AHRI Standard 550/590 (I-P)

2015 Standard for

Performance Rating of Water-chilling and Heat Pump Water-heating Packages Using the Vapor Compression Cycle

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ERRATA SHEET FOR AHRI STANDARD 550/590 (I-P)-2015, *PERFORMANCE RATING OF WATER-CHILLING AND HEAT PUMP WATER-HEATING PACKAGES USING THE VAPOR COMPRESSION CYCLE*

February 2016

The corrections listed in this errata sheet apply to AHRI Standard 550/590 (I-P)-2015.

Page	Erratum
3	**Section 3.14** The section reference for significant figures must be changed from Section 4.2 to Section 4.3. This was a typo that must be corrected.

Current sentence: …starting from the first nonzero digit (Refer to Sections 4.2 and 6.2).

New sentence: … starting from the first nonzero digit (Refer to Sections 4.3 and 6.2).

Pages	Errata
14	**Section 5.4.1.3.1,**
14	**Section 5.4.1.3.2,**
14	**Section 5.4.1.3.3,**
15	**Section 5.4.1.4.1,**
15	**Section 5.4.1.4.2, and**
16	**Section 5.4.1.4.3**

The table reference for temperature limits must be changed to Table 12 instead of Table 11. This was a typo that must be corrected.

Current sentence: …be within the required temperature limits per Table 11.

New sentence: …be within the required temperature limits per Table 12.

Page	Erratum
16	**Section 5.4.1.5.1** The table reference for tolerances must be changed to Table 11 instead of Table 10. This was a typo that must be corrected.

Current sentence: …within 3% of the target and per Table 10 where tolerance of ±5% is allowed.

New sentence: …within 3% of the target and per Table 11 where tolerance of ±5% is allowed.

Page	Erratum
17	**Section 5.4.1.5.2** The table reference for tolerances must be changed to Table 11 instead of Table 10. This was a typo that must be corrected.

Current sentence: …within the allowable tolerance as defined in Table 10 so the test data can be used directly in the NPLV.IP calculations for rating point A.

New sentence: …within the allowable tolerance as defined in Table 11 so the test data can be used directly in the NPLV.IP calculations for rating point A.

Page	Erratum
20	**Section 5.4.1.5.4** The table reference for tolerances must be changed to Table 11 instead of Table 10. This was a typo that must be corrected.

Current sentence: …is with 0.6% of the rated capacity and above the -5.0% tolerance as defined in Table 10.

New sentence: …is within 0.6% of the rated capacity and above the -5.0% tolerance as defined in Table 11.

Page	Erratum
21	**Section 5.4.1.5.5** The table reference for tolerances must be changed to Table 11 instead of Table 10. This was a typo that must be corrected.

Current sentence: …than the capacity tolerance defined in Table 10.

New sentence: …than the capacity tolerance defined in Table 11.

Page	Erratum
31	**Section 6.2** The table reference for significant figures must be changed to Table 14 instead of Table 13. This was a typo that must be corrected.

Current sentence: Published Ratings shall be rounded to the number of significant figures shown in Table 13…

New sentence: Published Ratings shall be rounded to the number of significant figures shown in Table 14…

Page	Erratum
54	**Section C3.6** The table reference for significant figures must be changed to Table 14 instead of Table 13. This was a typo that must be corrected.

Current sentence: …making use of the significant figure requirements of Table 13.

New sentence: …making use of the significant figure requirements of Table 14.

Page	Erratum
55	**Table C1** In Note 7, the section reference for significant figures must be changed from Section 4.2 to Section 4.3. This was a typo that must be corrected.

Current sentence: Significant figures (also known as significant digits) determined in accordance with Section 4.2.

New sentence: Significant figures (also known as significant digits) determined in accordance with Section 4.3.

Page	Erratum
61	**Section C4.3.1** The table reference for operation condition tolerances must be changed to Table 12 instead of Table 11. This was a typo that must be corrected.

Current sentence: …to maintain the mean and standard deviation within the tolerances defined in Table 11.

New sentence: …to maintain the mean and standard deviation within the tolerances defined in Table 12.

Page	Erratum
61	**Section C4.3.2** The table reference for tolerances must be changed to Table 11 instead of Table 10. This was a typo that must be corrected.

Current sentence: The performance test results shall meet the tolerances defined in Table 10.

New sentence: The performance test results shall meet the tolerances defined in Table 11.

Page	Erratum
62	**Section C4.5.1** The table reference for test validity tolerances must be changed to Table 13 instead of Table 12. This was a typo that must be corrected.

Current sentence: Test validity tolerance for energy balance is found in Table 12.

New sentence: Test validity tolerance for energy balance is found in Table 13.

Page	Erratum
63	**Section C4.5.3** The table reference for test validity tolerances must be changed to Table 13 instead of Table 12. This was a typo that must be corrected.

Current sentence: Test validity tolerance is found in Table 12.

New sentence: Test validity tolerance is found in Table 13.

Page	Erratum
88	**Section H2.1** The table reference for operating condition tolerances must be changed to Table 12 instead of Table 11. This was a typo that must be corrected.

Current sentence: … all other applicable Table 11 non-frosting parameters used in evaluating equilibrium...

New sentence: … all other applicable Table 12 non-frosting parameters used in evaluating equilibrium...

Page	Erratum
89	**Section H2.4.1** The table reference for operating condition tolerances must be changed to Table 12 instead of Table 11. This was a typo that must be corrected.

Current sentence: … a 30-minute interval where the Table 11 non-frosting test tolerances are satisfied:

New sentence: … a 30-minute interval where the Table 12 non-frosting test tolerances are satisfied:

Page	Erratum
89	**Section H2.4.1.3** The table reference for operating condition tolerances must be changed to Table 12 instead of Table 11. This was a typo that must be corrected.

Current sentence: One or more of the applicable Table 11 non-frosting test tolerances are exceeded

New sentence: One or more of the applicable Table 12 non-frosting test tolerances are exceeded

Page	Erratum
89	**Section H2.4.5** The table reference for operating condition tolerances must be changed to Table 12 instead of Table 11. This was a typo that must be corrected.

Current sentence: The test, for which the Table 11 test tolerances for non-frosting apply...

New sentence: The test, for which the Table 12 test tolerances for non-frosting apply...

Page	Erratum
89	**Section H2.5.1** The table reference for operating condition tolerances must be changed to Table 12 instead of Table 11. This was a typo that must be corrected.

Current sentence: …a 30-minute interval where the Table 11 non-frosting test tolerances are satisfied:

New sentence: …a 30-minute interval where the Table 12 non-frosting test tolerances are satisfied:

Page	Erratum
89	**Section H2.5.1.3** The table reference for operating condition tolerances must be changed to Table 12 instead of Table 11. This was a typo that must be corrected.

Current sentence: One or more of the applicable Table 11 non-frosting test tolerances are exceeded

New sentence: One or more of the applicable Table 12 non-frosting test tolerances are exceeded

Page	Erratum
90	**Section H2.5.5** The table reference for operating condition tolerances must be changed to Table 12 instead of Table 11. This was a typo that must be corrected.

Current sentence: The test, for which the Table 11 test tolerances for non-frosting apply…

New sentence: The test, for which the Table 12 test tolerances for non-frosting apply…

Page	Erratum
90	**Section H3.3** The table reference for operating condition tolerances must be changed to Table 12 instead of Table 11. This was a typo that must be corrected.

Current sentence: The test tolerance given in Table 11, “heat with frost,”

New sentence: The test tolerance given in Table 12, “heat with frost,”

Page	Erratum
91	**Section H3.6** The table reference for operating condition tolerances must be changed to Table 12 instead of Table 11. This was a typo that must be corrected.

Current sentence: In order to constitute a valid test, the test tolerances in Table 11 “heat with frost” shall…

New sentence: In order to constitute a valid test, the test tolerances in Table 12 “heat with frost” shall…

Page	Erratum
91	**Section H4** The table reference for operating condition tolerances must be changed to Table 12 instead of Table 11. This was a typo that must be corrected.

Current sentence: Tolerances and stability requirements are defined in Table 11 in Section 5.6.2.

New sentence: Tolerances and stability requirements are defined in Table 12 in Section 5.6.2.

IMPORTANT

SAFETY DISCLAIMER

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

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

Note:

This 2015 standard supersedes AHRI Standard 550/590 (I-P)-2011 with Addendum 3, and shall be effective April 1, 2016 except for Section 4 where the effective date for the 2015 edition is January 1, 2017. For SI ratings, see AHRI Standard 551/591 (SI)-2015. Accompanying this standard is an Excel Spreadsheet for the Computation of the Water Pressure Drop Adjustment Factors and an Excel spreadsheet for Calibration (http://www.ahrinet.org/site/686/Standards/HVACR-Industry-Standards/Search-Standards).

AHRI CERTIFICATION PROGRAM PROVISIONS

The current scope of the Air-cooled Chiller (ACCL) and Water-cooled Chiller (WCCL) Certification Programs can be found on AHRI website www.ahrinet.org. The scope of the Certification Programs should not be confused with the scope of the standard, as the standard also includes ratings for products that are not covered by a certification program.

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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 Heat Pump Water-heating Packages using the vapor compression cycle: definitions; test requirements; rating requirements; minimum data requirements for Published Ratings; marking and nameplate data; conversions and calculations; nomenclature; 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 compressors. These Water-chilling and Water-heating Packages include:

2.1.1 Water-cooled, Air-cooled, or Evaporatively-cooled Condensers
2.1.2 Water-cooled heat recovery condensers
2.1.3 Air-to-water heat pumps
2.1.4 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 ASHRAE/ANSI/AHRI/ISO Standard 13256

Note: This standard includes products that may not currently be covered under an AHRI certification program.

Section 3. Definitions

All terms in this document follow the standard industry definitions in the *ASHRAE Terminology* website (https://www.ashrae.org/resources--publications/free-resources/ashrae-terminology) unless otherwise defined in this section.

Common conversion factors are defined in Section 7 and are referred to in equations by the use of K1, K2, etc.

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. Capacity is the product of the water mass flow rate and the change in water enthalpy entering and leaving the heat exchanger, measured at the point of the field connection. 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 energy balance during testing. (Refer to Equations C4 and C5).

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 energy balance during testing. (Refer to Equation C3).

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 9 and 10).

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 it is calculated using only the sensible heat transfer. (Refer to Equation 7).

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 including any refrigerant circuit components like mufflers, oil separators and discharge valves at the point of field connection. 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 Recovery Condenser.* A component or components 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. The heat rejected can be done through a single or multiple heat exchangers, including desuperheaters as defined in ANSI/AHRI Standard 470.

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 to the total input power at any given set of Rating Conditions. (Refer to Equation 1).

3.7.1.2 *Energy Efficiency Ratio (EER).* A ratio of the Net Refrigerating Capacity to the total input power at any given set of Rating Conditions. (Refer to Equation 2)

3.7.1.3 *Power Input per Capacity (kw/ton_R).* A ratio of the total input power to the Net Refrigerating Capacity at any given set of Rating Conditions. (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 to the total input power at any given set of Rating Conditions. (Refer to Equation 4).

3.7.3 *Simultaneous Cooling and Heating Energy Efficiency*.

3.7.3.1 *Heat Recovery Coefficient of Performance (COP_{HR}).* A ratio of the Net Heat Recovery Capacity plus the Net Refrigerating Capacity to the total input power at any given set of Rating Conditions. 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).

3.7.3.2 *Simultaneous Heating and Cooling Coefficient of Performance (COP_{SHC}).* A ratio of the Net Heating Capacity plus the Net Refrigerating Capacity to the total input power at any given set of Rating Conditions. COP_{SHC} applies to units that are operating in a manner that uses both the net heating and refrigerating capacities generated during operation. COP_{SHC} takes into account the beneficial capacity, Q_{ev}, as well as the heating capacity, Q_{cd}, (Refer to Equation 6).

3.8 *Fouling Factor (R_{foul}).* The thermal resistance due to fouling accumulated on the water side or air side heat transfer surface.

3.8.1 *Fouling Factor Allowance ($R_{foul,sp}$).* A specified value for published ratings as a provision for anticipated thermal resistance due to water side or air side fouling during use, expressed in $h \cdot ft^2 \cdot °F/Btu$.

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.IP and NPLV.IP.)

3.10.1 *Integrated Part-Load Value (IPLV.IP)*. 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.IP)*. A single number part-load efficiency figure of merit calculated per the method described in this standard referenced to conditions other than IPLV.IP conditions. (i.e. For units with Water-Cooled Condensers that are not designed to operate at Standard Rating Conditions but is not used for Air-cooled and evaporatively-cooled chillers.)

3.11 *Percent Load (%Load).* The ratio of the part-load net capacity to the full-load rated net capacity at the full-load rating conditions, stated in decimal format. (e.g.100% = 1.0). The full load rating conditions, corresponding to 100% Load, shall be at Standard Rating Conditions. Exception: Configurable Units shall use Application Rating Conditions.

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 *Significant Figure*. Each of the digits of a number that are used to express it to the required degree of accuracy, starting from the first nonzero digit (Refer to Sections 4.3 and 6.2).

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

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

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

3.16 *Total Input Power (W_{input}).* Combined power input of all components of the unit, including Auxiliary Power and excluding integral pumps.

3.17 *Turn Down Ratio*. The ratio of the maximum to the minimum instrument measurement value in the range over which the measurement system meets the specified accuracy. Applicable only to measurements using a scale with an absolute zero value (negative values not allowed).

3.18 *Unit Type*.

3.18.1 *Configurable Unit***.** A chiller that has been selected to run at a full load point less than its maximum possible capacity

3.18.2 *Packaged Unit***.** A chiller that has been selected to run at full load at its maximum capacity.

3.19 *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. 19.1 *Heat Recovery Water-chilling Package*. A factory-made package, designed for the purpose of chilling water and containing a condenser for recovering 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. 19.2 *Heat Pump Water-heating 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. 19.3 *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.

3.19.4 *Condenserless Chiller.* A factory-made package designed for the purpose of chilling water but is not supplied with a condenser. A separate air, water or evaporatively cooled condenser will be supplied to interface with the condenserless chiller.

3.20 *Water Pressure Drop*. The reduction in static water pressure associated with the flow through a water-type heat exchanger.

Section 4. Test Requirements

4.1 *Test Requirements*. Ratings shall be established at the Rating Conditions specified in Section 5. All ratings shall be based on tests conducted in accordance with the test method and procedures described in Appendix C, effective January 1, 2017. Prior to this date, refer to AHRI Standard 550/590 (I-P)-2011 with Addendum 3.

4.2 Tests shall report measurement values and calculated results in accordance with methods and procedures described in Appendix C.

4.3 Calculations shall use measurement values without rounding as defined below. Reported values on test reports shall round values of measurements and calculated results to a number of significant figures per Table 14.

4.3.1 Numerical data are often obtained (or at least calculations can be made) with more digits than are justified by their accuracy or precision. In order not to be misleading, such data shall be rounded to the number of figures consistent with the confidence that can be placed in them when reported in final form. However, more digits shall be retained at intermediate stages of calculation to avoid compounding of rounding errors; retain no less than two additional significant figures than the final reported value, or as many digits as possible. The number of significant figures is the number of digits remaining when the data are rounded.

4.3.2 The rules for identifying significant figures when writing or interpreting numbers are as follows:

4.3.2.1 All non-zero digits are considered significant. For example, 91 has two significant figures (9 and 1), while 123.45 has five significant figures (1, 2, 3, 4 and 5).

4.3.2.2 Zeroes appearing anywhere between two non-zero digits are significant. Example: 101.1203 has seven significant figures: 1, 0, 1, 1, 2, 0 and 3.

4.3.2.3 Leading zeroes are not significant. For example, 0.00052 has two significant figures: 5 and 2.

4.3.2.4 Trailing zeroes in a number containing a decimal point are significant. For example, 12.2300 has six significant figures: 1, 2, 2, 3, 0 and 0. The number 0.000122300 still has only six significant figures (the zeros before the 1 are not significant). In addition, 120.00 has five significant figures since it has three trailing zeros. This convention clarifies the precision of such numbers; for example, if a measurement precise to four decimal places (0.0001) is given as 12.23 then it might be misunderstood that only two decimal places of precision are available. Stating the result as 12.2300 makes clear that it is precise to four decimal places (in this case, six significant figures).

4.3.2.5 The significance of trailing zeroes in a number not containing a decimal point can be ambiguous. For example, it may not always be clear if a number like 1300 is precise to the nearest unit (and just happens coincidentally to be an exact multiple of a hundred) or if it is only shown to the nearest hundred due to rounding or uncertainty. Various conventions exist to address this issue:

4.3.2.5.1 A bar may be placed over the last significant figure; any trailing zeros following this are insignificant. For example, $13\overline{0}0$ has three significant figures (and hence indicates that the number is precise to the nearest ten).

4.3.2.5.2 The last significant figure of a number may be underlined; for example, "$2\underline{0}00$" has two significant figures.

4.3.2.5.3 A decimal point may be placed after the number; for example "100." indicates specifically that three significant figures are meant.

4.3.2.5.4 In the combination of a number and a unit of measurement, the ambiguity can be avoided by choosing a suitable unit prefix. For example, the number of significant figures in a power measurement specified as 1300 W is ambiguous, while a power of 1.30 kW is not.

4.3.2.5.5 Scientific notation or exponential notation may be used; for example 1.30×10^3 W.

4.3.2.6 In multiplication and division, the operation with the least number of significant figures determines the numbers to be reported in the result. For example, the product 1256 x 12.2 = 15323.2 is reported as $15\bar{3}00$. In addition and subtraction, the least number of figures to either the right or the left of the decimal point determines the number of significant figures to be reported. Thus, the sum of 120.05 + 10.1 + 56.323 = 156.473 is reported as 156.5 because 10.1 defines the reporting level. In complex calculations involving multiplications and additions, for example, the operation is done serially, and the final result is rounded according to the least number of significant figures involved. Thus: (1256 x 12.2) + 125 = 15323.2 + 125 =$15\bar{4}00$.

4.3.3 The following rules shall be used in rounding values:

4.3.3.1 When the digit next beyond the one to be retained is less than five, the retained figure is kept unchanged. For example: 2.541 becomes 2.5 to two significant figures.

4.3.3.2 When the digit next beyond the one to be retained is greater than or equal to five, the retained figure is increased by one. For example; 2.453 becomes 2.5 to two significant figures.

4.3.3.3 When two or more figures are to the right of the last figure to be retained, they are to be considered as a group in rounding decisions. Thus in 2.4(501), the group (501) is considered to be >5 while for 2.5(499), (499) is considered to be <5.

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 atmospheric pressure as shown in Appendix F or by a part load degradation factor as detailed in Section 5.4.1.2.

5.1.1.1 The Cooling Coefficient of Performance (COP_R), kW/kW, shall be calculated as follows:

$$COP_R = \frac{Q_{ev}}{K3 \cdot W_{input}} \tag{1}$$

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

$$EER = \frac{Q_{ev}}{K7 \cdot W_{input}} \tag{2}$$

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

$$kW/ton_R = \frac{K5 \cdot W_{input}}{Q_{ev}} \tag{3}$$

5.1.2 *Heating Energy Efficiency*.

5.1.2.1 The Heating Coefficient of Performance (COP_H), kW/kW, shall be calculated as follows:

$$COP_H = \frac{Q_{cd}}{K3 \cdot W_{input}} \tag{4}$$

5.1.2.2 The Heat Recovery Coefficient of Performance (COP_{HR}), kW/kW shall be calculated as follows:

$$COP_{HR} = \frac{Q_{ev} + Q_{hrc}}{K3 \cdot W_{input}} \tag{5}$$

5.1.2.3 The Simultaneous Heating and Cooling Coefficient of Performance (COP_{SHC}), kW/kW, shall be calculated as follows:

$$COP_{SHC} = \frac{Q_{cd} + Q_{ev}}{K3 \cdot W_{input}} \qquad 6$$

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

$$Q_{ev} = m_w \cdot c_p \cdot (T_{in} - T_{out}) \qquad 7$$

Specific heat c_p is taken at the average of entering and leaving water temperatures. When expressing water flow rate in volumetric terms for ratings, the conversion from mass flow rate shall use water density corresponding to entering water temperature (Refer to Equation 27). The volumetric flow rate shall be calculated as follows:

$$V_w = \frac{m_w}{\rho_{in}} \cdot K10 \qquad 8$$

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:

$$Q_{cd} = m_w \cdot c_p \cdot (T_{out} - T_{in}) \qquad 9$$

$$Q_{hrc} = m_w \cdot c_p \cdot (T_{out} - T_{in}) \qquad 10$$

Specific heat c_p is taken at the average of entering and leaving water temperatures. When expressing water flow rate in volumetric terms for ratings, the conversion from mass flow rate shall use water density corresponding to entering water temperature (Refer to Equations 8 and 27).

5.1.5 *Water Pressure Drop*. For this standard, the Water Pressure Drop shall include pressure losses due to nozzles, piping, or other interconnections included with the Water-chilling or Water-heating Package and shall include all pressure losses across the external unit connection points for water inlet and water outlet. For Published Ratings, this value is expressed in feet H_2O at a reference water temperature of 60°F. For test measurements, this is a differential pressure expressed in psid. (Refer to Section 7 for converting units of measure). For the calculation of Water Pressure Drop, Refer to Equation C18 and Appendix G.

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 recovery, 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 (Water Conditions)[3]			Heat/Recovery (Water Conditions)[4]		Evaporatively-cooled Entering Temperature[5, 8]		Air-cooled (AC) Entering Temperature[6, 8]		Without Condenser			
														Air-cooled Refrigerant Temp.		Water & Evaporatively Cooled Refrigerant Temp.	
		Entering Temp. ºF	Leaving Temp. ºF	Flow Rate gpm/ton_R	Entering Temp. ºF	Leaving Temp. ºF	Flow Rate gpm/ton_R	Entering Temp. ºF	Leaving Temp. ºF	Dry-Bulb ºF	Wet-Bulb ºF	Dry-Bulb ºF	Wet-Bulb ºF	SDT[11] ºF	LIQ[12] ºF	SDT[11] ºF	LIQ[12] ºF
All Cooling	Std	54.00	44.00	2.4[9]	85.00	94.30	Note - 10	--	--	95.00	75.00	95.00	--	125.00	105.00	105.00	98.00
AC Heat Pump High Heating[7]	Low	--	105.00	Note - 1	--	--	--	--	--	--	--	47.00	43.00	--	--	--	--
	Medium	--	120.00	Note - 1	--	--	--	--	--	--	--	47.00	43.00	--	--	--	--
	High	--	140.00	Note - 1	--	--	--	--	--	--	--	47.00	43.00	--	--	--	--
AC Heat Pump Low Heating[7]	Low	--	105.00	Note - 1	--	--	--	--	--	--	--	17.00	15.00	--	--	--	--
	Medium	--	120.00	Note - 1	--	--	--	--	--	--	--	17.00	15.00	--	--	--	--
	High	--	140.00	Note - 1	--	--	--	--	--	--	--	17.00	15.00	--	--	--	--
Water Cooled Heating	Low	54.00	44.00	2.4[9]	--	--	--	95.00	105.00	--	--	--	--	--	--	--	--
	Medium	54.00	44.00	2.4[9]	--	--	--	105.00	120.00	--	--	--	--	--	--	--	--
	High	54.00	44.00	2.4[9]	--	--	--	120.00	140.00	--	--	--	--	--	--	--	--
	Boost	75.00	65.00	2.4[9]	--	--	--	120.00	140.00	--	--	--	--	--	--	--	--
Heat Recovery	Low	54.00	44.00	2.4[9]	75.00	--	Note - 10	--	--	--	--	--	--	--	--	--	--
	Medium	54.00	44.00	2.4[9]	75.00	--	Note - 10	105.00	120.00	40.00	38.00	40.00	38.00	--	--	--	--
	Hot Water 1	54.00	44.00	2.4[9]				90.00	140.00	--	--	--	--				
	Hot Water 2	54.00	44.00	2.4[9]	--	--	--	120.00	140.00	--	--	--	--	--	--	--	--

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 allowance for the cooling mode evaporator or the heating condenser for AC reversible cycles shall be $R_{foul} = 0.000100\ h \cdot ft^2 \cdot {}^\circ F/Btu$.
3. The rating fouling factor allowance for tower heat exchangers shall be $R_{foul} = 0.000250\ h \cdot ft^2 \cdot {}^\circ F/Btu$.
4. The rating fouling factor allowance for heating and heat recovery heat exchangers shall be $R_{foul} = 0.000100\ h \cdot ft^2 \cdot {}^\circ F/Btu$ for closed loop and $R_{foul} = 0.000250\ h \cdot ft^2 \cdot {}^\circ F/Btu$ for open loop systems.
5. Evaporatively cooled condensers shall be rated with a fouling factor allowance of zero, R_{foul} $0.000\ h \cdot ft^2 \cdot {}^\circ F/Btu$.
6. Air-Cooled Condensers shall be rated with a fouling factor allowance of zero, $R_{foul} = 0.000\ h \cdot ft^2 \cdot {}^\circ F/Btu$.
7. A reversible cycle is assumed where the cooling mode evaporator becomes the condenser circuit in the heating mode.
8. Air-cooled & evaporatively-cooled unit ratings are at standard atmospheric condition (sea level). Measured test data will be corrected to an atmospheric pressure of 14.696 psia per Appendix F.
9. Rated water flow is determined by the water temperatures at the rated capacity. The normalized flow rate shown, per unit of evaporator capacity, is for reference only at Standard Rating Conditions.
10. Rated water flow is determined by the water temperatures at the rated capacity and rated efficiency.
11. Saturated Discharge Temperature (SDT).
12. Liquid Refrigerant Temperature (LIQ).

5.3 *Application Rating Conditions.* Full and part-load Application Ratings shall 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 (0.00$\overline{0}$) 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. Full and Part-load Application Rating Conditions</th></tr>
<tr><td></td><td colspan="3">Evaporator</td><td colspan="3">Condenser</td></tr>
<tr><td rowspan="9">Cooling</td><td colspan="3">Water Cooled</td><td colspan="3">Water Cooled</td></tr>
<tr><td>Leaving Temperature[1], °F</td><td>Temperature Difference Across Heat Exchanger[5], °F</td><td>Fouling Factor Allowance, h·ft^2·°F/Btu</td><td>Entering Temperature[2], °F</td><td>Flow Rate, gpm/ton$_R$ [5, 7]</td><td>Fouling Factor Allowance, h·ft^2·°F/Btu</td></tr>
<tr><td rowspan="7">36.00 to 60.00</td><td rowspan="7">5.00 to 20.00</td><td rowspan="7">0.000 to 0.001000</td><td>55.00 to 105.00</td><td>1.0 to 6.0</td><td>0.00$\overline{0}$ to 0.001000</td></tr>
<tr><td colspan="3">Air-Cooled</td></tr>
<tr><td colspan="3">Entering Air Dry Bulb[3, 6], °F</td></tr>
<tr><td colspan="3">55.00 to 125.00</td></tr>
<tr><td colspan="3">Evaporatively Cooled</td></tr>
<tr><td colspan="3">Entering Air Wet Bulb[4, 6], °F</td></tr>
<tr><td colspan="3">50.00 to 80.00</td></tr>
<tr><td rowspan="6">Heating</td><td colspan="3">Water Source Evaporator</td><td colspan="3">Water Cooled Condenser</td></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[5], °F</td><td>Fouling Factor Allowance, h·ft^2·°F/Btu</td></tr>
<tr><td colspan="2">40.00 to 80.00</td><td>0.000 to 0.001000</td><td rowspan="4">105.00 to 160.00</td><td rowspan="4">5.00 to 20.00</td><td rowspan="4">0.00$\overline{0}$ to 0.001000</td></tr>
<tr><td colspan="3">Air Source Evaporator</td></tr>
<tr><td colspan="3">Entering Air Temperature[6], °F</td></tr>
<tr><td colspan="3">15.00 to 60.00</td></tr>
</table>

Notes:

1. Evaporator water temperatures shall be published in rating increments of no more than 4.00°F.
2. Condenser water temperatures shall be published in rating increments of no more than 5.00°F.
3. Entering air temperatures shall be published in rating increments of no more than 10.00°F.
4. Air wet bulb temperatures shall be published in rating increments of no more than 2.50°F.
5. Applies to design point only, not part-load points.
6. Atmospheric pressure in the range of 11.56 to 15.20 psia. Measured test data will be corrected per Appendix F to the application rating atmospheric pressure.
7. The normalized flow rate is per unit of evaporator capacity.

5.3.1 For the purpose of this standard, published application ratings shall use a standardized relationship between rated geometric altitude (Z_H) above mean sea level and atmospheric pressure (p_{atm}). The intent is to allow chiller Application Ratings to be published based on the altitude at the installation location without consideration of local weather variations on atmospheric pressure. Test data however must be corrected on the basis of atmospheric pressure at the time of the test. See Section 7 and Appendix F.

5.4 *Part-load Ratings.* 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 recovery), part-load ratings are only required for cooling mode operation.

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

a. IPLV.IP. Based on the conditions defined in Table 3 and method defined in Section 5.4.1.
b. NPLV.IP. Water-cooled condenser only. Based on the conditions defined in Table 3 and method defined in Section 5.4.1.
c. Individual part-load data point(s) suitable for calculating IPLV.IP or NPLV.IP as defined in Table 3.
d. Other part-load points, within the application rating limits of Table 2 and method defined in Section 5.4.2, 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.IP nor NPLV.IP shall be calculated for such points.

Note*:* Optionally, Heat Pump Water-heating Packages and Heat Recovery Water-chilling Packages may be rated at individual part load points. Neither IPLV.IP nor NPLV.IP shall be calculated for such points.

5.4.1 *Determination of Part-load Performance*. For Water-chilling Packages covered by this standard, the IPLV.IP or NPLV.IP 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.IP or NPLV.IP for units rated with COP_R and EER.

$$\text{IPLV.IP or NPLV.IP} = 0.01 \cdot A + 0.42 \cdot B + 0.45 \cdot C + 0.12 \cdot D \qquad 11$$

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.IP or NPLV.IP for units rated with kW/ton_R:

$$\text{IPLV.IP or NPLV.IP} = \frac{1}{\frac{0.01}{A}+\frac{0.42}{B}+\frac{0.45}{C}+\frac{0.12}{D}} \qquad 12$$

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 11 and 12, and an example of an IPLV.IP or NPLV.IP 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.IP[5]	NPLV.IP
Evaporator (All Types)		
All loads LWT, ºF [2]	44.00	Selected LWT
Flow Rate, gpm/ton_R [3]	Per Table 1	Per Table 1, Note 10[3]
R_{foul}, h·ft²·ºF/Btu	0.000100	As Specified
Water-cooled Condenser [1,2]		
100% load EWT, ºF	85.00	Selected EWT
75% load EWT, ºF	75.00	Note[4]
50% load EWT, ºF	65.00	Note[4]
25% load EWT, ºF	65.00	Note[4]
Flow rate, gpm/ton_R [3]	Note[3]	Selected flow rate
R_{foul}, h·ft²·ºF/Btu	0.000250	As Specified
Air-cooled Condenser[1, 6]		
100% load EDB, ºF	95.0	No Rating Requirements (NPLV.IP not applicable)
75% load EDB, ºF	80.0	
50% load EDB, ºF	65.0	
25% load EDB, ºF	55.0	
R_{foul}, h·ft²·ºF/Btu	0.000	
Evaporatively-cooled Condenser[1, 6]		
100% load EWB, ºF	75.00	No Rating Requirements (NPLV.IP not applicable)
75% load EWB, ºF	68.75	
50% load EWB, ºF	62.50	
25% load EWB, ºF	56.25	
R_{foul}, h·ft²·ºF/Btu	0.000	
Air-cooled Without Condenser		
100% load SDT, ºF	125.00	No Rating Requirements (NPLV.IP not applicable)
75% load SDT, ºF	107.50	
50% load SDT, ºF	90.00	
25% load SDT, ºF	72.50	
R_{foul}, h·ft²·ºF/Btu	0.000	
Water-cooled or Evaporatively-cooled Without Condenser		
100% load SDT, ºF	105.00	No Rating Requirements (NPLV.IP not applicable)
75% load SDT, ºF	95.00	
50% load SDT, ºF	85.00	
25% load SDT, ºF	75.00	
R_{foul}, h·ft²·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. If head pressure control is active below the rating temperature then tests should be run at a temperature above which head pressure control is activated.
2. Correct for Fouling Factor Allowance by using the calculation method described in Section C3.3.4.
3. The flow rates are to be held constant at full-load values for all part-load conditions as per Table 1.
4. For part-load entering condenser water temperatures, the temperature should vary linearly from the selected EWT at 100% load to 65.00ºF at 50% loads, and fixed at 65.00°F for 50% to 0% loads.
5. Reference Equations 13 through 17 for calculation of temperatures at any point that is not listed.
   - 5.1 - Entering air dry-bulb temperature (EDB).
   - 5.2 - Entering water temperature (EWT).
   - 5.3 - Entering air wet-bulb temperature (EWB).
   - 5.4 - Compressor Saturated discharge temperature (SDT for air-cooled).
   - 5.5 - Compressor Saturated discharge temperature (SDT for water-cooled or Evaporatively-cooled).
6. Air-cooled and evaporatively-cooled unit ratings are at standard atmospheric condition (sea level). Measured data shall be corrected to standard atmospheric pressure of 14.696 psia per Appendix F.

5.4.1.2 The IPLV.IP or NPLV.IP 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. The capacity points as close as possible to the rating load shall be used. 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.IP shall be based on the measured part-load percentage for the actual test point using the Equations 13 through 17. For example for an air-cooled chiller test point run at 83% capacity, the entering air temperature for the test shall be 84.80 °F (60·0.83 + 35).

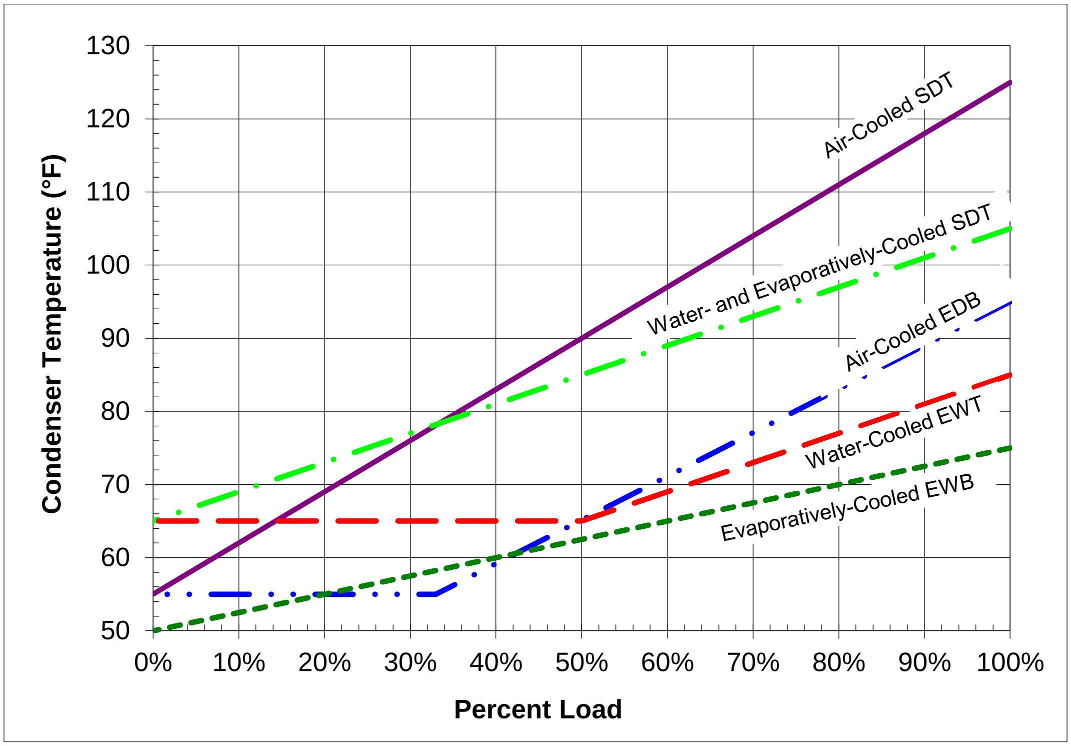

Figure 1. Part-load Condenser Temperature for IPLV.IP

Entering air dry-bulb temperature (EDB), °F, for an Air-cooled Condenser at IPLV.IP part load conditions (refer to Figure 1) shall use Equation 13:

$$\mathrm{EDB} = \begin{vmatrix} 60\cdot\%\,\mathrm{Load}+35 & \text{for Load } >33\% \\ 55 & \text{for Load } \leq 33\% \end{vmatrix} \qquad 13$$

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 atmospheric pressure correction.

Entering water temperature (EWT), °F, for a Water-cooled Condenser at IPLV.IP part load conditions (refer to Figure 1) shall use Equation 14:

$$\mathrm{EWT} = \begin{vmatrix} 40\cdot\%\,\mathrm{Load}+45 & \text{for Load } >50\% \\ 65 & \text{for Load } \leq 50\% \end{vmatrix} \qquad 14$$

Entering air wet-bulb temperature (EWB), °F, for an Evaporatively-cooled Condenser at IPLV.IP part load conditions (refer to Figure 1) shall use Equation 15:

$$\mathrm{EWB} = 25\cdot\%\mathrm{Load} + 50 \qquad 15$$

Saturated discharge temperature (SDT), °F, for an air-cooled unit without condenser at IPLV.IP part load conditions (refer to Figure 1) shall use Equation 16:

$$\text{AC SDT} = 70 \cdot \%\text{Load} + 55 \tag{16}$$

Saturated discharge temperature (SDT), °F, for a water-cooled (WC) or evaporatively-cooled (EC) unit without condenser at IPLV.IP part load conditions (refer to Figure 1) shall use Equation 17a or 17b:

$$\text{WC SDT} = 40 \cdot \%\text{Load} + 65 \tag{17a}$$
$$\text{EC SDT} = 40 \cdot \%\text{Load} + 65 \tag{17b}$$

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:

$$EER_{CD} = \frac{\text{EER}_{\text{Test}}}{\text{C}_\text{D}} \tag{18}$$

$$\text{COP}_{\text{R,CD}} = \frac{\text{COP}_{\text{Test}}}{\text{C}_\text{D}} \tag{19}$$

$$\left(\frac{\text{kW}}{\text{ton}_\text{R}}\right)_{CD} = \left(\frac{\text{kW}}{\text{ton}_\text{R}}\right)_{Test} \cdot \text{C}_\text{D} \tag{20}$$

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

C_D shall be calculated using the following equation:

$$\text{C}_\text{D} = (-0.13 \cdot \text{LF}) + 1.13 \tag{21}$$

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

$$\text{LF} = \frac{(\%\text{Load})\ (\text{Q}_{\text{ev 100\%}})}{(\text{Q}_{\text{ev min\%Load}})} \tag{22}$$

Where:

%Load = One of the standard rating points, i.e. 75%, 50%, or 25%
$Q_{ev\ 100\%}$ = The rated unit net capacity at design conditions
$Q_{evmin\%Load}$ = The measured or calculated unit net capacity at the minimum step of capacity including atmospheric pressure corrections as applicable

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 *Procedures for Testing and Calculation of IPLV/NPLV for Continuous Capacity Control Units.*

For fully continuous capacity controlled units or units with a combination of staged capacity and continuous capacity covered by this standard, the IPLV.IP/NPLV.IP shall be calculated using test data and or rating data using the following procedures.

For test purposes, units shall be provided with manual means to adjust the unit refrigeration capacity by adjusting variable capacity compressor(s) capacity and or the stages of refrigeration capacity as defined by the manufacturer's instructions.

The following sequential steps shall be followed:

5.4.1.3.1 *Step 1.* The unit shall be configured per the manufacturer’s instructions, including setting of stages of refrigeration and variable capacity compressor loading percent for each of the 4 rating percent load rating points of 100%, 75%, 50%, and 25%.

The condenser entering temperature shall be adjusted per the requirements of Table 3 as determined by the rating percent load of 100%, 75%, 50% and 25% and be within the required temperature limits per Table 12 for the 100% rating point and for the 75%, 50% and 25% points if the adjusted capacity is within 2% of the rating percent load. If the adjusted measured percent load difference is outside the 2% tolerance, then the actual adjusted measured percent load shall be used to determine the condenser temperature using equations in Section 5.4.1.2 depending on the condenser type. If the unit would operate with head pressure control active during the test at a the specified condenser temperature which would cause cycling and stable test conditions cannot be maintained then the condenser temperature should be increased to a condition where the cycling will not occur.

If the unit is an air-cooled chiller or evaporatively-cooled then the measured capacity and efficiency shall be adjusted for atmospheric pressure using the procedures of Appendix F. No adjustment is required for water-cooled units.

If the unit is a Packaged Unit, the adjusted full load capacity must be greater than the tolerance defined in Table 11. If the unit is a Configurable Unit then the capacity must be within the tolerance range defined by Table 11. If the capacity is not in compliance with the requirements, the test shall be repeated.

If the adjusted part load test capacity is within ±2% of the target percent load of 75%, 50% and 25% then the adjusted efficiency can be used directly to calculate the IPLV.IP/NPLV.IP. If the adjusted capacity of any point is not within the ±2% tolerance then the test shall be repeated or move to Step2 or Step 3.

5.4.1.3.2 *Step 2.* If the unit, due to its capacity control logic cannot be operated at the rating 75%, 50%, or 25% percent load point within ±2%, then additional test points for use in linear interpolation are required. Capacity staging and variable capacity shall be selected to have one test as close as possible to the desired rating point with an adjusted capacity above the desired rating percent load rating point of 75%, 50% and 25% and a second test as close as possible to the desired rating percent load with an adjusted capacity below the desired rating percent load of 75%, 50%, and 25%.

The condenser entering temperature shall be adjusted per the requirements of Table 3 using the test point adjusted percent load and be within the required temperature limits per Table 12.

The test capacity and efficiency for air and evaporatively-cooled chillers shall then be adjusted for atmospheric pressure using the procedures of Appendix F. No adjustment is required for water-cooled units.

Linear interpolation between the two adjusted capacity points shall then be used to determine the efficiency at the rating 75%, 50% or 25% percent load point, using the entering condenser temperature per Table 3 at the tested capacity. Extrapolation of the data is not allowed and there must be a test point above and below the rating percent load point.

5.4.1.3.3 *Step 3.* If the unit cannot be unloaded to any of the 75%, 50%, or 25% rating points at the minimum stage of unloading then the unit shall be run at the minimum stage of capacity for each of the test points where appropriate.

The condenser entering temperature shall be adjusted per the requirements of Table 3 using the rating percent load of 75%, 50%, or 25% and be within the required temperature limits per Table 12. If the unit would operate with head pressure control active during the test at a specified condenser temperature which would cause cycling and stable test conditions cannot be maintained then the condenser temperature should be increased to a condition where the cycling will not occur.

The capacity and efficiency for air and evaporatively-cooled chillers shall then be adjusted for atmospheric pressure using the procedures of Appendix F. No adjustment is required for Water-cooled chillers.

If the data for the lowest stage of capacity is above the desired rating point load with allowance for the 2% tolerance then the efficiency shall then be adjusted for cyclic degradation using the Equation 18, 19, or 20.

5.4.1.3.4 *Step 4.* Once the adjusted efficiency for each of the 100%, 75%, 50% and 25% rating percent load rating points is determined using Steps 1, 2, or 3 as appropriate, then the IPLV.IP/NPLV.IP shall be calculated using Equation 11 or 12.

5.4.1.4 *Procedures for Testing and Calculation of IPLV/NPLV for Discrete Capacity Step Controlled Units*

For discrete capacity step controlled units, including units with only a single stage of capacity, the IPLV.IP/NPLV.IP shall be calculated using test data and or rating data obtained using the following procedures.

For test purposes, units shall be provided with manual means to adjust the unit refrigeration capacity by adjusting the stages of refrigeration capacity as defined by the manufacturer's instructions.

The following sequential steps shall be followed:

5.4.1.4.1 *Step 1.* The unit shall be configured per the manufacturer's instructions, including setting of stages of refrigeration for each of the 4 rating percent load rating points of 100%, 75%, 50%, and 25%.

The condenser entering temperature shall be adjusted per the requirements of Table 3 as determined by the rating percent load of 100%, 50%, 75% and 25% and be within the required temperature limits per Table 12. If the unit would operate with head pressure control active during the test at the specified condenser temperature which would cause cycling and stable test conditions cannot be maintained then the condenser temperature should be increased to a condition where the cycling would not occur.

If the unit is an air-cooled chiller then the measured capacity and efficiency shall be adjusted for atmospheric pressure using the procedures of Appendix F. No adjustment is required for Water-cooled and evaporatively-cooled units.

If the adjusted part load test capacity is within 2% of the target percent load of 75%, 50% and 25% then the adjusted efficiency can be used directly to calculate the IPLV.IP/NPLV.IP. If the adjusted capacity of any point is not within the 2% tolerance then move to Step 2 or 3.

5.4.1.4.2 *Step 2.* If the unit, due to its capacity control logic cannot be operated at the rating 75%, 50%, or 25% percent load point within 2%, then additional test points for use in linear interpolation are required. Capacity staging shall be selected to have one test as close as possible to the desired rating point with an adjusted capacity above the desired rating percent load rating point of 75%, 50% and 25% and a second test as close as possible to the desired rating percent load with an adjusted capacity below the desired rating percent load of 75%, 50%, and 25%. Capacity staging with a capacity greater or less than the capacity staging closest to the desired rating point shall not be used.

The condenser entering temperature shall be adjusted per the requirements of Table 3 using the test point corrected percent load and be within the required temperature limits per Table 12.

The test capacity and efficiency shall then be adjusted for atmospheric pressure using the procedures of Appendix F.

Linear interpolation between the two adjusted capacity points shall then be used to determine the efficiency at the rating 75%, 50% or 25% percent load point. Extrapolation of the data is not allowed and there must be a test point above and below the rating percent load point.

5.4.1.4.3 *Step 3* If the unit cannot be unloaded to any of the 75%, 50%, or 25% rating points at the minimum stage of unloading then the unit shall be run at the minimum stage of capacity for each of the test points where appropriate.

The condenser entering temperature shall be adjusted per the requirements of Table 3 using the rating percent load of 75%, 50%, or 25% and be within the required temperature limits per Table 12. If the unit would operate with head pressure control active during the test at the specified condenser temperature which would cause cycling and stable test conditions cannot be maintained then the condenser temperature should be increased to a condition where the cycling would not occur.

The capacity and efficiency shall then be adjusted for atmospheric pressure for air and evaporatively-cooled chillers using the procedures of Appendix F.

The efficiency shall then be adjusted for cyclic degradation using the Equations 18, 19, or 20.

5.4.1.4.4 *Step 4.* Once the adjusted efficiency for each of the 100%, 75%, 50% and 25% rating percent load rating points is determined using step 1, 2, or 3 as appropriate, then the IPLV.IP/NPLV.IP shall be calculated using Equations 11 or 12.

5.4.1.5 *Sample Calculations.* The following are examples of the IPLV.IP/NPLV.IP calculations:

5.4.1.5.1 *Example 1.*

The chiller is a Water-cooled centrifugal chiller that has proportional capacity control and can be unloaded to less than 25%. The chiller has a full-load rated capacity of 500 ton_R and a full-load rated efficiency of 0.600 kW/ton_R. The unit can be run at the required conditions in Table 3 for IPLV.IP calculation. Table 4A shows the test results obtained. Because this is a water-cooled unit no corrections need to be made for atmospheric pressure.

Table 4A. Chiller Performance - IPLV.IP for Example 1 Test Results

Test No	Target Rating % Load, %	Target Capacity, ton_R	Measured Net Capacity, ton_R	Measured % Load, %	Different from target capacity, %	Target Condenser EWT, ºF	Measured Power, kW	Efficiency, kw/ton_R
1	100.0	500.0	515.0	103.0	3.00	85.00	296.6	0.5760
2	75.0	375.0	381.0	76.20	1.20	75.00	196.6	0.5160
3	50.0	250.0	266.0	53.20	3.20	66.28	140.7	0.5289
4	50.0	250.0	239.0	47.80	-2.20	65.00	131.1	0.5487
5	25.0	125.0	130.0	26.00	1.00	65.00	97.50	0.7500

Test 1 can be used for the full load IPLV.IP rating point A directly as the capacity is within 3% of the target and as per Table 11 where a tolerance of ±5% is allowed. Test 2 can also be used for the IPLV.IP rating point B because it is within 2% of the target capacity as required by Section 5.4.2. Test 3 cannot be used directly for the IPLV.IP rating point C because the capacity is 3.2% greater than the required load of 50%. Another test could be run to try to get the capacity within 2%, but for this case it was chosen to run a lower capacity test for and then use interpolation. Test 5 can be used directly for the IPLV.IP rating point D as the capacity is within 1% of the target capacity for 25% load. In the following Table 4B you will find the IPLV.IP rating point data that can then be used to calculate the IPLV.IP.

Table 4B. Example 1 IPLV.IP Calculations					
Rating Point	Target Rating % Load, %	Measured Net Capacity, ton_R	Measured Power, ton_R	Efficiency, kw/ton_R	Comment
A	100.0	515.0	296.6	0.5760	Use test point 1 directly.
B	75.0	381.0	196.6	0.5160	Use test point 2 directly.
C	50.0	-	-	0.5406	Interpolate Test 3 and 4.
D	25.0	130.0	97.50	0.7592	Use test point 5 directly.

The IPLV.IP calculations are shown below using the rating data for IPLV.IP ratings points A, B, C and D. Because the ratings are in kw/ton Equation 12 should be used.

$$\mathrm{IPLV.IP} = \frac{1}{\frac{0.01}{0.5760} + \frac{0.42}{0.5160} + \frac{0.45}{0.5406} + \frac{0.12}{0.7592}} = 0.5489$$

5.4.1.5.2 *Example 2.*

The chiller is a Water-cooled centrifugal chiller that has proportional capacity control. The unit can be run at the 100%, 75%, and 50% part load rating points, but it can only unload to 27.7% and the required test point D cannot be run. The chiller has a full-load rated capacity of 800 ton_R and a full-load efficiency of 0.632 kW/ton_R. The full load design conditions for the evaporator have a 42.00ºF leaving water with a 50.00ºF entering temperature. The condenser conditions at full load design are 89.00ºF entering water temperature with a 98ºF leaving water temperature. Because this unit is a Configurable Unit and was selected for the lift associated with these non-standard operating conditions the IPLV.IP rating metric should not be used and instead the NPLV.IP metric used with the NPLV.IP conditions and requirements of Table 3. Note that for an NPLV.IP test the condenser temperature should very from the rated full load condition of 89.00ºF down to 65.00°F and then remain constant. Because this is a Water-cooled machine no atmospheric pressure corrections are required. Shown below in Table 5A is the test data that was obtained as part of a verification test.

Table 5A. Chiller Performance - NPLV.IP for Example 2 Test Results								
Test No	Target Rating % Load, %	Target Capacity, ton_R	Measured Net Capacity, ton_R	Measured % Load, %	Different from target capacity, %	Target Condenser EWT, ºF	Measured Power, kW	Efficiency, kw/ton_R
1	100.0	800.0	803.7	100.5	0.5	89.0	507.9	0.6320
2	75.00	600.0	608.2	76.0	1.0	77.0	316.9	0.5210
3	50.00	400.0	398.5	49.8	-0.2	65.0	183.3	0.4600
4	25.00	200.0	221.5	27.7	2.7	65.0	125.2	0.5652

Test 1 was within 0.5% of the rated full load rated capacity so it is within the allowable tolerance as defined in Table 11 so the test data can be used directly in the NPLV.IP calculations for rating point A. Test 2 is within 1% of the 75% load point capacity so it can also be used to directly for the NPLV.IP rating point B. Test 3 is also within -0.2% of the 50% load point capacity point so it can be used directly for the NPLV.IP rating point C. Test 4 was run at the lowest capacity unloading capability of the chiller and the capacity load of 27.7% is greater than 2% above the required rating point of 25% so it cannot be used directly. Because this is the lowest capacity point it is not acceptable to use interpolation where a rating point above and below the 25% would be required. Extrapolation is not allowed by the standard. Therefore for NPLV.IP rating point D determination a degradation factor needs to be applied to the measured efficiency to reflect that the unit will be cycling at a 25% load point. The calculations for the NPLV.IP are shown below.

Table 5B. Example 2 NPLV.IP Calculations

Rating Point	Target Rating % Load, %	Measured Net Capacity, ton_R	Measured Power, kW	LF	CD	Efficiency, kw/ton_R	Comment
A	100.0	803.7	507.9	-	-	0.6320	Use Test 1 directly.
B	75.00	608.2	316.9	-	-	0.5210	Use Test 2 directly.
C	50.00	398.5	183.3	-	-	0.4600	Use Test 3 directly.
D	25.00	-	-	0.9029	1.0126	0.5724	Use Test 4 with C_D.

The following is a summary of the calculations for the degradation of rating Test 4:

$$\text{LF} = \frac{0.25 \cdot 800}{221.5} = 0.9029$$

$$C_D = (-0.13 \cdot 0.9029) + 1.13 = 1.0126$$

$$\text{kW/ton}_{\text{R25\%,CD}} = 1.0126 \cdot 0.5652 = 0.5724$$

With the data for the 4 NPLV.IP rating points A, B, C, and D the NPLV.IP can then be calculated using Equation 12

$$\text{NPLV.IP} = \frac{1}{\frac{0.01}{0.6320} + \frac{0.42}{0.5210} + \frac{0.45}{0.4600} + \frac{0.12}{0.5724}} = 0.4975$$

5.4.1.5.3 *Example 3.*

The chiller is an air-cooled chiller rated at 150 ton_R. The test data was run when the atmospheric pressure was 14.420 psia. The full-load measured capacity is 148.2 ton_R with an EER of 10.44. After atmospheric adjustment to sea level conditions, the adjusted capacity is 148.4 ton_R with a full-load adjusted EER of 10.48. The unit has 10 stages of capacity control and can unload down to a minimum of 15% of rated load. Only 7 stages of capacity control are needed for the computation of rating point data for the IPLV.IP calculations. The standard procedures require that for interpolation capacity points closest to the desired ratings point must be used. Larger or smaller capacity points from other stages cannot be used. Shown below is the test data that was obtained for the 7 points that will be used for interpolation and calculation of the IPLV.IP. Because this is an air cooled chiller the test performance must be corrected to standard atmospheric pressure of 14.696 using the procedures in Appendix F. Shown in Table 6A are the test data and the corrections for atmospheric pressure.

Table 6A. Chiller Performance - IPLV.IP for Example 3 Test Results

Test No	Target Rating % Load, %	Target Capacity, ton_R	Measured Net Capacity, ton_R	Measured Total Power, kW	Measured Efficiency, Btu/W	Capacity Correction Factor	Efficiency Correction Factor	Corrected Capacity, ton_R	Corrected Efficiency, Btu/W	Corrected Measured % Load, %	Different from target capacity, %	Target EDB, ºF
1	100.0	150.0	148.2	170.3	10.44	1.0017	1.0039	148.4	10.48	98.97	-1.03	95.00
2	75.0	112.5	124.5	125.8	11.88	1.0014	1.0032	124.7	11.91	83.12	8.12	84.87
3	75.0	112.5	105.7	93.8	13.52	1.0012	1.0028	105.8	13.56	70.55	-4.45	77.33
4	50.0	75.0	82.4	66.8	14.80	1.0009	1.0021	82.5	14.83	54.98	4.98	67.99
5	50.0	75.0	62.8	49.5	15.22	1.0007	1.0016	62.8	15.24	41.90	-8.10	60.14
6	25.0	37.5	45.2	36.2	14.97	1.0005	1.0012	45.2	14.99	30.15	5.15	55.00
7	25.0	37.5	22.5	19.0	14.20	1.0003	1.0006	22.5	14.21	15.00	-10.00	55.00

Before using the test data to calculate the IPLV.IP the data must be corrected for the atmospheric pressure of 14.420 psia using Appendix F. The calculations for test point 2 are shown below as an example of the atmospheric correction calculations.

Capacity $Q_{ev75\%Load}$ = 124.5 ton_R
Capacity $Q_{ev100\%Load}$ = 148.2 ton_R
Efficiency $\eta_{tested\ FL}$ = 10.44 EER
Efficiency η_{test} = 11.88 EER
Atmospheric pressure p_{atm} = 14.42 psia

Correction factor $D_Q = 0.0011273 \cdot 14.42^2 - 0.041272 \cdot 14.42 + 1.36304 = 1.0023$

Correction factor $D\eta = 0.0024308 \cdot 14.42^2 - 0.090075 \cdot 14.42 + 1.79872 = 1.0053$

Correction Factor $CF_Q = 1 + (124.5/148.2) \cdot (1.0023-1) \cdot exp[-0.35 \cdot (1.0053 \cdot 10.44-9.6)] = 1.0014$

Correction Factor $CF\eta = 1 + (124.5/148.2) \cdot (1.0053-1) \cdot exp[-0.35 \cdot (1.0053 \cdot 10.44-9.6)] = 1.0032$

Corrected capacity $Q_{corrected} = 124.5 \cdot 1.0014 = 124.7$ ton_R

Corrected efficiency $\eta_{corrected} = 11.88 \cdot 1.0032 = 11.91$ EER

Once the corrections are made then the following Table 6B shows the calculations that are done to determine the IPLV.IP rating points.

Table 6B. Example 3 IPLV.IP Calculations

Rating Point	Target Rating % Load, %	Measured Net Capacity, kW	Measured Power, kW	LF	CD	Efficiency, Btu/W	Comment
A	100.0	148.2	170.3	-	-	10.48	Use Test 1 directly.
B	75.00	-	-	-	-	12.98	Interpolate Test 2 and 3.
C	50.00	-	-	-	-	14.99	Interpolate Test 4 and 5.
D	25.00	-	-	-	-	14.72	Interpolate Test 6 and 7.

For the IPLV.IP rating point A then Test 1 can be used. It is within the required tolerance of -5% on capacity so the corrected efficiency can be used for the point A rating efficiency. For rating points B, C, and D the measure capacity is greater or less than the required load ±2% so interpolation must be used. 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 13 to determine the target entering dry-bulb temperature (EDB) using the corrected measured percent load. Use these target outdoor air temperatures when evaluating tolerance criteria in Table E2. So for rating point B linear interpolation is used using corrected Test 2 and 3 and similar linear interpolation for rating point C and D

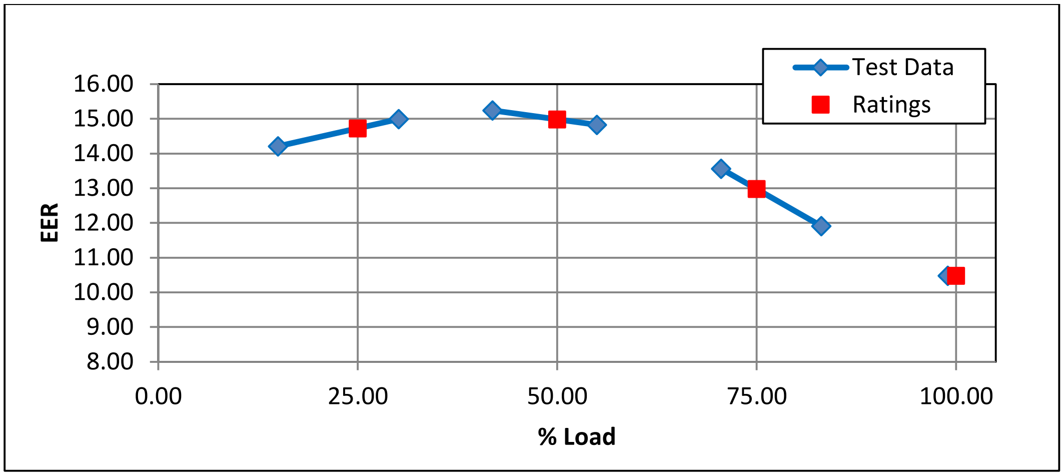

Figure 2. Rating Point Interpolation

An example of the linear interpolation is shown below for the 25% point D calculation.

$$\mathrm{EER}_{25\%} = \left(\frac{25\% - 15.00\%}{30.15\% - 15.00\%}\right)(14.99 - 14.21) + 14.21 = 14.72$$

The IPLV.IP can then be calculated using the efficiencies determined from the interpolation for the IPLV.IP rating point A, B, C and D. Note: because the ratings are in EER, Equation 11 is used.

$$\mathrm{IPLV.IP} = (0.01 \cdot 10.48) + (0.42 \cdot 12.98) + (0.45 \cdot 14.99) + (0.12 \cdot 14.72) = 14.07$$

5.4.1.5.4 *Example 4.*

For this example we have an air-cooled chiller rated at 110 $\mathrm{ton_R}$. The full-load measured capacity is 110.2 $\mathrm{ton_R}$ with an EER of 9.558. After atmospheric adjustment to sea level conditions for a measured atmospheric pressure of 14.200, the capacity is 110.7 $\mathrm{ton_R}$ with a full-load EER of 9.667. The unit has 3 stages of capacity with the last stage of capacity greater than the required 25% rating point. The tests run for verification of the IPLV.IP are shown in Table 7A. In the table it also shows the corrections for atmospheric pressure which is required because the unit is an air-cooled chiller.

Table 7A. Chiller Performance - IPLV.IP for Example 4 Test Results

Test No	Target Rating % Load	Target Capacity, $\mathrm{ton_R}$	Measured Net Capacity, $\mathrm{ton_R}$	Measured Total Power, kW	Measured Efficiency, Btu/W	Capacity Correction Factor	Efficiency Correction Factor	Corrected Capacity, $\mathrm{ton_R}$	Corrected Efficiency, Btu/W	Measured % Load, %	Difference from target capacity, %	Target EDB, ºF
1	100.0	110.0	110.2	138.1	9.576	1.0042	1.0096	110.7	9.667	100.60	0.60	95.00
2	75.0	82.5	79.30	77.70	12.25	1.0030	1.0069	79.5	12.33	72.31	-2.69	78.38
3	50.0	55.0	45.40	39.00	13.97	1.0017	1.0039	45.5	14.02	41.34	-8.66	59.81
4	25.0	27.5	46.50	40.90	13.64	1.0018	1.0040	46.6	13.70	42.35	17.35	55.00

Test 1 is an acceptable test for the full load IPLV.IP rating point A because the corrected capacity is within 0.6% of the rated capacity and above the -5.0% tolerance as defined in Table 11. Tests 2 and 3 are not within the 2% acceptable tolerance for part load points so interpolation will be required. Because the unit cannot unload to 25% and because the Test 3 at minimum unloading was run at the required 59.81°F target ambient temperature an additional test has been run at the 25% ambient temperature of 55.00°F and this point will then be used along with a degradation factor to calculate the 25% point D rating point. Shown below in Table 7B is the calculations for the IPLV.IP.

Table 7B. Example 4 IPLV.IP Calculations							
Rating Point	Target Rating % Load, %	Measured Net Capacity, kW	Measured Power, kW	LF	CD	Efficiency, Btu/W	Comment
A	100.0	110.2	138.1	-	-	9.667	Use Test 1 directly.
B	75.00	-	-	-	-	12.08	Interpolate Test 1 and 2.
C	50.00	-	-	-	-	13.55	Interpolate Test 2 and 3.
D	25.00	-	-	0.5904	1.0533	13.01	Use Test 4 with $C_{D.}$

Test 1 is used directly for the IPLV.IP rating point A. Test 1 and 2 are used for the IPLV.IP rating point B interpolation and Test 2 and 3 are used to interpolate the C rating point. For the IPLV.IP rating point D the degradation factor is applied to the Test 4 corrected efficiency to account for the cycling of the last stage of capacity.

The IPLV.IP can then be calculated using the efficiencies determined from the interpolation for the IPLV.IP rating point A, B, C and D. Note: because the ratings are in EER, Equation 11 is used.

$$\mathrm{IPLV.IP} = (0.01 \cdot 9.667) + (0.42 \cdot 12.08) + (0.45 \cdot 13.55) + (0.12 \cdot 13.01) = 12.83$$

5.4.1.5.5 *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 must be used to generate the rating data for the 75%, 50%, and 25% IPLV.IP rating points. The units can only run at full-load, thus additional performance information is required with the unit running at the 75.00ºF entering condenser water temperature for the B rating point and at 65.00ºF condenser entering water for the C and D rating point. The condenser water temperature is 65.00ºF for both the 50% and 25% rating points because the load is equal to or less than 50%, thus only 3 test points are required to generate the IPLV.IP rating data. The chiller has the following test information as shown in in Table 8A. Note that because this is a water cooled unit no atmospheric pressure corrections are required.

Table 8A. Chiller Performance - IPLV.IP for Example 5 Test Results								
Test No	Target Rating % Load, %	Target Capacity, ton_R	Measured Net Capacity, ton_R	Measured % Load	Different from target capacity	Target Condenser EWT, ºF	Measured Power, kW	Efficiency, kw/ton_R
1	100.0	15.00	15.30	102.0	2.00	85.00	11.90	0.7778
2	75.0	11.25	17.30	115.33	40.33	75.00	10.60	0.6127
3	50.0	7.500	19.80	132.00	82.00	65.00	11.30	0.5707

The test point 1 can be used for the A rating point because it has a capacity greater than the capacity tolerance defined in Table 11. For the B, C, and D rating points degradation factors need to be applied to the ratings test results. The IPLV.IP rating point data is shown in Table 8B

Table 8B. Example 5 IPLV.IP Calculations							
Rating Point	Target Rating % Load, %	Measured Net Capacity, ton_R	Measured Power, kW	LF	CD	Efficiency, kw/ton_R	Comment
A	100.0	15.3	11.9	-	-	0.7778	Use Test 1 directly
B	75.00	-	-	0.6503	1.0455	0.6406	Use Test 2 directly
C	50.00	-	-	0.3788	1.0808	0.6168	Use Test 3 directly
D	25.00	-	-	0.1894	1.1054	0.6308	Use Test 4 with C_D

The IPLV.IP can then be calculated using the efficiencies determined from the interpolation for the IPLV.IP rating point A, B, C and D. Note: because the ratings are in kW/ton_R, Equation 12 is used.

$$\mathrm{IPLV.IP} = \frac{1}{\frac{0.01}{0.778} + \frac{0.42}{0.6406} + \frac{0.45}{0.6168} + \frac{0.12}{0.6308}} = 0.6296$$

5.4.1.5.6 *Example 6.*

For this example the chiller is an air-cooled chiller with continuous unloading rated at 200 ton_R. The full-load measured capacity is 197.2 ton_R with an EER of 9.718. After atmospheric adjustment to sea level conditions, the capacity is 199.2 ton_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 9A. The atmospheric pressure measured during the test was 13.50 psia.

Table 9A. Chiller Performance - IPLV.IP for Example 6 Test Results												
Test No	Target Rating % Load, %	Target Capacity, ton_R	Measured Net Capacity, ton_R	Measured Total Power, kW	Measured Efficiency, Btu/W	Capacity Correction Factor	Efficiency Correction Factor	Corrected Capacity, ton_R	Corrected Efficiency, Btu/W	Measured % Load, %	Different from target capacity, %	Target EDB, ºF
1	100.0	200.0	197.2	243.5	9.718	1.0099	1.0226	199.2	9.938	99.58	-0.42	95.00
2	75.0	150.0	149.1	146.0	12.25	1.0075	1.0171	150.2	12.46	75.11	0.11	80.00
3	50.0	100.0	100.2	87.00	13.82	1.0051	1.0115	100.7	13.98	50.35	0.35	65.00
4	25.0	50.0	56.50	51.30	13.22	1.0029	1.0065	56.66	13.30	28.33	3.33	55.00

Test number 1 can be used for the full load IPLV.IP rating point A, because the corrected capacity is within -0.42% of the rating and the tolerance is -5.0% for a packaged unit. Test 2, can be used for rating point B as it is within the 2% tolerance for capacity and the same is true for Test 3 which can be used for rating point C. Test 4 cannot be used directly because the capacity is 3.33% above the required capacity and because it is the lowest unloading capability of the unit, a degradation factor has to be applied.

The rating point data calculations are shown in Table 9B.

Table 9B. Example 6 IPLV.IP Calculations							
Rating Point	Target Rating % Load, %	Corrected Capacity, ton_R	Corrected Efficiency, Btu/W	LF	CD	Efficiency, Btu/W	Comment
A	100.0	197.2	9.938	-	-	9.938	Use Test 1 directly.
B	75.00	149.1	12.46	-	-	12.46	Use Test 2 directly.
C	50.00	100.2	13.98	-	-	13.98	Use Test 3 directly.
D	25.00	56.5	13.30	0.8824	1.0153	13.10	Use Test 4 with C_D.

The IPLV.IP can be calculated using the efficiencies determined from the IPLV.IP rating point A, B, C and D. Note: because the ratings are in EER, Equation 11 is used.

$$\mathrm{IPLV.IP} = (0.01 \cdot 9.938) + (0.42 \cdot 12.46) + (0.45 \cdot 13.98) + (0.12 \cdot 13.10) = 13.20$$

5.4.1.5.7 *Example 7.*

The unit is an evaporatively-cooled chiller with a rated capacity of 150 $\mathrm{ton_R}$. It has a rated full load efficiency 14.50. The unit has proportional capacity control, but can only unload to 28.01%. The atmospheric pressure during the test was 14.100 psia. Because this is an evaporatively-cooled machine corrections for atmospheric pressure need to be made using Appendix F. The test data and atmospheric correction data are shown in the following Table 10A.

Table 10A. Chiller Performance - IPLV.IP for Example 7 Test Results

Test No	Target Rating % Load, %	Target Capacity, $\mathrm{ton_R}$	Measured Net Capacity, $\mathrm{ton_R}$	Measured Total Power, kW	Measured Efficiency, Btu/W	Capacity Correction Factor	Efficiency Correction Factor	Corrected Capacity, $\mathrm{ton_R}$	Corrected Efficiency, Btu/W	Measured % Load, %	Different from target capacity, %	Target EWB, °F
1	100.0	150.0	151.0	125.2	14.47	1.0009	1.0020	151.1	14.50	100.76	0.76	75.00
2	75.0	112.5	114.0	84.55	16.18	1.0007	1.0015	114.1	16.20	76.05	1.05	68.75
3	50.0	75.00	73.50	57.27	15.40	1.0004	1.0010	73.5	15.42	49.02	-0.98	62.50
4	25.0	37.50	42.00	45.82	11.00	1.0002	1.0006	42.01	11.01	28.01	3.01	57.00

Test 1 can be used for the IPLV.IP rating point A because the capacity is within 0.76% of the rated capacity and the tolerance is ± 5 for a configured unit. Test 2 can be used directly for IPLV.IP rating point B and Test 3 can be used for rating point C as they adjusted capacities within 2% of the target rating load. Because Test 4 can only unload to 28.01% a degradation correction needs to be made to Test 4 to determine the IPLV.IP rating point D. The IPLV.IP rating point data is shown in the Table 10B.

Table 10B. Example 7 IPLV.IP Calculations

Rating Point	Target Rating % Load, %	Corrected Capacity, $\mathrm{ton_R}$	Corrected Efficiency, Btu/W	LF	C_D	Efficiency, Btu/W	Comment
A	100.0	151.0	14.50	-	-	14.50	Use Test 1 directly.
B	75.00	114.0	16.20	-	-	16.20	Use Test 2 directly.
C	50.00	73.5	15.42	-	-	15.42	Use Test 3 directly.
D	25.00	-	-	0.8926	1.0140	10.85	Use Test 4 with C_D.

The IPLV.IP can be calculated using the efficiencies determined from the IPLV.IP rating points A, B, C and D. Note: because the ratings are in EER, Equation 11 is used.

$$\mathrm{IPLV.IP} = (0.01 \cdot 14.50) + (0.42 \cdot 16.20) + (0.45 \cdot 15.42) + (0.12 \cdot 10.85) = 15.19$$

5.4.2 *Determination of Part-load Performance within Application Rating Limits*. Part load points not meeting the requirements of IPLV.IP or NPLV.IP, but within the Application Rating Condition limits in Table 2, shall be calculated as follows:

5.4.2.1 For continuous capacity control chillers that can run at the application percent load within ±2% of the desired percent load determine the part-load energy efficiency at the application percent load and condenser entering temperature.

5.4.2.2 If the chiller is expected to have cycling at the application ratings conditions, due to either compressor on/off staging or discrete step capacity control, then the rating method shall use two-way linear interpolation between two other rating points. The two points can vary the load capacity to points above and below the rating condition, to conditions where the chiller is not expected to be cycling. The condensing temperature shall be held constant at the desired part load rating point. Linear interpolation shall then be used determine the efficiency at the application rating conditions. Extrapolation shall not be used.

5.4.2.3 If the application percent load is below the lowest capacity stage of the unit then a performance point shall be determined at the application part load condenser entering temperature and lowest stage of capacity and the efficiency adjusted for cyclic degradation using Equations 18, 19 or 20.

5.4.2.4 If the application percent load in operation where condenser heat pressure control is active and stable test results cannot be maintained, then the entering condenser temperature shall be increased until stable operation is obtained.

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.*

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) $h \cdot ft^2 \cdot °F/Btu$.

5.5.1.3 To determine the capacity of the Water-chilling Package at the rated water-side fouling conditions, the procedure defined in Section C4.4 shall be used to determine an adjustment for the evaporator and or condenser water temperatures.

5.6 *Tolerances*.

5.6.1 *Tolerance Limit*. The tolerance limit for test results for Net Capacity, full and part load Efficiency, and Water Pressure Drop shall be determined from Table 11. The limiting value (i.e. minimum or maximum acceptable value for Capacity, Efficiency, or Water Pressure Drop) shall be rounded to the number of significant figures in Table 14.

The tolerance limits are intended to be used when testing a unit to verify and confirm performance. They take into consideration the following:

5.6.1.1 *Uncertainty of Measurement*. When testing a unit, there are variations that result from instrumentation accuracy and installation affects, as well as test facility stability.

5.6.1.2 *Uncertainty of Test Facilities*. The tested performance of the same unit tested in multiple facilities will vary due to setup variations and may not yield the same performance.

5.6.1.3 *Uncertainty due to Manufacturing*. During the manufacturing of units, there are variations due to manufacturing production tolerances that will impact the performance from unit to unit.

5.6.1.4 *Uncertainty of Performance Prediction Models*. Due to the large complexity of options, manufacturers may use performance prediction models to determine ratings.

To comply with this standard, any test per Section 4.1 to verify published or reported values shall be in accordance with Table 14.

Table 11. Definition of Tolerances

		Limits	Related Tolerance Equations[2,3]
Capacity	Cooling or heating capacity for units with continuous unloading[1]	Full Load minimum: 100%- Tol_1 Full Load maximum: 100%+ Tol_1	$Tol_1 = 0.105 - (0.07 \cdot \%Load) + \left(\frac{0.15}{\Delta T_{FL} \cdot \%Load}\right)$ (23) ΔT_{FL} = Difference between entering and leaving water temperature at full-load, °F See Figure 3 for graphical representation of the Tol_1 tolerance.
Capacity	Cooling or heating capacity for units with discrete capacity steps	Full Load minimum: 100% - Tol_1 Full load maximum: no limit (Full Load shall be at the maximum stage of capacity)	
Efficiency	EER	Minimum of: (rated EER) / (100%+ Tol_1)	
Efficiency	kW/ton_R	Maximum of: (100%+ Tol_1)·(rated kW/ton_R)	
Efficiency	COP	Minimum of: (rated COP) / (100%+ Tol_1)	
Efficiency	IPLV.IP NPLV.IP EER	Minimum of: (rated EER) / (100%+ Tol_2)	$Tol_2 = 0.065 + \left(\frac{0.35}{\Delta T_{FL}}\right)$ (24) See Figure 4 for graphical representation of the Tol_2 tolerance.
Efficiency	IPLV.IP NPLV.IP kW/ton_R	Maximum of: (100%+ Tol_2)·(rated kW/ton_R)	
Efficiency	IPLV.IP NPLV.IP COP_R	Minimum of: (rated COP_R) / (100%+ Tol_2)	
Water Pressure Drop		$\Delta p_{corrected} \leq Tol_3$	$Tol_3 = \max\begin{cases} 1.15 \cdot \Delta p_{rated} \\ \Delta p_{rated} + 2\text{ ft } H_2O \end{cases}$ (25)

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 and evaporatively-cooled units, all tolerances are computed for values after the atmospheric correction is taken into account.
3. %Load, Tol_1 and Tol_2 are in decimal form.

Figure 3 is a graphical representation of the related tolerance equation for capacity, efficiency, and energy balance as noted in Table 11.

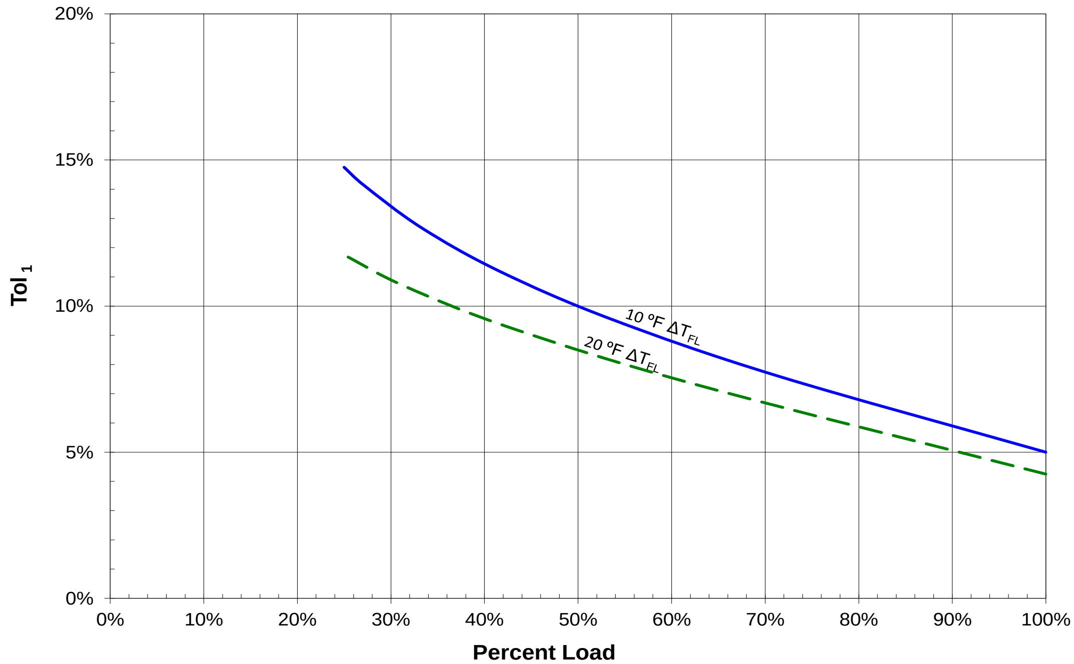

Figure 3. Allowable Tolerance (Tol_1) Curves for Full and Part-load Points

Figure 4 is a graphical representation of the related tolerance equation for IPLV.IP and NPLV.IP as noted in Table 11. The PLV line shown can represent either IPLV.IP or NPLV.IP depending on use.

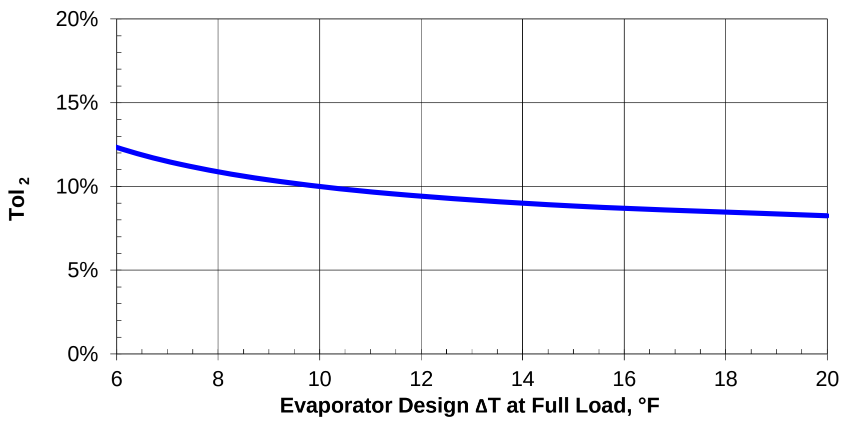

Figure 4. IPLV.IP and NPLV.IP Tolerance (Tol_2) Curve

5.6.2 *Allowable Operating Condition Tolerances.* Tests shall be conducted while maintaining the following tolerance limits on operating conditions. Measurement values and calculation results shall not deviate from published rating values more than the operating condition tolerance limits determined from Table 12.

Table 12. Definition of Operating Condition Tolerances and Stability Criteria

Measurement or Calculation Result	Applicable Operating Mode(s)	Values Calculated from Data Samples: Mean	Values Calculated from Data Samples: Std Dev	Operating Condition Tolerance Limits	Stability Criteria
Net Capacity, Q (Cooling or Heating)	Cooling, Heating, Heat Recovery	-	-	Unit with Continuous Unloading: [1] Part Load test capacity shall be within 2% of the target part-load capacity[2] $\frac{\lvert Q - Q_{target} \rvert}{Q_{100\%}} \leq 2.000\%$	No Requirement
				Units with Discrete Capacity Steps: Part Load test points shall be taken as close as practical to the specified part-load rating points as stated in Table 3	
Cooling Mode Evaporator					
Entering Water Temperature	Cooling, Heating, Heat Recovery	$\overline{T}$	s_T	No Requirement	$s_T \leq 0.18$ °F
Leaving Water Temperature	Cooling, Heating, Heat Recovery	$\overline{T}$	s_T	$\lvert \overline{T} - T_{target} \rvert \leq 0.50$ °F Exception for heating mode only: no requirement during defrost portion.	$s_T \leq 0.18$ °F
Entering Water Temperature	Heating	$\overline{T}$	s_T	Only during defrost portion of cycle: $\lvert \overline{T} - T_{target} \rvert \leq 2.00$ °F	$s_T \leq 0.50$ °F
Cooling Mode Heat Rejection Heat Exchanger (Condenser)					
Entering Water Temperature	Cooling	$\overline{T}$	s_T	$\lvert \overline{T} - T_{target} \rvert \leq 0.50$ °F	$s_T \leq 0.18$ °F
Leaving Water or Fluid Temperature	Heating, Heat Recovery	$\overline{T}$	s_T	$\lvert \overline{T} - T_{target} \rvert \leq 0.50$ °F	$s_T \leq 0.18$ °F
Cooling Mode Heat Rejection Heat Exchanger (Condenser)					
Entering Air Mean Dry Bulb Temperature[3]	Cooling, Heating (non-frosting)	$\overline{T}$	s_T	$\lvert \overline{T} - T_{target} \rvert \leq 1.00$ °F	$s_T \leq 0.75$ °F
	Heating (frosting)[4]			Heating portion: $\lvert \overline{T} - T_{target} \rvert \leq 2.00$ °F	Heating portion: $s_T \leq 1.00$ °F
				Defrost portion: no requirement for $\overline{T}$	Defrost portion: $s_T \leq 2.50$ °F
Entering Air Mean Wet Bulb Temperature[3]	Cooling, Heating (non-frosting)			$\lvert \overline{T} - T_{target} \rvert \leq 1.00$ °F	$s_T \leq 0.50$ °F
	Heating (frosting)[4]			Heating portion: $\lvert \overline{T} - T_{target} \rvert \leq 1.50$ °F	Heating portion: $s_T \leq 0.75$ °F
				Defrost portion: no requirement for $\overline{T}$	No requirement

Table 12. Definition of Operating Condition Tolerances and Stability Criteria

Measurement or Calculation Result	Applicable Operating Mode(s)	Values Calculated from Data Samples		Operating Condition Tolerance Limits	Stability Criteria
		Mean	Std Dev		
Water Flow (Volumetric, Entering)	Cooling, Heating, Heat Recovery	$\bar{V}_w$	s_{Vw}	$\frac{\lvert V_w - V_{w,target}\rvert}{V_{w,target}} \leq 5.000\%$	$\frac{s_V}{\bar{V}} \leq 0.750\%$
Voltage[5] (if multiphase, this is the average of all phases)	Cooling, Heating, Heat Recovery	$\bar{V}$	s_V	$\frac{\lvert \bar{V} - V_{target}\rvert}{V_{target}} \leq 10.00\%$	$\frac{s_V}{\bar{V}} \leq 0.500\%$
Frequency[5]	Cooling, Heating, Heat Recovery	$\bar{\omega}$	s_ω	$\frac{\lvert \bar{\omega} - \omega_{target}\rvert}{\omega_{target}} \leq 1.000\%$	$\frac{s_\omega}{\bar{\omega}} \leq 0.500\%$
Condenserless Refrigerant Saturated Discharge Temperature	Cooling	$\bar{T}$	s_T	$\lvert \bar{T} - T_{target}\rvert \leq 0.50\ °F$	$s_T \leq 0.25\ °F$
Condenserless Liquid Temperature	Cooling	$\bar{T}$	s_T	$\lvert \bar{T} - T_{target}\rvert \leq 1.00\ °F$	$s_T \leq 0.50\ °F$
Steam Turbine Pressure/Vacuum[6]	Cooling, Heating, Heat Recovery	$\bar{p}$	s_p	$\lvert \bar{p} - p_{Rating}\rvert \leq 0.500$ psid	$s_p \leq 0.250$ psid
Gas Turbine Inlet Gas Pressure[6]	Cooling, Heating, Heat Recovery	$\bar{p}$	s_p	$\lvert \bar{p} - p_{Rating}\rvert \leq 0.500$ psid	$s_p \leq 0.250$ psid
Governor Control Compressor Speed[7]	Cooling, Heating, Heat Recovery	$\bar{n}$	s_n	$\frac{\lvert \bar{n} - n_{target}\rvert}{n_{target}} \leq 0.500\%$	$\frac{s_n}{\bar{n}} \leq 0.250\%$

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. The ± 2.0% tolerance shall be calculated as 2.0% of the full load rated capacity (ton_R). For example, a nominal 50.0% part load point shall be tested between 48.0% and 52.0% of the full load capacity to be used directly for IPLV.IP and NPLV.IP calculations. Outside this tolerance, interpolation shall be used..
3. 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.
4. When computing average air temperatures for heating mode tests, omit data samples collected during the defrost portion of the cycle.
5. For electrically driven machines, voltage and frequency shall be maintained at the nameplate rating values within tolerance limits and stability criteria on voltage and frequency when measured at the locations specified at Appendix C. For dual nameplate voltage ratings, tests shall be performed at the lower of the two voltages.
6. For steam turbine and gas turbine drive machines the pressure shall be maintained at the nameplate rating values within the tolerance limits.
7. For speed controlled compressors the speed shall be maintained at the nameplate rating value within the tolerance limits.

5.6.3 *Test Validity Tolerances.* Tests shall be conducted while maintaining the following limits. Measurement values and calculation results shall not deviate more than the validity tolerance limits in Table 13.

Table 13. Definition of Validity Tolerances

Parameter	Limits	Related Tolerance Equations[3]
Energy Balance[1]	$\lvert E_{bal}\rvert \le Tol_4 \times 100\%$	$Tol_4 = 0.074 - (0.049 \cdot \%Load) + \left(\frac{0.105}{\Delta T_{FL} \cdot \%Load}\right)$ 26
Voltage Balance[2]	$V_{bal} \le 2.0\%$	

Notes:
1. Energy balance where applicable shall be calculated in accordance with Section C3.4.1.
2. Not applicable to single phase units. Voltage unbalance calculated per Section C3.4.2.
3. %Load and Tol_4 are in decimal form.

Figure 5 is a graphical representation of the related tolerance equation for energy balance as noted in Table 13.

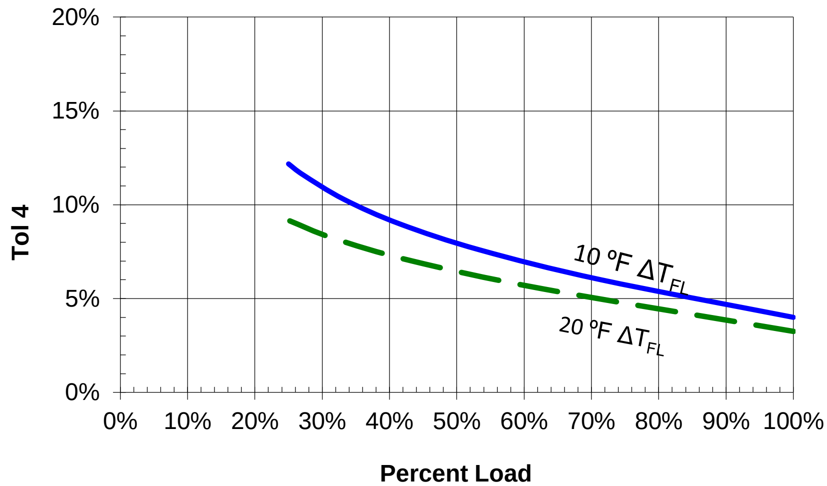

Figure 5. Energy Balance Tolerance (Tol_4) Curve

5.6.4 *Full-load Tolerance Examples*. The tolerance limit on full-load capacity and efficiency shall be determined from Section 5.6.1.

Full-Load Examples.

Full-Load Example in EER

Rated Full-load Performance:

Rated Capacity= 100.0 ton_R
Rated Power = 111.0 kW
Cooling ΔT_{FL} = 10.00ºF

$$EER = \frac{100.0\ ton_R \cdot K5}{111.0\ kW \cdot K7} = 10.81\ \frac{Btu}{W \cdot h}$$

$$\text{Tolerance Limit} = Tol_1 = 0.105 - (0.07 \cdot 1.00) + \left(\frac{0.15}{10.00 \cdot 1.00}\right) = 0.05000$$

$$\text{Minimum Allowable Capacity}(ton_R) = (1.00 - 0.05) \cdot 100.0\ ton_R = 95.00\ ton_R$$

$$\text{Minimum Allowable EER}\left(\frac{\text{Btu}}{\text{W}\cdot\text{h}}\right) = \frac{10.81}{1.000+0.050} = 10.30\ \left(\frac{\text{Btu}}{\text{W}\cdot\text{h}}\right)$$

Full-load Example in kW/ton$_R$

Rated Full-load Performance:

Rated Capacity = 100.0 ton$_R$
Rated Power = 70.00 kW
Cooling ΔT_{FL} = 10.00°F

$$\text{kW/ton}_\text{R} = \frac{70.00\ \text{kW}}{100.00\ \text{ton}_\text{R}} = 0.7000\ \frac{\text{kW}}{\text{ton}_\text{R}}$$

$$\text{Tolerance Limit} = \text{Tol}_1 = 0.105 - (0.07\cdot 1.00) + \left(\frac{0.15}{10.00\cdot 1.00}\right) = 0.05 = 0.05000$$

$$\text{Min. Allowable Capacity} = (1.00 - 0.05)\ \cdot\ 100.0\ \text{ton}_\text{R} = 95.00\ \text{ton}_\text{R}$$

$$\text{Max. Allowable kW/ton}_\text{R} = (1.00 + 0.05)\cdot\ 0.7000\ \text{kW/ton}_\text{R} =\ 0.7350\ \text{kW/ton}_\text{R}$$

Full-load Example in COP (Heat Pump)

Rated Full-load Performance:

Rated Heating Capacity = 1,500,000 Btu/h
Rated Power = 70.00 kW
Condenser ΔT_{FL} = 10.00°F

$$\text{Heating COP}_\text{H} = \frac{1{,}500{,}000\,\frac{\text{Btu}}{\text{h}}}{70\ \text{kW}\cdot 3{,}412.14\ \ \text{Btu/h}\cdot\text{kW}} = 6.280\ \frac{\text{kW}}{\text{kW}}$$

$$\text{Tolerance Limit} = \text{Tol}_1 = 0.105 - (0.07\cdot 1.00) + \left(\frac{0.15}{10.00\cdot 1.00}\right) = 0.05 = 0.05000$$

$$\text{Min. Allowable Capacity} = (1.00 - 0.05)\ \cdot 1{,}500{,}000\ \ \text{Btu/h} = 1{,}425{,}000\ \text{Btu/h}$$

$$\text{Min. Allowable COP}_\text{H} = \frac{6.280\ \frac{\text{W}}{\text{W}}}{1.00+0.05} =\ 5.981\ \ \frac{\text{W}}{\text{W}}$$

5.6.5 *Part-load Tolerance Examples*. The tolerance on part-load EER shall be the tolerance as determined from Section 5.6.1.

Part-load Example in EER

Rated Part-load Performance:

Power at 69.5% Rated Capacity = 59.60 kW
69.5% Rated Capacity = 69.50 ton$_R$
Cooling ΔT_{FL} = 10.00°F

$$\text{EER} = \frac{69.50\ \text{ton}_\text{R}\cdot 12{,}000\ \text{Btu/(h}\cdot\text{tonR)}}{59.6\ \text{kW}\cdot 1{,}000\ \text{W/kW}} = 13.99\ \frac{\text{Btu}}{\text{W}\cdot\text{h}}$$

$$\text{Tolerance Limit} = \text{Tol}_1 = 0.105 - (0.07\cdot 0.695) + \left(\frac{0.15}{10.00\cdot 0.695}\right) = 0.07793$$

$$\text{Minimum Allowable EER} = \frac{13.99}{1.00 + 0.07793} \frac{\text{Btu}}{\text{W} \cdot \text{h}} = 12.98 \frac{\text{Btu}}{\text{W} \cdot \text{h}}$$

Part-load Example in kW/ton$_R$

Rated Part-load Performance:

Power at 50% Rated Capacity = 35.00 kW
50% Rated Capacity = 50.00 ton$_R$
Cooling ΔT_{FL} = 10.00 °F

$$\text{kW/ton}_\text{R} = \frac{35\ \text{kW}}{50\ \text{ton}_\text{R}} = 0.7000\ \text{kW/ton}_\text{R}$$

$$\text{Tolerance Limit} = \text{Tol}_1 = 0.105 - (0.07 \cdot 0.50) + \left(\frac{0.15}{10.00 \cdot 0.50}\right) = 0.1000$$

$$\text{Maximum Allowable kW/ton}_\text{R} = (1.00 + 0.10) \cdot 0.700 = 0.7700\ \text{kW/ton}_\text{R}$$

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. Metrics at Standard Rating Conditions shall be per Sections 5.1 and 5.2. Exception: chillers using centrifugal type compressors shall use Application Rating Conditions per Section 5.3 using the water-side Fouling Factor Allowance as given in the notes section of Table 1 unless the specified application states that a different Fouling Factor Allowance value shall be used. Rated capacity $Q_{100\%}$, ton$_R$, for positive displacement chillers is the net capacity at full-load AHRI Standard Rating Conditions per Table 1. Rated capacity $Q_{100\%}$, ton$_R$, for centrifugal chillers is the net capacity at full-load AHRI Application Rating Conditions within the range permitted in Table 2.

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 be rounded to the number of significant figures shown in Table 14, using the definitions, rounding rules and formats in Section 4.3. Published Ratings and Application Ratings shall state all of the operating conditions used to establish the ratings 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 7), and/or Net Heating Capacity (Equation 9), Btu/h or ton$_R$

6.2.1.4 Total Input Power to chiller, kW, including all Auxiliary Power.

6.2.1.4.1 Exclude input power to integrated water pumps, when present.

6.2.1.4.2 For electric-drive packages provided with starters, transformers, gearboxes, or variable speed drives, whether self-contained or remote-mounted (free-standing), the input power shall include the power losses due to those components.

6.2.1.4.3 For non-electric drive packages, such as turbine or engine drive, the input power shall include the losses due to the prime mover and other driveline components such as a gearbox.

6.2.1.4.4 When the Water-chilling or Water-heating Package does not include some components, which are provided by another party independently from the chiller manufacturer, the input power and any losses associated with those components shall be determined as follows:

For electric-drive packages rated for fixed-speed operation but not including a starter, use the compressor motor terminal input power when determining the Total Input Power.

For electric-drive packages rated for variable-speed operation but not including a variable speed drive, assume a variable speed control method and variable speed drive type consistent with the chiller manufacturer installation requirements, and use the compressor motor terminal input power when determining the Total Input Power.

When a motor or other non-electric drive is not included with the Water-chilling or Water-heating Package, assume a speed control method consistent with the chiller manufacturer installation requirements, and use the compressor shaft input power when determining the Total Input Power.

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

It is important to note that pump energy associated with Water 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, $h·ft^2·°F/Btu$, as stated in Table 1 or Table 2

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

6.2.1.8 Integral Pumps.

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

6.2.1.8.2 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.IP/NPLV.IP, 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 Condenser water entering and leaving temperatures, ºF, as stated in Table 1, or leaving water temperature and temperature difference, ºF

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

6.2.2.4 Condenser Fouling Factor, $h \cdot ft^2 \cdot °F/Btu$, as stated in Table 1 or Table 2

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 Altitude, ft

6.2.3.3 Optional itemization, input power 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 Statement of condenser Fouling Factor Allowance on heat exchanger, $h \cdot ft^2 \cdot °F/Btu$

6.2.4.3 Altitude, ft

6.2.4.4 Optional itemization

6.2.4.4.1 Input power to fan(s), kW

6.2.4.4.2 Optional itemization, condenser spray pump power consumption, kW

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 Recovery Condenser(s).*

6.2.6.1 Heat Recovery net capacity, MBtu/h

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

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

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

6.2.6.5 Fouling Factor, $h \cdot ft^2 \cdot °F/Btu$, as stated in Table 1

6.2.7 *Water-to-Water Heat Pumps.*

6.2.7.1 Net Heating Capacity, MBtu/h

6.2.7.2 Condenser 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, $h \cdot ft^2 \cdot °F/Btu$, as stated in Table 1 or Table 2

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 Net Heating Capacity, MBtu/h

6.2.8.2 Condenser 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

6.2.8.7 Optional itemization, input power to fan(s), kW

6.3 *Summary Table of Data to be Published.* Table 14 provides a summary of Section 6 items. In case of discrepancy, the text version shall be followed.

Table 14. Published Values

Published Values	Units	Significant Figures[3]	Water-Cooled Chiller (Cooling)	Water-Cooled Heat Recovery Chiller	Evaporatively Cooled Chiller	Air-Cooled Chiller	Condenserless Chiller	Air-Cooled HP (Cooling)	Air-Cooled HP (Heating)	Air Cooled Heat Recovery Chiller	Water to Water HP (Cooling)	Water to Water HP (Heating)
General												
Voltage	V	3	■	■	■	■	■	■	■	■	■	■
Frequency	Hz	3	■	■	■	■	■	■	■	■	■	■
Refrigerant Designation		-	■	■	■	■	■	■	■	■	■	■
Model Number		-	■	■	■	■	■	■	■	■	■	■
Net Capacity												
Refrigeration Capacity	ton_R	4	■	■	■	■	■	■		■	■	■
Heat Rejection Capacity	Btu/h	4	■	■			■		■	■	■	■
Heat Recovery Capacity	Btu/h	4		■						■		
Efficiency												
Cooling EER	Btu/W·h	4										
Cooling COP	kW/kW	4	■	■	■	■	■	■		■	■	■
Cooling kW/ton_R	kW/ton_R	4										
Heating COP	kW/kW	4							■			■
Heat Recovery COP	kW/kW	4		■						■		
	Btu/W·h	4										
IPLV.IP/NPLV.IP	kW/kW	4	■		■	■	■	■			■	
	kW/ton_R	4										
Power												
Total Power	kW	4	■	■	■	■	■	■	■	■	■	■
Condenser Spray Pump Power *[optional]*	kW	4			■							
Fan Power *[optional]*	kW	4			■	■		■	■	■		
Cooling Mode Evaporator												
Entering Water[1]	ºF	Note 4	■	■	■	■	■	■	■	■	■	■
Leaving Water[1]	ºF	Note 4	■	■	■	■	■	■	■	■	■	■
Flow	gpm	4	■	■	■	■	■	■	■	■	■	■
Water Pressure Drop	ft H_2O	3	■	■	■	■	■	■	■	■	■	■
Fouling Factor	$h\cdot ft^2\cdot ºF/Btu$	3	■	■	■	■	■	■	■	■	■	■
Cooling Mode Heat Rejection Exchanger												
Tower Condenser												
Entering Water[1]	ºF	Note 4	■	■								
Leaving Water[1]	ºF	Note 4	■	■								
Flow	gpm	4	■	■								
Water Pressure Drop	ft H_2O	3	■	■								
Fouling Factor	$h\cdot ft^2\cdot ºF/Btu$	3	■	■								

Table 14. Published Values

Published Values	Units	Significant Figures[3]	Water-Cooled Chiller (Cooling)	Water-Cooled Heat Recovery Chiller	Evaporatively Cooled Chiller	Air-Cooled Chiller	Condenserless Chiller	Air-Cooled HP (Cooling)	Air-Cooled HP (Heating)	Air Cooled Heat Recovery Chiller	Water to Water HP (Cooling)	Water to Water HP (Heating)
Heat Recovery Condenser												
Entering Water[1]	ºF	Note 4		■						■		
Leaving Water[1]	ºF	Note 4		■						■		
Flow	gpm	4		■						■		
Water Pressure Drop	ft H_2O	3		■						■		
Fouling Factor	h·ft^2·ºF/Btu	3		■						■		
Dry-bulb air	ºF	Note 4								■		
Heat Rejection Condenser												
Entering Water[1]	ºF	Note 4									■	■
Leaving Water[1]	ºF	Note 4									■	■
Flow	gpm	4									■	■
Water Pressure Drop	ft H_2O	3									■	■
Fouling Factor	h·ft^2·ºF/Btu	3									■	■
Evaporatively Cooled												
Dry-bulb	ºF	Note 4			■							
Wet-bulb	ºF	Note 4			■							
Altitude[2]	ft	3			■							
Air Cooled												
Dry-bulb	ºF	Note 4				■		■	■	■		
Wet-bulb	ºF	Note 4							■			
Altitude[2]	ft	3				■		■	■	■		
Without Condenser												
Saturated Discharge	ºF	Note 4					■					
Liquid Temperature or Subcooling	ºF	Note 4					■					

Notes:

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
2. Altitude based on standard atmosphere; refer to Section 7 for conversion to atmospheric pressure.
3. Published Ratings and final reported test values shall be rounded to the number of significant figures shown in this table.
4. Commonly used units of measure for temperature are not on an absolute scale; however, proper use of significant figures requires an absolute scale. For simplicity, this standard will specify the number of decimal places as follows:
   a. Water temperatures round to two decimal places.
   b. Refrigerant temperatures (actual or saturated) round to one decimal place.
   c. Air temperatures round to one decimal place.

Section 7. Conversions and Calculations

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

Table 15. Conversion Factors[1, 2]

To Convert From	Factor Name	To	Multiply By
1 ft H_2O (at 60ºF)	K1	psi	0.43310
inch Hg (at 32ºF)	K2	psia	0.49115
kilowatt (kW)	K3	Btu/h	3412.14
watt (W)	K4	Btu/h	3.41214
ton of refrigeration (ton_R)	K5	Btu/h	12,000
ton of refrigeration (ton_R)	K6	kilowatt (kW)	3.51685
kilowatt (kW)	K7	watt (W)	1,000
MBtu/h	K8	Btu/h	1,000,000
$\frac{lb_f}{in^2} \cdot \frac{ft^3}{lb_m}$	K9	$\frac{Btu}{lb_m}$	0.18505
cubic feet per hour $\left(\frac{ft^3}{h}\right)$	K10	Gallon per minute (gpm)	0.124675

Notes:

1. For Water Pressure Drop, the conversion from water column "ft H_2O" to "psi" is per ASHRAE Fundamentals Handbook. 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*. One of the following calculation methods shall be utilized. In both cases, the value of the water temperature or pressure to be used as input is dependent on the context of the calculation using the density and specific heat terms.

Method 1. Use NIST REFPROP software (version 9.1 or later) to calculate physical properties density and specific heat, as a function of both pressure and temperature.

Method 2. Use the following polynomial equations to calculate density and specific heat of water as a function of temperature only.

$$\rho = (\rho_4 \cdot T^4) + (\rho_3 \cdot T^3) + (\rho_2 \cdot T^2) + (\rho_1 \cdot T) + \rho_0 \tag{27}$$

$$c_p = \left(c_{p5} \cdot T^5\right) + \left(c_{p4} \cdot T^4\right) + \left(c_{p3} \cdot T^3\right) + \left(c_{p2} \cdot T^2\right) + \left(c_{p1} \cdot T\right) + c_{p0} \tag{28}$$

	IP (°F)
T	32~212

	IP (lbm/ft³)
-	
ρ_4	$-7.4704 \cdot 10^{-10}$
ρ_3	$5.2643 \cdot 10^{-7}$
ρ_2	$-1.8846 \cdot 10^{-4}$
ρ_1	$1.2164 \cdot 10^{-2}$
ρ_0	62.227

	IP (Btu/lbm·°F)
c_{p5}	$-4.0739 \cdot 10^{-13}$
c_{p4}	$3.1031 \cdot 10^{-10}$
c_{p3}	$-9.2501 \cdot 10^{-8}$
c_{p2}	$1.4071 \cdot 10^{-5}$
c_{p1}	$-1.0677 \cdot 10^{-3}$
c_{p0}	1.0295

Note*:* Density and specific heat polynomial equations are curve fit from data generated by NIST REFPROP v9.1 (see Normative Appendix A) at 100 psia and using a temperature range of 32 ºF 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.

7.3 *Converting Altitude to Atmospheric Pressure*. The relationship is based on the International Standard Atmosphere (ISA) and represents a mean value of typical weather variations. The ISA is defined by ICAO Document 7488/3. The slight difference between geometric altitude (Z_H) and geopotential altitude (H) is ignored for the purposes of this standard ($Z_H \cong H$).

$$p_{atm} = p_0 \cdot \left[\frac{T_0}{T_0 + \beta \cdot (Z_H - Z_{H0})}\right]^{\left(\frac{g_0}{\beta} \cdot \frac{M_0}{R*}\right)} \tag{29}$$

Where:

$\beta = -0.00198 \dfrac{K}{ft}$

$Z_{H0} = 0 ft$

$g_0 = 9.80665\ m/s^2$

$M_0 = 28.96442\ kg/kmol$

$R* = 8314.32\ J/(K \cdot kmol)$

$p_0 = 14.696\ psia$

Section 8. Symbols and Subscripts

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

Table 16. Symbols and Subscripts

Symbol		Description	Unit Name	Unit Symbol
A	=	Efficiency at 100% load. COP, EER, or kW/ton$_R$ depending on use.		varies
A_w	=	Heat transfer surface area used in fouling factor adjustment calculations (water-side), as used in Appendix C.	square foot	ft^2
A_Q	=	Correction factor, capacity, polynomial equation coefficient	dimensionless	-
A_η	=	Correction factor, efficiency, polynomial equation coefficient	dimensionless	-
B	=	Efficiency at 75% load. COP, EER, or kW/Ton depending on use.		varies
B_Q	=	Correction factor, capacity, polynomial equation coefficient	dimensionless	-
B_η	=	Correction factor, efficiency, polynomial equation coefficient	dimensionless	-
C	=	Efficiency at 50% load. COP, EER, or kW/Ton depending on use.		varies
c	=	Y-axis intercept (offset) of the regression line when used in the form of $\hat{y}=m\hat{x}+c$		varies
C_D	=	Degradation factor	dimensionless	-
CF_Q	=	Atmospheric correction factor for capacity	dimensionless	-
CF_η	=	Atmospheric correction factor for efficiency	dimensionless	-
COP_H	=	Efficiency, coefficient of performance, heating	dimensionless	-
COP_{HR}	=	Efficiency, coefficient of performance, heat recovery	dimensionless	-
COP_R	=	Efficiency, coefficient of performance, cooling	dimensionless	-
$COP_{R,CD}$	=	Efficiency, coefficient of performance, cooling, corrected with degradation factor	dimensionless	-
$COP_{R,Test}$	=	Efficiency, coefficient of performance, cooling, test value	dimensionless	-
COP_{SHC}	=	Efficiency, coefficient of performance, simultaneous cooling & heating	dimensionless	-
c_p	=	Specific heat at constant pressure	British thermal unit (IT) per pound degree Fahrenheit	Btu/(lb·°F)
C_Q	=	Correction factor, capacity, polynomial equation coefficient	dimensionless	-
C_η	=	Correction factor, efficiency, polynomial equation coefficient	dimensionless	-
CWH	=	Cooling water hours	degree Fahrenheit hour	°F·h
D	=	Efficiency at 25% load. COP, EER, or kW/Ton depending on use.		varies
d	=	Pipe inside diameter dimension	foot	ft
DBH	=	Dry bulb hours	degree Fahrenheit hour	°F·h

Table 16. Symbols and Subscripts (continued)

Symbol		Description	Unit Name	Unit Symbol
D_Q	=	Correction factor, capacity, equation term	dimensionless	-
D_η	=	Correction factor, efficiency, equation term	dimensionless	-
e	=	Base of natural logarithm; Euler's number; a mathematic constant	dimensionless	-
E_{bal}	=	Energy balance	dimensionless	-
EDB		Entering Dry Bulb Temperature	degree Fahrenheit	°F
EER	=	Efficiency, energy efficiency ratio	Btu per watt hour	Btu/(W·h)
EER_{CD}	=	Efficiency, energy efficiency ratio, corrected with degradation factor	Btu per watt hour	Btu/(W·h)
EER_{Test}	=	Efficiency, energy efficiency ratio, test value	Btu per watt hour	Btu/(W·h)
E_{in}		Energy into a system	British thermal unit (IT) per hour	Btu/h
E_{out}		Energy out of a system	British thermal unit (IT) per hour	Btu/h
EWB	=	Entering wet bulb temperature	degree Fahrenheit	°F
EWT	=	Entering Water Temperature	degree Fahrenheit	°F
f	=	Darcy friction factor	dimensionless	-
%FS	=	Percent of full scale for the measurement instrument or measurement system	dimensionless	-
%RDG	=	Percent of reading for the measurement instrument or measurement system	dimensionless	-
g	=	Standard gravitational term	foot per second squared	ft/s^2
h_L	=	Head loss (pressure drop, pressure differential)	foot	ft
H	=	Geopotential altitude (see section 7.3)	foot	ft
IPLV	=	Efficiency, Integrated Part Load Value. kW/Ton, COP, or EER		varies
IPLV.IP	=	Efficiency, Integrated Part Load Value when calculated and reported in accordance with AHRI Standard 550/590 in IP units. COP, EER, or kW/Ton		varies
K	=	Resistance coefficient used in water side pressure drop calculations	dimensionless	-
K1 – K10	=	Refer to Table 15		varies
kW/ton_R	=	Efficiency, power input per capacity	kilowatt per ton of refrigeration	kW/ton_R
$\left(\frac{kW}{ton_R}\right)_{CD}$	=	Efficiency, power input per capacity, corrected with degradation factor	kilowatt per ton of refrigeration	kW/ton_R
$\left(\frac{kW}{ton_R}\right)_{Test}$	=	Efficiency, power input per capacity, test value	kilowatt per ton of refrigeration	kW/ton_R

Table 16. Symbols and Subscripts (continued)				
Symbol		Description	Unit Name	Unit Symbol
L_F	=	Load factor	dimensionless	-
L_H, L_W, L_1, L_2… $L_{overall}$,	=	Length dimensions (as used in Appendix E)	foot	ft
LWT	=	Leaving Water Temperature	degree Fahrenheit	°F
m_w	=	Mass flow rate, water	pound (avoirdupois) per hour	lb/h
m	=	Slope of the regression line when used in the form of $\hat{y}=m\hat{x}+c$		varies
$\bar{n}$	=	Average rotational speed (compressor)	revolution per minute	RPM
n	=	Number of calibration data points	dimensionless	-
NPLV	=	Efficiency, Non-standard Part Load Value. kW/Ton, COP, or EER depending on use.		varies
NPLV.IP	=	Efficiency, Non-standard Part Load Value when calculated and reported in accordance with AHRI Standard 550/590 in IP units. kW/Ton, COP, or EER depending on use.		varies
p	=	Pressure	pound-force per square inch	psia
p_{atm}	=	Atmospheric pressure	pound-force per square inch	psia
p_0	=	Standard atmospheric pressure	pound-force per square inch	psia
PI	=	Prediction Interval		varies
p_{ring}	=	Pressure, piezometer ring	pound-force per square inch	psia
Q	=	Capacity (heat flow rate); net capacity	Btu per hour	Btu/h
Q'	=	Capacity (heat flow rate); gross capacity	Btu per hour	Btu/h
Q_{cd}	=	Net capacity, condenser (heating)	Btu per hour	Btu/h
Q'_{cd}	=	Gross capacity, condenser (heating)	Btu per hour	Btu/h
Q_{ev}	=	Net capacity, evaporator (cooling)	Btu per hour	Btu/h
Q'_{ev}	=	Gross capacity, evaporator (cooling)	Btu per hour	Btu/h
Q_{hrc}	=	Net capacity, condenser (heat recovery)	Btu per hour	Btu/h
Q'_{hrc}	=	Gross capacity, condenser (heat recovery)	Btu per hour	Btu/h
$Q_{ev\%Load}$	=	Test results for unit net capacity at test point	Btu per hour	Btu/h
Q_{test}	=	Test results for net capacity, uncorrected for atmospheric pressure	Btu per hour	Btu/h
$Q_{corrected,standard}$	=	Test results for net capacity, corrected to standard atmospheric pressure	Btu per hour	Btu/h

Table 16. Symbols and Subscripts

Symbol		Description	Unit Name	Unit Symbol
$Q_{corrected,application}$	=	Test results for net capacity, corrected to application rating atmospheric pressure	Btu per hour	Btu/h
$Q_{ev100\%}$	=	Test results for unit net capacity at 100% load point	Btu per hour	Btu/h
$Q_{ev\%Load}$	=	Test results for unit net capacity at any part load test point	Btu per hour	Btu/h
r	=	Radius of the centerline of the elbow	foot	ft
Re	=	Reynolds number	dimensionless	-
R_{foul}	=	Fouling factor allowance	hour-foot squared-degree Fahrenheit per Btu	$h \cdot ft^2 \cdot °F/Btu$
s	=	Standard deviation of a sample from a population		varies
SDT	=	Saturated Discharge Temperature	degree Fahrenheit	°F
SS_x	=	Sum of squares of x value differences to the mean		varies
s_t	=	Standard deviation of temperature measurement samples		varies
s_{vw}	=	Standard deviation of volumetric water flow measurement samples		varies
s_ε	=	standard error of estimate		varies
T	=	Temperature	degree Fahrenheit	°F
t	=	Time. Hour, minute, second depending on use.		varies
T_{in}	=	Entering water temperature	degree Fahrenheit	°F
$T_{in,cd}$	=	Entering condenser water temperature	degree Fahrenheit	°F
$T_{in,DB}$	=	Entering dry bulb temperature	degree Fahrenheit	°F
$T_{in,w}$	=	Temperature, water entering	degree Fahrenheit	°F
$T_{in,WB}$	=	Entering wet bulb temperature	degree Fahrenheit	°F
$T_{liq,r}$	=	Liquid refrigerant temperature	degree Fahrenheit	°F
$T_{MC,WB}$	=	Mean coincident wet bulb	degree Fahrenheit	°F
$T_{OA,DB}$	=	Outdoor air dry bulb	degree Fahrenheit	°F
Tol_1	=	Tolerance 1, performance tolerance limit	dimensionless	-
Tol_2	=	Tolerance 2, IPLV and NPLV performance tolerance limit	dimensionless	-
Tol_3	=	Tolerance 3, Tolerance on water side pressure drop	foot of water	ft H_2O (at 60°F)
Tol_4		Tolerance 4, energy balance validity tolerance limit	dimensionless	-
T_{out}	=	Leaving water temperature	degree Fahrenheit	°F
$T_{out,w}$	=	Temperature, water leaving	degree Fahrenheit	°F
T_{sat}	=	Temperature, saturated refrigerant	degree Fahrenheit	°F
$T_{sat,disch}$	=	Saturated discharge temperature	degree Fahrenheit	°F

Table 16. Symbols and Subscripts (continued)

Symbol		Description	Unit Name	Unit Symbol
$T_{sat,disch,AC}$	=	Saturated discharge temperature	degree Fahrenheit	°F
$T_{sat,disch,EC}$	=	Saturated discharge temperature	degree Fahrenheit	°F
$T_{sat,disch,WC}$	=	Saturated discharge temperature	degree Fahrenheit	°F
$T_{WB,mean}$	=	Mean condenser wet bulb temperature	degree Fahrenheit	°F
$t_{\alpha/2,n-2}$	=	Critical value of Student's t distribution, at confidence level α/2 and degrees of freedom n-2	dimensionless	-
V_w	=	Volumetric flow rate	gallon per minute	gpm
V	=	Voltage	volt	V
v	=	Velocity	foot per second	ft/s
W	=	Power	kilowatt	kW
W_{input}	=	Power, total input power	kilowatt	kW
W_{refrig}	=	Power, total of compressor work and auxiliary devices transferring energy into the refrigerant	kilowatt	kW
W_{shaft}	=	Power, mechanical power input to the compressor shaft	kilowatt	kW
W_{gear}	=	Power, friction loss in an external gear box	kilowatt	kW
$W_{prime\ mover}$	=	Power, output of prime mover	kilowatt	kW
x	=	Variable representing any measurement value (corrected indicated value)		varies
$\hat{x}$	=	Any value of x at which to evaluate the curve fit and prediction interval		varies
$\bar{x}$	=	Mean of all corrected indicated values		varies
y	=	Reference standard value		varies
$\hat{y}$	=	Linear regression curve fit of the calibration data evaluated at $\hat{x}$		varies
Z	=	Equation term (as used in Appendix C)	dimensionless	-
Z_H	=	Geometric altitude	foot	ft
Δp	=	Pressure differential	pound force per square inch	psid
Δt	=	Time interval	second	s
ΔT	=	Temperature differential	degree Fahrenheit	°F
ΔT_{adj}	=	Temperature differential, additional temperature differential due to fouling	degree Fahrenheit	°F
$\Delta T_{adj,weighted}$	=	Weighted temperature adjustment	degree Fahrenheit	°F
ΔT_{FL}	=	Temperature differential, at full load design conditions	degree Fahrenheit	°F
ΔT_{iLMTD}	=	Incremental log mean temperature difference	degree Fahrenheit	°F

Table 16. Symbols and Subscripts (continued)

Symbol		Description	Unit Name	Unit Symbol
ΔT_{LMTD}	=	Log mean temperature difference	degree Fahrenheit	°F
ΔT_{range}	=	Temperature differential when referenced to entering and leaving heat exchanger fluid temperatures	degree Fahrenheit	°F
ΔT_{small}	=	Temperature differential when calculating LMTD	degree Fahrenheit	°F
$\Delta T_{small,clean}$	=	Small temperature difference as tested in clean condition	degree Fahrenheit	°F
$\Delta T_{small,sp}$	=	Small temperature difference as specified	degree Fahrenheit	°F
ε	=	Absolute roughness	foot	ft
η		Efficiency (EER) for atmospheric pressure correction.	Btu per watt hour	Btu/(W·h)
$\eta_{test, FL}$		Efficiency (EER) measured in Full Load test, for atmospheric pressure correction.	Btu per watt hour	Btu/(W·h)
ρ	=	Density	pound (avoirdupois) per cubic foot	lb/ft^3
ω	=	Frequency (electrical)	Hertz	Hz

Section 9. Marking and Nameplate Data

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

9.1.1 Manufacturer's name and location
9.1.2 Model number designation providing performance-essential identification
9.1.3 Refrigerant designation (in accordance with ANSI/ASHRAE Standard 34)
9.1.4 Refrigerant Safety Group Classification (in accordance with ANSI/ASHRAE Standard 34)
9.1.5 Voltage, phase and frequency
9.1.6 Serial number

9.2 *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 10. Conformance Conditions

10.1 *Conformance.* While conformance with this standard is voluntary, conformance shall not be claimed or implied for products or equipment within 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 (SI)-2015, *Performance Rating of Water-Chilling and Heat Pump Water-Heating Packages Using the Vapor Compression Cycle*, 2015, Air-Conditioning, Heating and Refrigeration Institute, 2111 Wilson Boulevard, Suite 500, Arlington, VA 22201, U.S.A.

A1.2 ANSI/AHRI Standard 110-2012, *Air-Conditioning, Heating and Refrigerating Equipment Nameplate Voltages*, 2012, Air-Conditioning, Heating and Refrigeration Institute, 2111 Wilson Boulevard, Suite 500, Arlington, VA 22201, U.S.A.

A1.3 ANSI/ASHRAE Standard 34-2013 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*, 2013, American Society of Heating, Refrigeration, and Air-Conditioning Engineers, Inc., 2013, ASHRAE, 1791 Tullie Circle, N.E., Atlanta, Georgia, 30329, U.S.A.

A1.5 ASHRAE Standard 41.1-2013, *Measurements Guide - Section on Temperature Measurements*, 2013, 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, *Terminology*, https://www.ashrae.org/resources--publications/free-resources/ashrae-terminology, 2014, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., 1791 Tullie Circle, N.E., Atlanta, GA 30329, U.S.A.

A1.7 ASHRAE/ANSI/AHRI/ISO Standard 13256-2:1998 (RA 2012), *Water-to-Water and Brine-to-Water Heat Pumps – Testing and Rating for Performance*, 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 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.

A1.9 ASME Standard PTC 19.5-2004, *Flow Measurement, 2004,* 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-16-2014, Measurement of Liquid Flow in Closed Conduits With Electromagnetic Flowmeters, 2014, American Society of Mechanical Engineers. ASME, Three Park Avenue, New York, NY 10016, U.S.A.

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APPENDIX B. REFERENCES – INFORMATIVE

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B.9 Excel Spreadsheet for the Computation of the Water 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.

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

C1 *Purpose*. This appendix prescribes 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 steady-state 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*. Use definitions in Section 3.

C3 *Calculations.* This section includes required methods for determining water properties, data processing, performance calculations (capacity, power, efficiency, and corrections), test validation metrics, unit of measure conversions, and rules for rounding numbers.

C3.1 *Water Properties.* Calculate density and specific heat per equations in Section 7.

C3.2 *Data Processing.* Data point measurements collected during the duration of the testing period shall be processed to calculate sample mean and sample standard deviation per the following equations. Calculate final performance metrics (Capacity, Efficiency, Water Pressure Drop) and other test results (Energy Balance, Voltage Balance) from the mean values of measurement data (this method of test is not intended for transient testing).

C3.2.1 *Sample Mean.*

$$\bar{x} = \frac{1}{n}\sum_{j=1}^{n}\left(x_j\right) \qquad \text{C1}$$

C3.2.2 *Sample Standard Deviation*.

$$s = \sqrt{\frac{1}{n-1}\sum_{j=1}^{n}\left(x_j - \bar{x}\right)^2} \qquad \text{C2}$$

C3.3 *Performance.*

C3.3.1 *Capacity*. To provide increased accuracy for energy balance calculations, the energy associated with pressure loss across the heat exchanger is included in the equation for gross capacity. This formulation closely approximates 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.

The Gross (Q'_{ev}) and Net (Q_{ev}) Refrigerating Capacity of the evaporator, and the Gross (Q'_{cd}) and Net (Q_{cd}) Heating Capacity of the condenser or heat recovery condenser, Btu/h, shall be obtained by the following equations:

$$\mathrm{Q}'_{\mathrm{ev}} = m_W \cdot \left[\mathrm{c_p} \cdot (\mathrm{T_{in}} - \mathrm{T_{out}}) + \frac{\mathrm{K9} \cdot \Delta \mathrm{p_{corrected}}}{\rho}\right] \qquad \text{C3}$$

For Q_{ev}, refer to Equation 7.

$$Q'_{cd} = m_W \cdot \left[c_p \cdot (T_{out} - T_{in}) - \frac{K9 \cdot \Delta p_{corrected}}{\rho}\right] \qquad C4$$

$$Q'_{hrc} = m_W \cdot \left[c_p \cdot (T_{out} - T_{in}) - \frac{K9 \cdot \Delta p_{corrected}}{\rho}\right] \qquad C5$$

For Q_{cd}, refer to Equation 9.

For Q_{hrc}, refer to Equation 10.

$$m_W = \frac{V_w \cdot \rho}{K10} \qquad C6$$

Where:

1) The values for specific heat of water, and for water density in the term $\frac{\Delta p}{\rho}$, are evaluated at the average of inlet and outlet temperatures.
2) The value for Water Pressure Drop of the heat exchanger uses the final test result $\Delta p_{corrected}$ after adjustment per Appendix G.
3) If measuring volumetric flow rate, convert to mass flow using density ρ required in Section C4.1.3.
4) Capacity using the unit of measure, ton_R, to be calculated using conversion factor K5.

For reference, the conversion factor K9 in Section 7 was derived as follows:

$$K9 = 0.18505 = \frac{144\ in^2}{ft^2} \cdot \frac{Btu}{778.17\ ft \cdot lb_f}$$

$$\frac{Btu}{778.17\ ft \cdot lb_f} = \frac{1\ Btu}{1055.05585262\ J} \cdot \frac{1\ J}{1\ N \cdot m} \cdot \frac{0.3048\ m}{1\ ft} \cdot \frac{4.4482216152605\ N}{1\ lb_f}$$

C3.3.2 *Power.*

C3.3.2.1 For use in efficiency calculations, determine the chiller Total Input Power, by summation including compressor and all auxiliary power requirements. See Section C4.1.5 for detailed requirements regarding measurement locations and power values that shall be included.

$$W_{input} = \sum_{j=1}^{n} (W_j) \qquad C7$$

C3.3.2.2 For use in energy balance calculations, determine the portion of the Total Input Power that is transferred into the refrigerant circuit and rejected through the condenser. See Sections C3.4.1 and C4.5.1 for detailed requirements regarding which power values shall be included.

$$W_{refrig} = \sum_{j=1}^{n} (W_j) \qquad C8$$

C3.3.3 *Efficiency.* Calculate efficiency per equations in Section 5.1.

C3.3.4 *Corrections.* This section defines fouling factor related adjustments to target temperature values, as well as corrections to test measurements for water-side pressure drop, and corrections to test results for atmospheric pressure.

C3.3.4.1 *Method for Simulating Fouling Factor Allowance.* The calculations in this section apply to evaporators and condensers using water, for full load and part load operating conditions. The resultant fouling factor correction, ΔT_{adj}, is added or subtracted to the target test water temperature as appropriate to simulate the fouled condition.

$$\Delta T_{range} = \left|T_{out,w} - T_{in,w}\right| \qquad C9$$

$$\Delta T_{small} = \left| T_{sat,r} - T_{out,w} \right| \quad \text{C10}$$

Where $T_{sat,r}$ is the saturated vapor temperature for single component or azeotrope refrigerants, or for zeotropic refrigerants T_{sat} is the arithmetic average of the Dew Point and Bubble Point temperatures, corresponding to refrigerant pressure.

Calculate the log mean temperature difference (ΔT_{LMTD}) for the evaporator and/or condenser using the following equation at the Fouling Factor Allowance (R_{foul}) specified by the rated performance, and the corresponding specified small temperature difference, $\Delta T_{small,sp}$.

$$\Delta T_{LMTD} = \frac{\Delta T_{range}}{\ln\left(1 + \frac{\Delta T_{range}}{\Delta T_{small,sp}}\right)} \quad \text{C11}$$

Calculate the incremental log mean temperature difference (ΔT_{ILMTD}) using the following equation:

$$\Delta T_{ILMTD} = R_{foul}\left(\frac{Q}{A_w}\right) \quad \text{C12}$$

Where Q is the rated net capacity and A_w is the water-side heat transfer surface area for the heat exchanger, which may be inside or outside surface area depending on the heat exchanger design.

The water temperature adjustment needed to simulate the additional fouling, ΔT_{adj}, can now be calculated:

$$Z = \frac{\Delta T_{range}}{\Delta T_{LMTD} - \Delta T_{ILMTD}} \quad \text{C13}$$

$$\Delta T_{small,clean} = \frac{\Delta T_{range}}{e^{Z} - 1} \quad \text{C14}$$

$$\Delta T_{adj} = \Delta T_{small,sp} - \Delta T_{small,clean} \quad \text{C15}$$

Where $\Delta T_{small,sp}$ is the small temperature difference as rated at a specified fouling factor allowance, and $\Delta T_{small,clean}$ is the small temperature difference as rated in a clean condition with no fouling.

The calculation of ΔT_{adj} is used for both evaporator and condenser water temperature corrections. The correcting water temperature difference, ΔT_{adj}, is then added to the condenser entering water temperature or subtracted from the evaporator leaving water temperature to simulate the additional Fouling Factor.

C3.3.4.1.1 *Special Consideration for Multiple 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 $\Delta T_{adj,i}$ 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 $\Delta T_{adj,i}$ values shall be computed as follows:

$$\Delta T_{adj,weighted} = \frac{\sum(Q_i \cdot \Delta T_{adj,i})}{\sum(Q_i)} \quad \text{C16}$$

Where:

Q_i = Heat transfer rate for each heat exchanger
$\Delta T_{adj,i}$= Computed temperature adjustment for each heat exchanger

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

C3.3.4.1.2 *Example - Condenser Fouling Inside Tubes*.

Specified Fouling Factor Allowance, R_{foul} = 0.000250 h·ft²·°F/Btu
Condenser load, Q_{cd} = 2,880,000 Btu/h
Specified Condenser leaving water temp, $T_{out,w}$ = 95.00°F
Specified Condenser entering water temp, $T_{in,w}$ = 85.00 °F
Specified saturated condensing temperature with specified fouling, $T_{sat,r}$ = 101.00 °F
Inside tube surface area, A = 550 ft² (since fouling is inside tubes in this example)

$$\Delta T_{range} = \left|T_{out,w} - T_{in,w}\right| = 95.00 - 85.00 = 10.00°F$$

$$\Delta T_{small,sp} = \left|T_{sat,r} - T_{out,w}\right| = 101.00 - 95.00 = 6.00°F$$

$$\Delta T_{LMTD} = \frac{\Delta T_{range}}{\ln\left(1 + \frac{\Delta T_{range}}{\Delta T_{small,sp}}\right)} = \frac{10.00}{\ln\left(1 + \frac{10.00}{6.00}\right)} = 10.1955°F$$

$$\Delta T_{ILMTD} = R_{foul}\left(\frac{Q_{cd}}{A_w}\right) = 0.000250 \cdot \left(\frac{2{,}880{,}000}{550}\right) = 1.30909°F$$

$$Z = \frac{\Delta T_{range}}{\Delta T_{LMTD} - \Delta T_{ILMTD}} = \frac{10.00}{10.1955 - 1.30909} = 1.12532$$

$$\Delta T_{small,clean} = \frac{\Delta T_{range}}{e^{Z} - 1} = \frac{10.00}{e^{1.12532} - 1} = 4.80°F$$

$$\Delta T_{adj} = \Delta T_{small,sp} - \Delta T_{small,clean} = 6.00 - 4.80 = 1.20°F$$

The entering condenser water temperature for testing is then raised 1.20°F to simulate the Fouling Factor Allowance of 0.000250 h·ft²·°F/Btu. The entering condenser water temperature will be:

$$T_{in,adj} = 85.00 + 1.20 = 86.20°F$$

C3.3.4.1.3 *Derivation of LMTD.*

This derivation is included for reference only:

$$\Delta T_{LMTD} = \frac{\left(T_{sat,r} - T_{in,w}\right) - \left(T_{sat,r} - T_{out,w}\right)}{\ln\left[\frac{T_{sat,r} - T_{in,w}}{T_{sat,r} - T_{out,w}}\right]} = \frac{\left(T_{out,w} - T_{in,w}\right)}{\ln\left[\frac{\left(T_{sat,r} - T_{out,w}\right) + \left(T_{out,w} - T_{in,w}\right)}{T_{sat,r} - T_{out,w}}\right]}$$

$$= \frac{\left(T_{out,w} - T_{in,w}\right)}{\ln\left[1 + \frac{\left(T_{out,w} - T_{in,w}\right)}{T_{sat,r} - T_{out,w}}\right]} = \frac{\Delta T_{range}}{\ln\left(1 + \frac{\Delta T_{range}}{\Delta T_{small}}\right)} \qquad \text{C17}$$

C3.3.4.2 *Pressure Drop.* Determine the pressure drop adjustment per Appendix G and calculate the corrected Water Pressure Drop as follows:

$$\Delta p_{corrected} = \Delta p_{test} - \Delta p_{adj} \qquad \text{C18}$$

C3.3.4.3 *Atmospheric Pressure.* Calculate corrected capacity and efficiency per Appendix F.

C3.4 *Validation.* Test results are validated by checking an energy balance and a voltage balance, calculated as follows.

C3.4.1 *Energy Balance.* Based on the first law of thermodynamics, or the law of conservation of energy, an energy balance calculation evaluates all of the measured energy flow into and out of a control volume. If there is a non-zero difference between energy flow in and energy flow out, greater than the energy balance measurement uncertainty, then either (a) the system is not at steady state (lack of equilibrium), or (b) some significant heat gain or heat loss has been omitted from the calculation, or (c) there is a measurement error to be corrected. The control volume shall include the entire chiller package, especially the refrigerant circuit(s) that convey thermal energy from a source to a sink (from the evaporator water to the condenser water). In many cases, heat losses or heat gain caused by radiation, convection, bearing friction, oil coolers, etc., are relatively small and may be either included or excluded without a problem in the overall energy balance.

Gross capacity shall be used for energy balance calculations.

Sum all energy sources flowing in and out of the system through the control volume boundary:

$$E_{in} = \sum_i E_{in_i} \qquad \text{C19}$$

$$E_{out} = \sum_i E_{out_i} \qquad \text{C20}$$

The general energy balance equation is expressed as a percentage:

$$E_{bal} = \frac{E_{in} - E_{out}}{avg(E_{in}, E_{out})} \cdot 100\% = 2\frac{E_{in} - E_{out}}{E_{in} + E_{out}} \cdot 100\% \qquad \text{C21}$$

For water-cooled chillers, the total measured input power (W_{input}) to the chiller package is often assumed to equal the compressor work done on the refrigerant (W_{refrig}). In cases where the difference in the total power and the compressor work is significant, an analysis that provides a calculated value of W_{refrig} shall be performed and used in the energy balance equation. Cases for different chiller configurations are shown in Sections C3.4.1.1 through C3.4.1.5.

C3.4.1.1 A typical summation omitting the effect of the small heat losses and gains mentioned above.

$$E_{in} = \sum_i E_{in_i} = Q'_{ev} + (W_{refrig}) \cdot K3$$

$$E_{out} = \sum_i E_{out_i} = Q'_{cd} + Q'_{hrc}$$

C3.4.1.2 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, so W_{refrig} shall be assumed to equal W_{input} as electrical power input to the compressor motor terminals, plus any significant auxiliary input power transferred to the refrigerant.

C3.4.1.3 In a package using an open-type compressor with prime mover and external gear drive W_{refrig} shall be assumed to equal shaft input power (W_{shaft}), plus any significant auxiliary input power transferred to the refrigerant. W_{shaft} is typically the prime mover output power ($W_{prime\ mover}$) less the gear losses (W_{gear}).

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

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

C3.4.1.4 In a package using an open-type compressor with prime mover and internal gear W_{refrig} shall be assumed equal to shaft input power (W_{shaft}), plus any auxiliary input power transferred to the refrigerant.

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

C3.4.2 *Voltage Balance.* Voltage balance is defined as the maximum absolute value of the voltage deviation from the average voltage, expressed in relative terms as a percentage of the average voltage. For a three-phase system with three measured voltages the equations are:

$$V_{bal} = \frac{\max[|V_1 - V_{avg}|, |V_2 - V_{avg}|, |V_3 - V_{avg}|]}{V_{avg}} \times 100\% \qquad \text{C22}$$

$$V_{avg} = \frac{(V_1 + V_2 + V_3)}{3} \qquad \text{C23}$$

C3.5 *Conversions.* Calculations and reporting of test results using other units of measure shall use the conversion factors defined in Section 7.

C3.6 *Rounding.* Reported measurement data and calculated test results shall follow the rounding requirements in Section 4.3, making use of the significant figure requirements of Table 14.

C4. *Test Requirements.* This section defines requirements for instrumentation (accuracy, installation, calibration), test plan, tolerances when conducting a test, corrections and adjustments to make, and test validation.

C4.1 *Instrumentation.* This section defines requirements for each type of measurement (temperature, flow, pressure, power). Instruments shall be selected, installed, operated, and maintained according to the requirements of Table C1. Further details are provided in this section for each measurement type.

Table C1. Requirements for Test Instrumentation

Measurement	Measurement System Accuracy[2,3,4,5]	Display Resolution[6, 7]	Selected, Installed, Operated, Maintained in Accordance With
Liquid Temperature	±0.20°F	0.01°F	ANSI/ASHRAE Standard 41.1
Air Temperature	±0.20°F	0.1°F	ANSI/ASHRAE Standard 41.1
Liquid Mass Flow Rate [1]	±1.0% RDG	4 significant figures	ASME Power Test Code PTC 19.5 (flow measurement) ASME MFC-16 (electromagnetic type) ASME MFC-3M (orifice & venturi type) ASME MFC-6M (vortex type) ASME MFC-11 (coriolis type) ISA Standard RP31.1 (turbine type)
Differential Pressure	±1.0% RDG	3 significant figures	ASME Power Test Code PTC 19.2
Electrical Power ≤ 600V > 600 V	 ±1.0% FS, ±2.0% RDG ±1.5% FS, ±2.5% RDG	4 significant figures (V, A, kW, Hz)	IEEE 120 IEEE C57.13 -2008
Atmospheric Pressure	±0.15 psia	0.01 psia	ASME Power Test Code PTC 19.2
Steam condensate mass flow rate	±1.0% RDG	4 significant figures	
Steam pressure	±1.0% RDG	3 significant figures	
Fuel volumetric flow rate	±1.0% RDG	4 significant figures	
Fuel energy content	-	3 significant figures	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. Accuracy requirement also applies to volumetric type meters.
2. Measurement system accuracy shall apply over the range of use during testing, as indicated by the Turn Down Ratio determined during calibration, i.e. from full scale down to a value of full scale divided by the Turn Down Ratio. For many types of instruments and/or systems this may require exceeding the accuracy requirement at full scale.
3. %RDG = percent of Reading, %FS = percent of Full Scale for the useable range of the measurement instrument or measurement system.
4. If dual requirements are shown in the table, FS and RDG, then both requirements shall be met.
5. Current Transformers (CT's) and Potential Transformers (PT's) shall have a metering accuracy class of 0.3 or better, rated in accordance with IEEE C57.13-1993 (R2003).
6. Display resolution shown is the minimum requirement (most coarse resolution allowable). Better (finer) resolution is acceptable for instrument or panel displays, or computer screen displays.
7. Significant figures (also known as significant digits) determined in accordance with Section 4.3.

C4.1.1 *Accuracy and Calibration.*

C4.1.1.1 All instruments and measurement systems shall be calibrated over a range that meets or 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.

Note: 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.

C4.1.1.2 For each instrument device in a measurement system, the calibration process shall identify the range over which the required accuracy can be achieved (specified accuracy from Table C1). This range shall be documented in a readily accessible format for verification (such as a manual of calibration records, or instrument labeling system, or work instructions for test facility operators). Many types of instruments have a usable range or Turn Down Ratio of 10:1, though some types are quite different. Differential pressure type flow meters may be limited to 3:1 range of flow (due to a differential pressure measurement range of 10:1). Some types of instruments, such as electromagnetic and coriolis type flow meters, or current transformers with low burden, may be capable of wider ranges such as 20:1 or more.

To determine the range over which the calibration achieves the required accuracy, a linear regression analysis is performed on the calibration data. The data is plotted to show the residual errors versus the calibration reference standard. The standard error of estimate shall be calculated for the measurement system indicated values (post calibration) versus the calibration reference standard, then using Equation C24 plot a 95% prediction interval (α=5%) on both sides of the curve fit. The point(s) at which the prediction interval curve exceeds the required accuracy shall be the limit(s) of the range. Table C2 and the equations that follow explain the method of calculating the prediction interval. See example using sample data in Figures C1 and C2, in which the specified accuracy is ±1% of reading, and the useable range is from 100 to 13.4, or Turn Down Ratio of 7.5:1.

All test point readings (i.e. at any percent load, or at any operating test condition) shall be within the calibration range or Turn Down Ratio for each instrument device measurement. For a given type of measurement, multiple instruments may be required to cover a wide range of testing conditions for a given test facility, or a range of Water-Chilling or Water-Heating Package sizes. In the case of multiple instruments, procedures and protocols shall be established by the test facility for use by test operators regarding when and how to switch between instruments.

C4.1.1.3 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). Water chilling or heating packages 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. 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 C2. Prediction Interval to Determine Range of Acceptable Accuracy

	Reference Standard Value [1] y_j j=1 to n	Corrected (As Left) Indicated Value [2] x_j j=1 to n	Absolute Prediction Interval of Indicated Value	Relative Prediction Interval of Indicated Value %RDG
Calibration Data	y_1	x_1	$x_1 - \hat{y} \pm PI(x_1)$	$\frac{x_1 - \hat{y} \pm PI(x_1)}{x_1}$
	y_2	x_2	$x_2 - \hat{y} \pm PI(x_2)$	$\frac{x_1 - \hat{y} \pm PI(x_2)}{x_2}$
	y_3	x_3	$x_3 - \hat{y} \pm PI(x_3)$	$\frac{x_1 - \hat{y} \pm PI(x_3)}{x_3}$
	…	…	…	…
	y_n	x_n	$x_n - \hat{y} \pm PI(x_n)$	$\frac{x_1 - \hat{y} \pm PI(x_n)}{x_n}$
Regression Statistics	$\bar{x}$ SS_x	s_ε	continuous curve $\hat{x} - \hat{y} \pm PI(\hat{x})$ varying $\hat{x}$ from min to max values of x_j	continuous curve $\frac{\hat{x} - \hat{y} \pm PI(\hat{x})}{\hat{x}}$ varying $\hat{x}$ from min to max values of x_j

Notes:
1. Reference Standard Value is the actual value determined or measured by the calibration standard.
2. Corrected Indicated Value is the value of the measured quantity given directly by a measuring system on the basis of its calibration curve ("as left" when the calibration process has been completed, not "as found" at the beginning of the calibration process).

$$PI(\hat{x}) = s_\varepsilon \cdot t_{\frac{\alpha}{2},n-2} \cdot \sqrt{1 + \frac{1}{n} + \frac{(\hat{x} - \bar{x})^2}{SS_x}} \qquad \text{C24}$$

$$\bar{x} = \frac{1}{n}\sum_{j=1}^{n}\left(x_j\right) \qquad \text{C25}$$

$$SS_x = \sum_{j=1}^{n}\left(x_j - \bar{x}\right)^2 \qquad \text{C26}$$

$$s_\varepsilon = \sqrt{\frac{\sum_{j=1}^{n}\left(y_j - mx_j - c\right)^2}{n-2}} \qquad \text{C27}$$

$$m = \frac{n\sum_{j=1}^{n} x_j y_j - \sum_{j=1}^{n} x_j \sum_{j=1}^{n} y_j}{n\sum_{j=1}^{n}\left(x_j\right)^2 - \left(\sum_{j=1}^{n} x_j\right)^2} \qquad \text{C28}$$

$$c = \frac{\sum_{j=1}^{n}\left(x_j^2\right)\sum_{j=1}^{n} y_j - \sum_{j=1}^{n} x_j \sum_{j=1}^{n}\left(x_j y_j\right)}{n\sum_{j=1}^{n}\left(x_j^2\right) - \left(\sum_{j=1}^{n} x_j\right)^2} \quad \text{C29}$$

$$\hat{y} = m \cdot \hat{x} + c \quad \text{C30}$$

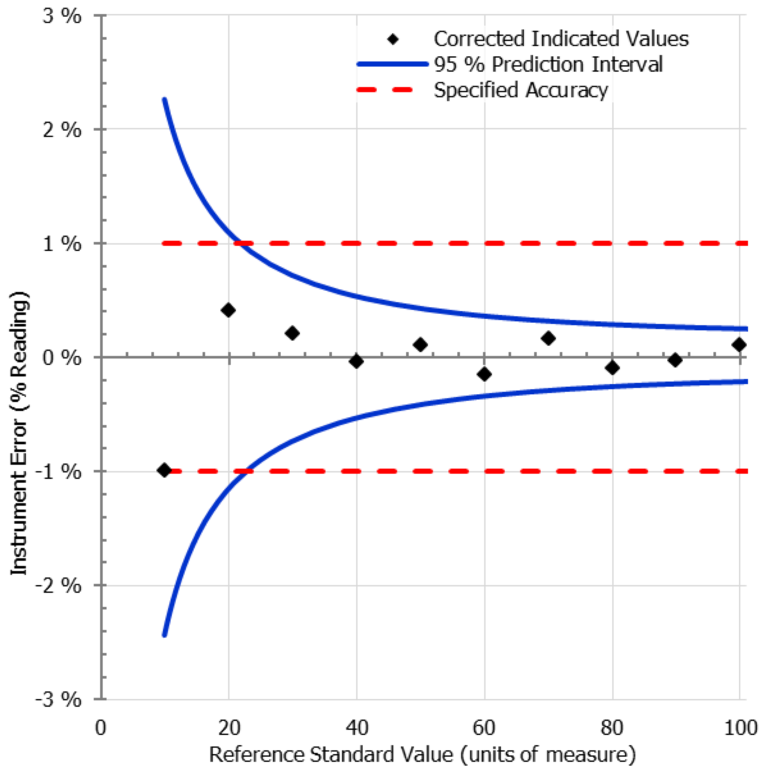


Figure C1. Sample of Relative Calibration Evaluation Data (Percent of Reading)

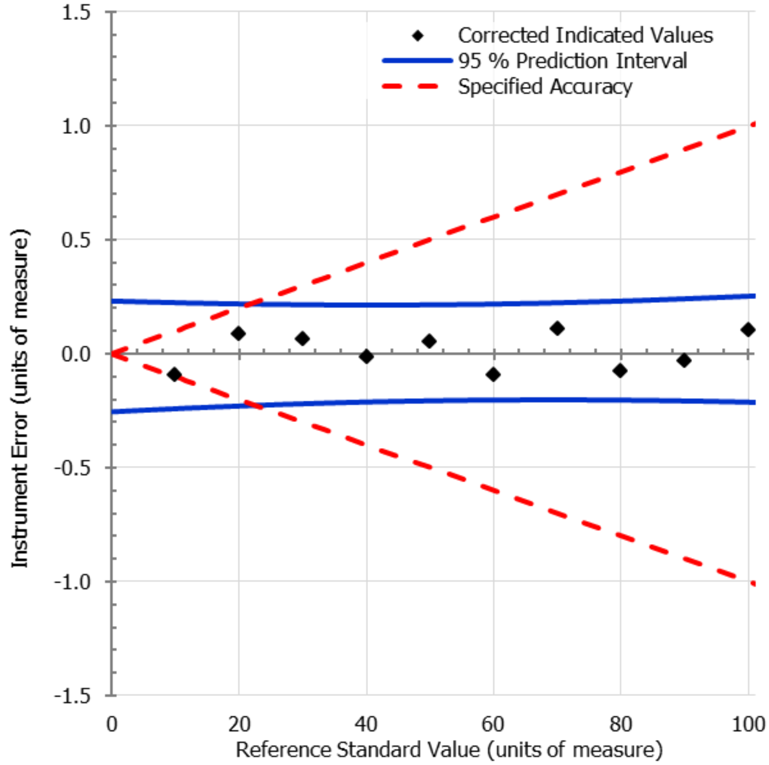


Figure C2. Sample of Absolute Calibration Evaluation Data

C4.1.2 *Temperature.*

C4.1.2.1 *Water.* Measure entering and leaving water temperatures, °F. Temperature sensor(s) shall be installed in a location that represents the average bulk fluid temperature.

Note: Non-mandatory but recommended practices to consider, especially if troubleshooting problems with energy balance. When deciding where to locate water temperature measurement sensors, consider the mixing effects of the piping configuration which may vary considerably across a range of flow rates. Check the spatial variation of temperature across a range of flow rates within a single plane perpendicular to the pipe, either with a movable (traversing) sensor or with multiple stationary sensors. If necessary add flow conditioners or mixers.

C4.1.2.1.1 Units with an optional integrated evaporator or condenser water pump shall be tested in either of the following two configurations.

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.

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.

C4.1.2.1.2 If evaporator or condenser water is used to add or remove heat to or from any other source(s) within the package, the temperature measurements shall be made at points so that the measurements reflect the Gross Capacity.

C4.1.2.2 *Air.* For chillers with either an air-cooled or Evaporatively-cooled Condenser, measure entering air temperature per Appendix E.

C4.1.3 *Flow.* Measure water mass flow rate, lbm/hr, or calculate mass flow rate from measured volumetric flow rate using Equation C6. If a volumetric flow meter was used for the measurement, the conversion to mass flow shall use the density corresponding to either of the following locations: (1) the temperature of the water at the location of flow meter; or (2) the water temperature measurement, either entering or leaving, which best represents the temperature at the flow meter.

C4.1.3.1 If volumetric flow rate is measured on the leaving side, then the target volumetric flow rate shall be adjusted to match the published ratings which are based on the entering side (see Sections 5.1 and 6.2), in order to have the same target mass flow rate.

C4.1.3.2 If evaporator or condenser water is used to add or remove heat to or from any other source(s) within the package, the flow measurement(s) shall be made at points so that the measurements reflect the Gross Capacity.

C4.1.4 *Pressure.*

Measure Water Pressure Drop across the heat exchanger, psid.

Static pressure taps shall be located per Appendix G. Depending on the design of the chiller water connections, Appendix G may or may not require additional piping external to the unit for accurate measurements. External piping for measurement purposes creates additional line losses between the static pressure tap and the unit connections. These additional losses are calculated and then subtracted from the average test measurement value as an adjustment method to obtain the reported test result for Water Pressure Drop across the unit connections. Appendix G specifies the calculation method for adjustment factors.

For units containing an integrated water pump, the measured Water 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.

C4.1.5 *Power.* Power is the rate at which work is performed or energy is converted in electrical and mechanical systems. The total input power including auxiliary power shall be measured, by summation of measurements at one or more locations defined below. Auxiliary power shall include those devices active during normal operation of the package; intermittent auxiliary power shall be reflected in the data points defined in Section C6, providing a time-averaged value over the duration of the test time period. For Water-cooled chillers, it will be necessary in some cases to make separate power measurements to segregate values and obtain valid energy balance.

Electrical measurements include voltage (for each phase), current (for each phase), power, and frequency (from a minimum of one phase). For units with a dual nameplate voltage rating, testing shall be performed at the lower of the two voltages. 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 of the test facility, and shall minimize losses due to cabling from the measurement location to the connection point on the chiller.

C4.1.5.1 For Air-cooled or Evaporatively-cooled Condensers, the test shall include the condenser fan power and Condenser spray pump power in the measurement(s) of total input power.

C4.1.5.2 For packages containing optional integrated water pumps, for evaporator or condenser, the test measurements shall exclude the pump power from the measurement of chiller input power (refer to Section 6.2.1.4). 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.

C4.1.5.3 *Power Measurement Considerations for Energy Balance.* Input power that enters the refrigerant side of the circuit impacts the energy balance calculation. This includes compressor shaft input power, and other refrigerant-cooled devices such as hermetic or semi-hermetic motors, variable speed drives, oil coolers, etc. If the energy balance validation criteria cannot be met, one possible cause is a significant quantity of input power lost to the ambient environment for air-cooled devices such as open-drive motors, starters, variable speed drives, oil coolers, controls, etc. If this is the case, then provide separate power measurement locations such that energy balance can be more accurately determined while also meeting the requirement to measure total input power to calculate efficiency.

C4.1.5.4 *Electric Drive.* The input power shall be determined by measurement of electrical input to the chiller.

C4.1.5.4.1 For electric-drive packages rated with starters, transformers, gearboxes, or variable speed drives, whether self-contained or remote-mounted (free-standing), the input power shall include the power losses due to those components and shall be measured on the input (line side). In the case of remote-mounted (free-standing) devices, substitute devices may be used during testing provided that they have similar power losses (within ±0.5%) and speed control method as the device supplied to the customer.

C4.1.5.4.2 For electric-drive packages not rated with a starter or variable speed drive (provided by others per Section 6.2.1.4.4), input power shall be measured as close as practical to the compressor motor terminals. For the case of such packages rated for variable speed operation, a variable speed control method and variable speed drive type consistent with the chiller manufacturer installation requirements shall be used for the test.

C4.1.5.4.3 When testing a chiller package that was rated with a motor supplied by others, the compressor shaft input power shall be measured by either of the following two methods:

Torque meter and rotational speed sensor installed between the compressor and another test motor used conduct the test. Power is calculated from torque multiplied times speed.

Calibrated test motor with dynamometer or similar test data to determine the relationship between input and output power at the required load points. Electrical input power to the test motor is measured, then the corresponding output power determined from calibration relationship.

C4.1.5.5 *Non-electric Drive.* Where turbine, engine drive, or other prime mover is employed, the total input power, and compressor shaft input power for energy balance calculations, shall be determined from steam or fuel consumption, at measured supply and exhaust conditions and prime mover manufacturer's certified performance data. The total input power shall include the losses due to the prime mover and other driveline components such as a gearbox.

C4.2 *Plan.* A test plan shall document the all requirements for conducting the test. This includes a list of the required full load and part load test points, the published ratings and associated operating conditions, including adjusted water temperature targets based on the rated Fouling Factor Allowance.

C4.3 *Tolerances.*

C4.3.1 *Operating Conditions.* Operating condition tolerances are defined to control two characteristics. The first is deviation of the mean value relative to the target value. The second is the stability, which is defined in statistical terms and allows excursions from the target that are brief in time and/or small in magnitude. Over the time period of each test point, the operating conditions shall be controlled to maintain the mean and standard deviation within the tolerances defined in Table 12.

C4.3.2 *Performance.* Performance tolerances are used when testing to validate published ratings in accordance with Section 4.1, and are applicable to full load capacity, and also applicable to efficiency and water pressure drop at both full and part load. The performance test results shall meet the tolerances defined in Table 11.

C4.4 *Corrections.* The following corrections shall be applied to test targets or test results when applicable.

C4.4.1 *Fouling Factor.* Target water temperatures shall be adjusted per the calculation method of Section C3.3.4.1, to simulate the effect of fouling at the Fouling Factor Allowance ($R_{foul,sp}$.) that was specified in the published rating. Testing shall be conducted with clean tubes (see Sections 5.5.1 and C6.1.1) and shall assume Fouling Factors to be zero ($R_{foul} = 0.00\overline{0}$ h·ft^2·°F/Btu).

C4.4.2 *Pressure Drop.* To account for measurement methods, the measured values of water pressure drop during testing shall be corrected per the methods defined in Section C3.3.4.2 and Appendix G prior to reporting the final test result for Water Pressure Drop.

C4.4.3 *Atmospheric Pressure.* For comparison to published ratings, whether at standard rating conditions or application rating conditions, units with air-cooled or evaporatively-cooled condenser type shall correct capacity and efficiency test results as defined in Appendix F such that the reported test results correspond to the standard atmospheric pressure.

C4.5 *Validation.*

C4.5.1 *Energy Balance.*

For the case of Water-cooled Condensers, measurement data shall be collected to calculate an energy balance (per Section C3.4.1) to substantiate the validity of each test point. Test validity tolerance for energy balance is found in Table 13. The energy balance (%) shall be within the allowable tolerance calculated per Section 5.6 for the applicable conditions.

For Air-cooled and Evaporatively-cooled Condensers, it is impractical to measure heat rejection in a test and an energy balance cannot be readily calculated. To validate test accuracy, concurrent redundant instrumentation method per Section C4.5.2 shall be used to measure water temperatures, flow rates, and power inputs.

For heat recovery 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 in Section C4.5.2 shall be used.

For heat recovery units with Water-cooled Condensers that fully condense the refrigerant, the energy balance method shall be used.

If evaporator water is used to remove heat from any other source(s) within the package, the temperature, pressure drop, and flow measurements of chilled water shall be made at points so that the measurements reflect the Gross Refrigerating Capacity.

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

C4.5.2 *Concurrent Redundant Instrumentation.*

For the case of Air-cooled or Evaporatively-cooled Condensers, or Air-source Evaporators for Heating Mode, redundant measurement data shall be collected to substantiate the validity of each test point.

C4.5.2.1 *Measurement Verification*: Redundant instrument measurements shall be within the limitations below:

C4.5.2.1.1 Entering water temperature measurements shall not differ by more than 0.20°F

C4.5.2.1.2 Leaving water temperature measurements shall not differ by more than 0.20°F

C4.5.2.1.3 Flow measurements shall not differ by more than 2%

C4.5.2.1.4 Power input measurements shall not differ by more than 2%

C4.5.2.2 *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.

C4.5.2.3 *Example Calculation for Capacity with Concurrent Redundant Verification.*

$T_{in,1}$ = 54.10°F, $T_{in,2}$ = 53.91°F (difference of 0.19°F is acceptable)
$T_{out,1}$ = 44.10°F, $T_{out,2}$ = 43.90°F (difference 0.20°F is acceptable)
$m_{w,1}$ = 101020 lbm/h, $m_{w,2}$ = 99080 lbm/h (difference of 1.94% is acceptable)

$T_{in,avg}$ = 54.005°F
$T_{out,avg}$ = 44.000°F
$m_{w,avg}$ = 100050 lbm/h

Properties of water are calculated per Section 7.2 as follows:
c_p = 1.0018 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 · 1.0018 Btu/lbm·°F · (54.005°F – 44.000°F)
Q_{ev} = 83.62 ton_R

C4.5.3 *Voltage Balance*. Multi-phase power supply with significant voltage unbalance can impact equipment operation and efficiency. Calculate voltage balance per Section C3.4.2. Test validity tolerance is found in Table 13. Voltage balance is not applicable to single phase units.

C4.6 *Uncertainty Analysis.* This standard does not require an uncertainty analysis for measurements and test results. It is recommended practice to perform uncertainty analysis following the procedures in ASME PTC 19.1 *Test Uncertainty*.

C5. *Data Collection.*

C5.1 *Primary Data.* The table below summarizes the data to be recorded during the test for each of the data point samples.

Table C3. Data to be Recorded During the Test

Type		Data Item	Units of Measure
All Condenser Types	General	Time of day for each data point sample	hh:mm:ss.s
		Atmospheric pressure	psia
	Evaporator	T_{in}	°F
		T_{out}	°F
		m_w or V_w	lb/h or gpm
		Δp_{test}	psid
Water-cooled Condenser	Condenser	T_{in}	°F
Water-cooled Heat Recovery Condenser		T_{out}	°F
		m_w or V_w	lb/h or gpm
		Δp_{test}	psid
Air-cooled Condenser	Condenser	Spatial average dry-bulb temperature of entering air	°F
Evaporatively-cooled Condenser	Condenser	Spatial average dry-bulb temperature of entering air	°F
		Spatial average wet-bulb temperature of entering air	°F
Without Condenser	Compressor	Discharge temperature	°F
		Discharge pressure	psia
	Liquid Line	Liquid refrigerant temperature entering the expansion device	°F
		Liquid pressure entering the expansion device	psia
Electric Drive	Chiller	W_{input} (and W_{refrig} if needed)	kW
		Voltage for each phase	V
		If 3-phase: average voltage	V
		Frequency for one phase	Hz
Non-Electric Drive	Chiller	W_{input} (and W_{refrig} if needed)	kW
		If Steam Turbine: Steam consumption Steam supply pressure Steam supply temperature Steam exhaust pressure	 lb/h psig ºF psia
		If Gas Turbine or Gas Engine: fuel consumption (natural gas or propane) Higher Heating Value	 ft^3/h Btu/ ft^3
		If Internal Combustion Engine: liquid fuel consumption (diesel or gasoline) Higher Heating Value	 gal/h Btu/lb

C5.2 *Auxiliary Data.* The table below summarizes the auxiliary data that shall be recorded for the test.

Table C4. Auxiliary Data to be Recorded

Type	Data Item	Units of Measure
All	Date, place, and time of test	dd-mmm-yyyy hh:mm:ss
	Names of test supervisor and witnessing personnel	-
	Ambient temperature at test site	°F
	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.	-
	Prime mover nameplate data (motor, engine or turbine).	-
Non-electric Drive	Fuel specification (if applicable) and calorific value	-

The table below summarizes optional auxiliary data (non-mandatory) that may be recorded during the test for diagnostic information.

Table C5. Optional Auxiliary Data to be Recorded		
Type	Data Item	Units of Measure
Open-type compressor	Compressor driver rotational speed	rpm
Electric Drive	Current for each phase of electrical input to chiller package	amp

C6. *Test Procedures.* For each test point at a specific load and set of operating conditions, the test will measure capacity, input power, and water-side pressure drop. Capacity, a measurement of the heat added to or removed from the water as it passes through the heat exchanger, may be cooling, heating, heat recovery, and/or heat recovery according to the test plan. Net capacity is always required, and gross capacity is required when an energy balance requirement applies. Each test point will collect multiple data points versus time. The test shall use instrumentation meeting the requirements in Section C4 and calculations in Section C3.

C6.1 *Setup.* The chiller package to be tested shall be setup at the test facility in accordance with the manufacturer's instructions, including but not limited to support of installation mounting points, connections for water, connections for power supply, test instrumentation, charging of refrigerant or oil, etc. Non-condensable gases if present shall be removed from the system.

C6.1.1 *Condition of Heat Transfer Surfaces.* The as tested Fouling Factors shall be assumed to be zero ($R_{foul} = 0.00\overline{0}$ $h \cdot ft^2 \cdot °F/Btu$). Tests conducted in accordance with this standard may require cleaning of the heat transfer surfaces (in accordance with manufacturer's instructions) prior to conducting the test.

C6.2 *Operation.* After setup is complete, the chiller will be started and operated to attain the target conditions of the test point per the test plan. The chiller is not required to operate continuously between different test points, shut down and re-start between test points is allowable.

C6.2.1 *General.*

Steady-state operating conditions and performance shall be maintained for a minimum test time period of 15 minutes, such that measurement parameters, associated standard deviations, and test results are within both the operating condition tolerances and test tolerances set forth in Section 5.6. If not within tolerance due to lack of stability, then continue testing until within tolerance. If not within tolerance but stability is acceptable, then stop testing to investigate and resolve instrumentation problems, then repeat the test. Resolving problems may require new calibration of instrumentation.

To minimize the effects of transient conditions, all measurement types should be taken as simultaneously as possible (flow, temperature, power, etc.). Software or other recording methods shall be used to capture time-stamped data points over the duration of the test time period. A minimum of 30 data point measurements shall be collected and recorded for each parameter at uniform time intervals. Intervals between time stamps shall not vary more than ±5% from the average time interval for all data points. Each data point measurement may represent either an individual reading from the measurement system, or a time averaged value from a larger number of data samples. In the case of using time averaging, whether in hardware or software, the time interval for averaging of data samples shall not exceed 1/60 of the total test time period. There is no limit on sampling rate and various time averaging methods may be employed.

Measurement values include temperatures, flow rates, differential pressure, power, voltage, fuel or steam consumption, and atmospheric pressure. Calculate the average and standard deviation for each measurement value.

Test results include Net Capacity, Efficiency, and Water Pressure Drop (corresponding to certification program published rating values). Calculate test results using the mean of the test measurement values. Capacity may be calculated for each data point for purposes of test facility control, but the final result for capacity shall be calculated from the mean of all measurement values.

C6.2.2 *Heat Pump.* Heating mode or heat recovery tests of chillers with Air-cooled or Evaporatively-cooled Condensers shall use the test procedure in Appendix H.

C6.3 *Adjustments.*

C6.3.1 *Controls.* Manual operation of chiller controls is allowed to avoid cycling and disruption of test stability.

C6.3.1 *Refrigerant.* Refrigerant charge may be adjusted during setup, prior to conducting the test, in accordance with manufacturer's instructions which may require operation of the chiller. Refrigerant charge quantity shall be held constant for the duration of the test, including all test points in a series of full load and part load tests.

C7 *Reporting of Results.* A written or electronic test report shall be generated including the following items for each test point at a specific load and set of operating conditions.

C7.1 *Data.* Report the test time period and number of data point measurements. Include the sample mean and sample standard deviation for each measurement value (temperature, flow, pressure drop, power, etc.). Data point measurements shall be made available upon request.

C7.2 *Calculations.* Report the correction adjustment values Δp_{adj} and ΔT_{adj}, correction factors CF_Q and $CF\eta$ when applicable, and associated input data used for the correction calculations. Report the density, specific heat capacity, and mass flow values used for capacity calculations. Report all values of Q used in energy balance calculations.

C7.3 *Results.* Report the following test results.

Table C6. Results to be Reported

Item	Units of Measure
Net Capacity (heating and/or cooling as applicable; corrected if applicable)	ton_R or Btu/h
Gross Capacity (heating and cooling, only for water-cooled condenser type)	ton_R or Btu/h
Power Input (W_{input}; and W_{refrig} as applicable)	kW or W
Efficiency (corrected if applicable)	kW/ton_R, EER, or COP
$\Delta p_{corrected}$	ft H_2O (at 60°F)
Energy Balance (if applicable)	%
Voltage Balance	%

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

D1 *Purpose.* This appendix is an informative appendix that has been included in the standard to document the derivation of the Integrated Part-Load Value (IPLV) weighting factors and temperatures.

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 Input Power per Capacity, kW/ton_R

These three alternative ratios are related as follows:

$COP_R = 0.293071$ EER, $\quad$ EER $= 3.41214\ COP_R$

$kW/ton_R = 12/EER$, $\quad$ $EER = 12/(kW/ton_R)$

$kW/ton_R = 3.51685/COP_R$ $\quad$ $COP_R = 3.51685/(kW/ton_R)$

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

$$IPLV = 0.01A + 0.42B + 0.45C + 0.12D \qquad \text{D1}$$

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 Input Power per Capacity, kW/ton_R:

$$\mathrm{IPLV} = \frac{1}{\frac{0.01}{A}+\frac{0.42}{B}+\frac{0.45}{C}+\frac{0.12}{D}} \qquad \text{D2}$$

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:

$$EER_{CD} = \frac{\mathrm{EER_{Test}}}{\mathrm{C_D}} \qquad \text{D3}$$

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:

$$\mathrm{C_D} = (\text{-}0.13 \cdot \mathrm{LF}) + 1.13 \qquad \text{D4}$$

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

$$\mathrm{LF} = \frac{\%\text{Load}\cdot(\text{Full Load unit capacity})}{(\text{Part}-\text{Load unit capacity})} \qquad \text{D5}$$

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 D1 and D2) 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

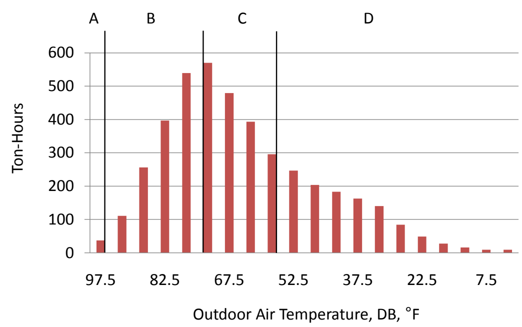


Figure D1. Ton$_R$-Hour Distribution Categories

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).

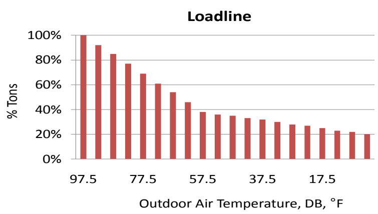
D3.3.5 A weighted average of buildings (with chiller plants only) with and without some form of economizer, based upon data from the DOE and BOMA reports, was included.

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:



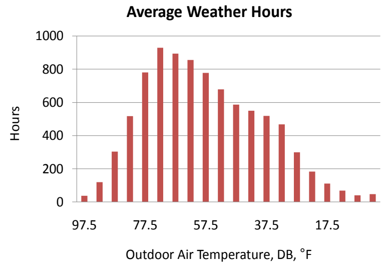


Figure D2. Bin Groupings –Ton$_R$ Hours

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:

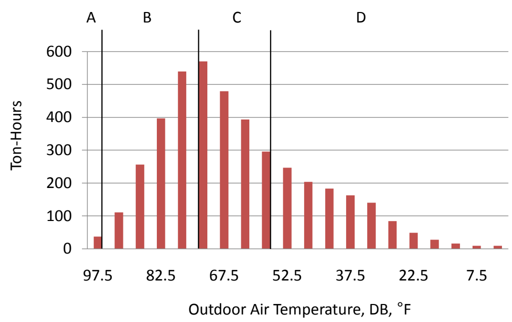


Figure D3. Group 1 Ton$_R$-Hour Distribution Categories

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.

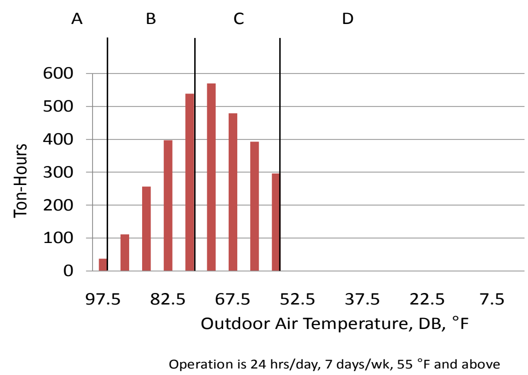


Figure D4. Group 2 Ton$_R$-Hour Distribution Categories

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$_R$):

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

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:

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

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 D2 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	8760	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	8760	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 =	$\frac{1}{0.009/A+0.309/B+0.413/C+0.269/D}$				IPLV =	$\frac{1}{0.012/A+0.423/B+0.565/C+0.0/D}$			
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 =	$\frac{1}{0.015/A+0.409/B+0.392/C+0.184/D}$				IPLV =	$\frac{1}{0.018/A+0.501/B+0.481/C+0.0/D}$			

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
Air Sampling Tree Average Temperature[1]	≤ ±1.0°F	≤ 0.1°F

Notes:
1. If a thermopile is used for this measurement, then the thermocouple wire must have special limits of error and all thermocouple junctions 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
Dry-bulb Temperature		°F
Deviation from the spatially-averaged mean air dry-bulb temperature to the time-averaged air dry-bulb temperature at any individual temperature measurement station[1]	Uniform temperature distribution (between two or more aspirating psychrometers)	±2.00 (≤200 ton_R) ±3.00 (>200 ton_R)
Difference between time-averaged dry-bulb temperature measured with each air sampler thermopile and with time-averaged aspirating psychrometer	Uniform temperature distribution (between two or more air sampler trees on a single aspirating psychrometers)	±1.50
Difference between time-averaged mean dry-bulb air temperature and the specified target test value[2]	Test condition tolerance, for control of air temperature	±1.00
Spatially-averaged mean dry-bulb air temperature variation over time (from the first to the last of the data points)	Test operating tolerance, total observed range of variation over data collection time	±1.00
Wet-bulb Temperature[3]		
Deviation from the Spatially-averaged mean wet-bulb temperature to the time-averaged wet-bulb temperature at any individual temperature measurement station[1]	Uniform humidity distribution (between two or more aspirating psychrometers)	±1.00
Difference between time-averaged mean wet-bulb air wet bulb temperature and the specified target test value[2]	Test condition tolerance, for control of air temperature	±1.00
Spatially-averaged mean wet-bulb air temperature variation over time (from the first to the last of the data points)	Test operating tolerance, total observed range of variation over data collection time	±1.00

Notes:
1. Each measurement station represents a time-averaged value as measured by a single Aspirating Psychrometer.
2. The mean dry-bulb temperature is the mean of all measurement stations using the time-averaged mean value from each aspirating psychrometer.
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. A typical configuration for the sampling tree is shown in Figure E1 for a tree with overall dimensions of 4 feet by 4 feet. 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/s. Average velocity in the holes may be determined by an inverse ratio from the average velocity through in the aspirating psychrometer, using the area through the aspirating psychrometer and the sum of the open area of the holes in the trees. 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 grid to measure the average temperature of the airflow over the sampling tree. The grid shall have at least 16 points per sampling tree, evenly spaced across the sampling tree. The 16 points can be measured by a thermopile wired in parallel or by individual measurement devices. If individual measurement devices are used, then an average will be calculated to determine the air sampling tree temperature. 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.

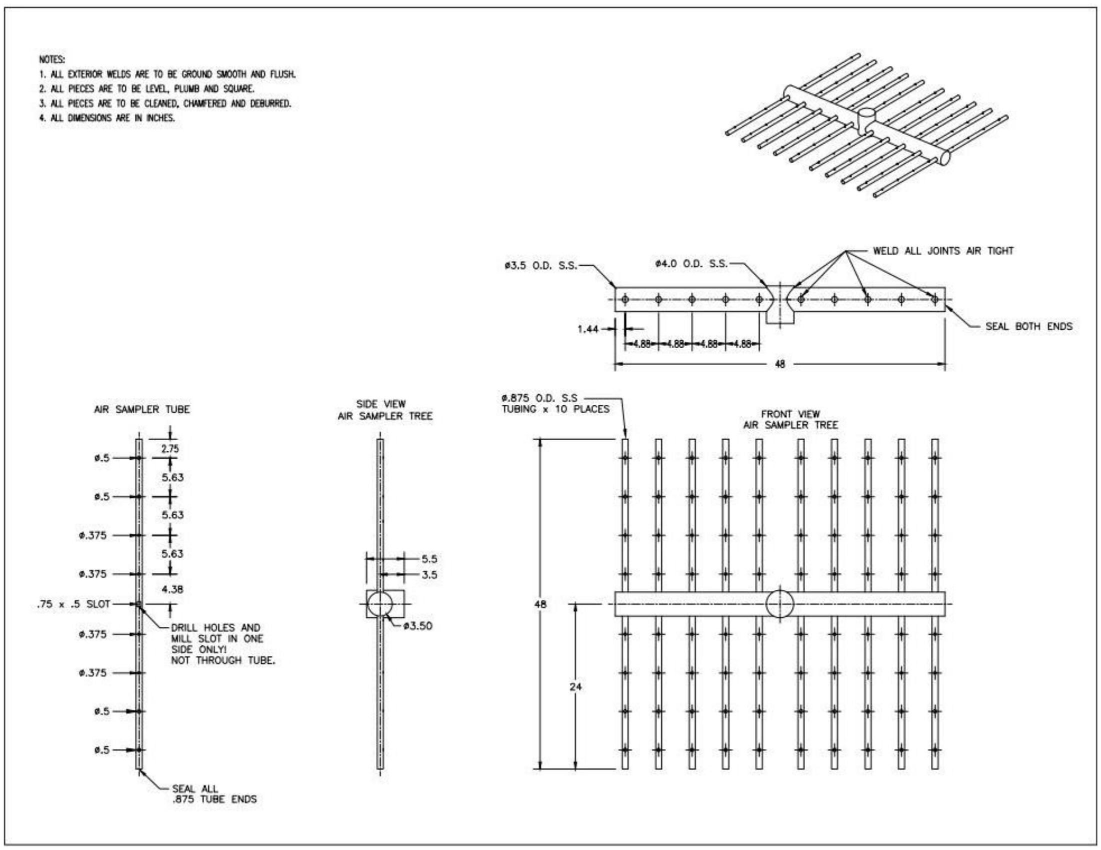


Figure E1. Typical Air Sampling Tree

Note: The 0.75" by 0.50" 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. Drill holes and mill slot shall be on the same side of the sampling tube. The holes and slot shall only pass through the one side of the sampling tube.

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.

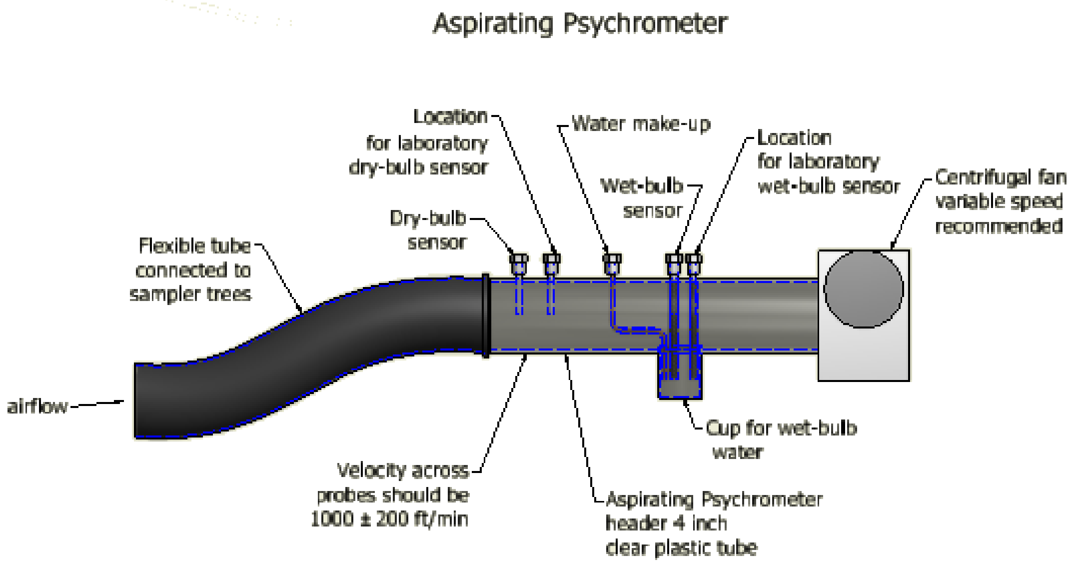


Figure E2. Aspirating Psychrometer

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.

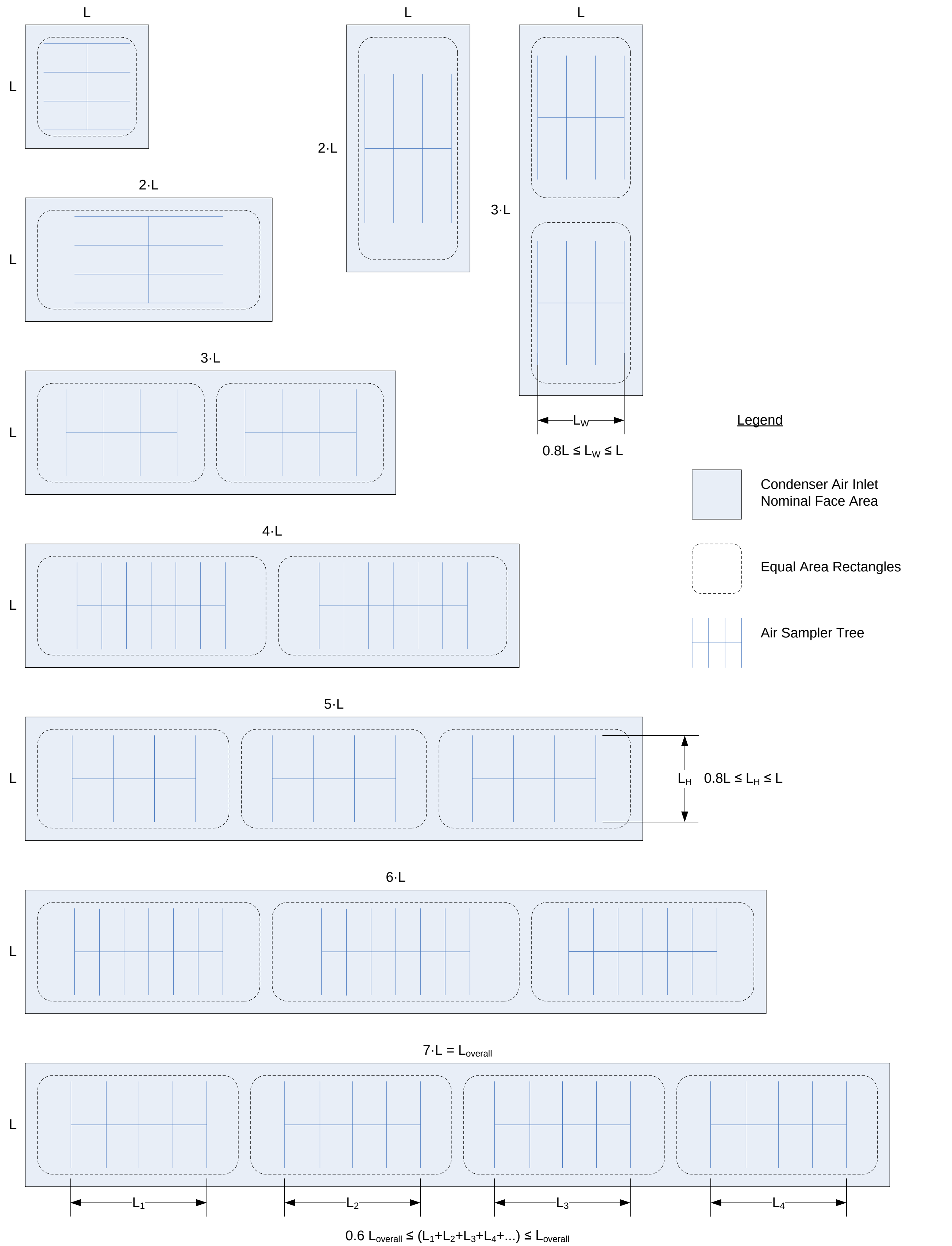


Figure E3. Determination of Measurement Rectangles and Required Number of Air Sampler Trees

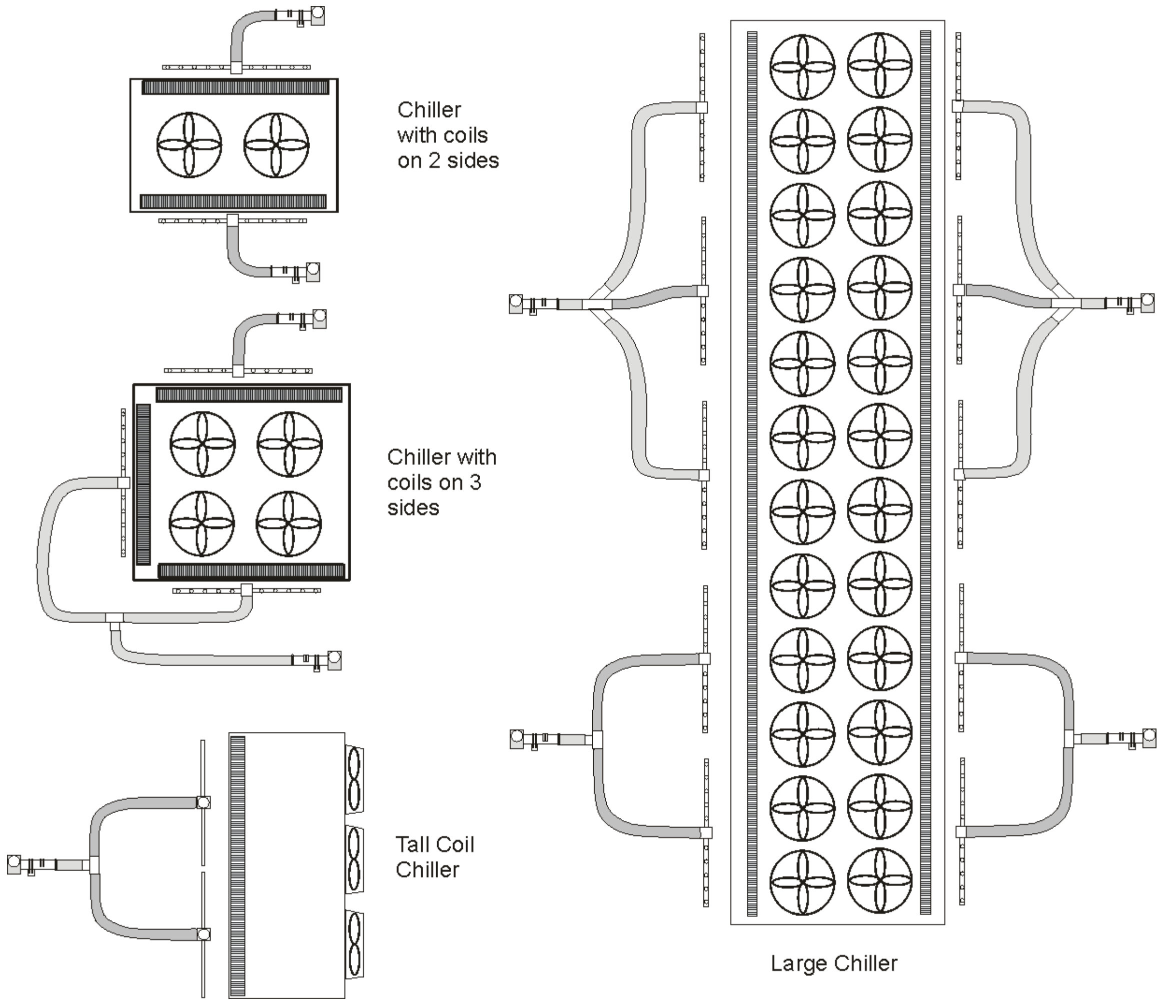


Figure E4. Typical Test Setup Configurations

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 E4 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.

For Part-load test points, aspirating psychrometers positioned at non-operating portions of the coil on the test chiller may be excluded from the calculations.

APPENDIX F. ATMOSPHERIC PRESSURE ADJUSTMENT – NORMATIVE

F1 *Purpose.* The purpose of this appendix is to prescribe a method of adjusting measured test data according to the local atmospheric 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 and evaporatively-cooled chillers shall be corrected for air density variations. To accomplish this, use the following two (2) correction factors (CF_Q, CFη) to correct test data at test load points back to standard atmospheric pressure at sea level, for Standard Rating Conditions, or to correct to another atmospheric pressure corresponding to a site altitude for Application Rating Conditions. 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 from actual tested atmospheric pressure to rated atmospheric pressure 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 calculated at each part-load point where the % load value is based on the measured capacity at that load point divided by the 100% load point measured 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 atmospheric pressure as related to fan, heat exchanger and compressor power and capacity.

The correction factors (CF_Q, CFη) will be limited to a value corresponding to an atmospheric pressure of 11.56 psia (approximately 6500 feet altitude). Correction factors for measured atmospheric pressure readings below the minimum will be equal to the value determined at 11.56 psia.

F3 *Procedure for Correcting Test Data to Standard Rating Condition Atmospheric Pressure.* Air-cooled and evaporatively-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.

$$DQ = AQ \cdot p2 + BQ \cdot p + CQ \tag{F1}$$

$$D\eta = A\eta \cdot p2 + B\eta \cdot p + C\eta \tag{F2}$$

$$\left(CF_Q\right)_{P=P_{test}} = 1 + \left(\frac{Q_{ev,\%Load}}{Q_{ev,100\%}}\right) \cdot \left(D_Q - 1\right) \cdot exp\left\{-0.35 \cdot \left[\left(D_\eta \cdot \eta_{test,100\%}\right) - 9.6\right]\right\} \tag{F3}$$

$$\left(CF_\eta\right)_{P=P_{test}} = 1 + \left(\frac{Q_{ev,\%Load}}{Q_{ev,100\%}}\right) \cdot \left(D_\eta - 1\right) \cdot exp\left\{-0.35 \cdot \left[\left(D_\eta \cdot \eta_{test,100\%}\right) - 9.6\right]\right\} \tag{F4}$$

$$Q_{corrected,standard} = Q_{test}\left(CF_Q\right)_{P=P_{test}} \tag{F5}$$

$$\eta_{corrected,standard} = \eta_{test}\left(CF_\eta\right)_{P=P_{test}} \tag{F6}$$

If efficiency η is expressed in kW/ton_R then divide, Equation F6, by the correction factor CFη instead of multiplying.

Table F1. 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 ton_R and 10.500 EER at sea level. The chiller is tested at an altitude of about 3500 feet.

The measured test results:

Capacity Q_{tested} = 198.5 ton_R
Efficiency η_{tested} = 10.35 EER
Air pressure P_{test} = 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 $CFq = 1 + (198.5 / 198.5) \cdot (1.0170 - 1) \cdot \exp[-0.35 \cdot (1.0386 \cdot 10.35 - 9.6)] = 1.0114$

Correction factor $CF\eta = 1 + (198.5 / 198.5) \cdot (1.0386 - 1) \cdot \exp[-0.35 \cdot (1.0386 \cdot 10.35 - 9.6)] = 1.0258$

Corrected test capacity $Q_{corrected\ standard} = 198.5 \cdot 1.0114 = 200.8$ ton_R

Corrected test efficiency $\eta_{corrected\ standard} = 10.35 \cdot 1.0258 = 10.62$ EER

Part load efficiency and capacity correction factors for the following example are determined using the same calculation process as for the 100% Load Point example:

With a part load test result capacity of 160 ton_R and a 198.5 ton_R test result for 100% load point capacity. The chiller is tested at an altitude of about 3500 feet,

The measured test results for the part load test;

Capacity Q_{test} = 160.00 ton_R
Efficiency $\eta_{tested,100\%}$ = 10.35 EER
Efficiency η_{test} = 12.60 EER
Air Pressure P_{test} = 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 $CFq = 1 + (160.0/198.5) \cdot (1.0170-1) \cdot \exp[-0.35 \cdot (1.0386 \cdot 10.35-9.6)] = 1.0092$

Correction Factor $CF\eta = 1 + (160.0/198.5) \cdot (1.0386-1 \cdot \exp[-0.35 \cdot (1.0386 \cdot 10.35-9.6)] = 1.0208$

Corrected test capacity $Q_{corrected\ standard} = 160.0 \cdot 1.0092 = 161.5$ ton_R

Corrected test efficiency $\eta_{corrected\ standard} = 12.60 \cdot 1.0208 = 12.86$ EER

F4 *Procedure for Correcting Test Data to Application Rating Condition Atmospheric Pressure.* First use the method in Section F3 to correct from tested atmospheric pressure to standard sea level atmospheric pressure. Then reverse the method to correct to the application rated atmospheric pressure P_{rated}.

$$Q_{corrected,application} = \frac{Q_{corrected,standard}}{(CF_Q)_{P=P_{rated}}} \qquad \text{F7}$$

$$\eta_{corrected,application} = \frac{\eta_{corrected,standard}}{(CF_\eta)_{P=P_{rated}}} \qquad \text{F8}$$

100% Load Point Example:

The same chiller from the example in Section F3 also has published application ratings of 199.3 ton_R and 10.42 EER at 1000 feet, corresponding to rated atmospheric pressure of 14.17 psia. The chiller is tested at an altitude of about 3500 feet.

The measured test results:

Capacity Q_{tested} = 198.5 ton_R
Efficiency η_{tested} = 10.35 EER
Air pressure P_{test} = 13.00 psia

From prior example calculations:

Corrected test capacity $Q_{corrected,standard}$ = 200.8 ton_R
Corrected test efficiency $\eta_{corrected,standard}$ = 10.62 EER

Next calculate correction factors for the application rating value of P_{rated}=14.17 psia:

Correction factor $D_Q = 0.0011273 \cdot 14.17^2 - 0.041272 \cdot 14.17 + 1.36304 = 1.0045$

Correction factor $D\eta = 0.0024308 \cdot 14.17^2 - 0.090075 \cdot 14.17 + 1.79872 = 1.0104$

Correction factor $CFq = 1 + (198.5 / 198.5) \cdot (1.0045 - 1) \cdot \exp[-0.35 \cdot (1.0104 \cdot 10.35 - 9.6)] = 1.0034$

Correction factor $CF\eta = 1 + (198.5 / 198.5) \cdot (1.0104 - 1) \cdot \exp[-0.35 \cdot (1.0104 \cdot 10.35 - 9.6)] = 1.0077$

Corrected test capacity $Q_{corrected,application}$ = 200.8 / 1.0034 = 200.1 ton_R

Corrected test efficiency $\eta_{corrected,application}$ = 10.62 / 1.0077 = 10.54 EER

APPENDIX G. WATER PRESSURE DROP MEASUREMENT PROCEDURE – NORMATIVE

G1 *Purpose.* The purpose of this appendix is to prescribe a measurement method for Water Pressure Drop and, when required, a correction method to compensate for friction losses associated with external piping measurement sections. The measurement method only applies to pipe of circular cross section.

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 with acceptable measurement uncertainty. In some cases, the measured Water Pressure Drop per this standard will be determined by using static pressure taps external to the unit in upstream and downstream piping. When using external piping, adjustment factors are allowed to compensate the reported pressure drop measurement. 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 to limit the magnitude of these potential errors. For units with small connection sizes it is feasible that straight pipe sections be directly connected to the units with adequate length to obtain static pressure measurements with acceptable systematic errors due to instrument installation location. This is the preferred connection methodology. Units with larger size connections may have spatial limits in the upstream and downstream connection arrangement such that elbows or pipe diameter changes may be necessary to accommodate the available space at the test facility, or to provide mechanical support for piping weight loads. While this may increase the measurement uncertainty it is a practical compromise considering capital costs of test facilities.

G3 *Measurement Locations.* Static pressure taps shall simultaneously meet all of the following requirements:

G3.1 Static pressure taps may be in either the unit connections (i.e. nozzles) or in additional external piping provided for the purpose of test measurements.

G3.2 If using additional external piping, the piping arrangement shall use rigid pipe and may include fittings such as elbows, reducers, or enlargers between the pressure tap locations and the unit connections. Flexible hose is prohibited between the unit connections and the pressure taps.

G3.3 Static pressure taps shall maintain the following lengths of cylindrical straight pipe in the flow path adjacent to each pressure tap location in Table G1.

Table G1. Straight Length in Flow Path

Unit Connection, Nominal Pipe Size	Straight Length in Flow Path	
	Upstream of Pressure Tap	Downstream of Pressure Tap
≤3 inches	Minimum 10 · d	Minimum 3 · d
4, 5, or 6 inches	Minimum 6 · d	Minimum 2 · d
≥8 inches	Minimum 3 · d	Minimum 1 · d

d = The greatest pipe inside diameter dimension, using the nominal pipe size and pipe schedule nominal wall thickness, of the following locations:

- The pipe diameter at the pressure tap location
- The largest diameter of any reducer or enlarger fittings between the pressure tap location and unit connections
- The largest diameter of the first reducer or enlarger fitting between the pressure tap location and the test facility if any

G4 *Static Pressure Taps.* Static pressure taps will be in a piezometer ring or piezometer manifold arrangement with a minimum of 3 taps located circumferentially around the pipe, all taps at equal angle spacing. To avoid introducing measurement errors from recirculating flow within the piezometer ring, each of the pipe tap holes shall have a flow resistance that is greater than or equal to 5 times the flow resistance of the piezometer ring piping connections between any pair of pressure taps. A "Triple-Tee" manifold arrangement using 4 pipe tap holes is the preferred arrangement, but not required if meeting the flow resistance requirement.

G4.1 For design or evaluation purposes, flow resistance may be estimated by resistance coefficient K factor calculation methods as found in Crane Technical Paper No. 410. Generally, manifold tubing or piping can be evaluated using the K factor and pressure tap holes can be evaluated using orifice flow equations (refer to Section G5.2).

G4.2 For more information about the design of piezometer rings see paper by Blake in the Informative References, see Appendix B.

G4.3 Provisions shall be made to bleed air out of the lines connected to pressure measurement devices. These provisions shall take into consideration the orientation of pressure taps and manifold connections.

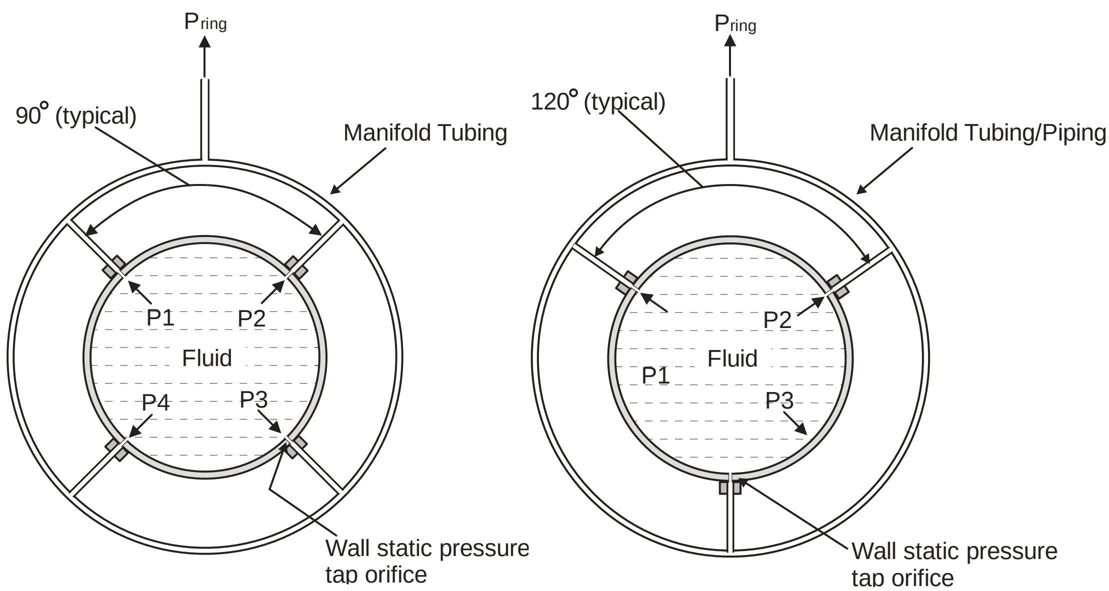


Figure G1. Examples of Piezometer Ring/Manifold

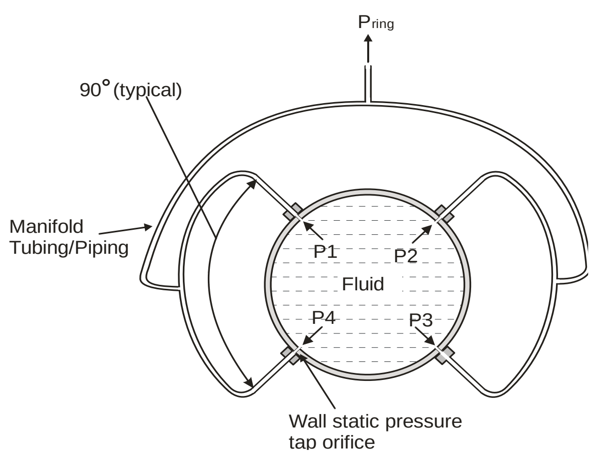


Figure G2. Example of Triple-tee Piezometer Ring/Manifold

G5 *Correction Method.* The average measured Water Pressure Drop values during test shall be adjusted to subtract additional static pressure drop due to external piping. The additional static pressure drop shall be the sum of all losses between the unit connections and the location of static pressure taps. Record the original measured value, the calculated adjustment value, and the final calculated corrected test result for Water Pressure Drop.

G5.1 The adjustment shall not exceed 10% of the measured Water Pressure Drop.

G5.2 The general form of the adjustment equations utilize the methods in the Crane Technical Paper No. 410. A Darcy friction factor is determined using the Swamee-Jain Equation G1.

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

Pipe roughness values shall be either actual measurements or approximations based on handbook values. If using handbook values consideration shall be given and values adjusted accordingly to the actual conditions of the pipe interior surface, which may have higher roughness versus the clean conditions of new pipe. Typical pipe roughness handbook values for reference:

Commercial Pipe, New Condition	ϵ (rms) (ft)
Steel	1.8×10^{-4}
Plastic	6.0×10^{-6}

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. These are shown in Equations G2 and G3.

$h_L = f \cdot \frac{L}{d} \cdot \frac{V^2}{2g}$ when the Darcy friction factor is used for straight pipe sections G2

$h_L = K \cdot \frac{V^2}{2g}$ when a K factor is specified for elbows and expansions/contractions G3

$$\Delta p_{adj} = \rho g\left[\sum_i (h_f)_i + \sum_j (h_m)_j\right] \qquad \text{G4}$$

Where:
g = 32.174 ft/s^2 (for the purposes of this standard)

Table G2. K Factors for Elbow Arrangements

Description	K Factor
Smooth elbow with r/d = 1	$20\cdot f$
Smooth elbow with r/d= 1.5	$14\cdot f$
Smooth elbow with r/d = 2	$12\cdot f$
Smooth elbow with r/d = 3	$12\cdot f$
Smooth elbow with r/d = 4	$14\cdot f$
Segmented with 2·45° miters	$30\cdot f$
Segmented with 3·30° miters	$24\cdot f$
Segmented with 6·15° miters	$24\cdot f$

The determination of the K factor for the expansion and contraction sections is a function of the diameter ratio ($\beta = \frac{d_1}{d_2}, d_1 < d_2$) as well as the angle θ of the expansion or contraction. For typical commercially available gradual expansion or contraction fittings, an equation has been developed for θ as a function of the smaller diameter d_1 that best represents the pressure drop results found in the ASHRAE Technical Report 1034-RP. The equation is valid in the range of 10° to 45°. For sudden expansion or contractions use the table's values for θ > 45°. See Crane Technical Paper No. 410 for a more complete description of the loss coefficient equations.

Table G3. Angle for expansion and reduction fittings

	$d_1 \leq 5$ inch	5 inch < d_1 < 20 inch	$d_1 \geq 20$ inch
gradual expander, gradual reducer	$\theta = 45°$	$\theta = \frac{170 - 7d_1}{3}$	$\theta = 10°$
sudden expansion, sudden contraction	45° < θ < 180°	45° < θ < 180°	45° < θ < 180°

Table G4. Resistance coefficient for expansion and reduction fittings		
	θ ≤ 45°	θ > 45°
expansion	$K = \frac{2.6\sin(\frac{\theta}{2})(1-\beta^2)^2}{\beta^4}$	$K = \frac{(1-\beta^2)^2}{\beta^4}$
reduction	$K = \frac{0.8\sin(\frac{\theta}{2})(1-\beta^2)^2}{\beta^4}$	$K = \frac{0.5\sqrt{\sin(\frac{\theta}{2})}(1-\beta^2)}{\beta^4}$

An Excel® spreadsheet is available from AHRI for computation of the water pressure drop adjustment factors.

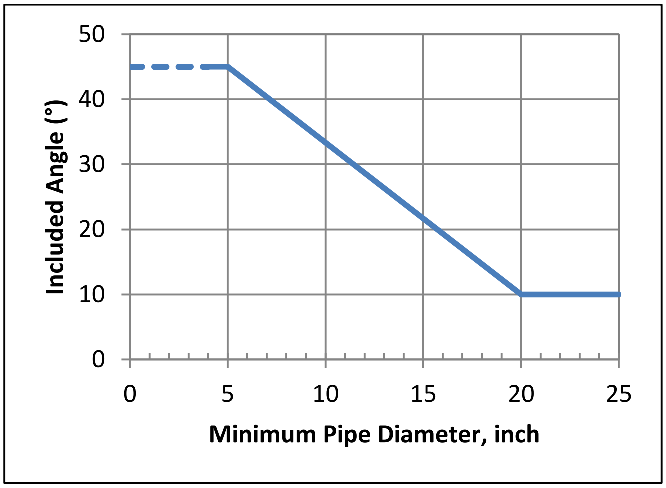


Figure G3. Correction Term for Included Angle for Expansion/Contraction Fittings

APPENDIX H. HEATING CAPACITY TEST PROCEDURE – NORMATIVE

H1 *Purpose*. This appendix prescribes methods of testing for measurement of water-side heating capacity for Air Source Heat Pump Water-heating Packages.

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:

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

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 12 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 11.

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 12 non-frosting test tolerances are satisfied:

H2.4.1.1 The heat pump undergoes a defrost

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

H2.4.1.3 One or more of the applicable Table 12 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 12 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 12 non-frosting test tolerances are satisfied:

H2.5.1.1 The heat pump undergoes a defrost

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

H2.5.1.3 One or more of the applicable Table 12 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 12 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 12, "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 12 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 12 "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 12 "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 12 "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 12 "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.

$$(Q_{cd})_{avg} = \frac{1}{t_2 - t_1} \int_{\tau t_1}^{\tau t_2} Q_{cd} \cdot \delta\tau = \frac{1}{t_2 - t_1} \sum_{i=1}^{n} (Q_{cd})_i \cdot \Delta t_i \qquad \text{H2}$$

$$\left(W_{input}\right)_{avg} = \frac{1}{t_2 - t_1} \int_{t_1} W_{input} \cdot \delta t = \frac{1}{t - t_1} \sum_{i=1}^{n} \left(W_{input}\right)_i \cdot \Delta t_i \qquad \text{H3}$$

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:

$$COP_{H,avg} = \frac{(Q_{cd})_{avg}}{\left(W_{input}\right)_{avg}} \qquad \text{H4}$$

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. Tolerances and stability requirements are defined in Table 12 in Section 5.6.2