COOLING TECHNOLOGY INSTITUTE

Acceptance Test Code For Cooling Towers

February 2019 ATC-105 (19)

This document was developed using the consensus procedure outlined in the CTI Operating Procedure 304 and has been approved for publication by the CTI Board of Directors

FOREWORD

This Cooling Technology Institute (CTI) publication is published as an aid to cooling tower purchasers and designers. It may be used by anyone desiring to do so, and efforts have been made by CTI to assure the accuracy and reliability of the data contained herein. However, CTI makes no warranty of fitness for particular purpose or merchantability nor any other warranty expressed, implied or statutory. In no event shall CTI be liable or responsible for Incidental, Consequential or Commercial losses or damage of any kind resulting from this publication's use; or violation of any federal, state, or municipal regulation with which this publication may conflict or for the infringement of any patent resulting from the use of this publication.

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This guideline document summarizes the best current state of knowledge regarding the specific subject. This document represents a consensus of those individual members who have reviewed this document, its scope and provisions. It is intended to aid all users or potential users of cooling towers.

Approved by the CTI Executive Board.

This document has been revised and approved as part of CTI's Five Year Review Cycle. This document is again subject to review in 2024.

Approved by the
CTI Executive Board

ATC-105 (19)

Table of Content

Part I - Test Procedure

1. Scope and Purpose

1.1 Scope - This Code covers the determination of the thermal capability of water cooling towers.

1.2 Purpose - The purpose of this Code is to describe instrumentation and procedures for the testing and performance evaluation of water cooling towers.

1.3 Flexibility - It is recognized that the data limitations specified throughout this test procedure represent desired conditions which may not exist at the time the test is performed. In such cases, existing conditions may be used if mutually agreed upon prior to the test by authorized representatives of the manufacturer, the purchaser, and the CTI.

1.4 Other Users - Although intended primarily for thermal acceptance testing, all or parts of this Code may be used for other purposes, such as the determination of any of the individual measurements described. It may be used for the certification of tower product lines in accordance with CTI Bulletin STD-201, in which case references to the purchaser are not applicable. Portions of this code and/or other codes, such as "ASME PTC-30 Air Cooled Heat Exchangers", may be used as applicable as a basis for water conservation and plume abatement tests by prior mutual agreement of purchaser and manufacturer.

1.5 Impartial Testing Service - It is the intention of the Cooling Technology Institute to provide independent third-party thermal performance tests. This means that any situation in which the testing agency has a material or any other interest in the outcome of the test must be avoided. The CTI-licensed Test Agency referred to in this code shall have no connection with the manufacturer, the purchaser, or the Cooling Technology Institute, other than a contractual agreement with the latter. For the purposes of this code the CTI-licensed Test Agency is hereinafter referred to as the CTI.

1.5.1 Certain categories of business interests or activities may compromise the objectivity of an agency and are considered by CTI as inappropriate for an organization licensed to provide impartial testing services. When any portion of the revenues of a testing agency or any of its corporate parents or subsidiaries is derived from these interests or activities this will preclude that organization from consideration. The following is a non-exclusive listing of examples of these categories of business interests or activities:

a) The manufacture, repair, replacement, or upgrade of cooling towers or cooling tower components.

b) Operation or ownership of cooling towers related to primary income generating processes.

1.5.2 Further, any specific situation in which the testing agency can be said to have an interest in the outcome of a particular test is considered a conflict of interest and is to be avoided. The following is a non-exclusive listing of examples of such situations:

a) Performing cooling tower design, engineering, construction, or related consultation services, such as bid evaluation, for a party to a particular test (for the purposes of this paragraph, operation and maintenance recommendations do not constitute "related consultation services").

b) Engaging in a contractual or business relationship with the cooling tower manufacturer, constructor, or supplier of components or equipment on a particular tower to be tested. (A conflict of interest shall not exist if the testing agency has conducted independent testing of other cooling towers for cooling tower owners, manufacturers, constructors, or suppliers of components and equipment).

1.6 System of Units - This standard is written in Primary Rational SI (International System of Units) units with secondary I-P (Inch-Pound) units. For full details on appropriate conversions between I-P and SI units refer to CTI Bulletin STD-145.

1.7 Uncertainty Analysis - An uncertainty analysis is not required per this Code; however, if an uncertainty analysis is desired, it should be agreed to by all interested parties prior to the test and it is recommended that agreement occur before signing the contract. Agreement prior to testing should be reached regarding whether, how and by whom an uncertainty calculation shall be made to assess the quality of the test data and, also, any criteria for rejection of the test data based on uncertainty. The purpose of an uncertainty analysis is to evaluate the quality of the test result. Test uncertainty should not be confused with test tolerance. Test tolerance is further discussed in the next paragraph. The details of an uncertainty analysis are discussed in Appendix U.

1.8 Test Tolerance - A test tolerance, if it is utilized, is a part of the contractual agreement that allows for a test capacity that is lower than 100% to be considered as acceptable. For example, a 3% test tolerance would allow for a 97% test capability to be deemed acceptable by the purchaser. A test tolerance value (expressed in percent of tower capacity, or as a deviation from design basis cold water temperature) can be written into the contract between the Owner and Manufacturer.

1.9 Nomenclature - The symbols used in this Code are identified in the following tabulation (some additional symbols are listed independently in the Appendices).

Symbol	Description	Units
a	Area of transfer surface per unit of fill volume	m^2/m^3 (ft^2/ft^3)
C	Tower capability	% of design flow
c_{pw}	Specific heat of water at constant pressure	kJ/kg·°C (BTU/lb_m·°F)
g	Acceleration due to gravity	m/s^2 (ft/s^2)
G	Mass flow rate of dry air through the cooling tower	kg dry air/s (lb_m dry air/hr)
G_d	Mass flow rate of dry air through the cooling tower at design conditions	kg dry air/s (lb_m dry air/hr)
G_t	Mass flow rate of dry air through the cooling tower at test conditions	kg dry air/s (lb_m dry air/hr)
h	Enthalpy	kJ/kg dry air (BTU/lb_m dry air)
h_a	Enthalpy of air	kJ/kg dry air (BTU/lb_m dry air)
h_i	Enthalpy of inlet air	kJ/kg dry air (BTU/lb_m dry air)
h_o	Enthalpy of outlet, or exit air	kJ/kg dry air (BTU/lb_m dry air)
h_w	Enthalpy of water at saturated wet-bulb temperature	kJ/kg dry air (BTU/lb_m dry air)
H_{pc}	Computed pumping head at design fluid flow rate, predicted from test measurements	m (ft) of flowing fluid
H_{pt}	Test (measured) pumping head	m (ft) of flowing fluid
h_w	Enthalpy of air-water vapor mixture at bulk water temperature	kJ/kg dry air (BTU/lb_m dry air)
K	Overall heat and mass transfer coefficient	$kg/s{\cdot}m^2$ ($lb_m/hr{\cdot}ft^2$)
KaV/L	Tower characteristic	dimensionless
L	Mass flow rate of water entering the cooling tower	kg/s (lb_m/hr)
L_d	Mass flow rate of water entering the cooling tower at design conditions	kg/s (lb_m/hr)
L_t	Mass flow rate of water entering the cooling tower at test conditions	kg/s (lb_m/hr)
L_1	Mass flow rate of water adjusted to design fan power	kg/s (lb_m/hr)
L_2	Tower capability expressed as mass flow rate of water at design fan power and design temperature conditions	kg/s (lb_m/hr)
$(L/G)_d$	Ratio of mass flow rate of water to that of air at design conditions	dimensionless
$(L/G)_t$	Ratio of mass flow rate of water to that of air at test conditions	dimensionless
$(L/G)_1$	L/G ratio at the intersection of the test characteristic curve and the design approach curve	dimensionless
P_t	Total pressure referred to atmospheric	Pa (inches of water)
P_{st}	Test static pressure, measured at (or referred to) the centerline of the tower piping inlet	m (ft) of flowing fluid
P_{vt}	Test velocity pressure (computed from $v^2/2g$) at the centerline of tower piping inlet	m (ft) of flowing fluid
q_{dry}	Dry heat load for wet/dry cooling tower	kJ/s (BTU/hr)
q_{wet}	Wet heat load for wet/dry cooling tower	kJ/s (BTU/hr)
q_{tot}	Total heat load for wet/dry cooling tower	kJ/s (BTU/hr)
Q_a	Airflow rate	L/s (CFM)
Q_{bd}	Blowdown water flow rate	L/s (gpm)
Q_{mu}	Make-up water flow rate	L/s (gpm)

Symbol	Description	Units
Q_w	Circulating water flow rate	L/s (gpm)
$Q_{w\ pred}$	Circulating water flow rate, predicted for test conditions	L/s (gpm)
Q_{wd}	Circulating water flow rate, design	L/s (gpm)
Q_{wt}	Circulating water flow rate, test	L/s (gpm)
$Q_{wt\ adj}$	Circulating water flow rate, test adjusted to design fan driver power	L/s (gpm)
R	Cooling range	°C (°F)
T_a	Temperature air wet-bulb	°C (°F)
T_{bd}	Temperature of blowdown water	°C (°F)
T_{cw}	Temperature of cold water leaving the tower	°C (°F)
T_{db}	Temperature of air dry-bulb entering the cooling tower	°C (°F)
T_{hw}	Temperature of hot water entering the cooling tower	°C (°F)
T_{mu}	Temperature of makeup water	°C (°F)
T_w	Temperature of water	°C (°F)
T_{wb}	Temperature of air wet-bulb entering the tower	°C (°F)
v	Velocity	m/s (ft/s)
V	Effective cooling tower fill volume	m^3 (ft^3)
W	Fan driver output power	kW (bhp)
y	Exponent for fan driver output power for the dry section of wet/dry cooling tower	dimensionless
Z_i	Vertical distance from basin curb to centerline of tower piping inlet	m (ft)
Z_s	Effective stack height	m (ft)
$\Delta\rho$	Air density difference	kg mixture/m^3 (lb_m mixture/ft^3)
$\Delta\rho_d$	Air density difference at design	kg mixture/m^3 (lb_m mixture/ft^3)
$\Delta\rho_{df}$	Air density difference driving force	kg mixture/m^3 (lb_m mixture/ft^3)
$\Delta\rho_r$	Resistance to air flow	kg mixture/m^3 (lb_m mixture/ft^3)
$\Delta\rho_t$	Air density difference at test	kg mixture/m^3 (lb_m mixture/ft^3)
Δh	Enthalpy difference	kJ/kg dry air (BTU/lb_m dry air)
ΔP_{rt}	Total pressure differential at test velocity	Pa (inches of water)
ΔP_t	Total pressure differential	Pa (inches of water)
ΔP_{td}	Total pressure differential at design	Pa (inches of water)
v	Specific volume of air	m^3 mixture/kg dry air (ft^3 mixture/lb_m dry air)
v_1	Specific volume of inlet air	m^3 mixture/kg dry air (ft^3 mixture/lb_m dry air)
v_2	Specific volume of exit air	m^3 mixture/kg dry air (ft^3 mixture/lb_m dry air)
ρ	Air density	kg mixture/m^3 (lb_m mixture/ft^3)
ρ_d	Average air density at design	kg mixture/m^3 (lb_m mixture/ft^3)
ρ_t	Average air density at test	kg mixture/m^3 (lb_m mixture/ft^3)

2. Conditions of Test

2.1 **Conduct of Test** – The tower shall be prepared for testing in accordance with CTI Document PTG-156, *Preparation for An Official CTI Thermal Performance Test*. The test shall be conducted by CTI in the presence of authorized representatives of the purchaser and the manufacturer, if they desire to be present. For acceptance testing, these representatives shall be given adequate notice prior to the test. The manufacturer shall be given permission and adequate notice to inspect the tower and prepare it for the test. In no case shall any directly involved party be barred from the test site. Acceptance test(s) shall be conducted within 12 months after structural completion of the tower, unless otherwise stipulated by contractual agreement of purchaser and manufacturer.

2.2 Condition of Equipment - At the time of the test the tower shall be in good operating condition.

a) Water distribution system shall be essentially clear and free of foreign materials which may impede the normal water flow.

b) Mechanical equipment, if involved, shall be in good working order. Fans shall be rotating in the correct direction, with proper orientation of leading and trailing edges. Fan blades shall be at a uniform angle that will yield within ± 15 percent of the specified fan driver output power loading. In addition, centrifugal fans shall be free of foreign material and properly secured to the shafts.

c) Drift eliminators shall be essentially clear and free of algae and other deposits which may impede normal air flow.

d) Fill shall be essentially free of foreign materials such as oil, tar, scale, or algae.

e) Cooling towers with polymer film fill shall have been operated a minimum of 1000 hours with a heat load (microscaling of the surface ie. seasoning) prior to the thermal acceptance test, unless otherwise stipulated by mutual agreement of purchaser and manufacturer. Cooling towers with splash fill do not require seasoning prior to an acceptance test unless specified by the manufacturer.

f) Water level in the cold water basin shall be at normal operating elevation and shall be maintained substantially constant during the test in order to provide proper air flow to the fill.

g) Make-up and/or blow-down streams may be stopped prior to testing if other test condition requirements are not adversely affected.

h) Air-cooled portions shall be essentially free of foreign materials, both inside and outside. Representatives of purchaser and manufacturer shall agree prior to commencement of testing that the cleanliness of the equipment is within the tolerance specified by the manufacturer. Prior establishment of cleanliness criteria is recommended.

2.3 **Operating Conditions** - The test shall be conducted within the following limitations:

2.3.1 The wet-bulb and dry-bulb temperatures shall be the inlet values, measured in accordance with Paragraph 3.3 of this test procedure.

2.3.2 The wind velocity shall be measured in accordance with Paragraph 3.4 of this test procedure, and unless otherwise specified as a design condition in the cooling tower contract it shall not exceed the following:

2.3.2.1 For mechanical-draft towers:

a) average wind velocity....4.5 m/s (10 miles per hour)

b) one-minute duration......7.0 m/s (15 miles per hour)

2.3.2.2 For natural-draft towers, the average wind velocity and peak wind velocity at the top of the tower and at the middle of the air inlet height shall lie within the design limits specified in the cooling tower purchase contract. If no limits were specified in that contract, an average wind velocity of 4.5 m/s (10 mph) and a peak one-minute velocity of 7.0 m/s (15 mph) at both locations, tower discharge and one-half air inlet height, shall be used as the limitation.

2.3.2.3 In the absence of wind velocity readings at the top of the shell, an indicator of the wind conditions at the top of the shell shall be the appearance of plume at the exit. For an acceptable test, visual observations of the plume shall indicate that the plume completely fills the shell outlet and rises vertically for a minimum distance of approximately one half of the outlet diameter. Refer to Figures 1 and 2 for Conforming and Non-Conforming schematics. In order to accommodate these visual observations, the test crew must have access around the tower perimeter at an adequate distance away from the shell to clearly observe the exiting plume. Observations of the crosswind plume profile is required to ensure compliance. If the wind velocity is specified in the contract, a wind speed correction curve may be supplied to correct for the actual wind speed measured at a given location.

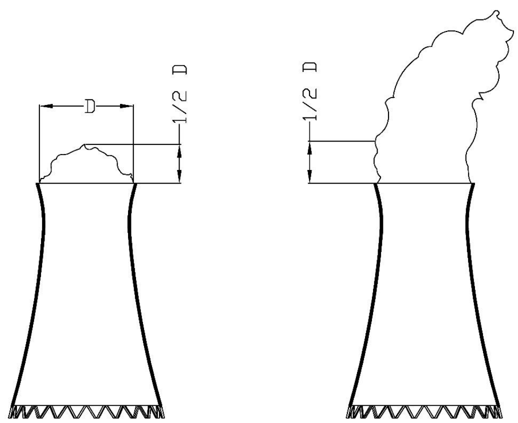


Figure 1. Plume Conforms. The plume completely fills the shell outlet and rises vertically for a minimum distance of approximately one half of the outlet diameter. The left side image represents operation on a hot day when plume extension is not as visible. The right-side image shows taller plume extension that conforms.

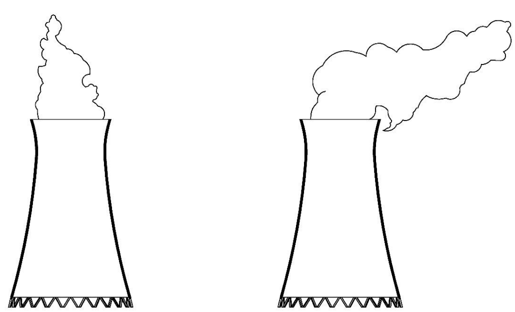


Figure 2. Plume Does Not Conform. The image on the left does not have a plume that completely fills the shell exit. The plume on the right-side image exhibits a condition of strong cross wind. The plume neither fills the shell nor rises vertically in a suitable manner.

2.3.3 The following variations from design conditions shall not be exceeded:

2.3.3.1 Wet-bulb temperature ± 8.5°C (15°F)

2.3.3.2 Dry-bulb temperature ± 14.0°C (25°F)*

2.3.3.3 Range…………………………..± 20%

2.3.3.4 Circulating water flow………….± 10%

2.3.3.5 Barometric Pressure ± 3.5 kPa (1"Hg)

2.3.3.6 Fan driver output power……….± 15% after the density correction

** If applicable*

2.3.4 The water shall be distributed to all operating cells and/or parts of the tower as recommended by the manufacturer. For multi-cell towers, one or more cells may be shut down, providing the circulating water flow to each operating cell is within the allowable limits. (For the purposes of this Code, a "cell" is defined as the smallest subdivision of the tower, bounded by exterior walls and partition walls, which can function as an independent unit. Each cell may have one or more fans or stacks and one or more distribution systems).

2.3.5 The total dissolved solids in the circulating water shall be limited as follows:

2.3.5.1 For all towers except wet/dry towers, the dissolved solids shall not exceed the greater of the following:

a) 5,000 ppm

b) 1.1 times the design concentration

The circulating water shall contain not more than 10 ppm oil, tar, or fatty substances as determined by the procedure outlined in "Standard Methods for the Examination of Water, Sewage, and Industrial Wastes", published by the American Public Health Association.

2.3.5.2 For wet/dry towers, the limits for foreign substances in the circulating water shall be by prior mutual agreement between the purchaser and manufacturer.

2.3.6 For natural-draft towers, there should not be rain during or at least two hours before the test.

2.3.7 For natural-draft towers, an additional criteria for defining acceptable test periods is the vertical ambient dry-bulb temperature gradient. Where it is practical to measure this parameter directly, the following criteria may be used for an acceptable test period:

The average vertical ambient dry-bulb temperature gradient between the elevation of the center of the cooling tower air inlet and twice the height of the top of the tower shell, assessed in 60 m (200 ft) maximum increments, shall lie within the design limits specified in the cooling tower purchase contract. If no limits were specified in that contract, an average gradient of at least a 0.65°C/100 m (3.5°F/1000 ft) decrease in ambient dry bulb temperature with height (U.S. Standard Lapse Rate) shall be used as a guideline.

An indicator of the cooling tower vertical ambient dry-bulb temperature gradient shall be the difference in dry-bulb temperatures between ground level and the top of the air inlet. For an acceptable test, the average of the dry-bulb temperature probes at or near the top of the air inlet shall be at least 0.15°C (0.25°F) less than the average dry-bulb temperature probes measured at the lowest level of the inlet (U.S. Standard Lapse Rate for a typical air inlet height). If this criteria is not met an inversion condition may exist.

2.4 Constancy of Test Conditions - For a valid test, variations in test conditions shall be within the following limits during each test run:

2.4.1 Circulating water flow rate shall not vary by more than 2 percent.

2.4.2 Heat load is defined as the amount of heat to be removed from the water within the tower which is proportional to the mass flow rate of water times the water's temperature differential between inlet and outlet. The linear least squares trend of the heat load shall not vary by more than 5 percent. Secondly, the maximum deviation of a reading may not exceed the overall test period average by more than plus or minus 5 percent.

2.4.3 Range, also called cooling range, is defined as the temperature difference between the hot water entering the tower and cold water leaving the tower. The linear least squares trend of the range shall not vary by more than 5 percent. Secondly, the maximum deviation of a reading may not exceed the overall test period average by more than plus or minus 5 percent.

2.4.4 Individual psychrometer air temperature readings may fluctuate during the test, but the linear least squares trend in the reading average (average for all stations) for the test period shall not exceed the following:

2.4.4.1 Wet-bulb temperature…1°C (2°F) per hour

2.4.4.2 Dry-bulb temperature…3°C (5°F) per hour*

** If applicable*

Secondly, the maximum deviation of a reading average (e.g. the average of all wet-bulb psychrometer readings at one time interval) may not exceed the overall test period average by more than the following:

2.4.4.3 Wet-bulb temperature…..±1.5°C (3°F)

2.4.4.4 Dry-bulb temperature…±4.5°C (7.5°F)*

** If applicable*

2.5 Duration of Test - After reaching steady state conditions, the requirements for test duration shall be as follows:

2.5.1 For mechanical-draft towers, the duration of the test run shall be not less than one hour. If thermal lag time is greater than five minutes, the timed test period shall be at least one hour plus thermal lag time; refer to Appendix J for additional information.

2.5.2 For natural-draft towers, it is recommended that the test duration include a minimum of six one-hour periods of test data where the operating conditions are within the limitations stated in Paragraph 2.3 and that they shall be collected over at least a 2-day period. Alternatively, if it is agreed to by both parties, a longer test period with additional tests may be utilized to more accurately determine the overall performance.

2.6 Frequency of Readings - Readings shall be taken at regular intervals and recorded in the units and to the number of digits shown in the following tabulation:

Measurement	Minimum Number Each Hour Per Station	Unit	Record to Nearest
Wet-bulb temperature[1]	12	°C (°F)	0.05 (0.1)
Dry-bulb temperature[1]	12	°C (°F)	0.05 (0.1)
Cold water temperature[1]	12	°C (°F)	0.05 (0.1)
Hot water temperature[1]	12	°C (°F)	0.05 (0.1)
Circulating water flow[2]	3	L/s (gpm)	0.5%
Tower pumping head	1	m (ft)	0.01 (0.1)
Fan driver power input[3]	1	kW (hp)	0.5%
Wind velocity	continuous	m/s (mph)	0.5 (1)
Make-up temperature	2	°C (°F)	0.05 (0.1)
Make-up flow[4]	2	L/s (gpm)	0.5%
Blow-down temperature	2	°C (°F)	0.05 (0.1)
Blow-down flow[4]	2	L/s (gpm)	0.5%
Barometric Pressure	1	kPa (In of Hg)	0.01 (0.01)

[1] *Or recording*

[2] *Single center point readings for comparison with full traverse reading (when measurement is made by pitot tube) to verify that the flow has not changed by the allowable limit*

[3] *If applicable or required*

[4] *Or totalized values per test*

2.7 Test Uncertainty - The overall test uncertainty depends on the type and number of instruments used for the various measurements, and on the stability of the test conditions. Details on the recommended method for evaluation of overall test uncertainty are presented in ASME PTC-19.1 "Measurement Uncertainty" and in Appendix U.

3. Instruments and Measurements

Instruments used for Code compliant tests shall meet or exceed the following accuracy criteria.

Instrumentation Accuracy

Measurement	Medium	Minimum Accuracy		Instrument Examples
		S-I	I-P	
Temperature	Air Wet-Bulb Temperature	± 0.05 ºC	± 0.10 ºF	RTD
	Air Dry-Bulb Temperature			
	Hot Water Temperature			
	Cold Water Temperature			
	Recirculating Water Temperature			
	Make-up Water Temperature	See Note 1		NA
	Blow-down Water Temperature	See Note 2		NA
Flow	Circulating Liquid	± 2.0%		Magnetic Flowmeter
	Circulating Liquid	± 3.0%		Pitot Tube
	Make-up Liquid	See Note 1		NA
	Blow-down Liquid	See Note 2		NA
Electrical	Fan Driver Input Power	± 3.0%		Power Meter
	Pump Driver Power (Note 3)			
Pressure	Air	0.34 kPa	±0.1 ”Hg	Barometer
	Water	± 1.0%		Transducer, Manometer
Velocity	Air Wind Speed	0.5 m/s	1.0 mph	Wind Meter

Note 1: The code provides a technique to calculate make-up water flow due to evaporation during the test. The influence of make-up water temperature and its entry point into the system must be understood in order to properly account for its influence during the test period. The methodology of how make-up flow will be accommodated must be mutually agreed by all parties prior to data collection. It is suggested to refer to Appendix U for information regarding the sensitivity of Make Up Flow and Temperature on the test result.

Note 2: It is generally accepted practice that blow-down flow is shut off during thermal testing. The technique of how blow-down is accounted for during the test must be mutually agreed on by all parties prior to data collection.

Note 3: The pump power reference is provided for recirculation water pumps encountered with ATC-105S tests. Pump power is not a required measurement for a standard ATC-105 test.

Data acquisition and recording are critical components of maintaining the integrity of these accuracy tolerances.

All instruments used on a test must be calibrated and approved by CTI prior to the test. Calibration must be traceable to the U.S. National Institute of Standards Technology (NIST), other national measurement laboratories, or derived from accepted values of natural physical constants. The minimum calibration frequencies will be as follows:

Instrument	Minimum Calibration Frequency
Temperature sensors	Within 3 months prior to use
Water flow measurement devices	Three years if undamaged
Electric power meters	Yearly
Wind speed and direction devices	Yearly

Calibrations shall be performed whenever an instrument is damaged, or its accuracy is called into question for any other reason. Sufficient on-site comparisons of temperature sensors, after all wiring connections are made, is required before testing. If in doubt, a temperature sensor will be compared to another similar sensor and replaced if it is outside the accuracy specified for the appropriate measurement. In addition, at the request and expense of the test purchaser, any and all instrumentation used on a test can be calibrated before and after a test. The test agency shall have a written procedure for calibration of each instrument. The test agency shall maintain records showing the calibration history for each instrument and make them available upon request.

The test report shall include the individual identification and location for each instrument used on the test so that calibration date and history can be traced.

3.1 Water Flow Measurements

3.1.1 Circulating water flow measurements may be made by any of CTI-approved methods specified in CTI Bulletin STD-146, namely a Pitot tube or other flow measurement devices that have a calibration accuracy of ±1.0 percent or better. CTI Bulletin STD-146 covers in detail the pitot tube including all the associated equations, piping requirements, table of equal annular mid-point factors for traverse point locations, and properties of water and manometer fluids. For the alternative flow measurement devices, sections are presented on operating principle, installation, and accuracy. The selection of the method and location of measurement will be dependent upon the nature of the installation to be tested and may decrease the actual installed accuracy.

3.1.2 If the circulating water flow to the tower is measured by pitot tube traverse, an air-over-water (inverted type) manometer shall be used unless otherwise agreed by the parties to the test. One full traverse (along two axes at 90° to each other) shall be made either immediately preceding the test or during the test. A minimum of two center-points readings shall be taken during the test to monitor for changes in flow, possibly requiring another pitot traverse.

3.1.3 Additional water quantities may be required

a) Make-up water flow*

b) Blow-down water flow*

* *if not stopped prior to test*

These two flow measurements may be directly measured, by using plant flowrate instrumentation, or calculated by heat balance from the other measurements.

3.2 Water Temperature Measurements - Water temperature measurements shall be made using temperature devices that comply with the Instrument Accuracy Table in Section 3 and be carefully located where the water will be thoroughly mixed.

3.2.1 Hot circulating water temperature measurement shall be made in the tower riser(s) or at the discharge of the inlet riser(s) into the flume or distribution system or, for a multi-cell tower, in the supply header just upstream of the first riser. If the source is a mixture of two or more streams of different temperatures, complete mixing must be assured at the point of measurement or sufficient flow rate and temperature measurements shall be made to ensure an accurate weighted average water temperature.

3.2.2 Cold circulating water temperature measurements should preferably be made in a full-flowing bleed stream at the circulating pump discharge, and the average corrected for heat added by the pump (see Appendix I). If the measurement is made at a location where temperatures and velocities are not uniform over the stream cross-section, sufficient flow rate and temperature measurements shall be made to ensure an accurate weighted average water temperature.

3.2.3 Make-up water temperature measurement, if applicable, shall be made at the point the make-up water enters the system.

3.2.4 Blow-down temperature measurement, if applicable, shall be made at the point the blow-down leaves the system.

3.3 Inlet Air Temperature Measurements - The measurement of inlet wet-bulb temperature is required for the testing of all types of cooling towers covered by this Code. The measurement of inlet dry-bulb temperature is required for natural-draft, fan-assisted, and wet/dry cooling towers. The measurement of inlet dry-bulb temperature is also required for mechanical-draft cooling towers of the forced draft design, in order to determine the fan inlet air density. It is recommended that the dry-bulb temperature also be recorded for mechanical-draft towers of the induced draft design to allow accurate calculation of inlet air psychometric properties and/or estimating evaporation.

3.3.1 Inlet Wet-bulb Temperature. The inlet wet-bulb temperature shall be measured with a mechanically aspirated wet bulb instrument, utilizing sensors that comply with the Instrument Accuracy Table in Section 3 and specification criteria per Appendix O.

3.3.1.1 *For measurement of inlet wet-bulb temperature, the instruments shall be located within 1.5 m (5 ft) of the air intake(s). A sufficient number of measurement stations shall be designated to ensure that the test average is an accurate representation of the true average inlet wet-bulb temperature. Guidance on the location of measurement stations is given in Appendix G.*

3.3.1.2 *For manually recorded data, the average of 3 successive readings taken at 10-second intervals at each station shall be considered the wet-bulb temperature at that time at that instrument station. The station averages shall be averaged to obtain the effective wet-bulb temperature for that run. Such runs shall be made every 5 minutes during the test period, and the arithmetic average of these runs shall be considered the inlet wet-bulb temperature and shall be used for the evaluation of results.*

3.3.1.3 *For computer data acquisitions systems, the number of readings taken must be at least equal to that of the manually recorded system, a minimum of 36 readings per hour. The number recorded must also be at least equal to the manually recorded system, a minimum of 12 per hour.*

3.3.2 Inlet Dry-bulb Temperature. The inlet dry-bulb temperature, where applicable, shall be measured with a wickless mechanically aspirated wet bulb instrument utilizing sensors that comply with the Instrument Accuracy Table in Section 3 and specification criteria per Appendix O.

3.3.2.1 *For the measurement of inlet dry-bulb temperature, instrument location, the number of stations, the frequency of readings, and the reading and averaging procedures shall be as described in Paragraphs 3.3.1.1, 3.3.1.2, and 3.3.1.3 for wet-bulb temperature measurement.*

3.3.2.2 *For natural-draft cooling towers, the inlet dry-bulb gradient temperature is very important since it affects the inlet air density which is part of the driving force that makes the cooling tower perform. The following are the dry-bulb temperature measurement requirements for natural-draft towers:*

a) For the direct measurement of natural-draft cooling tower vertical ambient dry-bulb temperature gradient, the plan location of the instruments shall neither be down-wind of nor within the influence of the cooling tower nor any other site sources of turbulence or thermal gradients which might substantially affect the data. The vertical elevations shall be as specified in Paragraph 2.3.7. Instrumentation locations and techniques will be site-specific and shall be based upon mutual agreement prior to testing.

b) As an INDICATOR of the vertical dry-bulb temperature gradient on natural-draft towers, two dry-bulb instruments shall be located at the same circumferential position around the tower, one at 1.5 m (5 ft) above grade level, and the other near or above the top of the air inlet. The instrument placed near or above the top of the air inlet may be located on a stairway at or above the elevation of the top of the air inlet if the location otherwise satisfies the criteria herein. The instruments shall not be placed downwind of the cooling tower. Attempt shall be made to place the instruments in locations not subject to radiation or to convective air heating effects due to solar heating of the shell. The placement of the gradient indicator instruments shall be mutually agreed upon by the parties to the test.

3.4 Wind Velocity (Speed and Direction) - Wind velocity shall be measured with a meteorological type anemometer and wind vane, preferably remote reading and recording. Rotating cup anemometers with separate wind direction vane or combination self-aligning propeller and direction vane devices are readily available and acceptable. Measurements shall be made in an open and unobstructed location, upwind of the tower and beyond the influence of the inlet air approach velocity. Care shall be taken to assure recorded wind speed and direction are representative of wind conditions affecting the tower. Placement of the wind measurement device shall be subject to mutual agreement by all parties to the test. Wind direction shall be recorded in compass degrees with the tower orientation and reference North clearly indicated.

3.4.1 For mechanical-draft towers with an overall height of 6 m (20 ft) or less, wind velocity shall be measured 1.5 m (5 ft) above curb elevation, at a point within 15 to 30 m (50 to 100 ft) of the tower, if practical. For mechanical-draft towers where the distance between curb and discharge elevations exceeds 6 m (20 ft), the wind velocity shall be measured at an elevation above the curb elevation that is approximately one-half the difference between the curb and discharge elevations and at a point at least 30 m (100 ft) from the tower, if practical.

3.4.2 For natural-draft towers, the wind shall be measured at the shell discharge elevation at a point within 45 to 450 m (150 to 1500 ft) from the curb of the tower, if practical. In addition, the wind velocity shall be measured at a height of one-half the inlet height and at a distance greater than twice the air inlet height and at least a quarter of the base diameter from the cooling tower. Section 2.3.2.2 gives the wind speed limitations.

3.5 Tower Pumping Head - Tower pumping head shall be the sum of:

a) The total pressure above atmospheric expressed in m (ft) of water at the centerline of the tower piping inlet, and

b) The vertical distance measured in m (ft) of the centerline of the piping inlet above the basin curb.

Test pumping head is given by:

$$H_{pt} = P_{st} + P_{vt} + Z_i \quad (3.1)$$

where:

H_{pt} = Test pumping head

P_{st} = Test static pressure at the centerline of the tower piping inlet

P_{vt} = Test velocity pressure at the centerline of the tower piping inlet

Z_i = Vertical distance from basin curb to centerline tower piping inlet

Corrected pumping head is given by:

$$H_{pc} = \left[(P_{st} + P_{vt}) \left(\frac{L_d}{L_t} \right)^2 \right] + Z_i \quad (3.2)$$

where:

H_{pc} = Corrected pump head at design flow rate

L_d = Mass flow rate of water at design conditions

L_t = Mass flow rate of water at test conditions

3.6 Fan Driver Output Power - For mechanical-draft and fan-assist towers, power shall be determined as power input to the motor or driver. In the case of electric motors, power input shall be determined by direct measurement of the kilowatt input, or by measurement of the voltage, current, and power factor per ASME PTC-19.6 "Electrical Measurement in Power Circuits". If motor input power is not directly measured at the motor, then a line loss correction shall be made, unless agreed upon by all parties. If a tower is equipped with a Variable Frequency Drive (VFD) the fan speed and power measurement can be affected, therefore, the VFD should be bypassed during the test. If a performance guarantee is based on driver output, efficiencies stated by the manufacturer of the driver may be used.

3.7 Water Analysis - A sample of the circulating water shall be taken during the test. If there are any questions concerning the condition of the circulating water, the sample shall be analyzed by a reputable testing laboratory to determine conformance with Paragraph 2.3.5 of this test procedure.

3.8 Make-Up and Blow-Down - If the Make-Up Water Flow has not been shut off during the test period and the Tower Cold Water Temperature $(T_{CW,t})$ has been measured downstream of the Make-Up Water insertion point, a correction must be made for the impact of the temperature and flow rate of the Make-Up (and Blow-down if not shut off during the test) using Equation 3.3 below:

$$T_{CW,adj} = \frac{(T_{CW,t})(Q_{w,t}) - (T_{mu})(Q_{mu}) + (Q_{bd})(T_{bd})}{(Q_{w,t}) - (Q_{mu}) + (Q_{bd})} \quad (3.3)$$

where:

$T_{cw,\,adj}$ = Cold water temperature adjusted for make up and blow down

$T_{cw,\,t}$ = Cold water temperature at test conditions

$Q_{w,\,t}$ = Circulating water flow rate at test conditions

T_{mu} = Temperature of make-up water

Q_{mu} = Make-up water flow rate

Q_{bd} = Blow down water flow rate

T_{bd} = Temperature of blow down water

4. Report of Results

4.1 Scope - The report of the results of the test shall include, the data required by this test procedure, the manufacturer's data, a description of the cooling tower with its orientation and principal dimensions, and a sketch of the installation showing the location of points where water flow, temperatures, and other required measurements were taken. Notation shall be made of any buildings, obstructions, or other equipment in the immediate vicinity of the tower tested. Notation shall also be made of other equipment or facilities discharging heat or vapor in the immediate vicinity.

4.2 Test Data Documentation - The observations shall be included in a comprehensive test report that shall constitute:

- A summary page having design and test values
- All original log sheets of the data collected
- The manufacturer's performance data as applicable
- Identification of the parameters that were out of code
- Capability analysis documentation

Documentation of instrument calibration shall be made available if requested.

4.3 Distribution - Upon completion of the test, one copy of the final test report becomes the property of the purchaser, one copy becomes the property of the manufacturer, and the original becomes the property of the CTI Representative.

A single summary sheet listing all important data and tower performance shall be prepared by the CTI Representative, who shall mail copies to the purchaser and the manufacturer within 10 days following the test.

4.4 Security - Information on any test will be available only to the purchaser, the manufacturer, and the CTI Representative. Any site or manufacturer identifiable information will not be released by CTI.

4.5 Limitations - Adherence to the limits of wet-bulb temperature, dry-bulb temperature, cooling range, and circulating water flow imposed by this Code will yield results with accuracy commensurate with the stability of the test conditions and the accuracy of the instruments specified for measurements.

When used for test conditions outside the limits described, errors may result due to the following considerations:

4.5.1 The effects of wide deviations from design in the following variables may not be adequately described by the equations and/or graphs used for adjustment of the test data:

a) water circulation rate
b) water temperatures
c) air flow rate

d) air wet-bulb temperature
e) air dry-bulb temperature

4.5.2 Strong and/or gusting winds are likely to result in cooling tower malperformance.

4.5.3 Poor air and/or water distribution will result in malperformance.

4.5.4 For evaluation by the characteristic curve method, the accuracy of the result will depend on the accuracy of the characteristic curve.

4.5.5 For evaluation by the performance curve method, the accuracy of the result will depend on the accuracy of the performance curves.

4.6 Tower Capability - Tower Capability is the thermal performance result calculated for a test conducted in accordance with this Code. Tower Capability is defined as the ratio of the adjusted water flow rate (or L/G) to the predicted water flow rate (or L/G) at the test conditions, expressed as a percent. The calculation procedures required to calculate the adjusted water flow rate and predicted water flow rate at the test conditions are dependent on the type of tower tested and are defined in Part II of this code.

PART II - EVALUATION OF RESULTS

5. Mechanical-Draft Cooling Tower – Characteristic Curve Method

5.1 Scope and Purpose - This part of the Code outlines a method for evaluation of the performance of mechanical-draft cooling towers from test data using characteristic curves. The results are expressed in terms of water cooling capability.

5.2 Manufacturer's Data - The manufacturer shall submit tower performance data relating tower characteristic, *KaV/L*, and water/air ratio, *L/G*, indicating the design *L/G* to be used for the test evaluation. This relationship may be presented as an equation with all constants listed, or as a curve or properly identified family of curves, expressing the effects of variables (such as hot water temperature and air velocity) that may have a significant effect on the result, and shall cover a range of *L/G* values of at least ± 20 percent from design *L/G*. If the relationship is presented as a curve, the graphical scaling must permit the determination of *KaV/L* to a minimum precision equal to that provided by the CTI *Cooling Tower Performance Curves* (CTI ToolKit).

The design *L/G* value shall be identified, as well as a listing of the design point conditions of hot water temperature, cold water temperature, inlet wet-bulb temperature, inlet dry-bulb temperature (forced draft tower), fan drive power (motor output power, unless otherwise stated), fan motor efficiency (if available at the time of quote), and either design barometric pressure or design basis elevation.

5.3 Determination of Test *L/G* Value - From the average water flow rate and fan driver output power at time of test, the test value of $\left(\frac{L}{G}\right)_t$ shall be calculated from:

$$\left(\frac{L}{G}\right)_t = \left(\frac{L}{G}\right)_d \left(\frac{Q_{wt}}{Q_{wd}}\right) \left(\frac{\dot{W}_d}{\dot{W}_t}\right)^{1/3} \left(\frac{\rho_t}{\rho_d}\right)^{1/3} \left(\frac{\nu_t}{\nu_d}\right) \quad (5.1)$$

where

$(L/G)_d$ = Ratio of mass flow rate of water to air from the manufacturer's design data

v = Specific volume of air, either test($_t$) or design($_d$)

ρ = Density of air, either test($_t$) or design($_d$)

W = Fan driver output power, either test($_t$) or design($_d$)

Q_w = Circulating water flow rate, either test($_t$) or design($_d$)

The design and test values of air specific volume (v) and density (ρ) are evaluated at:

a) the tower inlet conditions for a forced draft tower, or

b) the tower exit conditions for an induced draft tower (by heat balance).

Where the test was conducted with one or more cells of the tower shut down, in accordance with Paragraph 2.3.4, design water rate and fan driver output power shall be taken as the design values for the cells in operation.

5.4 Determination of Test *KaV/L* Value - Using average test values of hot water, cold water and wet-bulb temperatures, and test value of L/G, the test value of *KaV/L* shall be calculated from:

$$\frac{KaV}{L} = c_{pw} \int_{T_{cw}}^{T_{hw}} \frac{dT}{h_w - h_a} \quad (5.2)$$

where

c_{pw} = Specific heat of water

T_{hw} = Temperature of the hot water

T_{cw} = Temperature of cold water

h_w = Enthalpy of saturated air at the hot water temperature

h_a = Enthalpy of air entering the tower

Values for the enthalpies, h_w and h_a, are available in CTI ToolKit for the specific barometric pressure, or site elevation.

5.5 Determination of Tower Capability - The point representing calculated values of *L/G* and *KaV/L* from the test data shall be located on the manufacturer's tower characteristic graph. Through this test point a curve shall be drawn parallel and using the same form as the manufacturer's tower characteristic curves. Points on this curve may be calculated by using the CTI ToolKit software. The intersection of the line so drawn with the design approach curve determines the value of *L/G* at which the tower would produce design cold water temperature when operating at design conditions. The tower capability in percent of design water flow is the ratio of the *L/G* so determined to design *L/G*, multiplied by 100. If multiple characteristic curves have been submitted, this design *L/G* shall be adjusted to the applicable characteristic curve for the test conditions.

An example evaluation is given in Section III, the Appendices.

6. Performance Curve Method

Using the performance curve analysis method to evaluate cooling tower test data relative to design conditions provides for an expedient method of analysis in that one merely needs to cross-plot and interpolate from the curves. Alternatively, CTI Toolkit can be used to conduct the performance curve evaluation.

It is recommended the owner and their consultant review cooling tower performance curves for inconsistencies associated with predicted cold water temperatures at various operating conditions. That review should be conducted prior to execution of a contract with a cooling tower manufacturer and the review should result in the owner accepting the performance curves associated with the contract.

Performance curves associated with competing bids should be analyzed and compared at various conditions within the boundaries of the curves (i.e., range varying from design basis range, inlet wet bulb temperature varying from design basis, etc.) in order to identify inconsistencies.

Session 4 of the 2018 Educational Seminar is available for free download at cti.org that can assist in evaluating proposed performance curves for their veracity.

To improve the ability to analyze cooling tower capability at the design point, it is preferable to test the tower within 10% of the design water flow. In cases when the predicted water flow is beyond the boundary of the manufacturers performance curves, the analyst shall extrapolate a straight line using the nearest two data points on the cold water temperature versus predicted water flow graph in crossplot 2 on mechanical draft or crossplot 3 on natural draft until it meets the tested condition cold water temperature. The same analysis can be done using CTI ToolKit software.

7. Mechanical-Draft Cooling Tower – Performance Curve Method

7.1 Scope and Purpose - This part of the Code outlines a method for evaluation of the performance of mechanical-drafting cooling towers from test data using performance curves. The results are expressed in terms of water cooling capability.

7.2 Manufacturer's Data - The manufacturer shall submit a family of performance curves consisting of a minimum of 3 sets of 3 curves each. One set shall apply to 90%, one to 100%, and the other to 110% of the design water circulation rate. Each set shall be presented as a plot of wet-bulb temperature as abscissa versus cold water temperature as ordinate, with cooling range as parameter. Graphical scaling shall be incremented so as to provide a minimum of 0.2°C (0.5°F) increments and no more than 0.1°C/mm (5°F/inch) for both wet-bulb and cold water temperatures. Curves shall be based on constant fan pitch angle. In addition to the design range curve, bracketing range curves of approximately 80% and 120% of design to the nearest 0.5°C (1°F) shall be shown, at a minimum. The curves shall fully cover (but not necessarily be limited to) the range of variables specified in Paragraph 2.3. The design point shall be shown on the appropriate curve, as well as a listing of the design point conditions of hot water temperature, cold water temperature, inlet wet-bulb temperature, inlet dry-bulb temperature (forced draft tower), design *L/G* (induced draft only), fan driver output power (fan motor output power, unless otherwise designated), fan motor efficiency (if available at time of quote), and either design barometric pressure or design basis elevation.

7.3 Determination of Tower Capability - The manufacturer's performance curves shall be cross plotted at test conditions to determine capability. A suitable procedure consists of first preparing a set of 3 curves, based on test wet-bulb temperature, relating cooling range, cold water temperature, and water circulation rate. From this set of curves, a single curve is then prepared, based on test cooling range, relating cold water temperature to water circulation rate. The determination of performance capability consists of comparing the predicted water flow rate ($Q_{w,pred}$) from this curve with the adjusted test water flow rate ($Q_{wt,adj}$) using equations (7.1) and (7.2) following:

$$Q_{wt_{adj}} = Q_{wt}\left(\frac{\dot{W}_d}{\dot{W}_t}\right)^{1/3}\left(\frac{\rho_t}{\rho_d}\right)^{1/3} \qquad (7.1)$$

where:

Q_{wt} = Circulating water flow rate, test

ρ = Density of air, either test($_t$) or design($_d$)

W = Fan driver output power, either test($_t$) or design($_d$)

The capability, C, expressed in percent of design circulating flow is computed using Equation (7.2):

$$C = \frac{Q_{wt_{adj}}}{Q_{W_{pred}}} \text{ x } 100 \qquad (7.2)$$

where:

$Q_{w,pred}$ = Circulating water flow rate, predicted for the test conditions

An example evaluation is given in Section III, the Appendices.

8. Natural-Draft Cooling Towers – Performance Curve Method

8.1 Scope and Purpose - This part of the Code outlines a method for evaluation of the performance of natural-draft cooling towers from test data using performance curves. The results are expressed in terms of water cooling capability.

8.2 Manufacturer's Data - The manufacturer shall submit a family of curves which correctly relate the pertinent performance variables.

Performance curves applicable to 90%, 100%, and 110% of the design water flow rate shall be furnished. Each set shall consist of three or more cooling range curves and three or more relative humidity curves, arranged to show the effects of wet-bulb temperature, relative humidity, and cooling range on outlet water temperature. The range curves shall be presented in uniform increments of 0.5°C (1°F), with sufficient scope to cover approximately ± 20% of design range. The relative humidity curves shall be presented in uniform increments of % RH, with sufficient scope to cover approximately ± 40% of the design value with maximum increments of 20% RH. The design point shall be shown on the appropriate curve, as well as a listing of the design point conditions of hot water temperature, cold water temperature, inlet wet-bulb temperature, inlet dry-bulb temperature, design *L/G*, and either design barometric pressure or design basis elevation.

8.3 Determination of Tower Capability - The manufacturer's performance curves shall be cross plotted at test conditions to determine capability.

From the test wet-bulb temperature, relative humidity, and cooling range, a cross-plot relating outlet water temperature and predicted circulation rate will be prepared. From this graph the predicted water circulation rate capability at the test outlet water temperature is determined. Performance is then computed from Equation (8.1):

$$C = \frac{Q_{wt}}{Q_{w_{pred}}} \text{ x } 100 \qquad (8.1)$$

where:

Q_{wt} = Circulating water flow rate, test

$Q_{w,pred}$ = Circulating water flow rate, predicted for test conditions

An example evaluation is given in Section III, the Appendices.

9. Natural-Draft Cooling Towers With Fan-Assist – Performance Curve Method

9.1 Scope and Purpose - This part of the Code outlines a method for evaluation of the performance of fan-assist cooling towers from test data using performance curves. The results are expressed in terms of water cooling capability.

9.2 Manufacturer's Data - The manufacturer shall submit a family of curves which correctly relate the pertinent performance variables.

Performance curves applicable to 90%, 100%, and 110% of the design water flow rate shall be furnished. Each set shall consist of three or more cooling range curves and three or more relative humidity curves, arranged to show the effects of wet-bulb temperature, relative humidity, and cooling range on outlet water temperature. The range curves shall be presented in uniform increments of 0.5°C (1°F), with sufficient scope to cover approximately ± 20% of design range. The relative humidity curves shall be presented in uniform increments of % RH, with sufficient scope to cover approximately ± 40% of the design value with maximum increments of 20% RH. The design point shall be shown on the appropriate curve, as well as a listing of the design point conditions of hot water temperature, cold water temperature, inlet wet-bulb temperature, inlet dry-bulb temperature, design *L/G*, and either design barometric pressure or design basis elevation.

Additional data must be furnished, however, to cover the effect of fan power. This may be accomplished in any one of several different ways,

a) Three sets of curves shall be furnished, as for the pure natural-draft type, but with the addition of a nomograph extension covering the effects of fan power.

b) Nine sets of curves shall be furnished, as for the pure natural-draft type; three of these applicable to design fan power and three each applicable to 80% and 120% of design fan power.

c) Three sets of curves shall be furnished, as for the pure natural-draft type, plus the following additional design data to enable adjustment of test data to design fan power:

1) design *L/G* ratio

2) effective stack height

3) design fan power.

9.3 Determination of Tower Capability - The manufacturer's performance curves shall be cross plotted at test conditions to determine capability.

The procedure is similar to that described for the pure natural-draft tower (Paragraph 8.3) but must include the additional variable fan driver output power.

If the manufacturer's curves include fan driver output power as a parameter, as described in Paragraph 9.2 (a), the evaluation procedure is essentially the same as for the pure natural-draft type, requiring a single cross plot.

When the manufacturer's curves include fan driver output power as a curve identification as mentioned in Paragraph 9.2 (b), the procedure requires an additional cross plot. A graph is first prepared relating outlet water temperature and predicted circulation rate. This graph will consist of three curves, one each for 80%, 100%, and 120% of design fan power. A second cross plot is then prepared, relating fan power and predicted circulation rate at test outlet water temperature. From this curve the predicted water circulation rate capability at the test fan power is determined, and percent capability computed as before.

If fan power is not included in the manufacturer's curves, the additional data listed in Paragraph 9.2 (c) shall be used to compute the effects of variations in fan power. A trial-and-error or graphical procedure is required to determine the solution.

An example evaluation is given in Section III, the Appendices.

10. Wet/Dry Mechanical-Draft Cooling Towers – Performance Curve Method

10.1 Scope and Purpose - This part of the Code outlines a method for evaluation of the performance of mechanical-draft wet/dry cooling towers from test data using performance curves. The results are expressed in terms of water cooling capability. "Wet" and "Dry" refer to separate evaporative and non-evaporative heat exchanger elements incorporated into a cooling tower.

10.2 Manufacturer's Data - The manufacturer shall submit a family of curves which correctly relates the pertinent performance variables.

Performance curves applicable to 90%, 100%, and 110% of the design water flow rate shall be furnished. Each set shall consist of three or more cooling range curves and at least four relative humidity curves, arranged to show the effects of wet-bulb temperature, relative humidity, and cooling range on outlet water temperature. The range curves shall be presented in uniform increments of 0.5°C (1°F), with sufficient scope to cover approximately ± 20% of design range. The relative humidity curves shall be presented to cover the extent of expected conditions such as 5%, 20%, 40%, 60%, and 100% relative humidity. The design conditions shall be indicated on the set applicable to design water flow rate. The curve associated dry-bulb temperature to the wet-bulb on each fixed relative humidity graph shall be included. Note that for the case where the airflow dampers to the dry section are fully closed, then the tower can be evaluated by the standard mechanical draft cooling tower method, Paragraph 7.3.

In the specific case of separate fans serving dry and wet portions of a tower or separate towers, the manufacturer shall submit dry surface pressure drop and heat transfer exponents as required for use in evaluating the fan driver output power exponent in Equation (10.2).

10.3 Determination of Tower Capability - The manufacturer's performance curves shall be cross plotted at test conditions to determine capability.

From the test wet-bulb temperature, relative humidity, and cooling range, a cross-plot relating outlet water temperature and predicted circulation rate will be prepared, requiring an intermediate step to cross-plot relative humidity at constant wet-bulb temperature and range. From this graph the predicted water circulation rate capability at the test outlet water temperature is determined. The determination of performance capability consists of comparing the predicted water flow rate from this curve with the adjusted water flow rate using equation (10.1) or and equation (10.2).

For single fans for both wet and dry sections, the following equation (10.1) would apply:

$$Q_{wt_{adj}} = Q_{wt}\left(\frac{\dot{W}_d}{\dot{W}_t}\right)^{1/3}\left(\frac{\rho_t}{\rho_d}\right)^{1/3} \qquad (10.1)$$

For the special case with separate fans moving air through the wet and dry portions of a single tower, or through separate wet and dry towers, the following version of equation (10.1), namely equation (10.2) would apply:

$$Q_{wt_{adj}} = Q_{wt}\left[\left(\frac{\dot{q}_{wet}}{\dot{q}_{tot}}\right)\left(\frac{\dot{W}_{d_{wet}}}{\dot{W}_{t_{wet}}}\right)^{1/3}\left(\frac{\rho_{t_{wet}}}{\rho_{d_{wet}}}\right)^{1/3} + \left(\frac{\dot{q}_{dry}}{\dot{q}_{tot}}\right)\left(\frac{\dot{W}_{d_{dry}}}{\dot{W}_{t_{wet}}}\right)^{y}\left(\frac{\rho_{t_{dry}}}{\rho_{d_{dry}}}\right)^{1/3}\right] \qquad (10.2)$$

where:

$\dot{q}_{wet}$ = Wet heat load (c_{pw} * L_t * (T_{on} -T_{off})) for the wet section, or tower

$\dot{q}_{dry}$ = Dry heat load (c_{pw} * L_t * (T_{on} -T_{off})) for the dry section, or tower

$\dot{q}_{tot}$= Total heat load ($\dot{q}_{wet} + \dot{q}_{dry}$)

$\dot{W}$ = Fan driver output power, for either wet($_{wet}$) or dry($_{dry}$) at either

test($_t$) or design($_d$) conditions

$$y = \frac{1}{\left[\begin{array}{c}(1 + dry\ surface\ air\ speed\ exponent\ for\ the\ pressure\ drop) * \\ (dry\ surface\ air\ speed\ exponent\ for\ heat\ transfer)\end{array}\right]}$$

The capability is then given by Equation (10.3):

$$C = \frac{Q_{wt_{adj}}}{Q_{w_{pred}}} \times 100 \qquad (10.3)$$

11. Mechanical-Draft Helper Cooling Towers – Performance Curve Method

11.1 Scope and Purpose - This section of the Code outlines a method for evaluation of the performance of mechanical draft helper cooling towers from test data using performance curves where inlet water temperature is specified. The results are expressed in terms of water cooling capability. A helper tower is defined as either a once-through tower or a cooling tower where that the entering water temperature is controlled by another process.

11.2 Manufacturer's Data - The manufacturer shall submit a family of performance curves consisting of a minimum of 3 sets of 3 curves each. One set shall apply to 90%, one to 100%, and the other to 110% of the design water circulation rate. Each set shall be presented as a plot of wet-bulb temperature as abscissa versus cold (leaving) water temperature as ordinate, with inlet (hot) water temperature as parameter. Graphical scaling shall be incremented so as to provide a minimum of 0.2°C (0.5°F) increments and no more than 0.1°C/mm (5°F/inch) for both wet-bulb and cold water temperatures. Curves shall be based on constant fan pitch angle. In addition to the design inlet (hot) water temperature curve, bracketing inlet (hot) water temperature curves of the maximum and minimum inlet temperature curves shall be shown as a minimum. It is highly recommended that intermediate inlet temperature curves be shown to minimize interpolation errors when evaluating performance. The curves shall fully cover (but not necessarily be limited to) the range of variables specified in Paragraph 2.3. The design point shall be shown on the appropriate curve, as well as the listing of the design point conditions of hot water temperature, cold water temperature, inlet wet-bulb temperature, inlet dry-bulb temperature (forced draft tower), design *L/G* (induced draft only), fan driver output power (fan motor output power, unless otherwise designated), fan motor efficiency, and either design barometric pressure or design basis elevation.

11.3 Determination of Tower Capability - The manufacturer's performance curve shall be cross plotted at test conditions to determine capability. A suitable procedure consists of first preparing a set of 3 curves, based on test wet-bulb temperature, relating inlet (hot) water temperature, cold water temperature, and water circulation rate. From this set of curves, a single curve is then prepared, based on the test inlet (hot) water temperature, relating cold water temperature to water circulation rate. The determination of performance capability consists of comparing the predicted water flow rate ($Q_{w,pred}$) from this curve with the adjusted test water flow rate ($Q_{wt,adj}$) using equations (11.1) and (11.2) following:

$$Q_{wt_{adj}} = Q_{wt} * \left(\frac{\dot{W}_d}{\dot{W}_t} \right)^{\left(\frac{1}{3}\right)} * \left(\frac{\rho_t}{\rho_d} \right)^{\left(\frac{1}{3}\right)} \qquad (11.1)$$

Where:

$Q_{wt_{adj}}$ = Circulating water flow adjusted for test conditions

Q_{wt} = Measured circulating water flow

$\dot{W}_d$ = Design fan power

$\dot{W}_t$ = Measured fan power during the test

ρ_t = Air density during test

ρ_d = Design air density

The capability, C, expressed in percent of design circulating flow is computed using Equation (11.2):

$$Capability = \frac{Q_{wt_{adj}}}{Q_{w_{pred}}} X\ 100 \qquad (11.2)$$

Where:

Capability = Tower thermal performance compared to design, %

$Q_{w_{pred}}$ = Predicted flow obtained using supplier's performance curves

$Q_{wt_{adj}}$ = As defined above

An example evaluation is given in section III, the Appendices.

PART III - APPENDICES

APPENDIX A

SI Example Evaluation of Mechanical-Draft Cooling Tower Test Using the Characteristic Curve Method

Section A1. General

The purpose of this appendix is to describe and illustrate the characteristic curve methodology for evaluating a thermal performance test on a mechanical draft cooling tower, as described in Section 5 of this code.

Section A2. Design and Test Conditions

Design and Test conditions for the mechanical draft cooling tower are summarized in the following table:

Table A-1. Mechanical Draft Cooling Tower, Design and Test Data

	Design	Test
Water flow rate (Q_w)	220 L/s	209 L/s
Hot Water Temp. (T_{hw})	36 °C	33.4 °C
Cold Water Temp. (T_{cw})	30 °C	27.1 °C
Inlet Wet Bulb Temp. (T_{wb})	25 °C	21.1 °C
Inlet Dry Bulb Temp. (T_{db})	31 °C	30.6 °C
Total Fan Driver Power ($\dot{W}$)	60 kW	57.6 kW
Barometric Pressure (P_{bp})	101.325 kPa	101.325 kPa
Liquid to Gas Ratio (L/G)	1.700	

In accordance with Section 5.2 of this code, the manufacturer has submitted the Characteristic Curve shown in Figure A-1.

Section A3. Evaluation Procedure

Step 1 – Determine the Test *L/G*.

The test value of *L/G* is computed using the test values of hot water temperature, cold water temperature, entering dry-bulb and wet-bulb temperatures, barometric pressure, test flow and test total fan driver power.

To comply with this code, the design and test values for density (ρ), specific volume (v), and enthalpy (h) of air must be determined using either the CTI ToolKit or psychrometric tables in ASHRAE Handbook - Fundamentals (SI).

Since the evaluation is based upon the psychrometric properties of air at the fan inlet, different procedures must be employed to compute the *L/G* for forced draft and induced draft towers.

a. Forced Draft Tower

For a forced draft tower, the fan inlet air conditions are the same as the tower inlet air conditions. Therefore, the test density (ρ_t) and the test specific volume (v_t) are computed directly from the measured test wet-bulb temperature, dry-bulb temperature and barometric pressure. The design conditions at the tower's air inlet shall be supplied by the manufacturer.

Given the Design and Test values for wet-bulb and dry-bulb temperatures and the barometric pressure, determine the density and specific volume of the inlet air.

At Design values of 25°C wet-bulb, 31°C dry-bulb, 101.325 kPa barometric pressure:

$\rho_d = 1.1485$ kg/m³

$v_d = 0.88609$ m³/kg

$h = 76.274$ kJ/kg

At Test values of 21.1°C wet-bulb, 30.6°C dry-bulb, 101.325 kPa barometric pressure:

$\rho_t = 1.1539$ kg/m³

$v_t = 0.87694$ m³/kg

$h = 61.102$ kJ/kg

Using these data, calculate the test $\left(\frac{L}{G}\right)_T$ from Equation (A-1).

$$\left(\frac{L}{G}\right)_t = \left(\frac{L}{G}\right)_d \left(\frac{Q_{wt}}{Q_{wd}}\right) \left(\frac{\dot{W}_d}{\dot{W}_t}\right)^{1/3} \left(\frac{\rho_t}{\rho_d}\right)^{1/3} \left(\frac{v_t}{v_d}\right) \quad (A-1)$$

where:

$(L/G)_d$ = Ratio of mass flow rate of water to air from the manufacturer's design data

v = Specific volume of air, either test($_t$) or design($_d$)

ρ = Density of air, either test($_t$) or design($_d$)

$\dot{W}$ = Fan driver output power, either test($_t$) or design($_d$)

Q_w = Circulating water flow rate, either test($_t$) or design($_d$)

Substituting values:

$$\left(\frac{L}{G}\right)_t = (1.70)\left(\frac{209}{220}\right)\left(\frac{60.0}{57.6}\right)^{1/3}\left(\frac{1.1539}{1.1485}\right)^{1/3}\left(\frac{0.87694}{0.88609}\right)$$

$$\left(\frac{L}{G}\right)_t = 1.623$$

b. Induced Draft Tower

For an induced draft tower, the conditions at the inlet of the fan are the tower's discharge conditions. The code requires that both the design and test, discharge air properties be determined by a heat balance calculation. Calculating *design* discharge air properties is a straightforward procedure while calculating *test* discharge air properties requires combining the heat balance Equation (A-2) with Equation (A-1) and iterating for a solution.

The heat balance equation below simply states that the heat gain of the air equals the heat loss of the water:

$$L * c_{pw} * (T_{hw} - T_{cw}) = G * (h_o - h_i) \quad (A-2)$$

where:

L = Circulating water mass flow rate, kg/s

c_{pw} = Specific heat of water = 4.186 kJ/kg×°C

T_{hw} = Hot water temperature, °C

T_{cw} = Cold water temperature, °C

G = Mass flow rate of dry air through the tower, kg dry air/hr

h_o = Enthalpy of air leaving the tower, kJ/kg

h_i = Enthalpy of air entering the tower, kJ/kg

Rearranging to separate the exit air enthalpy, the heat balance equation becomes:

$$h_o = \left(\frac{L}{G}\right) * c_{pw} * (T_{hw} - T_{cw}) + h_i \quad (A-3)$$

Determine the enthalpy of the entering air at design conditions as:

$$h_{id} = 76.274 \frac{kJ}{kg}$$

Then, substituting all design values into Equation (A-3), calculate h_{od}.

h_{od} = (1.70) *4.186*(36 - 30) + 76.274 or

$$h_{od} = 118.971 \frac{kJ}{kg}$$

Assuming the discharge air is saturated at this enthalpy, determine the design discharge air temperature, density (ρ_d) and specific volume (v_d) as:

T_a = 33.35°C

ρ_d = 1.1295 kg/m³

v_d = 0.91488 m³/kg

Next calculate the test L/G by substituting all known values into Equation (A-1).

$$\left(\frac{L}{G}\right)_t = (1.70)\left(\frac{209}{220}\right)\left(\frac{60.0}{57.6}\right)^{1/3}\left(\frac{\rho_t}{1.1295}\right)^{1/3}\left(\frac{v_t}{0.91488}\right)$$

$$\left(\frac{L}{G}\right)_t = 1.7183 * (\rho_t)^{\frac{1}{3}} * (v_t)$$

Next, using the measured test conditions, the enthalpy of the entering air is determined as:

$$h_{it} = 61.102 \frac{kJ}{kg}$$

This value along with the $(L/G)_t$ just computed and the measured water temperatures are now substituted into the heat balance equation (A-3):

$$h_{ot} = 1.7183 * (\rho_t)^{\frac{1}{3}} * v_t * 4.186 * (33.4 - 27.1) + 61.102$$

which then reduces to:

$$h_{ot} = 45.3147 * (\rho_t)^{\frac{1}{3}} * v_t + 61.102 \quad (A-4)$$

At this point, one must guess a discharge air temperature and, assuming the leaving air is saturated, determine h, ρ and v at that temperature. Then substitute these values into Equation (A-4) and solve for h_{ot}. Compare this calculated enthalpy, h_{ot}, to the actual enthalpy, h, of the discharge air at the assumed temperature. Continue iterating the discharge air temperature until a suitable temperature is selected that satisfies the heat balance expression.

For a first iteration, use the average of the test T_{hw} and T_{cw} as the leaving air temperature. Typical iteration values are given below.

T_a	ρ_t	v_t	h_{ot} actual	h_{ot} computed	Error
30.0	1.1459	0.89656	100.005	103.616	+3.60%
31.0	1.1410	0.90185	105.368	103.806	-1.48%
30.7	1.1425	0.90025	103.733	103.749	+0.015%

A leaving air temperature of 30.7°C is sufficiently accurate. Now, determine the $(L/G)_t$ by substituting the psychrometric values for saturated air at 30.7°C into Equation (A-1) and solve:

$$(L/G)_t = 1.7183 * (1.1425)^{\frac{1}{3}} * 0.90025$$

$$(L/G)_t = 1.617$$

Step 2 – Calculate *KaV/L* at the test *L/G*

Once the $(L/G)_t$ is determined, the procedure for calculating *KaV/L* is identical for forced draft and induced draft towers. The calculation is readily performed using CTI ToolKit, the significant portion of which is reproduced below. Continuing with the induced draft tower example, the value *KaV/L* is computed in the following manner:

For: T_{hw} = 33.4 and T_{cw} = 27.1°C

Then: $(T_{hw}-T_{cw})$ = (33.4 - 27.1) = 6.3°C

$h_2 = h_1 + c_{pw} \cdot (T_{hw}-T_{cw}) * (\frac{L}{G})$

h_2 = 61.102 + 4.186 × 6.3 × 1.617 = 103.745

= 0.1 * c_{pw} * $(T_{hw}-T_{cw})$ * $(\frac{L}{G})$

= 0.1 * 4.186 * 6.3 * 1.617 = 4.264

= 0.4 * c_{pw} * $(T_{hw}-T_{cw})$ * $(\frac{L}{G})$

= 0.4 * 4.186 * 6.3 * 1.617 = 17.057

= 0.1* $(T_{hw}-T_{cw})$ = 0.1 * 6.3 = 0.63

= 0.4* $(T_{hw}-T_{cw})$ = 0.4 * 6.3 = 2.52

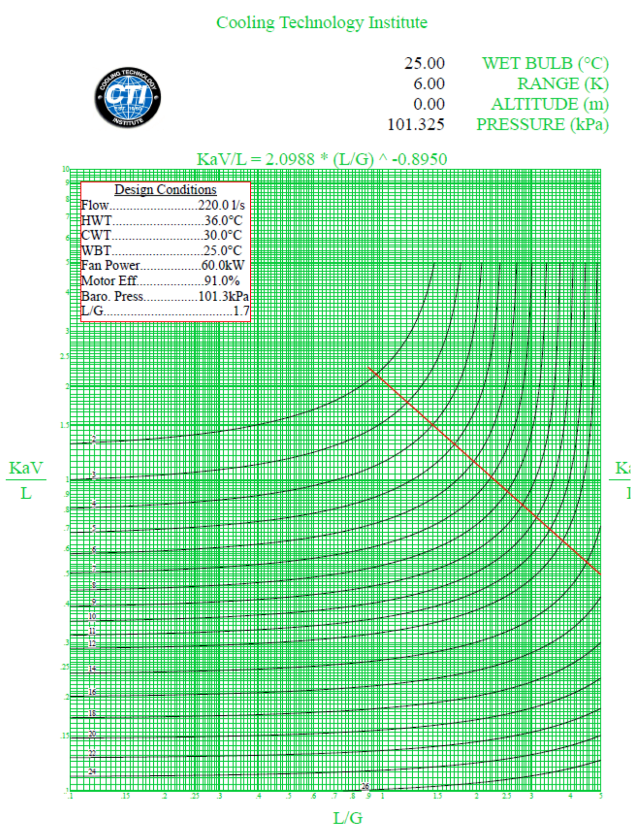
T, °C for h_w	h_w
T_{cw} = 27.1°C	
$T_{cw} + 0.1 * (T_{hw} - T_{cw}) = 27.73$	88.690
$T_{cw} + 0.4 * (T_{hw} - T_{cw}) = 29.62$	98.030
$T_{hw} - 0.4 * (T_{hw} - T_{cw}) = 30.88$	104.711
$T_{hw} - 0.1 * (T_{hw} - T_{cw}) = 32.77$	115.479
T_{hw} = 33.4°C	

h_a	$h_w - h_a$	$1/\Delta h$
$h_i = 61.102$		
$h_i + 0.1 * c_{pw}(T_{hw}\text{-}T_{cw}) * (\frac{L}{G}) = 65.659$	$\Delta h_1 = 23.031$	0.04342
$h_i + 0.4 * c_{pw}(T_{hw}\text{-}T_{cw}) * (\frac{L}{G}) = 78.452$	$\Delta h_2 = 19.578$	0.05108
$h_o - 0.4 * c_{pw}(T_{hw}\text{-}T_{cw}) * (\frac{L}{G}) = 86.981$	$\Delta h_3 = 17.731$	0.05640
$h_o - 0.1 * c_{pw}(T_{hw}\text{-}T_{cw}) * (\frac{L}{G}) = 99.774$	$\Delta h_4 = 15.705$	0.06367
$h_o = 104.038$		
	$\Sigma\ 1/\Delta h = \underline{0.21457}$	

$$\frac{KaV}{L} = c_{pw} * \left(\frac{T_{hw} - T_{cw}}{4}\right) * \sum\left(\frac{1}{\Delta h}\right)$$

$$\frac{KaV}{L} = 4.186 * \frac{6.3}{4} * 0.21457 = 1.415$$

Figure A-1. Manufacturer's Characteristic Curve with Design Point.



Step 3 – Determine the Interest $(L/G)_I$

At the $(L/G)_t$ calculated from Step 1 – Determine the Test *L/G*. and the *KaV/L* calculated from Step 2 – Calculate *KaV/L* at the test *L/G*, plot the test performance point on the manufacturer's characteristic curve, as shown in as shown in Figure A-2 below. A curve is drawn through this point parallel to the tower characteristic curve. This parallel curve intersects the 5°C design approach curve at $(L/G)_i = 1.72$. This is the predicted *L/G*, which is the *L/G* that the tower would produce if operating at design conditions.

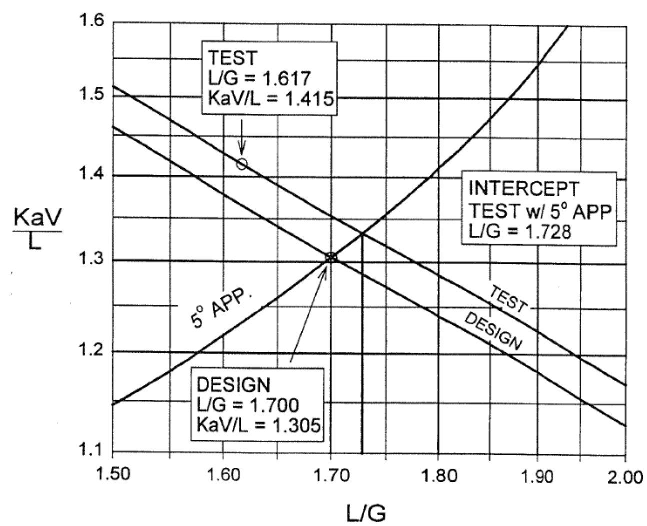
Step 4 – Calculate the Tower Capability

As set forth in Section 5.5, the tower capability, C, is the ratio of the predicted L/G to the design L/G

$$C = \frac{\left(\frac{L}{G}\right)_I}{\left(\frac{L}{G}\right)_d} * 100 \qquad (A-5)$$

$$C = \frac{1.728}{1.70} * 100 = 101.6\%$$

Figure A-2. Manufacturer's Characteristic Curve with Design Point and Test Point



Based on the test performed, the tower is capable of cooling 223.6 l/s from 36°C to 30°C at design wet-bulb temperature 25°C and design total fan driver power of 60 kW.

APPENDIX B

IP Example Evaluation of Mechanical-Draft Cooling Tower Test Using the Characteristic Curve Method

Section B1. General

The purpose of this appendix is to describe and illustrate the characteristic curve methodology for evaluating a thermal performance test on a mechanical draft cooling tower, as described in Section 5 of this code.

Section B2. Design and Test Conditions

Design and Test conditions for the mechanical draft cooling tower are summarized in the following table:

Table B-2. Mechanical Draft Cooling Tower, Design and Test Data

	Design	Test
Water flow rate (Q_w)	9500 gpm	9150 gpm
Hot Water Temp. (T_{hw})	115 °F	104.7 °F
Cold Water Temp. (T_{cw})	85 °F	79.3 °F
Inlet Wet Bulb Temp. (T_{wb})	80 °F	73.1 °F
Inlet Dry Bulb Temp. (T_{db})	90 °F	85.2 °F
Total Fan Driver Power ($\dot{W}$)	240 bhp	216 bhp
Barometric Pressure (P_{bp})	29.921 ”Hg	29.921 ”Hg
Liquid to Gas Ratio (L/G)	0.810	

In accordance with Section 5.2 of this code, the manufacturer has submitted the Characteristic Curve shown in Figure B-1.

Section B3. Evaluation Procedure

Step 1 – Determine the Test L/G.

The test value of *L/G* is computed using the test values of hot water temperature, cold water temperature, entering dry-bulb and wet-bulb temperatures, barometric pressure, test flow and test total fan driver power.

To comply with this code, the design and test values for density (ρ), specific volume (v), and enthalpy (h) of air must be determined using either the CTI ToolKit or psychrometric tables in ASHRAE Handbook - Fundamentals (IP).

Since the evaluation is based upon the psychrometric properties of air at the fan inlet, different procedures must be employed to compute the L/G for forced draft and induced draft towers.

a. Forced Draft Tower

For a forced draft tower, the fan inlet air conditions are the same as the tower inlet air conditions. Therefore the test density (ρ_t) and the test specific volume (v_t) are computed directly from the measured test wet-bulb temperature, dry-bulb temperature and barometric pressure. The design conditions at the tower's air inlet shall be supplied by the manufacturer.

Given the Design and Test values for wet-bulb and dry-bulb temperatures and the barometric pressure, determine the density and specific volume of the inlet air.

At Design values of 80□F wet-bulb, 90□F dry-bulb, and 29.921 ”Hg barometric pressure:

$\rho_d = 0.07131$ lbm/ft³

$v_d = 14.3025$ ft³/lbm

$h = 43.580$ Btu/lbm

At Test values of 73.1□F wet-bulb, 85.2□F dry-bulb, and 29.921 ”Hg barometric pressure:

$\rho_t = 0.07216$ lbm/ft³

$v_t = 14.0636$ ft³/lbm

$h = 36.711$ Btu/lbm

Using these data, calculate the test L/G from Equation (B-1).

$$\left(\frac{L}{G}\right)_t = \left(\frac{L}{G}\right)_d \left(\frac{Q_{wt}}{Q_{wd}}\right)\left(\frac{\dot{W}_d}{\dot{W}_t}\right)^{1/3} \left(\frac{\rho_t}{\rho_d}\right)^{1/3} \left(\frac{v_t}{v_d}\right) \quad (B-1)$$

where:

$\left(\frac{L}{G}\right)_d$ = Ratio of mass flow rate of water to air from the manufacturer's design data

v = Specific volume of air, either test($_t$) or design($_d$)

ρ = Density of air, either test($_t$) or design($_d$)

$\dot{W}$ = Fan driver output power, either test($_t$) or design($_d$)

Q_w = Circulating water flow rate, either test($_t$) or design($_d$)

Substituting values:

$$\left(\frac{L}{G}\right)_t = (0.81)\left(\frac{9150}{9500}\right)\left(\frac{240}{216}\right)^{1/3} \left(\frac{0.07216}{0.07131}\right)^{1/3} \left(\frac{14.0636}{14.3025}\right)$$

$$\left(\frac{L}{G}\right)_t = 0.7977$$

b. Induced Draft Tower

For an induced draft tower, the conditions at the inlet of the fan are the tower's discharge conditions. The code requires that both the design and test, discharge air properties be determined by a heat balance calculation. Calculating *design* discharge air properties is a straightforward procedure while calculating *test* discharge air properties requires combining the heat balance Equation (B-2) with Equation (B-1) and iterating for a solution.

The heat balance equation below simply states that the heat gain of the air equals the heat loss of the water:

$$L * c_{pw} * (T_{hw} - T_{cw}) = G * (h_o - h_i) \quad (B-2)$$

where:

- L = Circulating water mass flow rate, lbm/hr
- c_{pw}= Specific heat of water = 1.0 Btu/lbm·°F
- T_{hw}= Hot water temperature, °F
- T_{cw}= Cold water temperature, °F
- G = Mass flow rate of dry air through the tower, lbm dry air/hr
- h_o = Enthalpy of air leaving the tower, Btu/lbm
- h_i = Enthalpy of air entering the tower, Btu/lbm

Rearranging to separate the exit air enthalpy, the heat balance equation becomes:

$$h_o = \left(\frac{L}{G}\right) * c_{pw} * (T_{hw} - T_{cw}) + h_i \quad (B-3)$$

Determine the enthalpy of the entering air at design conditions as:

$$h_{id} = 43.58 \frac{BTU}{lbm}$$

Then, substituting all design values into Equation (B-3), calculate h_{od}.

h_{od} = (0.81) * 1.0 * (115 - 85) + 43.580 or

$$h_{od} = 67.880 \frac{BTU}{lbm}$$

Assuming the discharge air is saturated at this enthalpy and barometric pressure, determine the design discharge air temperature, density (ρ_d) and specific volume (v_d) as:

- T_a = 97.77 °F
- ρ_d = 0.06952 lbm/ft³
- v_d = 14.9624 ft³/lbm

Next calculate the test L/G by substituting all known values into Equation (B-1).

$$\left(\frac{L}{G}\right)_t = (0.81)\left(\frac{9150}{9500}\right)\left(\frac{240}{216}\right)^{\frac{1}{3}}\left(\frac{\rho_t}{0.06952}\right)^{\frac{1}{3}}\left(\frac{v_t}{14.9624}\right)$$

$$\left(\frac{L}{G}\right)_t = 0.13134 * (\rho_t)^{1/3} * (v_t)$$

Next, using the measured test conditions, the enthalpy of the entering air is determined as:

$$h_{it} = 36.711 \frac{BTU}{lbm}$$

This value along with the $(L/G)_t$ just computed and the measured water temperatures are now substituted into the heat balance equation (B-2):

$$h_{ot} = 0.13134 * (\rho_t)^{\frac{1}{3}} * v_t * 1.0 * (104.7 - 79.3) + 36.711$$

which then reduces to:

$$h_{ot} = 3.336 * (\rho_t)^{\frac{1}{3}} * v_t + 36.711 \quad (B-4)$$

At this point, one must guess a discharge air temperature and, assuming the leaving air is saturated, determine h, ρ and v at that temperature. Then substitute these values into Equation (B-4) and solve for h_{ot}. Compare this calculated enthalpy, h_{ot}, to the actual enthalpy, h, of the discharge air at the assumed temperature. Continue iterating the discharge air temperature until a suitable temperature is selected that satisfies the heat balance expression.

For a first iteration, use the average of the test T_{hw} and T_{cw} as the leaving air temperature. Typical iteration values are given below.

T_a	ρ_t	v_t	h_{ot} actual	h_{ot} computed	Error
92.0	0.07052	14.6532	58.794	56.907	-3.21%
90.0	0.07086	14.5528	55.950	56.801	+1.52%
90.6	0.07076	14.5826	56.788	56.832	+0.07%

A leaving air temperature of 90.6°F is sufficiently accurate. Now, determine the Test *L/G* by substituting the psychrometric values for saturated air at 90.6 °F into Equation (B-1) and solve:

$$(L/G)_t = 0.13134 * (0.07076)^{\frac{1}{3}} * 14.5826$$
$$(L/G)_t = 0.7922$$

Step 2 – Calculate *KaV/L* at the test *L/G*

Once the Test *L/G* is determined, the procedure for calculating *KaV/L* is identical for forced draft and induced draft towers. The calculation is readily performed using CTI form ATP-127, the significant portion of which is reproduced below. Continuing with the induced draft tower example, the value *KaV/L* is computed in the following manner:

For: T_{hw} = 104.7°F and T_{cw} = 79.3°F

Then: $(T_{hw} - T_{cw})$ = (104.7 – 79.3) = 25.4°F

$$h_2 = h_1 + c_{pw}(T_{hw} - T_{cw}) * \left(\frac{L}{G}\right)$$
$$h_2 = 36.827 + 1.0 * 25.4 * 0.7922 = 56.949$$
$$= 0.1 * c_{pw} * (T_{hw} - T_{cw}) * \left(\frac{L}{G}\right)$$
$$= 0.1 * 1.00 * 25.4 * 0.7922 = 2.012$$
$$= 0.4 * c_{pw} * (T_{hw} - T_{cw}) * \left(\frac{L}{G}\right)$$
$$= 0.4 * 1.00 * 25.4 * 0.7922 = 8.049$$
$$= 0.1 * (T_{hw} - T_{cw}) = 0.1 * 25.4 = 2.54$$
$$= 0.4 * (T_{hw} - T_{cw}) = 0.4 * 25.4 = 10.16$$

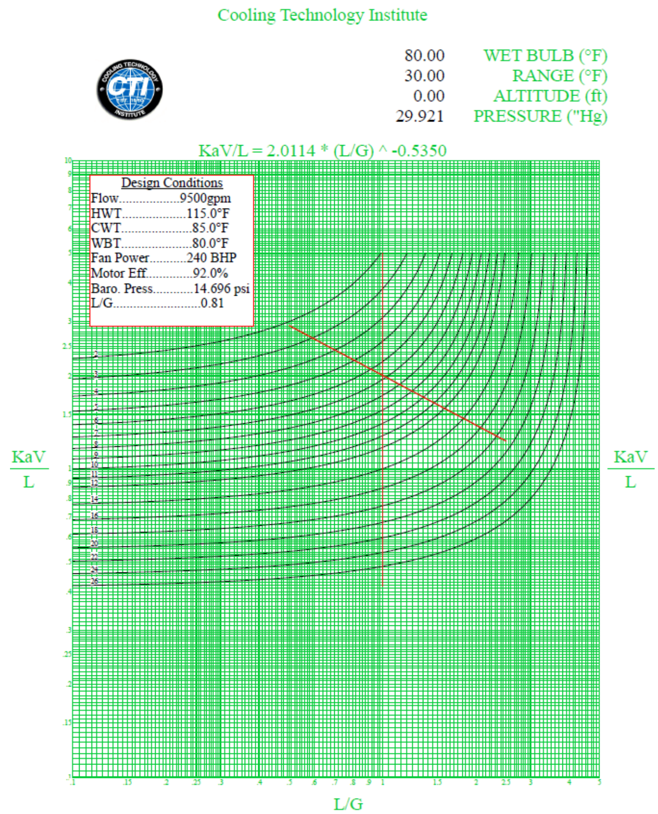
T, °F for h_w	h_w
T_{cw} = 79.3°F	
$T_{cw} + 0.1 * (T_{hw} - T_{cw}) = 81.84$	45.727
$T_{cw} + 0.4 * (T_{hw} - T_{cw}) = 89.46$	55.207
$T_{hw} - 0.4 * (T_{hw} - T_{cw}) = 94.54$	62.624
$T_{hw} - 0.1 * (T_{hw} - T_{cw}) = 102.16$	75.768
T_{hw} = 104.7°F	

h_a	$h_w - h_a$	$1/\Delta h$
$h_i = 36.827$		
$h_i + 0.1 * c_{pw}(T_{hw} - T_{cw}) * \left(\frac{L}{G}\right) = 38.840$	$\Delta h_1 = 6.887$	0.14519
$h_i + 0.4 * c_{pw}(T_{hw} - T_{cw}) * \left(\frac{L}{G}\right) = 44.876$	$\Delta h_2 = 10.330$	0.09680
$h_o - 0.4 * c_{pw}(T_{hw} - T_{cw}) * \left(\frac{L}{G}\right) = 48.901$	$\Delta h_3 = 13.724$	0.07287
$h_o - 0.1 * c_{pw}(T_{hw} - T_{cw}) * \left(\frac{L}{G}\right) = 54.937$	$\Delta h_4 = 20.831$	0.04880
$h_o = 56.949$		
	$\Sigma\left(\frac{1}{\Delta h}\right) = 0.36287$	

$$\frac{KaV}{L} = c_{pw} * \left(\frac{T_{hw} - T_{cw}}{4}\right) * \sum\left(\frac{1}{\Delta h}\right)$$

$$\frac{KaV}{L} = 1.00 * \frac{25.4}{4} * 0.36287 = 2.304$$

Figure B-2. Manufacturer's Characteristic Curve with Design Point.



Step 3 – Dertermine the Interest *(L/G)*$_I$

At the $(L/G)_t$ calculated from Step 1 – Determine the Test L/G and the KaV/L calculated from Step 2 – Calculate KaV/L at the test L/G, plot the test performance point on the manufacturer's characteristic curve, as shown in Figure B-2 below. A curve is drawn through this point parallel to the tower characteristic curve. This parallel curve intersects the 5°F design approach curve at $(L/G)_I = 0.8184$. This is the predicted L/G, which is the L/G that the tower would produce if operating at design conditions.

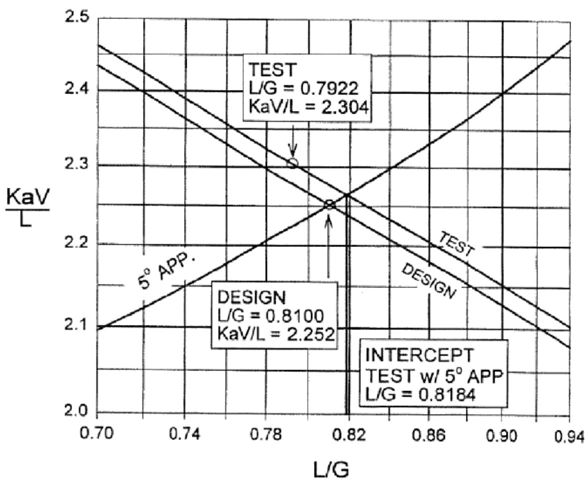
Step 4 – Calculate the Tower Capability

As set forth in Section 5.5, the tower capability, *C*, is the ratio of the predicted *L/G* to the design *L/G*.

$$C = \frac{\left(\frac{L}{G}\right)_I}{\left(\frac{L}{G}\right)_d} * 100 \qquad (B-5)$$

$$C = \frac{0.8184}{0.81} * 100 = 101.0\%$$

Figure B-2 Manufacturer's Characteristic Curve with Design Point and Test Point



Based on the test performed, the tower is capable of cooling 101.0% of design flow of 9594 gpm from 115 ºF to 85 ºF at design wet-bulb temperature 80 ºF and design total fan driver power of 240 bhp.

APPENDIX C
SI Example Evaluation of Mechanical-Draft Cooling Tower Test Using the Performance Curve Method

Section C1. General

The purpose of this appendix is to describe and illustrate the performance curve methodology for evaluating a thermal performance test on a mechanical draft cooling tower, as described in Section 7 of the standard.

Section C2. Design and Test Conditions

Design test conditions for a given mechanical draft cooling tower are summarized in the following table.

Parameter	Design	Test
Water Flow Rate (Q_{wt}) – L/s	3583	3623
Hot Water Temp. (T_{hw}) - °C	49.40	46.51
Cold Water Temp. (T_{cw}) - °C	30.60	29.04
Inlet Wet Bulb Temp. (T_{wb}) - °C	26.00	24.54
Inlet Dry Bulb Temp. (T_{db}) - °C	30.20	25.52
Fan Driver Power ($\dot{W}$) - kW	107.00	113.00
Barometric Pressure (P_{bp}) - kPa	101.325	98.86
Liquid-to-Gas Ratio	1.300	

In accordance with paragraph 7.2 of this standard, the manufacturer has submitted performance curves with Cold Water presented as a function of Wet Bulb Temperature with Range and Water Flow Rate as parameters (see Figures C-1, C-2 and C-3).

Section C3. Evaluation Procedure

The steps to be followed in evaluating the test are as follows:

Step 1. Determine the Predicted Cold Water Temperatures

Scribe the submitted performance curves vertically at the test wet bulb temperature (24.54°C) to determine the predicted Cold Water Temperatures associated with the test Wet Bulb Temperature at each of the three Range and three water Flow Rate conditions included on the performance curves. Values of these predicted Cold Water Temperatures are tabulated in Table C-2

Table C-2 Curve Points for Cold Water Temperature vs Range at 24.54°C Test Wet Bulb

Range	90% Flow	100% Flow	110% Flow
15.04°C	28.27	28.97	29.89
18.8°C	28.84	29.65	30.68
22.56°C	29.37	30.27	31.41

Step 2. First Crossplot

These values of Cold Water Temperature are plotted against Ranges using Flow Rate as a parameter (see Figure C-4).

Scribe Figure C-4 vertically at the test cooling Range (17.47°C) to determine the predicted Cold Water Temperatures of each of the three circulating Flow Rates. The values of these predicted Cold Water Temperatures are tabulated in Table C-3.

Table C-3 Curve Points for Cold Water Temperature vs Water Flow at 24.54°C Test Wet Bulb and 17.47°C Test Range

90 % Flow	100% Flow	110% Flow
28.64	29.41	30.41

Step 3. Second Crossplot

Using the values of Cold Water Temperature in Table C-3, plot these temperatures as a function of the three flow rate percentages (see Figure C-5).

Step 4. Determine the Predicted Flow Rate

On Figure C-5 scribe a horizontal line at the test Cold Water Temperature to intersect the curve. At the intersection, project a line vertically downward to find the predicted circulating water Flow Rate associated with the test Wet Bulb, Range and Cold Water Temperature.

$Q_{w,pred}$ = 95.35% of Design Flow = 3416.5 L/s

Step 5. Calculate the Adjusted Test Water Flow Rate

The adjusted test circulating water flow rate is computed from equation 7.1, using the values for air density at the fan inlet and the fan driver output power, at both the test and design conditions.

To comply with this standard, the design and test values for density (ρ), specific volume (ν) and enthalpy (h) of air must be determined using either CTI ToolKit or the psychrometric tables in ASHRAE Handbook – Fundamentals (SI). For this example, the CTI Toolkit was used to generate all psychrometric properties. Since the evaluation is based on the psychrometric properties of air at the fan inlet, different procedures must be utilized for forced draft and induced draft towers.

a. Forced Draft Tower

For a forced draft tower, the fan inlet air conditions are the same as the tower inlet air conditions. Therefore, the test density (ρ_t) and the test specific volume (ν_t) are computed directly from the measured test values of wet bulb, dry bulb and barometric pressure. In accordance with paragraph 7.2, the manufacturer shall supply the design conditions at the tower's air inlet.

Table C-4. Design and Test Values for Air Characteristics

Parameter	**Design**	**Test**
Barometric Pressure (P_{bp})	101.325 kPa	98.86 kPa
Wet Bulb Temp. (T_{wb})	26.00°C	24.54°C
Dry Bulb Temp. (T_{db})	30.20°C	25.52°C
Humidity Ratio (HR)	0.01966 kg/kg	0.01970 kg/kg
Specific Volume (ν)	0.8860 m^3/kg	0.8951 m^3/kg
Enthalpy (h_i)	80.6308 kJ/kg	75.8551 kJ/kg
Density (ρ)	1.15013 kg/m^3	1.13920 kg/m^3
Relative Humidity	71.98%	92.44%

b. Induced Draft Tower

For the induced draft tower, the fan air conditions are the tower discharge conditions. The code requires that both the design and test discharge air properties be determined by a heat balance calculation. Calculating design discharge air properties is a direct procedure while calculating **test** discharge air properties requires combining the heat balance, Equation (C-2), with Equation C-1 and iterating to a solution.

The heat balance equation states that the heat gain by the air equals the heat loss of the water. The formula follows:

$$L * C_{pw} * (T_{hw} - T_{cw}) = G * (h_o - h_i) \qquad (C-1)$$

Where:

L = Water mass flow rate, kg/hr

C_{pw} = Specific heat of water = 4.186 kJ/kg.K

T_{hw} = Hot water temperature, °C

T_{cw} = Cold water temperature, °C

G = Mass flow rate of dry air through the tower, kg/hr

h_o = Enthalpy of air leaving the tower, kJ/kg

h_i = Enthalpy of air entering the tower, kJ/kg

Rearranging to separate the exit air enthalpy, the heat balance equation becomes:

$$h_o = \left(\frac{L}{G}\right) * C_{pw} * (T_{hw} - T_{cw}) + h_i \qquad (C-2)$$

Determine the enthalpy of the entering air at design conditions as:

$$h_{id} = 80.6308 \frac{\text{kJ}}{\text{kg}}$$

Then substituting all design values into Equation (C-2), calculate h_{od}.

$$h_{od} = 1.300 * 4.186 * (49.4 – 30.6) + 80.6308$$

$$h_{od} = 182.937 \frac{\text{kJ}}{\text{kg}}$$

Assuming the discharge air is saturated at this enthalpy and barometric pressure, determine the design discharge air temperature, density (ρ_d) and specific volume (ν_d) as:

T_a= 41.86°C

ρ_d = 1.0863 kg/m^3

ν_d = 0.9708 m^3/kg

Next, calculate the discharge air characteristics at test conditions. First, calculate the test L/G by substituting all known values into Equation 5.1:

$$\frac{L}{G} = 1.300 * \left(\frac{3623}{3583}\right) * \left(\frac{\rho_t}{1.0863}\right)^{\frac{1}{3}} * \left(\frac{107.0}{113.0}\right)^{\frac{1}{3}} * \left(\frac{\nu_t}{0.9708}\right)$$

$$\left(\frac{L}{G}\right)_t = 1.2935 * (\rho_t)^{\frac{1}{3}} * \nu_t$$

Substitute this L/G expression into Equation (C-2) to calculate the test exit air enthalpy, h_{ot}

$$h_{ot} = 1.2935 * (\rho_t)^{\frac{1}{3}} * \nu_t * 4.186 * (46.51 – 29.04) + 75.8551$$

At this point, guess at a discharge air temperature and, assuming saturation, determine ρ and ν at that temperature. Then, substituting these values for ρ and ν into the final heat balance expression calculate an h_{ot}

Compare the calculated value for h_{ot} to the actual value of enthalpy at the assumed temperature and continue iterating discharge air temperature until a suitable temperature is selected for which the calculated value of h_{ot} matches the actual value.

For the first estimate of leaving air temperature, use the average of T_{hw} and T_{cw} at test conditions. Typical iteration values for this example are given in Table C-5 for Barometric Pressure of 98.86 kPa.

Table C-5. Iteration for Enthalpy of Leaving Air

T_a	ρ_t	ν_t	h_{ot} actual	h_{ot} computed	Error %
38	1.07815	0.9692	153.766	169.459	+10.21
40	1.06809	0.9835	170.139	170.544	+0.24
40.5	1.06554	0.9873	174.488	170.834	+2.09
40.1	1.06763	0.9842	170.914	170.598	+0.18

Step 6. Determine Cooling Tower Capability

Substituting the psychrometric values at 40.1°C, the adjusted flow rate is now calculated from Equation 7.1.

$$Q_{wt_{adj}} = 3623 * \left(\frac{107.0}{113.0}\right)^{\frac{1}{3}} * \left(\frac{1.06763}{1.08631}\right)^{\frac{1}{3}} = 3537 \frac{L}{s}$$

The tower capacity is then computed from equation 7.2.

$$C = \left(\frac{3537}{3416.5}\right) X100 = 103.5\%$$

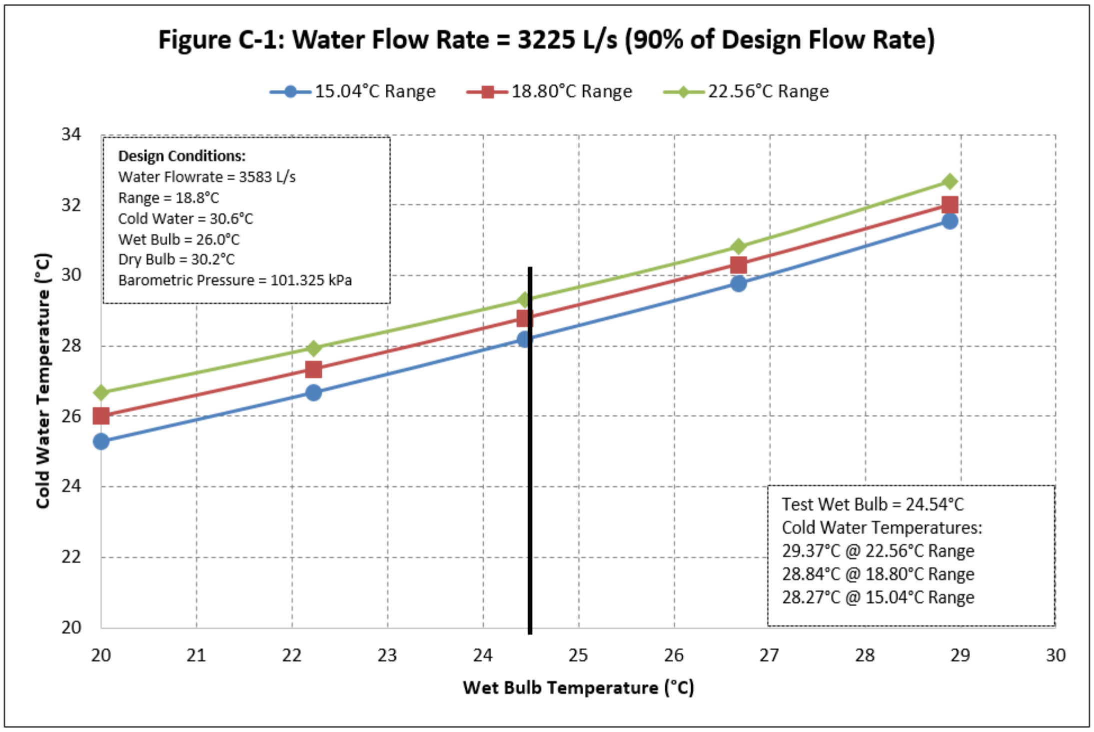


Figure C-1: Water Flow Rate = 3225 L/s (90% of Design Flow Rate)
15.04°C Range
18.80°C Range
22.56°C Range
Design Conditions:
Water Flowrate = 3583 L/s
Range = 18.8°C
Cold Water = 30.6°C
Wet Bulb = 26.0°C
Dry Bulb = 30.2°C
Barometric Pressure = 101.325 kPa
Test Wet Bulb = 24.54°C
Cold Water Temperatures:
29.37°C @ 22.56°C Range
28.84°C @ 18.80°C Range
28.27°C @ 15.04°C Range
Cold Water Temperature (°C)
Wet Bulb Temperature (°C)
34
32
30
28
26
24
22
20
20
21
22
23
24
25
26
27
28
29
30

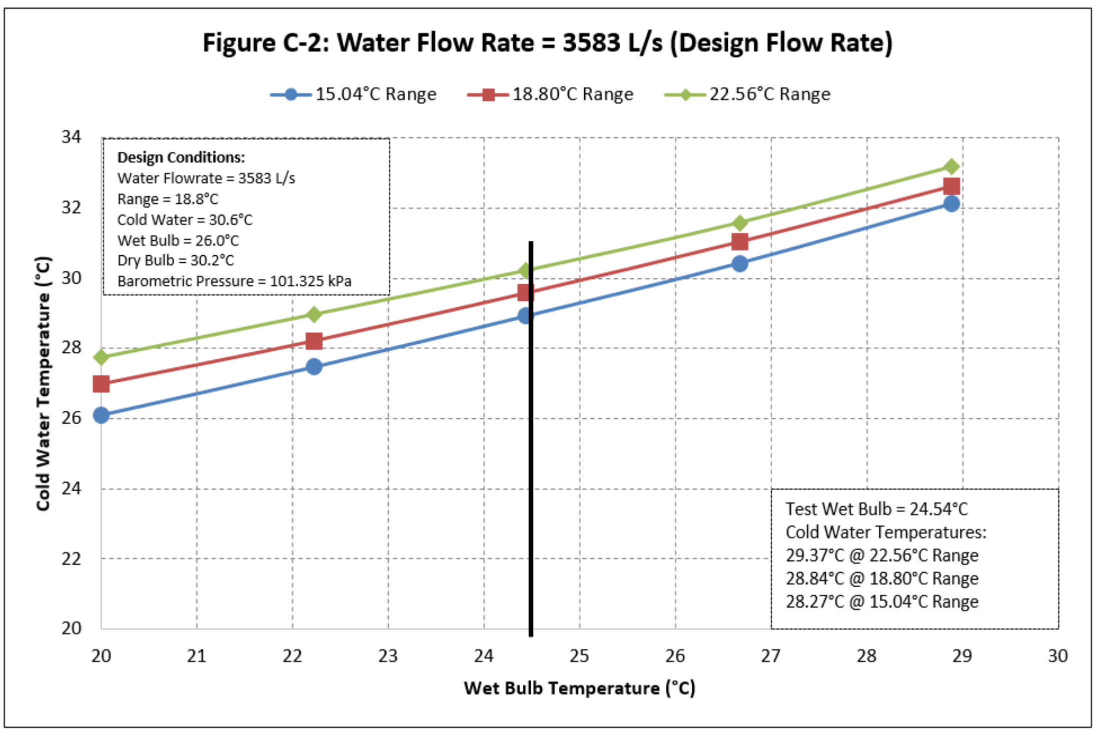


Figure C-2: Water Flow Rate = 3583 L/s (Design Flow Rate)
15.04°C Range
18.80°C Range
22.56°C Range
Design Conditions:
Water Flowrate = 3583 L/s
Range = 18.8°C
Cold Water = 30.6°C
Wet Bulb = 26.0°C
Dry Bulb = 30.2°C
Barometric Pressure = 101.325 kPa
Test Wet Bulb = 24.54°C
Cold Water Temperatures:
29.37°C @ 22.56°C Range
28.84°C @ 18.80°C Range
28.27°C @ 15.04°C Range
Cold Water Temperature (°C)
Wet Bulb Temperature (°C)
34
32
30
28
26
24
22
20
20
21
22
23
24
25
26
27
28
29
30

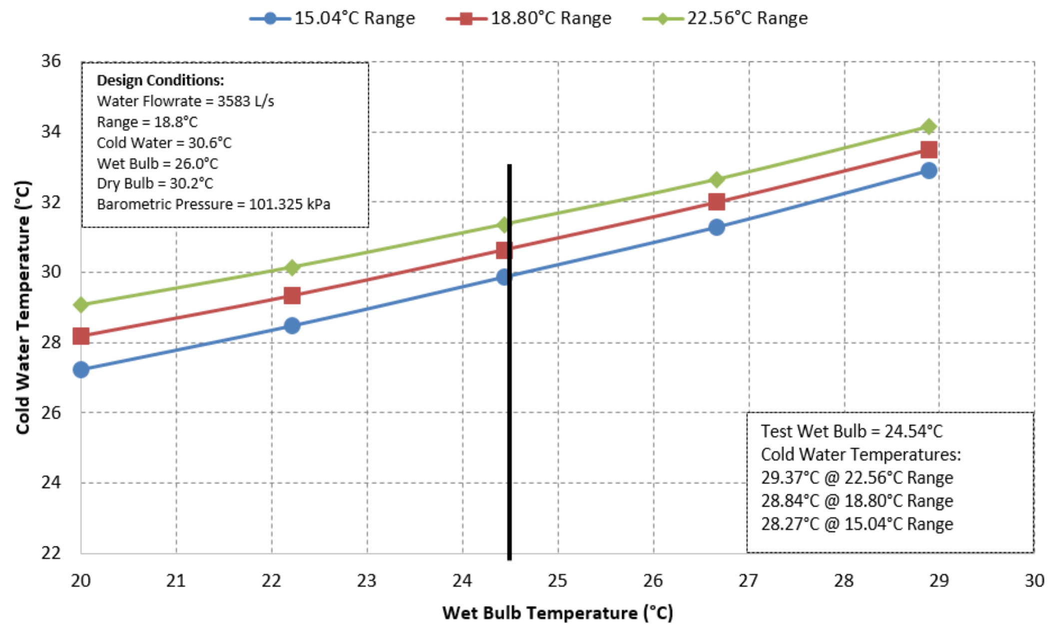


Figure C-3: Water Flow Rate = 3941 L/s (110% of Design Flow Rate)

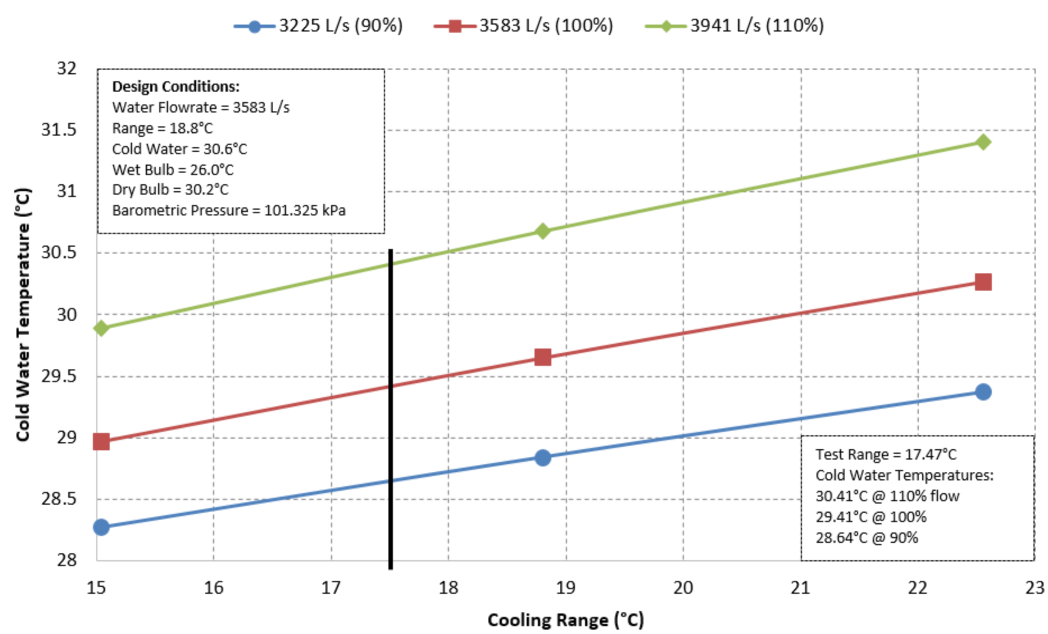


Figure C-4: Crossplot-1, Wet Bulb 24.54°C

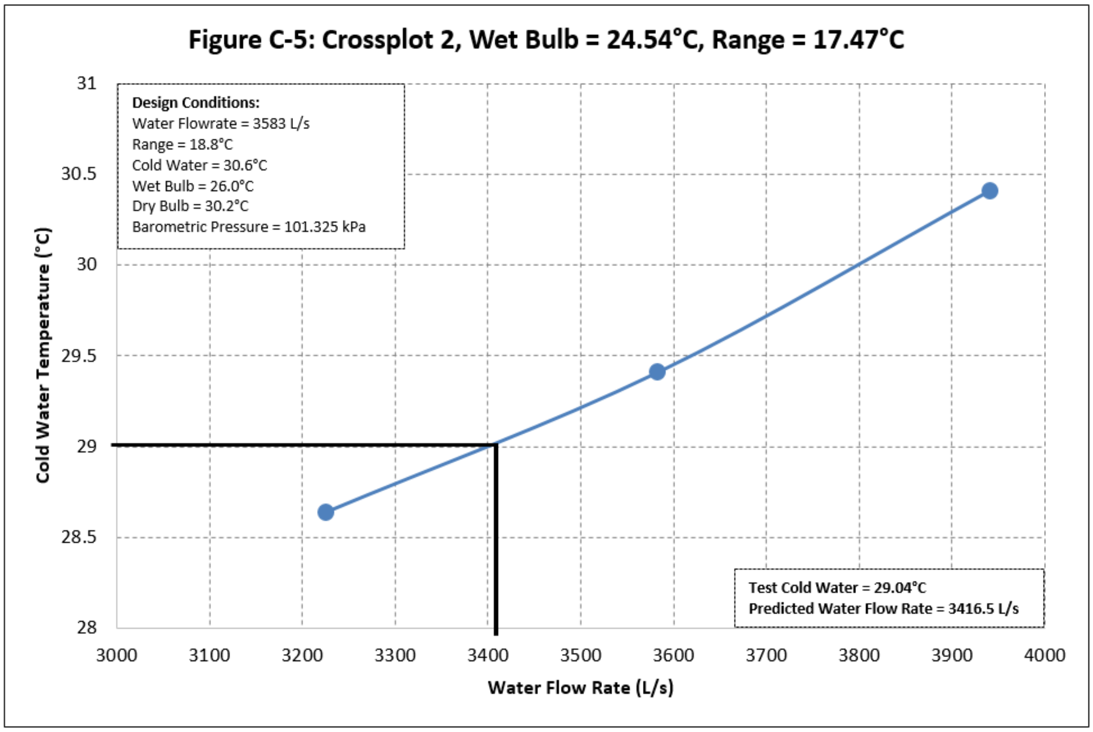


Figure C-5: Crossplot 2, Wet Bulb = 24.54°C, Range = 17.47°C
Design Conditions:
Water Flowrate = 3583 L/s
Range = 18.8°C
Cold Water = 30.6°C
Wet Bulb = 26.0°C
Dry Bulb = 30.2°C
Barometric Pressure = 101.325 kPa
Test Cold Water = 29.04°C
Predicted Water Flow Rate = 3416.5 L/s
Cold Water Temperature (°C)
Water Flow Rate (L/s)
31
30.5
30
29.5
29
28.5
28
3000
3100
3200
3300
3400
3500
3600
3700
3800
3900
4000

APPENDIX D

IP Example Evaluation of Mechanical-Draft Cooling Tower Test Using the Performance Curve Method

Section D1. General

The purpose of this appendix is to describe and illustrate the performance curve methodology for evaluating a thermal performance test on a mechanical draft cooling tower, as described in Section 7 of the standard.

Section D2. Design and Test Conditions

Design test conditions for a given mechanical draft cooling tower are summarized in the following table.

Parameter	Design	Test
Water Flow Rate (Q_{wt}) - gpm	56,792	57,426
Hot Water Temp. (T_{hw}) - °F	120.92	115.7
Cold Water Temp. (T_{cw}) - °F	87.08	84.27
Inlet Wet Bulb Temp. (T_{wb}) - °F	78.80	76.18
Inlet Dry Bulb Temp. (T_{db}) - °F	86.36	77.94
Fan Driver Power (W) - bhp	143.40	151.50
Barometric Pressure (P_{bp}) - "Hg	29.921	29.18
Liquid-to-Gas Ratio	1.300	

In accordance with paragraph 7.2 of this standard, the manufacturer has submitted performance curves with Cold Water presented as a function of Wet Bulb Temperature with Range and Water Flow Rate as parameters (see Figures D-1, D-2 and D-3).

Section D3. Evaluation Procedure

The steps to be followed in evaluating the test are as follows:

Step 1. Determine the Predicted Cold Water Temperatures

Scribe the submitted performance curves vertically at the test wet bulb temperature (76.18°F) to determine the predicted Cold Water Temperatures associated with the test Wet Bulb Temperature at each of the three Range and three water Flow Rate conditions included on the performance curves. Values of these predicted Cold Water Temperatures are tabulated in Table D-2

Table D-2 Curve Points for Cold Water Temperature vs Range at 76.18°F Test Wet Bulb

Range	90% Flow	100% Flow	110% Flow
27.07°F	82.88°F	84.14°F	85.81°F
33.84°F	83.91°F	85.37°F	87.23°F
40.62°F	84.86°F	86.49°F	88.53°F

Step 2. First Crossplot

These values of Cold Water Temperature are plotted against Ranges using Flow Rate as a parameter (see Figure D-4).

Scribe Figure D-4 vertically at the test cooling Range (31.43°F) to determine the predicted Cold Water Temperatures of each of the three circulating Flow Rates. The values of these predicted Cold Water Temperatures are tabulated in Table D-3.

Table D-3 Curve Points for Cold Water Temperature vs Water Flow at 76.18°F Test Wet Bulb and 31.43°F Test Range

90 % Flow	100% Flow	110% Flow
83.55	84.94	86.73

Step 3. Second Crossplot

Using the values of Cold Water Temperature in Table D-3, plot these temperatures as a function of the three flow rate percentages (see Figure D-5).

Step 4. Determine the Predicted Flow Rate

On Figure D-5 scribe a horizontal line at the test Cold Water Temperature to intersect the curve. At the intersection, project a line vertically downward to find the predicted circulating water Flow Rate associated with the test Wet Bulb, Range and Cold Water Temperature.

$$Q_{w,pred} = 95.35\% \text{ of Design Flow} = 54{,}152 \text{ gpm}$$

Step 5. Calculate the Adjusted Test Water Flow Rate

The adjusted test circulating water flow rate is computed from equation 7.1, using the values for air density at the fan inlet and the fan driver output power, at both the test and design conditions.

To comply with this standard, the design and test values for density (ρ), specific volume (ν) and enthalpy (h) of air must be determined using either CTI ToolKit or the psychrometric tables in ASHRAE Handbook – Fundamentals (IP). For this example, the CTI Toolkit was used to generate all psychrometric properties. Since the evaluation is based on the psychrometric properties of air at the fan inlet, different procedures must be utilized for forced draft and induced draft towers.

Since the evaluation is based on the psychrometric properties of air at the fan inlet, different procedures must be utilized for forced draft and induced draft towers.

a. Forced Draft Tower

For a forced draft tower, the fan inlet air conditions are the same as the tower inlet air conditions. Therefore, the test density (ρ_t) and the test specific volume (ν_t) are computed directly from the measured test values of wet bulb, dry bulb and barometric pressure. In accordance with paragraph 7.2, the manufacturer shall supply the design conditions at the tower's air inlet.

Table D-4 Design and Test Values for Air Characteristics

Parameter	Design	Test
Barometric Pressure (P_{bp})	29.921" Hg	29.18" Hg
Wet Bulb Temp. (T_{wb})	78.80°F	76.18°F
Dry Bulb Temp. (T_{db})	86.36°F	77.94°F
Humidity Ratio (HR)	0.01971lb/lb	0.01935 lb/lb
Specific Volume (ν)	14.2009 ft^3/lb	14.3382 ft^3/lb
Enthalpy (h_i)	42.3329 BTU/lb	40.2958 BTU/lb
Density (ρ)	0.07180 lb/ft^3	0.07112 lb/ft^3
Relative Humidity	71.95%	92.44%

b. Induced Draft Tower

For the induced draft tower, the fan air conditions are the tower discharge conditions. The code requires that both the design and test discharge air properties be determined by a heat balance calculation. Calculating **design** discharge air properties is a direct procedure while calculating **test** discharge air properties requires combining the heat balance, Equation (D-2), with Equation 5.1 and iterating to a solution.

The heat balance equation states that the heat gain by the air equals the heat loss of the water. The formula follows:

$$L * C_{pw} * (T_{hw} - T_{cw}) = G * (h_o - h_i) \qquad (D-1)$$

Where:

L = Water mass flow rate, lbm/hr

C_{pw} = Specific heat of water = 1.0 BTU/lbm.°F

T_{hw} = Hot water temperature, °F

T_{cw} = Cold water temperature, °F

G = Mass flow rate of dry air through the tower, lbm/hr

h_o = Enthalpy of air leaving the tower, BTU/lbm

h_i = Enthalpy of air entering the tower, BTU/lbm

Rearranging to separate the exit air enthalpy, the heat balance equation becomes:

$$h_o = \left(\frac{L}{G}\right) * C_{pw} * (T_{hw} - T_{cw}) + h_i \qquad (D-2)$$

Determine the enthalpy of the entering air at design conditions as:

$$h_{id} = 42.333 \frac{BTU}{lbm}$$

Then substituting all design values into Equation (D-2), calculate h_{od}.

$$h_{od} = 1.300 * (120.92 - 87.08) + 42.333$$

$$h_{od} = 86.325 \frac{BTU}{lbm}$$

Assuming the discharge air is saturated at this enthalpy and barometric pressure, determine the design discharge air temperature, density (ρ_d) and specific volume (ν_d) as:

T_a= 107.323°F

ρ_d = 0.06782 lbm/ft^3

ν_d = 15.5500 ft^3/lbm

Next, calculate the discharge air characteristics at test conditions. First, calculate the test L/G by substituting all known values into Equation 5.1:

$$\frac{L}{G} = 1.300 * \left(\frac{57.426}{56.792}\right) * \left(\frac{\rho_t}{0.06782}\right)^{\frac{1}{3}} * \left(\frac{143.4}{151.5}\right)^{\frac{1}{3}} * \left(\frac{\nu_t}{15.550}\right)$$

$$\left(\frac{L}{G}\right)_t = 0.20353 * (\rho_t)^{\frac{1}{3}} * \nu_t$$

Substitute this L/G expression into Equation (D-2) to calculate the test exit air enthalpy, h_{ot}

$$h_{ot} = 0.20353 * (\rho_t)^{\frac{1}{3}} * \nu_t * 1.0 * (115.7 - 84.27) + 40.2958$$

At this point, guess at a discharge air temperature and, assuming saturation, determine ρ and ν at that temperature. Then, substituting these values for ρ and ν into the final heat balance expression calculate an h_{ot}

Compare the calculated value for h_{ot} to the actual value of enthalpy at the assumed temperature and continue iterating discharge air temperature until a suitable temperature is selected for which the calculated value of h_{ot} matches the actual value.

For the first estimate of leaving air temperature, use the average of T_{hw} and T_{cw} at test conditions. Typical iteration values for this example are given in Table D-5 for Barometric Pressure of 29.18" Hg.

Table D-5. Iteration for Enthalpy of Leaving Air

T_a	ρ_t	ν_t	h_{ot} actual	h_{ot} computed	Error %
100	0.06738	15.5006	73.0653	80.6437	+10.37
105	0.06650	15.8207	82.9340	81.2980	-1.97
104	0.06668	15.7542	80.8517	81.1617	+0.383
104.16	0.06665	15.7648	81.1810	81.1833	+0.003

Step 6. Determine Cooling Tower Capability

Substituting the psychrometric values at 104.16°F, the adjusted flow rate is now calculated from Equation 7.1.

$$Q_{wt_{adj}} = 57{,}426 * \left(\frac{143.4}{151.5}\right)^{\frac{1}{3}} * \left(\frac{0.06665}{0.06782}\right)^{\frac{1}{3}} = 56{,}057 gpm$$

The tower capacity is then computed from equation 7.2.

$$C = \left(\frac{56{,}057}{54{,}152}\right) X100 = 103.5\%$$

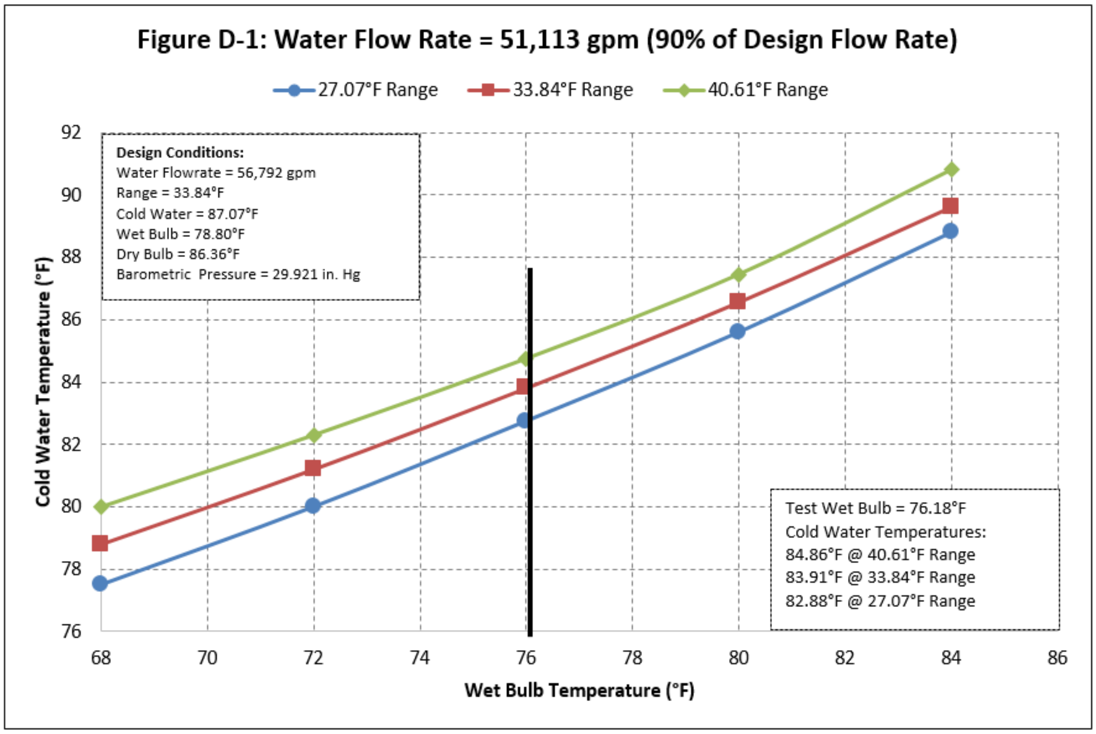


Figure D-1: Water Flow Rate = 51,113 gpm (90% of Design Flow Rate)
27.07°F Range
33.84°F Range
40.61°F Range
Design Conditions:
Water Flowrate = 56,792 gpm
Range = 33.84°F
Cold Water = 87.07°F
Wet Bulb = 78.80°F
Dry Bulb = 86.36°F
Barometric Pressure = 29.921 in. Hg
Test Wet Bulb = 76.18°F
Cold Water Temperatures:
84.86°F @ 40.61°F Range
83.91°F @ 33.84°F Range
82.88°F @ 27.07°F Range
Cold Water Temperature (°F)
Wet Bulb Temperature (°F)

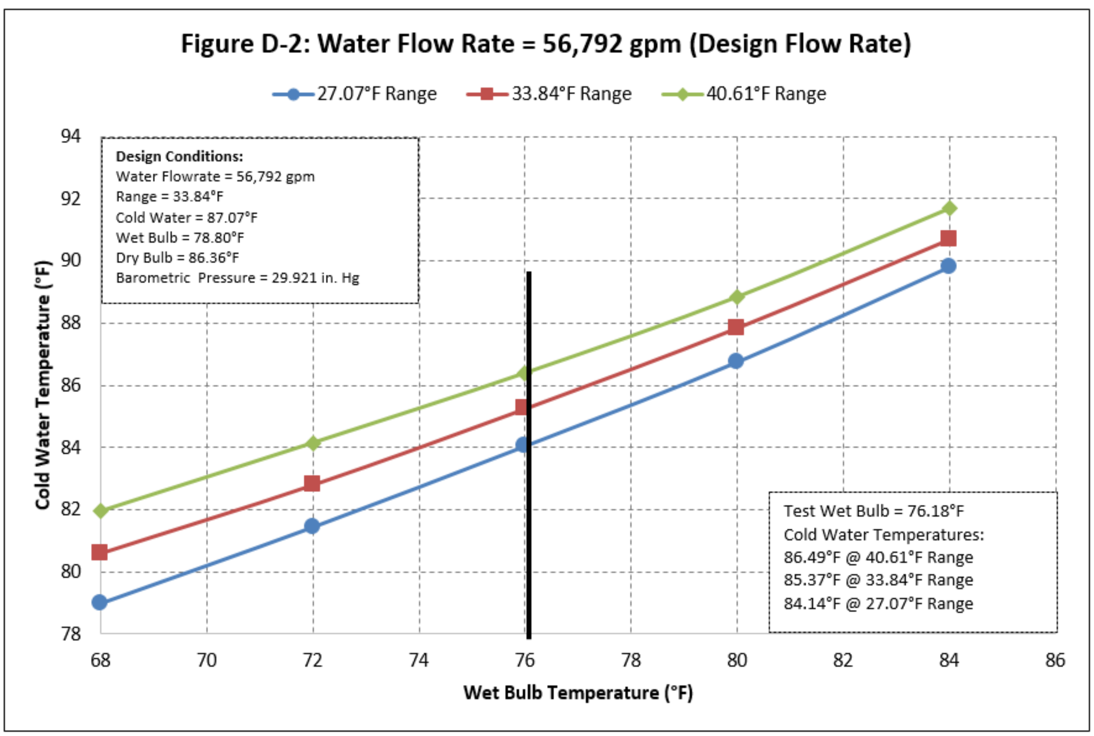


Figure D-2: Water Flow Rate = 56,792 gpm (Design Flow Rate)
27.07°F Range
33.84°F Range
40.61°F Range
Design Conditions:
Water Flowrate = 56,792 gpm
Range = 33.84°F
Cold Water = 87.07°F
Wet Bulb = 78.80°F
Dry Bulb = 86.36°F
Barometric Pressure = 29.921 in. Hg
Test Wet Bulb = 76.18°F
Cold Water Temperatures:
86.49°F @ 40.61°F Range
85.37°F @ 33.84°F Range
84.14°F @ 27.07°F Range
Cold Water Temperature (°F)
Wet Bulb Temperature (°F)

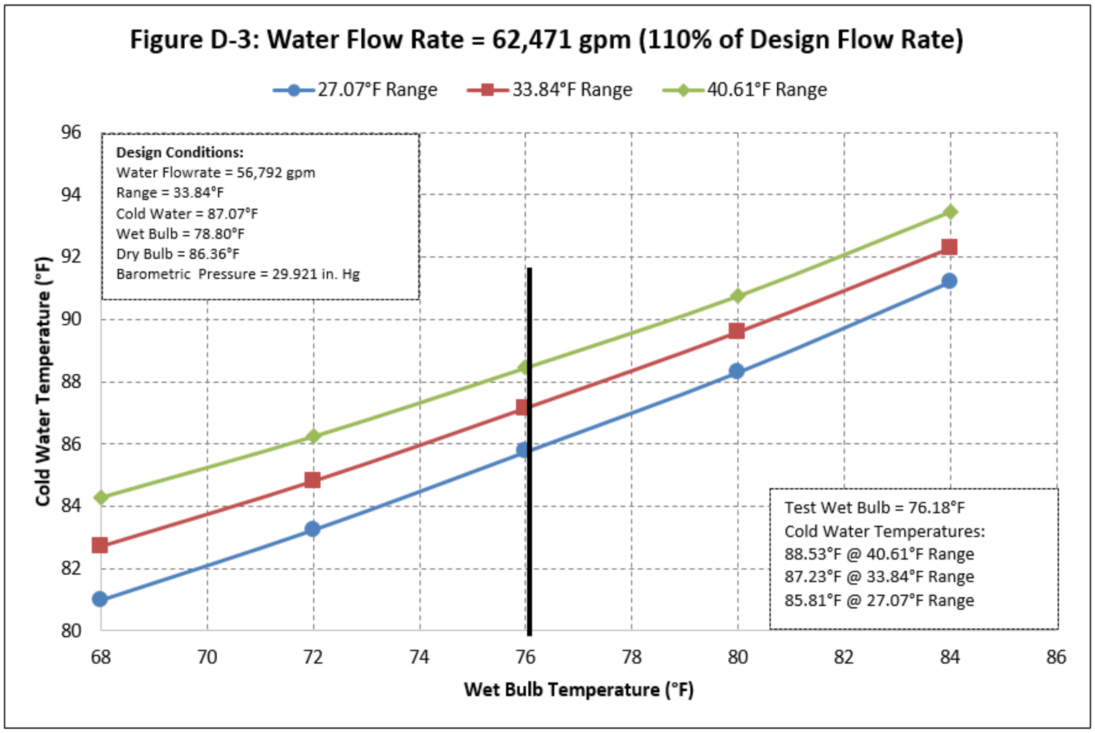


Figure D-3: Water Flow Rate = 62,471 gpm (110% of Design Flow Rate)
27.07°F Range
33.84°F Range
40.61°F Range
Design Conditions:
Water Flowrate = 56,792 gpm
Range = 33.84°F
Cold Water = 87.07°F
Wet Bulb = 78.80°F
Dry Bulb = 86.36°F
Barometric Pressure = 29.921 in. Hg
Test Wet Bulb = 76.18°F
Cold Water Temperatures:
88.53°F @ 40.61°F Range
87.23°F @ 33.84°F Range
85.81°F @ 27.07°F Range
Cold Water Temperature (°F)
Wet Bulb Temperature (°F)

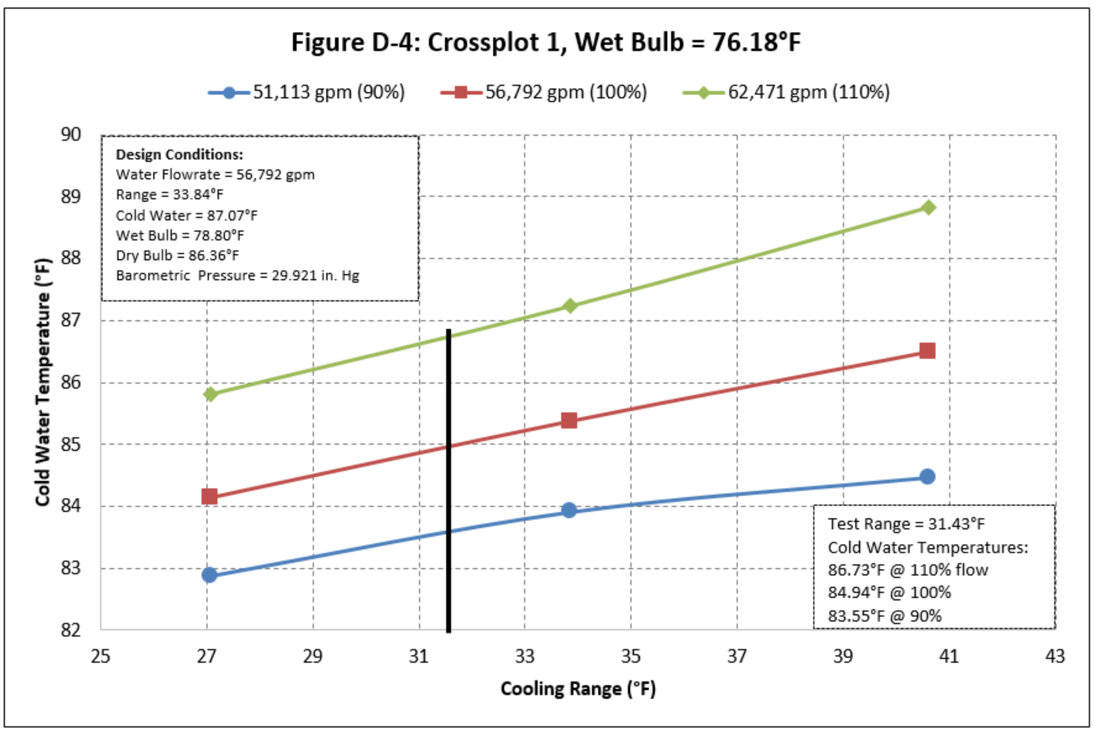


Figure D-4: Crossplot 1, Wet Bulb = 76.18°F
51,113 gpm (90%)
56,792 gpm (100%)
62,471 gpm (110%)
Design Conditions:
Water Flowrate = 56,792 gpm
Range = 33.84°F
Cold Water = 87.07°F
Wet Bulb = 78.80°F
Dry Bulb = 86.36°F
Barometric Pressure = 29.921 in. Hg
Test Range = 31.43°F
Cold Water Temperatures:
86.73°F @ 110% flow
84.94°F @ 100%
83.55°F @ 90%
Cold Water Temperature (°F)
Cooling Range (°F)

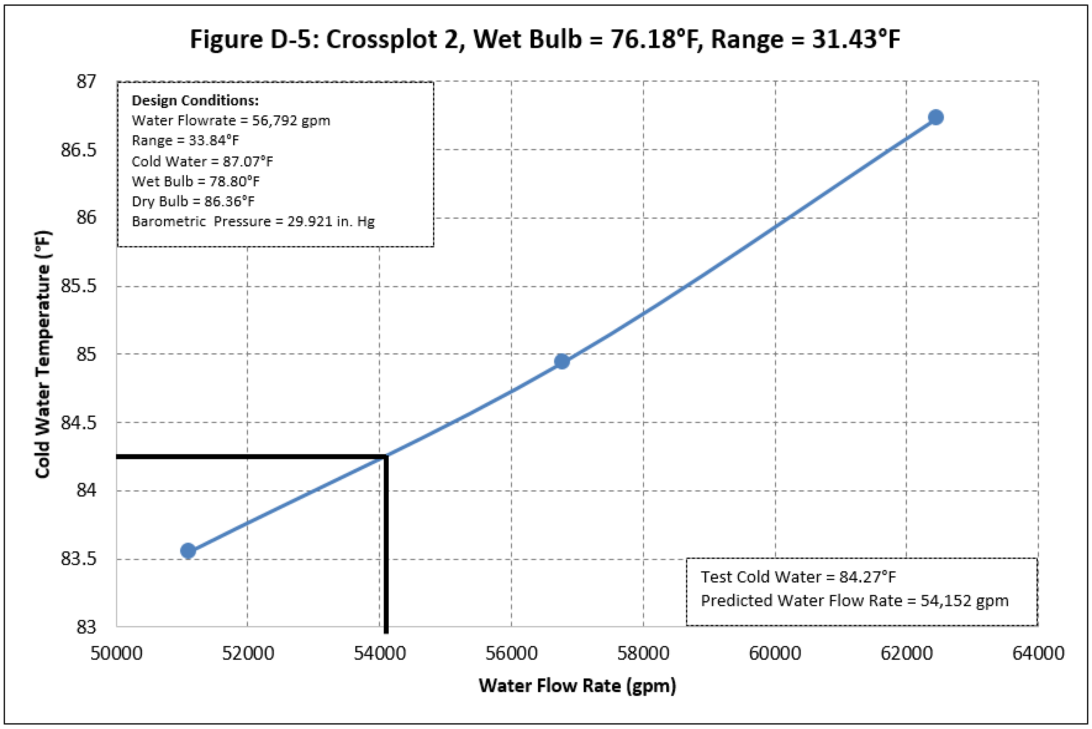


Figure D-5: Crossplot 2, Wet Bulb = 76.18°F, Range = 31.43°F
Design Conditions:
Water Flowrate = 56,792 gpm
Range = 33.84°F
Cold Water = 87.07°F
Wet Bulb = 78.80°F
Dry Bulb = 86.36°F
Barometric Pressure = 29.921 in. Hg
Test Cold Water = 84.27°F
Predicted Water Flow Rate = 54,152 gpm
Cold Water Temperature (°F)
87
86.5
86
85.5
85
84.5
84
83.5
83
50000
52000
54000
56000
58000
60000
62000
64000
Water Flow Rate (gpm)

APPENDIX E
SI Example Evaluation of Natural-Draft Cooling Tower Test Using the Performance Curve Method

Section E1. General

The purpose of this appendix is to describe and illustrate the performance curve methodology for evaluating a thermal performance test on a natural draft cooling tower, as described in Section 8 of the standard.

Section E2. Design and Test Conditions

Design and test conditions for the natural draft cooling tower are summarized in the Table E-1 below:

Table E-1 Natural Draft Cooling Tower Design and Test Data

	Design	Test
Water Flow Rate (W)	23 889 L/s	22 299 L/s
Hot Water Temp (T_{hw})	32.80 °C	27.80 °C
Cold Water Temp (T_{cw})	25.40 °C	20.50 °C
Cooling Range (R)	7.40 °C	7.30 °C
Wet Bulb Temp (T_{wb})	16.00 °C	9.50 °C
Dry Bulb Temp (T_{db})	18.20 °C	13.40 °C
Barometric Pressure(P_{bp})	101.325 kPa	103.70 kPa
Relative humidity (RH)	80.17%	60.42%

In accordance with Paragraph 8.2 of this standard, the manufacturer has submitted Performance Curves presenting Cold Water Temperature as a function of the air Dry Bulb Temperature with the Relative Humidity of the air as a parameter (see Figures E-1 to E-9).

E-1 to E-3 correspond to 90 % of design water circulating rate;

E-4 to E-6 correspond to 100 % of design water circulating rate;

E-7 to E-9 correspond to 110 % of design water circulating rate.

Section E3. Evaluation Procedure

The five steps to be followed in evaluating the tower performance capability are as follows.

Step 1 - Determine the Predicted Cold Water Temperatures

Using the nine Performance Curves, three for each of the three water circulation rates, enter the curves at the test Dry Bulb Temperature (13.40°C) and determine the Cold Water Temperature for 60%. 80%, and 100% Relative Humidity at each Flow Rate and Range.

Table E-2 Predicted Cold Water Temperatures at 13.4°C Entering Dry Bulb Temperature.

Flow	Range	60% RH	80% RH	100% RH
90%	6.7°C	20.24	21.10	21.95
	7.4°C	20.46	21.31	22.16
	8.1°C	20.59	21.44	22.30
100%	6.7°C	21.11	21.94	22.78
	7.4°C	21.38	22.20	23.03
	8.1°C	21.56	22.39	23.23
110%	6.7°C	21.94	22.78	23.63
	7.4°C	22.17	23.02	23.86
	8.1°C	22.41	23.24	24.06

Step 2 - First Crossplot

Then for each Flow Rate, prepare a crossplot of Cold Water Temperature as a function of the Relative Humidity, with the cooling Range as a parameter (see figures E-10 through E-12).

Step 3 - Second Crossplot

Using these new curves, enter each at the Test Relative Humidity (60.42%) and determine the Cold Water Temperature for each Flow Rate and Range.

Table E-3 Predicted Cold Water Temperatures at 13.4°C Entering Dry Bulb Temperature and 60.42%RH

Range	90% Flow	100% Flow	110% Flow
6.7°C	20.26	21.13	21.95
7.4°C	20.48	21.39	22.19
8.1°C	20.61	21.58	22.43

Then, for each Flow Rate, develop a crossplot of the Cold Water Temperature as a function of the cooling Range, (see Figure E-13).

Step 4 - Third Crossplot

Enter Figure E-13 at the Test Range (7.30°C) and determine the Cold Water Temperature for each of the three Flow Rates.

Table E-4. Predicted Cold Water Temperatures at 13.4°C Entering Dry Bulb Temperature, 60.42% RH, and 7.3°C Range.

90% Flow	100% Flow	110% Flow
20.45	21.36	22.16

Then crossplot the water Flow Rate as a function of the Cold Water Temperature (see Figure E-14).

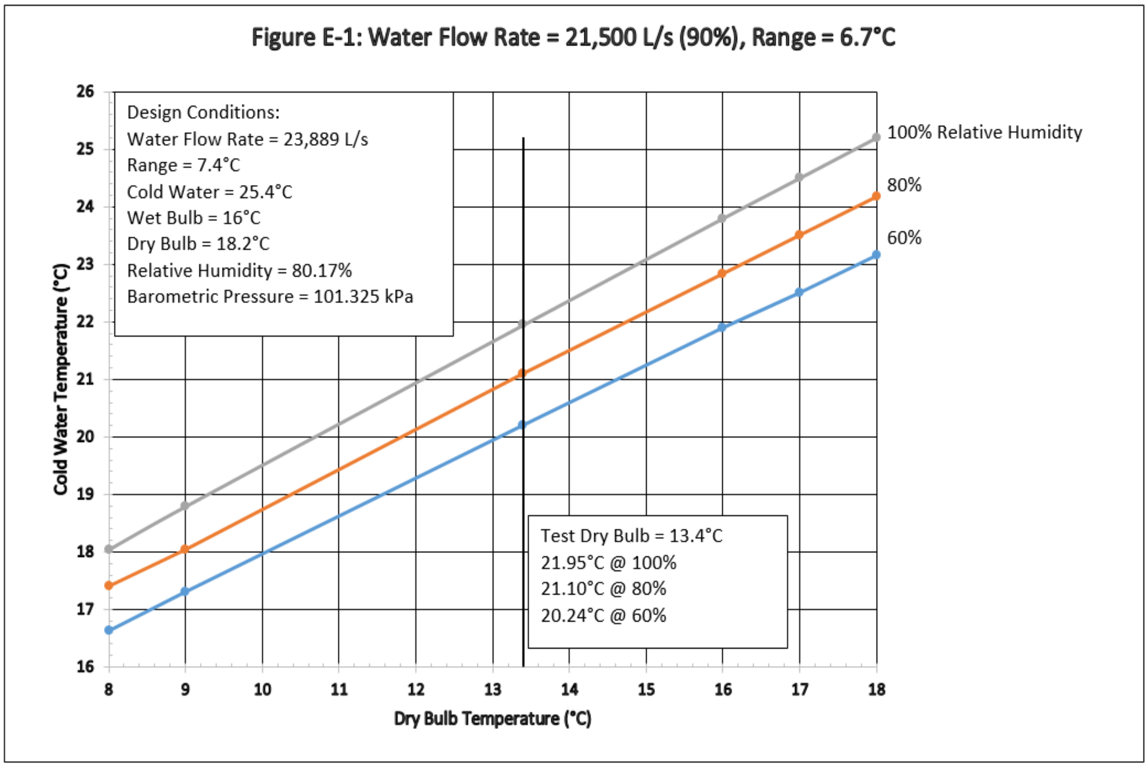
Step 5 - Determine Predicted Flow Rate

Enter Fig E-14 at the Measured Cold Water Temperature (20.50°C) and from the intersection with the curve, determine the Predicted Water Flow Rate at the Test Cold Water Temperature as 21 639 L/s.

Step 6 - Determine Cooling Tower Capability

Using Equation 8.1, find the cooling tower Thermal Performance Capability as:

$$\%Capability = \left(\frac{22299\ L/s}{21639\ L/s}\right) * (100) = 103.1\%$$



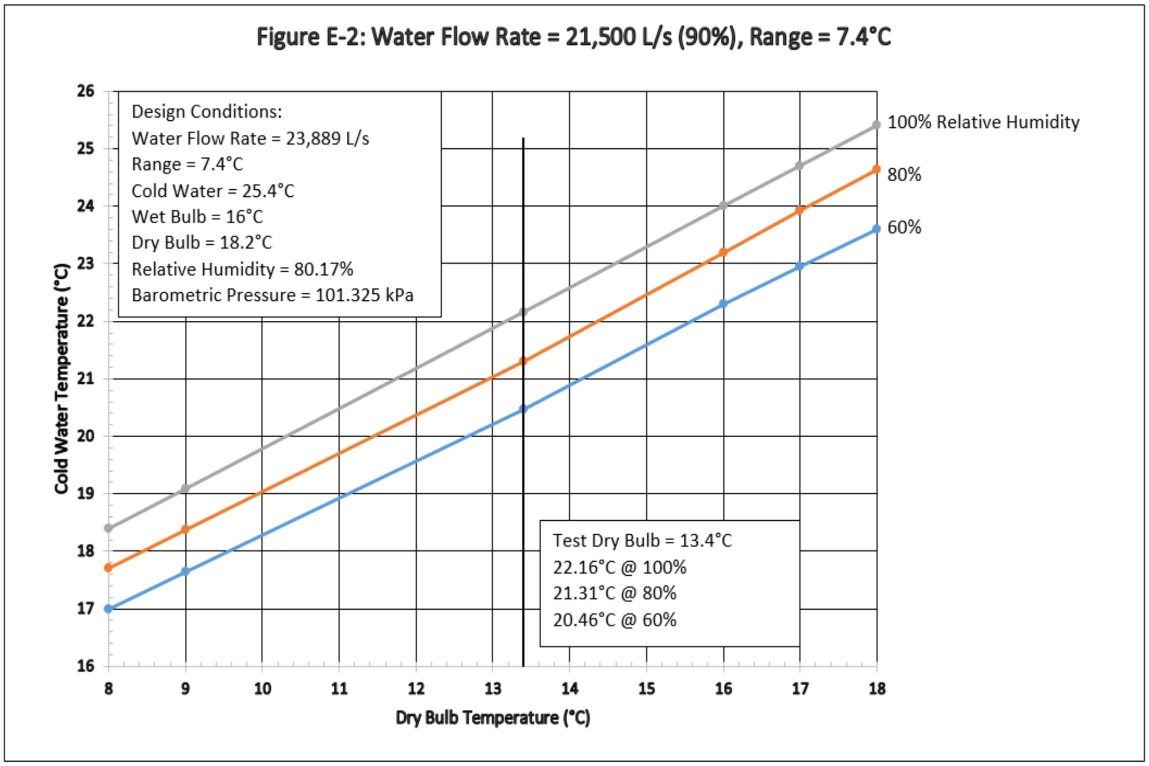


Figure E-2: Water Flow Rate = 21,500 L/s (90%), Range = 7.4°C
Design Conditions:
Water Flow Rate = 23,889 L/s
Range = 7.4°C
Cold Water = 25.4°C
Wet Bulb = 16°C
Dry Bulb = 18.2°C
Relative Humidity = 80.17%
Barometric Pressure = 101.325 kPa
Test Dry Bulb = 13.4°C
22.16°C @ 100%
21.31°C @ 80%
20.46°C @ 60%
100% Relative Humidity
80%
60%
Cold Water Temperature (°C)
Dry Bulb Temperature (°C)
16
17
18
19
20
21
22
23
24
25
26
8
9
10
11
12
13
14
15
16
17
18

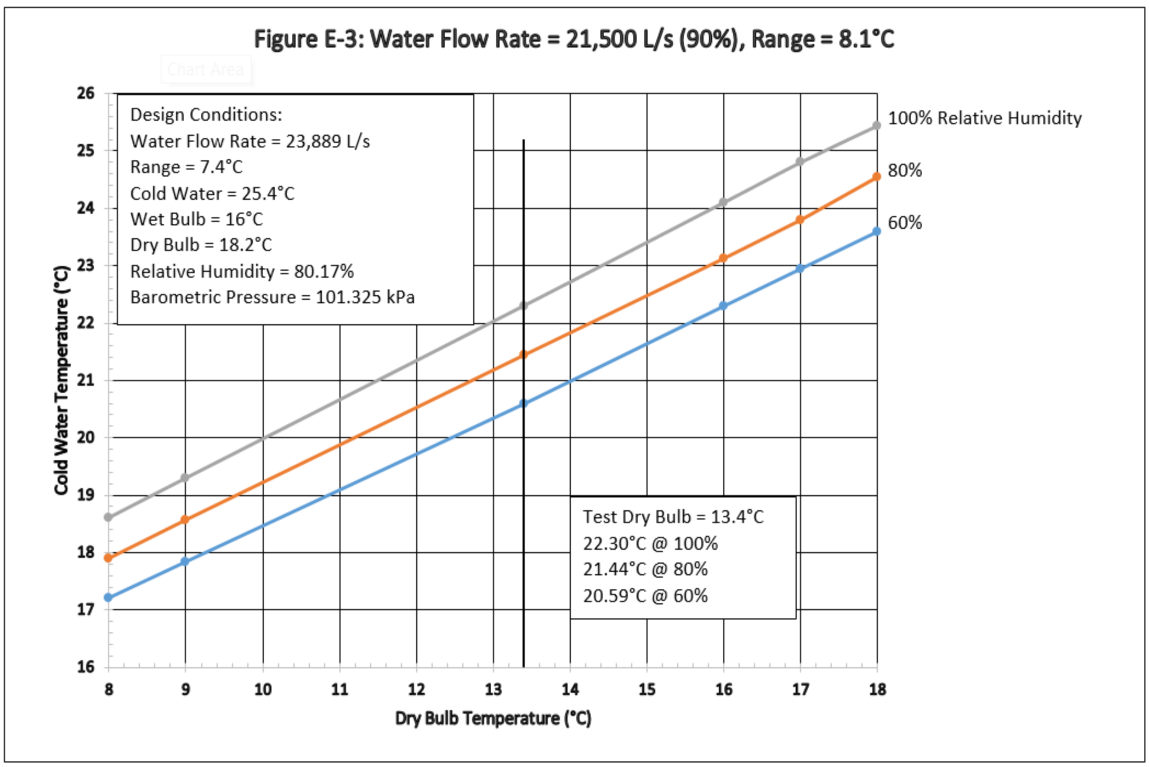


Figure E-3: Water Flow Rate = 21,500 L/s (90%), Range = 8.1°C
Design Conditions:
Water Flow Rate = 23,889 L/s
Range = 7.4°C
Cold Water = 25.4°C
Wet Bulb = 16°C
Dry Bulb = 18.2°C
Relative Humidity = 80.17%
Barometric Pressure = 101.325 kPa
Test Dry Bulb = 13.4°C
22.30°C @ 100%
21.44°C @ 80%
20.59°C @ 60%
100% Relative Humidity
80%
60%
Cold Water Temperature (°C)
Dry Bulb Temperature (°C)
16
17
18
19
20
21
22
23
24
25
26
8
9
10
11
12
13
14
15
16
17
18

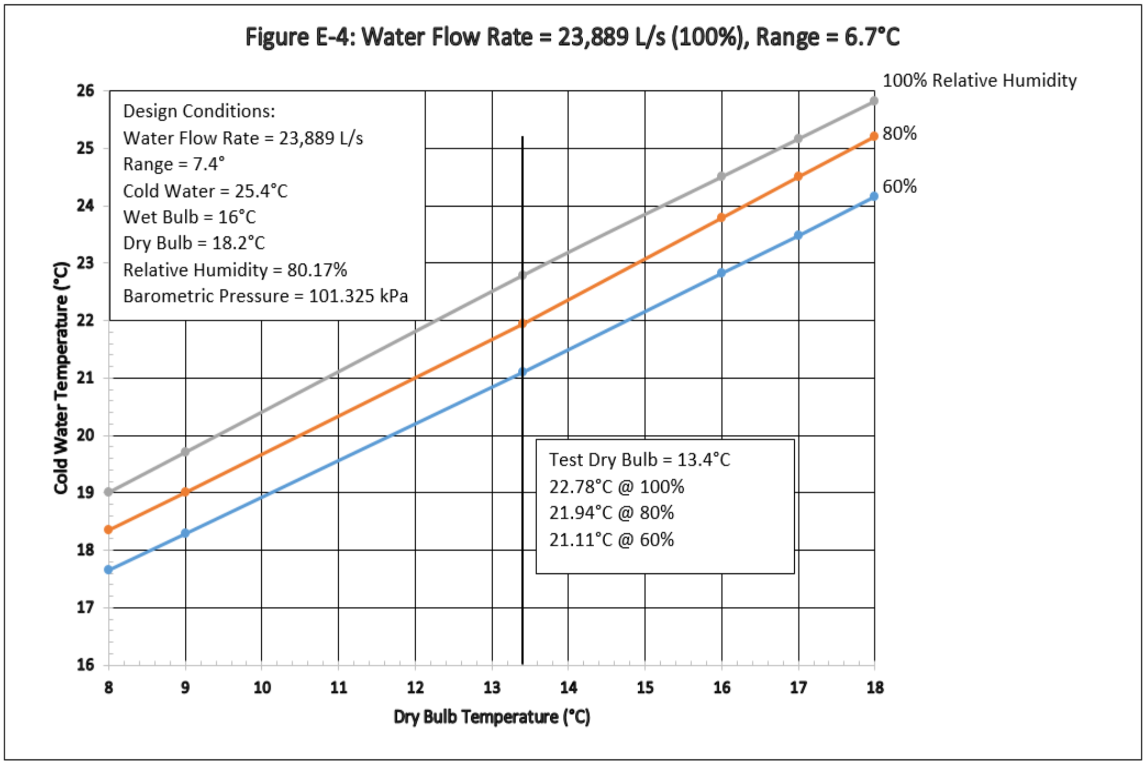


Figure E-4: Water Flow Rate = 23,889 L/s (100%), Range = 6.7°C
Design Conditions:
Water Flow Rate = 23,889 L/s
Range = 7.4°
Cold Water = 25.4°C
Wet Bulb = 16°C
Dry Bulb = 18.2°C
Relative Humidity = 80.17%
Barometric Pressure = 101.325 kPa
Test Dry Bulb = 13.4°C
22.78°C @ 100%
21.94°C @ 80%
21.11°C @ 60%
100% Relative Humidity
80%
60%
Cold Water Temperature (°C)
Dry Bulb Temperature (°C)

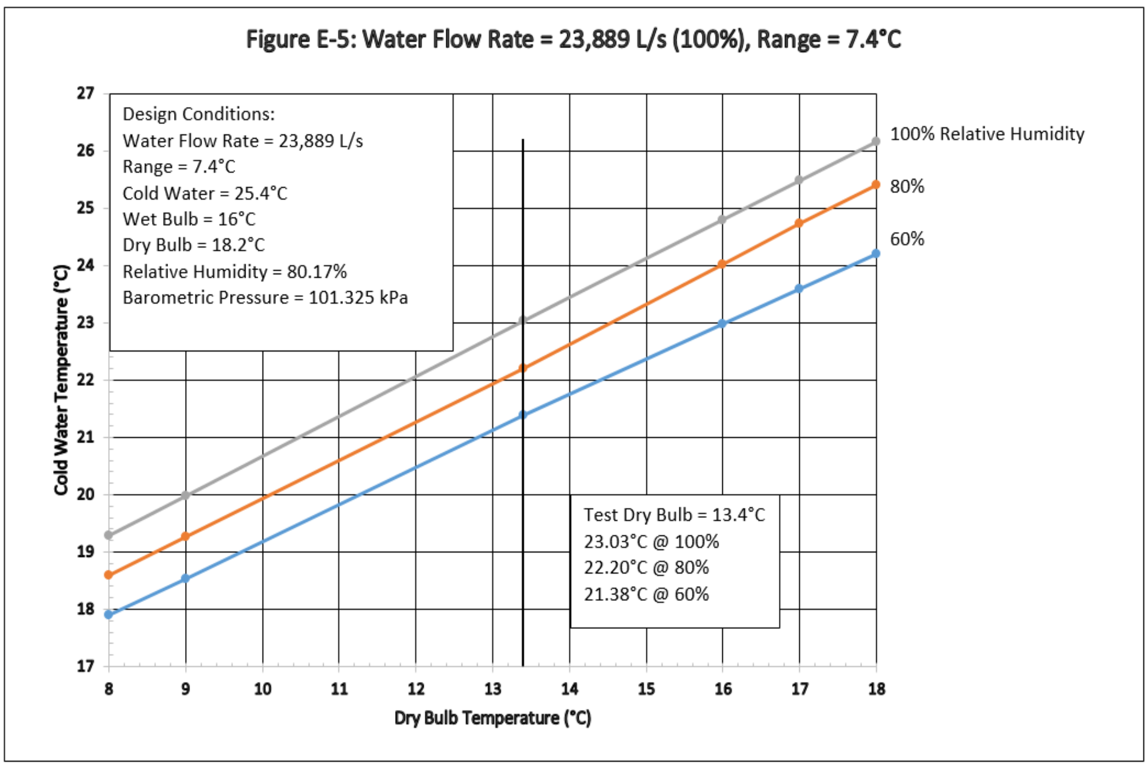


Figure E-5: Water Flow Rate = 23,889 L/s (100%), Range = 7.4°C
Design Conditions:
Water Flow Rate = 23,889 L/s
Range = 7.4°C
Cold Water = 25.4°C
Wet Bulb = 16°C
Dry Bulb = 18.2°C
Relative Humidity = 80.17%
Barometric Pressure = 101.325 kPa
Test Dry Bulb = 13.4°C
23.03°C @ 100%
22.20°C @ 80%
21.38°C @ 60%
100% Relative Humidity
80%
60%
Cold Water Temperature (°C)
Dry Bulb Temperature (°C)

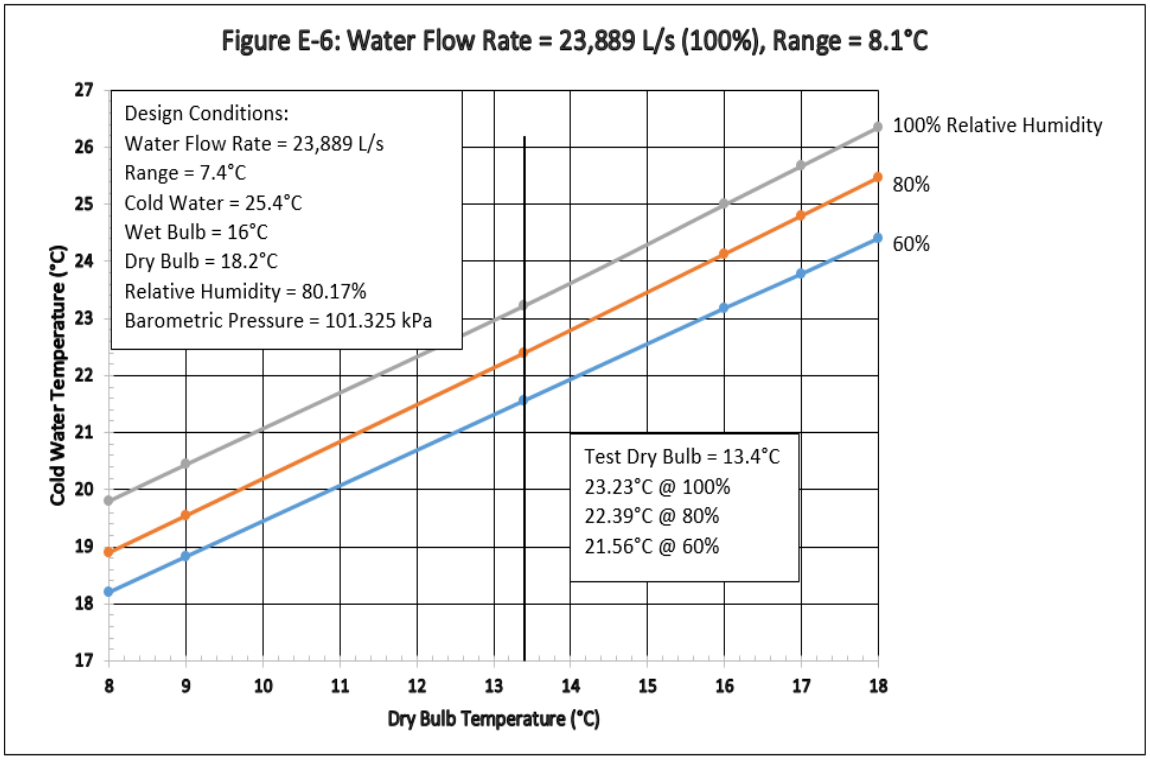


Figure E-6: Water Flow Rate = 23,889 L/s (100%), Range = 8.1°C
Design Conditions:
Water Flow Rate = 23,889 L/s
Range = 7.4°C
Cold Water = 25.4°C
Wet Bulb = 16°C
Dry Bulb = 18.2°C
Relative Humidity = 80.17%
Barometric Pressure = 101.325 kPa
Test Dry Bulb = 13.4°C
23.23°C @ 100%
22.39°C @ 80%
21.56°C @ 60%
100% Relative Humidity
80%
60%
Cold Water Temperature (°C)
Dry Bulb Temperature (°C)

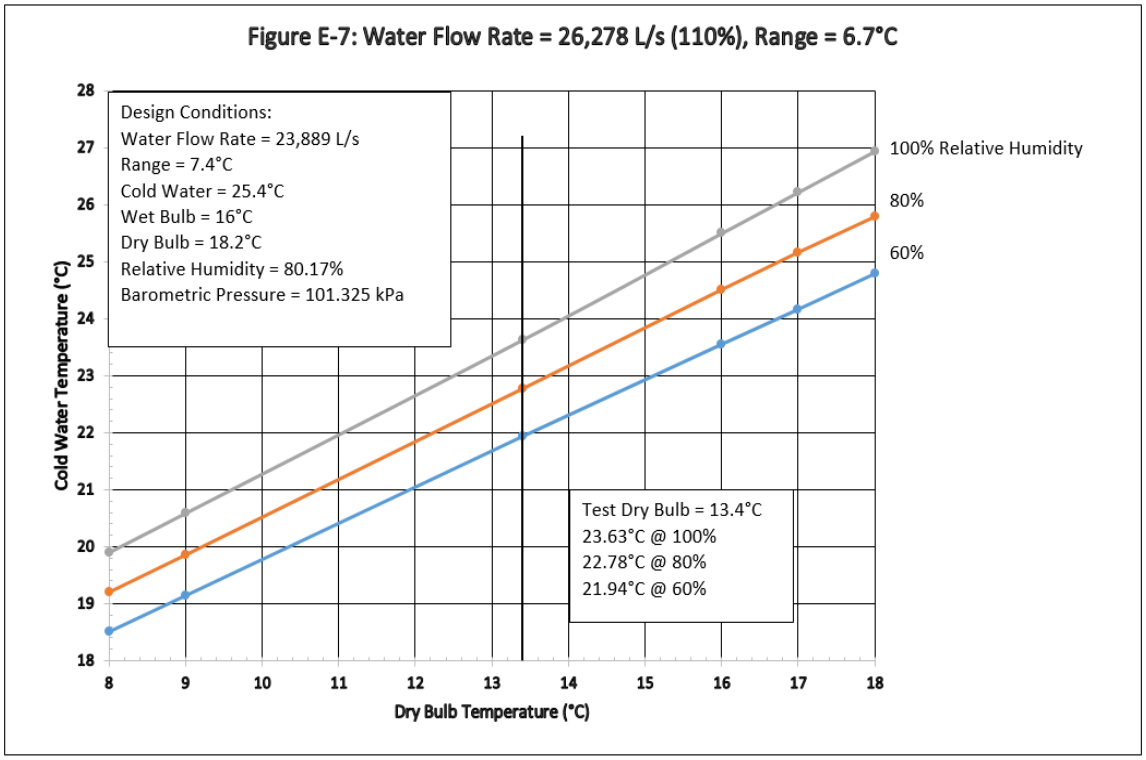


Figure E-7: Water Flow Rate = 26,278 L/s (110%), Range = 6.7°C
Design Conditions:
Water Flow Rate = 23,889 L/s
Range = 7.4°C
Cold Water = 25.4°C
Wet Bulb = 16°C
Dry Bulb = 18.2°C
Relative Humidity = 80.17%
Barometric Pressure = 101.325 kPa
Test Dry Bulb = 13.4°C
23.63°C @ 100%
22.78°C @ 80%
21.94°C @ 60%
100% Relative Humidity
80%
60%
Cold Water Temperature (°C)
Dry Bulb Temperature (°C)

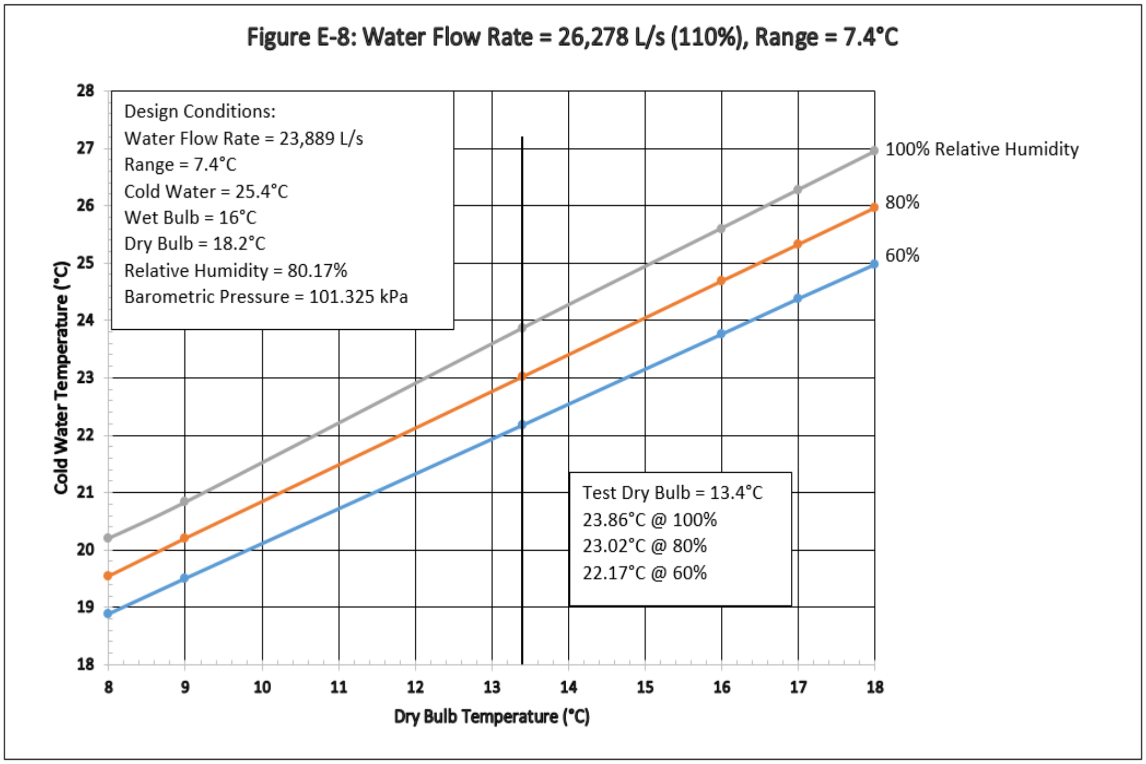


Figure E-8: Water Flow Rate = 26,278 L/s (110%), Range = 7.4°C
Design Conditions:
Water Flow Rate = 23,889 L/s
Range = 7.4°C
Cold Water = 25.4°C
Wet Bulb = 16°C
Dry Bulb = 18.2°C
Relative Humidity = 80.17%
Barometric Pressure = 101.325 kPa
100% Relative Humidity
80%
60%
Test Dry Bulb = 13.4°C
23.86°C @ 100%
23.02°C @ 80%
22.17°C @ 60%
Cold Water Temperature (°C)
Dry Bulb Temperature (°C)
28
27
26
25
24
23
22
21
20
19
18
8
9
10
11
12
13
14
15
16
17
18

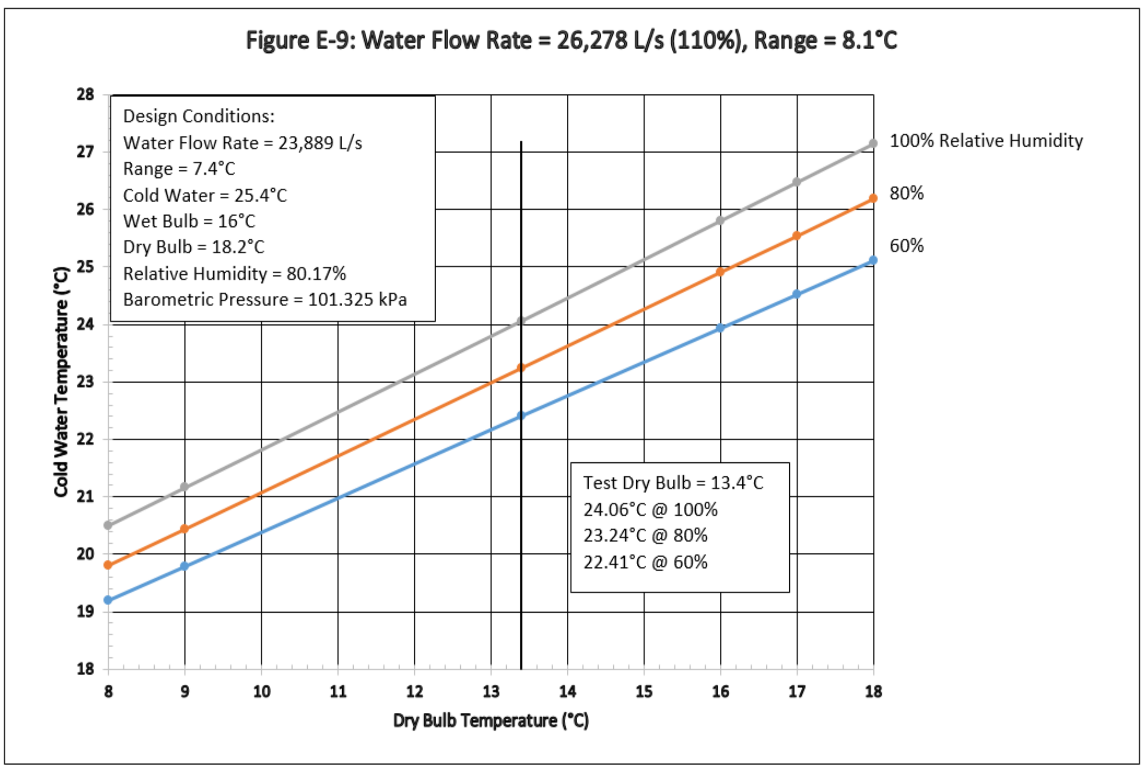


Figure E-9: Water Flow Rate = 26,278 L/s (110%), Range = 8.1°C
Design Conditions:
Water Flow Rate = 23,889 L/s
Range = 7.4°C
Cold Water = 25.4°C
Wet Bulb = 16°C
Dry Bulb = 18.2°C
Relative Humidity = 80.17%
Barometric Pressure = 101.325 kPa
100% Relative Humidity
80%
60%
Test Dry Bulb = 13.4°C
24.06°C @ 100%
23.24°C @ 80%
22.41°C @ 60%
Cold Water Temperature (°C)
Dry Bulb Temperature (°C)
28
27
26
25
24
23
22
21
20
19
18
8
9
10
11
12
13
14
15
16
17
18

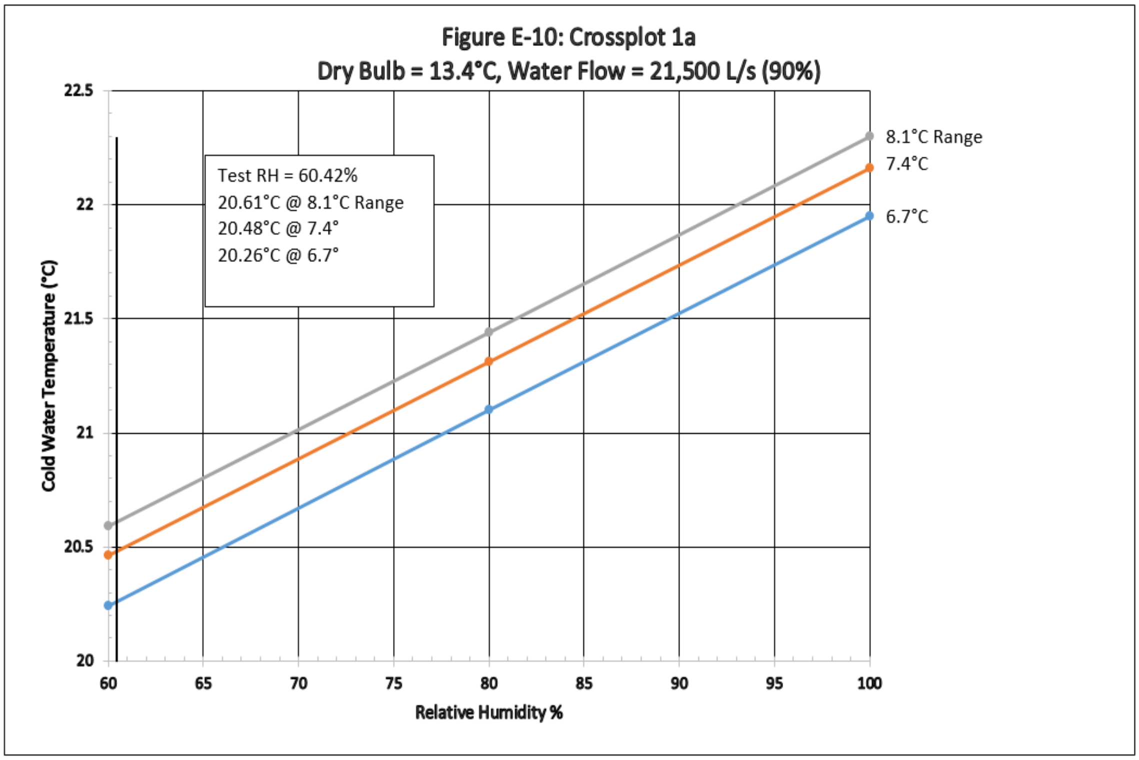


Figure E-10: Crossplot 1a
Dry Bulb = 13.4°C, Water Flow = 21,500 L/s (90%)
Test RH = 60.42%
20.61°C @ 8.1°C Range
20.48°C @ 7.4°
20.26°C @ 6.7°
8.1°C Range
7.4°C
6.7°C
Cold Water Temperature (°C)
Relative Humidity %
22.5
22
21.5
21
20.5
20
60
65
70
75
80
85
90
95
100

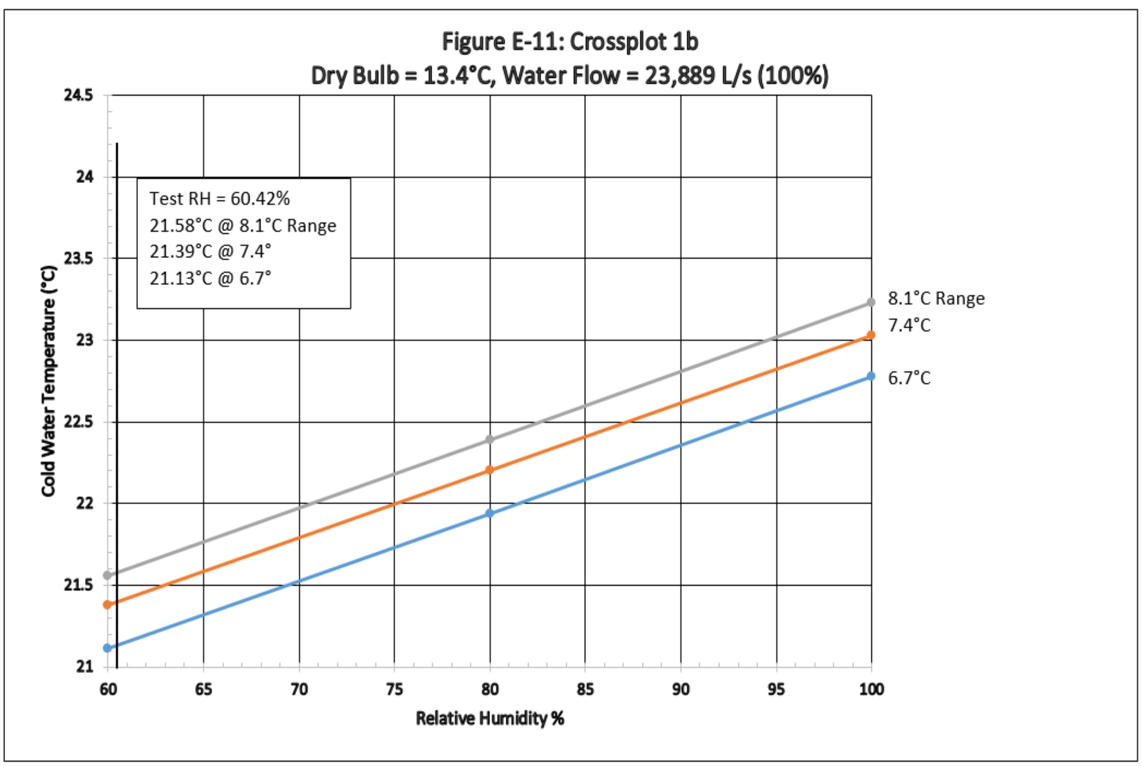


Figure E-11: Crossplot 1b
Dry Bulb = 13.4°C, Water Flow = 23,889 L/s (100%)
Test RH = 60.42%
21.58°C @ 8.1°C Range
21.39°C @ 7.4°
21.13°C @ 6.7°
8.1°C Range
7.4°C
6.7°C
Cold Water Temperature (°C)
Relative Humidity %
24.5
24
23.5
23
22.5
22
21.5
21
60
65
70
75
80
85
90
95
100

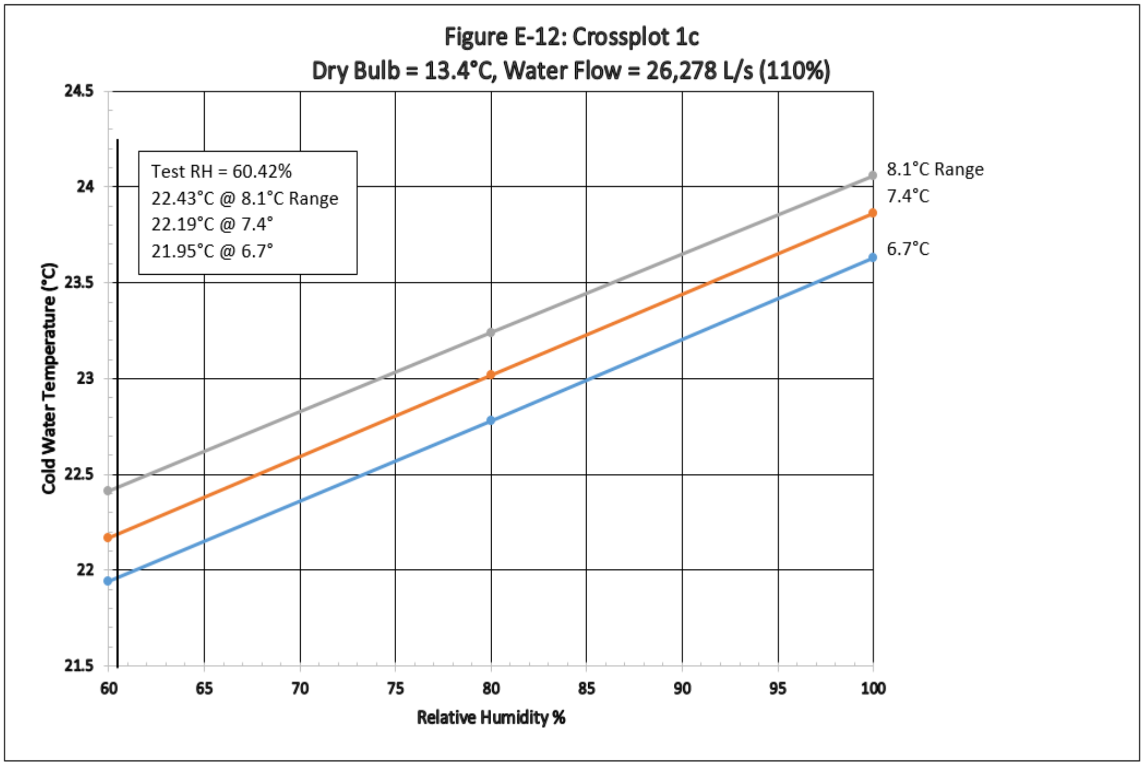


Figure E-12: Crossplot 1c
Dry Bulb = 13.4°C, Water Flow = 26,278 L/s (110%)
Test RH = 60.42%
22.43°C @ 8.1°C Range
22.19°C @ 7.4°
21.95°C @ 6.7°
8.1°C Range
7.4°C
6.7°C
Cold Water Temperature (°C)
Relative Humidity %
24.5
24
23.5
23
22.5
22
21.5
60
65
70
75
80
85
90
95
100

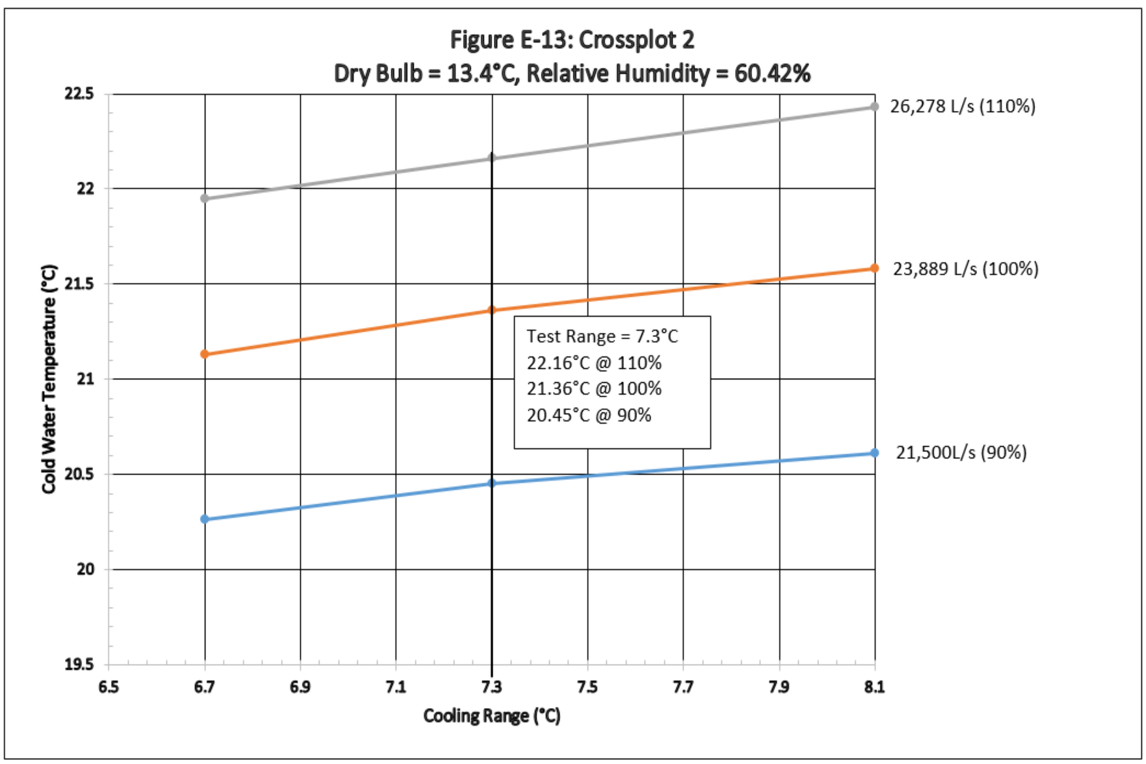


Figure E-13: Crossplot 2
Dry Bulb = 13.4°C, Relative Humidity = 60.42%
26,278 L/s (110%)
23,889 L/s (100%)
21,500L/s (90%)
Test Range = 7.3°C
22.16°C @ 110%
21.36°C @ 100%
20.45°C @ 90%
Cold Water Temperature (°C)
Cooling Range (°C)
22.5
22
21.5
21
20.5
20
19.5
6.5
6.7
6.9
7.1
7.3
7.5
7.7
7.9
8.1

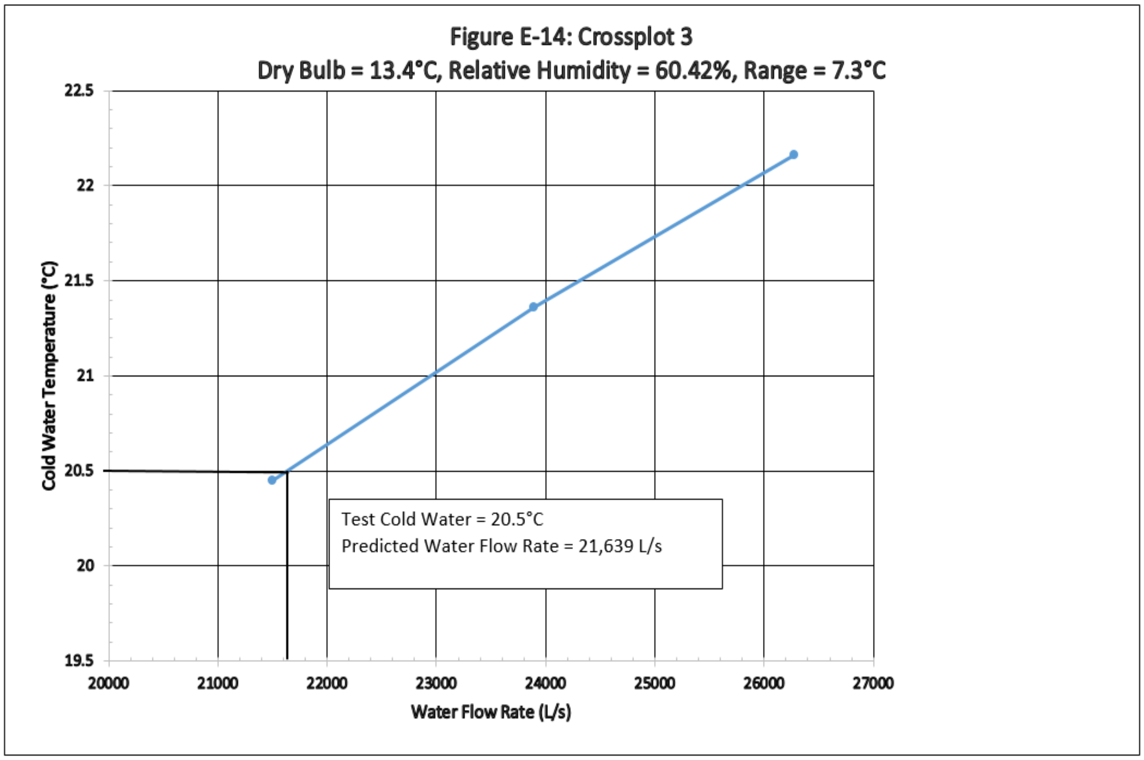


Figure E-14: Crossplot 3
Dry Bulb = 13.4°C, Relative Humidity = 60.42%, Range = 7.3°C
Cold Water Temperature (°C)
22.5
22
21.5
21
20.5
20
19.5
20000
21000
22000
23000
24000
25000
26000
27000
Water Flow Rate (L/s)
Test Cold Water = 20.5°C
Predicted Water Flow Rate = 21,639 L/s

APPENDIX F

IP Example Evaluation of Natural-Draft Cooling Tower Test Using the Performance Curve Method

Section F1. General

The purpose of this appendix is to describe and illustrate the performance curve methodology for evaluating a thermal performance test on a natural draft cooling tower, as described in Section 8 of the standard.

Section F2. Design and Test Conditions

The Design and measured Test Conditions for the natural draft cooling tower are summarized in Table F-1 below:

Table F-1 Natural Draft Cooling Tower Design and Test Data

	Design	Test
Water Flow Rate (W)	378,000 gpm	353,430 gpm
Hot Water Temp. (T_{hw})	91.0 °F	82.04 °F
Cold Water Temp. (T_{cw})	77.7 °F	68.90 °F
Cooling Range (R)	13.3 °F	13.14 °F
Wet Bulb Temp. (T_{wb})	60.8 °F	49.11 °F
Dry Bulb Temp. (T_{db})	64.8 °F	56.12 °F
Barometric Pressure (P_{bp})	29.921 in.Hg	30.570 in.Hg
Relative humidity (RH)	79.955%	42.5%

In accordance with Paragraph 8.2 of this standard, the manufacturer has submitted Performance Curves presenting Cold Water Temperature as a function of the air Dry Bulb Temperature with the Relative Humidity of the air as a parameter (see Figures F-1 through F-9).

F-1 to F-3 correspond to 90% of design water circulating rate;

F-4 to F-6 correspond to 100% of design water circulating rate;

F-7 to F-9 correspond to 110% of design water circulating rate.

Section F3. Evaluation Procedure

The five steps to be followed in evaluating the tower performance capability are as follows:

Step 1 - Determine the Predicted Cold Water Temperatures

Using the nine Performance Curves, three for each of the three water circulation rates, enter the curves at the test Dry Bulb Temperature (56.12°F) and determine the Cold Water Temperature for 60%. 80%, and 100% Relative Humidity at each Flow Rate and Range.

Table F-2 Predicted Cold Water Temperatures at 56.12°F Entering Dry Bulb Temperature.

Flow	Range	60% RH	80% RH	100% RH
90%	12.0°F	68.44	69.97	71.51
	13.3°F	68.83	70.36	71.90
	14.6°F	69.07	70.60	72.13
100%	12.0°F	70.00	71.49	73.00
	13.3°F	70.48	71.97	73.45
	14.6°F	70.80	72.30	73.81
110%	12.0°F	71.48	73.01	74.53
	13.3°F	71.91	73.43	74.95
	14.6°F	72.34	73.82	75.32

Step 2 - First Crossplot

Then for each Flow Rate, prepare a crossplot of Cold Water Temperature as a function of the Relative Humidity, with the cooling Range as a parameter. (See figures F-10 through F-12).

Step 3 - Second Crossplot

Using these new curves, enter each at the Test Relative Humidity (60.42%) and determine the Cold Water Temperature for each Flow Rate and Range.

Table F-3 Predicted Cold Water Temperatures at 56.12°F Entering Dry Bulb Temperature and 60.42%RH

Range	90% Flow	100% Flow	110% Flow
12.0°F	68.47	70.03	71.52
13.3°F	68.86	70.51	71.95
14.6°F	69.10	70.84	72.37

Then, for each Flow Rate, develop a crossplot of the Cold Water Temperature as a function of the cooling Range, (see Figure F-13).

Step 4 - Third Crossplot

Enter Figure F-13 at the Test Range (13.14°F) and determine the Cold Water Temperature for each of the three Flow Rates.

Table F-4 Predicted Cold Water Temperatures at 56.12°F Entering Dry Bulb Temperature, 60.42% RH, and 13.14°F Range.

90% Flow	100% Flow	110% Flow
68.81	70.45	71.89

Then crossplot the water Flow Rate as a function of the Cold Water Temperature (see Figure F-14).

Step 5 - Determine Predicted Flow Rate

Enter Fig F-14 at the Measured Cold Water Temperature (68.90°F) and from the intersection with the curve, determine the Predicted Water Flow Rate at the Test Cold Water Temperature as 342,224 gpm.

Step 6 - Determine Cooling Tower Capability

Using Equation 8.1, find the cooling tower Thermal Performance Capability as:

$$\%Capability = \left(\frac{353\ 430\ gpm}{342\ 224\ gpm}\right).(100) = 103.3\%$$

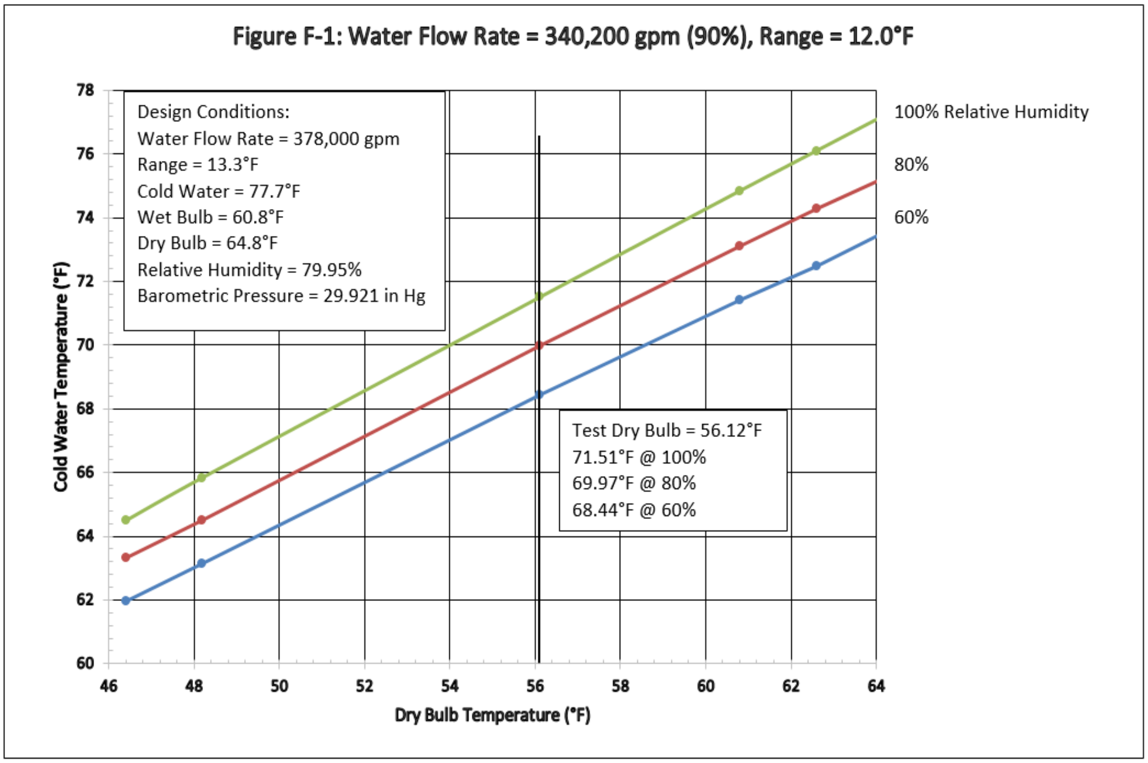


Figure F-2: Water Flow Rate = 340,200 gpm (90%), Range = 13.3°F

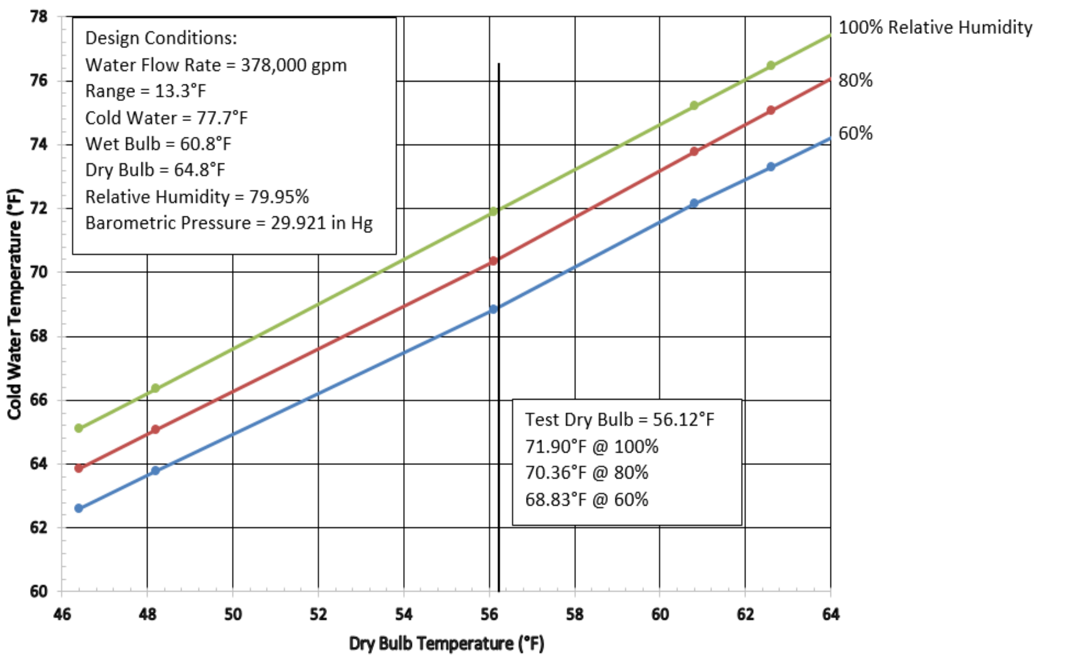


Figure F-3: Water Flow Rate = 340,200 gpm (90%), Range = 14.6°F

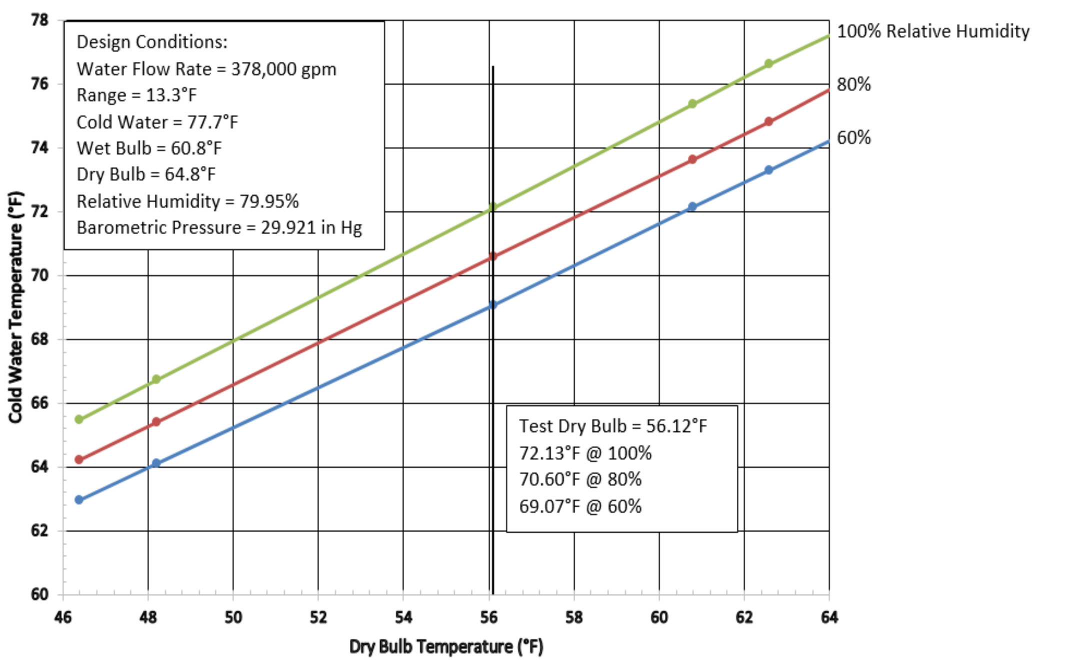


Figure F-4: Water Flow Rate = 378,000 gpm (100%), Range = 12°F

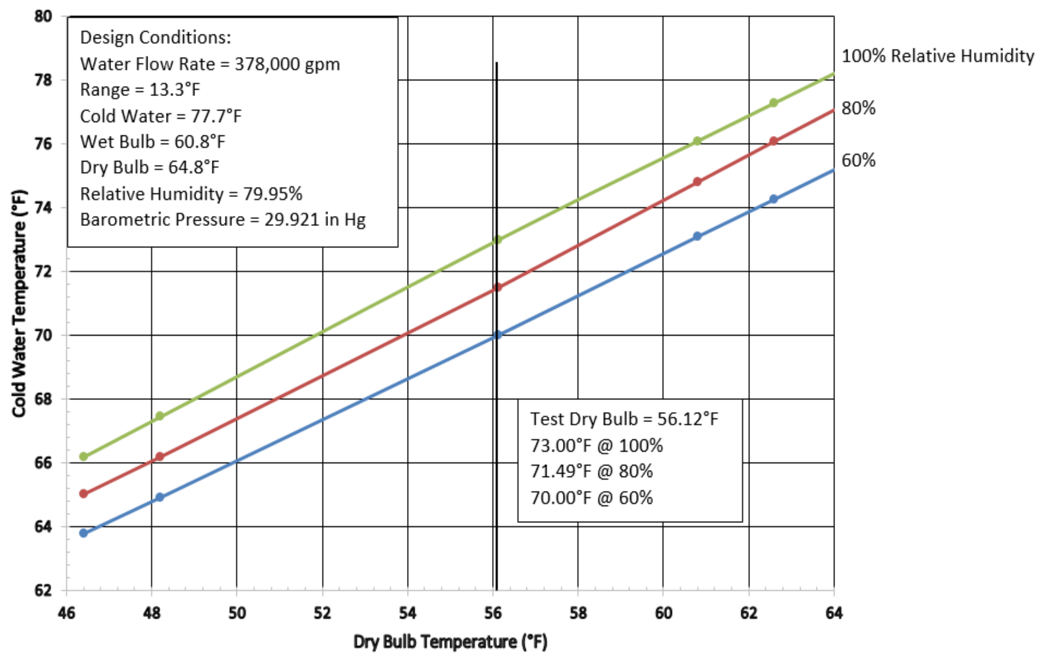
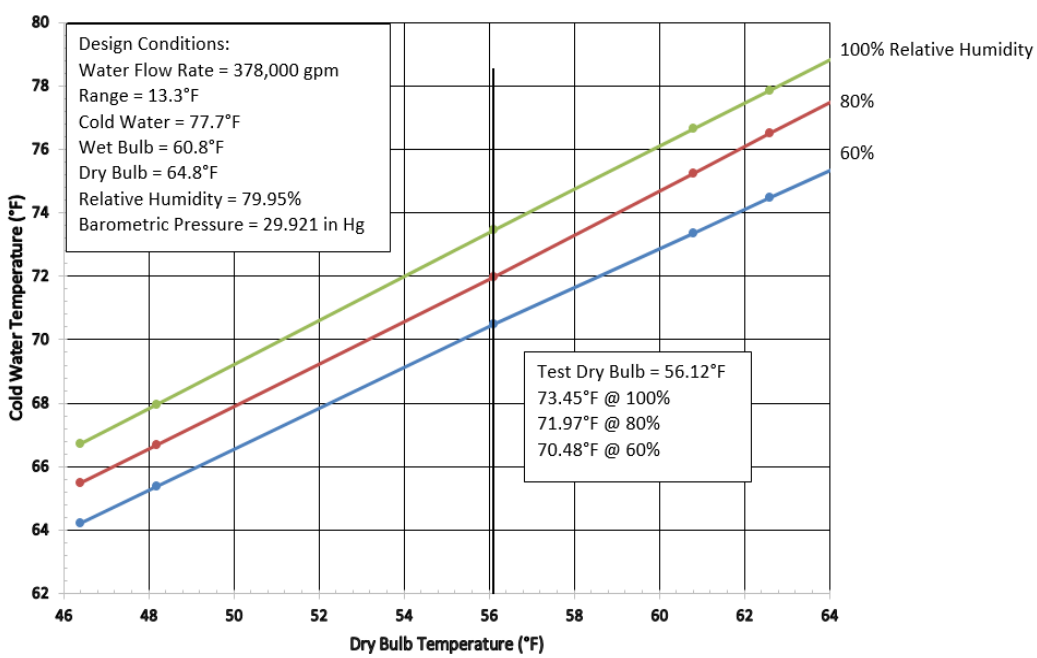


Figure F-5: Water Flow Rate = 378,000 gpm (100%), Range = 13.3°F



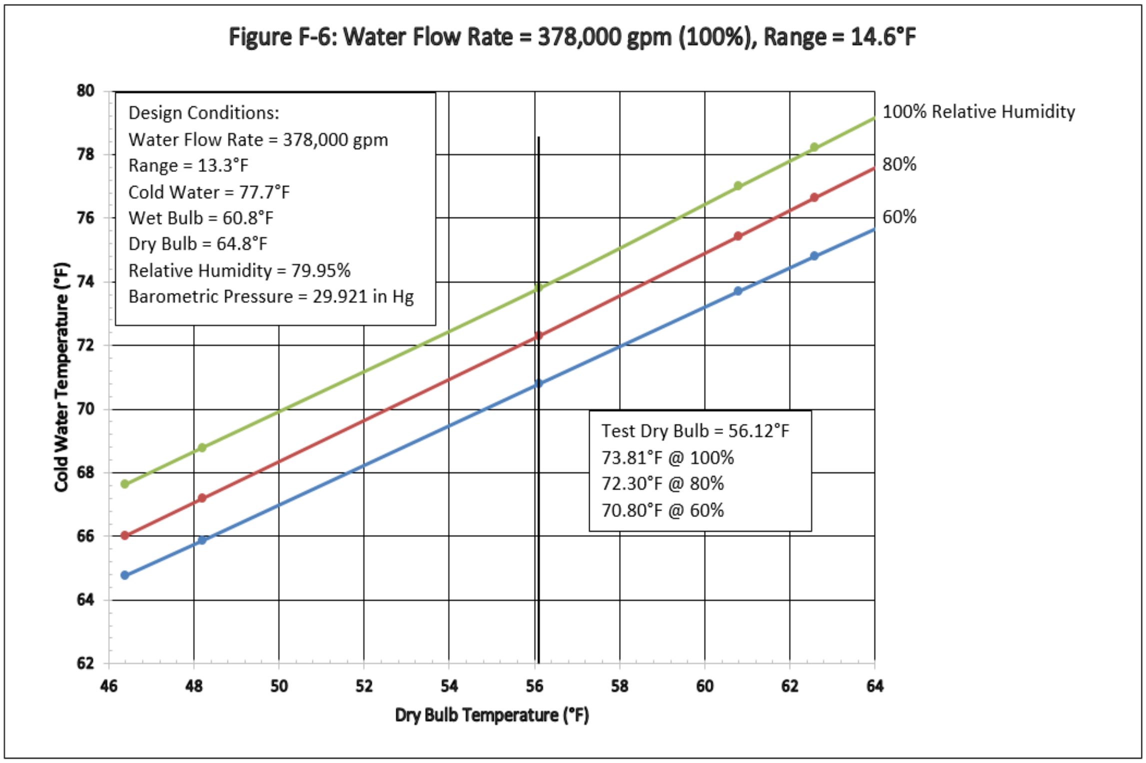


Figure F-6: Water Flow Rate = 378,000 gpm (100%), Range = 14.6°F
Design Conditions:
Water Flow Rate = 378,000 gpm
Range = 13.3°F
Cold Water = 77.7°F
Wet Bulb = 60.8°F
Dry Bulb = 64.8°F
Relative Humidity = 79.95%
Barometric Pressure = 29.921 in Hg
Test Dry Bulb = 56.12°F
73.81°F @ 100%
72.30°F @ 80%
70.80°F @ 60%
100% Relative Humidity
80%
60%
Cold Water Temperature (°F)
Dry Bulb Temperature (°F)
80
78
76
74
72
70
68
66
64
62
46
48
50
52
54
56
58
60
62
64

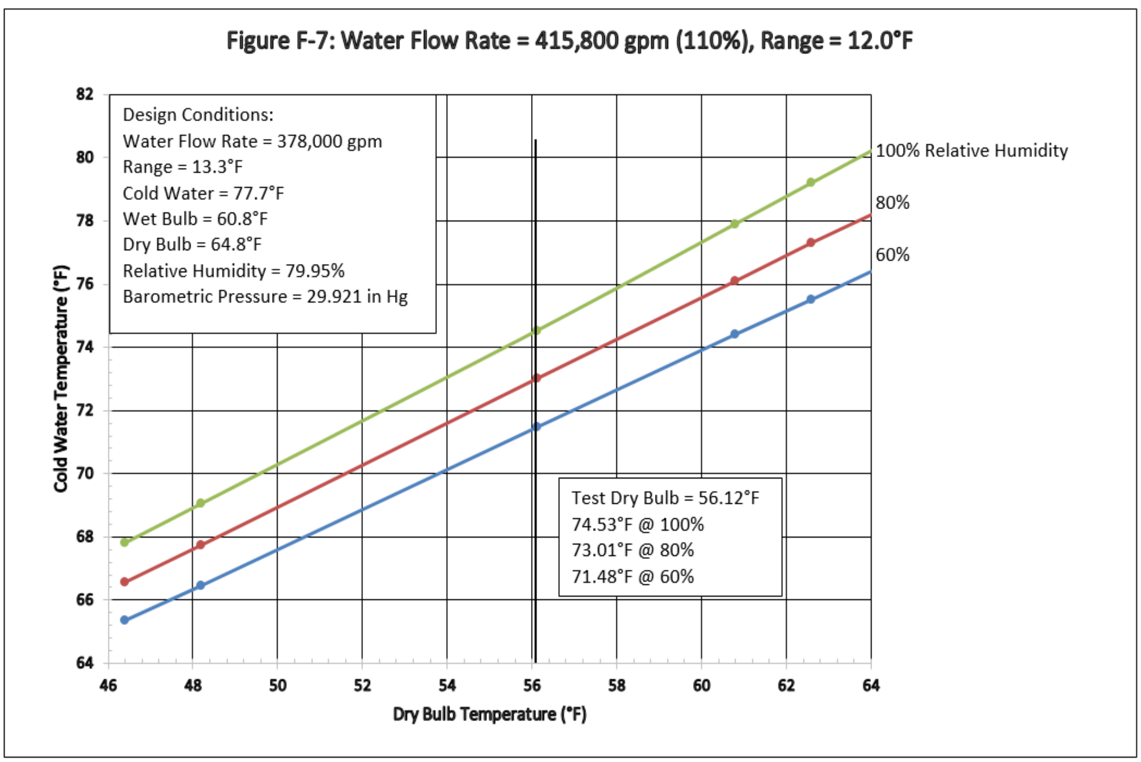


Figure F-7: Water Flow Rate = 415,800 gpm (110%), Range = 12.0°F
Design Conditions:
Water Flow Rate = 378,000 gpm
Range = 13.3°F
Cold Water = 77.7°F
Wet Bulb = 60.8°F
Dry Bulb = 64.8°F
Relative Humidity = 79.95%
Barometric Pressure = 29.921 in Hg
Test Dry Bulb = 56.12°F
74.53°F @ 100%
73.01°F @ 80%
71.48°F @ 60%
100% Relative Humidity
80%
60%
Cold Water Temperature (°F)
Dry Bulb Temperature (°F)
82
80
78
76
74
72
70
68
66
64
46
48
50
52
54
56
58
60
62
64

Figure F-8: Water Flow Rate = 415,800 gpm (110%), Range = 13.3°F

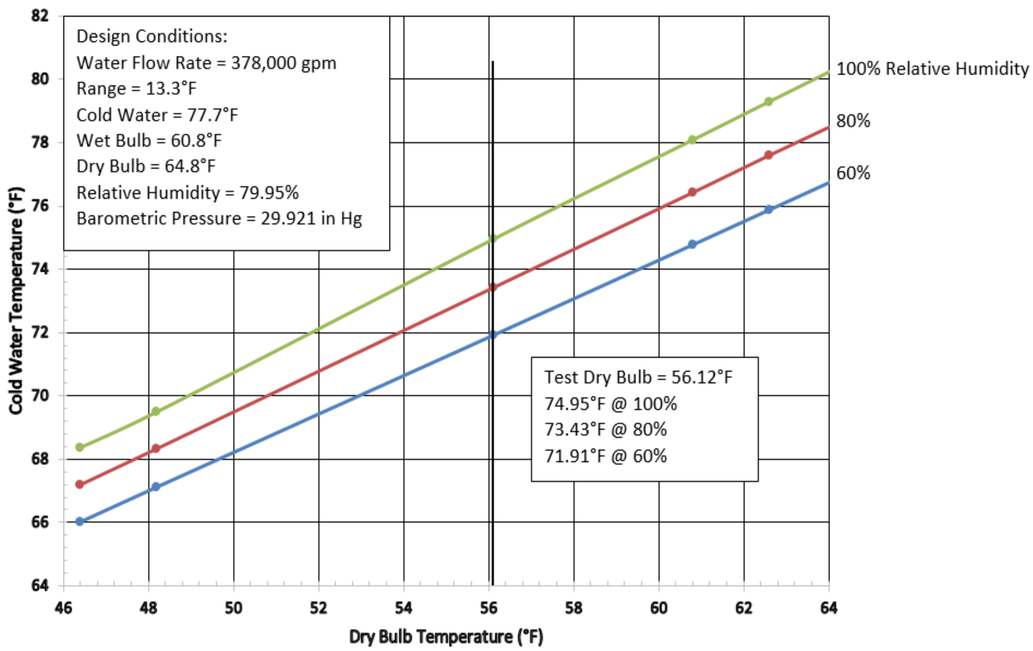
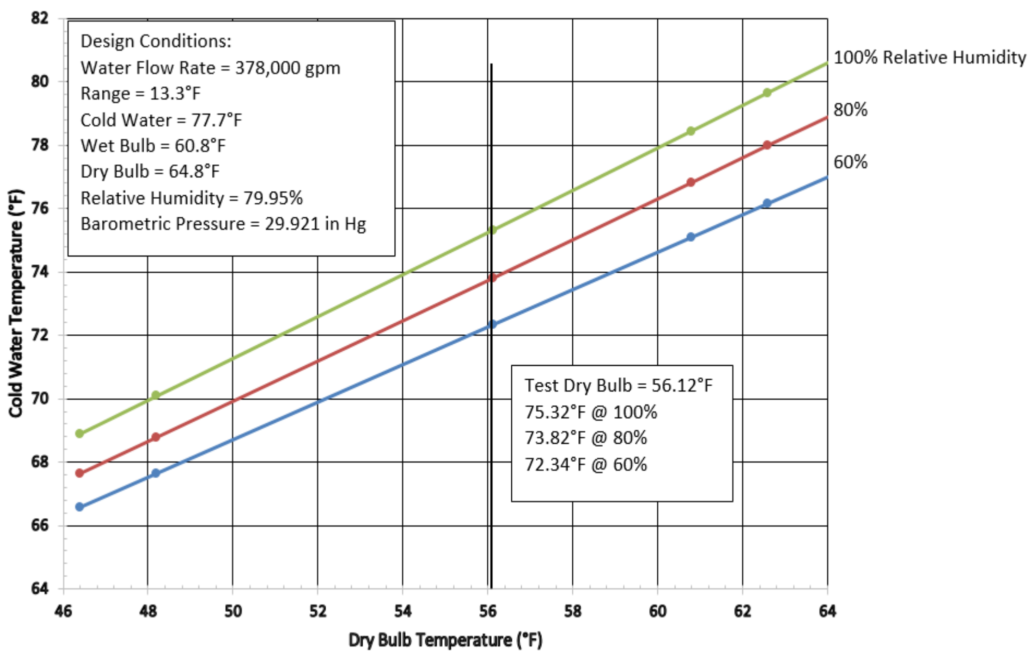


Figure F-9: Water Flow Rate = 415,800 gpm (110%), Range = 14.6°F



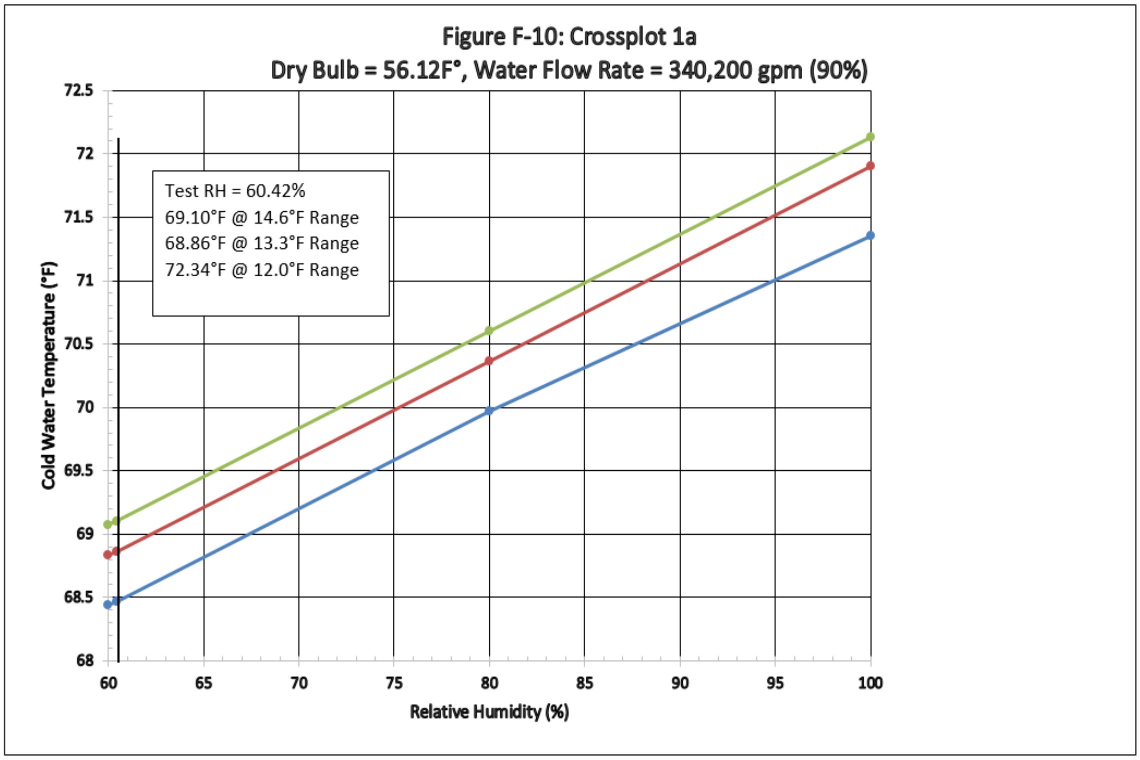


Figure F-10: Crossplot 1a
Dry Bulb = 56.12F°, Water Flow Rate = 340,200 gpm (90%)
Test RH = 60.42%
69.10°F @ 14.6°F Range
68.86°F @ 13.3°F Range
72.34°F @ 12.0°F Range
Cold Water Temperature (°F)
Relative Humidity (%)
72.5
72
71.5
71
70.5
70
69.5
69
68.5
68
60
65
70
75
80
85
90
95
100

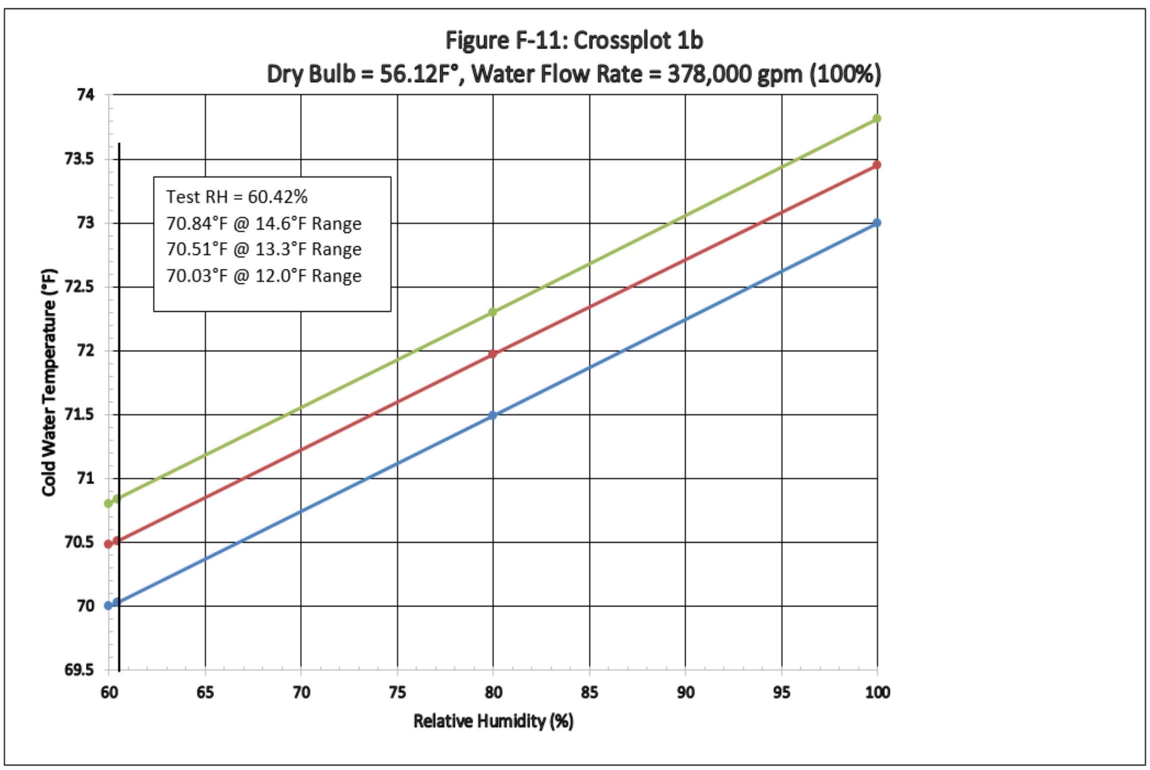


Figure F-11: Crossplot 1b
Dry Bulb = 56.12F°, Water Flow Rate = 378,000 gpm (100%)
Test RH = 60.42%
70.84°F @ 14.6°F Range
70.51°F @ 13.3°F Range
70.03°F @ 12.0°F Range
Cold Water Temperature (°F)
Relative Humidity (%)
74
73.5
73
72.5
72
71.5
71
70.5
70
69.5
60
65
70
75
80
85
90
95
100

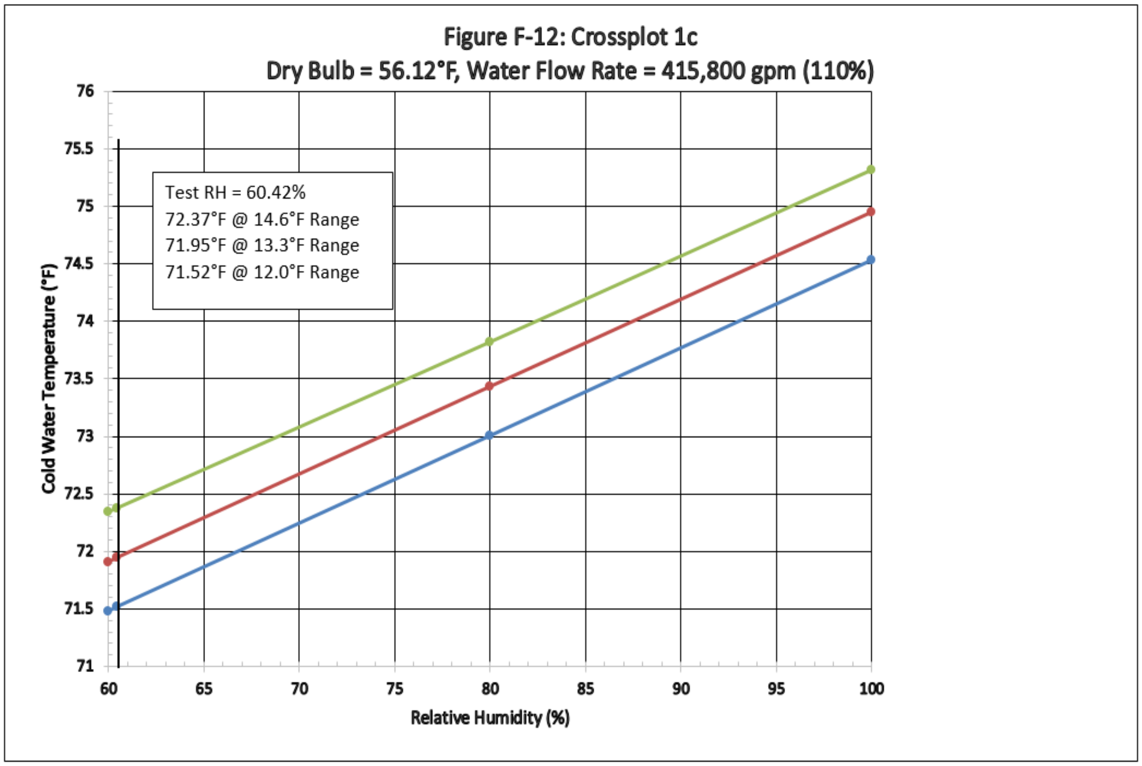


Figure F-12: Crossplot 1c
Dry Bulb = 56.12°F, Water Flow Rate = 415,800 gpm (110%)

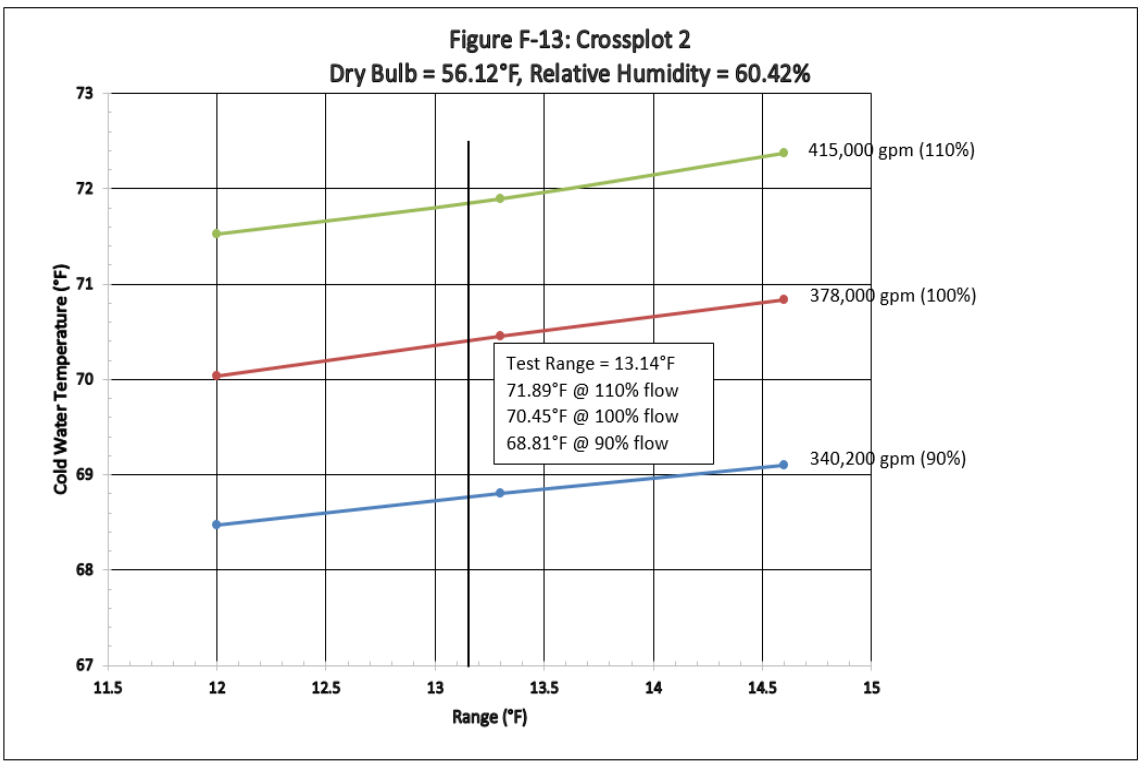


Figure F-13: Crossplot 2
Dry Bulb = 56.12°F, Relative Humidity = 60.42%

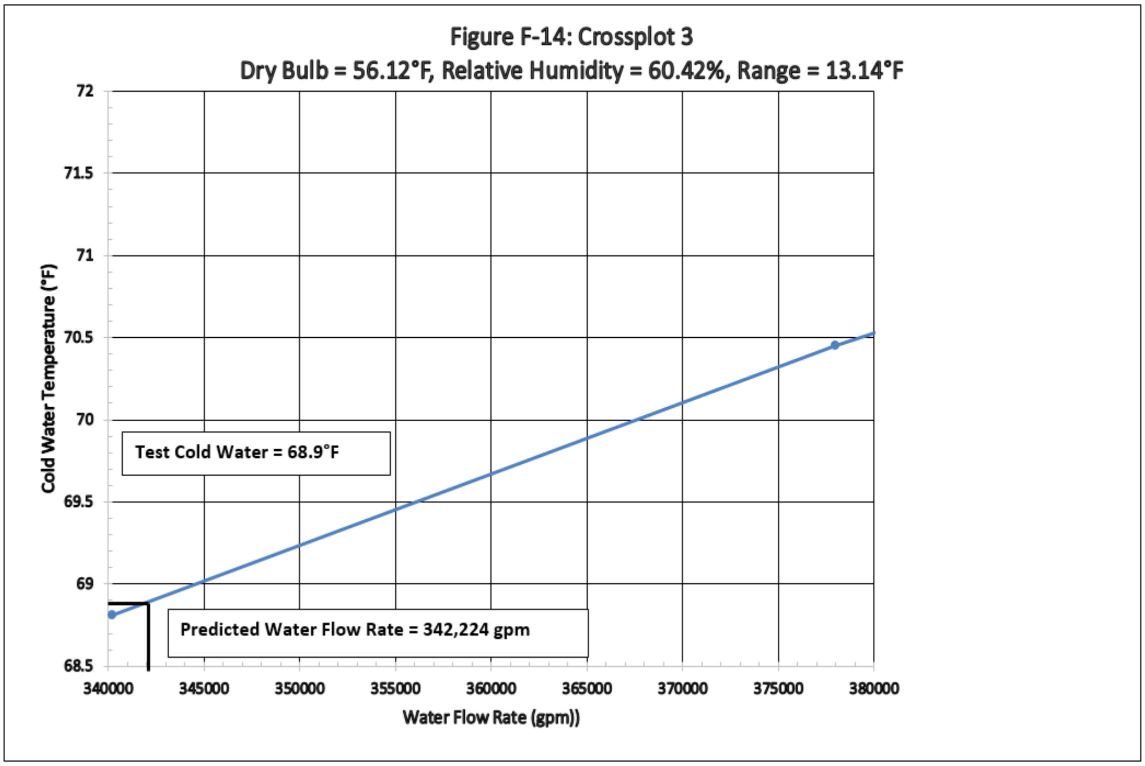


Figure F-14: Crossplot 3
Dry Bulb = 56.12°F, Relative Humidity = 60.42%, Range = 13.14°F
Cold Water Temperature (°F)
72
71.5
71
70.5
70
69.5
69
68.5
Test Cold Water = 68.9°F
Predicted Water Flow Rate = 342,224 gpm
340000
345000
350000
355000
360000
365000
370000
375000
380000
Water Flow Rate (gpm))

APPENDIX G
Number and Location of Wet-Bulb Temperature Measurement Stations

Section G1. General

The purpose of this appendix is to prescribe the **minimum** number and location of wet bulb temperature measurement stations that will produce an acceptable, average inlet wet-bulb temperature for the test period.

The equations and graphs presented below may produce results that require the test parties to exercise judgment and deploy more instruments than dictated by the formulas. For natural draft towers and round or octagonal mechanical draft towers, multiply the perimeter at half the air inlet height by the air inlet height to calculate the "inlet area of concern". Natural draft towers require no more than three vertical stations.

Section G2. Recommended Number of Stations.

The minimum number of wet bulb temperature measurement stations are shown on Figures G-1 (SI) and G-2 (IP) which plot the number of stations as a function of the size of the air inlet area of concern.

Alternatively, the minimum number of stations may be calculated using the equation below and rounding all answers to the next higher whole number (e.g., if n = 3.1, use 4):

$$n = (K)\ (A)^{0.4}$$

Where:

n = Minimum number of stations

K = A constant to adjust for units of measure equal to 0.52 for SI and 0.20 for IP

A = the inlet area of concern, m^2 (ft^2)

*Note: The "inlet area of concern" applies to each individual area, rather than the total area. Thus, when determining the number of stations for an induced-draft cooling tower having louvered inlet faces 4 m high by 35 m long on both longitudinal sides, the area "A" of each "area of concern" would be 4*35 = 140 m^2 and the recommended minimum number of stations per side would be 4.*

Both the graph and the equation assume both the air velocity and the temperatures are distributed uniformly (or nearly so) over the "inlet area of concern." A pre-test survey shall be conducted by the CTI Test Agency to confirm this assumption or detect any unusual conditions (see Paragraph G3).

Section G3. Location of Stations.

The area of concern shall be divided into equal area rectangles with measurement stations located at the geometric center of each rectangle, unless the pre-test survey reveals an unusual condition suggesting some other arrangement. In such instances, the Purchaser and the Manufacturer in consultation with the CTI Test Agency shall agree on the alternative locations.

The concentration of any recirculating air is normally greater at the top of the air intakes than at the bottom; to ensure detection and measurement of such gradients, multiple vertical stations should be employed. The number of vertical stations shall be specified by

$$n_v = \frac{z}{k_v}$$

Where

n_v = Number of vertical stations

z = Air inlet height, m (ft)

k_v= Constant, 4.25 m for SI units, 14 ft for IP units

The number of vertical measurements shall be rounded to the next higher integer value. The position of each vertical station is calculated by

$$z_i = z\frac{2(i-1)+1}{2n_v}$$

Where

z_i = Vertical height above bottom of air inlet for vertical position i.

The number of horizontal stations for each air inlet shall be calculated by

$$n_h = \frac{n}{n_v}$$

The number of horizontal stations shall be rounded to the next higher integer value.

$$L_j = L\frac{2*(j-1)+1}{2n_h}$$

Where

L = Length of the air inlet

L_j = Horizontal position of measurement j

For the case of a rectangular cooling tower without end walls, a single horizontal station shall be used at each open end.

Section G4. Air Velocity Effects.

If either air velocity or wet-bulb temperature is uniform throughout the inlet area of consideration, it will not be necessary to weight the wet-bulb temperatures.

In extreme cases, however, both air velocity and wet-bulb temperature may exhibit unacceptable variations for an extended period of time. If contractual agreement or other reasons prevent postponement of the test, sufficient measurements of the air velocity at each measurement station should be made and the measured test values of wet bulb temperature weighted in proportion to the velocity distribution to obtain an average inlet wet-bulb temperature.

Section G5. Example Situations.

G5.1 Examples in SI Units

G5.1.1 *Example #1* The tower to be tested is a counter-flow type, with louvered air intakes measuring 1.8 m high by 60 m long along each side. The area of concern for each intake is $1.8\ m * \ 60\ m\ =\ 108\ m^2$. From Figure G-1 determine the recommended minimum number of stations per intake is four.

Alternatively, using the equation yields the same number:

$$n = 0.52\ (108)^{0.4} = 3.4\ , \therefore \text{use } 4$$

The number of vertical stations on each inlet would be

$$n_v = \frac{z}{k_v} = \frac{1.8}{4.25} = 0.42 \therefore round\ to\ 1$$

The number of horizontal stations on each inlet would be

$$n_h = \frac{n}{n_v} = \frac{4}{1} = 4$$

Therefore, each intake should be divided into four imaginary rectangles, each measuring 1.8 m high by 15 m long with one wet-bulb measurement station located at the center-point of each rectangle.

G5.1.2 *Example #2* The tower to be tested is a double flow crossflow type, with louvered intakes measuring 11.6m high by 30m long along the two side opposing sides. The area of each intake is $11.6\ m * 30\ m = 348\ m^2$. From the graph or equation, the recommended minimum number of stations per intake sides is six.

The number of vertical stations on each air inlet will be

$$n_v = \frac{11.6}{4.25} = 2.7\ \therefore round\ to\ 3$$

The number of horizontal stations on each air inlet will be

$$n_h = \frac{6}{3} = 2$$

There would be a total of 6 psychrometers at each air inlet, each at the center of a rectangle 3.9 meters by 15 meters.

G5.2 Examples in IP Units

G5.2.1 *Example #1* The tower to be tested is an induced draft, counterflow tower having louvered air inlets on both longitudinal sides measuring 12 feet high by 120 feet long. The area of concern would be $12ft * 120ft = 1{,}440ft^2$ per individual area. Referring to Figure G-2, the recommended minimum number of stations per side would be 4 (8 total for the tower). Alternatively, using the equation:

$$n = 0.20\ (1{,}440)^{0.4} = 3.7,\ \therefore \text{use } 4$$

The number of vertical stations would be

$$n_v = \frac{z}{14} = \frac{12}{14} = 0.86\ \therefore round\ up\ to\ 1$$

The number of horizontal stations would be

$$n_h = \frac{n}{n_v} = \frac{4}{1} = 4$$

G5.2.2 *Example #2* Assume the tower to be tested is a counterflow type, with air intakes along each side, each measuring 18 feet high by 380 feet long by 48 feet wide. The cooling tower is 48 feet wide with open ends. The area of the air intake on each side is $6840ft^2$. From Figure G-2, the recommended minimum number of stations per intake is eight. The equation would yield approximately the same number:

$$n = 0.2\ (6840)^{0.4} = 6.8\ \therefore \text{round up to } 7$$

The number of vertical stations will be

$$n_v = \frac{18}{14} = 1.3\ \therefore roundup\ to\ 2$$

The number of horizontal stations along the tower side air inlets will be

$$n_h = \frac{n}{n_v} = \frac{7}{2} = 3.5\ \therefore round\ up\ to\ 4$$

There will be 4 horizontal stations each with psychrometers at two heights along each air inlet side and a single horizontal station with two vertically at each of the open ends, for a total of 20 psychrometers. The psychrometers will be placed at the midpoint of equal areas along the cooling tower sides. The height of the stations will be

$$z_i = z\frac{2(i-1)+1}{2n_v}$$

$$z_1 = 18\frac{2(1-1)+1}{2*2} = 4.5\ ft$$

$$z_2 = 18\frac{2(2-1)+1}{2*2} = 13.5\ ft$$

The horizontal position of the stations on the sides of the cooling tower will be

$$L_j = L\frac{2(j-1)+1}{2n_h}$$

$$L_1 = 380\frac{2(1-1)+1}{2*4} = 47.5ft$$

$$L_2 = 380\frac{2(2-1)+1}{2*4} = 142.5\ ft$$

$$L_3 = 380\frac{2(3-1)+1}{2*4} = 237.5ft$$

$$L_4 = 380\frac{2(4-1)+1}{2*4} = \ 332.5\ ft$$

Each of the psychrometers placed along each side of the cooling tower would be at the center of a rectangle 9 feet high by 95 feet long. Two psychrometers would be placed 4.5 feet and 13.5 feet high at the midpoint of each of the open ends.

The open ends computation is not illustrated, it requires rounding n and n_v down (i.e., judgement).

G5.2.3 *Example #3* Assume the tower to be tested is a double flow crossflow type, with louvered intakes along the two opposing sides, each measuring 36 feet high by 96 feet long. The area of each intake is 3,456 square feet. From the graph or equation, the recommended minimum number of stations per intake of six.

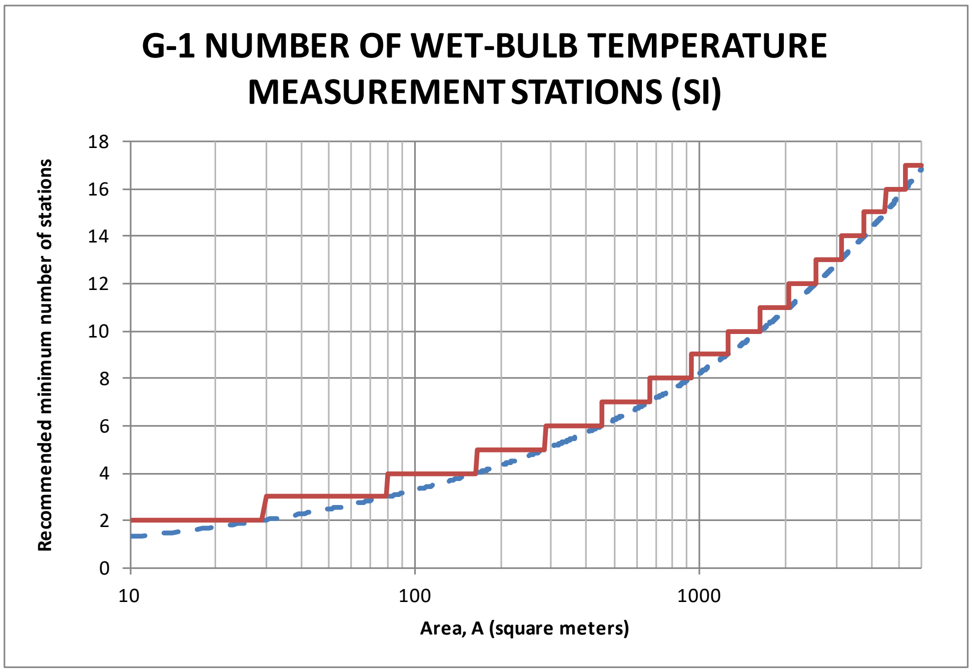
The number of vertical stations will be

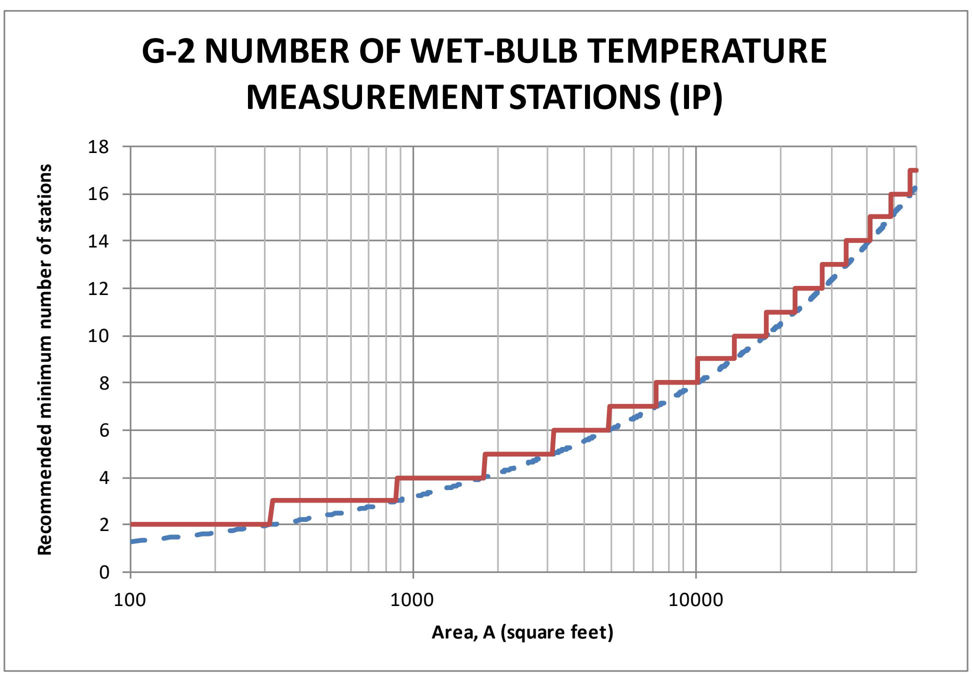
$$n_v = \frac{36}{14} = 2.6 \therefore roundup\ to\ 3$$

The number of horizontal stations will be

$$n_h = \frac{6}{3}$$

Each psychrometer will be placed at the center of a rectangle measuring 12ft high by 48ft long.





APPENDIX I
Correcting Cold Water Temperature for Heat Added by Pump

Section I1. General

When the cold water temperature is measured on the discharge side of the circulating pump by means of a bubbler and flow control valve, as illustrated in Figure I-1, the temperature reading is affected by the energy input from the pump to the water, which is reflected in two phenomena:

1) A temperature increase in the water due to the inefficiency of the pump.
2) A temperature increase as the water is throttled (adiabatically) from a higher pressure to the lower atmospheric pressure.

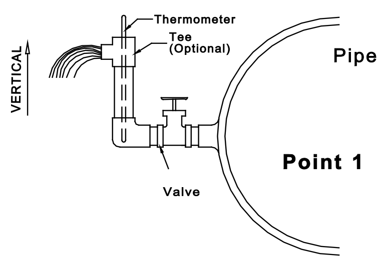


Figure I-1. Temperature Tap

Accordingly, the temperature measurement must be adjusted to more accurately reflect the true temperature of the water leaving the tower.

Section I2. Adjustments

To adjust the temperature measured at the bubbler, the total energy input from the pump to the water must be expressed as a temperature rise (ΔT) which can then be subtracted from the measured value. The value for ΔT is calculated as:

$$SI: \Delta T = \frac{P_1 * 0.0002390}{\eta_{pump}} \qquad (I-1)$$

$$IP: \Delta T = \frac{P_1 * 0.0002966}{\eta_{pump}} \qquad (I-1)$$

Where P_1 represents the gauge pressure (above atmospheric pressure) on the discharge side of the pump. This calculated value for ΔT is to be subtracted from the measured temperature to better reflect the true temperature of the water leaving the cooling tower.

Section I3. Derivation of Equations

I3.1 Pressure Considerations. As with all throttling processes, expansion across a flow control valve is an essentially adiabatic process where $\Delta h_w = 0$

But also:

$$\Delta h_w = \Delta U + \Delta\left(\frac{P.V}{J}\right) \qquad (I-2)$$

Therefore:

$$(U_2 - U_1) + \left(\frac{P_2 * V_2}{J}\right) - \left(\frac{P_1 * V_1}{J}\right) = 0 \quad (I-3)$$

Where:

Referring back to Figure I-1, Point 1 is inside the cold water main on the discharge side of the pump at some pressure P_1 above atmospheric.

Point 2 is downstream of the throttling valve at pressure P_2 = atmospheric = 0 gauge.

Thus, since $P_2 = 0$:

$$(U_2 - U_1) = \Delta U = \left(\frac{P_1 * V_1}{J}\right) \qquad (I-4)$$

And since $\Delta U = c_{pw} * \Delta T$, then:

$$\Delta T = \left(\frac{P_1 * V_1}{c_{pw} * J}\right) \qquad (I-5)$$

Next, substituting known values for both IP and SI:

$$SI: \Delta T = \frac{P_1 * \left(0.001 \frac{m^3}{kg}\right)}{\left(4.186 \frac{kJ}{kg.°C}\right)\left(1 \frac{N.m}{J}\right)} \qquad (I-6)$$

$$IP: \Delta T = \frac{P_1 * \left(144 \frac{in^2}{ft^2}\right) * \left(0.016 \frac{ft^3}{lbm}\right)}{\left(1.0 \frac{BTU}{lbm.°F}\right)\left(778 \frac{ft.lbf}{BTU}\right)} \qquad (I-6)$$

Which then reduce to:

$$SI: \Delta T = P_1 * 0.0002390 \qquad (I-7)$$

$$SI: IP = P_1 * 0.002966 \qquad (I-7)$$

Equation I-7 can be considered a representation of the useful work done by the pump in raising the pressure from atmospheric (in the cold water basin) to P1. Now the heat added due to the inefficiency of the pump must be considered.

I3.2 ***Pump Efficiency Considerations*** By definition, the efficiency of the pump is the useful work divided by the total energy input such that:

$$\eta_{pump} = \frac{W_{useful}}{W_{total}} \qquad (I-8)$$

Rearranging:

$$W_{total} = \frac{W_{useful}}{\eta_{pump}} \qquad (I-9)$$

Since the useful work of the pump, expressed as a ΔT, is described in Equation I-7, this can be substituted into Equation I-9 to express the total energy input to the water as a temperature rise in the water when the temperature is measured at the atmospheric side of the bubbler and valve:

$$SI: \Delta T = \frac{P_1 * 0.0002390}{\eta_{pump}} \qquad (I-1)$$

$$IP: \Delta T = \frac{P_1 * 0.0002966}{\eta_{pump}} \qquad (I-1)$$

SYMBOLS:

c_{pw} = Specific heat of water, 4.180 $kJ/kg.°C$
(1.0 $BTU/lbm.°F$)

h_w = Enthalpy of water, kJ/kg, (BTU/lbm)

U = Internal energy, kJ/kg (BTU/lbm)

P = Pressure, kPa (psi)

J = Mechanical equivalent of heat, 1 $N.m/J$
(778 $ft.lbf/BTU$)

V = Specific volume, 0.001 m^3/kg (0.016 ft^3/lbm)

T = Temperature, °C (°F)

η_{pump}= Pump efficiency, dimensionless

APPENDIX J
Adjusting Measured Data for Thermal Lag

Section J1. General

In most testing situations, the measurement point for the Cold Water Temperature can be located sufficiently close to the cooling tower that it is quite representative of the temperature of the water as it leaves the fill and falls into the cold water basin. However, where the basin is significantly large and/or the measurement station is at some distance from the tower, this may not be the case. This difference between the instantaneous temperatures of the water at the measurement point and as it leaves the fill is termed “Thermal Lag” and can affect the evaluation of the tower performance.

The calculation procedure contained herein to compute the theoretical Thermal Lag assumes the water inventory reacts in plug flow manner where the water first in is also first out. The tester is cautioned that plug flow cannot be guaranteed and it will be necessary to closely monitor system stability to assure data accuracy.

Section J2. Adjustment for Thermal Lag

The time interval from when the cooled water first reaches the collecting basin until it reaches the Cold Water Temperature measurement station can be estimated as:

$$lag = \frac{VOL_{basin}}{60 * Q_{tw}} \qquad (J-1)$$

Where:

lag = Thermal lag time, minutes

VOL_{basin} = Volume of water in basin during test, L (gal)

Q_{tw} = Test water circulation rate, L/s (gal/s)

If the cold water flows a significant distance in a canal from the tower to the measurement location, the lag time in the canal (VOL_{canal}/Q) should be added to the average basin lag time. For measurements at the side of a round basin or end of a longitudinal basin, the average lag time would be the value “lag” from Equation J-1 divided by two. For measurements adjacent to the middle of a longitudinal basin, the average lag time would be the value “lag” from Equation J-1 divided by four.

In those instances where the value of the average thermal lag equals or exceeds five minutes, the test period shall be lengthened by a like amount and the test averages based on compensating time spans to negate the effect of the thermal lag.

Specifically, for an overall test period ranging from a starting time t_1 to ending time t_2, the minimum duration of the test in minutes is $(t_2 - t_1) \geq 60$ + lag. The following readings shall be averaged over the time period t_1 through t_2:

(a) Hot Water Temperature (T_{hw})

(b) Dry-bulb Temperature - if required (T_{db})

(c) Wet-bulb Temperature (T_{wb})

(d) Flow Rate of Circulating Water (Q_w)

(e) Flow Rate and Temperature of any water other than the circulating water entering the tower basin [e.g. Make-up (Q_{mu} and T_{mu})]

The remaining data (listed below) shall be averaged over the time period $(t_1 + lag)$ through $(t_2 + lag)$:

(f) Temperature of main stream of Cold Water leaving the tower (T_{cw})

(g) Flow Rate and Temperature of any water other than the circulating water leaving the tower basin [e.g. Blowdown (Q_{bd} and T_{bd})]

The location of auxiliary flows (Make-up and Blowdown) relative to the Cold Water Temperature measurement location must be considered. Intermittent streams must be well understood and thoroughly documented as the effects of such streams on average thermal lag may be significant.

For example, for a system with an average 10 minute thermal lag the total duration of the test must extend for at least 70 minutes (60+lag). If the test started at 1:00, hot water, dry bulb, wet bulb, flowrate, and makeup flowrate would be averaged for the period 1:00 to 2:00, while cold water temperature and blowdown flowrate would be averaged for the period 1:10 to 2:10.

APPENDIX K
References

K.1 References — Normative.

Listed here are standards, handbooks, and other publications essential to the implementation of this standard. All references in this section are considered a part of this standard.

1. *CTI Bulletin STD-146(Current Revision), Standard for Water Flow Measurement,* Cooling Technology Institute, PO Box 73383, Houston, TX 77273
2. *Standard Methods for the Examination of Water, Sewage, and Industrial Wastes, 22nd Edition,* American Public Health Association, Publication Sales Office, P.O. Box 753, Waldorf, MD 20604-2014

K.2 References — Informative.

Listed here are standards, handbooks, and other publications which may provide useful information and background, but are not considered essential to the implementation of this standard. References in this section are not considered a part of this standard.

1. *CTI Bulletin PTG-156(Current Version), Preparation for an Official CTI Thermal Performance, Plume Abatement or Drift Emission Test,* Cooling Technology Institute, PO Box 73383, Houston, TX 77273
2. *CTI ToolKit Current Version,* Cooling Technology Institute, PO Box 73383, Houston, TX 77273-2010
3. *ASHRAE Handbook — Fundamentals,* Chapter 1, "Psychrometrics", American Society of Heating, Refrigerating and Air-Conditioning Engineers, Atlanta, GA. 2017
4. *ASHRAE Handbook — HVAC Systems and Equipment,* Chapter 40, "Cooling Towers", American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE), 1791 Tullie Circle, Atlanta, GA 30329. 2016.
5. *ASME Performance Test Code (PTC) 19.1 – (2013), Instruments and Apparatus, Part I, Measurement Uncertainty,* American Society of Mechanical Engineers (ASME), United Engineering Center, 345 East 47th Street, New York, NY 10017
6. *ASME Performance Test Code (PTC) 19.3 - (2016), Temperature Measurement,* American Society of Mechanical Engineers (ASME), United Engineering Center, 345 East 47th Street, New York, NY 10017
7. *CTI Bulletin STD-201(Current Version), Performance Rating of Evaporative Heat Rejection Equipment.* Cooling Technology Institute, PO Box 73383, Houston, TX 77273-2010

APPENDIX L
Determining Wind Velocity on Natural Draft Cooling Towers

Section Ll. General

On natural draft cooling towers, the wind at the top of the shell, can have a significant effect on the thermal performance. For this reason, the code imposes a limitation on the wind velocity at the top of the shell.

Section L2. Determining Wind Velocity

The preferred way is to ***measure*** the wind velocity at the discharge of the shell using an anemometer positioned at the right altitudes on a meteorological mast or suspended from a balloon, upwind of the cooling tower. However, too often this method is not practical.

L2.1 One alternative, visual observation of the plume, can give some indication of the wind velocity at the top of the shell, since the plume angle bears some relationship to the ratio of exit air velocity and wind velocity. Such visual evaluation is subject to human interpretation, assumes that all natural draft cooling towers have essentially the same exit air velocity, and is affected by the dilution rate of the plume into the atmosphere and the weather conditions. Also, in mild weather, there may be almost no visible plume rising from the cooling tower.

L2.2 A second alternative, measuring the wind velocity near ground level (3 to 10 m above grade) and then predicting the wind velocity at the discharge of the shell using an equation is easy, but does not always yield a good comparison to the actual velocity at the top of the shell. After extensive research using meteorological masts, equations were developed that represent the relationship of the wind velocity as a function of altitude. It should be noted that these equations were developed primarily for use in structural design of tall structures, where conservatism in estimated velocities (i.e. loads) for elevated structures was desired. There is no precise way to correlate velocities measured at a low elevation to actual velocities at a higher elevation. Two equations are in common use to ***estimate*** wind velocity at altitude, and yield similar results:- Hellman equation:

$$V = K[(1 + 2.81) * (Z + 4.75)\ ]$$

- German Cooling tower test code DIN 1947 equation:

$$V = (K_D)(Z)^{0.2}$$

Where:

V = wind velocity (m/s)

Z = altitude (m)

K, K_D = constant

Using these equations, the wind velocity at the top of the shell, in average, can be estimated from the wind velocity measured at ground level.

- Hellman equation:

$$V_t = V_g \left[\frac{[(1 + 2.81) * (Z_t + 4.75)\ ]}{[(1 + 2.81) * (Z_t + 4.75)\ ]}\right]$$

- German Cooling tower test code DIN 1947 equation:

$$V_t = V_g \left(\frac{Z_t}{Z_g}\right)^{0.2}$$

Where:

V_t = Wind velocity at the top of the shell (m/s)

V_g = Wind velocity at ground level between 3 to 10 m (10 to 33 ft) (m/s).

Z_t = Level of the top of the shell compare to the ground level (m).

Z_g = level of the wind velocity measurement from the ground level (m).

APPENDIX M

Expressing Tower Capability as a Function of Deviation from Design Cold Water Temperature

Section Ml. General

The purpose of this appendix is to provide a methodology for determining cooling tower capability as a function of temperature deviation from design cold water temperature.

Section M2. Methodology

The Capability of a cooling tower, as determined by a performance test, is usually reported as a percentage of the design water flow rate. It is calculated from the ratio of the Adjusted Test Flow Rate to the flow rate predicted from the manufacturer's data for the measured test conditions.

$$\%CAP = \left(\frac{Q_{adjust}}{Q_{pred}}\right) * 100 \qquad (M-1)$$

This equation shows that tower capability is inversely proportional to predicted flow (Q_{pred}). From that it follows that if a cooling tower is operating with design (100%) flow, design range, design fan power and at design wet bulb temperature, but producing a cold water temperature equal to that expected when operating at 90% design flow, the tower capability is:

$$\%CAP = \left(\frac{100\%}{90\%}\right) * 100 = 111.11\% \qquad (\mathrm{M}-2)$$

Similarly, if the tower were producing a cold water temperature equal to that expected when operating at 110% design flow, the tower capability would be:

$$\%CAP = \left(\frac{100\%}{110\%}\right) \cdot 100 = 90.91\% \qquad (M-3)$$

Using this approach in conjunction with the Performance Curves supplied by the tower manufacturer, three points of tower leaving water temperature as a function of tower capability can be readily obtained and plotted. The curve can then be used to determine the tower leaving water temperature at design flow, range, and fan motor power for any capability falling within the scope of the curve. Extrapolation of the curve to any significant degree is not recommended.

From the manufacturers performance curves, tabulate the cold water temperatures predicted for the design range, design wet bulb temperature and, if applicable, the design Relative Humidity for each flow rate for which performance curves are provided.

Section M3. Example

M3.1 SI Example. Refer to the example test evaluation in Appendix C.

Step 1 On each of the three performance curves submitted by the manufacturer, enter the abscissa at the design wet bulb temperature of 26.0°C and project a line vertically upward to intersect the design range curve (18.8°C). From the ordinate, read the cold water temperature corresponding to the intersect:

% Flow	% Capability	CWT (°C)
90%	111.11%	29.84°
100%	100.00%	30.60°
110%	90.91%	31.36°

Step 2 With Cold Water Temperature as the ordinate and Tower % Capability as the abscissa, plot the tabulated Cold Water Temperatures at the corresponding values for % Capability and fit a curve through the three points. Extrapolation beyond the submitted data is not recommended.

Step 3 Enter the abscissa at the % Tower Capability determined by the test (102.2%) and project a line vertically upward to intersect the curve. At the intersection, read the corresponding Cold Water Temperature as 30.44°C. The difference between this temperature and the Design Cold Water Temperature is the Cold Water Temperature Deviation associated with the tested tower capability, -0.16°C.

M3.2 IP Example. Refer to the example test evaluation in Appendix D.

Step 1 On each of the three performance curves submitted by the manufacturer, enter the abscissa at the design wet bulb temperature of 78.8°F and project a line vertically upward to intersect the 100% Range curve (33.84°F). From the ordinate, read the cold water temperature corresponding to the intersect:

% Flow	% Capability	CWT (°F)
90%	111.11%	85.71°
100%	100.00%	87.07°
110%	90.91%	88.84°

Step 2. With Cold Water Temperature as the ordinate and Tower % Capability as the abscissa, plot the tabulated Cold Water Temperatures at the corresponding values for % Capability and fit a curve through the three points. Extrapolation beyond the submitted data is not recommended.

Step 3. Enter the abscissa at the % Tower Capability determined by the test (101.4%) and project a line vertically upward to intersect the curve. At the intersection, read the corresponding Cold Water Temperature as 86.85°F. The difference between this temperature and the Design Cold Water Temperature is the Cold Water Temperature Deviation associated with the tested tower capability, -0.22°F

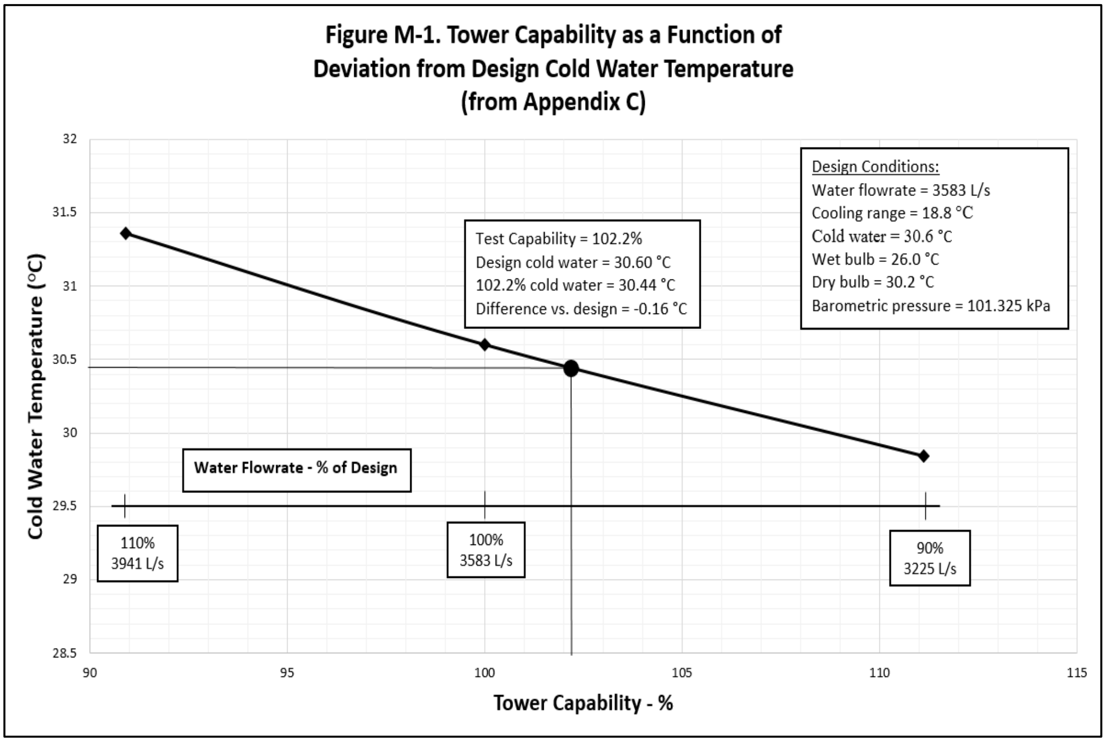


Figure M-1. Tower Capability as a Function of Deviation from Design Cold Water Temperature (from Appendix C)
Design Conditions:
Water flowrate = 3583 L/s
Cooling range = 18.8 °C
Cold water = 30.6 °C
Wet bulb = 26.0 °C
Dry bulb = 30.2 °C
Barometric pressure = 101.325 kPa
Test Capability = 102.2%
Design cold water = 30.60 °C
102.2% cold water = 30.44 °C
Difference vs. design = -0.16 °C
Water Flowrate - % of Design
110% 3941 L/s
100% 3583 L/s
90% 3225 L/s
Cold Water Temperature (°C)
Tower Capability - %
32
31.5
31
30.5
30
29.5
29
28.5
90
95
100
105
110
115

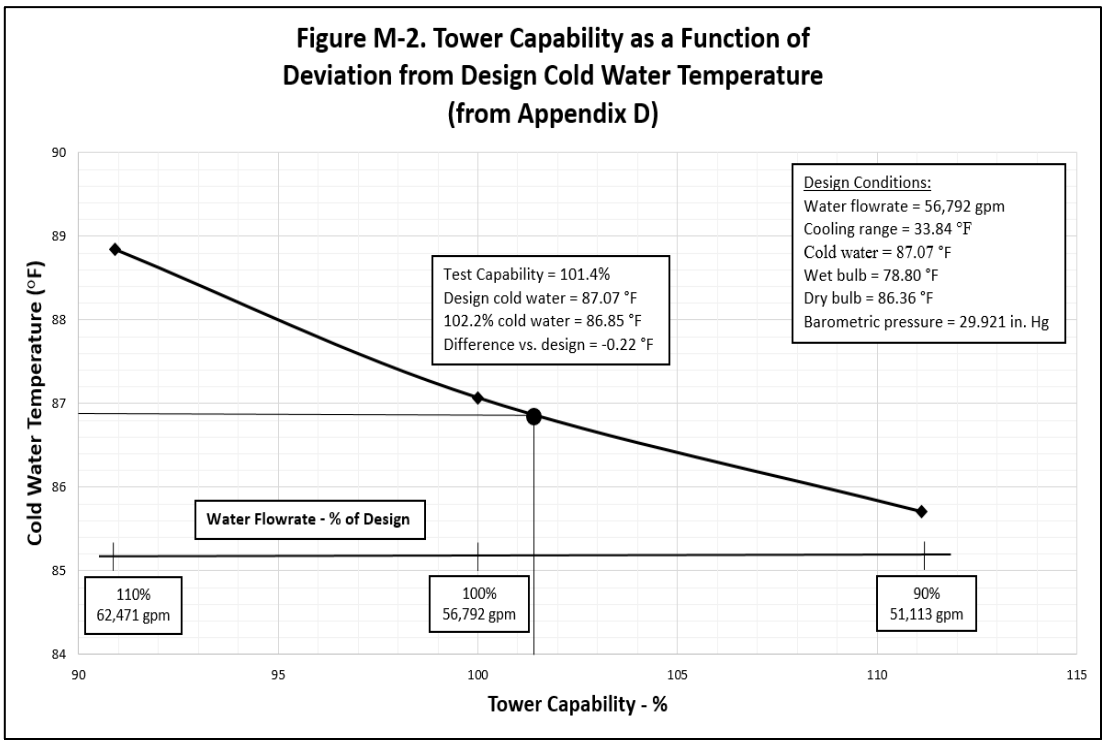


Figure M-2. Tower Capability as a Function of Deviation from Design Cold Water Temperature (from Appendix D)
Design Conditions:
Water flowrate = 56,792 gpm
Cooling range = 33.84 °F
Cold water = 87.07 °F
Wet bulb = 78.80 °F
Dry bulb = 86.36 °F
Barometric pressure = 29.921 in. Hg
Test Capability = 101.4%
Design cold water = 87.07 °F
102.2% cold water = 86.85 °F
Difference vs. design = -0.22 °F
Water Flowrate - % of Design
110% 62,471 gpm
100% 56,792 gpm
90% 51,113 gpm
Cold Water Temperature (°F)
Tower Capability - %
90
89
88
87
86
85
84
90
95
100
105
110
115

APPENDIX N
Motor Power Correction for Line Voltage Loss

Section N.1 General

Mechanical draft towers which have long cable distances between where the power is measured and the motor propelling the fan, a line voltage loss correction is necessary. If mutually agreed by all parties this section may be omitted. This appendix describes the technique to perform the line loss correction.

Table N-1 Test Conditions

Nominal Motor Size	110 KW (150 HP)
Cable Length between switchgear and motor	192 m (630 ft)
Cable Description	Size 4/0 metallic sheathed copper
Number of Phases on Power Supply	3
Measured KW at switchgear	115.0
Measured voltage at switchgear	460
Measured amperage at switchgear	175.5
Motor Efficiency at Test Load	0.94
Average Temperature of conductors during test	25°C (77°F)

Section N.2 Example

Step 1: Determine reactance for wires ‘X_L’ – From Table 9 of NEC (National Electrical Code)

$$X_L = 0.051(ohms\ to\ neutral\ per\ 1000\ feet\ (75°C))$$

Step 2: Determine AC resistance for uncoated copper wires ‘R’ – From Table 9 of NEC

$$R = 0.063\ (ohms\ to\ neutral\ per\ 1000\ feet\ (75°C))$$

Step 3: Calculate power factor and angle from the power, voltage and amperage measured at the switchgear

$$PF = \frac{P * 1000}{I * V * \sqrt{3}} \quad (N-1)$$

$$PF = Power\ Factor$$

$$I = Measured\ Amperage\ at\ switchgear$$

$$V = Measured\ Voltage\ at\ switchgear$$

$$P = Measured\ Power\ at\ switchgear\ (kW)$$

$$PF = \frac{115 * 1000}{\sqrt{3} * 175.5 * 460}$$

$$PF = 0.8224$$

$$\theta = \cos^{-1}(PF) \quad (N-2)$$

$$\theta = Power\ Factor\ Angle$$

$$\theta = (.8224)$$

$$\theta = 0.6052\ radians$$

Step 4: Calculate Power Factor Correction ‘Z_c’

$$Z_c = R * cos\ \theta\ + X_L * \ sin\ \theta \quad (N-3)$$

$$Z_c = 0.063 * cos\ 0.6052\ + 0.051_L * sin\ 0.6052$$

$$Z_c = 0.0808\ ohms\ to\ neutral\ per\ 1000\ feet\ at\ 75°C$$

Step 5: Determine Temperature coefficient of resistance ‘α’* for Copper at 75°C

Cautionary Statement: This calculation includes a correction for an average conductor temperature of 25°C, this is done in order to illustrate that methodology. However, field measurement of conductor temperature is most likely impractical and it will have at most a minimal effect on the calculated cooling tower capacity. Therefore, it is reasonable to ignore the conductor temperature correction.

*Found in Note 2 below Table 8 of NEC (All of the table data is based on 75°C)

α = 0.00323 for copper and 0.00330 for aluminum at 75°C

Step 6: Correct impedance to 25°C and Power Factor

$$R_2 = Z_c * \ [1 + \alpha * (T_2 - 75)] \quad (N-4)$$

$$R_2 = Impedance\ correction\ (ohms\ to\ neutral\ /1000ft\ (@\ T2)$$

$$T_2 = Temperature\ of\ the\ wire$$

$$R_2 = \ 0.0808 * \ [1 + 0.00323 * (25 - 75)]$$

$$R_2 = \ 0.0678$$

Step 7: Calculate voltage drop, line to neutral and line to line

$$VD_{L-N} = R_2 * \left(\frac{L}{1000} * I\right) \quad (N-5)$$

$$L = Cable\ Length\ between\ switchgear\ and\ motor\ (ft)$$

$$VD_{L-N} = Voltage\ drop\ line\ to\ neutral$$

$$VD_{L-N} = 0.0678 * 630/1000 * 175.5$$

$$VD_{L-N} = 7.496\ volts$$

$$VD_{L-L} = VD_{L-N} * \sqrt{3} \quad (N-6)$$

$$VD_{L-L} = Voltage\ drop\ line\ to\ line$$

$$VD_{L-L} = 7.496 * \sqrt{3}$$

$$VD_{L-L} = 12.983\ volts$$

Step 8: Calculate power loss

$$PL = \frac{VD_{L-L} * \sqrt{3} * \ I\ * \ PF}{1000} \quad (N-7)$$

$$PL = Power\ loss\ in\ kW$$

$$PL = \frac{12.983 * \sqrt{3} * 175.5 * .8224}{1000}$$

$$PL = 3.246\ kW$$

Step 9: Calculate power input to motor and brake horsepower

$$P_{in} = P - PL \quad (N-8)$$

$P_{in} = Input\ power\ at\ the\ motor$

$P_{in} = 115 - 3.246$

$P_{in} = 111.75\ kW$

$$BHP = \frac{P_{in} * \eta}{0.746} \quad (N-9)$$

$P_{in} = Input\ power\ at\ the\ motor\ kW$

$\eta = Motor\ Efficiency$

$$BHP = \frac{111.75 * .94}{0.746}$$

$$BHP = 140.81$$

APPENDIX O
Standard Specification for a Mechanically Aspirated Wet & Dry-Bulb Temperature Measuring Instrument

Section O.1 Introduction

This appendix specifies the main features of an apparatus to accurately measure wet and dry-bulb temperatures for the purposes of cooling tower testing. It does not intend to specify the full details of such devices so as to ensure well designed instruments, which have gained acceptance in the industry, are not arbitrarily excluded. However, it does intend to present the critical elements in the design and construction of such an apparatus for the accurate and repeatable measurements of wet-bulb and dry-bulb temperature suitable for testing cooling towers in their outdoor environment.

The approach of specifying only the essential features of a measuring instrument necessarily has some limitations. It should be understood, therefore, that good practice should be followed, both in implementing the requirements of this appendix and in detailing aspects of the design and construction that are not specified.

Section O.2 Definitions

For the purposes of this appendix the following definitions shall apply:

Aspirated Psychrometer: An instrument for determining relative humidity by using two thermometers to measure the wet-bulb and dry-bulb temperatures of air in which the air is drawn over the sensors by mechanical means at some predetermined velocity.

Dry-bulb Temperature: The temperature of air measured by a thermometer with a dry sensor, properly shielded from extraneous radiation.

Relative Humidity: The ratio of the actual mass of water vapor in an air sample to the mass of water vapor in a saturated sample at the same temperature and pressure.

Sensor: That portion of the temperature probe or sheath that actually senses the temperature.

Stem: The portion of a thermometer that does not contain the sensor.

Thermometer: Any temperature measuring device.

Wet-bulb Depression: The temperature difference between the dry-bulb temperature and the wet-bulb temperature of an air sample.

Wet-Bulb Instrument: An instrument with only one thermometer fitted with a wetted wick that is used to measure the wet-bulb temperature of the air.

Wet-bulb Temperature: The equilibrium temperature measured by a thermometer with a sensor covered with a wetted wick, shielded from extraneous radiation, and placed in a moving airstream. It closely approximates but is not equal to the thermodynamic wet-bulb temperature (also known as the adiabatic saturation temperature).

Wick: The entire cotton covering used to cover the thermometer sensor and a portion of which extends into the water reservoir.

Section O.3 References

ANSI/ASHRAE 41.6-1994 (RA 2006), Standard Method for Measurement of Moist Air Properties

ASHRAE Design Specification for Wet-Bulb Aspirator Apparatus, Final Report, November 2010, Technical Research Project 1460

ISO Cooling Tower Test Code Draft Standard Annex - Wet-bulb Determination

Section O.4 Psychrometers & Wet-Bulb Instruments

O.4.1 Scope This appendix addresses the use of mechanically aspirated instruments to determine the wet-bulb and dry-bulb temperature of air wet-bulb above 0.5°C (33°F).

O.4.2 Principle Wet-bulb instruments and Psychrometers, if properly designed and constructed are rugged and well suited for cooling tower testing. Psychrometers consist of two thermometers, one of which is maintained in a wetted condition with a moistened wick over the sensor portion. Wet-bulb instruments consist of only one thermometer fitted with a moistened wick over the sensor portion. The sensor portion of the thermometers are well shielded from extraneous radiation* and a powered blower is utilized to ventilate the device at a specific velocity of air over the sensors. Evaporation from the surface of the wetted wick into the airstream cools the sensor to a steady state temperature through evaporation such that there is a balance between the heat lost and that gained through conduction, convection and radiation.

O.4.3 Thermometers: Thermometers may be liquid-in-glass (in which case they shall be the partial immersion type), resistance temperature devices, thermistors or other types complying with the requirements specified herein. The accuracy shall be such that the uncertainty in the value of the instrument reading is no greater than 0.1°C (0.2°F) including any uncertainty associated with the calibration.

O.4.3.1 Each thermometer shall consist of a temperature sensor of essentially cylindrical shape that is supported on a single stem. The diameter of the sensors shall not differ substantially from that of the stem. The free end of each sensor shall be smoothly rounded.

O.4.3.2 With transverse ventilation, in which the airstream flows essentially perpendicular to the thermometer, the diameters of the sensors (excluding wet covering) shall be not less than 1 mm (0.04in.) and not greater than 7 mm (0.3in.).

O.4.3.3 With axial ventilation, in which the airstream flows essentially parallel to and coaxial with the thermometer, the diameters of the sensors (excluding wet covering) shall be not less than 2 mm (0.08 in.) and not greater than 7 mm (0.3 in.).

O.4.3.4 The stem of each thermometer shall be clear of obstructions and freely exposed to the airstream over a length measured from the sensor of not less than 1.5 times the length of the wicking covering the sensor as prescribed in O.3.4.

O.4.3.5 Heat from the fan or its motor shall not affect thermometer readings.

O.4.4 Wet-Bulb Wick and Water Reservoir

O.4.4.1 The wet-bulb wick shall be fabricated from hydrophilic, undressed white cotton muslin.

O.4.4.2 After fabrication, the wick shall be clean and thoroughly rinsed.

O.4.4.3 The wick shall tightly and completely cover the sensor.

O.4.4.4 The wick portion extending into the water reservoir shall have a minimum cross section consistent with an adequate water feed to the wet-bulb sensor for the highest rates of evaporation.

O.4.4.5 Routine inspection and replacement of the wick material is required to ensure accurate wet bulb temperatures are being obtained.

O.4.4.6 The water reservoir shall contain sufficient volume of water to adequately supply water to the wick for a period of at least one hour at the highest rates of evaporation.

O.4.4.7 A water reservoir shall not obstruct the airflow, and its contents shall not affect the humidity of the sample air.

O.4.5 Water The water used to fill the instrument reservoir and wet the wick and probe cover shall be distilled or deionized water.

O.4.6 Airflow

O.4.6.1 The velocity of the airflow over both the wet and dry-bulbs shall be 4.0 ± 0.5m/s (800 ± 100 ft/min) for transverse ventilation and 4.5 to 5.5 m/s (950 to 1050 ft/min) for axial ventilation.

O.4.6.2 The sample air shall not pass over any obstruction or through a fan before it passes over the wet and dry-bulb sensors.

O.4.6.3 With probes in axial orientation with respect to airflow, the flow direction shall be from the free end of each sensor toward the support end.

O.4.6.4 No air that has been cooled by interaction with the wet-bulb or by the wick shall impinge on the dry-bulb. This may be ensured by having two separate incoming airstreams. Air that has been discharged from the instrument shall be directed away from the instrument's intake as to not affect the temperature of the incoming air.

O.4.7 Radiation Shields

O.4.7.1 With transverse ventilation, radiation shields in the form of parallel plates, shall be provided to shield the wet and dry-bulbs from extraneous radiation. A shield may also be provided between sensors. The clearance between the wet and dry-bulbs and the shields shall be not less than half the diameter of the wet-bulb.

O.4.7.2 With axial ventilation, concentric radiation shields shall be provided for the wet and dry-bulb devices. The shield around the wet-bulb plays a vital role in reducing the radiative heat transfer between the bulb and its surroundings by approximately a factor of three.

APPENDIX-P (INFORMATIVE)

Methodology for Selection of the Test Period

This appendix presents a method for selecting the most stable test periods from a thermal test data set. As no methodology can handle every situation, this appendix is provided for guidance. However, in cases where this method is not used to select the reported test periods it is the test agency's responsibility to explain, in the test report, what method was used.

Ideally the complete thermal data set will conform to the requirements of Chapter 2.4. But, in such cases where parts or all of the data set do not conform to Chapter 2.4 and all parties to the test have agreed to use the data set then this methodology can be used in conjunction with good engineering judgement (supported by graphical methods, etc.).

As alluded to in the previous paragraph the most stable test period may be identified by graphically analyzing the data. Such a graphical analysis would review plots of hot water, cold water, wet bulb and dry bulb (if applicable) temperatures throughout the period of data collection. The period(s) showing the least fluctuation/rate of change are candidates for further analysis and reporting. Nonetheless, using a graphical analysis alone might overlook the weighting of some parameters on the results. For example, the dry bulb has almost no influence on the performance of a wet mechanical draft tower, whereas the wet-bulb has significant influence. It is recommended that the weighted analysis method be used in conjunction with graphical methods.

The rigorous way to assess the influence of each parameter is to compute a sensitivity table as per Appendix U and then to compute the uncertainty of each reading. The best period would then be the one with the lowest uncertainty. However, such calculations are time consuming, difficult to perform and not necessary in many situations. For this reason, weighting factors are commonly used. The weighting factors, suggested below, may not be ideal for every situation, but if they are used with graphical analysis and engineering judgement they should be appropriate.

For mechanical draft towers the most stable periods can be ranked with the product of the sum of the factored slopes of the data trend lines (SFSTL) x Std Dev Weighted. The lowest value is the most stable where:

$$SFSTL_{Weighted} = 0.1 * |Slope_{Hw}| + 0.4 * |Slope_{Cw}| + 0.2 * |Slope_{Wb}|$$

$Slope_{Hw}$ = Slope of the hot water temperature trend line per hour

$Slope_{Cw}$ = Slope of the cold water temperature trend line per hour

$Slope_{Wb}$ = Slope of the wet bulb temperature trend line per hour

$$StdDev_{Weighted} = 10 * \sqrt{(0.4 * Variance_{Cw})^2 + (0.1 * Variance_{Hw})^2 + (0.2 * Variance_{Wb})^2}$$

$Variance_{Cw}$ = Variance for cold water along the data trend line

$Variance_{Hw}$ = Variance for hot water along the data trend line.

$Variance_{Wb}$ = Variance for wet bulb along the data trend line.

APPENDIX Q

SI Evaluation of Mechanical-Draft Helper Cooling Tower Test Where Inlet (Hot) Water Temperature is Specified Using the Performance Curve Method

Section Q1. General

The purpose of this appendix is to describe and illustrate the performance curve methodology for evaluating a thermal performance test on a mechanical draft helper cooling tower where inlet (hot) water temperature is specified, as described in Section 11 of the standard.

Section Q2. Design and Test Conditions

Design and test conditions for a given mechanical draft helper cooling tower are summarized in the following table.

Table Q-1 Cooling Tower Design and Test Data

Parameter	Design	Test
Water Flow Rate (Q_{wt})	3583 l/s	3623 l/s
Hot Water Temp (T_{hw})	49.4 °C	46.5 °C
Cold Water Temp (T_{cw})	30.6 °C	29.04 °C
Inlet Wet Bulb Temp (T_{wb})	26.0 °C	24.53 °C
Inlet Dry Bulb Temp (T_{db})	30.2 °C	25.52 °C
Fan Driver Power ($\dot{W}$)	107.0 kW	113.0 kW
Barometric Pressure (P_{bp})	101.325 kPa	98.8 kPa
Liquid to Gas Ratio	1.300	

In accordance with paragraph 11.2 of this standard, the manufacturer has submitted Performance Curves with Cold Water presented as a function of Wet Bulb Temperature with Inlet (Hot) Water Temperature and Water Flow Rate as parameters (see Figures Q-1, Q-2, and Q-3).

Alternatively, thermal test data analyzed with constant range performance curves, as demonstrated in Appendix C, will result in the same analysis result (tower capacity).

Section Q3. Evaluation Procedure

The steps for evaluating the test are as follows:

Step 1. Determine the Predicted Cold Water Temperatures

Scribe the submitted performance curves vertically at the test wet bulb temperature (24.53°C) to determine the predicted Cold Water Temperature associated with the test Wet Bulb Temperature at each of the three Inlet (Hot) Water Temperatures and three Water Flow Rate conditions included on the performance curves. Values of these predicted Cold Water Temperatures are tabulated in Table

Q-2.

Table Q-2 Curve Points for Cold Water Temperature vs Inlet (Hot) Water Temperature at 24.53 °C Test Wet Bulb

Inlet (Hot) Water Temperature	90% Flow	100% Flow	110% Flow
45.64 °C	28.66	29.28	30.10
49.4 °C	29.12	29.86	30.75
53.16 °C	29.54	30.39	31.33

Step 2. Construct the First Cross Plot

The values of Cold Water Temperature are plotted against Inlet (Hot) Water Temperature using Flow Rate as a parameter (see Figure Q-4).

Scribe Figure Q-4 vertically at the test Inlet (Hot) Water Temperature (46.5°C) to determine the predicted Cold Water Temperatures at each of the three Water Flow Rates. The values of these predicted Cold Water Temperatures are tabulated in Table Q-3.

Table Q-3 Curve Points for Cold Water Temperature vs Water Flow Rate at 24.53 °C Test Wet Bulb and 46.5 °C Test (Hot) Inlet Water Temperature**.**

90% Flow	100% Flow	110% Flow
28.73	29.41	30.28

Step 3. Second Cross Plot

Using the values of Cold Water Temperature in Table Q-3, plot these temperatures as a function of the three flow rate percentages (see Figure Q-5).

Step 4. Determine the Predicted Flow Rate

On Figure Q-5 scribe a horizontal line at the test Cold Water Temperature to intersect the curve. At the intersection, project a line vertically downward to find the predicted Water Flow Rate associated with the test Wet Bulb, Inlet (Hot) Water Temperature, and Cold Water Temperature.

$$Q_{w,pred} = 95.35\%\ Design\ Flow = 3416\ L/s$$

Step 5. Calculating the Adjusted Test Water Flow Rate

The adjusted test Water Flow Rate is computed from equation 11.1, using the values for air density at the fan inlet and the fan driver output power, at both the test and design conditions.

To comply with this standard, the design and test values for density (ρ), specific volume (v) and enthalpy (h) of air must be determined using either CTI ToolKit or the psychrometric tables in ASHRAE Handbook – Fundamentals (SI). For this example, the CTI Toolkit was used to generate all psychrometric properties.

Since the evaluation is based upon the psychrometric properties of air at the fan inlet, different procedures must be employed for forced draft and induced draft towers.

a. Forced Draft Tower

For a forced draft tower, the fan inlet air conditions are the same as the tower inlet air conditions. Therefore, the test density (ρ_t) and test specific volume (υ_t) are computed directly from the measured test values of wet-bulb, dry-bulb and barometric pressure. In accordance with paragraph 11.2, the manufacturer shall supply the design conditions at the tower's air inlet.

Table Q-4. Design and Test Values for Air Characteristics

Parameter	Design	Test
Barometric Pressure (P_{bp})	101.325 kPa	98.8 kPa
Inlet Wet Bulb Temp (T_{wb})	26.0 °C	24.53 °C
Inlet Dry Bulb Temp (T_{db})	30.2 °C	25.52 °C
Humidity Ratio (HR)	0.01966 kg/kg	0.01969kg/kg
Specific Volume (υ)	0.8866m^3/kg	0.8952 m^3/kg
Inlet Enthalpy (h_i)	80.6308 kJ/kg	75.8212 kJ/kg
Density (ρ)	1.15013 kg/m^3	1.13902 kg/m^3
Relative Humidity (RH)	71.98%	92.36%

b. Induced Draft Tower

For the induced draft tower, the fan air conditions are the tower discharge conditions. The code requires that both the design and test discharge air properties be determined by a heat balance calculation. Calculating design discharge air properties is a straightforward procedure while calculating test discharge air properties requires combining the heat balance Equation (Q-2) with Equation 5.1 and iterating for a solution.

The heat balance equation states that the heat gain by the air equals the heat loss of the water. The formula follows:

$$L * c_{pw} * (T_{hw} - T_{cw}) = G * (h_o - h_i) \qquad (Q-1)$$

Where:

L = Water mass flow rate, kg/hr

c_{pw} = Specific Heat of Water = 4.186 kJ/kg °C

T_{hw} = Hot Water Temperature, °C

T_{cw} = Cold Water Temperature, °C

G = Mass flow rate of dry air through the tower, kg/hr

h_o = Enthalpy of air leaving the tower, kJ/kg

h_i = Enthalpy of air entering the tower, kJ/kg

Rearranging to separate the exit air enthalpy, the heat balance equation becomes:

$$h_o = \left(\frac{L}{G}\right) * c_{pw} * (T_{hw} - T_{cw}) + h_i \qquad (Q-2)$$

Determine the enthalpy of the entering air at design conditions as:

$$h_i = 80.6308 \frac{kJ}{kg}$$

Then, substituting all design values into Equation (Q-2), calculate the design outlet enthalpy.

$$h_{od} = (1.300) * 4.186 * (49.4 - 30.6) + 80.6308$$

$$h_{od} = 182.9366 \frac{kJ}{kg}$$

Assuming the discharge air is saturated at this enthalpy and barometric pressure, determine the design discharge air temperature (T_d), density (ρ_d) and specific volume (υ_d) as:

$$T_d = 41.866\ °C$$

$$\rho_d = 1.0863 \frac{kg}{m^3}$$

$$\upsilon_d = 0.9708 \frac{m^3}{kg}$$

Next the discharge air characteristics at test conditions are calculated. First calculate the test L/G by substituting all known values into equation (5.1):

$$\frac{L}{G} = 1.3 * \left(\frac{3623}{3583}\right) * \left(\frac{107}{113}\right)^{\frac{1}{3}} * \left(\frac{\rho_t}{1.0863}\right)^{\frac{1}{3}} * \left(\frac{\upsilon_t}{0.9708}\right)$$

$$\left(\frac{L}{G}\right) = 1.2935 * (\rho_t)^{1/3} * (\upsilon_t)$$

Substituting this L/G expression into equation (Q-2) to calculate the test exit air enthalpy, h_{ot}

$$h_{ot} = 1.2935 * (\rho_t)^{1/3} * (\upsilon_t) * 4.186 * (46.5 - 29.04) + 75.8212$$

Simplifying,

$$h_{ot} = 94.5388 * (\rho_t)^{1/3} * (\upsilon_t) + 75.8212$$

At this point, guess at a discharge air temperature and, assuming saturation, determine density (ρ) and specific volume (υ) at that temperature. Then, substituting these values for density (ρ) and specific volume (υ) into the final heat balance expressions, calculate an enthalpy h_{ot}.

Compare the calculated value for enthalpy h_{ot} to the actual value of enthalpy at the assumed temperature and continue iterating discharge air temperature until a suitable temperature is selected for which the calculated value of enthalpy hot matches the actual value.

For the first estimate of leaving air temperature, use the average of T_{hw} and T_{cw} at test conditions. Typical iteration values for this example are given in Table Q-5 for Barometric Pressure of 98.8 kPa.

Table Q-5 Iteration for Enthalpy of Leaving Air

T_{ao}	ρ_t	v_t	h_{ot} actual	h_{ot} computed	Error %
38.0	1.07797	0.9694	153.79	169.79	+10.4%
40.0	1.06791	0.9837	170.16	170.88	+0.04%

Step 6. Determine Cooling Tower Capability

Substituting the psychrometric values at 40.0°C, the adjusted flow rate is now calculated from equation 11.1.

$$Q_{wt\ adj} = 3623 * \left(\frac{107}{113}\right)^{\frac{1}{3}} * \left(\frac{1.0791}{1.0863}\right)^{1/3} = 3537\ l/s$$

The tower capacity is computed from equation 11.2.

$$C = \left(\frac{3537}{3416}\right) * 100 = 103.5\%$$

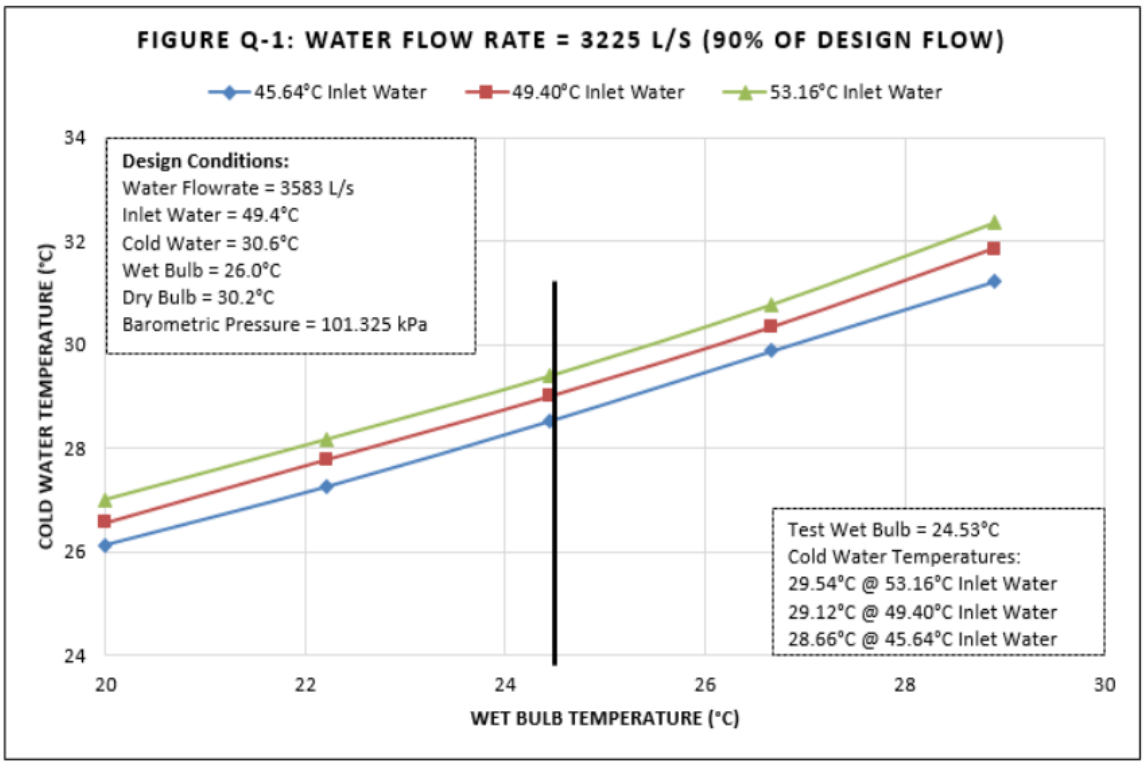


FIGURE Q-2: WATER FLOW RATE = 3583 L/S (DESIGN FLOW)

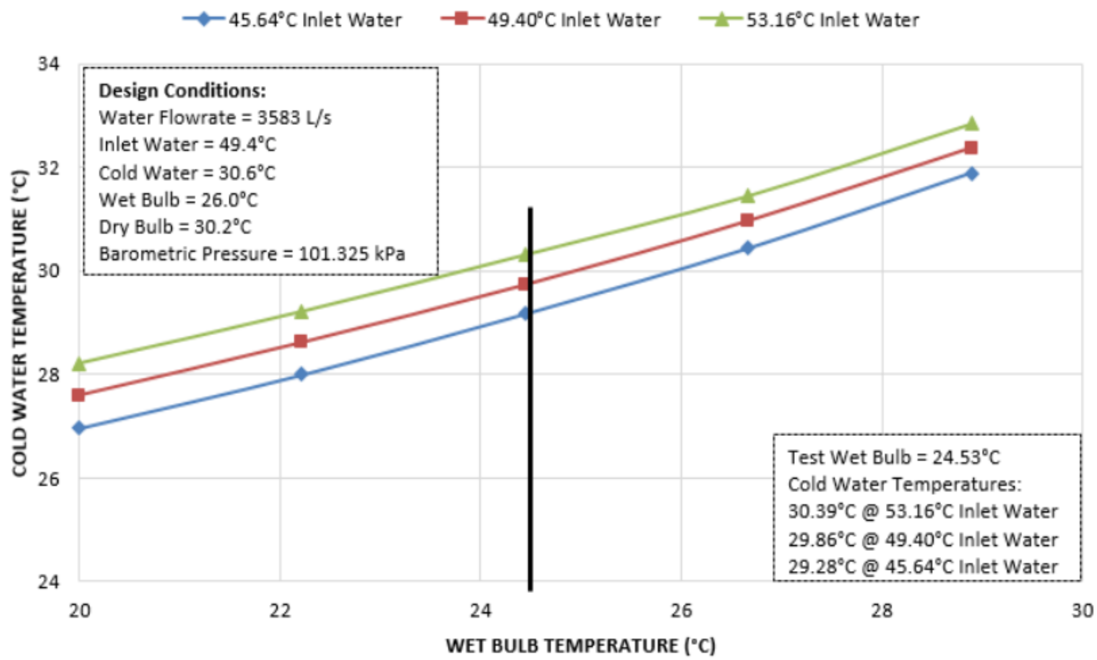
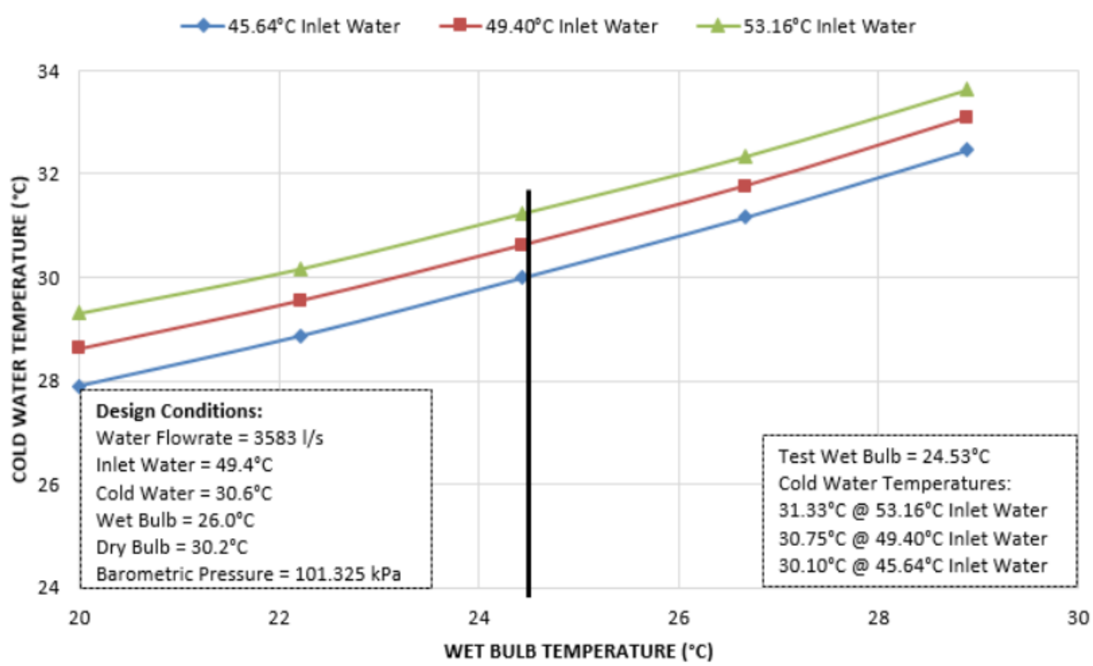


FIGURE Q-3: WATER FLOW RATE = 3941 L/S (110% OF DESIGN FLOW)



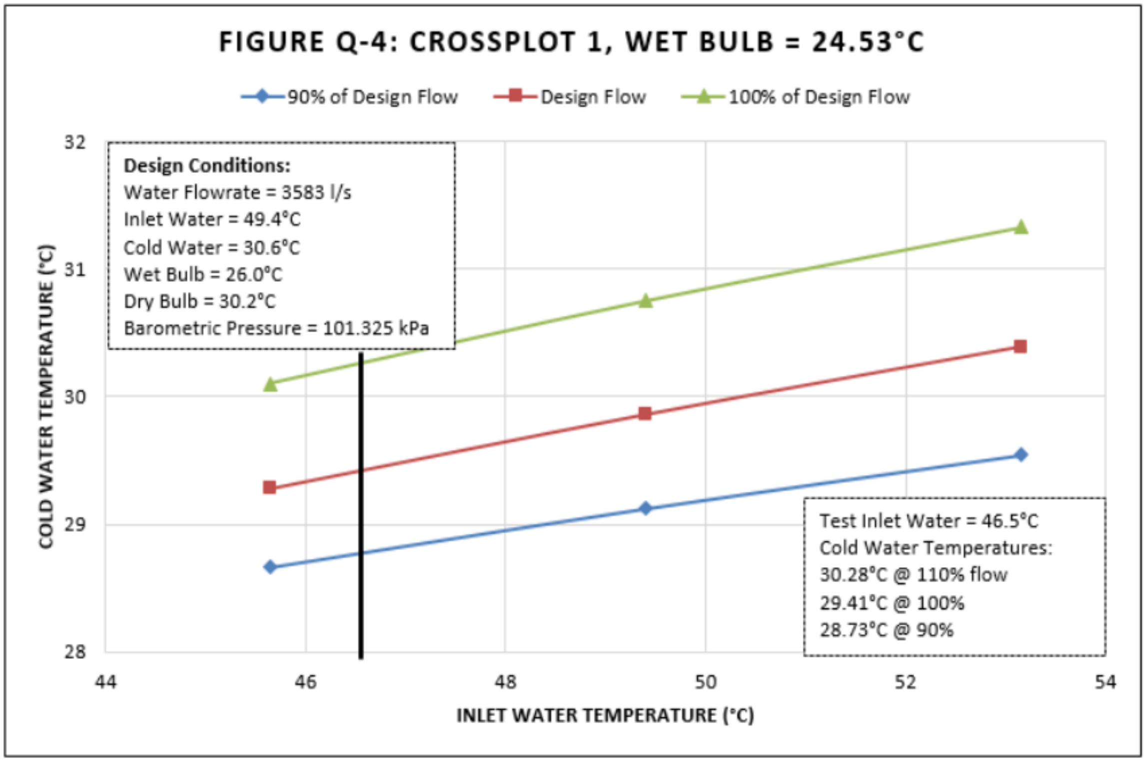


FIGURE Q-4: CROSSPLOT 1, WET BULB = 24.53°C
90% of Design Flow
Design Flow
100% of Design Flow
Design Conditions:
Water Flowrate = 3583 l/s
Inlet Water = 49.4°C
Cold Water = 30.6°C
Wet Bulb = 26.0°C
Dry Bulb = 30.2°C
Barometric Pressure = 101.325 kPa
Test Inlet Water = 46.5°C
Cold Water Temperatures:
30.28°C @ 110% flow
29.41°C @ 100%
28.73°C @ 90%
COLD WATER TEMPERATURE (°C)
INLET WATER TEMPERATURE (°C)
32
31
30
29
28
44
46
48
50
52
54

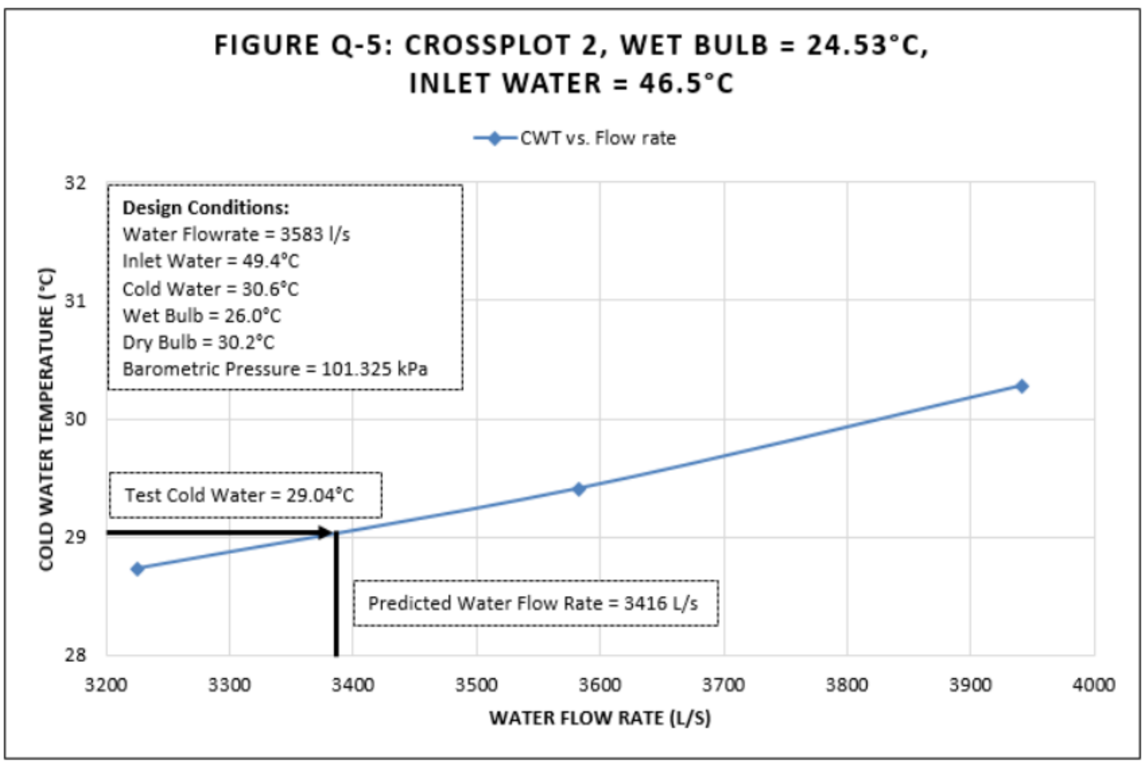


FIGURE Q-5: CROSSPLOT 2, WET BULB = 24.53°C,
INLET WATER = 46.5°C
CWT vs. Flow rate
Design Conditions:
Water Flowrate = 3583 l/s
Inlet Water = 49.4°C
Cold Water = 30.6°C
Wet Bulb = 26.0°C
Dry Bulb = 30.2°C
Barometric Pressure = 101.325 kPa
Test Cold Water = 29.04°C
Predicted Water Flow Rate = 3416 L/s
COLD WATER TEMPERATURE (°C)
WATER FLOW RATE (L/S)
32
31
30
29
28
3200
3300
3400
3500
3600
3700
3800
3900
4000

APPENDIX R

IP Evaluation of Mechanical-Draft Helper Cooling Tower Test Where Inlet (Hot) Water Temperature is Specified Using the Performance Curve Method

Section R1. General

The purpose of this appendix is to describe and illustrate the performance curve methodology for evaluating a thermal performance test on a mechanical draft helper cooling tower where inlet (hot) water temperature is specified, as described in Section 11 of the standard.

Section R2. Design and Test Conditions

Design and test conditions for a given mechanical draft helper cooling tower are summarized in the following table.

Table R-1 Cooling Tower Design and Test Data

Parameter	Design	Test
Water Flow Rate (Q_{wt})	56,792 gpm	57,426 gpm
Hot Water Temp (T_{hw})	120.92 °F	115.7 °F
Cold Water Temp (T_{cw})	87.08 °F	84.27 °F
Inlet Wet Bulb Temp (T_{wb})	78.80 °F	76.18 °F
Inlet Dry Bulb Temp (T_{db})	86.36 °F	77.94 °F
Fan Driver Power ($\dot{W}$)	143.40 bhp	151.50 bhp
Barometric Pressure (P_{bp})	29.921 in Hg	29.18 in Hg
Liquid to Gas Ratio	1.300	

In accordance with paragraph 11.2 of this standard, the manufacturer has submitted Performance Curves with Cold Water presented as a function of Wet Bulb Temperature with Inlet (Hot) Water Temperature and Water Flow Rate as parameters (see Figures R-1, R-2, and R-3).

Alternatively, thermal test data analyzed with constant range performance curves, as demonstrated in Appendix D, will result in the same analysis result (tower capacity).

Section R3. Evaluation Procedure

The steps for evaluating the test are as follows:

Step 1. Determine the Predicted Cold Water Temperatures

Scribe the submitted performance curves vertically at the test wet bulb temperature (76.18 °F) to determine the predicted Cold Water Temperature associated with the test Wet Bulb Temperature at each of the three Inlet (Hot) Water Temperatures and three Water Flow Rate conditions included on the performance curves. Values of these predicted Cold Water Temperatures are tabulated in Table R-2.

Table R-2 Curve Points for Cold Water Temperature vs Inlet (Hot) Water Temperature at 76.18°F Test Wet Bulb

Inlet (Hot) Water Temperature	90% Flow	100% Flow	110% Flow
114.15 °F	83.58	84.7	86.18
120.92 °F	84.42	85.75	87.35
127.69 °F	85.18	86.7	88.4

Step 2. Construct the First Cross Plot

The values of Cold Water Temperature are plotted against Inlet (Hot) Water Temperature using Flow Rate as a parameter (see Figure R-4).

Scribe Figure R-4 vertically at the test Inlet (Hot) Water Temperature (115.7 °F) to determine the predicted Cold Water Temperatures at each of the three Water Flow Rates. The values of these predicted Cold Water Temperatures are tabulated in Table R-3.

Table R-3 Curve Points for Cold Water Temperature vs Water Flow Rate at 76.18 °F Test Wet Bulb and 115.70 °F Test (Hot) Inlet Water Temperature

90% Flow	100% Flow	110% Flow
83.72	84.94	86.5

Step 3. Second Cross Plot

Using the values of Cold Water Temperature in Table R-3, plot these temperatures as a function of the three flow rate percentages (see Figure R-5).

Step 4. Determine the Predicted Flow Rate

On Figure R-5 scribe a horizontal line at the test Cold Water Temperature to intersect the curve. At the intersection, project a line vertically downward to find the predicted Water Flow Rate associated with the test Wet Bulb, Inlet (Hot) Water Temperature, and Cold Water Temperature.

$$Q_{w,pred} = 95.35\%\ Design\ Flow = 54{,}152\ gpm$$

Step 5. Calculating the Adjusted Test Water Flow Rate

The adjusted test Water Flow Rate is computed from equation 11.1, using the values for air density at the fan inlet and the fan driver output power, at both the test and design conditions.

To comply with this standard, the design and test values for density (ρ), specific volume (v) and enthalpy (h) of air must be determined using either CTI ToolKit or the psychrometric tables in ASHRAE Handbook – Fundamentals (IP). For this example, the CTI Toolkit was used to generate all psychrometric properties.

Since the evaluation is based upon the psychrometric properties of air at the fan inlet, different procedures must be employed for forced draft and induced draft towers.

a. Forced Draft Tower

For a forced draft tower, the fan inlet air conditions are the same as the tower inlet air conditions. Therefore the test density (ρ_t) and test specific volume (υ_t) are computed directly from the measured test values of wet-bulb, dry-bulb and barometric pressure. In accordance with paragraph 11.2, the manufacturer shall supply the design conditions at the tower's air inlet.

Table R-4 Design and Test Values for Air Characteristics

Parameter	**Design**	**Test**
Barometric Pressure (P_{bp})	29.921 in Hg	29.18 in Hg
Inlet Wet Bulb Temp (T_{wb})	78.80 °F	76.18 °F
Inlet Dry Bulb Temp (T_{db})	86.36 °F	77.94 °F
Humidity Ratio (HR)	0.01965 lbm/lbm	0.01971 lbm/lbm
Specific Volume (υ)	14.2009 ft^3/lbm	14.3382 ft^3/lbm
Inlet Enthalpy (h_i)	42.3329 Btu/lbm	40.2958 Btu/lbm
Density (ρ)	0.07180 lbm/ft^3	0.07112 lbm/ft^3
Relative Humidity (RH)	71.95%	92.44%

b. Induced Draft Tower

For the induced draft tower, the fan air conditions are the tower discharge conditions. The code requires that both the design and test discharge air properties be determined by a heat balance calculation. Calculating design discharge air properties is a straightforward procedure while calculating test discharge air properties requires combining the heat balance Equation (R-2) with Equation 5.1 and iterating for a solution.

The heat balance equation states that the heat gain by the air equals the heat loss of the water. The formula follows:

$$L * c_{pw} * (T_{hw} - T_{cw}) = G * (h_o - h_i) \qquad (R-1)$$

Where:

L = Water mass flow rate, lbm/hr

c_{pw} = Specific Heat of Water = 1.0 BTU/lbm °F

T_{hw} = Hot Water Temperature, °F

T_{cw} = Cold Water Temperature, °F

G = Mass flow rate of dry air through the tower, lbm/hr

h_o = Enthalpy of air leaving the tower, BTU/lbm

h_i = Enthalpy of air entering the tower, BTU/lbm

Rearranging to separate the exit air enthalpy, the heat balance equation becomes:

$$h_o = \left(\frac{L}{G}\right) * c_{pw} * (T_{hw} - T_{cw}) + h_i \qquad (R-2)$$

Determine the enthalpy of the entering air at design conditions as:

$$h_i = 42.333 \frac{BTU}{lbm}$$

Then, substituting all design values into Equation (R-2), calculate the design outlet enthalpy.

$$h_{od} = (1.300) * 1.0 * (120.92 - 87.08) + 42.333$$

$$h_{od} = 86.325 \frac{BTU}{lbm}$$

Assuming the discharge air is saturated at this enthalpy and barometric pressure, determine the design discharge air temperature (T_d), density (ρ_d) and specific volume (υ_d) as:

$$T_d = 107.323\,°F$$

$$\rho_d = 0.0678 \frac{lbm}{ft^3}$$

$$\upsilon_d = 15.550 \frac{ft^3}{lbm}$$

Next the discharge air characteristics at test conditions are calculated. First calculate the test L/G by substituting all known values into equation (5.1):

$$\frac{L}{G} = 1.3 * \left(\frac{57{,}426}{56{,}792}\right) * \left(\frac{143.4}{151.5}\right)^{\frac{1}{3}} * \left(\frac{\rho_t}{0.0678}\right)^{\frac{1}{3}} * \left(\frac{\upsilon_t}{15.55}\right)$$

$$\left(\frac{L}{G}\right) = 0.20355 * (\rho_t)^{1/3} * (\upsilon_t)$$

Substituting this L/G expression into equation (R-2) to calculate the test exit air enthalpy, h_{ot}

$$h_{ot} = 0.20355 * (\rho_t)^{\frac{1}{3}} * (\upsilon_t) * 1.0 * (115.70 - 84.27) + 40.2958$$

Simplifying,

$$h_{ot} = 6.3976 * (\rho_t)^{1/3} * (\upsilon_t) + 40.2958$$

At this point, guess at a discharge air temperature and, assuming saturation, determine density (ρ) and specific volume (υ) at that temperature. Then, substituting these values for density (ρ) and specific volume (υ) into the final heat balance expressions, calculate an enthalpy h_{ot}.

Compare the calculated value for enthalpy h_{ot} to the actual value of enthalpy at the assumed temperature and continue iterating discharge air temperature until a suitable temperature is selected for which the calculated value of enthalpy hot matches the actual value.

For the first estimate of leaving air temperature, use the average of T_{hw} and T_{cw} at test conditions. Typical iteration values for this example are given in Table R-5 for Barometric Pressure = 29.18 in Hg.

Table R-5 Iteration for Enthalpy of Leaving Air

T_{ao}	ρ_t	v_t	h_{ot} actual	h_{ot} computed	Error %
100	0.06738	15.5006	73.0653	80.6488	+10.38%
105.0	0.06650	15.8207	82.9340	81.3020	-1.97%
104.0	0.06668	15.7542	80.8517	81.1665	+0.389%
104.16	0.06665	15.7648	81.1811	81.1879	+0.008%

Step 6. Determine Cooling Tower Capability

Substituting the psychrometric values at 104.16 °F, the adjusted flow rate is now calculated from equation 11.1.

$$Q_{wt\ adj} = 57{,}426 * \left(\frac{143.4}{151.5}\right)^{\frac{1}{3}} * \left(\frac{0.06665}{0.0678}\right)^{1/3}$$
$$= 56{,}058\ gpm$$

The tower capacity is computed from equation 11.2.

$$C = \left(\frac{56{,}058}{54{,}152}\right) * 100 = 103.5\%$$

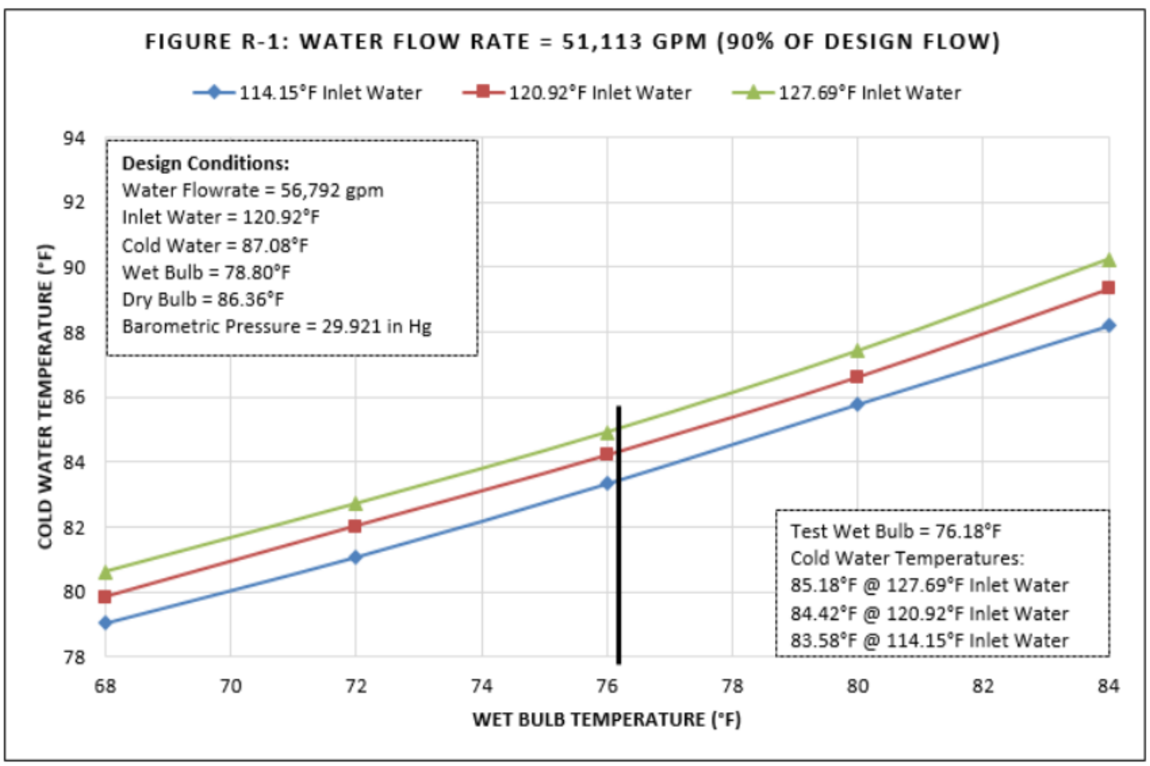


FIGURE R-1: WATER FLOW RATE = 51,113 GPM (90% OF DESIGN FLOW)

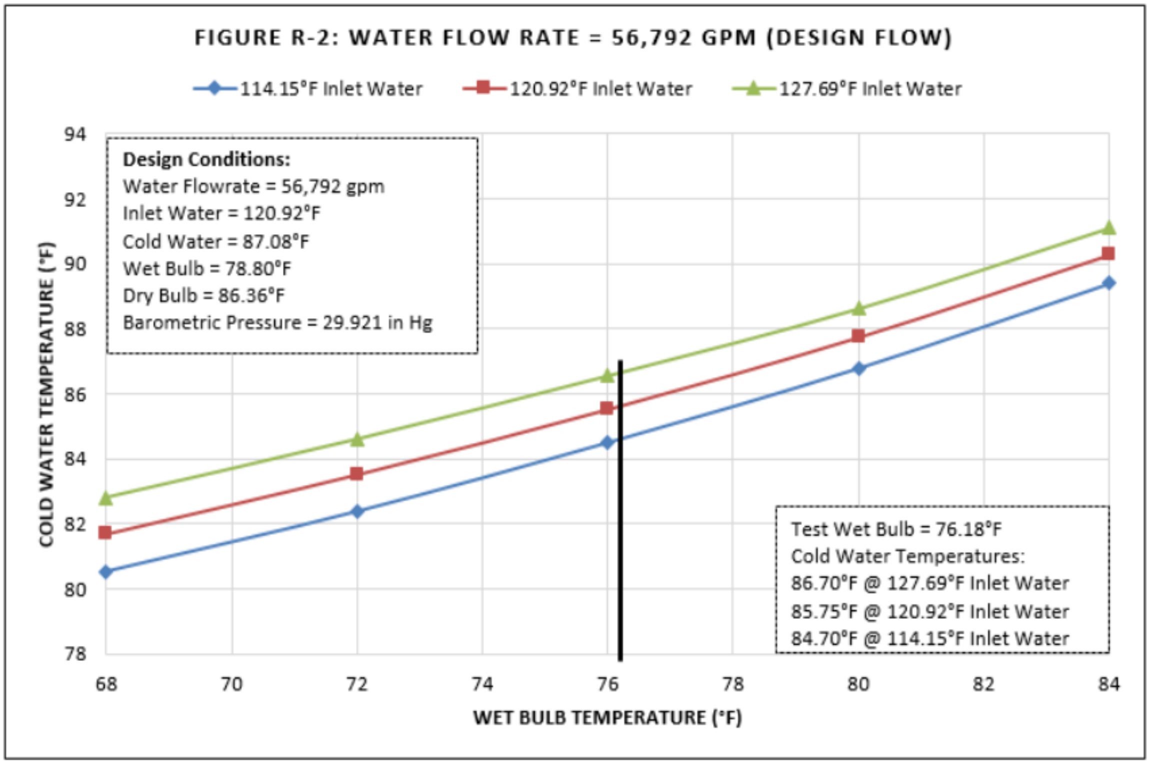


FIGURE R-2: WATER FLOW RATE = 56,792 GPM (DESIGN FLOW)
114.15°F Inlet Water
120.92°F Inlet Water
127.69°F Inlet Water
Design Conditions:
Water Flowrate = 56,792 gpm
Inlet Water = 120.92°F
Cold Water = 87.08°F
Wet Bulb = 78.80°F
Dry Bulb = 86.36°F
Barometric Pressure = 29.921 in Hg
Test Wet Bulb = 76.18°F
Cold Water Temperatures:
86.70°F @ 127.69°F Inlet Water
85.75°F @ 120.92°F Inlet Water
84.70°F @ 114.15°F Inlet Water
COLD WATER TEMPERATURE (°F)
94
92
90
88
86
84
82
80
78
68
70
72
74
76
78
80
82
84
WET BULB TEMPERATURE (°F)

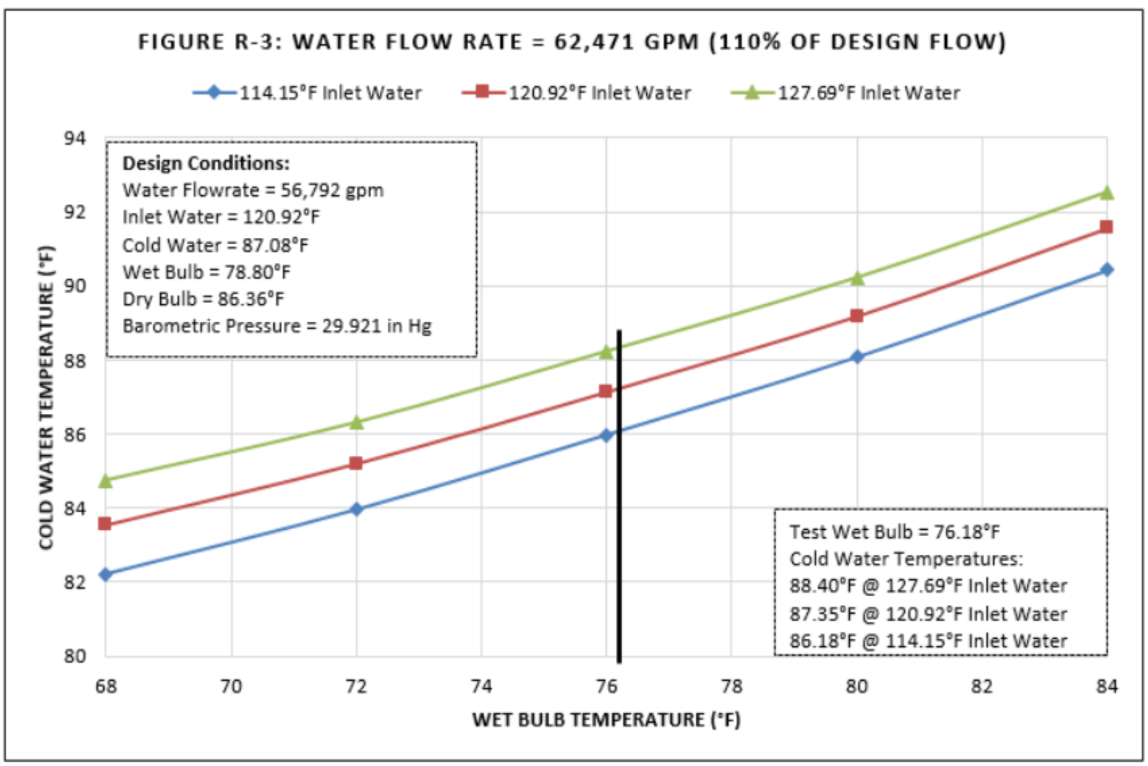


FIGURE R-3: WATER FLOW RATE = 62,471 GPM (110% OF DESIGN FLOW)
114.15°F Inlet Water
120.92°F Inlet Water
127.69°F Inlet Water
Design Conditions:
Water Flowrate = 56,792 gpm
Inlet Water = 120.92°F
Cold Water = 87.08°F
Wet Bulb = 78.80°F
Dry Bulb = 86.36°F
Barometric Pressure = 29.921 in Hg
Test Wet Bulb = 76.18°F
Cold Water Temperatures:
88.40°F @ 127.69°F Inlet Water
87.35°F @ 120.92°F Inlet Water
86.18°F @ 114.15°F Inlet Water
COLD WATER TEMPERATURE (°F)
94
92
90
88
86
84
82
80
68
70
72
74
76
78
80
82
84
WET BULB TEMPERATURE (°F)

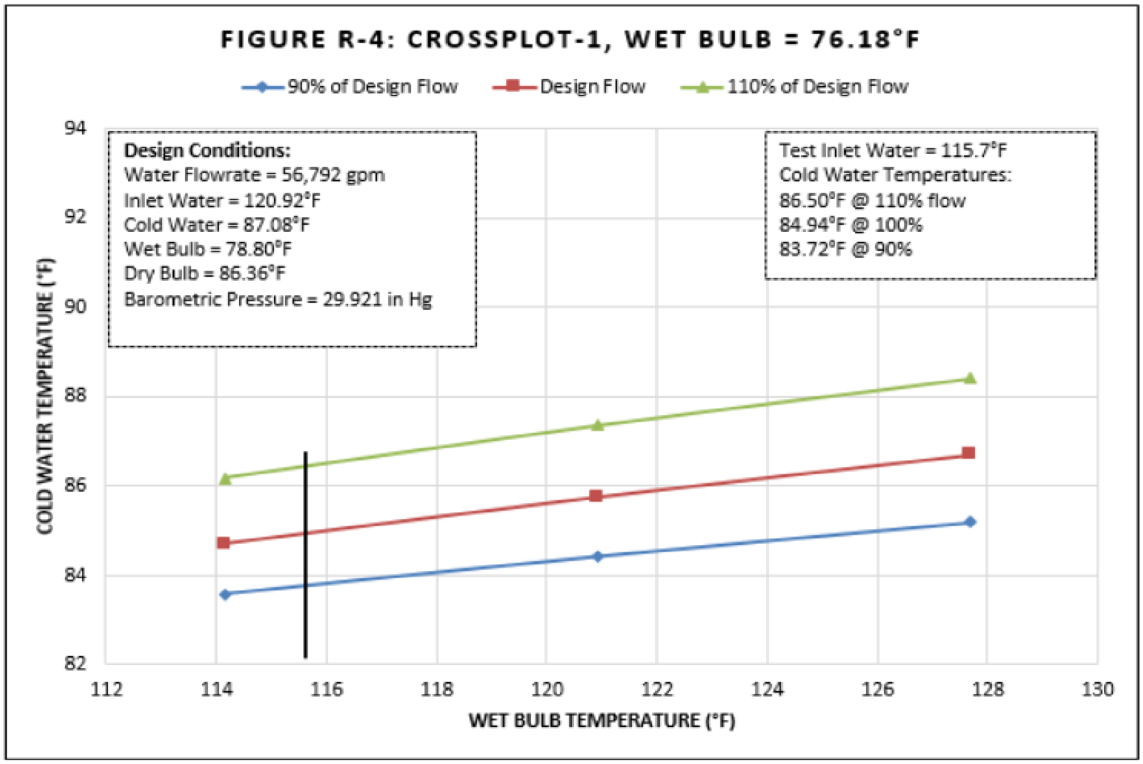


FIGURE R-4: CROSSPLOT-1, WET BULB = 76.18°F
90% of Design Flow
Design Flow
110% of Design Flow
Design Conditions:
Water Flowrate = 56,792 gpm
Inlet Water = 120.92°F
Cold Water = 87.08°F
Wet Bulb = 78.80°F
Dry Bulb = 86.36°F
Barometric Pressure = 29.921 in Hg
Test Inlet Water = 115.7°F
Cold Water Temperatures:
86.50°F @ 110% flow
84.94°F @ 100%
83.72°F @ 90%
COLD WATER TEMPERATURE (°F)
WET BULB TEMPERATURE (°F)
94
92
90
88
86
84
82
112
114
116
118
120
122
124
126
128
130

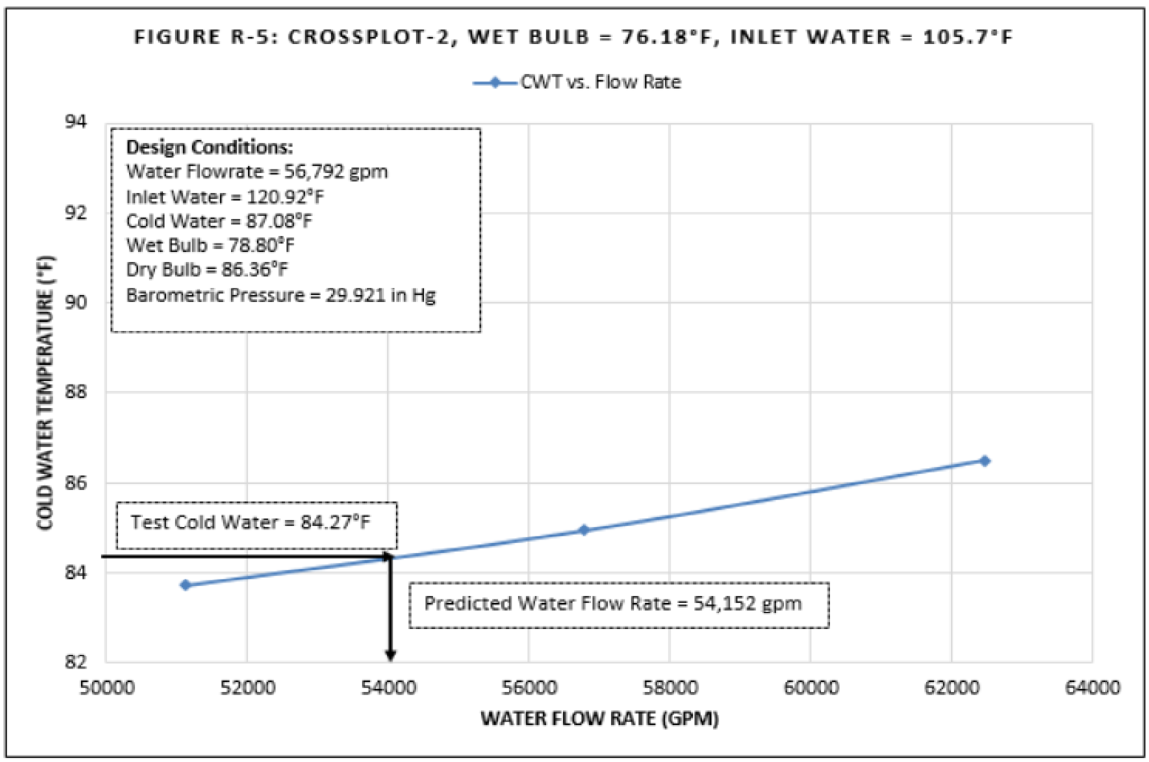


FIGURE R-5: CROSSPLOT-2, WET BULB = 76.18°F, INLET WATER = 105.7°F
CWT vs. Flow Rate
Design Conditions:
Water Flowrate = 56,792 gpm
Inlet Water = 120.92°F
Cold Water = 87.08°F
Wet Bulb = 78.80°F
Dry Bulb = 86.36°F
Barometric Pressure = 29.921 in Hg
Test Cold Water = 84.27°F
Predicted Water Flow Rate = 54,152 gpm
COLD WATER TEMPERATURE (°F)
WATER FLOW RATE (GPM)
94
92
90
88
86
84
82
50000
52000
54000
56000
58000
60000
62000
64000

APPENDIX U
UNCERTAINTY

U.1 Uncertainty Calculations: Scope and Purpose

Uncertainty is the estimate of the limits of error of a measurement. As used in this section, uncertainty refers to the limits of error of the measurements required to determine the cooling tower capability. The results will be summarized in terms of the uncertainty of the individual parameters measured to determine the cooling tower capability and the effect of these uncertainties on the capability. The uncertainty is an estimate of limits of error about the calculated capability. If the uncertainty intervals are equally distributed about the mean value, the probability of a high result is equal to that of a low result. Therefore, the best estimate of the cooling tower capability is the calculated capability.

The uncertainty of calculation methods utilized in this code or the uncertainty in the performance data provided by the manufacturer (performance or characteristic curves) are not addressed. The effects of parameters known to have an effect on the performance but not used in the capability calculation, such as wind speed and lapse rate, are also not addressed. The uncertainty calculated by the methods specified in this section should, therefore, be interpreted as an indication of the quality of the test data used in the performance calculation.

The purpose of a post-test uncertainty analysis is to determine the accuracy of the test result. It is sometimes used as a test tolerance, but this is a matter of contractual agreement between the parties to the test.

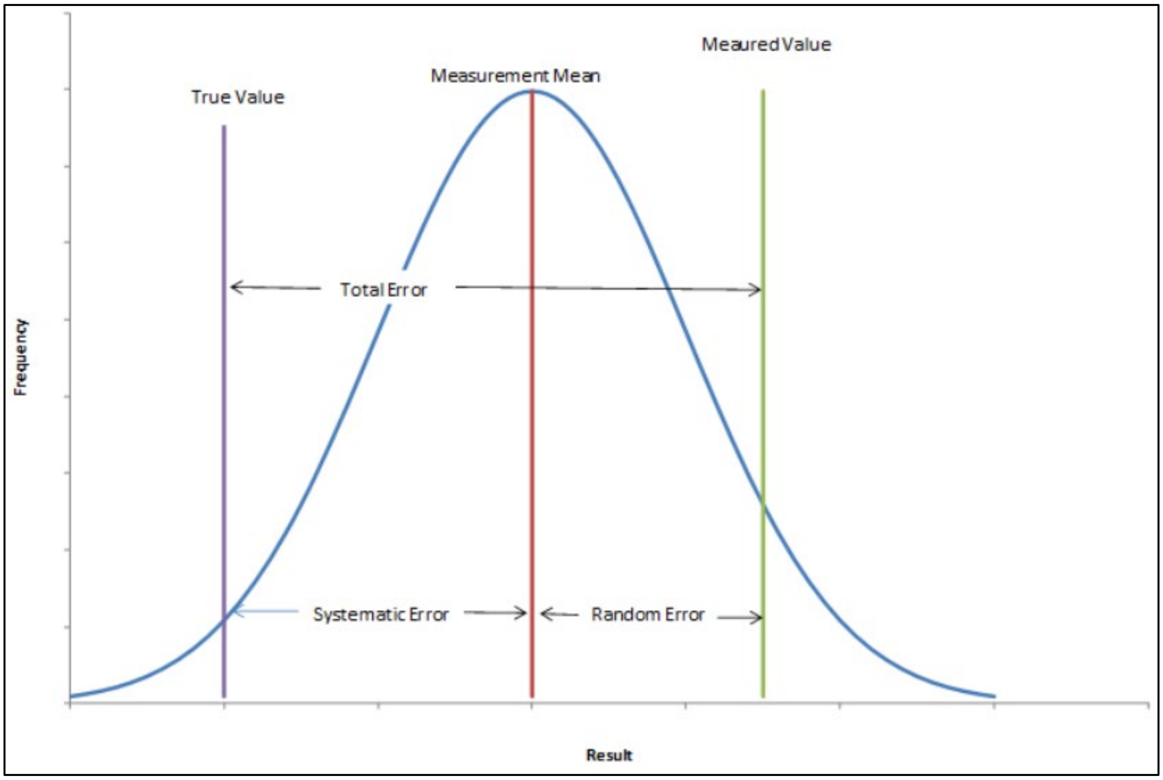
It is possible to do pretest uncertainty analysis. However, usually the most significant contributors to the overall uncertainty, the uncertainty due to the spatial variation in wet bulb temperature and the velocity distribution at the flow measurement location will not be known before the test. If pretest uncertainty analysis is performed conservative (high) values of the unknown parameters should be used for the unknown quantities. An accurate assessment of the test uncertainty (post-test uncertainty) requires data taken during the test.

U.2 Fundamental Concepts

Every measurement has error. This error results in a difference between the measured value and the true value. Since the true value is unknown the actual error cannot be known. Uncertainty estimates define the probable limits of error about test value. In order to define these limits, it is necessary to define an acceptable level of confidence for error estimate. This document uses a 95 percent confidence interval. This means that there is less than 5 percent probability that the true result will be outside the limits defined by the uncertainty interval.

As defined by ASME PTC 19.1 Test Uncertainty, the total error consists of random and systematic errors. Figure U-1 illustrates the relationship between the random and systematic errors.

Figure U-1 Random and Systematic Error



Random errors are sampling errors. The error in an average result will be reduced by repeated measurements. The random uncertainty methods described in Section U.4 provide an estimate of the random error in cooling tower capability. Systematic errors are bias errors. Averaging repeated measurements do not reduce bias errors since they are repeated with each measurement. Section U.5 provides procedures for calculating the systematic uncertainty which is an estimate of the limits of the systematic error. There are multiple sources of error in any measurement. Sensitivity factors are used to evaluate the effect of each error source on the result. Methods for evaluating the sensitivity factors are described in Section U.3.

The overall test uncertainty, in terms of capability, is calculated by

$$U_{cap} = \sqrt{B_{cap}^2 + S_{cap}^2}$$

Where:

U_{cap}= Overall test uncertainty in terms of capability

B_{cap} = The test systematic uncertainty in terms of capability

S_{cap} = The test random uncertainty in terms of capability

Table U-1 furnishes an example summary calculation of the overall uncertainty for a mechanical draft cooling tower.

Table U-1 Overall Uncertainty for Cooling Tower Capability

				Systematic Uncertainty				Random Uncertainty	
Parameter	Units	Sensitivity	Units	Instrumental	Spatial	Total	Capability	Total	Capability
Water Flow	gpm	7.15E-04	%/gpm	3346	2841	4389	3.14	NA	NA
Hot Water Temperature	°F	2.240	%/°F	0.17	NA	0.17	0.38	0.050	0.11
Cold Water Temperature	°F	-7.105	%/°F	0.17	0.1	0.20	1.40	0.026	0.19
Wet Bulb Temperature	°F	3.435	%/°F	0.34	0.74	0.82	2.81	0.022	0.08
Fan Motor Power	bhp	-0.172	%/hp	3.8	NA	3.78	0.65	NA	NA
Barometric Pressure	inHg	1.320	%/inHg	0.05	NA	0.05	0.07	NA	NA
Total Systematic Uncertainty		**4.50**	**%**						
Total Random Uncertainty		**0.23**	**%**						
Total Uncertainty		**4.51**	**%**						

The major parameters used to calculate cooling tower capability are:

- Water flow rate
- Hot water temperature
- Cold water temperature
- Wet bulb temperature
- Dry bulb temperature (natural draft and wet/dry cooling towers)
- Fan motor power (mechanical draft cooling towers)
- Barometric pressure

In order to evaluate the uncertainty of the test result, the uncertainty of the measurement of each of these parameters must be evaluated in terms of cooling tower capability.

For nearly every test there will be other measured parameters, such as pump discharge pressure, makeup water flow, makeup water temperature, blowdown flow and blowdown temperature, which have a minor impact on cooling tower capability. The effect of these parameters may normally be neglected without changing the value of the overall uncertainty. In unusual circumstances (for instance, if the cold water temperature measured at the pump discharge and the makeup temperature differ by 30°F), it may be worthwhile to include such parameters in the uncertainty analysis. An example of the calculation of the uncertainty of cold water temperature including the correction for makeup flow and temperature is included as Attachment U-1 of this appendix.

U.3 Sensitivity

When there is an analytical expression between the test result (R) and the test parameters (P1, P2,….Pn), the sensitivity factor may be determined by taking the partial derivative with respect to the test parameter. If the result is specified by an analytical expression,

$$R = f(P_1, P_2, P_3)$$

The sensitivity factors would be

$$\theta_{P_1}^R = \frac{\partial R}{\partial P_1}, \theta_{P_2}^R = \frac{\partial R}{\partial P_2}, \theta_{P_3}^R = \frac{\partial R}{\partial P_3}$$

The overall systematic uncertainty of the result would be

$$B_R = [(\theta_1 B_{P_1})^2 + (\theta_2 B_{P_2})^2 + (\theta_3 B_{P_3})^2 + 2\theta_1\theta_2 B_{P_1P_2} + 2\theta_1\theta_3 B_{P_1P_3} + 2\theta_2\theta_3 B_{P_2P_3}]^{1/2}$$

Where

B_{P_1} = The systematic uncertainty for parameter P_1

B_{P_2} = The systematic uncertainty for parameter P_2

B_{P_3} = The systematic uncertainty for parameter P_3

$B_{P_1P_2}$ = The covariance term which are products of perfectly correlated uncertainty for parameters P_1 and P_2

$B_{P_1P_3}$ = The covariance term which are products of perfectly correlated uncertainty for parameters P_1 and P_3

$B_{P_2P_3}$ = The covariance term which are products of perfectly correlated uncertainty for parameters P_2 and P_3

The subject of correlated uncertainty is more fully addressed in Appendix UC. For parameters for which the sources of error are independent, the total systematic uncertainty would be calculated by

$$B_R = \sqrt{\left(\theta_{P_1}^R B_{P_1}\right)^2 + \left(\theta_{P_2}^R B_{P_2}\right)^2 + \left(\theta_{P_3}^R B_{P_3}\right)^2}$$

For example, if the total water flow to the cooling tower were measured in three risers, the total circulating water flow to the cooling tower would be

$$q_{cw} = q_{r1} + q_{r2} + q_{r3}$$

Where q_{r1}, q_{r2}, and q_{r3} are the measured flow rates for each riser.

The sensitivity coefficients for the flow rates as measured in the risers would be

$\theta_{q_{r1}}^{q_{cw}} = \frac{\partial q_{cw}}{\partial q_{r1}} = 1,$

$\theta_{q_{r2}}^{q_{cw}} = \frac{\partial q_{cw}}{\partial q_{r2}} = 1,$ and

$\theta_{q_{r3}}^{q_{cw}} = \frac{\partial q_{cw}}{\partial q_{r3}} = 1.$

When there is no analytical expression for calculating the result from the test parameters the sensitivity factors may be determined numerically, by incremental variation of the test parameters. The sensitivity factor for parameter P_x may be approximated by

$$\theta_{P_x}^R = \frac{\Delta R}{\Delta P_x} = \frac{R\{X + \Delta x\} - R\{X - \Delta x\}}{2\Delta x}$$

Where

X = Base value of parameter P_x

Δx = Increment of parameter P_x

$R\{X + \Delta x\}$ = Value of the result with parameter P_x set to $X + \Delta x$ and all other parameters set to their base values

$R\{X - \Delta x\}$= Value of the result with parameter P_x set to $X - \Delta x$ and all other parameters set to their base values.

The value of Δx selected should be the minimum value required to yield sufficient variation to avoid rounding errors.

This approach is used when calculating the effect of the each of the test parameters on the cooling tower capability. The sensitivity factors are calculated using the same analysis tool used to evaluate the test data. Table U.2 summarizes the sensitivity factor calculations for a cooling tower test.

Table U.2 Sensitivity of Capability to Test Parameters

				Incremented Value		Capability			
Parameter	Units	Average	Increment	+	-	+	-	Sensitivity	Units
Water Flow	gpm	135405	1000	136405	134405	99.68	98.25	7.155E-04	%/gpm
Hot Water Temperature	°F	96.68	1	97.68	95.68	101.24	96.76	2.240	%/°F
Cold Water Temperature	°F	72.67	1	73.67	71.67	92.72	106.93	-7.105	%/°F
Wet Bulb Temperature	°F	52.90	1	53.90	51.90	101.59	94.72	3.435	%/°F
Fan Motor Power	bhp	189.22	10	199.22	179.22	97.31	100.74	-0.172	%/hp
Barometric Pressure	inHg	25.33	1	26.33	24.33	100.27	97.63	1.320	%/inHg

The performance curves and the test data were input into the CTI Toolkit software to determine the base case. The incremental values of the test parameters were determined by adding and subtracting the value increment from the base value for the parameter. The resulting value for the test parameter was then input into the Toolkit software which determined the capability. Using hot water temperature as an example,

$$T_{hw+\Delta x} = 96.68 + 1.00 = 97.68$$

The capability of the cooling tower, as determined by the Toolkit software, using 97.68 for the hot water temperature and all of the other parameters set to their base values was 101.24 percent.

$$T_{hw-\Delta x} = 96.68 - 1.00 = 95.68$$

The capability of the cooling tower, as determined by the Toolkit software, using 95.68 for the hot water temperature and all of the other parameters set to their base values was 96.76 percent. The sensitivity of cooling tower capability to hot water temperature is, therefore

$$\theta_{T_{hw}}^{cap} = \frac{101.24 - 96.76}{2(1.00)} = 2.24\frac{\%}{°\mathrm{F}}$$

This process is repeated for each of the test parameters. Sensitivity factors are a function of the cooling tower design and, to a lesser extent, the test conditions. Therefore, the sensitivity factors must be evaluated for each test.

U.4 Random Uncertainty

U.4.1 Random Uncertainty Based on Variation of Test Results – Natural Draft Cooling Towers

Random uncertainty is evaluated based on the standard deviation of the test results or by the evaluation of random uncertainty of individual test parameters. When multiple test windows, at least four, are analyzed to produce an average capability for the cooling tower, the best estimate of the random uncertainty is the standard deviation of the test results. Since the analysis of at least six test windows is required by this test code for natural draft cooling towers, The recommended procedure for calculating the precision uncertainty is to compute the standard deviation of the test results of all of the test windows used to calculate the average capability. The precision uncertainty of the average capability when multiple hours of data are analyzed is

$$S_{\underline{cap}} = \frac{t_{95\%,n-1} s_{cap}}{\sqrt{n}}$$

Where

$S_{\overline{cap}}$ = The precision uncertainty of the average capability (as determined by the average of the capabilities for the individual test windows) for the cooling tower

s_{cap} = The standard deviation of the capability values for the test windows

n = Number of test windows used to calculate the average capability

$t_{95\%,n-1}$ = Student's t value for a 95% confidence interval and $n-1$ degrees of freedom.

The standard deviation of the capability for the multiple test windows is

$$s_{cap} = \sqrt{\frac{\sum_{i=1,n}(cap_i - \overline{cap})^2}{n-1}}$$

Where

c_{api} = Capability of an individual test window

$\overline{cap}$= The average cooling tower capability as calculated from all of the analyzed test windows.

U4.1.2 Random Uncertainty Based on Variation in Test Parameters-Mechanical Draft Cooling Towers

For mechanical draft cooling towers, the standard practice is to analyze a single test window. When a single test window is used to calculate the capability, the precision uncertainty of the capability is calculated from the precision uncertainty of the test parameters. In terms of the standard deviation of the test parameters, the precision uncertainty of the test is calculated by

$$S_{cap} = \sqrt{\sum_{i=1,6} \left(\theta_i^{cap} \bar{S_i}\right)^2}$$

Where

θ_i^{cap} = Sensitivity of capability to the individual test parameters

$\bar{S_i}$ = The precision uncertainty of test parameters, wet bulb temperature, hot water temperature, cold water temperature, etc.

For parameters acquired by an automated data acquisition system the standard deviation for the test parameter is calculated by

$$\bar{S_i} = \frac{2\sqrt{\sum_{i=1,m} s_i}}{m\sqrt{n}}$$

Where

s_i = Standard deviation of the sensor reading over the test period

n = Number of sensor readings

m = Number of sensors.

The standard deviation of the sensor reading, s_i, is based on the time variation of the sensor reading over the test period.

Water flow and fan motor power are derived quantities considered to be sufficiently stable to be measured once per test. The precision uncertainty of these parameters can be considered negligible.

U.5 Systematic Uncertainty

Systematic errors are caused by bias errors in the instruments used to make measurements and by the errors caused by the calculation of an average value of spatially varying parameter from measurements taken at discrete points. Instrumental uncertainty for cooling towers includes estimates of the limits of error for calibration of temperature sensors, the calibration uncertainty of the pitot tube used to measure water flow, the accuracy limits of data loggers, and the effects of solar radiation on wet bulb and dry bulb temperature measurements made with psychrometers.

Instrumental errors can be expected to be duplicated in multiple measurements with the same instrument. When the same power meter is used to measure the fan motor power, the same errors would be expected to occur when measuring each fan motor power. When the same pitot tube is used to measure the flow in multiple risers, any error in the calibration of the pitot can be expected to be repeated in each flow measurement. When different instruments of the same type are used to measure the same parameter, many of the errors arise from the same source. For instance, all of the temperature sensors measuring cold water temperature would probably be calibrated in the same temperature bath using the same calibration standard. Very likely the sensor outputs would be measured by the same data logger. The recommended simple, conservative (leading to higher value of uncertainty) approach is to assume that these errors are totally correlated for measurements of the same parameter and that the errors are totally uncorrelated for measurements of different parameters. Totally correlated uncertainties are estimated by summing the uncertainties due to each error source.

$$B_{cor}^{P_x} = \sum_{i=1,n} \sqrt{(\theta_i^{Px} B_i)^2}$$

or, if all sensitivities are positive,

$$\mathrm{B_{cor}^{P_x}} = \sum_{\mathrm{i=1,n}} \theta_{\mathrm{i}}^{\mathrm{Px}} \mathrm{B_i}$$

Where

$B_{cor}^{P_x}$ = Correlated uncertainty for parameter P_x

θ_i^{Px}= Sensitivity factor relating error source i to parameter P_x

B_i = Systematic uncertainty for error source i.

The calculation of the systematic uncertainty of the average fan motor power can be used as an example of correlated uncertainty. The average fan motor power for a four cell mechanical draft cooling tower is calculated by

$$\bar{W} = \frac{\sum_{i=1,4} W_i}{4}$$

The sensitivity factor relating the average fan motor power to the measured power for an individual power meter is

$$\theta_{W_i}^{\bar{W}} = \frac{\partial \bar{W}}{\partial W_i} = \frac{1}{4}$$

If the accuracy of the power meter used to measure fan motor power is 2 percent of reading, then the uncertainty of a single reading is

$$B_{W_i} = 0.02 W_i$$

The uncertainty of the fan motor power is, therefore,

$$B_{\bar{W}} = \sum_{i=1,4} \frac{0.02 W_i}{4} = .02\bar{W}$$

This result can be generalized: the instrumental uncertainty for the fan motor power is equal to the instrumental uncertainty of a single power measurement if the same power meter is used for each measurement. This is also true for parameters where the average value of the parameter is determined based on the readings of multiple instruments of the same type (wet bulb temperature, dry bulb temperature, hot water temperature, cold water temperature). For these parameters,

$$B_{\overline{P_x}} = B_{P_{x,i}}$$

Where

$B_{\overline{P_x}}$ = The instrumental uncertainty of the average value of parameter, P_x

$B_{P_{x,i}}$ = The instrumental uncertainty for a single device measuring parameter P_x

For water flow rate determined by the sum of flow measured in multiple locations (e.g. risers), the instrumental uncertainty for the total water flow is

$$B_{WF} = \sum_{i=1,m} \theta_{wf,riser\ i}^{WF} B_{wf,riser\ i} = \sum_{i=1,m} (1) B_{wf,riser\ i}$$

Where

B_{WF} = Instrumental uncertainty for total water flow

$B_{wf,riser\ i}$ = Instrumental uncertainty for water flow in riser i.

When the output of two are more types of instruments are used in the measurement of a test parameter, it may be assumed that errors are uncorrelated. The uncertainty for uncorrelated sources of error is combined by

$$B_{uncor}^{P_x} = \sqrt{\sum_{i=1,n} (\theta_i^{Px} B_i)^2}$$

Where

$B_{uncor}^{P_x}$ = Uncorrelated systematic uncertainty for parameter P_x

Since many cooling tower test parameters involve the measurement of temperature, it is convenient to calculate the instrumental uncertainty for temperature measurements. Temperature measurements are made using calibrated sensors measured with a data logger. The significant error sources for this system are the calibration tolerance of the four-wire RTDs and the measurement tolerance of the data logger. These error sources are completely unrelated, so the uncertainties of these elements can be treated as uncorrelated. Information about the sensor calibration and data logger for an example temperature measurement system are:

- Four wire RTD calibration tolerance 0.1°F (.06°C)
- Sensitivity of resistance to change in temperature 0.214 Ω/°F (0.385 Ω/°C)
- Data logger accuracy 0.03 Ω

For the example temperature measurement system

$$\theta_{RTD}^{T} = 1 \frac{°\mathrm{F}}{°\mathrm{F}}$$

$$B_{RTD} = 0.1°\mathrm{F}$$

$$\theta_{DAS}^{T} = \frac{1}{0.214} = 4.67 \frac{°\mathrm{F}}{\Omega}$$

$$B_{DAS} = 0.03\Omega$$

$$\theta_{RTD}^{T} = 1 \frac{°\mathrm{C}}{°\mathrm{C}}$$

$$B_{RTD} = 0.06°\mathrm{C}$$

$$\theta_{DAS}^{T} = \frac{1}{0.385} = 2.60 \frac{°\mathrm{C}}{\Omega}$$

$$B_{DAS} = 0.03\Omega$$

The temperature measurement system uncertainty is

$$B_T = \sqrt{(1 \cdot 0.1)^2 + (4.67 \cdot 0.03)^2} = 0.17°\mathrm{F}$$

$$B_T = \sqrt{(1 \cdot 0.06)^2 + (2.60 \cdot 0.03)^2} = 0.10°\mathrm{C}$$

Wet bulb and dry bulb temperature measurements are made with mechanically aspirated psychrometers are subject to errors due to solar radiation, local variations in air velocity, wicking rate, etc. Since these errors are affected by local conditions, they are difficult to quantify, but, based on industry experience, it is recommended 0.3°F (0.17°C) be used for the instrumental uncertainty for wet bulb temperature. The recommended instrumental uncertainty for dry bulb temperature is 0.5°F (0.28°C).

U.6 Spatial Uncertainty

Wet bulb (and dry bulb) temperature for cooling tower tests are measured by arrays of psychrometers placed around the air inlet of the cooling tower. Water flow rate is measured by a pitot traverse where velocity is measured at multiple points across two diameters of a pipe at the measurement plane. Cold water temperature is sometimes determined by measuring the cold water temperature at an array of temperature sensors placed in the cold water discharge(s). In all of these cases, an average value of the parameter is determined from measurements conducted at a limited number of points at the measurement plane. The difference between the true average of the parameter across the measurement plane and the average of the point measurements of the parameter constitutes the spatial bias error. The spatial systematic uncertainty is an estimate of this error based on the variation of the point wise measurements by

$$B_{spatial,z} = \frac{t_{95\%,m-1} s_{spacial,z}}{\sqrt{m}}$$

Where

$B_{spatial,z}$ = Spatial systematic uncertainty for measurement plane z

m = Number of measurements at measurement plane z

$t_{95,m-1}$ = Student's t value for m-1 degrees of freedom

$s_{spacial,z}$ = Variance of parameter at plane z.

The variance of the parameter at the measurement plane is calculated by

$$s_{spacial,z} = \sqrt{\frac{\sum_{i=1,m}\left(\bar{X}_i - \bar{\bar{X}}\right)^2}{m-1}}$$

Where

$\bar{X}_i$ = Time averaged value of parameter at position i

$\bar{\bar{X}}$ = Global average of parameter at plane z.

The method described is based on the assumption that an adequate number of measurements is made to characterize the spatial variation of the parameter. This is not the case when the number of measurement locations is fewer than four.

The calculation of the spatial uncertainty for a cooling tower with closed inlets is illustrated in Figure U.2

Figure U.2 Spatial Uncertainty for a Counter Flow Cooling Tower

East Air Inlet of Cooling Tower				
25.49	26.17	25.62	26.21	
24.87	25.75	25.83	25.28	
Average	25.65	°C	Count	8
Variance	0.45	°C	Student's t	2.36
Spatial Uncertainty	0.37	°C		
West Air Inlet of Cooling Tower				
22.55	22.61	22.53	22.96	
22.54	23.01	22.28	22.81	
Average	22.66	°C	Count	8
Variance	0.25	°C	Student's t	2.36
Spatial Uncertainty	0.21	°C		
Total Uncertainty	**0.21**	**°C**		

The sensor values in Figure U2, represent the time average of the sensor readings. Since the measurement planes are separated by the cooling tower end walls, the spatial systematic uncertainty is calculated separately for the north side and for the south side of the cooling tower. For the south side of the cooling tower, the spatial systematic uncertainty for wet bulb temperature is

$$B_{spacial,TWBsouth} = \frac{0.45 * 2.36}{\sqrt{8}} = 0.37℃$$

For the north side of the cooling tower, the spatial systematic uncertainty for wet bulb temperature is

$$B_{spacial,TWBnorth} = \frac{0.25 * 2.36}{\sqrt{8}} = 0.21℃$$

The average inlet wet bulb temperature is mathematically equivalent to

$$\bar{T}_{wb} = \frac{\bar{T}_{wb,south} + \bar{T}_{wb,north}}{2}$$

Therefore, the sensitivity of the overall average wet bulb temperature to the average wet bulb temperature of each side is

$$\theta_{\bar{T}_{wb,south}}^{\bar{T}_{wb}} = \frac{\delta\bar{T}_{wb}}{\delta\bar{T}_{wb,south}} = \frac{1}{2} \text{ and}$$

$$\theta_{\bar{T}_{wb,north}}^{\bar{T}_{wb}} = \frac{\delta\bar{T}_{wb}}{\delta\bar{T}_{wb,north}} = \frac{1}{2}$$

The spatial uncertainty for wet bulb temperature is

$$B_{spatial,\bar{T}_{wb}} = \sqrt{(0.5 * 0.37)^2 + (0.5 * 0.21)^2} = 0.21℃$$

For rectangular mechanical draft cooling towers with open end walls, circular mechanical draft cooling towers, and natural draft cooling towers, there is one contiguous inlet area. For these types of cooling towers, the entire air inlet is considered one area and the spatial uncertainty is calculated based on the entire array of wet bulb temperature measurements.

When cold water temperature is measured in an open channel by an array of temperature sensors, the spatial uncertainty is calculated by the method illustrated in the previous example. When, as is usually the case, the cold water temperature is measured in a flowing (or thermal) well at the discharge of the pumps, the accuracy of the measurement is increased because of the high degree of mixing imparted by the pump.

The mixing is unlikely to be perfect, however, so it is suggested an estimated spatial uncertainty of 0.1°F (0.05°C) be used for cold water temperature measured at the discharge of the pumps.

Unless there is an unmeasured difference in water flow between cold water pumps in multiple pump systems, the variation in temperature between multiple pumps causes no error in the average cold water temperature. Therefore, no spatial uncertainty is calculated based on the variations of cold water temperatures measured at the discharge of multiple pump systems.

The cooling tower is usually far from the heat source so that spatial variation in the hot water temperature can be considered negligible.

U.7 Water Flow

The uncertainty in the water flow is, usually, the dominant factor in the uncertainty in the capability of a cooling tower test. The uncertainty calculations for this parameter have special aspects which justify detailed discussion. When, as is usually the case, the water flow to the cooling tower is determined by a pitot traverse of the inlet water pipe(s), the circulating water flow is calculated by

$$q = A_t \bar{V}$$

Where

q = Circulating water flow at measurement plane

A_t = Area of traverse plane

$\bar{V}$ = Average velocity in traverse plane = $\frac{\sum_{i=1,m} V_i}{m}$

V_i = Velocity at measurement location i

m = Number of measurement locations

U.7-1 Determination of Pipe Area Uncertainty

The diameter of the pipe at the traverse plane is usually determined by measuring the diameter of the pipe at each of the pitot taps. The area of the pipe is calculated by

$$A_t = \frac{\pi}{4} d_t^2$$

The diameter of the pipe at the traverse location is normally determined by contacting the pipe wall with the pitot tip, marking this position on the pitot tube, and incrementally withdrawing the pitot tube until the differential pressure between the total and the static ports is zero. The difference between the fully extended location and the null differential location plus the offset between the total port of the pitot tube determines the diameter of the pipe at the traverse location. Possible sources of error in this determination include

- Misalignment of the pitot tap to the diameter of the pipe
- Misalignment of the pitot tap such that it is not perpendicular to the pipe in the flow direction
- Resolution of distance measurement
- Flow in the pipe tap which would result in a measured differential pressure while the total port is outside the pipe wall
- Pipe at the traverse section is elliptical rather than round
- Welding material inside the pipe wall at the pitot tap.

Many of these errors are difficult to estimate because they depend on the quality of the pitot tap installation at the specific traverse site. In the absence of data quantifying these errors, the following table is offered as a conventional basis for estimating uncertainty in the pipe diameter.

Table U.3 Traverse Diameter Uncertainty

Pipe Diameter		Diameter Systematic Uncertainty	
inches	meters	inches	meters
<10	<0.25	0.0625	0.0016
10-20	0.25-0.51	0.125	0.0032
20-36	0.51-0.91	0.25	0.0064
36-72	0.91-1.83	0.375	0.0095
>72	>1.83	0.5	0.0127

In order to avoid the errors described above, the nominal pipe inside diameter is sometimes used to determine the area of the pipe. In that case, the pipe specifications used for the purchase of the pipe (possibly by reference to ASTM or other standards) usually include tolerances of the outside diameter and wall thickness. These tolerances should be used to determine the uncertainty in the inside diameter of the pipe at the traverse location.

The uncertainty in flow due to pipe diameter is

$$B_{d_t}^q = \theta_{d_t}^q B_{d_t}$$

The sensitivity factor for flow rate to pipe diameter is

$$\theta_{d_t}^q = \frac{\partial q}{\partial d_t} = 2 C_0 C_1 \bar{V} d_t$$

Where

C_o = Units conversion factor

$$= 7.4805 \frac{gal}{ft^3} * 60 \frac{sec}{min} = 448.83 \frac{gpm}{ft^3/sec}$$
$$= 1000 \frac{L}{m^3}$$

C_1= Units conversion factor

$$= \frac{\pi\, ft^2}{4 \cdot 144 in^2} = 0.005454 \frac{ft^2}{in^2}$$
$$= \frac{\pi}{4} = 0.7854$$

For a pipe with an inside diameter of 53 inches and velocity of 9.0 ft/s with the diameter at the traverse plane being determined using the pitot tube, the uncertainty in water flow due to measurement of the pipe diameter would be

$$B_{d_t}^q = 2 \cdot 448.83 \frac{gpm}{ft^3/sec} * 0.005454 \frac{ft^2}{in^2} * 9 \frac{ft}{s} * 53\, in * 0.375\, in = 876\, gpm$$

U.7-2 Pitot Tube Calibration Uncertainty

The velocity at each measurement point is determined by

$$V_i = C_c C_{pitot} \sqrt{\Delta P_{v,i}}$$

and

$$\bar{V} = C_c C_{pitot} \frac{\sum_{i=1,m} \sqrt{\Delta P_{v,i}}}{m} = C_c C_{pitot} \overline{\Delta P_v^{1/2}}$$

Where

C_c= Units conversion factor

$$\sqrt{\frac{2 * 32.17}{12} \frac{ft}{s * inwg^{1/2}}}$$

C_{pitot} = Calibration coefficient for pitot tube

$\Delta P_{v,i}$ = Differential pressure between total and static ports of pitot tube

$$\overline{\Delta P_v^{1/2}} = \frac{\sum_{i=1,m} \sqrt{\Delta P_{v,i}}}{m}$$

The pitot calibration coefficient is typically determined by traversing the pipe at the calibration facility with the pitot tube under calibration while measuring the pipe volumetric flow with a traceable volumetric (or mass) standard. The uncertainty of the pitot tube coefficient is determined from the uncertainty of the volumetric calibration standard used by the laboratory, the spatial variation of the velocity at the measurement point for the calibration, the area uncertainty at the traverse location, and degree of fit for the pitot calibration versus Reynolds number. Current calibration practices support a coefficient uncertainty of approximately 2 percent.

In terms of water flow rate, the systematic uncertainty for the pitot coefficient is:

$$B_{C_{pitot}}^q = \theta_V^q B_{C_{pitot}}^V = C_0 A_t B_{C_{pitot}}^V$$

U.7-3 Differential Pressure Measurement Uncertainty

The differential pressure at each measurement location may be measured using a differential pressure transmitter. In that rather unusual case, the uncertainty in the differential pressure should be determined from the calibration tolerance of the pressure transmitter. More typically the differential pressure is measured using an air-over-water manometer with a rule or tape measurer to measure the difference in the water levels in the legs of the manometer. Unless the velocity in the pipe is below 4 ft/sec (1.2 m/s), resulting in a differential pressure reading of approximately 5 inches (0.13 m), the resolution of the rule or tape measure will be an insignificant source of error. Another source of error in the differential pressure will result from the time variation of the differential level between the legs of the manometer. For an “unbiased” (literally, one whose errors are random) observer this source of error is usually insignificant, as illustrated by the following example.

The sensitivity of average velocity to the differential pressure at point i is

$$\theta_{\Delta P_{v,i}}^{\bar{V}} = \frac{C_C C_{pitot}}{2m\sqrt{\Delta P_{v,i}}}$$

Experience has shown that human ability to “eyeball average” this variation is very good. A very conservative (high) estimate of the uncertainty of the differential level in the manometer is ¼ of the maximum variation in the differential level. For a time variation 10 percent of the value of the time averaged velocity pressure at a point i, the systematic uncertainty of velocity pressure would be,

$$B_{\Delta P_{v,i}} = 0.1 \cdot 0.25 \cdot \Delta P_{v,i}$$

The uncertainty of the average velocity is

$$B_{\Delta P_v}^{\bar{V}} = \sqrt{\sum_{i=1,m} \left(\theta_{\Delta P_{v,i}}^{\bar{V}} B_{\Delta P_{v,i}}\right)^2}$$
$$= 0.1 \cdot 0.25 \frac{C_C C_{pitot} \overline{\Delta P_v^{1/2}}}{2\sqrt{m}}$$

For an average velocity of 9 ft/s, a pitot tube coefficient of 0.80, $\overline{\Delta P_v^{1/2}} = 4.86\, inwg^{1/2}$, and 40 point pitot traverse

$$B_{\Delta P_v}^{\bar{V}} = 0.1 \cdot 0.25 \frac{2.316 \cdot 0.8 \cdot 4.86}{2\sqrt{40}} = 0.018\, ft/s$$

The uncertainty of 0.018 ft/s is 0.2 percent of the average velocity. This uncertainty is not sufficiently significant to justify recording the variation in velocity pressure at each traverse point. Instead it is recommended that component be neglected unless the variation in the manometer level is more than 15 percent of the reading. In that case, the variation in the velocity pressure should be recorded at a least five points on a single radius. The average of the maximum variation at these points could be used to estimate the uncertainty of the “eyeball” average reading.

In terms of water flow rate, the systematic uncertainty for differential pressure is

$$B^{q}_{\Delta P_V} = \theta^{q}_{V} B^{V}_{\Delta P_V} = C_0 A_t B^{V}_{\Delta P_V}$$

When a pressure transmitter is used to read the velocity pressure, the accuracy of the pressure transmitter (calibration tolerance plus signal conditioning and environmental effects) is used to calculate the uncertainty of the velocity due to differential pressure. Since the same instrument is used to make all of the measurements, the systematic uncertainty due to velocity pressure is perfectly correlated.

$$B^{\bar{V}}_{\Delta P_v} = \sum_{i=1,m} \theta^{\bar{V}}_{\Delta P_{v,i}} B_{\Delta P_{v,i}}$$

For a transmitter with an accuracy of 0.15 percent of reading and the same 9 ft/sec velocity,

$$B^{\bar{V}}_{\Delta P_v} = \frac{2.316 \cdot 0.8 \cdot 4.86}{2} 0.0015 = 0.0067\ ft/s$$

In terms of water flow rate, the systematic uncertainty for the velocity pressure is

$$B^{q}_{\Delta P_v} = \theta^{q}_{V} B^{V}_{\Delta P_v} = C_0 A_t B^{V}_{\Delta P_v}$$

For the example of the 53 inch pipe with an area of $15.09\ ft^2$, the uncertainty would be

$$B^{q}_{\Delta P_v} = 448.85 \cdot 15.09 \cdot 0.0067 = 45\ gpm$$

U.7-4 Uncertainty Due to Spatial Variation

There are two methodologies available to model the spatial variation in the velocity profile, the radial approach and the random approach. Each of these methods is based on a set of assumptions. The validity of these will vary according to the proximity of the flow measurement location to flow disturbances such as valves and elbows. Because these situations vary greatly from test to test, it is necessary that parties to a test agree on a methodology to be used to calculate the uncertainty due to spatial variation for a specific test.

Radial Approach

This methodology is consistent with that presented in ASME PTC 23 Section D.4.5.1 and ASME PTC 19.1 Section 10.3 Flow Measurement Using Pitot Tubes. The method is based on the axial symmetry of fully developed turbulent flow. The assumption is made the radial variation of the average velocity among the radii approximates the error due to spatial variation. The assumption of a fully developed turbulent flow will be valid if the flow location meets the recommendations of CTI STD 46 Section 3.1, 10 pipe diameters downstream of a disturbance and 5 pipe diameters upstream of a disturbance. Using the radial approach, the bias uncertainty is calculated by

$$\mathrm{B}_{\bar{\mathrm{v}}} = \frac{2\mathrm{s}_{\overline{\mathrm{v_{rad}}}}}{\sqrt{\mathrm{N_{rad}}}}$$

Where

$B_{\bar{v}}$ = Spatial uncertainty estimate for velocity

$s_{\overline{v_{rad}}}$= Standard deviation of the radially averaged velocity

$\overline{\mathrm{v_{rad,k}}}$= Average velocity for radius k

N_{rad} = Number of radii measured, usually 4

The use of the factor 2 in the denominator of the equation above, instead of the student’s T value, is based the measurement of the velocity at greater than 30 measurement points.

A velocity profile for a pipe with 50 pipe diameters upstream of the measurement location is illustrated in Figure U.3 and tabulated in Table U.4.

Figure U.3 Velocity Profile for Main Line

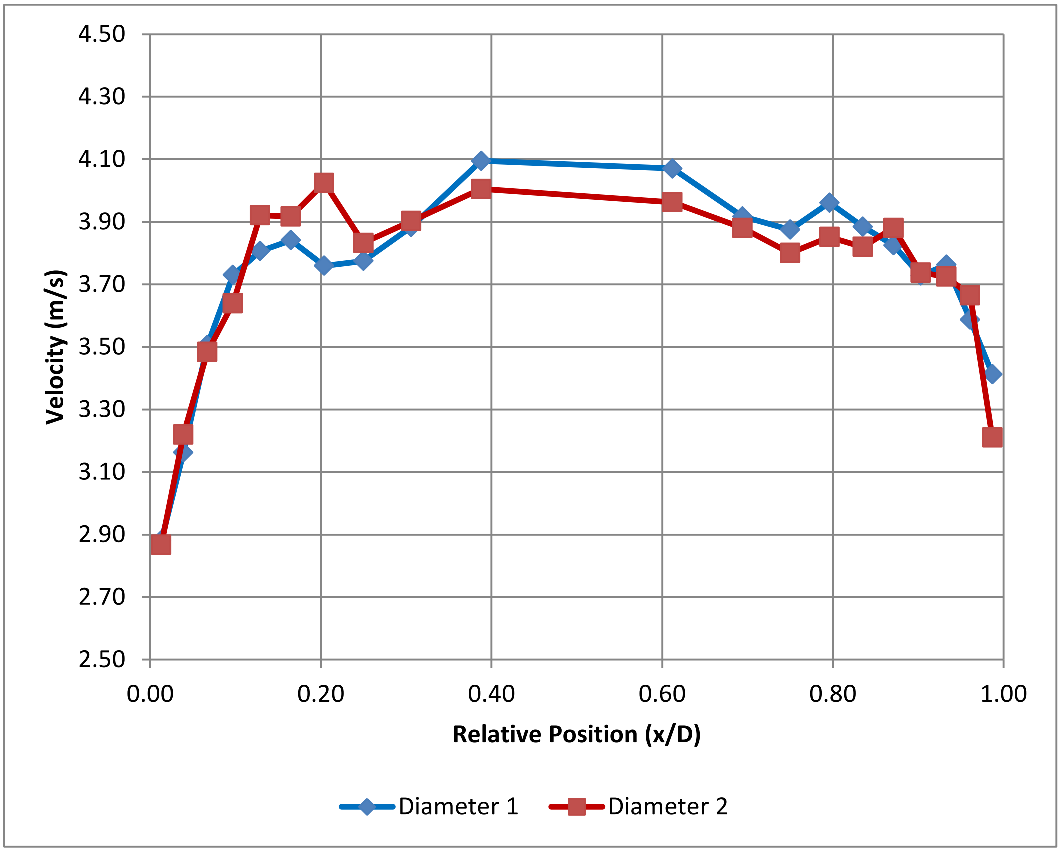


Table U.4 Velocity Profile for Main Line

	Relative	Diameter 1		Diameter 2	
Position	Position	Deflection	Velocity	Deflection	Velocity
	x/D	m	m/s	m	m/s
1	0.013	0.711	2.88	0.705	2.87
2	0.039	0.857	3.16	0.889	3.22
3	0.067	1.054	3.51	1.041	3.48
4	0.097	1.194	3.73	1.137	3.64
5	0.129	1.245	3.81	1.321	3.92
6	0.165	1.270	3.84	1.321	3.92
7	0.204	1.219	3.76	1.397	4.02
8	0.250	1.232	3.77	1.270	3.83
9	0.306	1.308	3.88	1.321	3.90
10	0.388	1.461	4.09	1.397	4.00
Average			3.64		3.68
11	0.612	1.461	4.07	1.384	3.96
12	0.694	1.359	3.92	1.334	3.88
13	0.750	1.334	3.87	1.283	3.80
14	0.796	1.397	3.96	1.321	3.85
15	0.835	1.346	3.88	1.302	3.82
16	0.871	1.308	3.83	1.346	3.88
17	0.903	1.245	3.73	1.251	3.74
18	0.933	1.270	3.76	1.245	3.72
19	0.961	1.156	3.59	1.207	3.66
20	0.987	1.048	3.41	0.927	3.21
Average			3.80		3.75

For the data in Table U.4, the standard deviation of the four average radial velocities is 0.071 m/s.

$$B^{V}_{spatial} = \frac{2 * 0.071}{\sqrt{4}} = 0.071 ft/s$$

For a flow area of $2.627m^2$, the uncertainty due to spatial variation would be

$$B^{q}_{spatial} = 1000 * 2.627 * 0.071 = 187\ L/s$$

Random Approach

The random approach is consistent with that presented in PTC 23 Section D.4.5.2. This method is used where the disturbances upstream and downstream of the measurement location make the assumptions involved in the radial method invalid.

The velocity measured at each point is randomly distributed about the average value.

The point measurement grid adequately captures the true variation in velocity at the measurement location.

The first assumption is clearly not valid in the case of fully developed turbulent flow and even highly disturbed flow profiles are continuous rather than randomly distributed about the average. To the extent that there is a continuous velocity profile along a single radius, the random approach will overestimate the error to spatial variation. In highly disturbed flow situations, those approaching the minimum limits specified by STD 146 (5 pipe diameters downstream of a disturbance, 1 pipe diameter downstream of a disturbance) it is not clear that number of radii sampled, usually 4, is sufficient to establish the true radial variation. This could cause the method to underestimate the uncertainty due to spatial variation. These errors will be in opposite directions and may tend to cancel.

Using the random approach, the uncertainty due to spatial variation in velocity is

$$B_{\overline{v}} = \frac{2s_{\overline{v}}}{\sqrt{m}}$$

Where

$B_{\overline{v}}$ = Spatial uncertainty estimate for velocity

$s_{\overline{v}}$ = The standard deviation of velocity using all measurements

m = Number of measurement points

The variance of the spatial variation in velocity at the traverse location is calculated by

$$s^{V}_{spatial} = \sqrt{\frac{\sum_{i=1,m}(V_i - \overline{V})^2}{m}}$$

The spatial systematic uncertainty for the velocity at the traverse location is

$$B^{V}_{spatial} = \frac{2s^{V}_{spatial}}{\sqrt{m}}$$

In terms of water flow, the spatial systematic uncertainty is

$$B^{q}_{spatial} = \theta^{q}_{V} B^{V}_{spatial} = C_0 A_t B^{V}_{spatial}$$

A flow profile for a riser is illustrated in Figure U.4 and summarized in Table U.5.

Figure U.4 Riser Velocity Profile

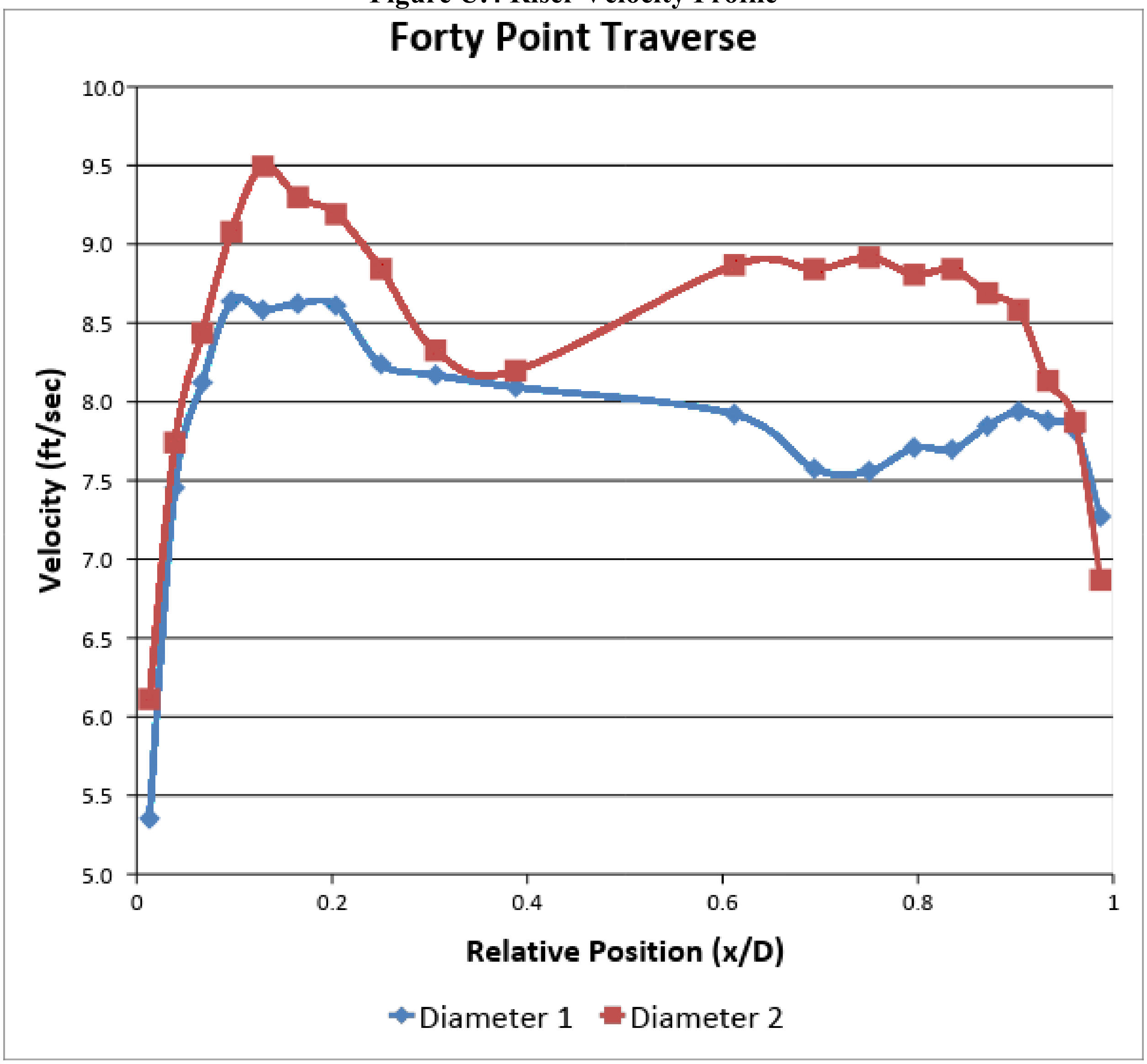


Table U.5 Velocity Values for Riser Flow Location

	Relative	Diameter 1		Diameter 2	
Position	Position	Deflection	Velocity	Deflection	Velocity
		inches	ft/s	inches	ft/s
1	0.013	8.25	5.35	10.8	6.11
2	0.039	16	7.45	17.3	7.74
3	0.067	19	8.12	20.5	8.44
4	0.097	21.5	8.64	23.8	9.08
5	0.129	21.25	8.58	26.0	9.49
6	0.165	21.5	8.62	25.0	9.30
7	0.204	21.5	8.61	24.5	9.19
8	0.25	19.75	8.24	22.8	8.84
9	0.306	19.5	8.17	20.3	8.32
10	0.388	19.25	8.09	19.8	8.20
Average			7.99		8.47
11	0.612	18.75	7.92	23.50	8.87
12	0.694	17.25	7.57	23.50	8.84
13	0.75	17.25	7.56	24.00	8.92
14	0.796	18.00	7.71	23.50	8.81
15	0.835	18.00	7.70	23.75	8.84
16	0.871	18.75	7.84	23.00	8.69
17	0.903	19.25	7.94	22.50	8.58
18	0.933	19.00	7.88	20.25	8.13
19	0.961	18.75	7.82	19.00	7.87
20	0.987	16.25	7.27	14.50	6.87
Average			7.72		8.44

For the traverse in Table U.5 and a traverse area of 15.40 ft,

$$s^{V}_{spatial} = 0.818\ ft/s$$

$$B^{V}_{spatial} = \frac{2 * 0.818}{\sqrt{40}} = 0.259\ \ ft/s$$

$$B^{q}_{spatial} = 448.85 \cdot 15.40 \cdot 0.259 = 1787\ gpm$$

U.7-5 Total Systematic Uncertainty for Water Flow

The total systematic uncertainty of the flow measurement includes the uncertainty of the diameter measurement, the pitot coefficient, the differential pressure measurement and the spatial variation. The total systematic uncertainty of the measurement of the water flow at a single traverse location is

$$B_q = \sqrt{(B^{q}_{d_t})^2 + (B^{q}_{\Delta P_v})^2 + (B^{q}_{C_{pitot}})^2 + (B^{q}_{spatial})^2}$$

For the example described previously (velocity pressure measured with at transducer) the total uncertainty in the water flow would be

$$B_q = \sqrt{(876)^2 + (45)^2 + (1219)^2 + (1271)^2}$$
$$= 1965\ gpm$$

When flow is measured in multiple locations with the same pitot tube, errors due to the pitot calibration and the velocity pressure measurement will be totally correlated. If it is conservatively assumed that all of the errors are totally correlated

$$B_q = \sum_{i=1,m} B_{q,i}$$

Where

m = The number flow measurements

$B_{q,i}$ = The systematic uncertainty of a single flow measurement, i

In most cases, the flow profiles will be different. It is, therefore, reasonable to assume the errors due to spatial uncertainty are uncorrelated. For this (recommended) approach, the systematic uncertainty for the flow measurement would be

$$B_q = \sum_{i=1,m} B^q_{d_t i} + \sum_{i=1,m} B^q_{\Delta P_v i} + \sum_{i=1,m} B^q_{C_{pitot} i} + \left[\sum_{i=1,m} (B^q_{spatial,i})^2\right]^{1/2}$$

Attachment U-1 Uncertainty in Cold Water Temperature

Influence of Pump Discharge Pressure

When the cold water temperature is measured in a flowing well downstream of a pump, the measured cold water temperature is corrected for the influence of throttling and pump efficiency. From Appendix, the correction equation is

$$\Delta T = C_T \frac{P_d}{\eta_p}$$

Where

ΔT = Correction for heat added by the pump

C_T = Conversion factor, 0.000239°C/kPa for SI units, 0.00296°F/psi for IP units

P_d= Pump discharge pressure, kPa (psig)

η_{pump}= Pump efficiency

The sensitivity coefficient of the temperature correction to pump discharge pressure is

$$\theta^{\Delta T}_{P_d} = \frac{\partial \Delta T}{\partial P_D} = \frac{C_T}{\eta_{pump}}$$

The sensitivity coefficient of the temperature correction to pump efficiency is

$$\theta^{\Delta T}_{\eta_{pump}} = \frac{\partial \Delta T}{\partial \eta_{pump}} = -\frac{C_T}{\eta^2_{pump}}$$

For a pump discharge pressure of 30 psig and a pump efficiency of 85 percent,

$$\Delta T = 0.002966 \frac{30}{0.85} = 0.105°F$$

$$\theta^{\Delta T}_{P_d} = \frac{0.002966}{0.85} = 0.00349 \frac{°\mathrm{F}}{psi}$$

$$\theta^{\Delta T}_{\eta_{pump}} = -\frac{0.002966}{0.85^2} = 0.00411°\mathrm{F}$$

For a systematic uncertainty of 3 psi in the discharge pressure and 5 percent in the pump efficiency, the systematic uncertainty in the temperature correction would be

$$B_{\Delta T} = \sqrt{[\theta^{\Delta T}_{P_d} B_{P_d}]^2 + [\theta^{\Delta T}_{\eta_{pump}} B_{\eta_{pump}}]^2}$$

$$B_{\Delta T} = \sqrt{[0.00349 * 3]^2 + [0.0041 * 0.05]^2} = 0.010°\mathrm{F}$$

This uncertainty is insignificant in comparison to the other uncertainties in measuring the cold water temperature.

Influence of Makeup Flow and Temperature

When makeup enters the cold water basin upstream of the measurement location, the cold water temperature is calculated by

$$T_{cw} = \frac{qT_{cwm} - q_{mu}T_{mu}}{q - q_{mu}}$$

Where

q = Circulating water flow

q_{mu}= Makeup water flow

T_{cwm} = Measured cold water temperature

T_{mu}= Makeup water temperature

In this situation the overall systematic uncertainty of the cold water temperature is calculated by

$$B_{T_{cw}} = \sqrt{(\theta^{T_{cw}}_{T_{cwm}} B_{T_{cwn}})^2 + (\theta^{T_{cw}}_{T_{mu}} B_{T_{mu}})^2 + (\theta^{T_{cw}}_{q} B_q)^2 + (\theta^{T_{cw}}_{q_{mu}} B_{q_{mu}})^2}$$

The sensitivity coefficients of the cold water temperature with respect to the measured parameters are calculated by

$$\theta^{T_{cw}}_{T_{cwm}} = \frac{\partial T_{cw}}{\partial T_{cwm}} = \frac{q}{q - q_{mu}}$$

$$\theta^{T_{cw}}_{T_{cwm}} = \frac{\partial T_{cw}}{\partial T_{mu}} = \frac{q_{mu}}{q - q_{mu}}$$

$$\theta^{T_{cw}}_{q} = \frac{\partial T_{cw}}{\partial T_{mu}} = \frac{q_{mu}(T_{mu} - T_{cwm})}{(q - q_{mu})^2}$$

$$\theta^{T_{cw}}_{q_{mu}} = \frac{\partial T_{cw}}{\partial q_{mu}} = \frac{q \quad (T_{cwm} - T_{mu})}{(q - q_{mu})^2}$$

The uncertainty in the cold water temperature will normally be very close to uncertainty of the measured cold water temperature. An example where the uncertainty in the makeup flow rate had a small effect on the cold water temperature uncertainty is detailed in Table U-5.

Table U-6 Cold Water Temperature Uncertainty

Parameter	Units	Value	Systematic Uncertainty	Sensitivity Factor	Cold Water Temperature Uncertainty
Circulating water flow	gpm	139,685	3850	-5.571E-06	0.021
Measured cold water temperature	°F	95.1	0.18	1.0283	0.185
Makeup water flow	gpm	3850	385	0.000202	0.078
Makeup water temperature	°F	68.4	0.18	0.028	0.005
Cold Water Temperature Systematic Uncertainty(°F)					**0.20**

The values in Table U-5 are atypical in that they maximize the difference between the uncertainty of the measured cold water temperature and that of the cold water temperature corrected for makeup. The difference in the measured cold water and the makeup temperature was 27°F.The makeup flow rate was 3 percent of the circulating water flow rate and the uncertainty in the makeup flow rate was 10 percent of the measured value. For this case, the uncertainty in the corrected cold water temperature was 0.20°F compared to the uncertainty of the measured cold water temperature of 0.18°F.

APPENDIX UC
CORRELATED UNCERTAINTY

The treatment of correlated uncertainty in this section is based on Section 8-1 of ASME PTC 19.1 Test Uncertainty. The expanded form of the systematic uncertainty of result calculated based multiple parameters is

$$B_r = \left[\sum_{i=1}^{I}(\theta_i B_i)^2 + 2\sum_{i=1}^{I-1}\sum_{k=i+1}^{I}\theta_i\,\theta_k B_{ik}\right]^{1/2}$$

Where

I = number of measured parameters

B_i = Uncertainty for parameter i

θ_i = Sensitivity of result to parameter i

θ_k = Sensitivity of result to parameter k

B_{ik}= Covariance term representing the portion of the systematic uncertainties of parameter i and that arise from the same source and, therefore, are perfectly correlated

The covariance term is calculated by

$$B_{i,k} = \sum_{l=1}^{L} B_{i,l}\,B_{k,l}$$

Where

L = Number of perfectly correlated error sources

l = Index for error source

$B_{i,l}$ = Systematic uncertainty of parameter i due to error source l

$B_{k,l}$ = Systematic uncertainty of parameter k due to error source l

The calculation of correlated uncertainty is illustrated by the following example. Average cold water temperature is calculated averaging the readings of four temperature sensors. Correlated uncertainty will be used to calculate the systematic uncertainty of the average cold water temperature. The cold water measurements are made with the same data logger. The uncertainty of significant error sources are tabulated in Table U-7.

Table U-7 Temperature Measurement Systematic Uncertainty Sources

Index	Source	Value (°F)	Correlated
1	Calibration standard	0.036	C
2	Meter reading resistance of standard	0.018	C
3	Meter reading resistance of sensor	0.08	C
	Spatial variation in water bath	0.05	U
	Lack of fit – calibration equation	0.03	U
4	Data logger analog to digital converter	0.035	C
5	Data logger resistance card	0.08	C
6	Ambient temperature effect on data logger	0.08	C
	Data logger read relay	0.043	U
Total		0.165	

The correlated column indicates whether the uncertainty source is correlated for two cold water temperature measurements.

The sensitivity of the average cold water temperature to the sensor readings is

$$\theta_{T1} = \theta_{T2} = \theta_{T3} = \theta_{T4} = \frac{1}{4}$$

The systematic uncertainty of a single temperature measurement, calculated as the square root of the sum of the square of the uncertainty due to all error sources, is 0.165°F. The covariance term for any two temperature sensors $B_{i,k}$, is

$$B_{i,k} = 0.036 * 0.036 + 0.018 * 0.018 + 0.08 * 0.08 + 0.035 * 0.035 + 0.08 * 0.08 + 0.08 * 0.08 = 0.02205$$

The total systematic uncertainty for the average cold water temperature measurement is

$$B_{\bar{T}_{cw}} = \begin{bmatrix} (\theta_{T1}B_{T1})^2 + (\theta_{T2}B_{T2})^2 + \ 2\theta_{T1}\theta_{T2}B_{1,2} + 2\theta_{T1}\theta_{T3}B_{1,3} + 2\theta_{T1}\theta_{T4}B_{1,4} \\ (\theta_{T3}B_{T3})^2 + (\theta_{T4}B_{T4})^2 + 2\theta_{T2}\theta_{T3}B_{2,3} + 2\theta_{T2}\theta_{T4}B_{2,4} + 2\theta_{T3}\theta_{T4}B_{2,4} \end{bmatrix}^{0.5}$$

$$B_{\bar{T}_{cw}} = \begin{bmatrix} (0.25 * 0.165)^2 + (0.25 * 0.165)^2 + 2 * 0.25 * 0.25 * 0.02205 + 2 * 0.25 * 0.25 * 0.02205 + 2 * 0.25 * 0.25 * 0.02205 \\ (0.25 * 0.165)^2 + (0.25 * 0.165)^2 + 2 * 0.25 * 0.25 * 0.02205 + 2 * 0.25 * 0.25 * 0.02205 + 2 * 0.25 * 0.25 * 0.02205 \end{bmatrix}^{0.5}$$

$$B_{\bar{T}_{cw}} = 0.153°\mathrm{F}$$

Because many of the error sources are correlated, the systematic uncertainty for measurement of cold water temperature with four sensors is only slightly less the systematic uncertainty for a single sensor.

Cooling Technology Institute

PO Box 681803 Houston, Texas 77268

281.583.4087 – Fax: 281.537.1721 – www.cti.org

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