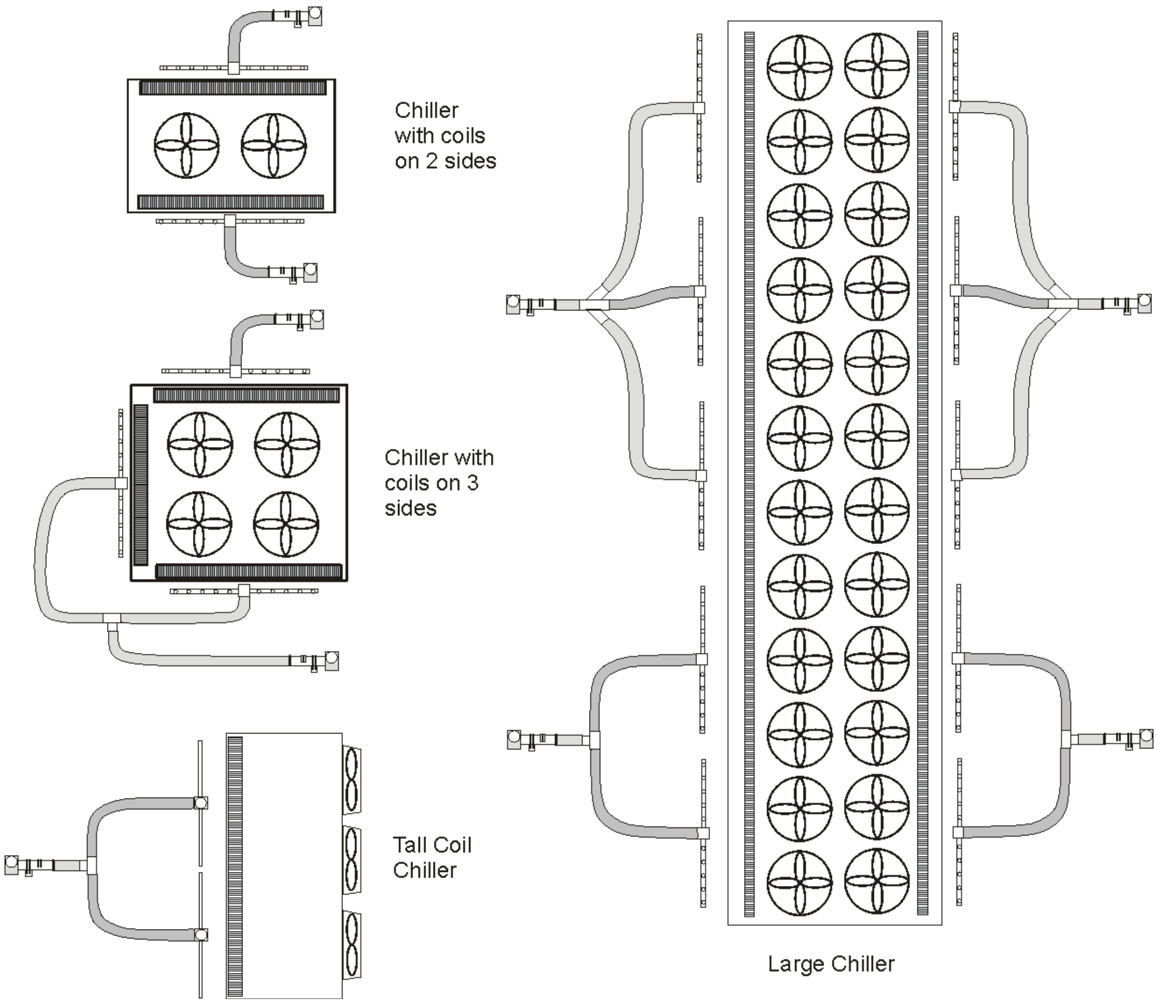

Figure E4. Typical Test Setup Configurations

A minimum of one aspirating psychrometer per side of a chiller shall be used. For units with three (3) sides, two (2) sampling aspirating psychrometers can be used but will require a separate air sampler tree for the third side. For units that have air entering the sides and the bottom of the unit, additional air sampling trees should be used.

A minimum total of two (2) air sampler trees shall be used in any case, in order to assess air temperature uniformity.

The air sampler trees shall be located at the geometric center of each rectangle; either horizontal or vertical orientation of the branches is acceptable. The sampling trees shall cover at least 80% of the height and 60% of the width of the air entrance to the unit (for long horizontal coils), or shall cover at least 80% of the width and 60% of the height of the air entrance (for tall vertical coils). The sampling trees shall not extend beyond the face of the air entrance area. It is acceptable to block all branch inlet holes that extend beyond the face of the unit. Refer to Figure E3 for examples of how an increasing number of air sampler trees are required for longer condenser coils.

A maximum of four (4) sampling trees shall be connected to each aspirating psychrometer. The sampling trees should be connected to the aspirating psychrometer using flexible tubing that is insulated and routed to prevent heat transfer to the air stream. In order to proportionately divide the flow stream for multiple sampling trees for a given aspirating psychrometer, the flexible tubing should be of equal lengths for each sampling tree. Refer to Figure E4 for some typical examples of air sampler tree and aspirating psychrometer setups.

APPENDIX F. BAROMETRIC PRESSURE ADJUSTMENT – NORMATIVE

F1 *Purpose.* The purpose of this appendix is to prescribe a method of adjusting measured test data according to the local barometric pressure.

F2 *Background.* In order to ensure that performance can be uniformly compared from one unit to another and from one manufacturer to another, performance testing for air-cooled chillers shall be corrected for air density variations. To accomplish this, use the following two (2) correction factors (CF_Q, $CF\eta$) to correct test data at 100% load points back to standard barometric pressure at sea level. These correction factors use an empirical method of correction based on industry average values across a wide variety of chillers. The correction factors are based on pressure rather than altitude, in order to include the effects of weather variations. Test data shall be corrected to Standard Air conditions for comparison to Published Ratings. The correction multiplier for efficiency and capacity at the 0% load point will be 1.0. Intermediate correction multipliers at part-load points will be a linear interpolation of values between the 0 and 100% load points where the % load value is based on the measured capacity divided by the design capacity.

> *Note:* These factors are not intended to serve as selection code correction factors. For selection codes it is best to use component models that properly adjust for variation in barometric pressure as related to fan, heat exchanger and compressor power and capacity.

The correction factors (CF_Q, $CF\eta$) will be limited to a value corresponding to a barometric pressure of 12.23 psia (approximately 5000 feet elevation). Correction factors for measured barometric readings below the minimum will be equal to the value determined at 12.23 psia.

F3 *Procedure.* Air-cooled chillers are tested at the local conditions. The data is then corrected to sea level and standard pressure by multiplying the measured data by the appropriate correction factor (CF). Both factors are in the form of a second order polynomial equations, Table F2.

$$D_Q = A_Q \cdot P^2 + B_Q \cdot P + C_Q \qquad \text{F1}$$

$$D_\eta = A_\eta \cdot P^2 + B_\eta \cdot P + C_\eta \qquad \text{F2}$$

$$CF_Q = 1 + (D_Q - 1) \cdot e^{[-0.35 \cdot (D\eta \cdot \eta_{test,\,FL} - 9.6)]} \qquad \text{F3}$$

$$CF_\eta = 1 + (D_\eta - 1) \cdot e^{[-0.35 \cdot (D\eta \cdot \eta_{test,\,FL} - 9.6)]} \qquad \text{F4}$$

$$Q_{corrected} = Q_{test} \cdot CF_Q \qquad \text{F5}$$

$$\eta_{corrected} = \eta_{test} \cdot CF_\eta \qquad \text{F6}$$

Where:

Variables are defined in Tables F1 and F2.

Table F1. Terms		
Variable or Subscript	**Description**	**Units of Measure**
P	pressure, absolute	psia
Q	Capacity	ton_R
η	Efficiency [1]	EER
$\eta_{test, FL}$	Efficiency measured in Full Load test	EER
CF_Q	capacity correction factor	-
CFη	efficiency correction factor	-
A, B, C	polynomial constants	-
Test	actual measured value during test at local conditions	-
Corrected	adjusted value equivalent to operation at sea level with standard pressure	-
1. If efficiency is expressed in kW/ton_R then divide, Equation F6, by the correction factor CFη instead of multiplying.		

Table F2. Correction Factor (CF) Coefficients						
	Capacity D_Q			**Efficiency D_η**		
Units of Measure for P	A_Q	B_Q	C_Q	A_η	B_η	C_η
IP (psia)	1.1273E-03	-4.1272E-02	1.36304E+00	2.4308E-03	-9.0075E-02	1.79872E+00
Note: E indicates scientific notation, example: 1E-02 = 0.01						

100% Load Point Example:

A chiller has published ratings of 200.0 $tons_R$ and 10.500 EER at sea level. The chiller is tested at an elevation of about 3500 feet with overcast skies.

The measured test results:

Capacity Q_{tested} = 198.50 $tons_R$
Efficiency η_{tested} = 10.350 EER
Air pressure P = 13.00 psia

Correction factor $D_Q = 0.0011273 \cdot 13.00^2 - 0.041272 \cdot 13.00 + 1.36304 = 1.0170$

Correction factor $D\eta = 0.0024308 \cdot 13.00^2 - 0.090075 \cdot 13.00 + 1.79872 = 1.0386$

Correction factor $CFq = 1+(1.0170 - 1) \cdot \exp[-0.35 \cdot (1.0386 \cdot 10.350 - 9.6)] = 1.0114$

Correction factor $CF\eta = 1+(1.0386 - 1) \cdot \exp[-0.35 \cdot (1.0386 \cdot 10.350 - 9.6)] = 1.0258$

Corrected capacity $Q_{corrected} = 198.50 \cdot 1.0114 = 200.76$ $tons_R$

Corrected efficiency $\eta_{corrected} = 10.350 \cdot 1.0258 = 10.62$ EER

Part load efficiency and capacity correction factors for the above example are determined using a linear interpolation method based on the 0 and 100% correction values:

With a part load measured capacity of 160 $tons_R$ and a 200 ton_R rating point,

$CF\eta = 1 + (160/200) \cdot (1.0258 - 1) = 1.0206$
$CF_Q = 1 + (160/200) \cdot (1.0114 - 1) = 1.0091$

APPENDIX G. WATER-SIDE PRESSURE DROP CORRECTION PROCEDURE – NORMATIVE

G1 *Purpose.* The purpose of this appendix is to prescribe a method of compensating for friction losses associated with external piping sections used to determine water-side Water Pressure Drop.

G2 *Background.* As a certified test point for the liquid to refrigerant heat exchangers, the water-side pressure drop needs to be determined by test. Since the measured pressure drop for this standard will be determined by using static pressure taps external to the unit in upstream and downstream piping, adjustment factors are allowed to compensate the reported pressure drop measurement for the external piping sections. For units with small connection sizes it is felt that straight pipe sections should be connected to the units with adequate spacing to obtain reasonable static pressure measurements. This is the preferred connection methodology. Units with larger size connections may be restricted in the upstream and downstream connection arrangement such that elbows or pipe diameter changes may be necessary. Numerous studies conclude that the determination of a calculated correction term for these external components may contain significant sources of error and therefore the use of external correction factors will be restricted as follows:

G2.1 A requirement of the test arrangement is that the static pressure taps will be in a manifolded arrangement with a minimum of 3 taps located circumferentially around the pipe at equal angle spacing.

G2.2 Correction factors will be limited to 10% of the pressure drop reading.

G2.3 Unit connections with piping that have an internal diameter of 4.5 inches and below will only allow for a frictional adjustment for a straight pipe section not to exceed 10 diameters of flow length between the unit and the static pressure measurement. The absolute roughness for the pipe will be assumed to be typical of clean steel piping.

G2.4 Units with pipe connections greater than 4.5" internal diameter, may have an additional allowance for elbow (s) and/or diameter change(s) in both the upstream and downstream unit connection. These static pressure taps will be located at least 3 diameters downstream of a flow expansion and at least 1 diameter away from either an elbow or a flow contraction. The sum of all corrections may not exceed 10% of the pressure drop reading.

G3 *Procedure.* Derivation of Correction Factors– The general form of the adjustment equations utilize the methods in the Crane Technical Paper No. 410. A friction factor is determined using the Swamee-Jain equation of

$$\mathrm{f} = \frac{0.25}{\left[\log_{10}\left(\frac{\epsilon}{3.7 \cdot \mathrm{D}} + \frac{5.74}{\mathrm{Re}^{0.9}}\right)\right]^2} \qquad \text{G1}$$

Where:

$\frac{\epsilon}{\mathrm{D}}$ is the relative roughness, with ϵ the absolute roughness assumed to be 0.00015 ft and D the internal pipe diameter (ft).
Re is the Reynolds number for the flow in the pipe.

The pressure drop (h_L) associated with a flow component or fitting may be calculated using the friction factor as detailed above or the equation may use a K factor. The forms of the equations are:

$h_L = f \cdot \frac{L}{D} \cdot \frac{V^2}{2g}$ when friction factor is used for straight pipe sections, or

$h_L = K \cdot \frac{V^2}{2g}$ when a K factor is specified for elbows and expansions/contractions

Where:

L/D is the ratio of pipe length to internal diameter
V is the average velocity calculated at the entrance to the component
g is the standard gravitational term 32.174 ft/sec^2

The K factors for the elbows utilize the equation set found in the Crane Technical Publication 410. A correction factor is computed for the following elbow arrangements as detailed in Table G1:

Table G1. K Factors for Elbow Arrangements

Description	K Factor
Smooth elbow with r/D = 1	20·f
Smooth elbow with r/D= 1.5	14·f
Smooth elbow with r/D = 2	12·f
Smooth elbow with r/D = 3	12·f
Smooth elbow with r/D = 4	14·f
Segmented with 2·45° mitres	30·f
Segmented with 3·30° mitres	24·f
Segmented with 6·15° mitres	24·f

Where:

f = Darcy friction factor described above, and r/D is the radius (r) to the centerline of the elbow divided by the internal pipe diameter (D)

The determination of the K factor for the expansion and contraction sections is a function of the inlet to outlet diameter ratio as well as the angle of expansion and contraction. For purposes of this standard, the equation has been calibrated by assigning an angle term that best represents the pressure drop results found in the ASHRAE technical report 1034-RP for these expansion and contraction fittings. The user is directed to the Crane Technical Paper for a more complete description of the equations. The angle of expansion or contraction is detailed on the accompanying chart (Figure G1) with limits placed at 45 degrees and 10 degrees.

An excel spreadsheet is available from AHRI for computation of the pressure drop adjustment factors.

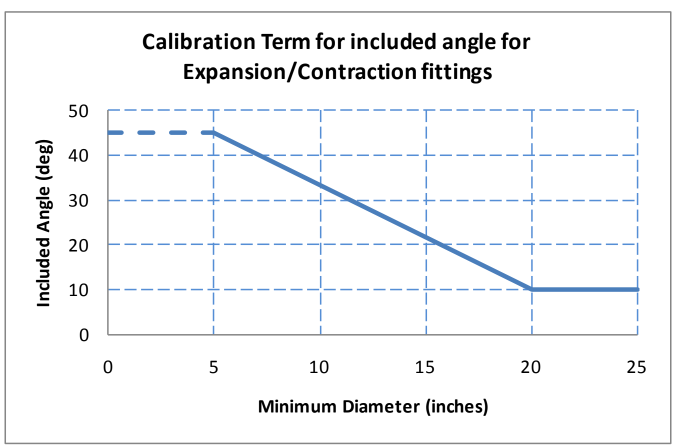

Figure G1. Calibration Term for Included Angle for Expansion/Contraction Fittings

APPENDIX H. HEATING CAPACITY TEST PROCEDURE – NORMATIVE

H1 *Purpose*. This appendix prescribes methods of testing for measurement of water-side Heating Capacity for Heat Pump Water-Heating Packages with outdoor air-side.

H1.1 *General.* Net Heating Capacity is determined from water-side measurements of temperature change and flow rate. Redundant instrumentation, rather than two separate capacity measurements methods, is used to check for erroneous measurements.

H1.1.1 During the entire test, the equipment shall operate without damage to the equipment.

H1.1.2 During the entire test, the heat rejection water flow rate shall remain constant at the cooling mode test conditions derived from Table 1 or Table 2 as shown in Section 5 of AHRI Standard 550/590 (I-P).

H1.1.3 For the duration of the test all ice or melt must be captured and removed by drain provisions.

H1.2 Heating capacity tests used to evaluate the heating performance of a heat pump when operating at conditions that are conducive to frost accumulation on the outdoor coil should be conducted using the "T" test procedure described in Section H3. Otherwise, the manufacturer shall have the option of first trying to use the "S" test procedure of Section H2. If the requirements of the "S" test procedure cannot be achieved, then the Heating Capacity test shall be conducted using the "T" test procedure described in Section H3.

H1.3 Except as noted, overriding of automatic defrost controls shall be prohibited. The controls may only be overridden when manually initiating a defrost cycle is permitted.

H1.4 For heat pumps that use a time-adaptive defrost control system, where defrost initiation depends on the length of previous defrost cycles, the defrost controls of the heat pump shall be defeated during the official data collection interval of all Heating Capacity tests. When the defrost controls are defeated, defrost cycles (if any) shall be manually induced in accordance with the manufacturer's instructions.

H1.5 Any defrost cycle, whether automatically or manually initiated, that occurs while conducting a Heating Capacity test shall always be terminated by the action of the heat pump's defrost controls.

H1.6 Defrost termination shall be defined as occurring when the controls of the heat pump actuate the first change in converting from defrost operation to normal heating operation. Whether automatically or manually initiated, defrost initiation shall be defined as occurring when the controls of the heat pump first alter its normal heating operation in order to eliminate possible accumulations of frost on the outdoor coil.

H1.7 Frosting capacity degradation ratio used in the "S" Test Procedure is defined as:

$$\text{Frosting capacity degradation ratio} = \frac{q_{cd(\tau=0)} - q_{cd(\tau)}}{q_{cd(\tau=0)}} \qquad \text{H1}$$

Where:

q_{cd} = Condenser Net Heating Capacity, ton_R
τ = Time, minutes

H2 *"S" Test Procedure.*

H2.1 The dry-bulb temperature and water vapor content of the air entering the outdoor-side shall be sampled at equal intervals of one minute throughout the preconditioning and data collection periods. Over these same periods, all other applicable Table H1 non-frosting parameters used in evaluating equilibrium shall be sampled at equal intervals of five minutes. All data collected over the respective periods, except for parameters sampled between a defrost initiation and ten minutes after the defrost termination, shall be used to evaluate compliance with the test tolerances specified in Table H1.

H2.2 The test room reconditioning apparatus and the equipment under test shall be operated until equilibrium conditions are attained, but for not less than one hour, before test data are recorded. At any time during the preconditioning period, the heat pump may undergo one or more defrost cycles if automatically initiated by its own controls. The preconditioning period may, in addition, end with a defrost cycle and this period ending defrost cycle may be either automatically or manually initiated. Ending the preconditioning period with a defrost cycle is especially recommended for Heating Capacity tests at low outdoor temperatures. If a defrost does occur, the heat pump shall operate in the heating mode for at least ten minutes after defrost termination prior to resuming or beginning the data collection described in Sections H2.1 and H2.3, respectively.

H2.3 Once the preconditioning described in Section H2.2 is completed, the data required for the specified test shall be collected. These data shall be sampled at equal intervals that span five minutes or less. The net Heating Capacity q_{cd} shall be evaluated at equal intervals of five minutes. The capacity evaluated at the start of the data collection period, $q_{cd(\tau=0)}$, shall be saved for purposes of evaluating Sections H2.4.1 or H2.5.1 compliance.

H2.4 *Test Procedures If the Pre-Conditioning Period Ends with a Defrost Cycle.*

H2.4.1 Data collection shall be suspended immediately if any of the following conditions occur prior to completing a 30-minute interval where the Table H1 non-frosting test tolerances are satisfied:

H2.4.1.1 If the heat pump undergoes a defrost;

H2.4.1.2 If the indoor-side water temperature difference degrades such that the degradation ratio exceeds 0.050 (refer to Equation H1); or

H2.4.1.3 If one or more of the applicable Table H1 non-frosting test tolerances are exceeded.

H2.4.2 If the “S” test procedure is suspended because of condition “a” of Section H2.4.1, then the “T” test procedure described in Section H3 shall be used.

H2.4.3 If the “S” test procedure is suspended because of condition “b” of H2.4.1, then the “T” test procedure described in H3 shall be used.

H2.4.4 If the “S” test procedure is suspended because of condition “c” of Section H2.4.1, then another attempt at collecting data in accordance with H2 and the “S” test procedure shall be made as soon as steady performance is attained. An automatic or manually initiated defrost cycle may occur prior to making this subsequent attempt. If defrost does occur, the heat pump shall operate in the heating mode for at least ten minutes after defrost termination prior to beginning the data collection described in Section H2.3. The preconditioning requirements in Section H2.2 are not applicable when making this subsequent attempt.

H2.4.5 If the “S” test procedure is not suspended in accordance with Section H2.4.1, then the sampling specified in Section H2.3 shall be terminated after 30 minutes of data collection. The test, for which the Table H1 test tolerances for non-frosting apply, shall be designated as a completed steady-state Heating Capacity test, and shall use the average of the seven (7) samples at the reported net Heating Capacity.

H2.5 *Test Procedure If the Pre-Conditioning Period Does Not End with a Defrost Cycle.*

H2.5.1 Data collection shall be suspended immediately if any of the following conditions occur prior to completing a 30-minute interval where the Table H1 non-frosting test tolerances are satisfied:

H2.5.1.1 If the heat pump undergoes a defrost;

H2.5.1.2 If the indoor-side water temperature difference degrades such that the degradation ratio exceeds 0.050 (refer to Equation H1); or

H2.5.1.3 If one or more of the applicable Table H1 non-frosting test tolerances are exceeded.

H2.5.2 If the "S" test procedure is suspended because of condition "a" of Section H2.5.1, then another attempt at collecting data in accordance with Sections H2.3 and H2.4 shall be made beginning ten minutes after the defrost cycle is terminated. The preconditioning requirements of Section H2.2 are not applicable when making this subsequent attempt.

H2.5.3 If the "S" test procedure is suspended because of condition "b" of Section H2.5.1, then another attempt at collecting data in accordance with Sections H2.3 and H2.4 shall be made. This subsequent attempt shall be delayed until ten minutes after the heat pump completes a defrost cycle. This defrost cycle should be manually initiated, if possible, in order to avoid the delay of having to otherwise wait for the heat pump to automatically initiate a defrost.

H2.5.4 If the "S" test procedure is suspended because of condition "c" of Section H2.5.1, then another attempt at collecting data in accordance with Section H2 and the "S" test procedure shall be made as soon as steady performance is attained. An automatic or manually initiated defrost cycle may occur prior to making this subsequent attempt. If a defrost does occur, the heat pump shall operate in the heating mode for at least ten minutes after defrost termination prior to beginning the data collection described in Section H2.3. The preconditioning requirements in Section H2.2 are not applicable when making this subsequent attempt.

H2.5.5 If the "S" test procedure is not suspended in accordance with Section H2.5.1, then the sampling specified in Section H2.3 shall be terminated after 30 minutes of data collection. The test, for which the Table H1 test tolerances for non-frosting apply, shall be designated as a completed steady-state Heating Capacity test, and shall use the average of the seven (7) samples at the reported net Heating Capacity.

H3 *"T" Test Procedure.*

H3.1 Average Heating Capacity shall be determined using the indoor water temperature method. The normal outdoor-side airflow of the equipment shall not be disturbed.

H3.2 No changes in the water flow or air flow settings of the heat pumps shall be made.

H3.3 The test tolerance given in Table H1, "heat with frost," shall be satisfied when conducting Heating Capacity tests using the "T" test procedure. As noted in Table H1, the test tolerances are specified for two sub-intervals. "Heat portion" consists of data collected during each heating interval; with the exception of the first ten minutes after defrost termination. "Defrost portion" consists of data collected during each defrost cycle plus the first ten minutes of the subsequent heating interval. In case of multiple refrigerant circuits, "Defrost portion" applies if any individual circuit is in defrost cycle. The test tolerance parameters in Table H1 shall be sampled throughout the preconditioning and data collection periods. For the purpose of evaluating compliance with the specified test tolerances, the dry-bulb temperature of the air entering the outdoor-side shall be sampled once per minute during the heat portion and once per 20 second intervals during the defrost portion. The water vapor content of the air entering the outdoor-side shall be sampled once per minute. All other Table H1 "heat with frost" parameters shall be sampled at equal intervals that span five minutes or less.

All data collected during each interval, heat portion and defrost portion, shall be used to evaluate compliance with the Table H1 "heat with frost" tolerances. Data from two or more heat portion intervals or two or more defrost portion intervals shall not be combined and then used in evaluating Table H1 "heat with frost" compliance. Compliance is based on evaluating data for each interval separately.

H3.4 The test room reconditioning apparatus and the equipment under test shall be operated until equilibrium conditions are attained, but for not less than one hour. Elapsed time associated with a failed attempt using the "S" test procedure of Section H2 may be counted in meeting the minimum requirement for one hour of operation. Prior to obtaining equilibrium and completing one hour of operation, the heat pump may undergo a defrost(s) cycle if automatically initiated by its own controls.

H3.5 Once the preconditioning described in Section H3.4 is completed, a defrost cycle shall occur before data are recorded. This defrost cycle should be manually initiated, if possible, in order to avoid the delay of having to otherwise wait for the heat pump to automatically initiate a defrost. Data collection shall begin at the termination of the defrost cycle and shall continue until one of the following criteria is met. If, at an elapsed time of three hours, the heat pump has completed at least one defrost cycle per refrigerant circuit, and a defrost cycle is not presently underway, then data collection shall be immediately terminated. If, at an elapsed time of three hours, the heat pump is conducting a defrost cycle, the cycle shall be completed before terminating the collection of data. If three complete cycles are concluded prior to three hours, data collection shall be terminated at the end of the third cycle, provided that each circuit in a multiple circuit design has had at least one defrost cycle. A complete cycle consists of a heating period and a defrost period, from defrost termination to defrost termination. For a heat pump where the first defrost cycle is initiated after three hours but before six hours have elapsed, data collection shall cease when this first defrost cycle terminates. Data collection shall cease at six hours if the heat pump does not undergo a defrost cycle within six hours.

H3.6 In order to constitute a valid test, the test tolerances in Table H1 "heat with frost" shall be satisfied during the applicable Section H3.5 test period. Because the test begins at defrost termination and may end at a defrost termination, the first defrost portion interval will only include data from the first ten-minute heating interval while the last defrost portion interval could potentially include data only from the last defrost cycle.

H3.7 The data required for the indoor water side capacity test method shall be sampled at equal intervals of five minutes, except during the following times when the water entering and leaving the indoor-side shall be sampled every ten seconds, during

H3.7.1 Defrost cycles and

H3.7.2 The first ten minutes after a defrost termination (includes the first ten minutes of the data collection interval).

H3.8 Average Heating Capacity and average input power shall be calculated in accordance with Section H3.9 using data from the total number of complete cycles that are achieved before data collection is terminated. In the event that the equipment does not undergo a defrost during the data collection interval, the entire six-hour data set shall be used for the calculations in Section H3.9.

H3.9 *Heating Calculation for "T" Test Method.* For equipment in which defrosting occurs, an average Heating Capacity and average input power corresponding to the total number of complete cycles shall be determined. If a defrost does not occur during the data collection interval, an average Heating Capacity shall be determined using data from the entire interval.

$$(q_{cd})_{avg} = \frac{1}{\tau_2-\tau_1}\int_{\tau_1}^{\tau_2} q_{cd} \cdot \delta\tau = \frac{1}{\tau_2-\tau_1}\sum_{i=1}^{n}(q_{cd})_i \cdot \Delta\tau_i \qquad \text{H2}$$

$$(W_{INPUT})_{avg} = \frac{1}{\tau_2-\tau_1}\int_{\tau_1}^{\tau_2} W_{INPUT} \cdot \delta\tau = \frac{1}{\tau_2-\tau_1}\sum_{i=1}^{n}(W_{INPUT})_i \cdot \Delta\tau_i \qquad \text{H3}$$

Where q_{cd} is calculated according to Section 5.1.4, at each data collection time interval specified by either the "S" test or the "T" test procedure, and *n* is the number of data collections.

The average efficiency is then calculated as:

$$COP_{H,avg} = \frac{(q_{cd})_{avg}}{(W_{INPUT})_{avg}} \qquad \text{H4}$$

H4 *Accuracy and Tolerances.* Redundant instrumentation shall be used according to requirements of Section C6.4.2. Instrumentation accuracy shall comply with requirements of Appendix C and Appendix E. Set up for air temperature measurements shall comply with requirements of Appendix E. Uniformity of air temperature distribution shall comply with requirements of Appendix E.

Table H1. Test Tolerances

Parameter	Test Operating Tolerance (Total Observed Range)			Test Condition Tolerance (Variation of Average from Specified Test Conditions)		
	Non-Frosting	With Frost[1]		Non-Frosting	With Frost[1]	
		Heat Portion	Defrost Portion		Heat Portion	Defrost Portion
Outdoor Mean Dry-Bulb Air Temperature, Entering (°F)	±1.50 (3.00 range)	±2.00 (4.00 range)	±5.00 (10.0 range)	±1.00	±2.00	–(Note[2])
Outdoor Mean Wet-Bulb Air Temperature, Entering (°F)	±1.00 (2.00 range)	±1.50 (3.00 range)	–	±1.00	±1.50	–(Note[2])
Indoor Condenser Leaving Water Temperature (°F)	±0.50 (1.00 range)	±0.50 (1.00 range)	–	±0.50	±0.50	–
Indoor Condenser Entering Water Temperature (°F)	–	–	±2.00 (4.00 range)	–	–	–
Water Flow Rate (% of reading)	±5.0%	±5.0%	±5.0%	±5.0%	±5.0%	±5.0%

Notes:

1. The "heat portion" shall apply when the unit is in the heating mode except for the first ten minutes after terminating a defrost cycle. The "defrost portion" shall include the defrost cycle plus the first ten minutes after terminating the defrost cycle.
2. When these data are sampled during the defrost portion of the cycle, they shall be omitted when computing average temperatures for the tests.