ASME PTC 23-2003

[Revision of ANSI/ASME PTC 23-1986 (R1997)]

REAFFIRMED 2009

FOR CURRENT COMMITTEE PERSONNEL
PLEASE E-MAIL CS@asme.org

ATMOSPHERIC WATER COOLING EQUIPMENT

PERFORMANCE TEST CODES

An American National Standard

The American Society of Mechanical Engineers

Intentionally left blank

The American Society of
Mechanical Engineers

A N A M E R I C A N N A T I O N A L S T A N D A R D

ATMOSPHERIC WATER COOLING EQUIPMENT

ASME PTC 23-2003

[REVISION OF ANSI/ASME PTC 23-1986 (R1997)]

PERFORMANCE TEST CODES

Date of Issuance: November 10, 2003

This Standard will be revised when the Society approves the issuance of a new edition. There will be no addenda issued to this edition.

ASME issues written replies to inquiries concerning interpretations of technical aspects of this Standard. Interpretations are published on the ASME Web site under the Committee Pages at http://www.asme.org/codes/ as they are issued.

The American Society of Mechanical Engineers
Three Park Avenue, New York, NY 10016-5990

CONTENTS

FOREWORD

In 1918, revision began on the original ten Codes that formed the 1915 edition of the ASME Power Test Codes and it was decided to include water cooling in the list of revised Codes. Following the appearance of the Test Code for Atmospheric Water Cooling Equipment in tentative form in the August 1928 issue of *Mechanical Engineering*, the Society presented this Code for discussion at a public hearing held in December 1928 during its annual meeting in New York. The Code was approved at the June 2, 1930 meeting of the Standing Committee, and was adopted by the Council as a standard practice of the Society on August 4, 1930.

A new Technical Committee for Atmospheric Water Cooling Equipment was formed in 1948 to update the Code. Agreement on the location for the wet-bulb temperature measurement was the major issue. While there was general recognition that tower performance was governed by the entering wet-bulb temperature, at that time there was concern about responsibility for plume re-circulation and interference. The difficulty of obtaining adequate air temperature coverage of both sides of large towers was considered insurmountable. For this reason, some believed it would be better to relate performance to an ambient temperature measured some distance upwind and to have performance stated and substantiated on that basis.

In 1954, a seven-person Subcommittee with representation from the American Society of Refrigeration Engineers and the Cooling Tower Institute (now known as Cooling Technology Institute) was appointed to abbreviate and bring to an early conclusion the work of the 1948 Technical Committee. The Subcommittee's work was completed and a Code based on ambient wet-bulb temperature measurement was adopted by the Society on January 29, 1958. The growing number and size of towers, the evolution of the natural draft tower, and continuing disagreement on ambient versus entering wet-bulb temperature led to the formation of a new Committee in December 1968. Modern instrumentation and the availability of data acquisition systems had advanced measurement methods since the 1958 Code. Model test results and error analysis made available to the Committee supported the position that tower performance should be related to entering wet-bulb temperature in the revised Code. This issue of the Code was approved by ANSI and published by the Society in November 1986.

Continued advances in instrumentation, experiences with the testing itself and test uncertainty, installation of a variety of other types of evaporative cooling equipment, and more stringent environmental regulations led to the convening of a new Committee in 1995. Its objective was to extend the Code to include plume abatement compliance of wet–dry towers and the performance test procedures for cooling towers, closed circuit evaporative coolers and wet surface air-cooled condensers. In the interim, the impact on plant economics of the cooling system's operating performance became better understood. Hence, Appendix A in the new Code addresses practical techniques of monitoring the performance of cooling towers.

As in past editions of PTC 23, the most accurate test methods were established as Code. However, the Committee was aware that for some towers, an elaborate test was not practical or economically viable. Therefore, nonmandatory Appendix K provides simpler test methods. These test methods, being less accurate, have a higher uncertainty.

To expedite the completion of this version of the Code, sections of CTI Code ATC-105 were used with the permission of the Cooling Technology Institute (CTI). That contribution is acknowledged and appreciated.

The Committee voted to approve the document on November 13, 2002. It was then approved and adopted by the Council as a Standard practice of the Society by action of the Board on Performance Test Codes on February 28, 2003. The Code was also approved as an American National Standard by the ANSI Board of Standards Review on March 13, 2003.

PERFORMANCE TEST CODE COMMITTEE 23
ON ATMOSPHERIC WATER COOLING EQUIPMENT

(The following is the roster of the Committee at the time of approval of this Standard.)

COMMITTEE OFFICERS

J. M. Burns, *Chair*
F. L. Michell, *Vice Chair*
J. H. Karian, *Secretary*

COMMITTEE PERSONNEL

J. M. Burns, Burns Engineering Services, Inc.
J. W. Cuchens, Southern Company Services, Inc.
D. H. Drew, *Alternate,* American Electric Power
R. D. Fulkerson, Fulkerson & Associates, Inc.
K. W. Hennon, Power Generation Technologies/E.S.C.
E. Hernandez, Hitachi America Ltd.
D. Hutton, Baltimore Aircoil Co.
D. C. Karg, Santee Cooper
J. H. Karian, The American Society of Mechanical Engineers
C. Lazenby, *Alternate,* Southern Company Services, Inc.
P. A. Lindahl, Jr, Marley Cooling Tower Co.
F. L. Michell, American Electric Power
G. R. Mirsky, *Alternate,* Hamon Cooling Towers
M. Monjoie, Hamon Thermal Europe
G. C. Morris, Niagara Blower Co.
D. Wheeler, *Alternate,* Power Generation Technologies

The Committee acknowledges with thanks the contributions made by Marcel R. Lefevre to the initial development of this revision and for his initiative in helping to reconstitute the Committee.

BOARD ON PERFORMANCE TEST CODES

CORRESPONDENCE WITH THE PTC 23 COMMITTEE

General. ASME Codes are developed and maintained with the intent to represent the consensus of concerned interests. As such, users of this Code may interact with the Committee by requesting interpretations, proposing revisions, and attending Committee meetings. Correspondence should be addressed to:

Secretary, PTC 23 Standards Committee
The American Society of Mechanical Engineers
Three Park Avenue
New York, NY 10016-5990

Proposing Revisions. Revisions are made periodically to the Code to incorporate changes that appear necessary or desirable, as demonstrated by the experience gained from the application of the Code. Approved revisions will be published periodically.

The Committee welcomes proposals for revisions to this Code. Such proposals should be as specific as possible, citing the paragraph number(s), the proposed wording, and a detailed description of the reasons for the proposal, including any pertinent documentation.

Interpretations. Upon request, the PTC 23 Committee will render an interpretation of any requirement of the Code. Interpretations can only be rendered in response to a written request sent to the Secretary of the PTC 23 Standards Committee.

The request for interpretation should be clear and unambiguous. It is further recommended that the inquirer submit his/her request in the following format:

Subject:	Cite the applicable paragraph number(s) and the topic of the inquiry.
Edition:	Cite the applicable edition of the Code for which the interpretation is being requested.
Question:	Phrase the question as a request for an interpretation of a specific requirement suitable for general understanding and use, not as a request for an approval of a proprietary design or situation. The inquirer may also include any plans or drawings, which are necessary to explain the question; however, they should not contain proprietary names or information.

Requests that are not in this format will be rewritten in this format by the Committee prior to being answered, which may inadvertently change the intent of the original request.

ASME procedures provide for reconsideration of any interpretation when or if additional information that might affect an interpretation is available. Further, persons aggrieved by an interpretation may appeal to the cognizant ASME Committee or Subcommittee. ASME does not "approve," "certify," "rate," or "endorse" any item, construction, proprietary device, or activity.

Attending Committee Meetings. The PTC 23 Standards Committee regularly holds meetings, which are open to the public. Persons wishing to attend any meeting should contact the Secretary of the PTC 23 Standards Committee.

INTRODUCTION

This Code describes instruments, test procedures, and analysis of test data to be used to determine the performance of all designs of cooling towers and evaporative cooling equipment. It defines procedures that determine the plume abatement compliance of wet-dry cooling towers. The Code outlines practical methods of monitoring the performance of cooling towers in an appendix. It provides explicit test procedures that will yield results of the highest level of accuracy, consistent with the best current engineering practices and knowledge in this field. The Code is not intended to be used for spray-cooling devices, cooling canals or ponds, or cooling lakes.

To aid in an overall study of the Code, the following review sequences are suggested.

A quick survey of the Code can be obtained by reading the introductions to each section followed by the Test Procedures.

At the plant design, contractual agreement, or specification stage, it is advisable to review in order:

(a) achievable test uncertainty stated in Object and Scope section

(b) test procedures or alternatively the particular special test from the appendix

(c) Guiding Principles section

(d) Instruments and Methods of Measurement section for the recommended requirements, particularly the water flow measurement

Performance monitoring projects should review Appendix A before reviewing the details of Code Sections.

When this Code is to be used as a means to determine fulfillment of contract obligations, the contracting parties shall agree in advance on the test procedures, uncertainty estimates and implications, methods of presentation of data, and presentation of results.

Considerable efforts were made to write this cooling tower Code so that all the related technology was contained within the document itself; however, in all instances this was not possible. In these cases and unless otherwise specified, all references to other codes refer primarily to ASME Performance Test Codes or to Cooling Technology Institute Standards. Any terms not defined herein are listed in the Code in Definitions and Values, ASME PTC 2. Descriptions of instruments and apparatus may be found in the PTC 19 series of supplemental codes. The general basis of the uncertainty analysis beyond that specified in this Code may be found in the supplement ASME PTC 19.1, Test Uncertainty. A careful study should be made of all the referenced codes, but in the event of discrepancies between specific directions contained herein and those codes incorporated by reference, ASME PTC 23 shall govern.

Intentionally left blank

ATMOSPHERIC WATER COOLING EQUIPMENT

Section 1
Object and Scope

1-1 OBJECT

This Code provides uniform test methods for conducting and reporting thermal performance characteristics of wet mechanical draft, natural draft, wet–dry cooling towers, closed circuit evaporative (wet) coolers, and wet surface air-cooled steam condensers. This Code also provides directions and rules for conducting and reporting plume abatement of wet–dry cooling towers and water consumption of any cooling tower. It provides explicit test procedures to yield results of the highest level of accuracy consistent with the best engineering knowledge and practice currently available. The purpose of this Code is to provide rules for monitoring thermal performance or for conducting acceptance tests on all of the cooling equipment referenced above. It provides rules for monitoring plume abatement and conducting plume-abatement acceptance tests on wet–dry cooling towers.

The test can be used to determine compliance with contractual obligations and can be incorporated into commercial agreements. A test shall be considered an ASME Code Test only if the test procedures comply with those allowed in this Code and the post-test uncertainty analysis results are in accord with para. 1.3.

1-2 SCOPE

This Code provides rules for determining the performance of all referenced cooling equipment with regard to the thermal capability, deviation from design thermal capability, or deviation from design cold water temperature. This Code also provides procedures for assessing the compliance to specified plume abatement requirements characteristic of a wet–dry cooling tower. It is not intended for tests of atmospheric wind towers, dry coolers, spray canals, or ponds, although sections of this Code may be useful for that purpose. The determination of special data or verification of guarantees that are outside the scope of this Code, shall be made only with the written agreement of the parties to the test. The agreed methods of measurement and computation shall be defined in writing and fully described in the test report.

1-3 UNCERTAINTY

The application of uncertainties to adjust test results or guarantees is specifically not permitted because the test results themselves provide the best indication of actual performance. The uncertainty values are used to determine the validity of the test and have no relationship to the expected performance of the equipment. The uncertainty values reflect the accuracy of the test instrumentation and stability of the test conditions.

The results of a thermal performance test, conducted in full compliance with the procedures and instrumentation specified in this Code, shall be considered valid if the calculated overall uncertainty in the thermal capability is less than:

(a) natural draft cooling tower: ±6.0%
(b) mechanical draft cooling tower: ±5.5%
(c) wet–dry cooling tower: ±6.0%
(d) closed circuit wet evaporative cooler: ±5.5%
(e) wet surface air-cooled steam condensers: ±8%

The results of plume characteristic and mixing quality tests of wet–dry cooling towers are greatly a function of the stability of the operating and environmental conditions. Because a plume characteristic test incorporates all of the same measurements as a thermal performance test plus other measurements, the uncertainty in the test result will be higher than for a mechanical draft tower. All efforts should be made to conduct these tests in fully code compliant conditions. As the stability of the test conditions deteriorates, the test uncertainty will increase. Because the sensitivity of the test result to environmental conditions (e.g., wind speed and wind direction) cannot be mathematically defined, the uncertainty in the test result is not calculated.

Because of the variety of methods and instruments used in the conduct of performance tests, the test uncertainty (for all tests with the exception of the plume characteristic and mixing quality tests) must be determined by an uncertainty analysis based on ASME PTC 19.1, but specifically applied to cooling equipment as indicated within this Code. It should be noted that the collection, reduction, and evaluation of thermal data is greatly facilitated through the use of a remote data acquisition system. The use of a data acquisition system is preferable when compared to a manual recording system because the increased number of test measurements will reduce the random error in the measurements. A post-test uncertainty analysis is required. If the calculated test uncertainty is greater than the above stated values, the test is to be considered inconclusive.

The application of uncertainties to adjust test results or guarantees is specifically not permitted because the test results themselves provide the best indication of actual performance.

Section 2
Definitions and Description of Terms

2-1 SYMBOLS

The following symbols are to be used unless otherwise defined in the text.

Symbol	Definition	Units: U.S. Customary	Units: SI
A	Area	ft^2	m^2
BHP	Brake horsepower	HP	kW
CI	Dye injection concentration	ppb	ppb
C_{pPF}	Specific heat of process fluid at the average temperature	Btu/lbm	J/kg-K
C_{pW}	Specific heat of spray water flowing over external surface of heat exchanger	Btu/lbm	J/kg-K
CS	Sample concentration	ppb	ppb
E_m	Voltage at motor	V	V
E_{np}	Nameplate voltage	V	V
E_{ob}	Observed voltage at test station	V	V
exp	Exponent	. . .	. . .
F_s	Fluorescence at standard temperature, T_s	. . .	. . .
F_θ	Measured fluorescence at temperature T, corrected for background and instrument offset	. . .	. . .
G	Total dry air flow through tower	lbm/hr	kg/s
HB	Barometric pressure	psia	Pa
h_e	Enthalpy of exhaust air	Btu/lbm dry air	J/kg dry air
H_e	Specific humidity of exhaust air	lbm water vapor/lbm dry air	kg water vapor/kg dry air
H_f	Friction loss	ft of water	m of water
h_{fg}	Change in steam enthalpy	Btu/lb	J/kg
h_{in}	Enthalpy of entering air	Btu/lbm dry air	J/kg dry air
H_{in}	Specific humidity of inlet air	lbm water vapor/lbm dry air	kg water vapor/kg dry air
H_n	Nozzle loss	ft of water	m of water
HP_d	Design horsepower	HP	kW
HP_{np}	Nameplate horsepower	HP	kW
HP_t	Calculated test horsepower	HP	kW
H_s	Static head	ft of water	m of water
H_{vi}	Velocity head	ft of water	m of water
i, j, k	Indices of an element	. . .	. . .
I_{np}	Full load nameplate current	A	A
I_{ob}	Observed current	A	A
kW_{ob}	Observed test power input to fan motor	kW	kW
L/G	Liquid-to-gas ratio	. . .	. . .
L_d	Design water flow rate	gpm	m^3/s
L_m	Measured water flow rate	gpm	m^3/s
M_Q	Mixing quality	%	%

Symbol	Definition	Units	
		U.S. Customary	SI
M_q	Mixing quality percentage	%	%
N_{GL}	Number of horizontal grid levels	. . .	. . .
N_{GS}	Number of equally spaced vertical grid strings	. . .	. . .
N_{WB}	Minimum number of wet-bulb instruments	. . .	. . .
P_{atm}	Atmospheric pressure during the test	psia	Pa
P_{diag}	Atmospheric pressure of the psychrometric diagram	psia	Pa
PF	Power factor	. . .	. . .
P_{Ht}	Measured (test) pumping head expressed in height of water column	ft	m
P_s	Measured static pressure expressed in height of water column	ft	m
P'_s, P''_s	Static pressure referred to the inlet centerline expressed in height of water column	ft	m
P_w	Circulating water pressure (main stream)	psig	Pa
P_{wd}	Circulating water pump discharge pressure	psig	Pa
P_{wi}	Circulating water pump inlet pressure	psig	Pa
Q_{adj}	Mass steam flow adjusted	lb/h	kg/s
q_d	Total design air flow through tower	ft^3/min	m^3/s
Q_D	Total design circulating water flow through tower	gal/min	m^3/s
Q_{dye}	Dye injection flow rate	ml/min	ml/min
Q_E	Expected flow rate	gpm	m^3/s
Q_{meas}	Mass steam flow measured	lb/h	kg/s
Q_{MU}	Volumetric flow rate of makeup water	gpm	m^3/s
Q_{PFt}	Volumetric flow rate of process fluid measured during test	gpm	m^3/s
$Q_{PFt\ \text{adj}}$	Volumetric flow rate of process fluid, measured and adjusted for makeup	gpm	m^3/s
q_t	Total measured air flow through tower	ft^3/min	m^3/s
R	Range	°F	°K
RH	Measured relative humidity (inlet, upwind, or exhaust)	%	%
RH_c	Corrected relative humidity (inlet, upwind, or exhaust)	%	%
RH_{gc}	Equivalent exhaust air relative humidity	%	%
$RH_{\max}$	Maximum limit of the exhaust air relative humidity	%	%
S	Thermal lag time	min	min
T_{21}	Cold water temperature corrected for throttling	°F	°C
T_{22}	Cold water temperature corrected for heat added by the pump	°F	°C
T_{23}	Cold water temperature corrected for makeup and blowdown	°F	°C
T_B	Temperature of blowdown leaving basin	°F	°C
T_{BE}	Temperature of basin exit stream	°F	°C
T_{CPF}	Temperature of the cold process fluid leaving the tower	°F	°C
T_{CW}	Cold water temperature	°F	°C
T_{DB}	Dry-bulb temperature of entering air (inlet, outlet, or exhaust)	°F	°C
T_{HE}	Temperature of water flow into the basin other than circulating water or makeup	°F	°C

Symbol	Definition	Units	
		U.S. Customary	SI
T_{HL}	Temperature of water flow from basin other than circulating water or blowdown	°F	°C
T_{HPF}	Temperature of the hot process fluid entering the tower	°F	°C
T_{HW}	Hot water temperature	°F	°C
T_{MU}	Temperature of the makeup water entering the tower	°F	°C
T_{RW}	Temperature of external spray water, measured at the spray water pump discharge	°F	°C
T_s	Standard temperature	°F	°C
T_w	Water temperature	°F	°C
T_{WB}	Wet-bulb temperature of entering air (inlet, outlet, or exhaust)	°F	°C
V	Volume of water in basin during test	ft^3	m^3
V	Pipe velocity	ft/s	m/s
V_{da}	Specific volume of dry air flowing through tower	ft^3 mixture/lbm dry air	m^3 mixture/kg dry air
V_e	Local exhaust air flows (air velocity)	ft/s	m/s
V_m	Measured air velocity at measured angle	ft/s	m/s
V_{me}	Marked exhaust local air flows (air velocity)	ft/s	m/s
V_v	Vertical component of air velocity	ft/s	m/s
W_B	Blowdown rate	lbm/hr	kg/s
W_{BE}	Basin exit flow	lbm/hr	kg/s
W_{HE}	Water flow into basin other than circulating water or makeup	lbm/hr	kg/s
W_{HL}	Water flow out of the basin other than circulating water or blow-down	lbm/hr	kg/s
W_L	Circulating water flow	lbm/hr	kg/s
W_M	Makeup water flow	lbm/hr	kg/s
x	Location of temperature sensors	ft	m
Z_{ai}	Height of air inlet	ft	m
Z_{GL1}	Height of each horizontal grid line	ft	m
α	Measured air velocity angle from vertical	deg	deg
ΔY	Vertical distance of the center of the inlet water line above the point of static pressure measurement	ft	m
η_f	Fan efficiency	%	%
η_m	Motor efficiency	%	%
η_p	Pump efficiency	%	%
ρ_a	Density of air	lbm/ft^3	kg/m^3
ρ_{ad}	Design exit air density	lbm/ft^3	kg/m^3
ρ_D	Design density	lbm/ft^3	kg/m^3
ρ_{PF}	Density of process fluid	lbm/ft^3	kg/m^3
ρ_T	Test density	lbm/ft^3	kg/m^3
ρ_W	Density of water	lbm/ft^3	kg/m^3

2-2 DEFINITIONS

air: mixtures of gases and associated water vapor surrounding the earth; dry air plus its associated water vapor. The term is used synonymously with atmosphere.

air density: mass of air per unit volume.

air density, standard: air at density of 0.075 lbm/ft^3 (1.201 kg/m^3).

air, dry: mixture of dry gases present in the atmosphere.

air flow, mass: mass of dry air flowing through tower for reducing circulating water temperature.

air flow, volume: volume of air mixture flowing through the tower for reducing circulating water temperature.

ambient temperature: temperature of the atmosphere measured windward of the tower.

approach: difference between cold water temperature and entering wet-bulb temperature.

basin: an open structure located beneath the tower fill for collecting the circulating water.

basin curb: the top elevation of the tower basin, usually the datum from which tower elevations are measured.

blowout: circulating water blown out of the tower that is wind induced. Also called *windage*.

cell: the smallest subdivision of a tower, bounded by exterior wall(s) and/or partitions(s), which can function as an independent unit.

circulating water flow: quantity of hot water flowing into the tower to be cooled.

cold water temperature: average temperature of water as it leaves the tower basin.

cooling tower: a semi-enclosed device for cooling water by direct contact with air.

counterflow tower: a type of tower in which the air and water streams flow in opposing directions.

crossflow tower: a type of tower in which the air and water streams are in crosscurrent (perpendicular) flow.

distribution system: a system of conduits, orifices, weirs, or nozzles for receiving the circulating water entering the tower and distributing it over the fill or heat transfer plan area where it is in contact with air.

drift: circulating water lost from the tower in the form of fine droplets entrained in the exhaust air.

drift eliminator: devices(s) to minimize drift.

entering wet-bulb temperature: wet-bulb temperature of air temperature entering the tower; includes any effect of recirculation and/or interference.

evaporation: water evaporated from the circulating water into the atmosphere during the cooling process. It is independent of drift.

exhaust air: the mixture of dry air and water vapor leaving the tower.

exit basin temperature: temperature of circulating water as it leaves the coldwater-collecting basin.

fill: heat transfer devices placed in the tower for the purpose of facilitating direct contact between circulating water and air.

forced draft tower: type of mechanical draft tower in which the air-moving device is located at the air inlet.

heat load: the rate of heat removal from the circulating water.

hot water temperature: weighted average temperature of circulating water entering the tower.

induced draft tower: type of mechanical draft tower in which the air-moving device is located at the air exhaust.

interference: the thermal contamination of tower inlet air by air from a source extraneous to the tower.

makeup: water added to the system to replace water lost by evaporation, drift, blowdown, and leakage.

mechanical draft tower: type of cooling tower through which the air movement is effected by mechanical devices. See *forced draft tower* and *induced draft tower.*

natural draft tower: type of cooling tower through which the air movement is effected by the difference in densities of the entering and exhaust air.

partition wall: vertical interior wall, which is either transverse, longitudinal, or radial, that subdivides a mechanical or natural draft tower into cells.

range: difference between hot water and cold water temperatures.

recirculation: that portion of the tower exhaust air that reenters the tower inlet. It can be expressed as a difference between the average entering and windward side wet-bulb temperatures.

sound level: a weighted sound pressure level obtained by the use of metering characteristics and the weighting A, B, or C specified in the American National Standard Specification for Sound Level Meters, ANSI SI.4.

sound pressure level: the sound pressure level, in decibels (dB), of a sound is 20 times the logarithm to the base 10 of the ratio of the pressure of this sound to the reference pressure, 0.0002 microbars. It is the generally accepted unit of sound pressure level.

specific volume: the volume of air-vapor mixture per unit mass of dry air.

splash out: circulating water splashed from the tower that is not wind-induced.

thermal lag: the time interval before the temperature of the water leaving the hot water temperature measurement point is detected at the point of cold water temperature measurement.

tower pumping head: total head of water at the centerline of the circulating water inlet to the cooling tower, referred to the tower basin curb as a datum. It is the sum of the static pressure measured at the centerline of the inlet connection to the cooling tower, the velocity pressure at this point, and the vertical distance between this point and the top of the basin curb.

wet-bulb temperature: the temperature indicated by a properly designed wet-bulb instrument. This closely approximates the thermodynamic wet-bulb temperature (i.e., temperature of adiabatic saturation).

windage: wind-induced loss of circulating water.

2-3 UNITS

All the equations and calculations used in this Code are given in U.S. Customary units. The corresponding values in SI units are calculated by using soft conversion and are given only for information.

Section 3
Guiding Principles

3-1 ADVANCED PLANNING FOR TEST

Plans for the proper locations of test instrument connections should be made so they are provided in the original design of the circulating water system and equipment. Before proceeding to select, construct, install, calibrate, or operate instruments, relevant sections of the PTC 19 Series of Supplements on Instruments and Apparatus, ASME MFC-3M, ASME Fluid Meters, and Cooling Technology Institute (CTI) Bulletins STD-146 and FSP 156 should be consulted for detailed instructions.

The effect of changes in the atmosphere and other extraneous sources require recognition of the changes in performance that will occur. The importance of conducting tests under stable operating and weather conditions cannot be overemphasized. Unstable conditions can be ample reason to disqualify the results of the test.

3-2 AGREEMENTS PRIOR TO TEST

The parties to the test shall reach a definite agreement as to the specific objectives of the test(s), the number of test runs, the method of operation of the equipment, and the instruments and apparatus to be used. Furthermore, this Code encompasses a number of test procedures. The specific scope to be included in the test program should be agreed to prior to the test. The main purpose of this Code is to provide adequate guidance for conducting acceptance tests. The tests shall be conducted in the manner provided herein. Any exceptions shall be by written mutual agreement by the parties to the test. All parties to a test should carefully consider the legal and technical ramifications of including in a contract any proposed language that modifies the provisions of this Code.

If the test involves contractual obligations, authorized representatives of the purchaser, the manufacturer, and/or supplier shall be given adequate notice and the opportunity to be present. To the extent possible, the owner shall ensure that the tower is in a condition to be tested prior to arrival of the test parties on site. The manufacturer and/or supplier shall be given permission and adequate notice to inspect the tower and determine preparations needed for the test. Note that the tower may have to remain in service during the inspection, depending on the purchaser/user's needs. In no case shall any directly involved party be barred from being a test participant or be barred from the test site.

3-3 TEST OVERVIEW

This Code encompasses a variety of cooling tower equipment and also a variety of tests.

Whether the equipment is a natural draft, a mechanical draft or a wet–dry cooling tower, an evaporative cooler, or a wet surface air-cooled steam condenser (WSACC), and whether or not the purpose of the test is to demonstrate plume compliance or water consumption, the thermal capability must be determined at the measured test heat load of the equipment for the particular weather conditions of the test. That measured test thermal performance result is then adjusted by a prescriptive modification of the as-tested parameters of heat load, atmospheric factors, and performance to the design or reference guarantee condition. With respect to plume compliance, the level of operation of the tower is specifically related to the characteristics of the plume produced and therefore its thermal performance must be established.

In broad terms, test measurement of the tower heat load is assessed by an accurate measurement of the water flow to the tower and the temperature change of that water. The atmospheric conditions must be measured to determine the particular impact the weather has on the tower performance or the plume, i.e., the tower approach temperature to the wet-bulb or the exit air characteristics to assess the influence of the related air flow parameters on the entire cooling process. It should be recognized that natural draft tower air flows are particularly sensitive to the weather conditions of the test. All test measurements specified in this Code, other than the following three exceptions, are either associated with the above measurements to ensure accuracy or are optional. The exceptions are:

(a) The wet–dry tower requires an extensive traverse and evaluation of one representative fan stack to determine the exit air characteristics during a plume compliance test.

(b) The evaporative cooler requires measuring and determining the process side fluid flow rate (usually water) and its inlet and outlet temperatures to determine the tower heat load.

(c) The wet surface air-cooled steam condenser (WSACC) heat load is measured and evaluated in accordance with Sections 4 and 5. The WSACC test heat load and steam flow can be established from energy balance methods. Because of the potential uncertainty inherent

in heat balance methods, it is recommended that the testing be at the same time as the test of the steam turbine and in accordance with ASME PTC 6, Steam Turbines. The heat rejected to the WSACC would then be computed by energy balance or calculated from condensate flow measured by calibrated flowmeters. The steamside condenser pressure shall be measured by two basket tips per cell distributed equally. In all other respects, the WSACC test requirements shall be identical to those of an evaporative cooler.

3-4 PREPARATION FOR TEST

Prior to the test, CTI Bulletin FSP-156 should be consulted and the equipment shall be examined and conditions shall be as follows:

(a) The water distribution system shall be essentially clean and free of foreign material that may clog or impede the normal water flow. The water shall be distributed to all operating cells and/or parts of the tower as recommended by the manufacturer. For multicell towers with full cell partitions, one or more cells may be shut down, provided that the circulating water flow to each operating cell is within ±10% of the per cell design specification.

(b) Mechanical equipment, if involved, shall be in good working order, with fans rotating in the correct direction, at the correct speed, and/or pitched to draw within ±10% of the design motor power at design thermal conditions.

(c) Drift eliminators shall be essentially free of foreign material such as oil, tar, scale, algae, and other deposits that may impede normal airflow.

(d) The heat transfer or fill media shall be essentially free of foreign material such as oil, tar, scale, or algae that may impede normal air and water flow, or alter heat transfer characteristics.

(e) Water in the collecting basin shall be maintained within the recommended range of operating level during the test to ensure proper air flow through the fill, i.e., eliminate air bypass in crossflow cooling towers.

(f) The quality or characteristics of circulating water shall be in accordance with para. 3-8(k) of this Code.

(g) Makeup and/or blowdown streams may be stopped prior to testing if other test condition requirements are not adversely affected. If the makeup, blowdown, or other auxiliary stream enter the test control volume such that they impact a measured parameter, then corrections must be made for the influence of the stream. For these instances, the stream flow rate and temperature must be recorded. If makeup is not measured, it can be determined by shutting down the blowdown and calculating the evaporation loss using heat load, water flow, and air quantity and quality.

(h) Prior establishment of cleanliness criteria is recommended for the water distribution system, drift eliminators, and heat transfer media. In the event that the equipment is not in satisfactory operating physical condition, adjustments or changes that will place it in proper operating condition shall be made prior to the test. No adjustments shall be made that are not practical for long-term commercial operation.

(i) For WSACCs, the infiltration of noncondensables shall be limited to the value stated in para. 3-8(m) under stable back pressure conditions.

(j) The dry section of wet–dry cooling towers shall be essentially free of foreign material, both inside and outside, that may impede normal air or water flow or alter heat transfer characteristics.

(k) If applicable, air and water control devices of wet–dry cooling towers shall be set essentially in accordance with the manufacturer's recommendation to achieve the required plume abatement and thermal performance.

(l) For closed-circuit evaporative coolers and WSACCs, any external heat load added to the spray water shall be turned off for the duration of the test.

3-5 DURATION OF TESTING

Crucial parameters must be monitored prior to testing to ensure stable test conditions. The stability of the test conditions and monitored crucial parameters must adhere to the requirements of para. 3-6.

3-5.1 Duration of Analyzed Test Period

After reaching steady state conditions, the requirements for test duration are as follows:

3-5.1.1 Mechanical Draft Equipment. The duration of the test run shall not be less than 1 hr.

3-5.1.2 Natural Draft Towers. To compensate in part for the inability to measure atmospheric lapse rate and vertical wind speed profiles, it is recommended that the duration of testing include a minimum of six nonoverlapping 1-hr periods, when the operating conditions are within the limitations of para. 3-8, and extend over a minimum of a 2-day period. When the atmospheric lapse rate and vertical wind speed profiles are measured and are within Code limitations, fewer test hours are permitted by mutual agreement.

3-5.1.3 Thermal Lag Adjustment. In the event that the thermal lag of the system exceeds 5 min, the duration of the test run shall be not less than the sum of 1 hr plus the thermal lag time.

3-5.1.4 Wet–Dry Towers. After reaching steady state conditions, the duration of one test run shall be the time used to complete the fan effluent traverse or not less than 1 hr.

3-5.2 Selection of Period for Analysis

The time period selected for evaluation shall be one in which the rate of change of temperatures is at a minimum, the scatter of the data with respect to a straight line curve fit is at a minimum, and both the rate of change and scatter are within the limits stated in para. 3-6. This demonstrates that the equipment is at a steady state operating condition.

3-6 STABILITY OF TEST CONDITIONS

Stability of test conditions is critical to the conduct of an accurate test. For a valid test during the selected period, the limitations from maximum to minimum shall be based on the computed average of each interval of the following data. Individual readings may fluctuate during the test. To ensure test stability, the slope of a linear least-squares-regression in the overall average of readings for all stations at each recording interval shall be evaluated as follows:

(a) *Heat load* shall not vary by more than 5% per hour.

(b) *Range* (heat load for WSACC) shall not vary by more than the greater of 5% per hour or 1°F (0.5°C).

(c) Circulating *water flow rate* (process fluid flow rate for closed-circuit evaporative coolers and steam mass flow rate for WSACC) shall not vary by more than 2% per hour.

(d) *Air temperatures* shall not vary by more than specified below (a discussion of the cautions associated with the measurement of inlet air temperature is presented in para. 5-1.1, including the effects of recirculation, external heat sources, and inversions):

(1) wet-bulb temperature: 2°F (1°C) per hour

(2) dry-bulb temperature (if applicable): 5°F (3°C) per hour

To limit scatter, the maximum deviation of the overall average of readings for all stations at one time interval may not exceed the overall test period average by more than the following:

(1) wet-bulb temperature: 3°F (1.5°C)

(2) dry-bulb temperature (if applicable): 7.5°F (4.5°C)

3-7 READINGS

If data acquisition systems are used, it is recommended that continuous reading with averaging over 1-min intervals be used for (a) through (e) and for (r) below. If manually measured, readings of each instrument shall be recorded at a regular frequency not less than the following:

(a) entering wet-bulb temperature: 12 per hour.

(b) dry-bulb temperature: 12 per hour.

(c) cold water temperature: 12 per hour.

(d) hot water temperature: 12 per hour.

(e) wind velocity (speed and direction): continuous recording.

(f) circulating water flow rate (process fluid flow rate for closed-circuit evaporative coolers): 3 single, center point readings when measurement is made by Pitot tube to verify that the flow rate has not changed beyond allowable limits [see para. 3-6(c)]. Other techniques could also be used to check flow stability. One full traverse (along two axes at 90 deg to each other) shall be made either immediately preceding, following, or during the test.

(g) For WSACC, the heat load steam flow and steam quality shall be computed from ASME PTC 6 results for the power cycle.

(h) makeup flow rate and temperature: 2 per hour if flow is not shut off during the test and the makeup stream affects the test result. Alternately, the makeup flow rate may be calculated by the difference between the initial and final readings of a totalizing flow meter or by calculation as described in para. 3-4.

(i) blowdown water flow rate and temperature: 2 per hour (or totalized values per test) if flow is not shut off during the test, and the blowdown affects the test result.

(j) pump discharge pressure(s) required for calculating the pump correction to cold water temperature measurements or for pumping head (if required to verify pumping head guarantee): one before and one after testing.

(k) barometric pressure: 1 per hour.

(l) water (process fluid) sample acquisition: 1 during the test if requested by any party to the test.

(m) fan motor power input: 1 per test (if applicable or required).

(n) exhaust air wet-bulb and dry-bulb temperatures and velocity (wet–dry towers): 1 complete traverse per test, per para. 4-4.5.

(o) spray water flow rate (closed-circuit evaporative coolers): 1 per test.

(p) circulating water pump speed: 1 per hour (if applicable).

(q) for a wet surface air-cooled condenser test the condenser steam-side pressure 12 per hour.

(r) for closed-circuit evaportive cooler heat exchanger pressure drop 1 per test.

3-8 LIMITATIONS

The following variations from design conditions shall not be exceeded:

(a) wet-bulb temperature

(1) ±15°F (±8°C) thermal performance test for all equipment

(2) −0.0°F to +15°F (−0.0°C to +8°C) wet–dry cooling tower (plume compliance test)

(3) minimum allowable wet-bulb shall be 32°F (0°C)

(b) dry-bulb temperature

(1) ±25°F (±14°C) thermal performance test for all equipment

(2) −0.0°F to +25°F (−0.0°C to +14°C) wet–dry cooling tower (plume compliance test)

(c) circulating water flow or process fluid flow (closed-circuit) ±10%

(d) cooling range ±20%

(e) power

(1) fan motor output power ±10%

(2) spray pump input power ±10%

(f) barometric pressure ±1 in. Hg (±3.5 kPa)

(g) heat load ±20%

(h) wind velocity: unless otherwise specified as a design condition in the cooling tower contract, the average wind velocity shall not exceed the following values:

(1) all equipment

(a) average wind velocity: 10 mph (4.5 m/s)

(b) 1-min duration: 15 mph (7 m/s)

For natural draft towers, this shall apply at the top of the tower shell and at the middle of the air inlet height if no specific limits were specified in the purchase contract. If wind conditions cannot be measured at the tower shell 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.

Indications of wind velocity at the tower shell exit may be obtained also by relating the wind velocity measured at a given height to the tower height.

(2) wet–dry cooling tower (plume compliance):

(a) average wind velocity: 6.5 mph (3 m/s)

(b) 1-min duration: 10 mph (4.5 m/s)

For wet–dry cooling towers, the reliability of the fan exit air measurements increases when the wind velocity decreases. The lowest possible wind speed conditions shall be sought for testing.

(i) Atmospheric gradient of temperature (natural draft cooling towers only). The average vertical ambient dry-bulb temperature gradient between the elevation of the center of the air inlet and twice the height of the top of the tower shell, assessed in 200 ft (60 m) maximum increments, shall decrease by at least 3.5°F/1000 ft (0.65°C/100 m) with height (U.S. Standard Lapse rate)

An indicator of the cooling tower vertical ambient dry-bulb temperature gradient shall be the difference in dry-bulb temperature between ground level and the top of the air inlet. For an acceptable test, the average dry-bulb temperature at or near the top of the air inlet shall be more than the average of the dry-bulb temperature measured 5 ft (1.5 m) above grade level.

(j) Atmospheric Conditions. There should be no precipitation during the test.

(k) Water Characteristics. The total dissolved solids in the circulating water shall not exceed the greater of 5000 ppm or 1.1 times the design value.

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

The limits for foreign substances, like surfactants, in the circulating water shall be by prior mutual agreement between the parties to the test.

(l) Operating cells. the circulating water shall be distributed to all operating cells and/or part of the tower in accordance with the operating instructions supplied by the manufacturer/supplier.

For mult-icell towers, one or more cells may be shut down, providing the circulating water flow rate and cooling range limitation are met on a per cell basis.

(m) Limitations on noncondensable gases (for WSACC)

(1) Excessive infiltration of noncondensable gases (primarily air) degrades the WSACC performance and therefore must be maintained within the limits shown in Table 3-8 (U.S. Customary units). It is necessary to verify that all noncondensable gas removal equipment is functioning properly prior to the WSACC performance test. Therefore, prior to the test, the noncondensable gas loading shall be measured to ensure adherence to Table 3-8. Techniques for measuring noncondensable gas loading are given in Interim Supplement PTC 19.5, Part II, *Fluid Meters*, and PTC 19.5.

(n) Wet–Dry Cooling Tower (Plume Compliance). The test will be performed on one cell of the entire cooling tower. If no single cell may be considered representative of the entire tower, the test shall be conducted on several cells. In that case, the final performance result will be the weighted average of the individual performance results by the number of cells having the same design as the cell tested.

3-9 MANUFACTURER'S PERFORMANCE CURVES

All performance curves shall cover the operating conditions defined in para. 3-8, or those conditions specified by the contractually responsible parties. Performance curves shall be plotted on a convenient and consistent scale readable to the nearest 0.1°F (0.05°C); rectangular coordinates are preferred. Performance curves prepared for wide range of water flow rates are recommended to facilitate extrapolation for thermal performance at off-design operating conditions.

3-9.1 Mechanical Draft Towers and Closed-Circuit Evaporative Coolers

The manufacturer shall submit three sets of performance curves based on constant fan blade pitch angle and constant fan speed covering circulating water (fluid) flow rates of 90%, 100%, and 110% of design rate. Each of the sets shall show cold water (fluid) temperature as ordinate and wet-bulb temperature as abscissa with lines of constant cooling range as a parameter. The cooling ranges shall cover 80% to 120% of the design range and shall be shown in uniform increments. The design thermal point shall be shown on the 100% water (fluid)

Table 3-8 Noncondensable Gas Load Limits

U.S. Customary Units			SI Units		
Number of Steam Turbine Exhausts	Total Exhaust Steam Flow to WSACC, lb/hr	Noncondensable Gas Load Limit, scfm	Number of Steam Turbine Exhausts	Total Exhaust Steam Flow to WSACC, kg/sec	Noncondensable Gas Load Limit, scm/hr
1	Up to 100,000	1.0	1	Up to 15	2.0
1	100,000–250,000	2.0	1	15–30	3.0
1	250,000–500,000	2.5	1	30–60	4.0
1	500,000–1,000,000	3.0	1	60–125	6.0
1	1,000,000–2,000,000	3.75	1	125–250	8.0
1	2,000,000–3,000,000	4.5	1	250–375	
2	200,000–500,000	3.5	2	25–60	6.0
2	500,000–1,000,000	4.0	2	60–125	7.0
2	1,000,000–2,000,000	6.0	2	125–250	10.0
2	2,000,000–4,000,000	7.5	2	250–500	12.0
2	4,000,000–6,000,000	8.5	2	500–750	15.0

flow rate curve. See Nonmandatory Appendix E and Nonmandatory Appendix H.

3-9.2 Natural Draft Towers

The cooling tower manufacturer shall submit a minimum of 9 sets of performance curves. Each set shall show cold water temperature as ordinate and wet-bulb temperature as abscissa with lines of constant relative humidity as parameter; i.e., 25%, 40%, 65%, and 100% are recommended. The performance curves will include three main groups at 90%, 100%, and 110% of design water flow rate. A minimum of three sets of curves within each water flow rate group shall be for a constant cooling range including 80%, 100%, and 120% of design cooling range. The design thermal point shall be shown on the appropriate curve of Nonmandatory Appendix F.

3-9.3 Wet Surface Air-Cooled Condensers (WSACC)

The manufacturer shall submit three sets of performance curves based on constant fan blade pitch angle and constant fan speed covering steam mass flow rates of 90%, 100%, and 110% of design rate. Each one of these curves shall be presented as a family of curves expressing steam vapor quality (80%, 90%, and 100%). Each set of curves shall show absolute pressure as ordinate and wet-bulb temperature as abscissa. See Nonmandatory Appendix I.

3-9.4 Plume Compliance

The tower manufacturer shall submit a family of plume characteristic performance curves that relate the pertinent performance variables. Performance curves shall consist of 9 sets of curves. These sets shall apply to 80%, 100%, and 120% of design range for each of 90%, 100%, and 110% of the design circulating water flow. Each set shall consist of four or more relative humidity curves. The full set of curves are arranged to show the effects of air temperature, cooling range, and circulating water flow rate on the plume.

Section 4
Instruments and Methods of Measurement

4-1 GENERAL REQUIREMENTS

This Code presents the mandatory requirements for test instrumentation and its use.

The Instruments and Apparatus Supplements (ASME PTC 19 Series) outline the governing requirements for all ASME performance testing. Before proceeding to select, construct, install, calibrate, or operate instruments, relevant sections of that PTC 19 Series, ASME Fluid Meters, relevant PTC codes cited within this Section, and the Cooling Technology Institute STD 146 should be consulted for detailed instructions.

Achievement of the required accuracy for each parameter measured is the single most important criterion in the selection of an appropriate method of measurement. This Code shall not be construed as preventing the use of advanced technologies or methods of measurement not described herein, provided that the accuracy requirements of para. 5-12 are achieved by the alternative method. Note that plant instrumentation is acceptable only if it can be demonstrated to meet the overall uncertainty requirements. If instruments or procedures other than those prescribed are used, they must be mutually agreed upon by the authorized representatives of the parties to the test, prior to the test. Any departure from the prescribed methods shall be described in the test report.

It is highly recommended that provisions for cooling tower testing, particularly the flow measurement, be incorporated into the design of the facility at which the cooling tower is located. Back-fitting an existing system for the required measurements can be very expensive and time-consuming and in some instances can be impossible.

All flow measuring devices, temperature sensors, and electric power meter test instrumentation shall be calibrated against reference standards traceable to NIST (National Institute of Standards and Technology) or recognized physical constants. Note that where this Code refers to NIST standards and calibrations, those of other nations' equivalent standards laboratories may be used as appropriate for the locale of the testing. Test variables of a secondary nature such as wind speed and direction devices need not be calibrated against a reference standard, but instead can be calibrated against other calibrated instruments or transfer standards, or can be checked in place with two or more instruments at the same location measuring the same variable.

Instruments shall have been calibrated and inspected in accordance with accepted engineering practice. Instrument calibration should be prepared in advance of the test. At the request and cost of the requesting party, a post-test calibration can be performed. Specifically, temperature sensors shall be calibrated within 3 months prior to use; flow-measuring devices shall be calibrated at least every 3 years if undamaged; electric meters and wind speed and direction devices shall be calibrated yearly. Before testing, but after the on-site wiring connections are made, there shall be sufficient comparisons of all like temperature sensors to ensure their relative accuracy.

There shall be a written procedure for the calibration of each type of instrument. Records shall be maintained showing the latest calibration of each instrument and make them available upon request. Calibrations should encompass the expected measurement range and comprise at least two points more than the order of any calibration curve fits. The test report shall include the individual identification and location for each instrument used on the test so that calibration history can be traced.

4-2 MEASUREMENT OF WATER FLOW

The rate of circulating water flow into the cooling tower and or individual cells is required to assess the performance of wet cooling towers and wet–dry towers. To test a closed-circuit evaporative cooler, in which the process fluid to be cooled is circulated inside the tubes/plates of a tube bundle/heat exchanger, the Main Circulating Flow Rate shall be understood to be the flow rate of the process fluid within the closed circuit. The process fluid within the closed-circuit can be any chemical element, compound or mixture, liquid or gas, for which physical properties are known, in single-phase flow. The external spray water flow rate is considered a secondary parameter in this case.

Wet-dry towers and evaporative coolers may require the flows defined for only one representative cell. Measurements of the makeup water flow rate, blowdown flow rate, dry tube bundle flow rate and other flows may also be required depending on the type of test and mode of operation.

To ensure the highest level of measurement accuracy, it is important that consideration be given to flow measurement at the design stage of any specific project and a suitable accurate method and location for measurement be agreed upon by the purchaser and manufacturer.

It is highly desirable that the flow devices be installed at points in the circulating water system where a fully developed velocity profile exists; e.g., distortions of the velocity traverse, helical swirls, or vortices should be minimized. The accuracy of any flow measurement, regardless of metering instrumentation, may be achieved as much by measurement location as by the degree of perfection of manufacture or the characteristics of the device.

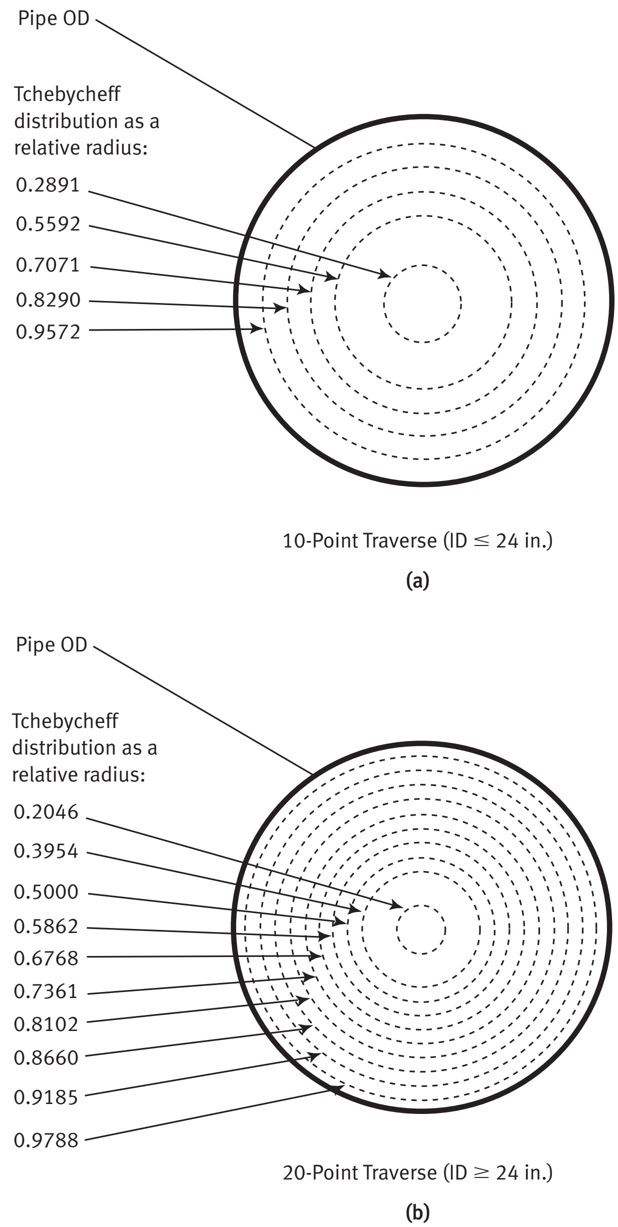
The methods described in paras. 4-2.1 through 4-2.4 of cooling water flow measurement in conduits that typically serve cooling towers can be capable of a test uncertainty of ±2% to ±4% when properly implemented. The velocity traverse or differential producer methods are also useful to accurately measure the water flow in smaller lines such as those that serve individual wet cooling tower cells, a dry section tube bundle of a wet–dry tower, or an evaporative cooler's spray water or the closed circuit cell flow. Due to the nature, variation, and many accurate measurements that are required, using the energy balance methods to determine the flow rate is not recommended except when performed concurrently with an ASME PTC 6 test. The use of pump curves or other methods not discussed herein could result in inaccurate determination of the water flow. Dye dilution has been successful in some installations, particularly once through cooling tower systems, but it has been unsuccessful in some recirculating systems. Ultrasonic time-of-travel meters have performed satisfactorily in metal piping systems, but have proven less reliable in certain piping systems with coated or nonmetallic pipes. Velocity traverse with Pitot tubes and differential producers are recommended.

Measurement of the circulating water flow rate shall be made in the piping leading to the tower or in its individual operating cells. Where undesirable conditions for flow measurement exist in the hot water piping to the tower, due either to an inaccessible location or one that is likely to contain distorted velocity profiles, measurements may be made in the equipment return piping from the tower. Because the desired test parameter is water delivered to the tower, corrections shall be made to the measured flow rate if the measurement is impacted by flow streams (e.g., makeup or blowdown) that are not reflected in the water flow at the flow measurement point.

Considerations concerning the application of these methods to cooling towers are covered in this Section.

4-2.1 Velocity Traverse Methods

Velocity traverse methods are generally most applicable to cooling towers since they may be used to measure flow in a relatively large range of conduits and the size



GENERAL NOTE: Each point is to be equally weighted.

Fig. 4-2.1 Recommended Velocity Traverse Probe Positions

of the traverse probe will not affect the measured flow. PTC 19.5 describes the general considerations for acceptable velocity traverse methods. For the purposes of PTC 23, the Pitot tube is recommended, and is the most common type of instrument employed.

If a Fechheimer probe is used, it is recommended that the directional sensing capabilities of the probe be used to determine the local flow angle, and that this flow angle be incorporated into the flow determination. Due to the probe geometry, it can only be used for a 10-point traverse [see Fig. 4-2.1(a)], regardless of pipe size.

Traverse probes shall be inspected and calibrated by a hydraulic laboratory to an uncertainty of ≤2.5%, using standards traceable to the NIST or a primary physical standard and inspected by the parties before the test. Calibrations shall cover the range of probe Reynolds numbers of the average pipe velocity expected in the flow measurement.

Differential pressure measuring devices used for either the Fechheimer or Pitot traverse probe impact and static pressure measurements shall be calibrated before the test to an accuracy of at least ±0.25% of the maximum differential pressure expected. Mechanical gauges, manometers, and electronic differential pressure transducers are acceptable, provided the stated accuracy requirements are met. However, an air-over-water, inverted type of manometer is recommended.

The velocity probe should be inspected for damage periodically during testing. If any damage to the probe is noted, all measurements made since the previous inspection shall be repeated with another calibrated probe.

A velocity probe may experience vibration under certain flow conditions. If such vibration is detected, the cause of the vibration shall be corrected, if possible. Individual point measurements taken under conditions of significant probe vibration may occur and shall be considered in the evaluation of the uncertainty of the measurement. Indicators of probe vibration include a sharp change (usually a rise) in the probe differential when the probe position is changed slightly, physical vibration of exposed part of the probe, or a relatively sudden onset of a high level of periodic pressure pulsation.

The traverses shall be made immediately preceding the test, during the test, or following the test. The Pitot taps should be installed such that the velocity profile is well defined at the measurement point. At least 10 diameters of straight, unobstructed piping is recommended upstream of the measuring station with at least 5 similar diameters of length downstream. Traverses shall be taken along two diameters, spaced 90 deg apart. The volume flow rate is determined by integrating velocities measured at a number of points in a plane perpendicular to the water direction. For this reason, accurate measurement of the pipe inside diameter, pipe shape (out-of-roundness), and location of the probe tip is extremely important. In addition, response times for specific tube designs can vary significantly and therefore attention should be given to being sure that equilibrium has been reached at each measurement point. The traverse locations shall be based on equal area weighting method or the Tchebycheff weighting scheme described in PTC 19.5. A 10-point diametrical traverse shall be used for pipes of 24 in. ID or less; a 20-point diametrical traverse shall be employed for larger diameter pipes. An example of acceptable traverse points for each is shown in Fig. 4-2.1.

(a) Pipe Internal Diameter. Internal pipe diameters are critical to the application of the traverse method. Errors in determination of the pipe internal diameters affect both the positions of the traverse locations and the area used to determine total water flow. Errors in area directly affect the calculated result for tower performance.

(b) Measurement of Internal Pipe Diameter. Another procedure is measurement of each pipe internal diameter using a device that can be deployed through the valve, as with the Pitot tube. Such a device would be deployed from far side to near side as nearly normal to the pipe centerline as possible, and the difference in insertion measured on the external portion of the device. Alternatively, the Pitot tube itself can be used to approximate this measurement by insertion to the far side of the pipe, and, while reading deflection on the manometer; retraction until the deflection exactly reaches zero. The difference in insertion is the pipe internal diameter. It is important to note that if the pipe fitting for the tap location is welded or otherwise attached such that it is not flush with the inside of the piping, an error in internal pipe diameter can result as the zero deflection point will not be at the pipe wall. If the taps are skewed with respect to the true pipe diameter, a dimension either greater or smaller than the diameter is possible. Internal diameter measurements shall be used for calculation of the traverse points, in any event.

(c) Determination of Internal Pipe Diameter by Calculation. If for some reason the internal pipe diameter can't be measured directly as above, nominal values may be used or the internal diameter can be determined by calculation. If the wall thickness of the piping is known from pipe drawings, the circumference can be measured and the internal diameter calculated by subtracting double the wall thickness from the external diameter determined from the measured circumference. The uncertainty of this method is very high. In new and clean installations the calculation method can yield acceptable results but adhering to the prescribed uncertainty requires verification of piping internal cleanliness and physical dimensions at the measurement point.

4-2.2 Dye Dilution Method

Applying the dye dilution method requires that the parties to the test agree upon the details of its implementation. It should be noted that because of the constantly rising background concentration, there is no industry consensus that the dye dilution method can be accurately applied to closed-circuit cooling towers. The method is best suited for once-through cooling circuits where background concentrations remain constant. Refer to PTC 19.5 and Nonmandatory Appendix C of this Code for sample dye dilution field procedure.

Key requirements are:

(a) Complete mixing of the tracer must be achieved for an accurate flow measurement. As a general guideline, 100 diameters of pipe are recommended between

the injection and sampling points. However, turbulence producers (e.g., pumps, bends, etc.) can reduce the number of pipe diameters required for complete mixing.

(b) The background concentration of the measured water cannot be altered by the recirculation of the injected dye.

(c) When the background concentration begins to rise, it is recommended that the testing be stopped.

(d) If the method must be used with a rising background concentration, a method to properly correct for the condition must be determined and agreed upon.

(e) Care must be taken to account for temperature effects if a temperature change occurs between the injection and sample points. For Rhodamine WT dye, the temperature correction of various samples may be made to a common temperature, using the following equation (see Smart and Laidlaw in Appendix M):

$$F_s = F_\theta{}^{-0.027\,(T_S - T_W)}$$

where

F_S = the fluorescence at standard temperature, T_S, °C

F_θ = the measured fluorescence at temperature T_W, °C corrected for background and instrument offset

(f) The dye shall exhibit minimal tendency to adsorb into organic or inorganic surfaces.

(g) The flow should be free of any chemicals (e.g., chlorine) or silt concentrations that can affect the ability to accurately measure the concentration of dye.

(h) If the mass or volume of the injected dye is not directly measured during the test, the injection apparatus shall be calibrated for injection flow with water from the system to be tested.

(i) The fluorometer or other concentration device shall be calibrated before the test with a minimum of three calibration solutions made with the system water that bracket the expected dye concentration.

(j) The chlorine injection system must be placed out of service several hours prior to testing. Chlorine and other chemicals may consume dye, affecting measured results.

4-2.3 Differential Producers

Devices that are left in the flow path for a period of time may not retain the ability to accurately measure the flow. Differential producers, which are recommended by this Code, include the orifice plate, the flow nozzle, and the venturi meter. Because of energy loss considerations, these devices will be most applicable to smaller cooling tower installations, i.e., those with circulating water system piping of less than about 3 ft (1.5 m) diameter.

The installation of the differential producer shall follow the requirements of PTC 19.5, particularly with regard to the length of upstream and downstream piping, construction and finish of the flow element, and location and finish of the piezometer taps. The flow element, together with any flow conditioning devices immediately upstream of the element, shall be calibrated as a unit by a hydraulic laboratory to an uncertainty of ±1% or better, using NIST traceable methods.

4-2.4 Ultrasonic Time-of-Travel

Flow may be measured using the multiple-path, time-of-travel, ultrasonic flow meter. An ultrasonic clamp-on transducer can be used provided that it complies with all requirements to follow. It is emphasized that the time-of-travel method is very different than the Doppler technique. The Doppler type instruments will not provide sufficient accuracy to satisfy the requirements of this Code.

The ultrasonic time of transit measurement shall be made immediately preceding the test, during the test, or immediately following the test. The internal pipe diameter is critical to the application of the ultrasonic method. Errors in determination of the pipe internal diameters affect the area used to determine total flow and that directly alters the calculated result for tower performance. The provisions given in PTC 19.5 shall be used as a guide in the application of this method. Because of the constantly improving technology in the ultrasonic field, the final application must be based on a combination of the requirements listed in Nonmandatory Appendix C, in the equipment manufacturer's recommendations, in PTC 19.5, and in the guidelines in PTC 18.

Experience in cooling tower testing has shown that the ultrasonic instrument and its specific application to a particular test may have a measurement inaccuracy which exceeds that required by this Code; therefore, careful consideration must be taken when using the acoustic method corresponding to the repeatability and accuracy of the measurement. All instruments and sensors used must be NIST traceable. The meter and transducers shall have been calibrated in similar conditions to that of the test in question. The similarity shall include geometry modeled and Reynolds Number range.

In addition, if the method is proposed, the calibration data and/or previous comparative experience of the particular meter shall be required to be provided as evidence of its measurement accuracy for the specific test conditions prior to its acceptance and installation for a test. That calibration information and data shall become part of the test report.

No less than two chordal or diametric paths shall be measured regardless of the size of the conduit. Wherever possible, the (location of the transducers) metering section shall be preceded by at least 20 diameters and followed by a minimum of 10 diameters of straight pipe. As a minimum, at least 10 diameters of straight pipe upstream and 5 diameters downstream shall be provided without any obstructions at either end such as an open butterfly valve.

The time-of-travel instrumentation shall be calibrated at the zero flow condition before and after the test to ensure proper and accurate operation. Zero flow calibration must be conducted in a full pipe condition only. To further verify accuracy during the calibration, the acoustic velocity should be measured with the acoustic device and compared with the published values for the speed of sound in the process fluid. Any discrepancies must be investigated.

Because this method may be adversely affected by the presence of silt and other particulate or air bubbles in the flow, it should also be verified that the source water is suitably clean before the metering system is installed.

Since permanent ultrasonic time-of-travel systems may be relatively difficult and expensive to install on large diameter conduits, it is recommended that provision for their installation be made during design and construction of the cooling system. At the time of installation, the true diameter of the conduit cross-sectional area shall be determined.

4-3 MEASUREMENT OF WATER TEMPERATURE

Water temperature measurements for cooling towers, or process fluid temperature measurements for closed-circuit evaporative coolers, or WSACCs, shall be made with instruments having a maximum uncertainty of ±0.2°F (±0.1°C). Several instruments capable of achieving this accuracy are suitable for use in cooling tower tests, such as resistance-temperature devices (RTDs), thermistors, and liquid-in-glass thermometers. Mercury thermometers are not recommended because of the potential environmental hazards posed by the mercury in case of breakage. The general procedures given in PTC 19.3, Temperature Measurement, should be followed for their use.

All temperature measuring devices shall be calibrated to within ±0.1°F (±0.05°C), using NIST traceable standards following the general procedures given in PTC 19.3. A minimum of five calibration points covering the expected range of temperatures shall be taken. Before testing, but after the wiring connections are in-place, sufficient comparisons of all water temperature sensors shall be made to verify their proper functioning.

4-3.1 Instruments

100-ohm platinum RTDs are recommended. Any thermistor with nominal impedance of greater than 10,000 ohms at 32°F (0°C) is acceptable. For all resistance devices, a 4-wire measurement arrangement is recommended. For thermistors with 2-wire arrangement, the wiring resistance measured on site prior to the test will be subtracted from the probe resistance reading. The uncertainty of this thermistor reading will be higher than a comparable RTD reading.

Thermometers, if used, should be of the total immersion type with etched stems.

Thermometers should have clearly readable graduation increments of 0.2°F (0.1°C). If the thermometer was calibrated for total immersion, an emergent stem correction factor as described in PTC 19.3 should be applied. The thermometer should be isolated from heat sources, and be well illuminated. The thermometer should be inspected before and after the tests to ensure that it is in good physical condition, with no breaks, cracks, or liquid separation.

4-3.2 Location of Measurement Points

4-3.2.1 Hot Water Temperature. The sensing elements shall be located so that the true weighted average temperature of the water/process fluid entering the cooling tower can be determined from the data. Because the water delivered to the cooling tower cell/tube bundle inlet is generally well mixed, a minimum of two temperature-measuring devices is necessary to measure the hot water temperature. One sensor acts as insurance for the other. Normal locations for the sensing elements are the common supply conduit or riser or inlet flume at the point of entrance. If the water enters as two or more separate streams of different temperatures, the temperature and flow rate of each stream shall be measured. Or, if the measurement point is located downstream of the mixing point of these streams, the sensing elements must be located such that the temperature of the delivered stream is well mixed. Measurements may be taken by inserting the temperature measuring devices directly into the flow, measuring the temperature from a flowing bleed stream, or from insertion in a thermowell. Care should be take to ensure that flow from a flowing bleed stream does not splash onto the exposed stem of the thermometer or sensor as evaporative heat transfer will then reduce the apparent temperature. If a thermometer is used in a well, the well should be clean and filled with a suitable heat transfer liquid such as ethylene glycol to sufficiently cover the bulb. Insulation should be used around the thermowell and probe to minimize heat exchange with the environment. Thermowell length should be sufficient for the measurement to reflect the bulk fluid temperature. It is cautioned that if a thermowell is used, it must be able to rapidly reflect any changing temperature that may occur during the test. For additional guidance, refer to PTC 19.3.

4-3.2.2 Cold Water Temperature. The cold water temperature must indicate the true average temperature of the water/process fluid leaving the equipment/tube bundles. Care must be taken to either avoid or account for any stratification at the point of measurement. For most installations, the most suitable location for the overall cold water temperature measurement will be in a flowing bleed stream or thermowell at the discharge of the circulating water pumps where the water is well mixed. All the precautions in para. 4-3.2.1 above should be followed. If the cold water temperature is measured on the discharge side of the pump(s), the measured temperature must be corrected for the heat effects of pump inefficiency, pump pressure, or throttling in accordance with Section 5 of this Code.

In some cases it may be necessary to measure the cold water temperature in an open channel, weir, or other conduit in which the water is discharged from the cooling tower by gravity. These tests must be conducted with a data acquisition system utilizing an equal area cross-sectional array of submersible probes. In extreme cases where the cold water temperatures and the flows are obviously non-uniform, flow measurements made through each of the equal areas should be used to flow weight the individual temperature stations. Reference should be made to applicable sections of Part I of Fluid Meters, PTC 19.5, and PTC 18, Hydraulic Turbines, for more information on the methods of open channel flow rate measurement.

Adjustments to the measured overall cold water temperature will depend on the locations of the measurement stations and the type of test operation as follows:

(a) Test run without makeup to the basin and without blowdown. In this mode the makeup and blowdown flows are stopped during the test. The mixed temperature in the basin or discharge conduit is the cold water temperature. The basin water level will drop and the dissolved solids in the circulating water will increase. The effects of the decreased volume shall be checked during the test and at its conclusion to confirm that none of the Section 3 limitations of flow rate, cooling range, or concentration of dissolved solids has been exceeded.

(b) Test run with makeup or other heat sources into the basin and with blowdown removal from the basin. In this mode the measured cold water temperature must be adjusted by a heat balance around the basin, taking into account the temperatures and flows of the makeup and blowdown and any other sources of heat added between the tower and measuring point. The temperatures of the makeup and blowdown streams shall be taken as near to the tower as possible. In the absence of a reasonable means of determining the makeup flow, it may be calculated from the estimated tower evaporation plus the blowdown, provided that there are no other significant system losses. For closed-circuit evaporative coolers or WSACCs, the adjustment for heat added or removed by makeup and blowdown shall be applied to the process fluid flow rate as provided in para. 5-10.2.

4-4 MEASUREMENT OF AIR TEMPERATURE

The measurement of inlet air 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 forced draft design in order to determine the fan inlet air density for fan power correction.

4-4.1 Instruments (Wet-Bulb)

The recommended instrument for measurement of wet-bulb temperatures is a mechanically aspirated instrument or psychrometer that meets the following requirements:

(a) The temperature-sensitive element shall be calibrated to within ±0.10°F (±0.05°C), using NIST traceable standards following the general procedures given in PTC 19.3. The calibration shall be at least beyond ±10°F (±5.5°C) of the expected temperature extremes during the test. Before testing, but after the wiring connections are in place, sufficient comparisons of all air temperature sensors shall be made to verify their proper functioning.

(b) Temperature-sensitive elements shall be shielded from the sun or from any other source of radiant heat. The innermost shield shall be essentially at the dry-bulb temperature.

(c) The temperature-sensitive element shall be covered with a wick that is continuously fed from a reservoir of distilled or demineralized water.

(d) The temperature of the water used to wet the wick shall be at approximately the wet-bulb temperature being measured. This may be obtained in practice by allowing adequate ventilated wick between the water supply and the temperature-sensitive element.

(e) The wick shall fit snugly over the temperature-sensitive element and extend at least one inch over the active portion of the sensor. The wick shall be kept clean and free from contaminants that would inhibit its wetting ability or change the partial pressure of the water.

(f) The air velocity over the temperature-sensitive element shall be between 950 fpm and 1050 fpm (4.8 m/s and 5.3 m/s).

4-4.2 Instruments (Dry-Bulb)

The dry-bulb temperature, where applicable, shall be measured with a mechanically aspirated instrument that meets the requirements of wet-bulb instruments less the wick and water supply.

(a) For the measurement of inlet dry-bulb temperature, instrument location, the number of stations, the frequency of readings, and the reading and averaging procedure shall be as described for wet-bulb temperature measurement.

(b) For natural draft cooling towers, the inlet dry-bulb temperature is very important because it affects the inlet air density, which determines the driving force that

moves the air through the tower. The following are the dry-bulb temperature measurement requirements for natural draft cooling towers:

(1) For the direct measurement of natural draft cooling tower vertical ambient dry-bulb temperature gradient, the plan location of the instruments shall neither be downwind of nor within the influence of the cooling tower nor any other site source of turbulence or thermal gradients that might substantially affect the data. Instrumentation location and techniques will be site-specific and shall be based upon mutual agreement prior to testing.

(2) 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 5 ft (1.5 m) 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. An attempt shall be made to place the instruments in locations not subject to radiation or convective air heating effects due to solar heating of the shell.

4-4.3 Inlet Air Temperature Measurement Locations

4-4.3.1 Location of Wet-Bulb Instrumentation

(a) Air Inlet Wet-Bulb Temperature. For the measurement of inlet wet-bulb temperature, the instruments shall be located no more than 5 ft (1.5 m) outside the air intake(s). Care should be taken to ensure that splashing at the air inlet does not affect the instruments. A sufficient number of measurement stations shall be applied to ensure that the test average is an accurate representation of the true average inlet wet-bulb temperature. The number of instrumentation stations is determined as follows by tower type. An instrumentation grid can then be developed for location of wet-bulb stations on the air inlet of the tower.

(b) Minimum Number of Locations for Each Tower Face. The following is the *minimum* total number of wet-bulb instrument stations for each face (rounded up to the next whole number):

Mechanical draft cooling tower:	$N_{WB} = 0.65\ (A)^{0.33}$
Wet-dry cooling tower:	$N_{WB} = 0.65\ (A)^{0.33}$
Closed circuit:	$N_{WB} = 0.65\ (A)^{0.33}$
WSACC	$N_{WB} = 0.65\ (A)^{0.33}$
Round or polygonal mechanical or fan-assisted natural draft cooling tower:	$N_{WB} = 0.65\ (A)^{0.4}$
Natural Draft Tower:	$N_{WB} = 12$, $Z_{ai} \le 50$ ft (15 m)
	$N_{WB} = 16$, $Z_{ai} \ge 50$ ft (15 m)

where

A = area of total air inlets, m^2

N_{WB} = minimum number of wet-bulb instruments

(c) Horizontal Levels. Wet-bulb instrument stations should be located at the intersection of vertical and horizontal grid lines determined from the total number of wet-bulb stations above and the vertical air inlet height. The number of horizontal levels (N_{GL}) is determined from the following guideline:

Number of Horizontal Grid Levels, N_{GL}	For Air Inlet Height
$N_{GL} = 1$	≤ 15 ft (4 m)
$N_{GL} = 2$	> 15 ft (4 m) ≤ 30 ft (8 m)
$N_{GL} = 3$	> 30 ft (8 m) ≤ 50 ft (15 m)
$N_{GL} = 4$	≥ 50 ft (15 m)

The horizontal grid lines are to be located based on the following formulae:

Number of Horizontal Grid Levels, N_{GL}	Height of Grid Levels(s), Z_{GL1}
$N_{GL} = 1$	$Z_{GL1} = Z_{ai} \cdot 0.5$
$N_{GL} = 2$	$Z_{GL1} = Z_{ai} \cdot 0.25$
	$Z_{GL2} = Z_{ai} \cdot 0.75$
$N_{GL} = 3$	$Z_{GL1} = Z_{ai} \cdot 0.167$
	$Z_{GL2} = Z_{ai} \cdot 0.5$
	$Z_{GL3} = Z_{ai} \cdot 0.833$
$N_{GL} = 4$	$Z_{GL1} = Z_{ai} \cdot 0.125$
	$Z_{GL2} = Z_{ai} \cdot 0.375$
	$Z_{GL3} = Z_{ai} \cdot 0.625$
	$Z_{GL4} = Z_{ai} \cdot 0.875$

where

Z_{ai} = air inlet height

$Z_{GL1} \ldots Z_{GL4}$ = height of each horizontal grid line

(d) Vertical Grid Strings. The number of equally spaced vertical grid strings (N_{GS}) is then determined from the following formula:

$$N_{GS} = N_{WB}/N_{GL}$$

(e) Position of instruments in equal area sections. If practical, the air temperature shall be measured at the central line of equal-area air inlet sections. In case of wet–dry cooling towers, the instruments shall be located both in front of wet and dry section air inlets, treated as separate faces.

4-4.3.2 Location of Inlet Air Dry-Bulb Instrumentation. For the measurement of inlet dry-bulb temperature on wet–dry, natural draft, and fan-assisted natural draft towers, the instruments shall be located on the same basis as for the wet-bulb temperature.

4-4.4 Air Upwind Wet-Bulb and Dry-Bulb Temperature for Plume Compliance for Wet-Dry Cooling Tower

The air upwind wet-bulb and dry-bulb temperatures are used for wet–dry cooling tower plume evaluation. It is the same as the inlet wet and dry bulb temperature measurements located at the immediate upwind faces of the tested cell of the cooling tower. Each row of a multitower installation must be tested separately.

4-4.5 Air Exhaust Wet-Bulb and Dry-Bulb Temperature for Plume Compliance for Wet-Dry Cooling Tower

Measurements of the exhaust air wet-bulb and dry-bulb temperatures are required to determine the characteristics of the effluent air of the cooling tower. The measurements shall be taken at the discharge plane of the fan stack or, if a forced draft wet–dry cooling tower, at the discharge plane area. These measurements must be performed with a remote sensing data acquisition system.

The psychrometers shall be designed to prevent any water droplets from impinging on the sensing element.

In order to avoid wind effects, the measurement shall be taken at the centers of equal areas of the fan stack in a plane within 2 ft (0.5 m) from the cooling tower exhaust plane, if this can be done safely. (It is dangerous to locate the psychrometer too close to the fan rotating plane). The measurements are to be taken at 20 equal area points in the sample plane. The size of the instrument may preclude measurement at the outermost stations.

For small stacks, up to 16 ft (5 m) in diameter, the number of measurement points may be reduced by mutual agreement between the parties, with a minimum of 2 points on each of the 4 radii for a total of 8 sampling points.

If the sampling plane is located within one half of the fan diameter of the fan, the sampling locations should be adjusted for the effect of the fan hub or seal disk.

For circular sample planes without fan hub effects, the sample locations will be based on the total area and located at the centers of equal annular sample zones with 4 radii and 5 points per radius.

For circular sample planes with fan hub effects, the sample location will be based on the net area and located at the centers of equal annular sample zones with 4 radii and 5 points per radius. For small stacks, 8 points may be used.

For all circular planes, the sample locations can be calculated by:

$$X_m = \frac{D_s}{2} - \sqrt{\left[\left(\frac{2M - 2m + 1}{8M}\right)(D_s^2 - D_h^2)\right] + \left(\frac{D_h}{2}\right)^2}$$

where

D_h = hub diameter
D_s = stack diameter
M = number of sampling points on a single radius
m = sampling point number
X_m = sample location, distance from wall

For rectangular sample planes, the sample locations will be at the centers of a matrix of equal area sample zones of similar length and width.

4-5 WIND VELOCITY

Wind speed and direction 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 air inlet approach velocity. Care shall be taken to ensure that wind speed and direction are representative of conditions affecting the tower. Placement of the wind measurement device shall be subject to mutual agreement by the parties to the test. Wind direction shall be recorded in compass degrees with the tower orientation and reference north clearly indicated.

4-5.1

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

4-5.2

In the absence of a direct measurement of the wind speed at the exhaust plane of a natural draft cooling tower, the wind near the ground is measured and the upper level winds are calculated. The wind station shall be placed a minimum of 100 ft (30 m) from the tower with the sensing element a minimum of 10 ft (3 m) above grade. The following procedure shall be applied to estimate the upper wind speed.

Upper level wind velocities may be approximated by the following equation (see Wark and Warner in Nonmandatory Appendix M):

$$\frac{u}{u_1} = \left(\frac{z}{z_1}\right)^p$$

where

p = coefficient as function of Pascal (atmospheric) stability
u = wind speed at target elevation
u_1 = wind speed at known elevation

The exponent p may determined according to Table 4-5.2.

Table 4-5.2 Upper Level Wind Exponent

Surface Wind at 10 m, m/s	Day			Night	
	Incoming Solar Radiation			Cloud Cover	
	Strong, Class 1 [Note (1)]	Moderate, Class 2 [Note (2)]	Slight, Class 3 [Note (3)]	Mostly Overcast, Class 4 [Note (4)]	Mostly Clear, Class 5 [Note (4)]
< 2	0.111	0.111	0.111	0.198	0.333
2–3	0.111	0.111	0.127	0.198	0.333
3–5	0.111	0.127	0.127	0.143	0.198
5–6	0.143	0.143	0.143	0.143	0.143
> 6	0.143	0.143	0.143	0.143	0.143

GENERAL NOTE: Incoming solar radiation of 0.143 should be assumed for overcast conditions during day or night; 0.111 is associated with unstable atmospheric conditions; and 0.333 is associated with the most stable atmospheric conditions (see Wark and Warner, and Turner in Appendix M).

NOTES:
(1) Clear skies, solar altitude greater than 60 deg above the horizontal, typical of a sunny summer afternoon. Very convective atmosphere.
(2) Summer day with a few broken clouds.
(3) Typical of a sunny fall afternoon, summer day with broken low clouds, or summer day with clear skies and solar altitude from only 15 deg to 35 deg above horizontal.
(4) Can also be used for a winter day.

4-6 TOWER PUMP HEAD

Tower pump head is the sum of the tower static head, velocity head, and piping losses. Piping losses will vary depending on the type of tower as well as the terminal point (or reference point) of connection to the tower inlet.

On mechanical draft towers, the terminal point (i.e., scope of supply) of connection is typically the centerline of the inlet piping connection. On natural draft towers, the terminal point of connection is typically the inlet connection to the tower riser at the base of the tower. System head includes the following components:

(a) Static head (H_s). Also known as geometric head, the static head is the vertical distance from the top of the basin wall (curb) to the centerline of the inlet piping to the tower [ft (m) of water].

(b) Nozzle Loss (H_n). Static pressure [ft (m) of water] necessary for compensation of nozzle pressure losses. This head loss should be the minimum head required to ensure proper piping and valve losses [ft (m) of water] from the terminal point of connection through the entire tower distribution system. On natural draft towers, H_f typically includes riser friction losses.

(c) Velocity head (H_{vi}). $V^2/2$ g where V is the pipe velocity [ft (m) of water].

If required, tower-pumping head is determined from measurement of static pressure in the inlet piping to the tower. Location and installation of pressure gages and/or instrumentation should be in compliance with requirements of Section 4.1 of PTC 19.2, Pressure Measurement. The measured static pressure [ft (m)] will be the linear average of measured gage pressures.

Tower pumping head is the average gage pressure [psig (Pa)] corrected for differential in elevations and friction losses from the point of reference (or terminal point of connection). Refer to para. 5-7 for determination of tower pump head (ft of water) from test data.

4-7 MEASUREMENT OF FAN POWER

When near the design point, the airflow rate through the tower is generally proportional to the cube root of the test fan motor power. Corrected test power is the shaft output power of the prime mover corrected for differences between the design and test air density.

For electric motors with constant speed drives, test measurements of the input power are made, and the output power is computed by multiplying input power by motor efficiency. The preferred instrument for determining power is a wattmeter. For the power measurement of high voltage motors (greater than 600 volts), panel readings of bus voltage and motor amperage may be used.

When readings are taken at a load center located a substantial distance from the motors, correction should be made by direct measurement of voltage drop (or by computation of line loss) between load center and motor. A measured or computed voltage drop between the load center and one motor may be applied to the other motors by prorating their distances from the load center. For prime movers other than electric motors, the method for determining power shall be mutually agreed upon prior to the test.

Test fan power shall be corrected for any difference between test and design air densities. The power correction for air density is necessary because the manufacturer's performance curves are based on constant fan blade pitch-angle (constant volume at constant fan speed). As atmospheric air conditions and/or heat load conditions vary from design, the mass air rate will vary, causing a direct variation in air pressure loss and air horsepower. (See para. 5-8 for the analytical method of fan power correction for the effect of air density.)

The airflow rate is directly proportional to the speed of the fan and thus directly proportional to the shaft speed of the motor. For electric motors with pulse width variable frequency drives (VFDs), the airflow rate is controlled through the use of the VFD. For fan motors with VFDs, it is recommended that the VFD be put in bypass mode for the duration of the test. When the VFD is bypassed, the fan power measurement is identical to that for a standard electric motor with a constant speed shaft. When in service, both the motor efficiency and motor shaft speed are typically reduced through the use of a VFD.

If the VFD does not have a bypass or if the test must be conducted with the VFD in service, the speed must be set to 100% and one of the following approaches must be taken for the evaluation of the fan motor power:

(a) The power on the line side of the VFD is measured. The VFD manufacturer's guaranteed voltage drop across the unit is used to calculate the power input into the motor shaft. The fan motor efficiency must be evaluated at the reduced power loading.

(b) Some speed controllers display power delivered to the motor. If agreeable to the test representatives, this may be used for the evaluation of the fan motor power.

(c) The fan motor power is measured with a true RMS wattmeter on the load side of the VFD and the motor efficiency is evaluated at the reduced power loading. The wattmeter must have a sampling frequency that is at least twice the highest frequency component in the wave being sampled. Switching frequencies of pulsed width modulation drives can range from 2 kHz to 15 kHz. In practice, a low-pass filter must be connected between the measurement location (e.g., switch gear) and the meter. The cut-off frequency of low pass filter should correspond to the sampling frequency of the digital meter.

It should be noted that use of the VFD would lower the fan motor efficiency.

The fan motor manufacturer should provide data for the expected fan motor efficiency at the reduced speed.

4-8 MEASUREMENT OF SOUND LEVEL

Measurement of sound levels is a subject beyond the scope of this Code. It is recommended that a standard consistent with the size and type of equipment be consulted for this type of test (see Wark and Warner in Nonmandatory Appendix M).

4-9 MEASUREMENT OF ATMOSPHERIC PRESSURE

Atmospheric pressure shall be measured with standard barometric instruments in accordance with PTC 19.2, Pressure Measurement, or calibrated equivalents. Adjustments shall be made to any readings taken indoors in spaces exhibiting lower pressure.

4-10 MEASUREMENT OF WSACC PRESSURE AND HEAT LOAD

4-10.1 Condensing Pressure

The steamside condenser pressure shall be measured by two basket tips per cell located in the two outboard bonnets at the entrance to the tube bundle with a maximum of 10 basket tips regardless of the number of cells in the WSACC.

The basket tips shall be in accordance with Section 4.3.1 of ASME PTC 12.2. The pressure instrument shall have an accuracy of ±0.04 in. Hg and be suitable for turbine exhaust vacuum conditions in accordance with ASME PTC 19.2. Measurements shall be taken at an elevation above the basket tip so that the tubing is continuously sloped down toward the tap. Tubing shall be purged with air just before each reading.

4-10.2 Heat Load and Steam Flow

The WSACC test heat load and steam flow can be established from energy balance methods. Because of the potential uncertainty inherent in heat balance methods, it is recommended that the testing be at the same time as the test of the steam turbine and in accordance with ASME PTC 6, Steam Turbines. It will be assumed that there is a flow meter capable of measuring the quantity of condensate or feedwater. Steam quality will be estimated from the heat balance and an extrapolation of the turbine expansion line to the pressure of the condenser on a Mollier chart. The steam flow will be computed from the feedwater or condensate flow corrected for extraneous drains or alternatively from above heat rejection and end point enthalpy corresponding to the WSACC pressure.

4-11 WATER ANALYSIS

4-11.1

If a sample of the circulating water is taken during the test and if there are any questions concerning the quality of circulating water, the sample shall be analyzed by a reputable testing laboratory to determine the constituent analysis of the water and/or conformance with para. 3-8(k). The design water analysis should define the operating limits (high/low ranges) of circulating water constituents and cycles of concentration.

The addition of chemical additives (biocides, surfactants, dispersants, etc.) can potentially impact tower performance and/or water distribution. Chemical injection should be discontinued approximately 24 hr prior to testing to avoid potentially impacting test results. In any event, injection of biocides should be discontinued during the tests to minimize exposure to test personnel.

The limits for foreign substances in the circulating water shall be established by prior mutual agreement between the purchaser and manufacturer/supplier. If no contractual agreements exist, the physical, chemical, and thermodynamic properties of the circulating water shall be determined by the ASHRAE Handbook (see Nonmandatory Appendix M).

The buildup of organic materials and/or inorganic substances on the surface of heat transfer (fill) media or structure can potentially indicate cooling water problems that could adversely affect test results. If the pretest inspection reveals such buildups, further analysis of cooling water samples should be considered.

4-11.2 Analysis of Process Fluid for Closed-Circuit Cooling Towers

The physical and thermodynamic properties of the process fluid or the reference source for these properties shall be specified in the contract documents and agreed upon by all parties to the test, prior to the test. A sample of the process fluid shall be taken during the test. Should any questions arise concerning the composition of the process fluid, the sample shall be analyzed by a testing laboratory acceptable to all parties to confirm the composition of the process fluid at the time of the test and, if necessary, the physical and thermodynamic properties. When the process fluid is a solution of ethylene or propylene glycol and water, the properties to be used in the test evaluation may be determined from CTI ATC-105, Supplement for Closed-Circuit Cooling Towers (see Nonmandatory Appendix M).

4-12 MEASUREMENT OF DRIFT LOSS

Measurement of a drift loss from a cooling tower is a subject beyond the scope of this Code. It is recommended that a cooling tower specific standard such as CTI ATC-140 be consulted.

Drift loss for a closed-circuit evaporative cooler shall be expressed as a percentage of the recirculating *external spray-water flow rate*.

4-13 DATA RECORDING

This paragraph is a guide for the implementation of a data-recording method. The Code preference is for an automatic data acquisition system (DAS).

(a) In contrast to the manual collection of performance test data, the DAS can enhance data collection and analysis by accomplishing the following:

(1) simultaneous reading and recording of all data points

(2) providing data collection frequency exceeding that described in para. 3

(3) providing data collection time measurements/synchronization as described in para. 3

(4) providing data accuracy exceeding that described in para. 1-3

(b) A portable, computer-based DAS, together with state-of-the-art sensor technology and analog-to-digital converter reliability, can accomplish the above objectives. This DAS could also provide the following:

(1) portability and ease of configuration

(2) flexible network, which can analyze sensor and acquisition faults

(3) flags and alarms for out-of-range values

(4) graphics for data trending and results presentation

(5) mass data storage and ease of data retrieval

(6) ease of calculation development and export data for third-party cooling tower parametric test studies, post-test uncertainty analysis, or thermodynamic model analysis

(c) The DAS can also provide certain enhancements that manual data collection cannot offer, such as:

(1) real-time data at very high sampling frequency

(2) reduction in manual data collection personnel

(3) reduction in data recording errors

(4) quick test condition/results validation, which reduces the need for retesting and associated costs

(5) measurement taking at otherwise inaccessible locations, e.g., cold water grids, elevated psychrometers, or exhaust air measurements needed for plume compliance

Section 5
Calculation and Results

5-1 GENERAL

This paragraph describes the numerical procedures and test data analysis that are used to establish the thermal performance and water consumption of wet mechanical draft, natural draft, and wet–dry cooling towers, and closed-circuit evaporative (wet) coolers and wet surface air-cooled steam condensers. The paragraph additionally addresses the data analysis required to determine the compliance of wet–dry towers to plume abatement specifications. This paragraph also includes the post-test data uncertainty estimates that ensure the quality of all the types of tests that are defined as appropriate and in accordance with Code requirements.

5-1.1 Philosophy of Air Temperature Measurements

Major performance and plume compliance variables of all types of wet, evaporative water cooling equipment are the wet- and dry-bulb temperatures at which the cooling occurs. Whether to use the ambient air conditions (considered to be those outside of the influence of the tested cooling tower and other towers in the vicinity) or the inlet air conditions (the air entering the cooling tower) has a profound effect on the specified guarantees and the test results of all equipment governed by the Code. This Code continues to recognize the entering air temperature and humidity measurements as the criteria of thermal performance and plume compliance for the following reasons:

(a) A test inlet air condition provides a physically correct gauge of the actual heat transfer performance or plume abatement capability of the equipment.

(b) Use of the entering air conditions minimizes the uncertainty of the recognized impacts of recirculation, humidity, and vertical temperature gradients on the performance.

(c) Measuring inlet air conditions provides a means of separately determining the actual tower performance and the effects of recirculation, humidity, or vertical temperature gradients. These parameters can be extracted from the test data regardless of testing conditions and as opposed to only global tower thermal performance or plume abatement capability. It also automatically accounts for upwind tower interference. Hence the test is more quantitatively reproducible.

(d) The purchaser is provided the capability of checking recirculation predictions at the time of the test.

(e) The use of inlet air conditions indirectly encourages progress toward improving tower designs that reduce recirculation or improve plume abatement capability. That is, it furnishes the capability of verifying design predictions during the tests and assigning credit to superior design during proposal evaluations.

(f) It also encourages purchasers to consider methods of accounting for recirculation, interference, and vertical temperature gradients during the plant design and specification writing phase. Weather history studies, model testing of overall plant and topography, computational fluid dynamic (CFD) techniques, or experience can be used and the results of these estimates incorporated into the cooling equipment thermal performance purchase specification requirements.

The advantages of the philosophy of using inlet air temperature and humidity measurements are amplified in subsequent paragraphs. Recirculation is not a new phenomenon in the consideration of the design of mechanical draft cooling towers; neither is interference. The vertical temperature gradient effects on thermal performance and plume abatement capability are also well known.

The adverse effects of severe vertical temperature gradients on the performance of natural draft towers have been widely documented. However, the vertical temperature gradient is not directly reflected by inlet air measurements.

Recirculation is a broader problem. Testing mechanical draft cooling equipment with ambient temperature measurements does not allow the purchaser to determine whether the tower purchased is performing properly or otherwise complying with specification requirements as guaranteed. Even when the performance test results were good, it could not be determined if this resulted from a lower recirculation at wind speed that was less than design. Conversely, the manufacturer could blame excessive recirculation for poor test results and cite adverse test conditions. Later, this same manufacturer could make a few minor modifications to the tower, come back to test on a day of lesser wind speed and recirculation, and show better overall performance.

It is important to distinguish between the tower's actual thermal performance or plume abatement capability and the effect of recirculation. For example, suppose a tower were tested and found to be deficient. From the point of view of a guarantee, both recirculation and thermal design or plume compliance deficiencies represent inadequacy. However, the design deficiency is permanent. On the other hand, the recirculation effect varies with wind speed, occurrence, and direction. The separate determination of tower effectiveness versus the recirculation effects may be made by the user of the Code.

Further, when an ambient performance test was specified in the past, it was sometimes a practice for a purchaser to define the associated recirculation allowance. With the inlet air temperature Code performance test, the purchaser can elect to specify only the design ambient temperature, and request that the manufacturer provide a guaranteed performance or plume compliance based on inlet conditions in relation to wind speed, direction, and occurrence. The purchaser may also elect to estimate the recirculation allowance needed before requesting proposals so that the optimum system performance design conditions can be established. For this alternative, the purchaser combines weather data, in the form of seasonal temperature, humidity if applicable, wind velocity, and frequency of occurrence, with the manufacturer's empirical data of cost and either thermal performance or plume abatement capability if applicable, and plant cost values.With that data, the purchaser calculates an optimum weighted average recirculation allowance for the seasonal, local temperatures. If project time for the sizing, design, and orientation of the tower allows, the recirculation can also be determined by specifying a wind or water tunnel model test or CFD analysis. The latter would estimate the recirculation with an appropriate scale model of the cooling equipment including the major tower geometry, and a simulation of the site conditions.

5-1.2 Thermal Capability Evaluation

The reason for conducting thermal performance test data calculations for cooling towers is to determine either the capability, or alternatively, to project the as-tested cold water temperature to design conditions so that its deviation from any guarantees may be evaluated. For a closed circuit evaporative cooler or a wet surface air-cooled condenser, it may also be expressed as the deviation in design or a reference turbine backpressure from a projection of the test data to that same point.

The tower capability, expressed as a percentage, is the ratio of the adjusted test circulating water flow rate to the predicted test circulating water flow rate at the test conditions. For a wet cooling tower, the capability is the percentage of the water flow that can actually be cooled to the test conditions, compared to the quantity that was guaranteed. For evaporative coolers or wet surface air-cooled condensers, the capability is a like term but instead uses the ratio of the adjusted process fluid flow or condenser steam flow to that guaranteed for the specific test conditions.

5-1.3 Plume Abatement Compliance

Wet–dry cooling towers may also require a plume abatement compliance test. This requires measurement of the thermal parameters at the time of the measurement of the exhaust plume's compliance with the nonvisibility requirements in a specification. The degree of visibility of the plume is determined by comparing a projection of the plume weighted average relative humidity data at the test upwind entering air conditions (see para. 4-4.4) to the guaranteed relative humidity at the specified design conditions. All required intermediate and supporting computations for the evaluation of that compliance are also described in this Section.

5-1.3.1 Wet-Dry Cooling Towers, Limited or Zero Visible Plume. Cooling towers are often located in factories with taller installations around them. It can then be acceptable to have light visible plume located above the fan exhaust area, extending around 50 ft (15 m). In such a case, the design of the wet–dry cooling tower is called "limited visible plume." In that case, light visible plume is not acceptable, meaning that, at the design point, no visible plume may occur, even just above the fan stack area, the design of the wet–dry cooling tower is called "zero visible plume."

5-1.3.2 Wet-Dry Plume Abatement Design Point. The definition of "limited or zero visible plume" has an effect on the design of the wet–dry cooling tower and on the criteria described in this code to fulfill the plume performance guarantee.

The design air temperature and humidity is defined as the entering air upwind from the cooling tower row, assuming isothermal atmosphere. (It may be different from the ambient air temperature and humidity).

It is recommended that a "winter mode" thermal performance design point be associated with the plume design point.

5-2 WATER FLOW

5-2.1 Average Test Water Flow

When more than one flow rate has been measured during a test period, the average test water flow rate shall be calculated as the sum of the water flow rates divided by the number of times flow was measured. Where flow rates have been measured for multiple portions of the system, such as at each of multiple riser pipes on a tower, the sum of the average flow rates from each portion of the tower shall be the total average test water flow rate for the test period.

5-2.2 Adjusted Test Water Flow

5-2.2.1 Mechanical Draft Towers. The water rate to the cooling tower measured during the test period shall be adjusted to account for any departure of the measured (test) fan power from the design fan power. This adjusted test water rate shall be defined as the product of the measured test water rate and the one-third power of the ratio of the design fan power to the corrected test fan power.

The corrected test fan power is the measured fan power corrected for any line loss between the point of measurement and the fan motor, as well as for any variation of the test fan air density from the design fan air density. [See para. 4-7 and Nonmandatory Appendix E, para. E1(a)].

5-2.2.2 Wet-Dry Towers. The adjustment is the same as for mechanical draft towers in para. 5-2.2.1, except when dry and wet sections have separate fans.

5-2.2.3 Natural Draft Towers. The measured test water flow rate may be used without adjustment for thermal evaluation. The manufacturer's performance curves shall include the effect of changes in mass airflow rate and thus reflect the true thermal capability.

5-2.2.4 Closed-Circuit Evaporative Coolers. The process fluid flow to the evaporative cooler measured during the test period shall be adjusted to account for any departure of the measured (test) fan power from the design fan power. This adjusted test fluid flow shall be defined as the product of the measured test fluid flow and the ratio of the design fan power to the corrected test fan power taken to an exponent between 0.25 and 0.33, as specified by the manufacturer. Without any specified values, the 0.33 exponent shall be used.

The corrected test fan power is the measured fan power corrected for any line loss between the point of measurement and the fan motor, as well as for any variation of the test fan air density from the design fan air density. [See para. 4-7 and Nonmandatory Appendix H, para. H1(a).]

5-2.2.5 Wet Surface Air-Cooled Steam Condensers. The steam flow to the WSACC measured during the test period shall be adjusted to account for any departure of the measured (test) fan power from the design fan power. This adjusted test steam flow shall be defined as the product of the measured test steam flow and the ratio of the design fan power to the corrected test fan power taken to an exponent between 0.25 and 0.33 as specified by the manufacturer. Without any specified values, the exponent 0.33 shall be used.

The corrected test fan power is the measured fan power corrected for any line loss between the point of measurement and the fan motor, as well as for any variation if the test fan air density from the design fan air density. (See para. 4-7 and Nonmandatory Appendix I, para. I1.1.)

5-3 WATER TEMPERATURES

The measured temperatures shall be corrected for instrument calibration.

(a) Hot Water Temperature. The test hot water temperature T_{HW} shall be the weighted average temperature of the entering streams.

Table 5-3 Temperature Corrections

Location of Well	Before Pump	After Pump
Thermowell	No correction	Apply Eq. (2)
Flowing well	Apply Eq. (1)	Apply Eq. (2a)

(b) Cold Water Temperature. If the temperature is obtained by measuring the temperature of water flowing from a full flowing well, a pressure differential from the main stream (at pressure P_W) to the discharge well will exist. This throttling process will increase the temperature of the sampling stream and the measured temperature shall be corrected by use of Eq. (1).

$$T_{21} = T_W - FP_W \tag{1}$$

If the measurement point is located at the discharge of the circulating pump(s), the temperature rise from the heat added by the pump work must also be considered. The measured temperature shall be corrected by use of the following:

Thermowell at pump discharge

$$T_{22} = T_W - F(P_{Wd} - P_{Wi})[(1 - \eta_p)/\eta_p] \tag{2}$$

Flowing well at pump discharge

$$T_{22} = T_W - F\left[(P_{Wd} - P_{Wi})/\eta_p\right] \tag{2a}$$

where

F = 0.002966, U.S. Customary units (2.39×10^{-7}, SI units)

These temperature corrections are summarized in Table 5-3.

The test cold water temperature t_2 shall be the weighted average temperature of the exiting streams corrected for the effects of makeup, blowdown, and any other heat added or removed between the equipment and the measured points computed from Eq. (3):

$$T_{23} = \frac{W_{HL}\,T_{HL} + W_{BE}\,T_{BE} + W_B T_B - W_M T_{21} - W_{HE}\,T_{HE}}{W_{HL} + W_{BE} + W_B - W_M - W_{HE}} \tag{3}$$

Equation (3) is based on the assumption that during the test, no streams other than those in the denominator entered or left the basin.

5-3.1 Hot and Cold Process Fluid Temperatures for Closed-Circuit Evaporative Coolers

Testing of closed-circuit evaporative coolers, in which the process fluid to be cooled is circulated inside the tubes/plates of a tube bundle/heat exchanger, the *hot water temperature* shall be deemed to be the weighted average temperature of the process fluid stream(s) entering the tube bundle/heat exchanger. The *cold water temperature* shall be deemed to be the weighted average temperature of the process fluid stream(s) leaving the tube bundle/heat exchanger. Temperature corrections for throttling, heat added by the pump, and any other additions or losses of heat between the measurement points and the equipment shall be applied, as appropriate. The adjustment for heat added or removed by makeup and blowdown shall be applied to the process fluid flow rate. See para. 5-10.2.1.

5-3.2 Thermal Lag

Normally the interval from the time the cooled water reaches the collecting basin to the time it reaches the t_2 measurement station is small, and computation of thermal lag is not required. The time interval, determined by Eq. (4), is the theoretical minimum (plug flow) lag time. The actual lag time under a mixing condition may be higher than the calculated lag time. This lag time calculation is appropriate for deep basins or other basins where the measured cold water response of the tower is abrupt. This calculation is not appropriate for cooling towers in which the thermal response time of the tower is drawn out over an extended period. If this time interval, determined by Eq. (4), is greater than the interval for which averages are computed (usually 5 min), the test time period shall be lengthened by a like amount. Test averages shall be based on compensating time spans, so that the readings chosen will represent the true tower performance.

$$S = \frac{V\rho_w}{W_L} \tag{4}$$

The following readings will be averaged over the first part of the timed period (usually 60 min):

T_{HW}, T_{DB}, T_{WB}, W_M, W_{HE}, T_{21}, T_{HE}, and W_L

And the fan power; and the following, over the latter part:

W_{HL}, W_{BE}, W_B, T_{HL}, T_{BE}, and T_B

Thermal lag time of less than the interval at which averages are computed (usually 5 min), may be ignored.

5-4 AIR TEMPERATURES

5-4.1 Average Entering Wet-Bulb Temperature

The entering wet-bulb temperature for a test period shall be calculated by first calculating the average of the values for each location over the duration of the period. The average of the individual averages from each location shall be the average entering wet-bulb temperature for the tower for the test period.

5-4.2 Average Entering Dry-Bulb Temperature (Wet–Dry and Natural Draft Towers)

The entering dry-bulb temperature for a test period shall be calculated by first calculating the average of the values for each location over the duration of the period. The average of the individual averages from each location shall be the average entering wet-bulb temperature for the tower for the test period.

5-4.3 Average Discharge Wet-Bulb Temperature (Plume Compliance — Wet–Dry Towers)

The average discharge wet-bulb temperature shall be calculated from the average of the local enthalpy and humidity ratio weighted by the local mass flow. See para. 4-4.5.

5-4.4 Average Discharge Dry-Bulb Temperature (Plume Compliance — Wet–Dry Towers)

The average discharge dry-bulb temperature shall be calculated from the average of the local enthalpy and humidity ratio weighted by the local mass flow. See para. 4-4.5.

5-4.5 Upwind Wet- and Dry-Bulb Temperatures (Plume Compliance — Wet–Dry Towers)

The average individual location entering wet- and dry-bulb temperatures on the upwind faces of the tower shall be calculated. The average of the upwind values of each of wet-bulb and dry-bulb temperatures shall be the average of the individual location averages.

5-4.6 Vertical Dry-Bulb Temperature Gradient or Lapse Rate (Natural Draft Towers)

If available, the change in dry-bulb temperature from approximately the top of the air inlet of the tower to approximately twice the shell height shall be calculated for each of the 200 ft (60 m) increments, and averaged. The sign convention shall be negative for a decrease in temperature with increased height.

5-5 STEAM CONDENSING TEMPERATURE

The steam condensing temperature of a wet surface air-cooled condenser shall be determined from the saturation temperature corresponding to average of the measured condenser pressures.

5-6 AIR VELOCITY

5-6.1 Wind Velocity (Mechanical Draft Equipment)

The average wind speed and direction (in degrees) for the test period shall be calculated.

5-6.2 Maximum 1-min Duration Wind Gust Speed. The maximum value of wind speed shall be determined from 1-min calculated averages.

5-6.3 Top of Shell Wind Velocity (Natural Draft Towers)

If available, the average wind speed and direction (in degrees) at the top of shell shall be interpolated from the vertical velocity gradients taken during the test, or calculated according to the equation in para. 4-5.2.

5-6.4 Discharge Air Velocity (Wet/Dry Towers)

The vertical component of velocity for each traverse location (when the flow measurement device is aligned with the local flow stream) shall be calculated from the following equation:

$$V_v = V_m \cdot \cos\alpha$$

where

α = measured air velocity angle from vertical

V_m = measured air velocity at measured angle

V_v = vertical component of air velocity

5-7 TOWER PUMPING HEAD

Pump head pressure measurement must be corrected for differentials in static head and piping head losses from the point of measurement to the reference point (or terminal point of connection, see para. 4-6). The tower pumping head is obtained by correcting the measured gage static pressure for the following:

(a) differential in test port elevation to the centerline of the tower inlet connection elevation

(b) differential in measured water flow to tower design water flow (friction losses, velocity head, etc.)

On mechanical draft towers, the pressure measurement is typically converted to an equivalent pressure at the centerline of the inlet piping connection (flange). On natural draft towers, pressure measurement is typically converted at the centerline of the inlet piping to the tower riser. In both cases, this is done by subtracting the friction head losses in the piping from the point of measurement to the centerline of the tower inlet, and the differential in elevation between these two points of reference, from the measured pressure

$$P'_s = P_s - (P_f - \Delta Y) \tag{6}$$

Correcting the gage static pressure for tower design flow is done on the assumption that gage static pressure is proportional to the square of the flow rate. Friction losses at tower design flow rate is determined as follows:

$$P''_s = P'_s \, (L_d/L_m)^2 \tag{7}$$

where

L_d = design water flow rate, gpm (l/s)

L_m = measured water flow rate, gpm (l/s)

The measured tower pumping head at the design water flow rate is then obtained by adding the corrected static head (H_s) plus the design velocity head (H_v) and the friction losses from Eq. (7) above. This value may then be compared to the design tower pumping head

$$PH_t = P'_s + H_s + H_{vi} \tag{8}$$

where

H_s = static head, the vertical distance from the top of the basin wall (curb) to the centerline of the inlet piping to the tower

H_{vi} = $V^2/2g$

P'_s = $H_N + H_f$

5-7.1 Piping Losses

Piping losses include the following components:

(a) Nozzle Loss (H_N). Static pressure necessary for compensation of nozzle pressure losses. This head loss should be the minimum head required to ensure proper water distribution.

(b) Friction losses (H_f). Piping and valve losses from the terminal point of connection through the entire tower distribution system. On natural draft towers, H_f typically includes riser friction losses.

(c) Velocity head (H_{vi}). $V^2/2g$ where V = the pipe velocity.

If required, tower pumping head is determined from measurement of static pressure in the inlet piping to the tower. Location and installation of pressure gages and/or instrumentation should be in compliance with requirements of para. 4.1 in PTC 19.2, Pressure Measurement. The measured static pressure will be the linear average of measured gage pressures.

5-7.2 Tower Pumping Head for Closed-Circuit Evaporative Coolers or Wet Surface Air-Cooled Condensers

When the equipment to be tested is a closed-circuit evaporative cooler or wet surface air-cooled condenser, the tower pumping head shall be understood to be the head of the external spray water flow to the tower. Measurement and evaluation shall be as for an open cooling tower, as described above.

5-7.3 Heat Exchanger Pressure Drop for Closed-Circuit Evaporative Coolers

(a) Pressure Differential Instruments. When a pressure differential instrument is used to measure the pressure difference between the inlet nozzle and the outlet nozzle of the heat exchanger, this measurement, corrected for any losses that occur between the measurement points and the nozzles, is the pressure drop across the heat exchanger.

(b) Individual Pressure Measurements. If the pressures at the inlet and outlet nozzles of the heat exchanger are measured separately, an adjustment must be made to correct for any difference in elevation.

5-8 FAN POWER

5-8.1

Fan motor power may be calculated using one of the following equations:

(a) Electric motor power output from wattmeter measurements may be computed using the following equation:

$$HP_t = \frac{(kW_{ob})(\eta_m)(E_m)}{(0.746)(E_{ob})}\left(\frac{\rho_T}{\rho_D}\right)$$

where

ρ_D = design density

ρ_T = test density

(b) If both motor efficiency and power factor are supplied by the motor manufacturer, power from measured voltage and amperage may be computed from the following equation:

Customary Units:

$$HP_t = \frac{(\sqrt{3})(E_m)(I_{ob})(PF)(\eta_m)}{(746)}\left(\frac{\rho_T}{\rho_D}\right)^{.333}$$

SI Units:

$$kW = \sqrt{3}(E_m)(I_{ob})(PF\mu_m)\left(\frac{\rho_T}{\rho_D}\right)^{.333}$$

(c) If the available motor data does not include the power factor, this term shall be calculated from nameplate data using the following equation:

Customary Units:

$$PF = \frac{(746)(HP_{np})}{(\sqrt{3})(E_{np})(I_{np})(\eta_m)}$$

SI Units:

$$PF = \frac{kW_{np}}{(\sqrt{3})(E_{np})(I_{nm})(\mu_m)}$$

(d) If the power factor and efficiency are unknown, power from voltage and amperage measurements may be computed from the following equation, provided that the motor is operating within 15% of its nameplate rating:

Customary Units:

$$HP_t = \frac{(HP_{np})(E_m)(I_{ob})}{(E_{np})(I_{np})}\left(\frac{\rho_T}{\rho_D}\right)^{.333}$$

SI Units:

$$kW = \frac{k(W_{np})(E_m)(I_{ob})}{(E_{np})(I_{np})}\left(\frac{\rho_T}{\rho_D}\right)^{.333}$$

5-8.2 Fan Motor Efficiency

For towers with less than two years in service, nameplate NEMA motor efficiency shall be used for the calculation of brake horsepower. Rewinding a motor typically causes a decrease in motor efficiency of approximately 0.5% per rewind. If the fan motor efficiency cannot be measured directly, the motor efficiency of a rewound motor shall be assumed to be the original nameplate corrected for each rewinding loss of 0.5%. If it is unknown whether the motor has been rewound, or if the motor is older than two years and the motor efficiency cannot be measured directly, the nameplate motor efficiency shall be used for the calculation of fan motor brake horsepower. The fan motor manufacturer must provide the motor efficiency if a VFD is in service during the thermal performance test. See para. 4.7.

5-9 ATMOSPHERIC PRESSURE

The average atmospheric pressure shall be calculated for the test period.

5-10 THERMAL CAPABILITY

5-10.1 Test Results

The manufacturer's performance curves shall be used to evaluate cooling tower performance. Examples are provided in the appendices.

5-10.2 Adjusted Flow

5-10.2.1 Makeup Correction Flow Rate for Closed-Circuit Cooling Towers and WSACC. If the makeup water supply to the external spray water system cannot be isolated during the test, the measured flow rate shall be adjusted as indicated below. This will account for the heat added or removed as a result of the difference between the temperature of the makeup water entering the unit and the temperature of the spray water flowing in the external circuit.

(a) *Closed-Circuit Cooling Towers*

$$Q_{PFt\,\mathrm{adj}} = \frac{Q_{PFt} - [(T_{RW} - T_{MU})(\rho_W)(c_{pW})]}{[(T_{HPF} - T_{CPF})(\rho_F)(c_{pPF})]} \cdot (Q_{MU})$$

c_{pW} = specific heat of spray water flowing over external surface of heat exchanger

c_{pPF} = specific heat of process fluid at the average temperature

ρ_W = density of the spray water flowing over external surface of heat exchanger

ρ_{PF} = density of the process fluid

Q_{MU} = volumetric flow rate of makeup water

Q_{PFt} = volumetric flow rate of process fluid measured during test

$Q_{PFt\,\mathrm{adj}}$ = volumetric flow rate of process fluid, measured and adjusted for makeup

T_{RW} = temperature of external spray water, measured at the spray water pump discharge

T_{CPF} = temperature of the cold process fluid leaving the tower

T_{HPF} = temperature of the hot process fluid entering the tower
T_{MU} = temperature of the makeup water entering the tower

(b) *Wet Surface Air-Cooled Condenser*

$$Q_{adj} = Q_{meas} - [(T_{RW} - T_{MU}) \cdot D_W \cdot C_{pw}/h_{fg}] \cdot Q_{MU}$$

Q_{adj} = mass steam flow adjusted
Q_{meas} = mass steam flow measured
T_{RW} = temperature of external spray water, measured at the spray water pump discharge
T_{MU} = temperature of makeup water
h_{fg} = change in steam enthalpy
Q_{MU} = makeup flow

5-10.2.2 Fan Power Correction. (Not applicable to natural draft cooling towers). The adjusted test process flow rate shall be defined as the product of the measured test process fluid flow rate multiplied by the ratio of the design fan power to the corrected test fan power taken to an exponent. The value of the exponent must be supplied by the tower manufacturer prior to the test, and shall be limited in magnitude to $0.25 \leq \exp \leq 0.33$. The default value is 0.33, if not otherwise specified by the manufacturer.

$$Q_{adj} = Q_{meas} \cdot (DP_d/HP_t)^{\exp}$$

exp = exponent
HP_d = design fan power
HP_t = test fan power
Q_{adj} = adjusted process flow rate
Q_{meas} = measured process flow rate

5-10.3 Tower Capability Evaluation

5-10.3.1 Mechanical Draft, Natural Draft, Wet-Dry, and Closed-Circuit. The performance curves described in para. 3-9(a) shall be initially cross-plotted at the test wet-bulb temperature. The cross plots shall show cold water temperatures as ordinates and cooling ranges as abscissas with lines of constant water rate as parameters. This curve will yield the predicted cold water temperature at the test cooling range and wet-bulb temperature for each of the various water rates.

In the case of a wet–dry and natural draft cooling tower, an additional cross plot from curves with dry-bulb temperature or relative humidity as an additional variable is required.

A last cross plot showing cold water temperatures as ordinates and water rates as abscissas shall then be plotted from the points derived from the first cross plot. The predicted water rate at the measured test cold water temperature is then read from the latter curve. The thermal capability of the cooling tower is calculated from:

$$\text{Tower capability (\%)} = \frac{\text{Adjusted test water rate}}{\text{Predicted test water rate}} \cdot 100$$

Nonmandatory Appendix E, para. E-2 shows the predicted test cold water temperature at the test heat load and wet-bulb temperature, compared to the cold water temperature measured during the test. Nonmandatory Appendix E, para. E3 shows how to predict the design cold water temperature at design heat load, design fan motor power and design wet-bulb temperature based on the measured test tower capability.

5-10.3.2 Wet-Surface Air-Cooled Steam Condenser. The performance curves described in para. 3-9.3 shall be initially cross plotted at the test wet-bulb temperature. The cross plots shall show steam absolute pressure as ordinates and steam vapor quality as abscissas with lines of constant steam rate as parameters. These curves will yield the predicted steam absolute pressure at the test steam quality and wet-bulb temperature for each of the various steam rates.

A last cross plot showing steam absolute pressure as ordinate and steam rate as abscissa shall then be plotted from the points derived from the first cross plot. The predicted steam rate at the measured test steam absolute pressure is then read from the latter curve. The thermal capability of the WSACC is calculated from:

$$\text{WSACC capability (\%)} = \frac{\text{Adjusted test steam rate}}{\text{Predicted test steam rate}} \cdot 100$$

5-11 PLUME COMPLIANCE

5-11.1 Evaluation Using the Exhaust Air Characteristics Curves

The plume performance is evaluated in terms of tower plume indicator by the ratio of the equivalent guaranteed relative humidity (calculated from the guaranteed exhaust air wet-bulb and dry-bulb temperatures read on the manufacturer's plume performance curves for the test conditions) and the measured relative humidity of the exit air.

5-11.2 Correction of the Relative Humidities by the Atmospheric Pressure

As the barometric pressure at the day of the test is not necessarily the same as the barometric pressure on a psychrometric chart, a correction has to be calculated to refer to the chart barometric pressure. This correction will be applied to the inlet, upwind, and exhaust air relative humidities.

$$RH_C = RH \frac{P_{diag}}{P_{atm}}$$

where
P_{atm} = atmospheric pressure during the test
P_{diag} = atmospheric pressure of the psychrometric diagram (generally 1013.3 mbar, 29.92 in. Hg, or 14.697 psi)

RH = measured relative humidity (inlet, upwind, or exhaust)
RH_C = corrected relative humidity (inlet, upwind, or exhaust)

5-11.3 Drawing of the Measured Plume Dilution Line

On the psychrometric diagram:

(a) Locate the point representing the upwind air condition using the upwind dry-bulb temperature and the upwind relative humidity.

(b) Locate the point representing the exhaust air conditions, using the exhaust dry-bulb temperature and the exhaust relative humidity.

(c) Draw a straight line between these two points. This line is the measured plume dilution line.

5-11.4 Calculation of the Guaranteed Relative Humidity

On the manufacturer's guaranteed curves, the guaranteed exhaust wet-bulb and dry-bulb are read, using linear interpolation between the curves for the test conditions (flow, range, inlet wet-bulb temperature, corrected inlet relative humidity). The guaranteed relative humidity of the exhaust air is then calculated from these values and the standard barometric pressure (the one used in the psychrometric diagram), using the psychrometric diagram or software.

5-11.5 Drawing of the Guaranteed Plume Dilution Line

On the psychrometric diagram:

(a) Locate the point representing the upwind air conditions using the upwind dry-bulb temperature and the upwind relative humidity.

(b) Locate the point representing the guaranteed exhaust air conditions, using the guaranteed exhaust dry-bulb temperature and the exhaust relative humidity.

(c) Draw the straight line between these two points. This line is the guaranteed plume dilution line.

5-11.6 Equivalent Guaranteed Exhaust Air Relative Humidity

The relative location of the two dilution lines on the psychrometric chart (measured and guaranteed) shows whether the plume performance is fulfilled. If the measured dilution line is on or under the guaranteed dilution line, the plume performance is fulfilled. To quantify the performance, a tower plume indicator can be calculated. The guaranteed exhaust air relative humidity must be transferred to an equivalent one at the same enthalpy as the test condition.

(a) Draw the enthalpy line passing at the point representing the measured exhaust air conditions.

(b) Read the equivalent guaranteed exhaust air relative humidity at the intersection of the guaranteed dilution line and the above enthalpy line.

5-11.7 Calculation of the Tower Plume Indicator

The ratio expressed in percent of the equivalent exhaust air relative humidity and the measured exhaust air relative humidity is the tower plume indicator.

$$TPI = 100 \cdot \frac{RH_{gc}}{RH_c}$$

where

RH_c = corrected exhaust air relative humidity
RH_{gc} = equivalent exhaust air relative humidity

5-11.8 Evaluation of the Air Mixing Quality

This evaluation need only be done when the guarantee is based on zero visible plume. If the dry air coming from the dry heat exchanger does not mix well with the wet air coming form the wet heat exchanger, the temperature and the humidity of the rejected air will vary across the section of the cooling tower exhaust area. Some light visible plume due to incomplete mixing may exist on some locations above the fan stack exit level. If such light plume only extends generally within 50 ft (15 m) above the fan stack, it is often acceptable to the owner. If such light plume is acceptable, the design basis is limited visible plume and the mixing quality need not be verified.

If such light plume is not acceptable, the design basis refers to zero visible plume, and the quality of the air mixing shall then also be verified. The procedure is to check that any exhaust air measurement point is within an acceptable variation from the average point. This variation is called the scattering criteria.

(a) If all the points measured above the fan stack are within the scattering criteria, the mixing quality is acceptable.

(b) If some of the points are above the acceptable variation, but the airflow corresponding to these points is small (maximum air flow criteria, see para. 5-11.9.3), the plume will disappear very quickly above the fan stack, and the mixing quality is acceptable.

(c) If some points are over the acceptable variation, but the air flow associated with these points is large, the plume will not disappear quickly above the fan stack, and the mixing quality is not acceptable.

5-11.9 Scattering Criteria

5-11.9.1 Calculation of the Maximum Limit of the Exhaust Air Relative Humidity. Tests on models and on actual cooling towers show that a scattering from 1.2 to 0.8 times the average value of the exit air relative humidity is normal and acceptable when mixing wet and dry air. This means that the maximum measured relative humidity (corrected by the atmospheric pressure as described in para. 5-11.2) calculated for each individual point at the fan exhaust area must be within the average multiplied by 1.2:

$$RH_{max.} = RH_c \cdot 1.2$$

where

RH_c = corrected measured average relative humidity, %

$RH_{max.}$ = maximum limit of the exhaust air relative humidity, %

5-11.9.2 Calculation Criteria. On the psychrometric chart:

(a) Draw the enthalpy lines passing through the point representing the measured exhaust air conditions.

(b) Locate on this line the point corresponding to the allowed maximum limit for the exhaust air relative humidity ($RH_{max.}$).

(c) Draw the straight line between the above point and the point representing the upwind air conditions. This line is the maximum limit dilution line.

(d) Mark any individual measured exhaust air condition that is outside of the maximum limit dilution line.

5-11.9.3 Maximum Airflow Criteria. If the measured relative humidity at some points is above the normal scattering, these particular points have to be marked, and the associated airflow calculated. If the airflow associated with these particular points doesn't exceed 15% of the total airflow of the cooling tower, the mixing quality is acceptable.

5-11.9.4 Mixing Quality Coefficient. The mixing quality coefficient can be calculated from the above marked and recorded airflow.

$$M_q = \left(1 - \frac{\Sigma V_{em}}{\Sigma V_e}\right) \cdot 100$$

where

M_q = mixing quality percentage
= $\geq 85\%$

V_e = local exhaust air flows (air velocity)

V_{em} = marked exhaust local air flows (air velocity)

The air mixing quality is fulfilled if the mixing quality coefficient is greater than or equal to 85%.

5-12 UNCERTAINTY

Regardless of the wet cooling equipment type, the thermal performance of the equipment is evaluated from measurements of hot water temperature, cold water temperature, wet-bulb temperature, water flow rate, etc. Each of these variables is measured with instrumentation designed to be accurate but the measurement of each parameter is inexact to some degree. The amount of error within a measured variable and ultimately the total error in the test result cannot be directly measured but can be estimated through the application of an uncertainty analysis. Uncertainty U is defined as an interval, about the measured value or the final test result that has a 95% probability of containing the true value. The procedures developed in ASME PTC 19.1-1998 are used to estimate the uncertainty in calculated capability results of the particular wet cooling equipment test by:

(a) estimating the systematic and random uncertainty components in each variable

(b) combining the systematic and random components

(c) evaluating the sensitivity of the capability to the parameter

(d) combining the uncertainties in terms of capability

The procedure described in this Code provides a simplified method for calculating the uncertainty associated with a cooling equipment performance test. The confidence level of 95% indicates there is a 95% probability that the true test result would fall within the range of the calculated test capability, plus or minus the test uncertainty.

An example of an uncertainty calculation is shown in Nonmandatory Appendix D.

In this procedure, terms that typically have less than 0.20% impact on capability are not included. These terms are insignificant for normal operating and test conditions and may be excluded when test conditions are within the operating limits specified in para. 3-8. The effects of some excluded terms may increase if the test conditions vary significantly from the design point. Parameters that are not included in the calculation of tower capability are also not evaluated in an uncertainty estimate. Examples of variables that are not evaluated are effects of inaccurate performance curves, wind speed, and lapse rate.

It should be noted that the purpose of a post-test uncertainty analysis is to determine the accuracy of the final test results.

5-12.1 Systematic Uncertainty

Systematic uncertainties are approximations of the fixed errors inherent in a measurement. These errors are also called bias errors. Systematic errors are typically the largest source of error in a cooling tower performance analysis. These errors are primarily a result of the intrinsic accuracy of the instruments and of the calibration procedures employed. Systematic uncertainty errors are estimated from review and analysis of the instrument manufacturer's specifications, independent parameter measurement by additional means, and examination of typical calibration data.

5-12.2 Spatial Systematic Uncertainty

Spatial systematic uncertainty errors occur during the measurement of a spatially diverse sample. Spatial error is defined as the difference between the true average value of a parameter and the average produced by an array of instruments used to measure the parameter. Spatial errors for a standard cooling tower thermal performance test occur during the measurement of inlet wet-bulb temperature, inlet dry-bulb temperature, and cold water temperature when measured with an array of probes. Spatial uncertainties also occur during the measurement of water flow rate within a pipe as a result of the limited number of local measurements that are used to calculate average velocities across the measurement plane. Spatial uncertainties are calculated from the average of local measurements in space and are thus independent of time.

In general, spatial distributions are not random; there is definite pattern in the variation of the test parameter in space. In principle, the uncertainty associated with this variation could be calculated as an integration error. However, the measurement points for the test parameters containing a spatial distribution are not sufficiently dense to permit the treatment of spatial uncertainty as an integration error. Therefore, the variation is treated as random with the understanding that this will overestimate the spatial uncertainty. The calculation of the spatial uncertainty depends on an adequate sampling of the variation. For instance, because of access problems, water flow rates are frequently measured with a pitot traverse conducted on a single diameter. This single traverse provides no information about the variation of velocity with angular position and therefore contains insufficient data for the calculation of uncertainty due to spatial variation.

According to the guidelines of ASME PTC 19.1 [Eq. (10.7)], spatial uncertainties are calculated from the following formula, from which S_{spatial} is calculated:

$$B_{Sp} = \frac{S_{\text{spatial}}^2}{\sqrt{M}}$$

$$S_{\text{spatial}} = \sqrt{\frac{\sum_{k=1}^{M} (\overline{X}_k - \overline{\overline{X}})^2}{M-1}}$$

where

B_{Sp} = systematic uncertainty due to spatial variation

M = the number of measurements locations

$\overline{\overline{X}}$ = average value of the group of measurements

$\overline{X}_k$ = time-averaged value at the measurement location k

Spatial errors are associated with the calculation of the average from physically distinct regions. For a mechanical draft cooling tower with two air inlets, the upwind side of the cooling tower is physically separated from the downwind side. A separate spatial uncertainty is calculated for each side of the cooling tower. An example of this calculation is included in Nonmandatory Appendix D.

5-12.3 Random Uncertainty

Random errors are also referred to as precision errors. Random errors result from the scatter of data that result from repeated measurements of transient data (e.g., the variability in a wet-bulb temperature reading at a specific location). Precision errors can be reduced by increasing the number of measurement repetitions or by selecting data intervals with greater stability. The population standard deviation σ is a measure of the scatter about the true population mean μ. For a normally distributed population, the interval $\mu \pm 2\sigma$ will include approximately 95% of the population. An estimate of the population standard deviation is the standard deviation of a data sample S_x, which is determined by:

$$S_{X_k} = \sqrt{\frac{\sum_{i=1}^{N} (X_{i,k} - \overline{X}_k)^2}{N-1}}$$

where

N = number of measurements or readings per station (e.g., 60 channel scans of a data acquisition system)

S_{X_k} = standard deviation at the measurement location k

$X_{i,k}$ = time i reading at location k

$\overline{X}_k$ = average value of the measurements at a specific location k

For parameters calculated from multiple instruments, the standard deviation of the group is determined by:

$$S_{\overline{x}} = \frac{\sqrt{\sum_{k=1}^{M} S_{\overline{x}_k}^2}}{M}$$

where

M = number of instruments

$S_{\overline{x}}$ = standard deviation for the group of instruments measuring a single parameter

$S_{\overline{x}_k}$ = standard deviation for a group of instruments used to calculate an average of the same test parameter

The standard deviation of a time-averaged array is calculated by:

$$S_{\overline{\overline{X}}} = \frac{S_{\overline{x}}}{\sqrt{N}}$$

Systematic uncertainties are estimated or calculated for each assumption or parameter that affects the test result. In a similar manner, standard deviations are statistically calculated for thermal measurements that affect the test result. Although not used directly, the uncertainty, in terms of the test measurement, can be calculated from:

$$U_{p,x} = 2 \cdot S_{\overline{x}}$$

where

$U_{p,x}$ = random uncertainty in parameter x

5-12.4 Sensitivity Factors

Sensitivity factors θ relate a change in an independent measured parameter to the resulting change in the test result. These sensitivities are calculated as the partial derivative of the test result with respect to the parameter of interest. The sensitivity may also be calculated numerically as the ratio of the change in the test result to the change in the test parameter.

These sensitivity factors are analytically defined by:

$$\theta_{Pj}^{Cap} = \frac{\partial Cap}{\partial X_j}$$

where

$\theta \frac{Cap}{x_j}$ = sensitivity factor for relating test capability to test parameter j

Sensitivity may also be calculated by employing the central difference theorem. If the data reduction of a parameter (e.g., water flow rate) or test result (e.g., capability) is performed with a computer, the sensitivity of the parameter or test result to supporting measurements may be approximated by making a small change in the supporting measurement and evaluating the change in the parameter or test result. This is represented by the following equation:

$$\theta_{Pj}^{Cap} = \frac{\partial Cap}{\partial X_j} \approx \frac{\Delta Cap}{\Delta XP_j}$$

These sensitivities are easily calculated by manipulation of inputs into computerized data reduction programs or spreadsheets. The test result is calculated using the test parameter plus a small increment and then recalculated using the same value minus the small increment. The change in the test results at each of the two evaluation points is divided by the total difference between the two input values.

5-12.5 Combining Elemental Systematic Uncertainties

The systematic uncertainty of the parameter i is the root-sum-square of all elemental systematic uncertainties B_k for all K sources:

$$B_i = \sqrt{\sum_{k=1}^{K} B_k^2}$$

Uncorrelated systematic uncertainty occurs when the systematic uncertainties for a parameter arise from independent sources. Because they are independent, an error in one measurement introduces no errors in the measured value of other parameters. Uncorrelated uncertainties are combined using the square root of the sum of the squares. The systematic uncertainty of the test result for J parameters, when all elemental systematic uncertainties are uncorrelated, is:

$$B_R = \sqrt{\sum_{i=1}^{J} (\theta_i B_i)^2}$$

where

θ_i = sensitivity factor relating the specific parameter to the result

The ASME Test Code for Test Uncertainty, PTC 19.1-1998, introduces the concept of correlated systematic uncertainty. Systematic uncertainties are correlated when components of the uncertainty in two parameters arise from the same source. Examples of correlated errors include using the same device to measure different parameters or calibrating different parameters against the same standard. One common example for cooling tower tests includes the use of a single Pitot tube to measure the circulating water flow rate in multiple risers. For the case where the result is determined from two parameters that have correlated systematic errors, the systematic uncertainty is calculated by:

$$B_R = \left[(\theta_1 B_1)^2 + (\theta_2 B_2)^2 + 2\theta_1\theta_2 B_{1,2}\right]^{1/2}$$

The terms $B_{1,2}$ are the estimates of the covariances of the systematic uncertainties of parameters 1 and 2. When the systematic errors are uncorrelated for all parameters, this equation reduces to:

$$B_R = \sqrt{\sum_{i=1}^{J} (\theta_i B_i)^2}$$

When the systematic uncertainties for all parameters are perfectly correlated the equation for completely correlated systematic uncertainty reduces to:

$$B_R = \sum_{i=1}^{J} \theta_i B_i$$

When measured test parameters contain both correlated and uncorrelated errors, it is usually easier to combine the correlated terms into a total correlated term and to combine the uncorrelated terms into a total uncorrelated term for the result. For instance, if the circulating water flow rate is measured in two risers with the same Pitot tube, the calibration systematic uncertainty of the Pitot tube in each measurement is perfectly correlated. The spatial systematic uncertainty in the risers can be assumed to be uncorrelated. The total systematic uncertainty is calculated as a combination of the total correlated and uncorrelated terms.

5-12.6 Combining Elemental Systematic Uncertainties Fan Motor Power Example

For a mechanical draft cooling tower with nine fans, the average test fan power is calculated by summing the measured fan power for each fan and dividing by the number of fans.

$$HP_t = \frac{\sum_{k=1}^{9} HP_k}{9}$$

The same instrument is used to make each measurement of fan power. A systematic error due to the calibration of the power meter will be repeated for the power measurement on each fan. Thus, the systematic uncertainties for fan power measurements for each fan are perfectly correlated. Therefore, the systematic uncertainty for the test fan power is

$$HP_t = \sum_{k=1}^{9} \theta_{HP_k} B_{HP_k}$$

and

$$\theta_{HPk} = \frac{1}{9}$$

If the calibration uncertainty of the power meter used for the test measurements is expressed as 2% of the full scale reading:

$$B_{HP_t} = \sum_{k=1}^{9} \left(\frac{1}{9} \cdot B_{HP,k}\right) = .02FS$$

In contrast to this perfectly correlated example, in most cases the systematic uncertainties for measured parameters will be neither perfectly correlated nor perfectly uncorrelated. However, the treatment of partially correlated systematic uncertainties is complex and is usually not justified. To avoid the complexity of the treatment of partially correlated uncertainties, the following general guidelines are offered:

(a) Treat the measurement of different parameters as uncorrelated. In the computation of the systematic uncertainty in the predicted water flow for a cooling tower test, hot water temperature, cold water temperature and wet-bulb temperature should be treated as uncorrelated uncertainties. This is justified because many of the systematic uncertainties common to these measurements tend to be relatively small in comparison to the uncorrelated uncertainties.

(b) Treat measurements of the same parameter as correlated even if these measurements are made using different instruments of the same type. This is justified because many systematic uncertainties for multiple measurements arise from the same source. This rule is always conservative; the uncertainty calculated will always be greater than if the uncertainties were treated as uncorrelated or partially correlated. Using this rule, the systematic uncertainty of each of the wet-bulb temperature measurements would be considered totally correlated when calculating the average wet-bulb temperature. In practice this means instrument systematic uncertainty for any parameter will be equivalent to that of a single instrument measuring that parameter.

An important exception to the general rules occurs when a significant portion of the total systematic uncertainty is caused by spatial variation. Errors due to spatial variations are almost totally uncorrelated and treatment of these uncertainties as correlated may lead to a greatly inflated measurement uncertainty. This most often occurs when flow measurements are made at multiple locations using the same Pitot tube. In this case, the systematic uncertainties should be treated as partially correlated. Further guidance for combining uncertainties of partially correlated parameters is provided in Nonmandatory Appendix D of this Code and in ASME PTC 19.1, Appendix B.

5-12.7 Total Uncertainty of a Measurement

The total uncertainty of a measurement is a combination of the systematic and random uncertainties. The combination of the systematic and random uncertainties is expressed in the following equation:

$$U_{95}^{p} = \sqrt{B_R^2 + (2S_{\bar{\bar{x}}})^2}$$

5-12.8 Total Uncertainty of Result

The total uncertainty in the test result is calculated from the following:

$$U_{95}^{\text{cap}} = \sqrt{\sum_{j}^{k} (\theta_j^{\text{cap}} U_{95}^{j})^2}$$

Section 6
Report of Results

6-1 COMPOSITION OF REPORT

The report of the results of the test shall include as a minimum the following items:

(a) brief summary of the objective, results, and conclusions

(b) representative parties to the test

(c) a description of the cooling tower tested, including its related mechanical equipment

(d) tower thermal design conditions

(e) method of test (including a sketch of the cooling tower and test instrument locations) including, but not limited, to the following items:

(1) speed and direction of wind prevailing during the test

(2) overall dimensions of the installation

(3) piping/riser layout

(4) water flow rate and temperature measurement locations

(5) types of instruments used

(6) description of any methods of measurement not prescribed by the Code

(f) summary of measurements and observations

(g) methods of calculation from obtained data

(h) specified and/or agreed allowances for possible error, including method of application

(i) test results reported as follows:

(1) results computed on the basis of the test operating conditions; only correction applied is for instrument calibrations presented in tabular and graphical form

(2) results corrected to specified conditions if the test operating conditions have deviated from those specified presented in tabular and graphical form

(j) discussion of test, its results, and conclusions

(k) supporting documentation/information required to make the report complete such as

(1) appendices and illustrations to clarify description of the equipment, methods, and circumstances of the test

(2) descriptions of methods of calibration of the test instruments, calibration certificates

(3) sample test result calculations

(4) data sheets and applicable performance curves

(5) raw data as recorded during the test

(6) calibration checks performed on site

(7) uncertainty analysis

6-2 REPORT DATA

Nonmandatory Appendix L provides reporting guidelines for typical test parameters and other pertinent information for wet mechanical and natural draft cooling towers, closed-circuit evaporative wet coolers, wet–dry cooling towers, and wet surface air-cooled steam condensers.

Copies of the final test report shall be distributed to the test parties. Tests performed by independent agencies shall not be distributed beyond the official parties to the test.

NONMANDATORY APPENDIX A
PERFORMANCE MONITORING

A1 INTRODUCTION

This Appendix addresses techniques that permit trending and evaporative cooling equipment performance evaluations during operation. Although the main body of this Code is written for the purpose of acceptance testing, satisfactory performance monitoring can be achieved without the stringent instrument accuracy required for acceptance testing. The reduced need for such a high level of accuracy is what distinguishes the monitoring test plan focus, set-up, and data from acceptance testing. Relative measurements and repeatability are critical. If the data prove to be repeatable during the same operating conditions, correction factors to absolute performance levels can always be developed from an analysis of those data sets.

Historical trending can be handled differently than acceptance testing because less emphasis is placed on the actual measurement accuracy. Although exact values are important, the differences that exist between them are of greater interest. Using the cooling range of a cooling tower as an example, a 0.2°F inaccuracy for a single measurement, though important for acceptance testing, makes little difference if both values are biased in the same direction. In this case, presuming the biases are equal in magnitude, the inaccuracies will cancel out when calculating the difference. Therefore, ordinary operational sensors can be successfully used for trending purposes as long as their biases are considered and accounted for. Accounting for differences in measurements can be accomplished by the installation of test quality sensors and by comparing them to those permanently installed. Once the biases are determined, they can be used to correct the operational values. After the corrections have been incorporated, and incremental changes in the range correspond to operational changes in the range of the tower, the retrieved information can be used to start an historical file on the cooling tower.

The following discussion describes the considerations of evaporative cooling equipment performance monitoring tests.

A2 PERFORMANCE MONITORING TEST STRUCTURE

Performance monitoring can range from periodic to real-time online testing. Implementation of a performance monitoring program will vary significantly between plants and will be based on local needs, economics, and resources, including evaporative cooling equipment performance, instrumentation methods, and methods of data collection and interpretation.

A decision that significantly characterizes an evaporative cooling equipment program is whether to monitor periodically, continuously, or both. The benefits of continuous evaporative cooling equipment performance monitoring are: the knowledge of when changes occur and what the related circumstances were in order to develop the earliest operational or maintenance response; the ability to anticipate if there will be more severe changes from the initial indications; and the continuous assessment of how the evaporative cooling equipment influences the generation, the process, or the costs. Nevertheless, a compromise may be considered that balances the one-time high capital costs and maintenance cost of the continuous system's permanent instrumentation against the repetitive set-up costs and data collection of the periodic test. It should also be recognized that more complex and reliable levels of performance monitoring require increased quantities of instrumentation.

A3 MONITORING PARAMETERS

The following recommended monitoring parameters are listed in a general order of importance; however, the actual order of the list is always dictated by the program's overall objectives:

(a) ambient local wet-bulb and dry-bulb temperature
(b) tower approach terminal temperature difference
(c) tower approach deviation from design
(d) relative or actual circulating water flow
(e) range of cooling
(f) wind direction and approximate speed
(g) estimated recirculation if applicable
(h) makeup, blowdown
(i) generation or applicable plant output
(j) fan power, if applicable

A4 MONITORING MEASUREMENTS

If a modern plant DCS is not available, a recorder with a computer interface is recommended. Computers with data-logging capability can also be used. Manual readings using local instrumentation, although not recommended, are an alternative. A formal data sheet should be constructed so that no readings are left out. Data sheets should be filled out on a daily basis to establish the necessary historical trending.

The requirements for acceptance test measurements, described in the main body of this Code, can be slightly relaxed and adapted for performance monitoring, as long as the sensor in question is still sufficiently accurate to reliably reflect the same relative test value as conditions change. The following discussion and Table A4 below apply. Several notes are relevant to Table A4. Most installed plant flow devices are not sufficiently accurate to serve as a primary flow measurement device, so in order to monitor the flow accurately, it is a key requirement to calibrate plant devices during an accurate test. For example, correlate pump TDH, a self-averaging Pitot tube, or Pitot tube center point measurements, with previous full traverses of a Pitot tube as the circulating water flow is varied by throttling a valve or the line-up of the pumps is changed. With regard to pressure, water temperature, and air temperature measurements, refer to sections of this Code or the supporting 19 Series PTC Codes for instrumentation choices. Some new instrumentation is likely to be a requirement for a successful monitoring program.

A5 CALCULATIONS

Refer to Section 5 for details of the computations of the parameters for trending. All variables are recommended to be plotted with respect to time, such as percent capability, cold water approach, range, or wet-bulb temperature. Normalize the data with respect to design capability or approach. Benchmark milestone conditions such as cleaning the fill or tube bundle, revising the water-treating program, or adjusting the fan blade pitch or the individual cell water distribution.

To ensure data validity, examine the statistical data variation. The data should be precise, consistent, and dependable. Suitable approximations can be made, depending on the experience of the personnel and program goals.

Table A4 Performance Monitoring

Measurement	Code Requirement	Performance Monitoring Method	Potential Caveats and Inaccuracies
Tower cooling water flow	Pitot tube traverse	Ultrasonic meter, Pitot tube centerpoint, self-averaging Pitot tube, pump TDH and curve, heat balance method, electromagnetic flowmeter, differential pressure devices	Nonrepresentative velocity profile or large vorticity at location of meter; scale build-up in line; out of round pipe diameter; unknown open channel flows; periodic blockage of sensing ports by debris; ongoing pump deterioration; inaccurate hot or cold water temperatures; inaccurate gauge or correction to pump *C/L*; out of calibration transducers
Tower water temperatures	Arrayed probes, full flow bleed stream	Submerged sensors in flowing hot water return flumes or discharges, calibrated thermocouples, thermowells on CW lines downstream of pump or at the condenser	Nonrepresentative temperature profile near the basin pump; shallow or un-insulated thermowells; inadvertent contact of submerged sensor with boundary or wall; poor calibrations
Air temperatures	Proximate, accurate, mechanically aspirated psychrometers upwind and downwind	Station meteorological tower measurements, local airport measurements, upwind and downwind ground level psychrometers or dew point readings, sling psychrometers	Influence of sunlight; poor, shadowed location; influence of local water body or terrain vs. a distant weather station; poor water quality on wicks; influence of local wind on plant structures and heated exhausts; widely variable wind direction; poorly maintained or unknown equipment calibrations
Wind speed	Remote reading meteorological anemometer and wind vanes of various designs at proper height	Same as Code except closer to ground level and not as close to tower	Local influence of terrain and ground; in the case of natural draft, poor correlation estimates of wind speed at top of shell; poor instrument quality; effect of weather and time on basic meter correlation
Fan power	Local wattmeters or motor voltage and amperage	Same as Code but remotely recorded	Poor inherent instrument quality; changing motor efficiencies; poor calibration
Evaporative cooler inlet and outlet temperatures	Thermowells per PTC 19.3	Calibrated thermocouples in thermowells on process fluid lines before and a few diameters downstream of cooler	Nonrepresentative temperature profile near the basin pump; shallow or uninsulated thermowells; inadvertent contact of submerged sensor with boundary or wall; poor calibrations
Evaporative cooler process flows	Differential meter like orifice or venturi meter; time of travel ultrasonics; Pitot tube traverse	Ultrasonic meter, Pitot tube centerpoint, self-averaging Pitot tube, pump curve with TDH, heat balance method, electromagnetic flowmeter, differential pressure devices	Nonrepresentative velocity profile or large vorticity at location of meter; scale build-up in line; out of round pipe diameter; periodic blockage of sensing ports by debris; ongoing pump deterioration; inaccurate hot or cold water temperatures; inaccurate gauge or correction to pump *C/L*; out of calibration transducers
Evaporative cooler spray water flows	Differential meter like orifice or venturi meter; time of travel ultrasonics; Pitot tube traverse	Ultrasonic meter, Pitot tube centerpoint, self-averaging Pitot tube, pump curve with TDH, heat balance method, electromagnetic flowmeter, differential pressure devices	Nonrepresentative velocity profile or large vorticity at location of meter; scale build-up in line; out of round pipe diameter; periodic blockage of sensing ports by debris; ongoing pump deterioration; inaccurate hot or cold water temperatures; inaccurate gauge or correction to pump *C/L*; out of calibration transducers

Table A4 Performance Monitoring (Cont'd)

Measurement	Code Requirement	Performance Monitoring Method	Potential Caveats and Inaccuracies
WSACC steam pressure	Two basket tips per cell not to exceed 10	One basket tip in tube bundle	Water in lines, vacuum leaks, long sensing lines, use of wall taps rather than basket tips; out of calibration or poor initial calibration
WSACC air inleakage	Orifice or rotometer	Orifice or rotometer	Transducer out of calibration; blockage
Recirculation	Large number of wet bulb sensors, all at air inlet planes	Small number of sensors (presumably at air inlets if monitored)	Wind direction and speed changes; insufficient sensors; poor calibrations

NONMANDATORY APPENDIX B ULTRASONIC FLOWMETERS[1]

B1 TRANSIT TIME FLOW METERS

B1.1 Time Difference Type

Flow computed by transit time devices is calculated by measuring the time taken for a burst of ultrasonic energy to migrate across a pipe section. Fig. B1.3-1 depicts a typical installation of a generic type meter configured on a spool piece pipe section.

The governing equations relating the time interval to the other physical parameters are as follows:

$$T_{(ab)} = L/(C + V \cos \theta)$$

$$T_{(ba)} = L/(C - V \cos \theta)$$

where

C = speed of sound in the measured fluid
L = acoustic path length
$T_{(ab)}$ = time interval for the ultrasonic energy burst to go from Transducer A to Transducer B
$T_{(ba)}$ = time interval for the ultrasonic energy burst to go from Transducer B to Transducer A
V = velocity of the fluid
θ = angle of the path with respect to the pipe axis

Simplifying the above relationships yields the following:

$$\Delta T = T_{(ba)} - T_{(ab)} = 2LV \cos \theta / C$$

where
$\Delta T = T_{(ba)} - T_{(ab)}$

Further simplification finalizes the equation to:

$$V = L\, \Delta T / 2 \cos \theta\, T_A^2$$

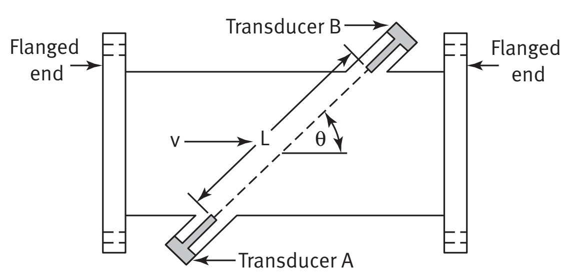


Fig. B1.3-1 Integral Transducer

where
T_A = average transit time between the transducers

Once the cross-sectional area of the pipe section or spool piece is known, the product of the corresponding area and velocity will yield volumetric flow rate.

B1.2 Frequency Difference Type

The metering concept for this type of device maintains one ultrasonic energy-generating module (also known as oscillator) at a frequency of $f_{AB} = 1/T_{(ab)}$ and the second oscillator at $f_{(ba)} = 1/T_{(ba)}$.

The velocity is related to the difference in frequency by the following relationship:

$$V = \Delta f\, L/2 \cos \theta$$

B1.3 Flowmeter Construction

The fabrication and corresponding packaging of modern ultrasonic flowmeters consists of an electronics housing for the central processing unit (CPU) module, transducers, and a pipe section. Most designs allow removal of the transducers without interrupting the process flow. A spool piece with integral transducers is one of the more common types of configuration (see Fig. B1.3-1). In the integral transducer type, the manufacturer mounts the transducers to a flanged pipe section or spool piece. The unit can then be calibrated by the manufacturer to customer specifications in a certified flow laboratory. The spool piece becomes an integral part of the hydraulic system and cannot be retrofitted into an existing system. Most manufacturers will supply a transducer assembly capable of being mounted outside of an existing pipe (see Fig. B1.3-2). This type of system can be configured by adjusting the inputs to the CPU module corresponding to pipe diameter, pipe wall thickness, process fluid, percent of solids concentration, process temperature, change of process temperature, compressibility factor, etc. Flowmeters of this type are easily retrofitted and/or temporarily installed into an existing system because no pipe sections need to be installed, thus ensuring that the dynamics and physical properties of the system remain intact. Mounting hardware for the

[1] This Appendix addresses the principle of operation, construction, and field applications for transit time and Doppler types. The objective of this Appendix is to provide further information to those considering the use of this flow measuring instrumentation. For more information, refer to References at the end of this Appendix; Nonmandatory Appendix M, References; and PTC 19.5.

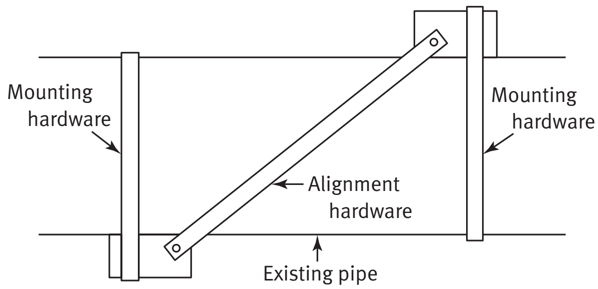


Fig. B1.3-2 Transducer Assembly Mounted Outside of an Existing Pipe

arrangement shown in Fig. B1.3-2 includes silicone gel pads to properly attach the transducer to the pipe and eliminate the possibility of any air gaps.

Several manufacturers have provided transducers and mounting hardware that can be installed into an existing pipe. Holes are drilled into the existing pipe and the transducer mounting hardware is attached by welding or other suitable means. The transducers are then mounted and aligned. Units of this type may be configured by changing the inputs to the CPU only after careful measurements of transducer angle, spacing, and pipe diameter are made.

B1.4 Application Considerations

Common to most meters, ensuring proper operation requires that the spool piece or pipe section be always full of the measured fluid. Manufacturers and corresponding codes will specify the minimum distance from valves, tees, elbows, pumps, and other flow obstructions that will minimize measurement inaccuracies. Typically, 10 to 20 diameters upstream and 5 diameters downstream are required. Since ultrasonic flowmeters rely on an acoustic signal traversing across the pipe section, the liquid must be relatively free of solids and/or air bubbles. Bubbles in the flowstream cause more attenuation of the acoustic signals than solids do. On average, most flowmeters can tolerate a single-digit percentage of solids but only a fraction of a percent of entrapped gas bubbles.

Proper transducer materials and protection must be selected to prevent transducer damage due to chemical action when measuring corrosive fluids. Temperature limitations must also be considered and the appropriate hardware chosen to ensure reliable and accurate flowmeter performance.

B1.5 Performance Specifications and Features

Accuracy, usually specified as a percent of rate, is 1% to 1.5% in laboratory application, but 4% to 6% in typical field cooling equipment applications. Differences can vary significantly with manufacturer, velocity, pipe size,

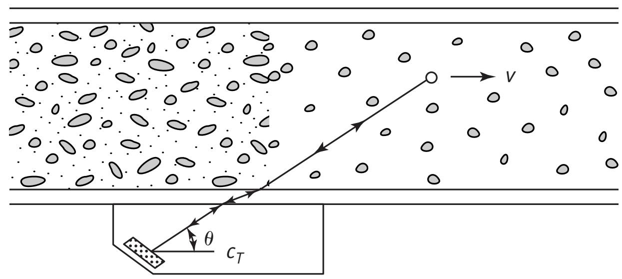


Fig. B2.1-1 Doppler Flowmeter

and process. Most manufacturers calibrate each flowmeter at one or more points under actual flow conditions. Repeatability, usually specified as a percent of rate, is typically better than 0.25% but depends upon the velocity range and manufacturer.

In an attempt to improve performance and accuracy for larger pipe sizes, some suppliers offer flowmeters with two, four, or more pairs of transducers arranged in multiple acoustical paths. The cost of the described configuration is higher than that of a single path flowmeter.

Bidirectional flowmeters, as the name implies, will measure flow in either direction. The display and output of the calculated flow information depends on the manufacturer. The CPU may be programmed to act as a totalizer in order to indicate total flow through the flowmeter. A current output (4 mA to 20 mA) is usually standard, and voltage, pulse train, or other digital outputs may be optionally available depending upon the manufacturer. Alarms for high or low flow are available from most manufacturers and included in most popular models.

B2 DOPPLER FLOWMETERS

Dopler flowmeters are not recommended because they are not sufficiently accurate.

B2.1 Principle of Operation

In a Doppler flowmeter, an ultrasonic wave is projected at an angle through the pipe wall into the liquid by a transmitting crystal in a transducer mounted on the outside of the pipe. Part of the energy is reflected by bubbles or particles in the liquid and is returned through the pipe wall to a receiving crystal. Since the reflectors are traveling at the fluid velocity, the frequency of the reflected wave is shifted according to the Doppler principle (see Fig. B2.1-1).

Combining Snell's Law and the classical Doppler equation, the velocity relationship becomes

$$V = \Delta f\, Ct/(2\, fo \cos\theta) = \Delta f\, K$$

where

Ct = velocity of sound in the transducer

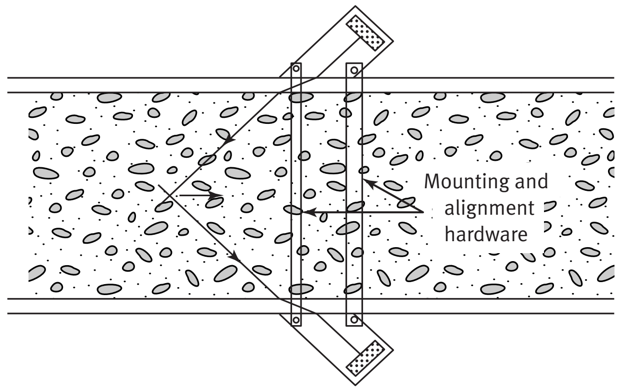


Fig. B2.1-2 Transducer Mounting and Alignment Hardware

fo = frequency of transmission
Δf = difference between the transmitted and received frequency
θ = angle of the transmitter and receiver with respect to the pipe axis

Volumetric flow rate can now be determined by the standard relationship

$$GPM = 2.45\ V(ID)^2$$

where
GPM = flow rate, gal/min
ID = inside diameter, ft
V = velocity, ft/min

The single transducer is the most popular design. Both the transmitter and receiver crystal are contained in a single transducer assembly that mounts onto the outside of the pipe. Alignment of the crystals is specified by the individual manufacturer (see Fig. B2.1-2).

In the dual transducer design, the transmitter crystal and the receiver crystal are mounted separately on the outside of the pipe. Alignment is maintained by a mounting assembly between the transducers, as shown in Fig. B2.1-2.

Each manufacturer will have specific instructions on how to mount the transducer or transducers to the pipe. The acoustic coupling to the pipe and the alignment of the transducer to the pipe must be maintained in spite of pipe temperature changes and vibration. Therefore, the manufacturer's instructions should be followed closely when mounting the transducer or transducers to ensure a stable, reliable installation.

B2.2 Application Considerations

As with transit time and other flowmeters, the pipe must always be full in order to properly indicate volumetric flow. However, a Doppler will indicate velocity in a partially full pipe as long as the transducer is mounted below the liquid in the pipe.

Most manufacturers will specify the minimum distance from valves, elbows, tees, and pumps that will ensure accurate flowmeter performance. Typically, 10 to 20 diameters upstream and 5 diameters downstream are required for relatively clean fluids, but this might change, depending on the process solids concentration or solids composition.

A Doppler flowmeter relies on bubbles or particles in the flow stream to reflect the ultrasonic energy. Most manufacturers specify a lower limit of the concentration and size of solids or bubbles in the liquid for reliable, accurate operation. The flow must also be fast enough to keep the solids or bubbles in suspension, typically 6 ft/sec (1.8 m/s) minimum for solids and 2.5 ft/sec. (0.75 m/s) for small bubbles. On horizontal pipes, the best place to locate the transducer around the circumference of the pipe is not always specified for all applications. The user should rely on the manufacturer's empirical testing, application experience, and instructions for various applications.

Since energy need not go across the entire pipe, the single transducer Doppler can work with wide variations and high levels of solids concentration or aeration. In the Doppler with two transducers, ultrasonic energy must go across the pipe, so some effects on the flowmeter may occur due to wide variations and high levels of solids concentration or aeration.

The Doppler will operate independently of pipe material provided that the pipe is sonically conductive. Such pipes as concrete, clay, and very porous cast iron absorb the ultrasonic energy and may not work with a Doppler. Depending on the manufacturer, some Dopplers will work with lined pipes as long as the liner is well bonded to the inside wall of the pipe.

Transducer temperature limits must also be considered for proper flowmeter operation over the full process temperature range.

B2.3 Performance Specifications and Features

Accuracy is usually specified as a percent of span. Typically, it is 2% to 3% depending on manufacturer, velocity, pipe size, and process fluid. Some manufacturers calibrate the flowmeter at one or more points under actual flow conditions. The inside diameter of the pipe must be measured very carefully to minimize the error in volumetric flow indication, because volumetric flowrate varies as the square of the diameter.

Repeatability is usually specified as a percent of span; typically it is better than 0.5% under simulated flow conditions. Under actual flow conditions it is typically 1% depending on manufacturer, velocity, pipe size, and process conditions.

Bidirectional flowmeters will measure flow in either direction, but they only measure flow magnitude, not direction. Totalizers are available in the form of an optional counter to indicate total flow through the flowmeter in some user-selected units.

Pipe vibration at no-flow conditions can sometimes cause an upscale flow indication due to particle or bubble motion. Some manufacturers simply turn down the sensitivity of the detection circuitry, while others have proprietary circuitry to ensure a zero indication at no-flow conditions.

A current output (4 mA to 20 mA) is usually standard. Voltage or pulse train outputs may be optionally available depending on the manufacturer. Alarms for high or low flow are optionally available, depending on the manufacturer.

B3 REFERENCES

J. Connery, L. DiNapoli, R. Faddick, C. Pouska, C. Punis, "Ultrasonic Velocity Meter." Paper presented at the Sixth International Conference on the Hydraulic Transport of Solids in Pipes, BRHA Fluid Engineering, Cranfield, Beds (UK), (September 1979)

B. C. Liptak, "Ultrasonic Instruments." *Instrumentation Technology* (September 1974)

L. C. Lynnworth, "Clamp-on Ultrasonic Flowmeters." *Instrumentation Technology* (September 1975)

L. C. Lynnworth, "Selected Alternatives to Conventional Ultrasonic Flowmeter." Paper presented at the Ultrasonics International Conference, Guildford, Surrey (UK), (1977)

J. L. Shane, "Ultrasonic Flowmeter Basics." *Instrumentation Technology* (July 1971)

E. M. Zachavias, "Sound Velocimeters Monitor Process Streams." *Chemical Engineering* (January 22, 1973)

NONMANDATORY APPENDIX C TRACER DILUTION METHOD FOR WATER FLOW DETERMINATION[1]

C1 TRACER DILUTION METHODS

C1.1 Constant Rate Injection Technique (Grab Sample)

The constant rate injection method is based on the injection of a tracer dye of a known concentration and at a measured constant rate into the stream being measured. A sample is taken far enough downstream of the injection point to allow for complete mixing. The flow rate is calculated by determining the concentration of a small sample of the downstream, fully mixed, measured fluid. The achievable uncertainty of the constant rate method has been confirmed at a range between 1% to 3%, depending on actual test conditions.

The following equation depicts the relationship associated with the constant rate method (see Fig. C1.1-1):

$$Q_1\,C_1 + Q_2\,C_0 = (Q + Q_1)\,C_2 \qquad (1)$$

where

C_0 = background concentration of measured fluid, ppm
C_1 = tracer dye concentration, ppm
C_2 = stream concentration, ppm
Q = flow of measured stream, gal/min
Q_1 = tracer dye injection rate in, gal/min
Q_2 = secondary transport flow (if required), gal/min

Since C_1 is typically much greater than C_2, the previous equation becomes:

$$Q = \frac{Q_1\,C_1}{(C_2 - C_0)}$$

Using the constant rate injection method requires that the following conditions be met:

(a) Sufficient mixing length must exist between the sampling point and the injection point.

(b) Homogeneous concentrations of the tracer dye must exist at the injection and sampling points.

(c) The tracer dye must be injected at a known constant and accurately measured rate.

(d) Any background concentrations of the tracer dye must be accounted for.

(e) No tracer dye can be lost between injection and sampling point.

(f) Effects of any chemical reducing agents must be calculated and corrections established.

(g) Tracer dye must behave in a known quantitative manner depicted by predictable relationship.

Water flow studies have been successfully conducted using various tracers and dyes. The most popular and predictable include rhodamine WT; rhodamine B; fluorescein; and lithium, sodium, and similar radioactive tracers.

Certain criteria pertain when using the said dyes and tracers. At a minimum, the selected tracers and/or dyes must adhere to but be limited by the following:

(a) mix easily with the measured fluid

(b) cause only negligible and accountable modifications to the measured fluid

(c) be able to be accurately analyzed at the expected concentrations

(d) be detectable at concentrations below the highest permissible for the application

(e) cannot react with measured flow or other substances in piping affecting tracer concentrations

(f) have background concentrations of tracer in measured flow that are correctable and constant

(g) have very low or correctable absorption qualities

One proven method of equipment configuration for the system described is detailed in Figs. C1.1-2 and C1.1-3.

C1.2 Constant Rate Injection Technique (Continuous Sample)

The continuous sample method is identical on the injection end to the grab sample method but differs in how the analyzed sample is handled. This method is more dynamic in nature and can detect disturbances in the system during the course of the test. The flow rate is determined by the output reading of continuous sampling device determining the concentration of a small sample of the downstream, fully mixed, measured fluid. Grab samples can be taken at the beginning and end of the test period to verify concentration levels. The achievable uncertainty of the constant rate method has

[1] This Appendix addresses the principle of operation, construction, and field applications for tracer dilution methods, and its objective is to provide further information to those considering the use of this flow-measuring instrumentation. For more information, refer to PTC 19.5; para. C3, References; and Nonmandatory Appendix M, References.

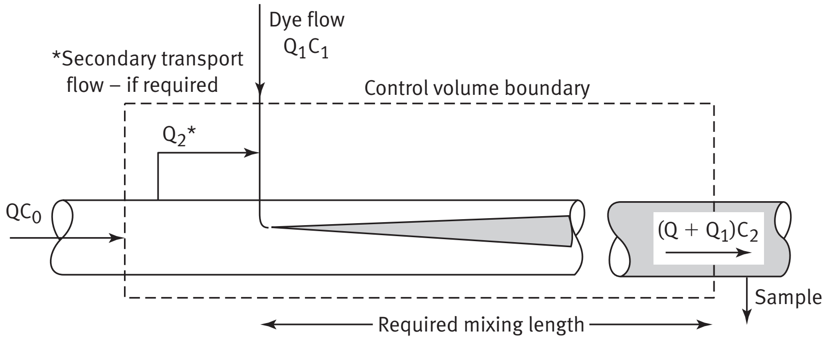


Fig. C1.1-1 Constant Rate Injection Method

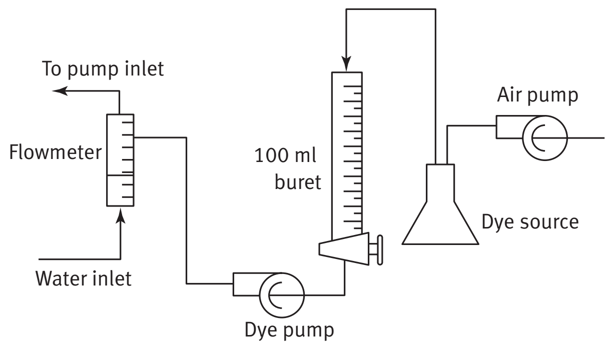


Fig. C1.1-2 Tracer Injection

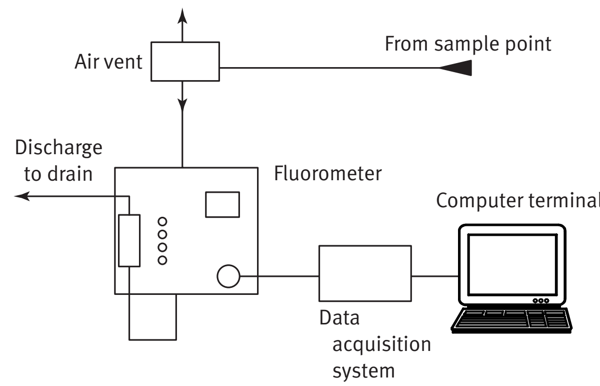


Fig. C1.1-3 Tracer Sampling

been confirmed at a range between 1% to 3%, depending on actual test conditions.

All previous recommended conditions, criteria, and equations pertain to the continuous sampling method.

C1.3 Tracer Injection Configuration

The tracer injection arrangement shown in Fig. C1.1-2 relies on timing the delivery of a precise volume of tracer dye from a calibrated burette to measure the injection rate Q_1. A metering pump delivers the dye to a mixing chamber where a measured carrier flow is introduced to ensure proper delivery of the tracer dye mixture.

The injection flow rate measurement is one of the primary contributors to the overall accuracy of the entire test. Therefore, the ability to accurately measure the flow rate is key to maintaining an overall testing uncertainty within the prescribed range. Burettes, injection pumps, scales, and any other measuring device associated with the injection flow rate measurement should be of the highest field test quality, previously certified through traceable calibrations, and should correspondingly contribute to the expected levels of overall testing uncertainty.

As with the injection flow rate, it is essential that the injected tracer dye solution be homogeneously mixed and its concentration verified before beginning the injection process. Vigorous mixing by means of a mechanical stirrer or a closed loop pumping arrangement can ensure solution homogeneity. The injection solution should be prepared using purified water to ensure that no absorption or other chemical reaction of the dye tracer takes place prior to its introduction into the measured fluid. The injection solution must remain homogeneously mixed throughout the test run. The aforementioned is especially critical during extended testing periods and multiple conduit surveys. The settling of undissolved particles in the solution container during the test run or idle periods must be eliminated. The ambient temperature of the injection solution must remain constant and at the same level experienced during the mixing period. Any variations in temperatures of the solution must be accounted for using applicable temperature-based correction equations. All containers must remain capped to avoid evaporative losses and subsequent changes in concentration.

C1.4 Sampling Configuration

The downstream sampling configuration shown in Fig. C1.1-3 depicts a continuous sample first passing through an air separation chamber and then through a continuous sampling fluorometer. The fluorescent intensity of the sample is recorded, as is its temperature, so that the proper corrections can be applied. The sample

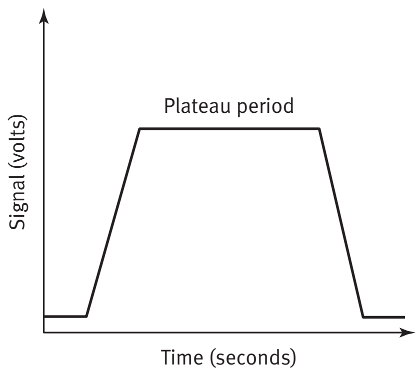


Fig. C1.5 Tracer Sampling Period

stream is then discharged into the draining system, which complies with the environmental requirements of the tested site.

Prior to the start of a sampling period but well within the injection run, samples should be taken across the sampling conduit's cross-section to verify sampling homogeneity. Verification sampling must take place at the start of the plateau period. More than two sampling points are recommended. If homogeneity is not verified, mixing distances or other means must be used to achieve proper mixing.

C1.5 Sampling Period

The sampling period is selected when the tracer dye concentration has reached a homogeneously constant plateau period. A graphical representation of the sampling period is shown in Fig. C1.5.

C1.6 Mixing Length

The section of conduit from the injection point to the sampling point is called the mixing length *L*. Effective mixing can usually take place at an equivalent distance between 100 and 200 pipe diameters from the injection point. Introducing additional turbulence to the measured stream can reduce mixing lengths. This turbulence can be induced by injecting upstream from pumps, elbows, orifices, and other internal piping disturbances.

C2 SAMPLING EQUIPMENT

The operation of a fluorometer is based on directing a beam of light at a select wavelength, which causes the tracer dye in the measured fluid to fluoresce. The wavelength is determined by a color filter placed in front of the light source. A secondary filter is used to absorb the transmitted beam and to pass only the fluorescent light. The concentration of tracer in the sample is linearly proportional to the intensity of light emitted.

The duration of a typical flow test ranges from 15 min to 30 min. Measurements are recorded during the plateau period of the concentration curve and averages are calculated. All appropriate corrections are applied and final flow calculation can be made based on Eq. (1).

When measuring water flows, fluorescence will be decreased when mixing tracer dyes with highly acidic streams. Other factors affecting fluorescence include but are not limited to the following:

(a) absorption of the exciting light beam

(b) absorption of the light emitted by the tracer dye solution

(c) chemical changes in the fluorescent compound

(d) biological absorption of the tracer dye solution

(e) silt and other similarly obtrusive particles in the sampling stream

C3 REFERENCES

ASME Performance Test Code 19.5-2002, Fluid Meters

Publisher: The American Society of Mechanical Engineers (ASME International), Three Park Avenue, New York, NY 10016-5990; Order Department: 22 Law Drive, Box 2300, Fairfield, NJ 07007-2300

I. M. S. Laidlaw and P. L. Smart, "An Evaluation of Fluorescent Dyes for Water Tracing," *Water Resources Research* 13, no. 1 (February 1977):15-33.

NONMANDATORY APPENDIX D
UNCERTAINTY CALCULATION

D1

The calculation procedure for evaluating cooling tower capability is too complex to be represented by an equation or series of equations. Some measured parameters, such as wet-bulb temperature, are used directly in the calculation. Others, such as water flow and cold water temperature, are the result of a series of calculations translating the measured parameter(s) into the capability component. Table D1 provides a breakout of the typical uncertainty constituents that are evaluated as part of a thermal performance uncertainty analysis for a selected window of data.

D2

Because the capability calculation is too complex to be represented by a single analytical expression, the sensitivity of the capability to the capability components must be calculated numerically as discussed in para. 5-12.6.

$$\theta_{P_j}^{Cap} = \frac{\partial Cap}{\partial P_j} \approx \frac{\Delta Cap}{\Delta P_j}$$

The uncertainty of each subset of measurements and calculations are evaluated independently in terms of engineering units and then multiplied by the appropriate sensitivity factor and combined.

Table D2 lists suggested perturbation values for each of the capability components and ranges of sensitivity values.

Some of the capability components, such as hot water and wet-bulb temperature, are directly measured test parameters. Other components, such as water flow, are derived from secondary parameters that are measured. For instance, when the water flow is measured by a Pitot traverse, the water flow rate is calculated from the measured velocity and the pipe cross-sectional area. The pipe cross-sectional area is calculated from the measured pipe diameter. The equations used to calculate the water flow and area are used to calculate sensitivity factors of water flow to the pipe diameter using the derivative form of the sensitivity equation as presented in para. 5-12.6.

$$\theta_D^Q = \frac{\delta Q}{\delta D} = \frac{\delta Q}{\delta A}\frac{\delta A}{\delta D}$$

where

A = area at measurement location
D = pipe diameter
Q = water flow

Similarly, if the measured cold water temperature must be corrected for makeup, the sensitivity of the corrected cold water temperature to the makeup flow and temperature may be calculated using the temperature correction expression provided in para. 5-3.

D3 TEMPERATURE MEASUREMENT SYSTEMATIC UNCERTAINTY

All thermal measurements have a calibration and instrument systematic uncertainty that is unique for each temperature measurement and calibration system. The systematic uncertainty of the temperature measurement includes:

(a) systematic uncertainty of the calibration standard

(b) systematic uncertainty of the instrument(s) used to read the standard

(c) uncertainty represented by the spatial variability of the temperature bath used to calibrate the temperature sensors

(d) uncertainty of the measurement system used to read the temperature sensors during calibration and during the test

(e) lack of fit of the sensor calibration equation

With quality precision RTDs or thermistor calibrated against a NIST traceable standard, an accuracy of 0.20°F may be achieved. To meet this level of accuracy, great care must be taken in the selection of instruments, including a pretest multiple-point calibration of all sensors. The calibration range must extend over the range of test measurements for which the temperature device is used.

D3.1 Cold Water Temperature Uncertainty

When the cold water temperature is measured in a well-mixed location, the cold water temperature systematic uncertainty is of the temperature measurement system (for the system described above, $B_{TCW} = B_T =$ 0.20°F). When the cold water temperature is measured in an equal area grid, the spatial uncertainty can be calculated directly by utilizing previously detailed techniques. If makeup enters the system in such a manner

Table D1 Typical Test Uncertainty Components

Capability Components	Measured Parameter	Error Source	Systematic	Derived Parameter	Constants
Test water flow rate by Pitot tube traverse	Traverse point velocity	B, Sp	Function of calibration laboratory standard, spatial variation at calibration location, coefficient variation with velocity, 2% typical	. . .	Pitot coefficient
	Pipe diameter	B	0.125 in.	Corrected area	. . .
Calculated cold water temperature measured with matrix of probes	Water velocity at temp. sensor location	B, Sp	See Note (1)	. . .	. . .
Calculated cold water — all measurement techniques	Raw CW temp.	B, P	See Note (1)	Corrected CW temp., range	. . .
	Makeup flow [Note (2)]	B		. . .	. . .
	Makeup temp. [Note (2)]	B	See Note (1)	. . .	. . .
	Blowdown flow [Note (2)]	B		. . .	. . .
	Blowdown temp. [Note (2)]	B	See Note (1)	. . .	. . .
Hot water temperature		B, P	See Note (1)	. . .	. . .
Wet-bulb temperature		B, Sp, P	See Note (1)	. . .	. . .
Dry-bulb temperature		B, Sp, P	See Note (1)	(Rel. Humidity) [Note (3)]	
Test fan power	Measured kW or voltage, current, and power factor	B	See Note (4)	Line loss	Motor efficiency
Test fan inlet density	All measured parameters + barometric pressure			. . .	. . .

GENERAL NOTE: B = Systematic, Sp = Spatial, P = Random

NOTES:

(1) This value is calculated from review of the temperature measurement and calibration system.

(2) If the makeup or blowdown streams enter the cold water basin. The systematic uncertainty values for these flow rate measurements are unique to the type of flow rate measurement system. The uncertainty in the cold water temperature due to makeup or blowdown uncertainty is usually negligible.

(3) Dry-bulb temperatures are measured for natural draft towers, forced draft towers, and wet-dry towers.

(4) This value is calculated from review of the measurement device.

Table D2 Capability Sensitivity Factors

Capability Component	Perturbation Increment	Typical Sensitivity Range
Water flow	±5% of test value	1% capability/1% water flow
Cold water temperature	±0.5°F	5–15%/°F
Hot water temperature	±1.0°F	0.5–5.0%/°F
Wet-bulb temperature	±1.0°F	2.0–7.0%/°F
Fan power	±5% of test value	0.33% capability/% fan power
Barometric pressure	±0.5 in. Hg	1–2%/in. Hg

Table D3.3 Calculation of the Cold Water Temperature Uncertainty

Cold Water Temperature	Pump 1	Pump 2	Pump 3	Pump 4	Average
Channel, °C	20.00	21.00	22.00	23.00	
Temperature, °F	84.63	84.37	84.01	84.15	84.29
Standard deviation	0.21	0.14	0.13	0.13	0.40
Standard deviation^2	0.04	0.02	0.02	0.02	
Number of stations (M) =	4				
Number of readings per station (N) =	60				
Group average standard deviation =	0.079				
Time average standard deviation =	0.010				

that the test cold water temperature is impacted, separate sensitivities for the additional flow rate and temperature must be calculated and incorporated into the cold water temperature uncertainty.

D3.2 Combined Cold Water Systematic Uncertainty

The total cold water systematic uncertainty, in terms of measured temperature, is calculated from the combination of the cold water systematic uncertainty and a spatial uncertainty.

D3.3 Cold Water Temperature Standard Deviation

The example in Table D3.3 illustrates the calculation of the cold water uncertainty for a test data set with the cold water measured on the discharge side of four pumps. The group standard deviation (para. 5-12.2) is:

$$S_{\overline{TCW}} = \frac{\sqrt{\sum_{k=1}^{M} S_{TCW_k}^2}}{M}$$

and standard deviation of the time average is:

$$S_{\overline{\overline{TCW}}} = \frac{S_{\overline{TCW}}}{\sqrt{N}}$$

D3.4 Inlet Wet-Bulb or Dry-Bulb Temperature Systematic Uncertainty

For either the inlet wet-bulb or dry-bulb temperatures, a calibration and instrument systematic uncertainty $B_{TWB,\text{cal}\,op}$ value of 0.28°F is assumed, based on a temperature measurement system meeting specifications previously described ($B_T = 0.020$°F) and a standard wet-bulb/dry-bulb instrument as defined in para. 4-4.

D3.5 Spatial Uncertainty

For any cooling tower, spatial systematic uncertainty may be calculated by the formulas previously provided, when based on the entire array of wet-bulb or dry-bulb instruments. Alternately, for a rectangular tower with air inlets on two sides, the spatial uncertainty is calculated according to the following formula:

$$B_{TWB,sp} = \frac{1}{N_{\text{side}}}\sqrt{\left[\frac{S_{sp_{\text{side1}}}\cdot 2}{\sqrt{M_{\text{side1}}}}\right]^2 + \left[\frac{S_{sp_{\text{side2}}}\cdot 2}{\sqrt{M_{\text{side2}}}}\right]^2}$$

where

$B_{TWB,sp}$ = spatial uncertainty for wet-bulb temperature

M_{side1} = number of wet-bulb instruments on side 1 of the cooling tower

M_{side2} = number of wet-bulb instruments on side 2 of the cooling tower

N_{side} = number of inlet planes evaluated for spatial uncertainty (e.g., 2)

$S_{sp_{\text{side1}}}$ = spatial standard deviation for wet-bulb temperature for side 1

$S_{sp_{\text{side2}}}$ = spatial standard deviation for wet-bulb temperature for side 2

The total combined systematic uncertainty in terms of wet-bulb temperature is calculated from:

$$B_{TWB} = \sqrt{(B_{TWB,\text{ Cal Op}})^2 + (B_{TWB,\ sp})^2}$$

The example in Table D3.5 illustrates the calculation of the wet-bulb temperature for a counter flow cooling tower with closed end walls and seven wet-bulb instruments on each side of the cooling tower.

D.3.6 Wet-Bulb or Dry-Bulb Temperature Random Uncertainty

The example in Table D3.6 illustrates the calculation of wet-bulb temperature random uncertainty. For this example, the wet-bulb temperature was measured with 14 instruments. Seven instruments were suspended on two sides of the cooling tower.

The group standard deviation (para. 5-12.2) is:

$$S_{\overline{TWB}} = \frac{\sqrt{\sum_{k=1}^{M} S_{TWB_k}^2}}{M} = 0.185,\ 0.126$$

for Side 1 and Side 2, respectively, and the standard deviation of the time average is:

Table D3.5 Calculation of the Inlet Wet-Bulb Temperature Uncertainty

Side 1								
Channel	42W	43W	44W	0W	1W	2W	3W	
Temperature (°F)	74.85	75.08	76.24	75.40	75.50	75.17	73.75	
Side 2								
Channel	28W	29W	40W	41W	5W	6W	7W	AVG
Temperature (°F)	74.97	74.53	74.68	74.81	74.82	74.88	74.80	74.96

Calibration and operation	Calibration and operation (assumed)		0.283
Spatial uncertainty	Number of sides		2
		Side 1	**Side 2**
	Number of stations per side	7	7
	Sample standard deviation (average station readings)	0.75	0.14
	Spatial systematic uncertainty	0.57	0.11
Combined wet-bulb spatial uncertainty		0.290	
Combined wet-bulb systematic uncertainty		0.405	

Table D3.6 Calculation of Wet-Bulb Temperature Random Uncertainty

Side 1								
Channel	42W	43W	44W	0W	1W	2W	3W	
Temp. (°F)	74.85	75.08	76.24	75.40	75.50	75.17	73.75	
Std. dev.	0.35	0.32	0.35	0.66	0.63	0.57	0.43	Total std.^{2}
Std. dev.^{2}	0.12	0.10	0.12	0.44	0.40	0.32	0.18	Total 1.68
Side 2								
Channel	28W	29W	40W	41W	5W	6W	7W	Average
Temp. (°F)	74.97	74.53	74.68	74.81	74.82	74.88	74.80	75.14
Std. dev.	0.32	0.31	0.31	0.32	0.35	0.35	0.38	
Std. dev.^{2}	0.10	0.09	0.10	0.10	0.12	0.12	0.14	Total 0.78

$$S_{\overline{\overline{TWB}}} = \frac{S_{\overline{TWB}}}{\sqrt{N}} = 0.024,\ 0.016$$

for Side 1 and Side 2, respectively.

The total combined random uncertainty in terms of wet-bulb temperature is calculated from:

$$S_{\overline{TWB}} = \frac{1}{N_{\text{sides}}}\sqrt{(S_{\overline{\overline{TWB}},\ \text{side}_1})^2 + (S_{\overline{\overline{TWB}},\ \text{side}_2})^2} = 0.01°\text{F}$$

D3.7 Hot Water Temperature Systematic Uncertainty

If the hot water temperature is well mixed at the tower, spatial bias may be ignored. Systematic uncertainty is then solely due to the calibration accuracy of the thermal sensors. If the temperature is not well mixed and the hot water is measured in multiple locations, the hot water temperature is calculated by flow weighting the individual temperature measurements. For tests with flow weighting, the sensitivity of the calculated hot water temperature to the water flow measurements must be determined.

D3.8 Hot Water Temperature Standard Deviation

Hot water temperature standard deviation is calculated in the same manner as the inlet wet-bulb and cold water temperature standard deviations.

D4 TEST WATER FLOW RATE UNCERTAINTY

When measured by a Pitot tube equipped with an air-over-water manometer, water flow rate is calculated by the formula:

$$Q = A \cdot \overline{V}$$

where

A = cross-sectional area of the pipe
Q = water flow rate
$\overline{V}$ = average velocity at each of the measurement stations corrected for blockage of the cross-sectional area

and

$$\overline{v} = \frac{\sum_{i=1}^{M} v_i}{M}$$

where
M = number of measurement stations

The velocity (in IP units) at each of the measurement stations is calculated from the following formula:

$$\overline{V}_i = (Def_i / 12 \cdot [2 \cdot 32.2])^{.5} \cdot (A - Block_i / 144)/A \cdot C_p$$

where
$Block_i$ = blockage at the measurement station i, equivalent to the cross sectional area of the Pitot tube in the pipe at the insertion point, in.^{2}
C_p = coefficient of the Pitot tube
Def_i = manometer deflection, in.
Q = water flow rate, gpm

The area of the pipe at the traverse location is determined by:

$$A = \frac{\pi}{4 \cdot 144} D^2$$

(a) When the differential pressure is measured by an air-over-water manometer, the Pitot traverse consists of at least 20 points, and the average velocity is at least 3 ft/sec, the following uncertainties are considered negligible:

(1) random uncertainty due to variations in the manometer deflection

(2) manometer reading uncertainty

(3) blockage uncertainty

(b) The following systematic uncertainties are calculated:

(1) calibration uncertainty-determination of the pitot tube coefficient

(2) pipe diameter measurement

(3) spatial uncertainty for the average velocity

The calculation of each of these uncertainties is detailed in the paragraphs that follow.

D4.1 Calibration Uncertainty Example

Pitot tubes are calibrated by traversing a pipe to determine the apparent velocity while measuring the volumetric flow rate with a reference device. Pitot tubes are calibrated at two or more flow rates, which will yield average velocities spanning the range typically experienced in field measurements. Significant systematic uncertainties include:

(a) uncertainty of the reference device

(b) spatial uncertainty at the measurement location

(c) variation of the Pitot tube coefficient with velocity

The Pitot tube is calibrated by traversing a pipe with the subject pitot tube, while the flow rate through the pipe is determined by a laboratory standard. The average velocity at the traverse section is determined by dividing the flow rate as measured by the standard by the area of the pipe, which has been determined by internal measurements of the pipe diameter. The measurement of the internal diameter is sufficiently accurate that the uncertainty in the area of the calibration section has a negligible effect on the uncertainty of the true average velocity.

The coefficient of the Pitot tube is determined by dividing the average velocity in the pipe by the apparent velocity as determined by the Pitot tube. The coefficient in this example was determined at two average velocities, 5 ft/sec and 9 ft/sec. The coefficient for the Pitot tube is calculated by averaging coefficient for each of the two average velocities. The apparent variation in the Pitot tube coefficient with the average pipe velocity must be included in estimation uncertainty of the Pitot tube coefficient.

The systematic uncertainty in the calibration of a Pitot tube is due to the uncertainty in the standard used to determine the calibration flow rate and the variability in the calculated Pitot coefficient at different velocities. The laboratory performing the calibration of the Pitot in this example used a flow standard with an uncertainty of 0.74%. The Pitot taps at this facility are located 15 diameters downstream from the nearest flow disturbance. Even at this location, significant variation exists between the average velocity calculated for each radius. Review of several calibration reports reveals a spatial uncertainty of 1.5% within the pipe during the calibration traverses. Thus the uncertainty in the Pitot coefficient attributable to the calibration laboratory may be calculated by:

$$B'_{C_p,\mathrm{cal}} = \sqrt{(B'_{C_p,\,Q_{st}})^2 + (B'_{C_p,\,sp})^2} = \sqrt{0.0074^2 + 0.015^2}$$
$$= 0.017$$

where
$B'_{C_p,\,Q_{st}}$ = systematic uncertainty of the calibration flow rate standard, fraction of flow
$B'_{C_p,\,sp}$ = spatial uncertainty in the calibration flow rate traverses, fraction of flow

Because the Pitot tube coefficient changes with Reynolds number (velocity), there is an uncertainty associated with the average Pitot coefficient that is a function of the variability of the coefficient with velocity. At each calibration flow, the velocity at an individual measurement station may vary by up to 2 ft/sec from the average value. There is no way to actually calculate the flow dependence of the coefficient with velocity. Rather, the difference in the coefficient as determined at each of the average velocities is used as an estimate of the systematic uncertainty, due to variation of the actual Pitot tube coefficient with velocity. For the Pitot tube used in this test, the average coefficient determined from two bounding average velocities was 0.810. The coefficients at each of the bounding velocities were 0.807 and 0.813. The uncertainty in the average coefficient due to variability in the bounding coefficients is calculated by:

$$B'_{Cp,\,V} = \frac{(C_{pV1} - C_{pV2})}{C_{p,\,\text{avg}}} = \frac{(0.813 - 0.807)}{0.810} = 0.007$$

where

C_{pV1} = coefficient calculated at the average velocity of the first calibration point V_1

C_{pV2} = coefficient calculated at the average velocity of the second calibration point V_2

D4.2 Total Pitot Tube Calibration Systematic Uncertainty

The total pitot tube coefficient systematic uncertainty on a relative basis is calculated by:

$$B'_{C_p} = \sqrt{(B'_{C_p,\,lab})^2 + (B'_{C_p,\,V})^2} = \sqrt{0.017^2 + 0.007^2} = 0.018$$

The total Pitot tube coefficient systematic uncertainty in terms of flow rate may then be calculated by:

$$B_{C_p} = B'_{C_p} C_p$$

$$B_{C_p} = 0.018 \cdot 0.810 = 0.0148$$

D4.3 Sensitivity of Flow to Pitot Coefficient

The sensitivity of the measured flow rate to the Pitot tube coefficient is:

$$\theta^Q_{C_p} = \frac{\delta Q}{\delta C_p} = \frac{Q}{C_p}$$

D4.4 Systematic Uncertainty in the Measurement of the Pipe Diameter

When the pipe diameter is determined by the methods described in para. 4-2.1(b), the uncertainty in the pipe diameter is considered to be 0.125 in. The sensitivity of the flow to the pipe diameter is:

$$\theta^Q_D = \frac{\delta Q}{\delta D} = \frac{2 \cdot \Pi \cdot D \cdot \overline{v}}{4 \cdot 144} \cdot 7.48 \cdot 60$$

In the following example, the water flow rate was measured in a single conduit with a 84.5-in. internal diameter. The Pitot taps were installed in the main hot water supply conduit but the velocity profile was not symmetrical across the measurement plane. The calculated water flow rate was 177,582 gpm. The average velocity of the radial traverses were:

Vel (fps)	V1	V2	V3	V4
	10.49	10.02	10.42	9.71

Substituting the average value of 10.16 ft/sec into the above equations yields:

$$\theta^Q_D = 4{,}203 \text{ gpm/ft}$$

D4.5 Velocity Spatial Systematic Uncertainty

D4.5.1 Velocity Spatial Systematic Uncertainty — Radial Approach. Spatial uncertainties occur from limited measurements across the measurement plane to calculate the average velocity. This method assumes that the velocity profile in the pipe should be uniform on a radial basis. This is often a fairly good assumption in flow measurement locations in long straight runs of pipe. From PTC 19.1-1998, spatial biases (in terms of velocity) for the water flow rate measurement within a single pipe may be calculated according to the following formula:

$$B_{\overline{V},\,sp} = \frac{2 \cdot S_{\overline{V}_{\text{rad}}}}{\sqrt{N_{\text{rad}}}}$$

where

N_{rad} = number of radial traverses, 4

$S_{\overline{V}_{\text{rad}}}$ = standard deviations of the average radial velocity

$\overline{V}_{\text{rad}}$ = average velocity of each of the radial traverses, ft/sec

For the preceding example, the standard deviation of the average radial velocity $S_{\overline{V}_{\text{rad}}}$ would be 0.37 ft/sec. The spatial uncertainty in the average velocity would be:

$$B_{\overline{V},\,sp} = \frac{2 \cdot 0.37}{\sqrt{4}} = 0.37 \text{ ft/sec}$$

$$B_{Q,\,sp} = \theta^Q_{\overline{V}} \cdot B_{\overline{V},\,sp} \text{ ft/sec}$$

$$\theta^Q_{\overline{V}} = \frac{\delta Q}{\delta \overline{V}} = A \cdot 7.48 \cdot 60 = 17{,}302 \text{ gpm}$$

Based on the radial traverse approach, spatial uncertainty for this example equals:

$$B^Q_{\overline{V}} = 17{,}302 \cdot 0.37 = 6{,}325 \text{ gpm}$$

D4.5.2 Velocity Spatial Systematic Uncertainty — Random Approach. Alternatively, the spatial uncertainty in the water flow may be calculated as if each of the measurement stations is independent from the others and that the overall profile is randomly distributed about a single point. This method is suited for locations with a skewed velocity profile. Spatial uncertainty (in terms of velocity) for the water flow rate measurement within a single pipe can be calculated according to the following formula:

$$B_{V,\,sp} = \frac{2 \cdot S_{\overline{V}}}{\sqrt{M_{\mathrm{Stat}}}}$$

$$S_{\overline{V}} = \sqrt{\frac{\sum_{i=1}^{M_{\mathrm{Stat}}} (V_i - \overline{V})^2}{M_{\mathrm{Stat}\,-\,1}}}$$

where

M_{stat} = number of measurement stations, typically either 20 or 40

V_i = time averaged velocity at each of the measurement stations, ft/sec

The velocity profile at most measurement locations is skewed because of upstream disturbances. If the Pitot location is less than 15 pipe diameters downstream of the nearest disturbance, the alternative calculation based on randomly distributed velocities should be used.

For the previously provided flow rate example, the standard deviation of the 40 point velocities was 0.622 ft/sec. In terms of the water flow rate, the spatial uncertainty calculated for the random distribution of point velocities in the pipe would be calculated by:

$$B_{\overline{V},\,sp} = \frac{2 \cdot 0.622}{\sqrt{40}} = 0.197 \text{ ft/sec}$$

$$\theta_{\overline{V}}^{Q} = \frac{\delta Q}{\delta D} = A \cdot 7.48 \cdot 60 = 17{,}302 \text{ gpm}$$

$$B_{Q,\,sp} = \theta_{\overline{V}}^{Q} \cdot B_{\overline{V},\,sp} \text{ ft/sec}$$

$$B_{\overline{V}}^{Q} = 17{,}302 \cdot 0.197 = 3{,}405 \text{ gpm}$$

Therefore, spatial uncertainty calculated by the randomly distributed value will be used because it is a lower value and a better representation than the radially distributed value.

D4.6 Combined Test Flow Rate Systematic Uncertainty

The combined systematic uncertainty for the flow measurement in terms of the test flow is calculated by:

$$B_Q = \left[(\theta_{c_p}^{Q} B_{c_p})^2 + (\theta_D^{Q} B_D)^2 + (\theta_{\overline{V}}^{Q} B_{V,\,sp})^2\right]^{1/2}$$

For this example:

$$B_Q = \left[(219{,}237 \cdot 0.0148)^2 + (4{,}203 \cdot 0.125)^2 + (17{,}302 \cdot 0.197)^2\right]^{1/2}$$
$$= 4{,}736 \text{ gpm}$$

D5 TEST FAN BRAKE HORSEPOWER

The test fan brake horsepower is calculated from the input motor power and the motor efficiency by:

$$HP_t = \frac{kW_I \cdot \eta_m}{0.746}$$

where

kW_I = motor power measured at the input to the fan motor

η_m = fan motor efficiency

There is no measurement uncertainty associated with motor efficiency. The sensitivity of the fan power to the motor input is:

$$\theta_{kW_I}^{HP_t} = \frac{\delta HP_t}{\delta kW_I} = \frac{\eta_m}{0.746}$$

When the fan motor input power is measured directly with a Wattmeter, the fan input power is calculated by:

$$kW_I = kW_{ob} \frac{E_m}{E_{ob}}$$

The ratio of the voltage at the motor E_m to the measured voltage E_{ob} is very close to 1 since their difference represents the voltage drop between the measurement station and the motor terminals. Therefore,

$$\theta_{kW_{ob}}^{kW_I} \approx 1$$

and

$$B_{kW_I} \approx B_{kW_{ob}}$$

When the fan motor power is calculated by measuring the voltage, current, and power factor,

$$kW_I = \sqrt{3}\, PF\, E_m I_{ob}$$

where

PF = power factor

The following example illustrates the calculation of the test fan power. The average fan power for the cooling tower is computed by

$$kW_I = \frac{\sum_{i=1}^{M} kW_i}{\mathrm{M}}$$

where

kW_I = average input fan motor power, kW

kW_i = input fan motor power for fan i, kW

M = number of fans, 12

Therefore,

$$\theta_{kW_i}^{kW_I} = \frac{\delta kW_I}{\delta kW_i} = \frac{1}{M}$$

Table D5-1 Sample Uncertainty Summary for Each Power Measurement Device

Parameter	Full Scale	Uncertainty % of Full Scale	Parameter Uncertainty
Power factor	1.0	1.0	0.01
Potential	4,000 volts	0.2	8.0 volts
Current	50	2.0	1.0 amps

The fan power for an individual fan is calculated by:

$$kW_i = \sqrt{3}PF_iE_iI_i$$

where

E_i = potential (voltage) for fan motor I, volts
I_i = current for fan motor I, amps
PF_i = power factor for fan motor i

The sensitivity factors for power factor, potential, and current are:

$$\theta^{kW_i}_{pf_i} = \frac{\delta kW_i}{\delta Pf_i} = \sqrt{3}E_iI_i$$

$$\theta^{kW_i}_{E_i} = \frac{\delta kW_i}{\delta E_i} = \sqrt{3}PF_iI_i$$

$$\theta^{kW_i}_{I_i} = \frac{\delta kW_i}{\delta I_i} = \sqrt{3}PF_iE_i$$

In this example, the fan current is measured using current transformers with a ratio of 10:1.

Uncertainty for each power measurement device is summarized in Table D5-1.

The systematic uncertainty in the current meter includes that of the current transformers. The uncertainty in the current measurement was calculated by multiplying the full scale for the meter by the uncertainty of the meter and the turndown ratio for the current transformers. The 12 fan motors are fed by 2 busses with a voltmeter on each bus.

Each of the voltmeters were of the same type, manufactured to the same tolerances as were the current meters and the power factor meters. Each type of meter was used in the same portion of its range. This analysis assumed that the uncertainties between the types of instrument were uncorrelated but that the uncertainties in a single type of instrument are totally correlated. Therefore, the overall systematic uncertainty in the fan power for a single fan motor is calculated by:

$$B_{kW_i} = \left[(\theta^{kW_i}_{PF_i} B_{PF})^2 + (\theta^{kW_i}_{E_i} B_E)^2 + (\theta^{kW_i}_{I_i} B_I)^2\right]^{1/2}$$

The overall systematic uncertainty in the average fan power is calculated by:

$$B_{kW_I} = \frac{1}{M}\sum_{i=1}^{M} B_{kW_i}$$

Table D5-2 includes a partial summary of fan motor power measurement for a 12-fan cooling tower. The table illustrates the uncertainty of the measurements and the sensitivities of the calculated fan power to the voltage, amperage, and power factor measurements that were made with panel instruments in the motor control center.

The average fan motor power systematic uncertainty is 6.26 kW.

D6 SENSITIVITY FACTORS

Table D6 illustrates the calculation of the sensitivity of the test capability to each test parameter. The table lists the average test parameter used to calculate the test capability, the amount that the test value is changed, and the corresponding capabilities when the change is added and subtracted to the test value.

D7 COMBINED TEST UNCERTAINTY

The combined systematic uncertainty is calculated by:

$$B_{\text{cap}} = \left[(\theta^{TWB}_{\text{cap}} B_{TWB})^2 + (\theta^{TCW}_{\text{cap}} B_{TCW})^2 + (\theta^{THW}_{\text{cap}} B_{THW})^2 + (\theta^{BP}_{\text{cap}} B_{BP})^2 + (\theta^{Q}_{\text{cap}} B_Q)^2\ (\theta^{P}_{\text{cap}} B_{HP})^2\right]^{1/2}$$

The total systematic uncertainty for the test was calculated to be 4.32%.

The total standard deviation for the test is calculated by:

$$S_{\text{cap}} = \left[(\theta^{TWB}_{\text{cap}} S_{TWB})^2 + (\theta^{TCW}_{\text{cap}} S_{TCW})^2 + (\theta^{THW}_{\text{cap}} S_{THW})^2\right]^{1/2}$$

The total standard deviation for the test was calculated to be 0.20%.

Table D7 summarizes each of the test parameter systematic and standard deviation components for this example, which are used to calculate the total systematic and random uncertainties in terms of tower capability.

Finally, the overall total systematic and random uncertainty components are combined to yield the uncertainty in the tower capability according to the following equation:

$$U_{\text{cap}} = 2 \cdot \sqrt{\left(\frac{4.32}{2}\right)^2 + 0.15^2} = 4.33\%$$

D8 ADDITIONAL UNCERTAINTY EXAMPLE: PARTIALLY CORRELATED SYSTEMATIC UNCERTAINTY — MULTIPLE FLOW MEASUREMENTS BY PITOT TRAVERSE

In a cooling tower test, flow was measured in two different inlet pipes by conducting 20-point Pitot traverses on each of two perpendicular diameters. Table D8 summarizes the results of the flow uncertainty for each pipe.

Table D5-2 Fan Motor Power Systematic Uncertainty

Fan number	1	2	3	4	5
Voltage	4,247.4	4,247.4	4,247.4	4,247.4	4,247.4
Power factor (PF)	0.805	0.805	0.805	0.805	0.805
Amps	27.72	27.16	28.28	25.79	26.87
Watts	164,173	160,850	167,489	152,749	159,156
Voltage systematic uncertainty (% of FS)	0.20%	0.20%	0.20%	0.20%	0.20%
Full scale (voltage)	4,000	4,000	4,000	4,000	4,000
Voltage systematic uncertainty (V)	8	8	8	8	8
Sensitivity of input kW to V	0.0387	0.0379	0.0394	0.0360	0.0375
kW uncertainty due to V	0.31	0.30	0.32	0.29	0.30
Amperage					
Amperage systematic uncertainty (% of full scale)	2.00%	2.00%	2.00%	2.00%	2.00%
Full scale (amp)	5	5	5	5	5
Turn down ratio	10	10	10	10	10
Amperage systematic uncertainty (I)	1	1	1	1	1
Sensitivity of input kW to I	5.9	5.9	5.9	5.9	5.9
kW uncertainty due to I	5.92	5.92	5.92	5.92	5.92
Power Factor					
Power factor uncertainty	1.00%	1.00%	1.00%	1.00%	1.00%
Sensitivity of input kW to PF	203.9	199.8	208.1	189.7	197.7
kW uncertainty due to PF	2.04	2.00	2.08	1.90	1.98
kW Systematic Uncertainty	6.271	6.257	6.285	6.225	6.251

Result: The average fan motor systematic uncertainty is 6.26 kW.

Table D6 Calculation of the Sensitivity of the Test Capability

Parameter	Average Value	Change ±	Capacity, High	Capacity, Low	Sensitivity
Inlet WB (°F)	74.96	1.00	110.93	99.82	5.56
Hot water (°F)	98.57	0.50	106.86	103.11	3.75
Cold water (°F)	84.29	0.50	99.34	111.16	−11.82
Water flow (gpm)	177,582.00	1,776.00	105.93	103.83	5.913E−04
Fan motor (kW)	158.64	5.00	103.80	106.00	−0.22
Bar pressure (in. Hg)	30.00	0.20	105.12	104.64	1.20
Capability	104.88	. . .	. . .	. . .	. . .

Table D7 Test Parameter Systematic and Standard Deviation Components

Parameter	Spatial	Total Systematic	Standard Deviation	Sensitivity	Capability Systematic Uncertainty	Capability Standard Deviation
Inlet WB (°F)	0.29	0.41	0.01	5.56	2.25	0.08
Hot water (°F)	0.00	0.16	0.01	3.75	0.60	0.03
Cold water (°F)	0.00	0.16	0.01	−11.82	−1.89	−0.12
Water flow (gpm)	3,405	4,736	. . .	5.913E−04	2.78	. . .
Fan motor (kW)	. . .	6.26	. . .	−0.22	−1.38	. . .
Bar pressure (in. Hg)	. . .	0.10	. . .	1.20	0.12	. . .
TOTAL	. . .	. . .	. . .	. . .	4.32	0.15

Table D8 Flow Systematic Uncertainty

Parameter	Units	Variable	Pipe 1	Pipe 2
Flow	gpm	Q_{total}	105,732	106,435
Coefficient uncertainty	gpm	$B_{c_p}^{Q}$	1,935	1,948
Diameter uncertainty	gpm	B_{D}^{Q}	678	673
Spatial uncertainty	gpm	B_{sp}^{Q}	3,538	4,773
Total uncertainty	gpm	B_Q	4,089	5,199
Sensitivity coefficient	. . .	$\theta_{Q}^{Q_{total}}$	1	1
			Total	. . .
Flow	gpm	Q_{total}	212,511	. . .
Covariance term	gpm^2	$B_{Q_1Q_2}$	4,240,939	. . .
Uncertainty	gpm	B_Q	6,930	. . .

The total flow to the cooling tower was computed by summing the flow for both pipes.

$$Q_{\text{total}} = Q_1 + Q_2$$

The sensitivity coefficients relating the total flow to the flow in each pipe are

$$\theta_{Q_1}^{Q_{\text{total}}} = \theta_{Q_2}^{Q_{\text{total}}} = \frac{\delta Q_{\text{total}}}{\delta Q_1} = 1$$

Both measurements were made using the same Pitot tube. The same methods were used to determine the pipe diameter for both pipes. The piping and Pitot tap configurations were sufficiently different to indicate that the spatial distribution of the velocity at the Pitot locations were uncorrelated. This was confirmed by examination of the flow profile. It was decided to treat the systematic uncertainties for the Pitot tube coefficient and the determination of the pipe diameter as correlated uncertainties (e.g., uncertainties arising from the same source). The spatial systematic uncertainties were treated as uncorrelated uncertainties.

Using the methods described in PTC 19.1, Section 8, the total uncertainty is calculated by:

$$B_{Q_{\text{total}}} = \left[(\theta_{Q_1}^{Q_{\text{total}}} B_{Q_1})^2 + (\theta_{Q_2}^{Q_{\text{total}}} B_{Q_2})^2 + 2\theta_{Q_1}^{Q_{\text{total}}}\, \theta_{Q_2}^{Q_{\text{total}}} B_{Q_1Q_2}\right]^{1/2}$$

The covariance term is calculated by:

$$B_{Q_1Q_2} = B_{C_p}^{Q_1} B_{C_p}^{Q_2} + B_{D}^{Q_1} B_{D}^{Q_2}$$

The covariance term is:

$$B_{Q_1Q_2} = 1{,}935 \cdot 1{,}948 + 678 \cdot 673 = 4{,}225{,}674 \text{ gpm}^2$$

The total systematic uncertainty is:

$$\begin{aligned} B_{Q_{\text{total}}} &= \left[(1 \cdot 4{,}089)^2 + (1 \cdot 5{,}199)^2 + 2 \cdot 1 \cdot 1 \cdot 4{,}239{,}439\right]^{1/2} \\ &= 7{,}225 \text{ gpm} \end{aligned}$$

This represents 3.4% of the total flow rate of 212,167 gpm.

If the systematic uncertainties had been totally correlated, the flow rate uncertainty would have been:

$$B_{Q_{\text{total}}} \approx \theta_{Q_1}^{Q_{\text{total}}} B_{Q_1} + \theta_{Q_2}^{Q_{\text{total}}} B_{Q_2}$$

$$B_{Q_{\text{total}}} = 1 \cdot 4{,}089 + 1 \cdot 5{,}199 = 9{,}298 \text{ gpm}$$

or 4.4% of the total flow.

The spatial uncertainty is by far the most significant contributor to the systematic uncertainty of the flow measurement in this example. When the flow is determined by multiple measurements and the spatial uncertainties are uncorrelated, the calculated uncertainty of the flow measurement is significantly reduced by the consideration of partial correlation.

NONMANDATORY APPENDIX E
SAMPLE CALCULATION FOR MECHANICAL DRAFT TOWERS

The following sample calculations are typically used to evaluate the data of a mechanical draft cooling tower, using performance curves submitted by the manufacturer in accordance with para. 3-9.

E1 MECHANICAL DRAFT TOWER CAPABILITY

Table E1-1 presents the design conditions and a set of test conditions for a particular tower. The procedure for calculation of thermal capability is described in para. 5-10.2.

The test data are compared to the requirements of paras. 3-5, 3-6, 3-7, and 3-8 to ensure that the data meet the requirements stated therein.

(a) Determine the fan power at design air density (see para. 5-8) for calculation of adjusted test water rate. The test fan power has been adjusted for nameplate motor efficiency and power factor according to para. 5-8. The tower manufacturer shall also have provided some or all of the following information:

(1) design exit air density, lbm/ft^3

(2) design exit air temperature, °F

(3) design liquid-gas ratio, *L/G*

(4) design air rate, acfm

If design exit air temperature is provided, the air is assumed to be saturated.

Corresponding values of specific volume and specific humidity may be obtained from psychrometric tables or curves; either the design acfm or design *L/G* may be calculated depending on which is provided.

If the design air density is provided along with the design exit air temperature, the design *L/G* or acfm may be calculated by using the tables.

If the design *L/G* is provided, the exit air conditions may be calculated by using the tables.

$$(L)\ (T_{HW} - T_{CW}) = (G)\ (h_e - h_{in})$$

$$h_e = (L/G)\ (T_{HW} - T_{CW}) + h_{in}$$

Table E1-1 Design and Test Conditions

	Design Conditions	Test Conditions
Hot water temperature	104.0°F	96.2°F
Cold water temperature	87.0°F	82.3°F
Wet-bulb temperature	80.0°F	74.2°F
Cooling range	17.0°F	13.9°F
Circulating water rate	60,000 gpm	63,950 gpm
Fan power (per fan)	100 BHP	94.1 BHP
Liquid-gas ratio (*L/G*)	1.173	. . .

where

$$h_e = (1.173)\ (17) + 43.69$$
$$= 63.63 \text{ Btu/lbm dry air}$$
$$h_{in} = 43.69 \text{ Btu/lbm dry air at 80°F (from tables)}$$

Corresponding data (from tables):

Exit air temperature, T_{WB} (saturated) = 95.2°F

$$\text{Specific volume } (V_{da}) = 14.813\ \frac{\text{air-vapor mixture, ft}^3}{\text{dry air, lb}}$$

$$\text{Humidity ratio } (H_e/H_{in}) = 0.0370 \text{ lb water/lb dry air}$$

$$\text{Design exit air density, } \rho_a = \frac{1.0370}{14.813}$$

$$= 0.07001\ \frac{\text{air-vapor mixture, lb}}{\text{air-vapor mixture, ft}^3}$$

Since $q_d = (G)(V_{da})$ and $L/G = (Q_D)(8.33)/G$, then

$$\frac{q_d = (Q_D)(8.33)V_{da}}{L/G}$$

where

$$q_d \text{ (at design)} = (60{,}000)\ (8.33)\ (14.813)/1.173$$
$$q_d \text{ (at design)} = 6{,}311{,}626 \text{ acfm}$$

Calculate exit air conditions at test:

$$h_e = (L/G)(T_{HW} - T_{CW}) + h_{in}$$

Since $G = q_t/V_{da}$ and $L = (Q_T)$ (8.33), then

$$h_e = (Q_T)(8.33)(V_{da}/q_t)(T_{HW} - T_{CW}) + h_{in}$$

Determine the test air rate by using the following relationship:

$$q_t/q_d = (HP_t\ /\ HP_d)^{1/3}$$
$$q_t = q_d\ (HP_t\ /\ HP_d)^{1/3}$$
$$h_{in} = 37.85 \text{ Btu/lbm dry air at 74.2°F (from tables)}$$
$$h_e = \frac{(63{,}950)(8.33)(13.9)(v_{da})}{(6{,}311{,}626)(94.1/100)^{1/3}} + 37.85$$
$$h_e = 1.197\ V_{da} + 37.85$$

Calculate the test exit air properties next, using the above relationship.

Table E1-2 Test Exit Air Properties

Assumed Exit Air Temperature, °F	Specific Volume, v_{da}, ft^3 Air-Vapor Mixture/lb Dry Air	Humidity Ratio, lb Water/lb Dry Air	Calculated Exit Air Enthalpy, h_e, Btu/lb Dry Air	Corresponding Exit Air Temperature, °F
85.00	14.308	0.02642	54.98	89.31
89.31	14.511	0.03048	55.22	89.48
89.48	14.520	0.03065	55.23	89.49

Assume an exit air temperature somewhere between the hot and cold water temperature, obtain the corresponding specific volume (and humidity ratio) from psychrometric tables, and calculate the air exit enthalpy. Compare the corresponding temperature from the tables with the assumed air temperature. Repeat this procedure until the assumed temperature and the calculated temperature are equal, as shown in Table E1-2.

Test exit air density, $\rho_a = 1.03065/14.520$ (at 89.49°F)

$$\rho_a = 0.07098 \frac{\text{air-vapor mixture, lb}}{\text{air-vapor mixture, ft}^3}$$

Since fan power varies directly with air density for a constant system (constant volume) at constant fan speed and constant blade pitch angle, the test fan power is corrected for the difference between the design air density and the test air density as follows:

Corrected test fan power = (0.07001/0.07098) (94.1) = 92.81 HP_t

(b) Determine the adjusted test water rate according to para. 5-2.2.

$$\text{Design Fan Power} = 100\ HP_d$$

$$\text{Corrected test fan power} = 92.81\ HP_t$$

$$\begin{aligned}\text{Adjusted test water rate} &= (63{,}950)(100/92.81)^{1/3}\\ &= 65{,}560 \text{ gpm}\end{aligned}$$

(c) Determine tower capability from manufacturer's performance curves. The manufacturer will have submitted performance curves A through C at 90%, 100% and 110%, of design flow (see Figs. E1-1 through E1-3).

(d) Construct a cross plot no. 1 (Fig. E1-4) from the manufacturer's performance curves by plotting cooling range against cold water temperature at the test wet-bulb temperature (74.2°F). The following predicted cold water temperatures were obtained from cross plot no. 1 at the test cooling range of 13.9°F (see Table E1-3).

(e) Construct test cross plot no. 2 (see Fig. E1-5) by plotting water rates against the cold water temperatures shown in Table E1-2.

(f) The predicted test water rate is obtained from test cross plot no. 2 by entering the curve at the test cold water temperature (82.3°F).

Predicted test water rate = 63,280 gpm

$$\begin{aligned}\text{Tower thermal capability} &= \frac{\text{Adjusted test water rate}}{\text{Predicted test water rate}}\\ &= \frac{65{,}560}{63{,}280}\\ &= 1.036\ (103.6\%)\end{aligned}$$

See also Table E1-3.

E2 PREDICTED TEST COLD WATER TEMPERATURE

The predicted test cold temperature at the test conditions is obtained from test cross plot no. 2 by entering the curve at the adjusted test water rate (65,560 gpm).

$$\begin{aligned}\text{Predicted test cold water temperature} &= 82.62°\text{F}\\ \text{Actual test cold water temperature} &= 82.30°\text{F}\\ \text{Deviation} = 82.30°\text{F} - 82.62°\text{F} &= -0.32°\text{F}\end{aligned}$$

E3 PREDICTED DESIGN COLD WATER TEMPERATURE

Construct design cross plot no. 3 (see Fig. E3) at 17°F cooling range and 80°F wet-bulb temperature by directly reading the cold water temperatures at the three water rates of the manufacturer's curves A through C (see Figs. E1-1 through E1-3). No intermediate cross plot (such as cross plot no. 1) is necessary for the design cross plot since no interpolation is required at design conditions. In Table E3, the predicted cold water temperatures at the respective water rates are the basis for design cross plot no. 3.

The predicted cold water temperature at the design cooling range, wet-bulb temperature, and water based on the tested thermal capability is obtained by entering design cross plot no. 3 at a test-compensated water rate. This test-compensated water rate is simply the ratio of the design water rate to the test capability.

$$\begin{aligned}\text{Test-compensated water rate} &= (60{,}000/1.036)\\ &= 57{,}915 \text{ gpm}\\ \text{Predicted design cold water temperature} &= 86.75°\text{F}\\ \text{Design cold water temperature} &= 87.00°\text{F}\\ \text{Deviation} = 86.75°\text{F} - 87.0°\text{F} &= -0.25°\text{F}\end{aligned}$$

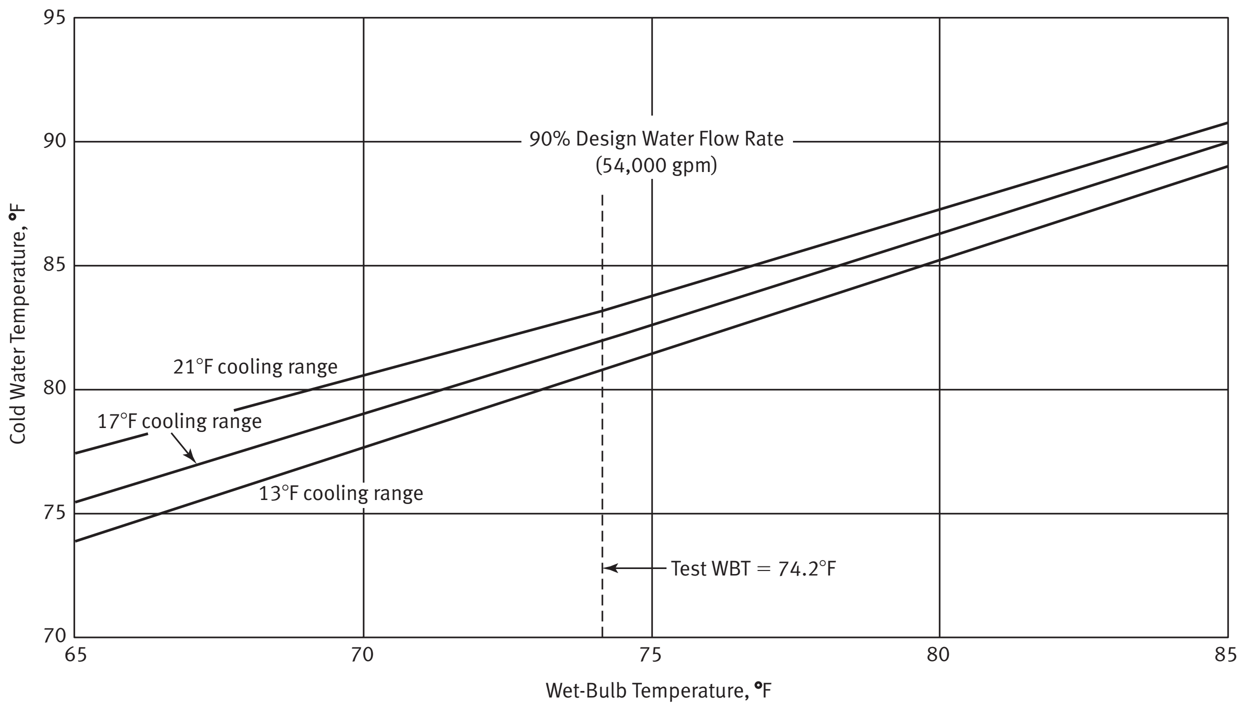


Fig. E1-1 Manufacturer's Performance Curve A

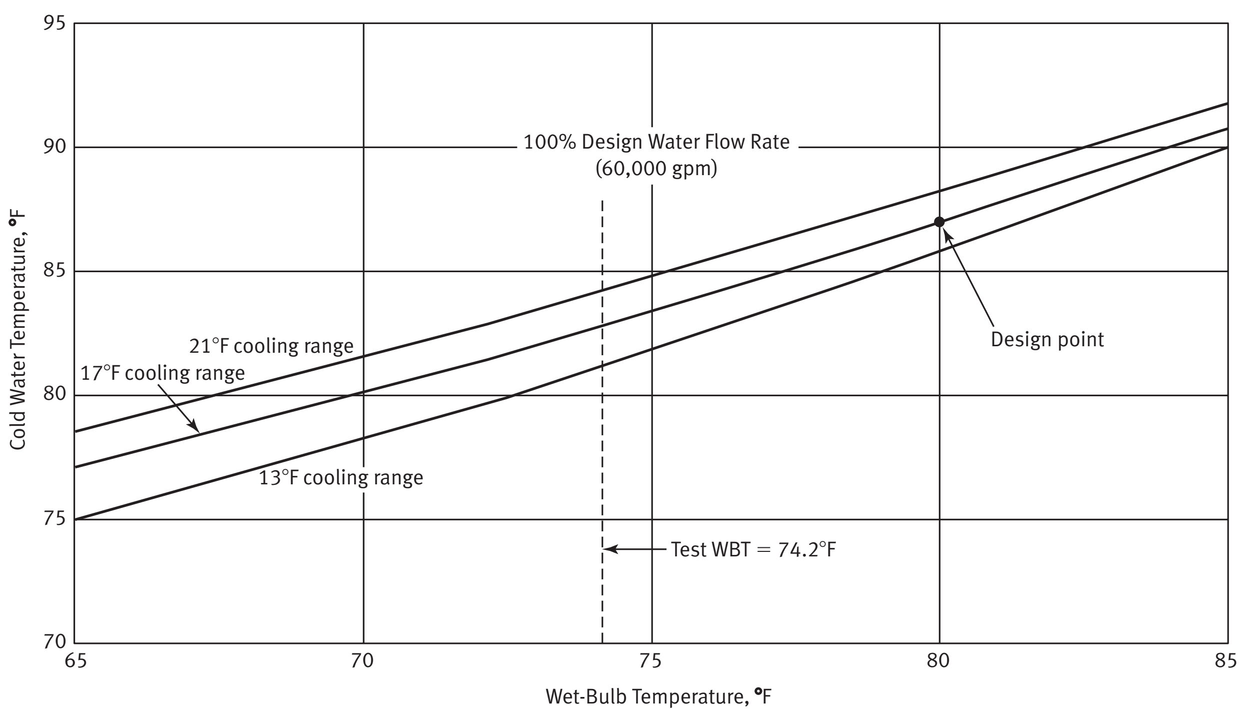


Fig. E1-2 Manufacturer's Performance Curve B

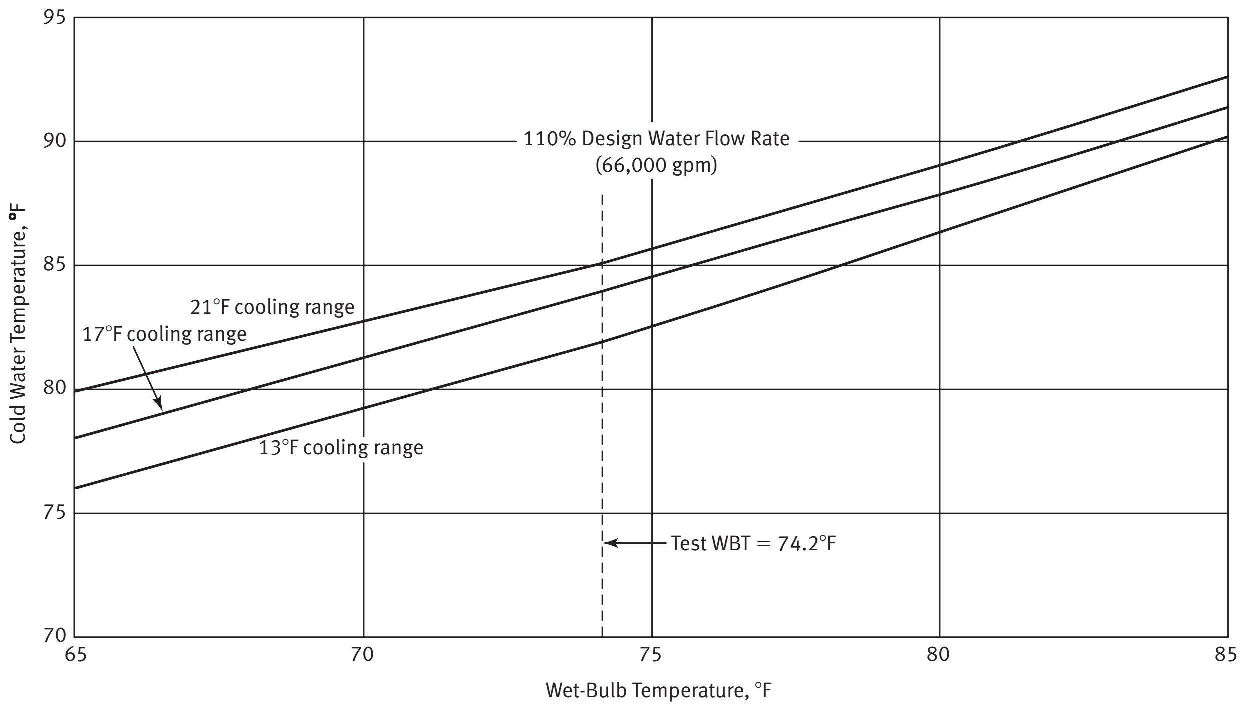


Fig. E1-3 Manufacturer's Performance Curve C

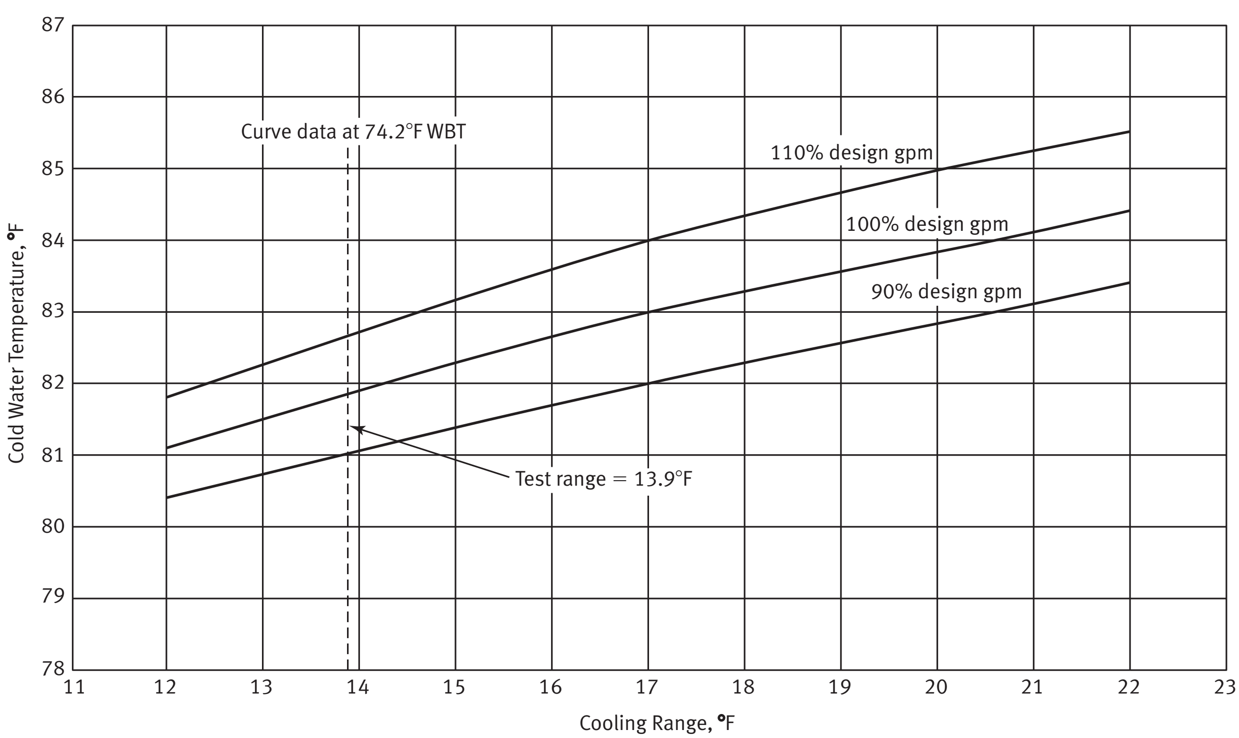


Fig. E1-4 Test Cross Plot No. 1

Table E1-3 Predicted Cold Water Temperatures Cross Plot 1

% Design Water Rate	Water Rate, gpm	Cold Water Temperature, °F
90	54,000	81.00
100	60,000	81.85
110	66,000	82.70

Table E3 Predicted Cold Water Temperatures Cross Plot 3

% Design Water Rate	Water Rate, gpm	Cold Water Temperature, °F
90	54,000	86.20
100	60,000	87.00
110	66,000	88.00

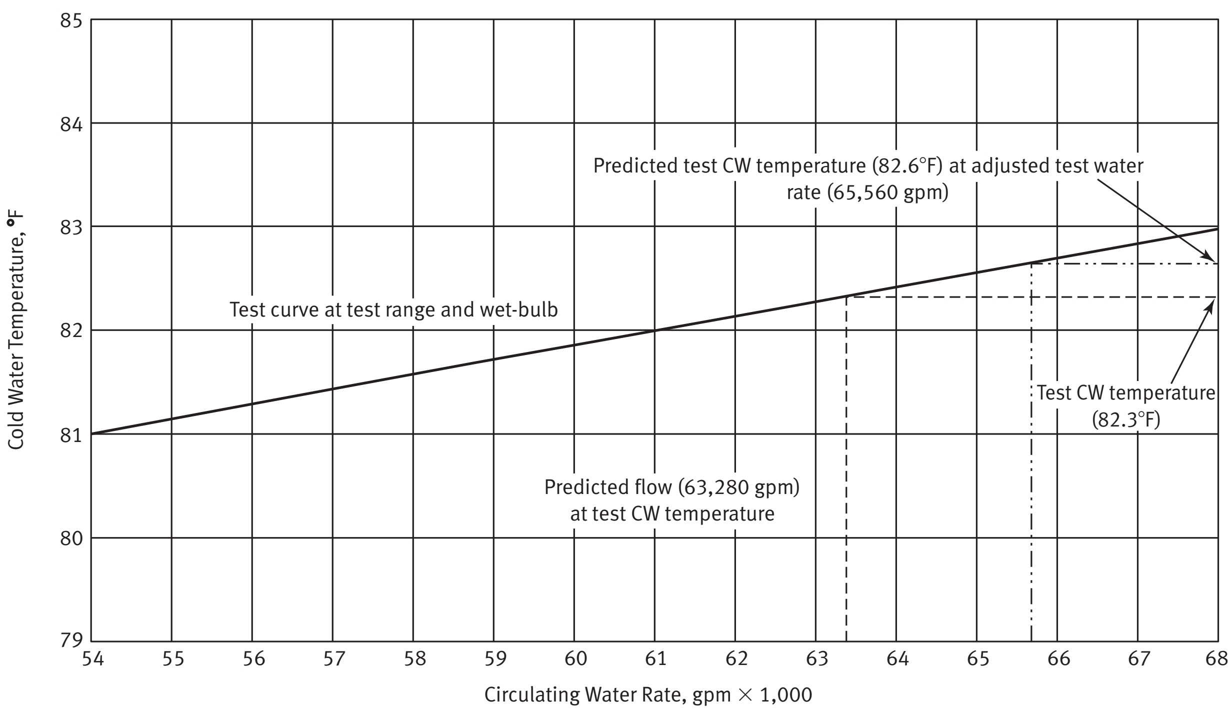


Fig. E1-5 Test Cross Plot No. 2

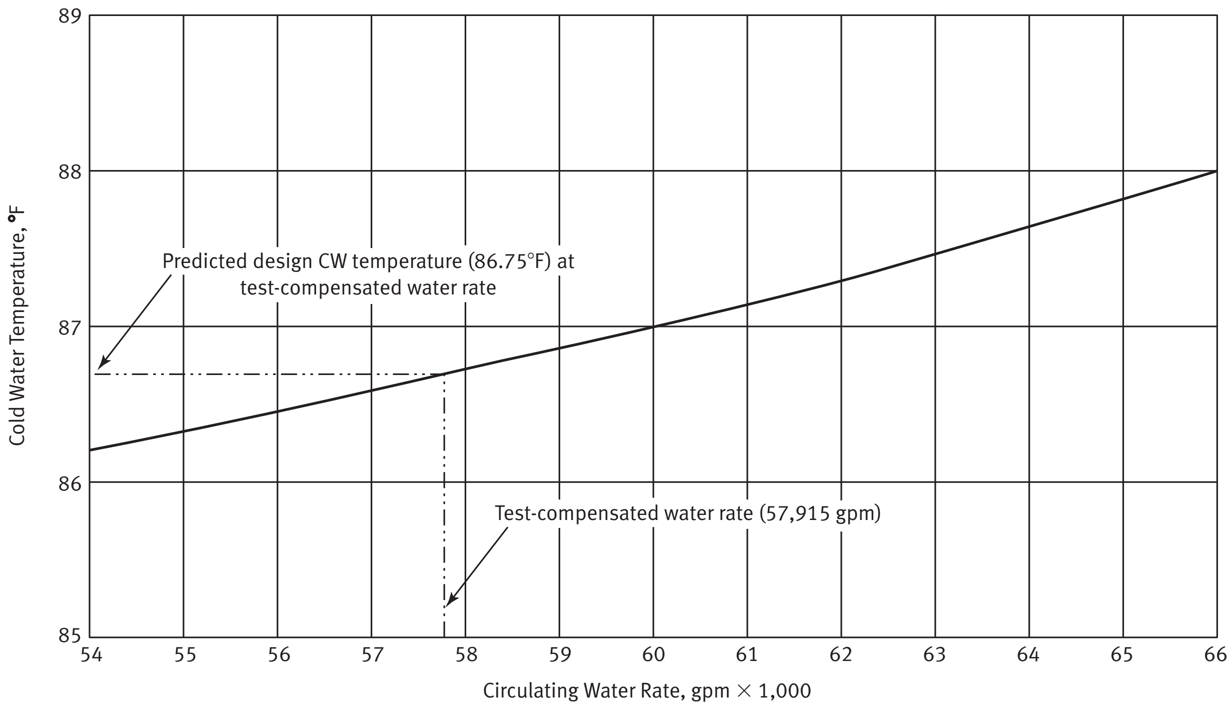


Fig. E3 Design Cross Plot No. 3

NONMANDATORY APPENDIX F
SAMPLE CALCULATION FOR NATURAL DRAFT TOWERS

The following sample calculations are typically used to evaluate the data of a natural draft cooling tower, using performance curves submitted by the manufacturer in accordance with para. 3-9.

F1 NATURAL DRAFT TOWER CAPABILITY

Table F1 presents the design conditions and a set of test conditions for a particular tower. The procedure for calculation of thermal capability is described in para. 5-10.

The manufacturer has submitted performance curves presenting cold water temperature as a function of the ambient air dry-bulb temperature with the relative humidity of the ambient air as a parameter (see Figs. F1-1 through F1-9). Figs. F1-1 to F1-3 correspond to 90% of design water circulating rate; Figs. F1-4 to F1-6 correspond to 100% of design water circulating rate; and Figs. F1-7 to F1-9 correspond to 110% of design water circulating rate.

F2 EVALUATING TOWER PERFORMANCE CAPABILITY

Follow these six steps to evaluate tower performance capability.

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.

Step 2. First Cross Plot. For each flow rate, prepare a cross plot of cold water temperature as a function of the relative humidity, with the cooling range as a parameter (see Figs. F2-1 through F2-3).

Step 3. Second Cross Plot. Using these new curves, enter each curve at the test relative humidity (60.42%) and determine the cold water temperature for each flow rate and range.
Next, for each flow rate, develop a cross plot of the cold water temperature as a function of the cooling range (see Fig. F2-4).

Step 4. Third Cross Plot. Enter Fig. F2-4 at the test range (13.14°F) and determine the cold water temperature for each of the three flow rates.
Next, cross plot the water flow rate as a function of the cold water temperature. See Fig. F2-5).

Step 5. Determine Predicted Flow Rate. Enter Fig. F2-5 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 the equation in para. 5-10.3.1, find the cooling tower thermal performance capability as:

$$\begin{aligned}\%\ \text{Capability} &= (353{,}430\ \text{gpm}/342{,}224\ \text{gpm}) \times 100\\ &= 103.3\%\end{aligned}$$

See also Tables F2-1, F2-2, and F2-3.

Table F1 Natural Draft Cooling Tower Design and Test Data

	Design Conditions	Test Conditions
Circulating water flow	378,000 gpm	353,430 gpm
Hot water temperature	91.0°F	82.04°F
Cold water temperature	77.7°F	68.90°F
Cooling range	13.3°F	13.14°F
Wet-bulb temperature	60.8°F	49.11°F
Dry-bulb temperature	64.8°F	56.12°F
Barometric pressure	29.921 in. Hg	30.570 in. Hg
Relative humidity	79.955%	60.425%

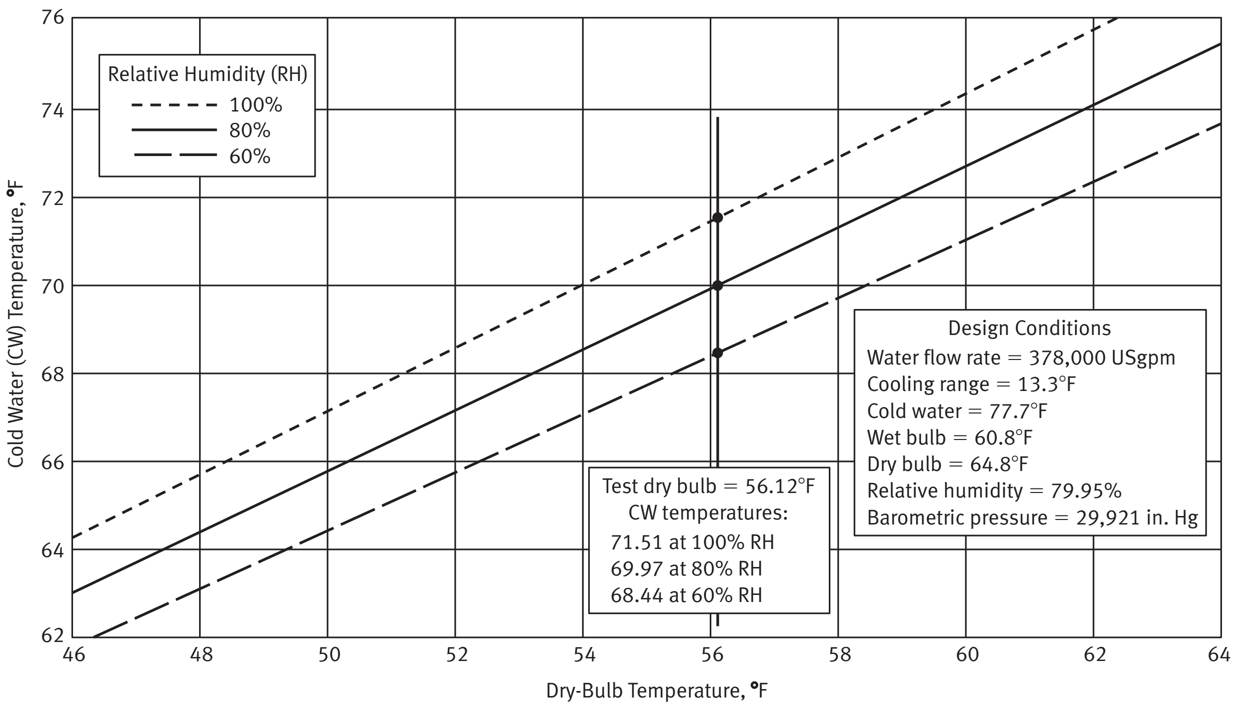


Fig. F1-1 Cold Water Temperature Versus Dry-Bulb Temperature
Water Flow Rate = 340,200 gpm (90%), Range = 12.0°F

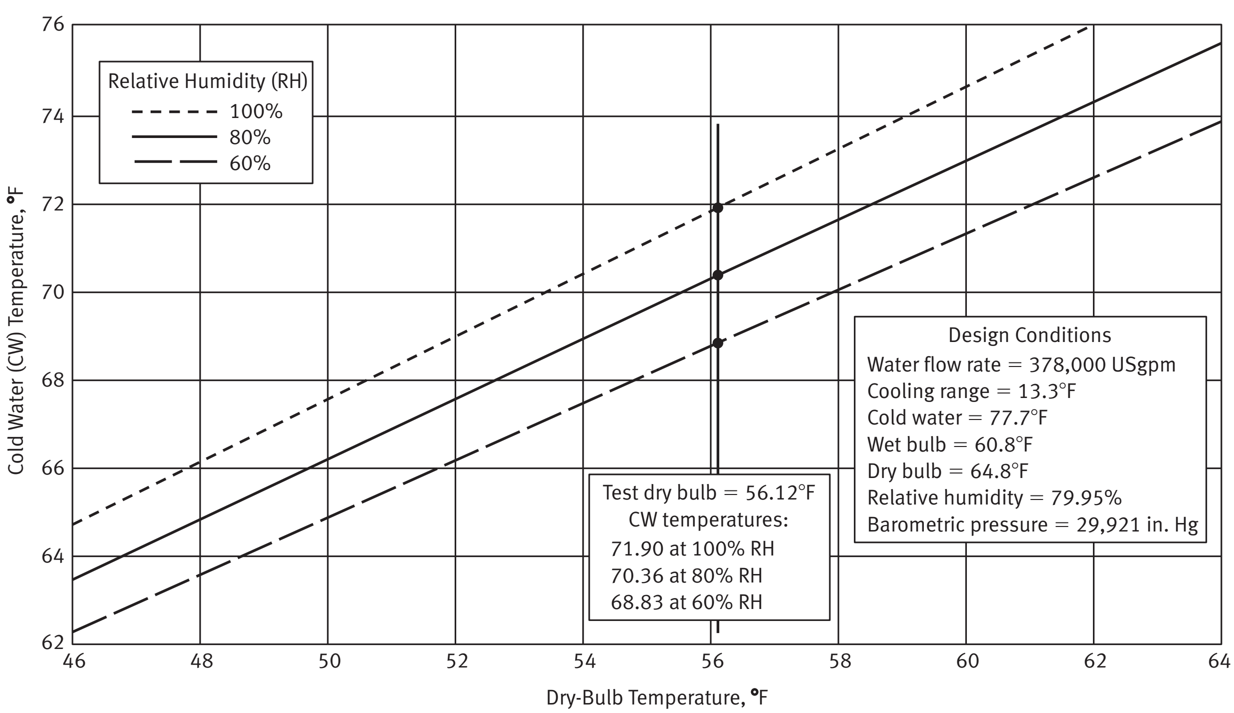


Fig. F1-2 Cold Water Temperature Versus Dry-Bulb Temperature
Water Flow Rate = 340,200 gpm (90%), Range = 13.3°F

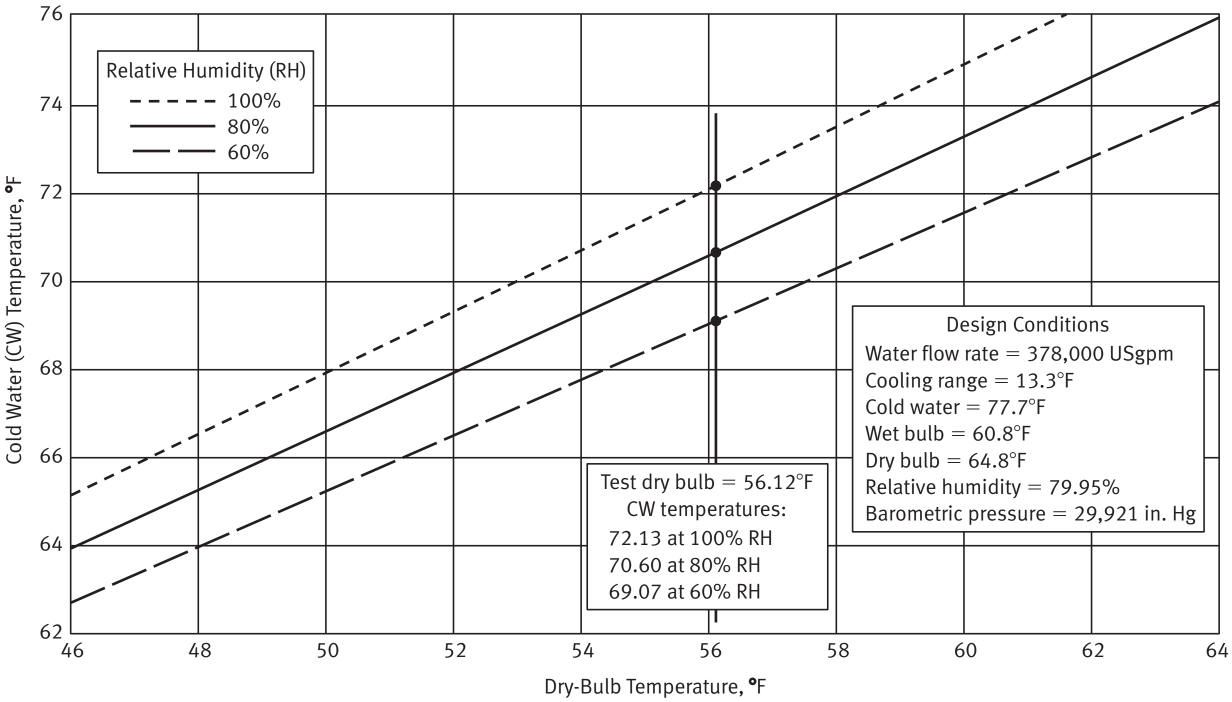


Fig. F1-3 Cold Water Temperature Versus Dry-Bulb Temperature
Water Flow Rate = 340,200 gpm (90%), Range = 14.6°F

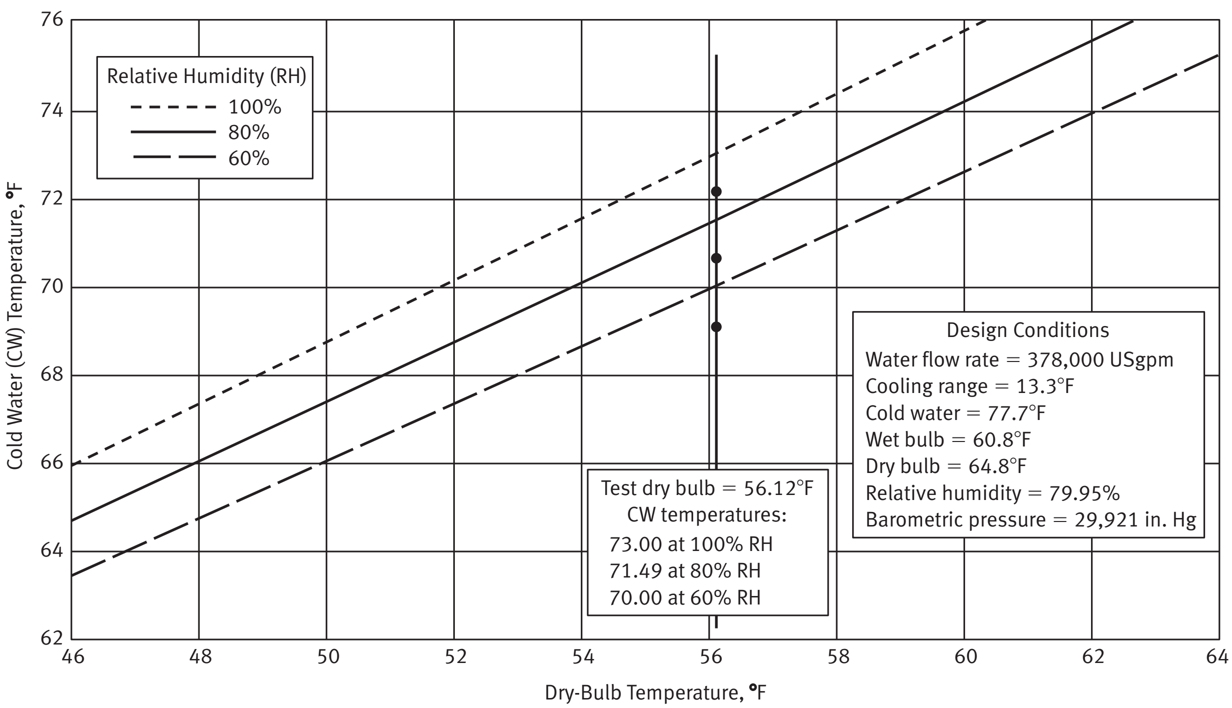


Fig. F1-4 Cold Water Temperature Versus Dry-Bulb Temperature
Water Flow Rate = 378,000 gpm (100%), Range = 12.0°F

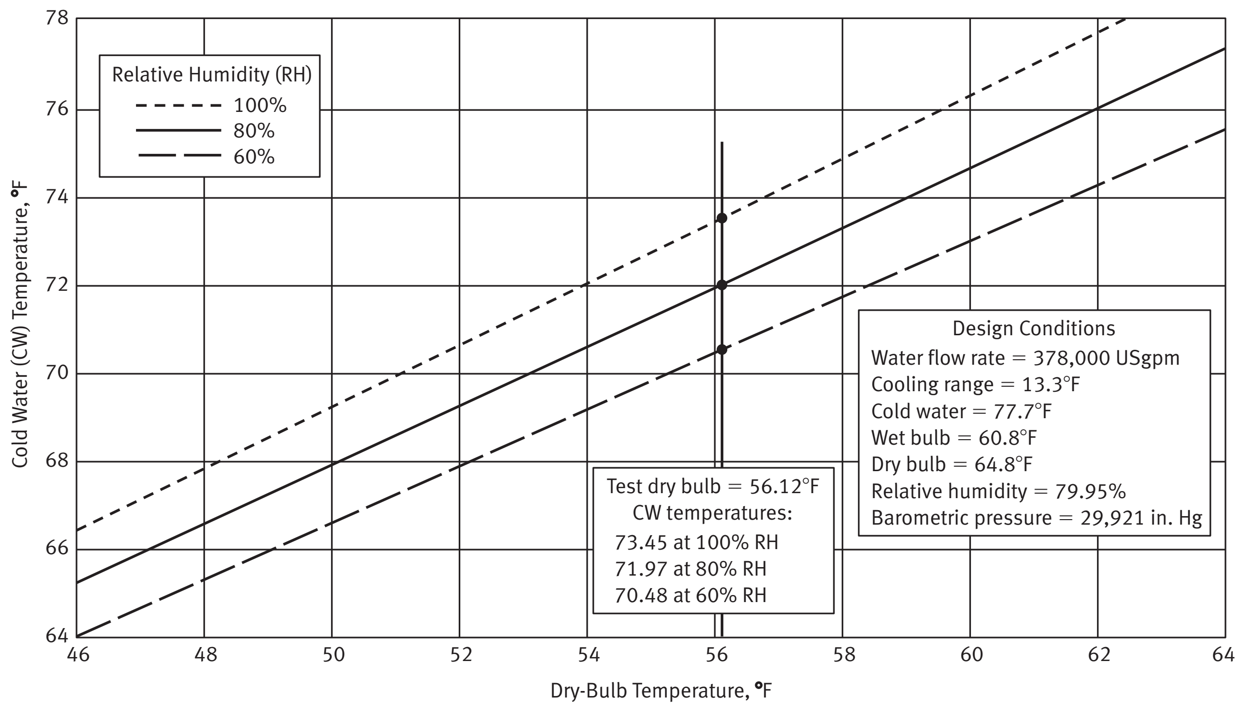


Fig. F1-5 Cold Water Temperature Versus Dry-Bulb Temperature
Water Flow Rate = 378,000 gpm (100%), Range = 13.3°F

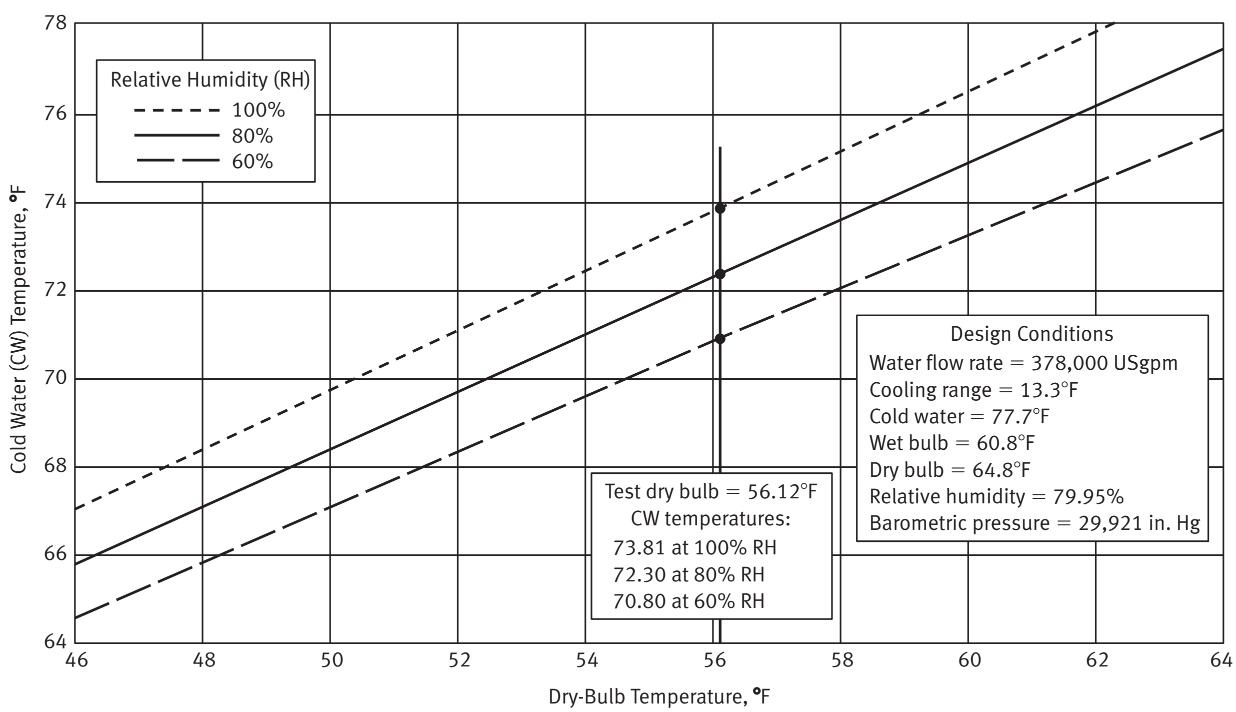


Fig. F1-6 Cold Water Temperature Versus Dry-Bulb Temperature
Water Flow Rate = 378,000 gpm (100%), Range = 14.6°F

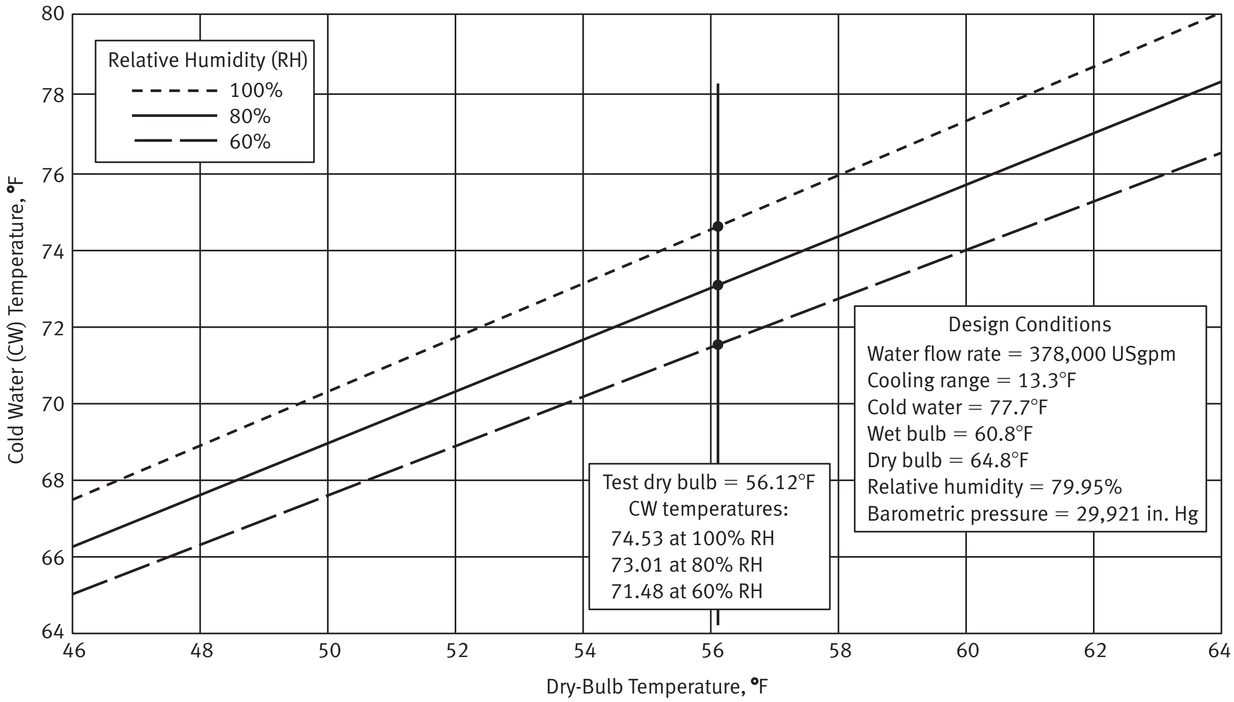


Fig. F1-7 Cold Water Temperature Versus Dry-Bulb Temperature
Water Flow Rate = 415,800 gpm (110%), Range = 12.0°F

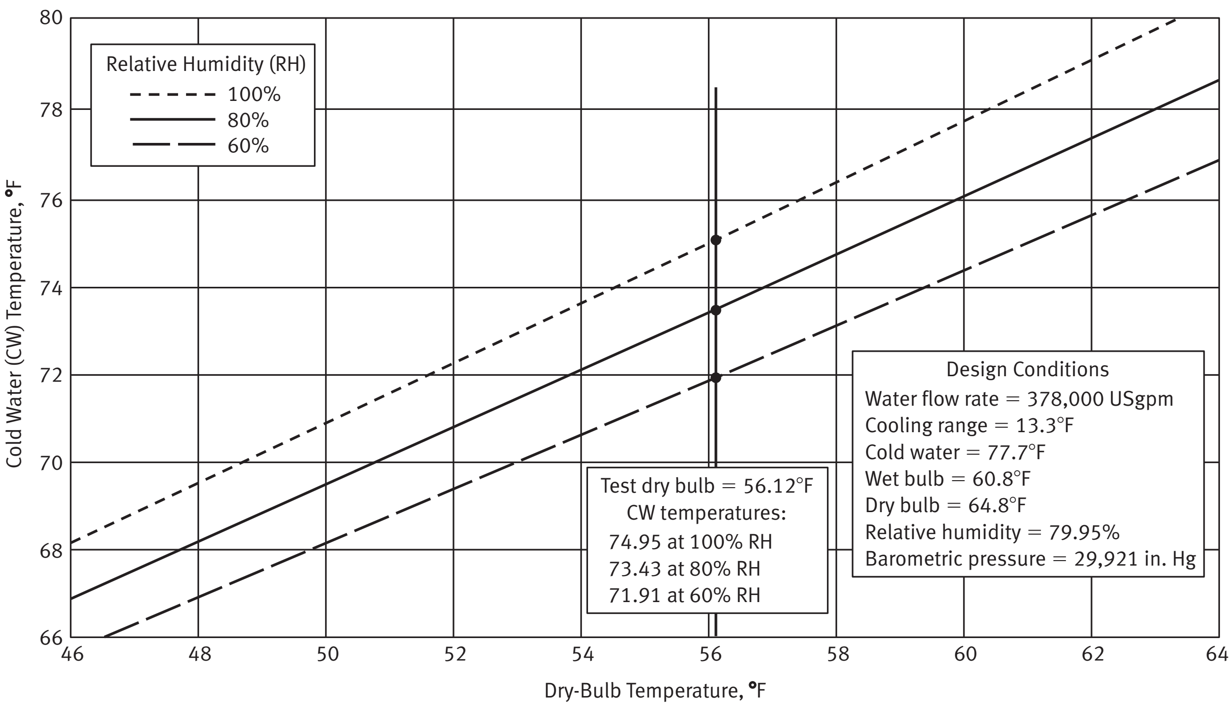


Fig. F1-8 Cold Water Temperature Versus Dry-Bulb Temperature
Water Flow Rate = 415,800 gpm (110%), Range = 13.3°F

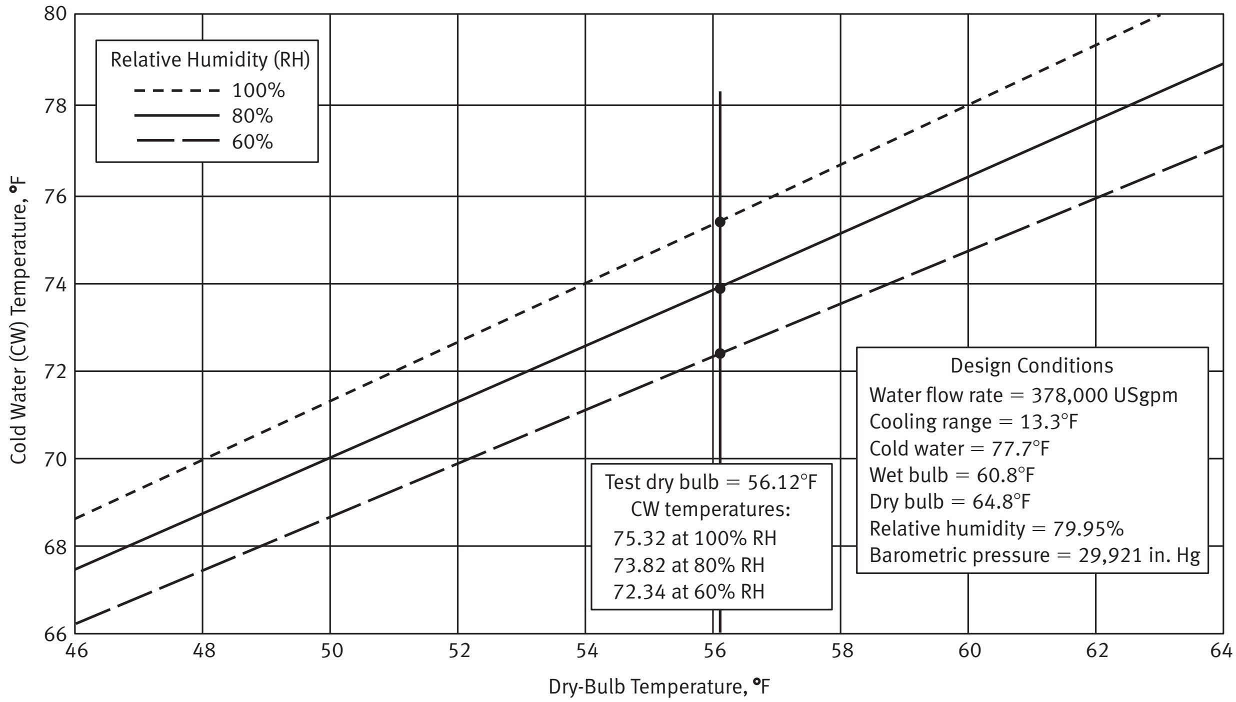


Fig. F1-9 Cold Water Temperature Versus Dry-Bulb Temperature
Water Flow Rate = 415,800 gpm (110%), Range = 14.6°F

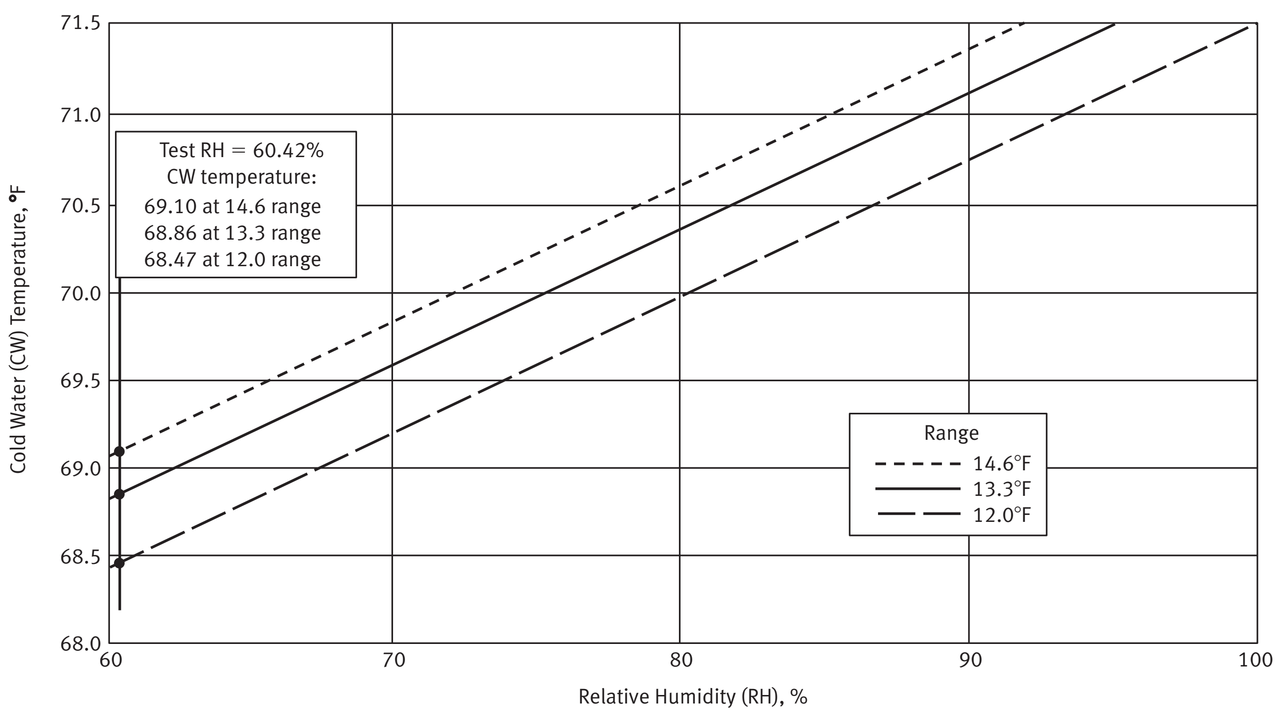


Fig. F2-1 Cross Plot No. 1A
Dry Bulb = 56.12°F, Water Flow Rate = 340,200 gpm (90%)

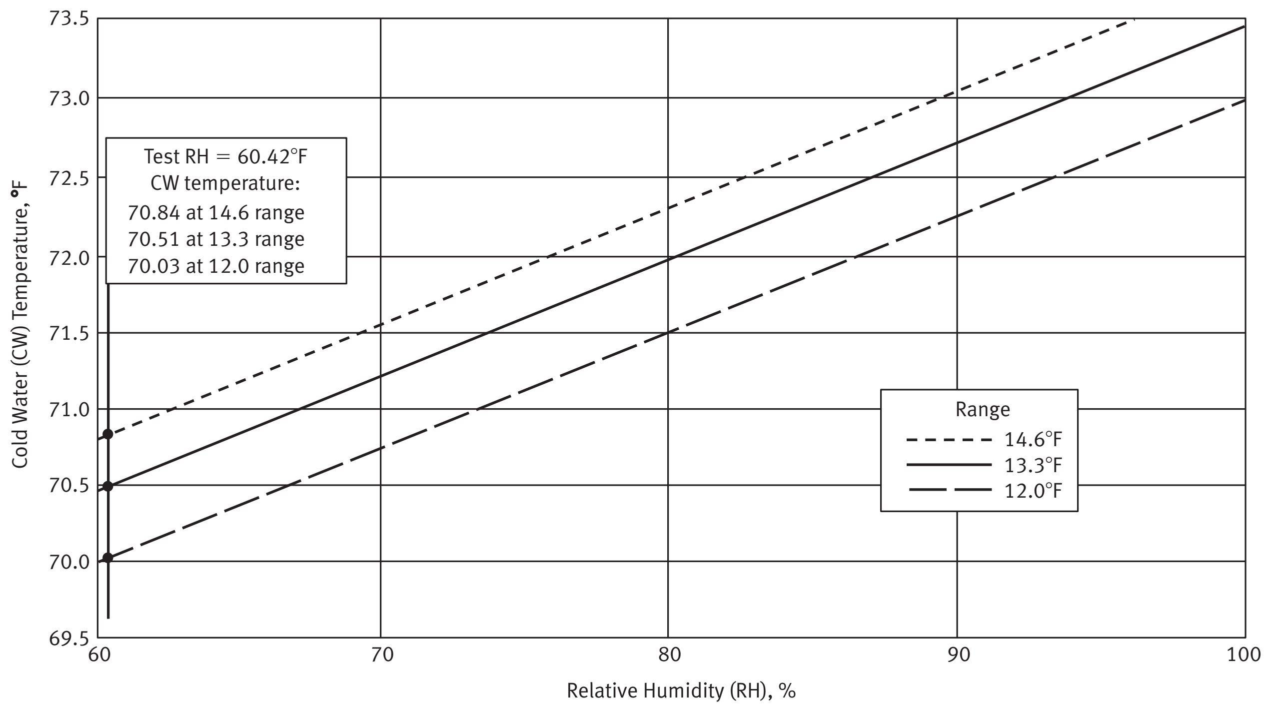


Fig. F2-2 Cross Plot No. 1B
Dry Bulb = 56.12°F, Water Flow Rate = 378,000 gpm (100%)

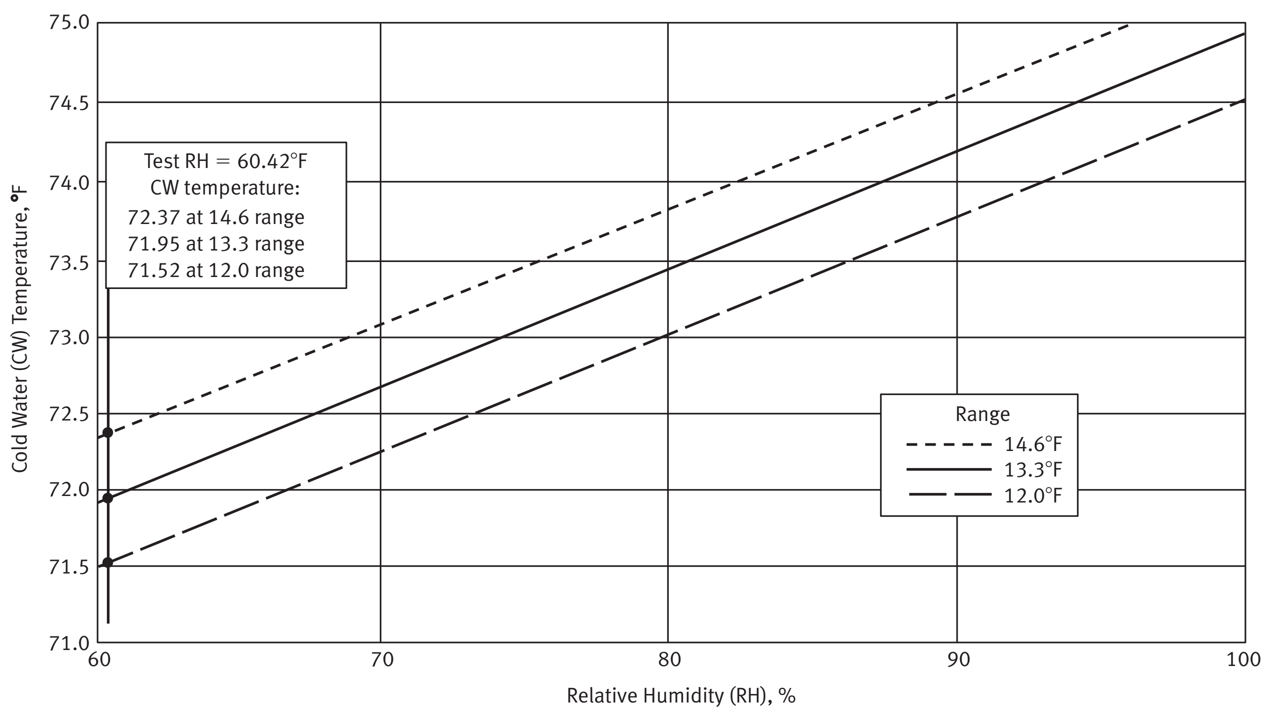


Fig. F2-3 Cross Plot No. 1C
Dry Bulb = 56.12°F, Water Flow Rate = 415,800 gpm (110%)

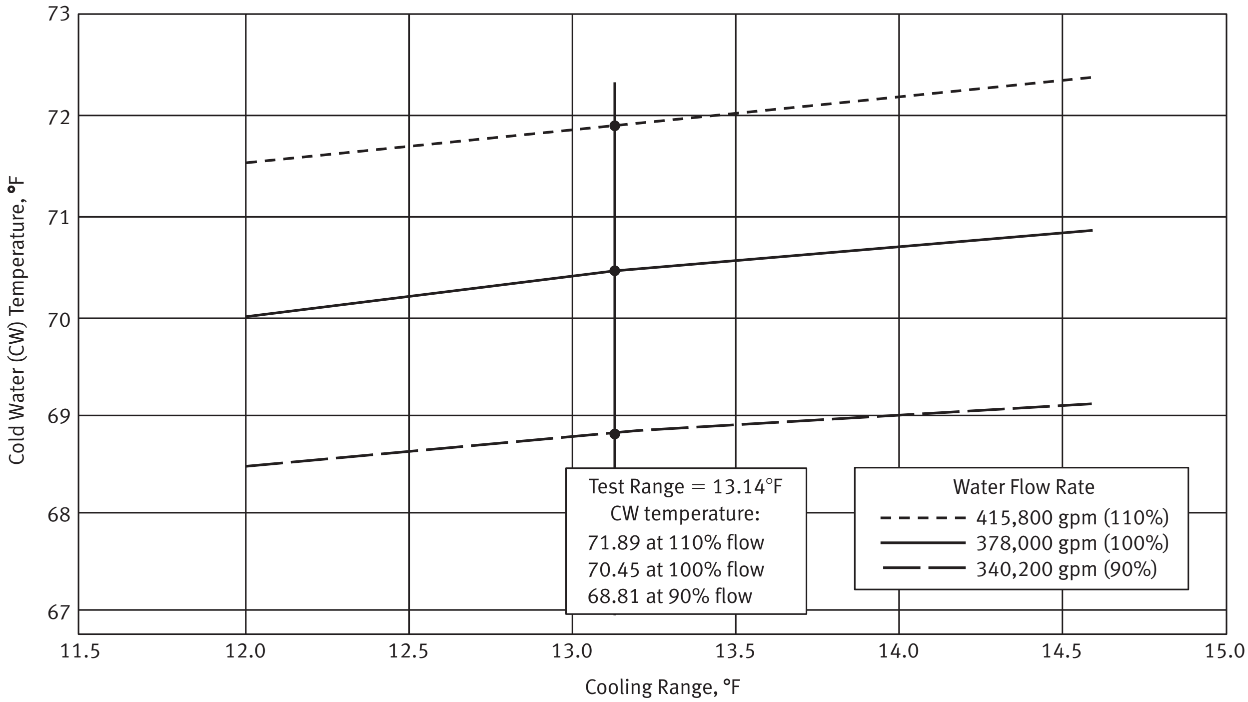


Fig. F2-4 Cross Plot No. 2
Dry Bulb = 56.12°F, Relative Humidity = 60.42%

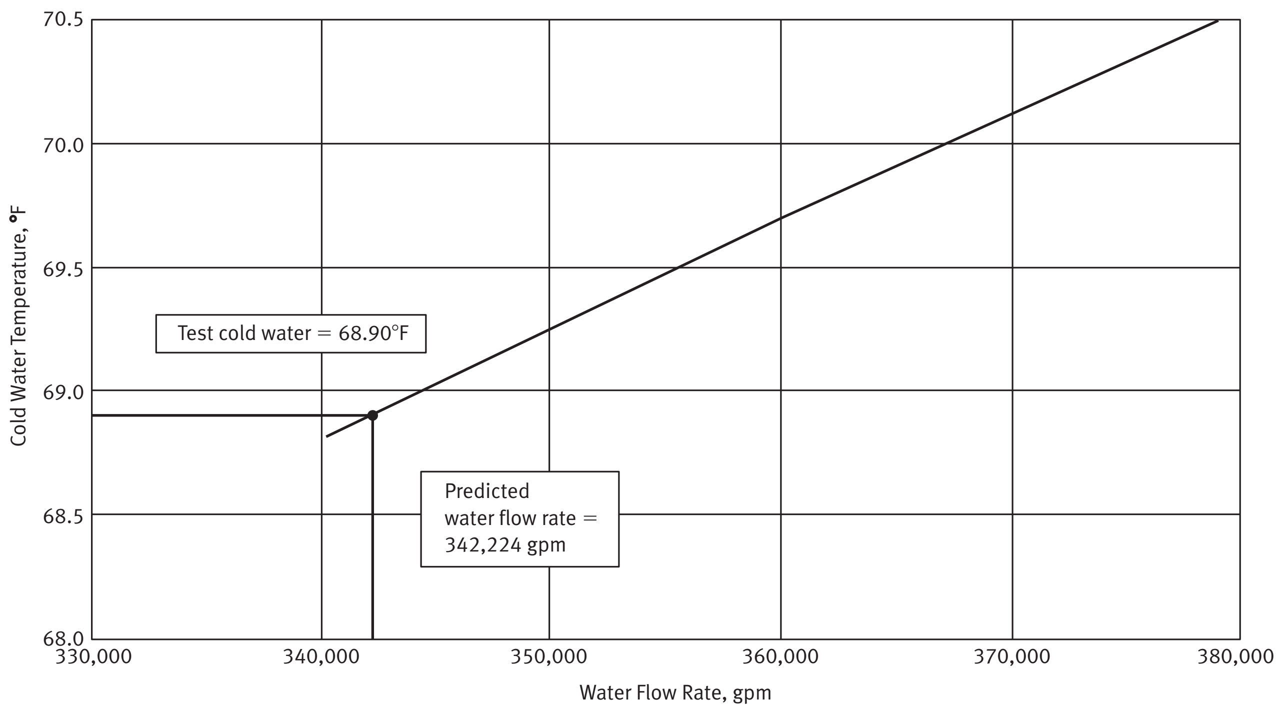


Fig. F2-5 Cross Plot No. 3
Dry Bulb = 56.12°F, Humidity = 60.42%, Range = 13.14°F

Table F2-1 Predicted Cold Water Temperatures

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

GENERAL NOTE: 56.12°F is the entering dry-bulb temperature.

Table F2-2 Predicted Cold Water Temperatures

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

GENERAL NOTE: 56.12°F is the entering dry-bulb temperature and 60.42% is the relative humidity.

Table F2-3 Predicted Cold Water Temperatures

90%	100%	110%
68.81°F	70.45°F	71.89°F

GENERAL NOTE: 56.12°F is the entering dry-bulb temperature, 60.42% is the relative humidity, and 13.14°F is the range.

NONMANDATORY APPENDIX G SAMPLE CALCULATION FOR PLUME COMPLIANCE

G1 DESIGN AND TEST CONDITIONS

See Table G1.

	Design	Test
Water flow, l/s	1100.0	1100.0
Hot water (T_{HW}), °C	25.0	24.5
Cold water (T_{CW}), °C	15.0	14.5
Cooling range (R), °C	10.0	10.0
Upwind dry-bulb (T_{DBu}), °C	5.0	7.0
Upwind wet-bulb (T_{WBu}), °C	4.3	3.4
Upwind relative humidity (R_{HV}), %	90.0	53.5
Inlet dry-bulb (T_{DBi}), °C	5.0	7.1
Inlet wet-bulb (T_{WBi}), °C	4.3	3.6
Inlet relative humidity (Rh_{in}), %	90.0	54.9
Barometric pressure (HB), kPa	101.3	100.6
Zero visible plume	yes	

Correction of the Relative Humidity by Barometric Pressure

Corrected upwind air relative humidity:
$RH_{uc} = 53.5 \cdot 101.3/100.6 = 53.87\%$
Corrected inlet air relative humidity:
$RH_{ic} = 54.9 \cdot 101.3/100.6 = 55.28\%$
Corrected exhaust air relative humidity:
$RH_{ie} = 78.09 \cdot 101.3/100.6 = 78.63\%$

G2 PLUME ABATEMENT GUARANTEE CURVE (EXHAUST AIR CHARACTERISTICS CURVES)

See Fig. G2.

G3 TEST INTERPRETATION (EXHAUST AIR CHARACTERISTICS CURVES)

G3.1 Drawing the Measured Plume Dilution Line

On the psychrometric diagram, locate (per G3.3)
Upwind air conditions:
DB = 7.0°C Rh = 53.87%
Exit air conditions:
DB = 18.84°C Rh = 78.63%
Draw a straight line between these two points.

G3.2 Calculation of the Guarantee Relative Humidity [as in 6-1(c)]

On the performance curves,
Water flow = 1100 l/s
Range = 10°C
Inlet wet-bulb = 3.6°C
Inlet relative humidity (graph conditions) = 55.28%

By interpolation between the inlet humidity curves,
Guarantee exhaust air wet-bulb = 15.7°C
Guarantee exhaust air dry-bulb = 17.5°C
Barometric pressure = 101.3 kPa
Guarantee exit air humidity = 83.35%

G3.3 Drawing the Guarantee Plume Dilution Line

On the psychrometric diagram, locate
Upwind air conditions:
DB = 7.0°C Rh = 53.87%
Exhaust air conditions:
DB = 17.5°C
Rh = 83.35%
Draw a straight line between these two points.

G3.4 Calculation of the Equivalent Guarantee Exhaust Air Relative Humidity

On the psychrometric diagram, draw a line from measured point, parallel to the enthalpy line. The cross point between the above line and the guarantee dilution line corresponds to 83.6% humidity.

G3.5 Tower Plume Indicator

Measured relative humidity = 78.4%
Equivalent guarantee relative humidity = 83.6%
Tower plume indicator = $100 \cdot 83.6/78.4 = 106.6\%$

G3.6 Evaluation of the Air Mixing Quality (Zero Visible Plume)

Maximum limit of exhaust air relative humidity = $78.4 \cdot 1.2 = 94.1\%$

On the psychrometric diagram, locate the point on the enthalpy line passing at the measured point and corresponding to 94.1%. Draw a straight line between above point and the point representing the upwind air conditions (maximum limit of the dilution line).

The measured points named *Y1* and *Z1* are located above the maximum limit of the dilution line. These points are marked "Yes" in the table.

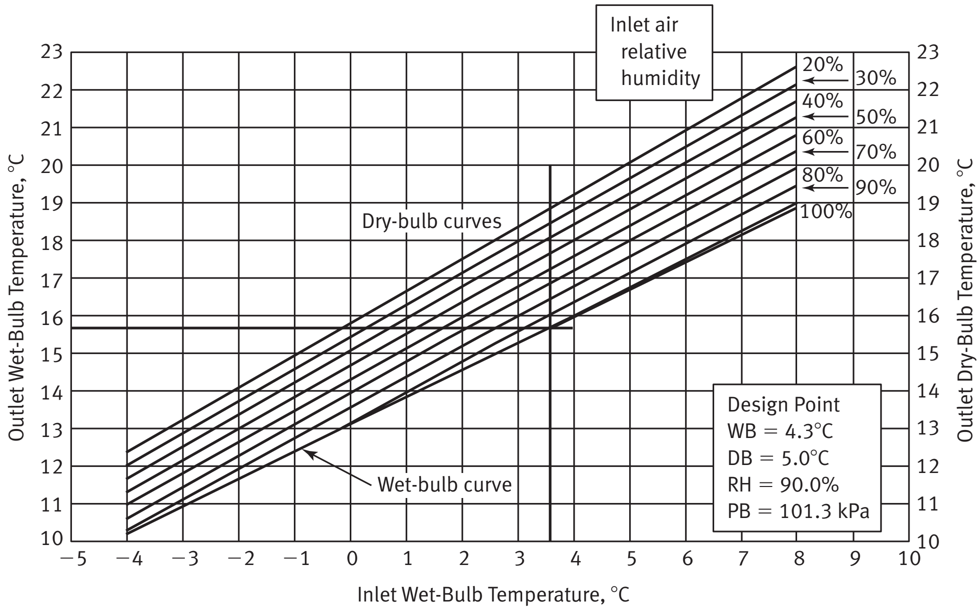
The air velocity V corresponding to these points is:
Y1 V = 5.1 m/s
Z1 V = 3.8 m/s
Sum of the concerned points = 8.9 m/s
Sum of all the air velocities = 227.7 m/s
Mixing quality:
$MQ = (1–8.9/227.7) \cdot 100 = 96.1\%$

The mixing quality coefficient is higher than 85%. The mixing quality criteria is well fulfilled.

Table G1

Exhaust Air Measurements					Air Characteristics				Average WB, DB Calculation		
Points, L	WB, °C	DB, °C	Velocity, m/s		Enthalpy, kJ/kg	Humidity Rating, kg/kg	Specific Volume, m^3/kg		Mass, kg/s.m^2	Enthalpy Mass	Humidity Mass
W1	17.7	18.9	3.4		50.2991	0.01234	0.84988		4.00	201.22	0.0494
W2	17.7	19.4	15.1		50.2867	0.01213	0.85106		17.74	892.22	0.2152
W3	17.2	19.6	16.4		48.7255	0.01144	0.85070		19.28	939.34	0.2205
W4	16.1	20.4	14.8		45.3929	0.00980	0.85082		17.39	789.61	0.1705
W5	16.7	20.6	6.7		47.1776	0.01042	0.85224		7.86	370.89	0.0819
X1	17.2	18.4	4.5		48.7541	0.01193	0.84788		5.31	258.76	0.0633
X2	17.0	18.8	14.6		48.1305	0.01153	0.84850		17.21	828.17	0.1984
X3	17.1	20.2	16.7		48.4039	0.01107	0.85195		19.60	948.82	0.2170
X4	16.3	20.9	15.4		45.9734	0.00983	0.85231		18.07	830.67	0.1776
X5	16.6	20.3	7.8		46.8832	0.01043	0.85138		9.16	429.52	0.0956
Y1	16.4	16.9	5.1		46.3605	0.01160	0.84307		6.05	280.45	0.0702
Y2	16.1	17.4	13.9		45.4584	0.01104	0.84378		16.47	748.86	0.1819
Y3	15.5	17.2	15.7		43.7124	0.01043	0.84239		18.64	814.69	0.1944
Y4	15.2	17.7	15.2		42.8426	0.00988	0.84311		18.03	772.39	0.1781
Y5	14.9	18.4	8.2		41.9794	0.00926	0.84430		9.71	407.71	0.0899
Z1	16.6	17.1	3.8		46.9560	0.01175	0.84386		4.50	211.45	0.0529
Z2	16.3	17.5	12.9		46.0488	0.01123	0.84433		15.28	703.55	0.1716
Z3	15.9	17.6	14.9		44.8660	0.01072	0.84394		17.66	792.12	0.1893
Z4	15.9	18.6	15.4		44.8445	0.01031	0.84629		18.20	816.04	0.1876
Z5	15.3	19.2	7.2		43.0973	0.00938	0.84678		8.50	366.45	0.0798
Average [Note (1)]	16.35	18.84	11.4	**Weighted Average**	46.1656	0.01074		**Sum**	268.66	12,402.93	2.8850

NOTE:
(1) Exhaust dry bulb, T_{DBe} 16.35°C
Exhaust wet bulb, T_{WBe} 18.84°C
Exhaust relative humidity, RH_e 78.09%



GENERAL NOTES:

(a) Water Flow = 1 100 l/s
Range = 10°C

(b) Conditions:
(1) parallel air circuit
(2) full flow in the dry section
(3) air shutters fully open

Fig. G2 Wet-Dry Plume Abatement Performance Curve: Outlet Temperatures as a Function of Inlet Conditions

NONMANDATORY APPENDIX H SAMPLE CALCULATION FOR CLOSED-CIRCUIT EVAPORATIVE COOLERS

The following sample calculations are typically used to evaluate test data of a closed-circuit evaporative cooler, using performance curves submitted by the manufacturer in accordance with para. 3-9.

H1 CLOSED-CIRCUIT EVAPORATIVE COOLER CAPABILITY

Table H1 presents the design conditions and a set of test conditions for a particular evaporative cooler. The procedure for calculation of thermal capability is described in para. 5-10.2.

The test data are compared to the requirements of paras. 3-5, 3-6, 3-7, and 3-8 to ensure that they meet the requirements stated therein.

(a) Correct the test fan power to design exit air density (see para. 5-8) for calculation of adjusted test fluid rate. The test fan power has already been adjusted for nameplate motor efficiency and power factor in accordance with para. 5-8. The CCEC manufacturer shall also have provided some or all of the following information:

(1) design exit air density, lbm/ft^3

(2) design exit air temperature, °F

(3) design air rate, acfm

If design exit air temperature is provided, the air may be assumed to be saturated. Corresponding values of specific volume and specific humidity may be obtained from psychrometric tables or curves, and the design acfm may then be calculated.

If the design exit air density is provided with the design exit air temperature, the acfm may be calculated by using the tables.

The test exit air density must be determined by an iterative process, as shown in Table H1.1. The test exit air density κ_a can be calculated from these data:

$$\kappa_a = 1.02752/14.364 \text{ (at 86.22°F)}$$

$$\kappa_a = 0.07153 \, \frac{\text{air-vapor mixture, Lb}_m}{\text{air-vapor mixture, ft}^3}$$

Since fan power varies directly with air density for a constant system (constant volume) at constant fan speed and constant blade pitch angle, the test fan power is corrected for the difference between design air density and test air density as follows:

$$\begin{aligned}\text{Corrected test fan power} &= (0.07122/0.07153)\,(98.8)\\ &= 98.37 \text{ HP}_t\end{aligned}$$

(b) Determine the adjusted test fluid rate according to para. 5-2.2.

$$\begin{aligned}\text{Design fan power} &= 100 \text{ HP}_d\\ \text{Corrected test fan power} &= 98.37 \text{ HP}_t\\ \text{Adjusted test fluid rate} &= (4{,}550)\,(100/98.37)^{0.25}\\ &= 4{,}569 \text{ gpm}\end{aligned}$$

(c) Determine tower capability from manufacturer's performance curves. The manufacturer will have submitted performance curves A through C at 90%, 100%, and 110% of design fluid rate (see Figs. H1.3-1 through H1.3-3).

(d) Construct a cross plot no. 1 (Fig. H1.4) from the manufacturer's performance curves by plotting cooling range against cold fluid temperature at the test wet-bulb temperature (72.2°F). The following predicted cold fluid temperatures are obtained from cross plot no. 1 at the test cooling range of 22.4°F.

(e) Construct a test cross plot no. 2 (see Fig. H1.5) by plotting fluid rates against the cold fluid temperatures shown in Table H1.5.

(f) The predicted test fluid rate is obtained from test cross plot no. 2 by entering the curve at the test cold fluid temperature (87.2°F).

$$\begin{aligned}\text{Predicted test fluid rate} &= 4{,}315 \text{ gpm}\\ \text{Tower thermal capability} &= \frac{\text{Adjusted test fluid rate}}{\text{Predicted test fluid rate}} \times 100\\ &= \frac{4{,}569}{4{,}315} \times 100\\ &= 105.9\%\end{aligned}$$

H2 PREDICTED TEST COLD FLUID TEMPERATURE

The predicted test cold fluid temperature at test conditions can be obtained from test cross plot no. 2 by entering the curve at the adjusted test fluid rate (4,569 gpm).

$$\begin{aligned}\text{Predicted test cold fluid temperature} &= 88.1°\text{F}\\ \text{Actual test cold fluid temperature} &= 87.2°\text{F}\\ \text{Deviation} &= 87.20°\text{F} - 88.10°\text{F} = -0.90°\text{F}\end{aligned}$$

Table H1 Design and Test Conditions

	Design Conditions	Test Conditions
Process fluid	Water	Water
Hot fluid temperature, °F	110.0	109.6
Cold fluid temperature, °F	90.0	87.2
Wet-bulb temperature, °F	75.0	72.2
Cooling range, °F	20.0	22.4
Circulating process fluid rate, gpm	5,000	4,550
Fan power (per fan), BHP	100	98.8
Exponent for fan power corr.	0.250	. . .
Heat rejection, Btu/hr	50×10^6	50.96×10^6
Design air rate, acfm	817,960	. . .
Design exit air density, lbm·ft^3	0.07122	. . .

Table H1.1 Test Exit Air Properties

Assumed Exit Air Temperature, °F	Specific Volume, v_{da}, ft^3/lb Dry Air	Humidity Ratio, lb Water/lb Dry Air	Calculated Exit Air Enthalpy, h_2, Btu/lb Dry Air	Corresponding Exit Air Temperature, °F
95.00	14.802	1.03674	51.38	86.58
86.00	14.354	1.02732	50.91	86.29
86.25	14.366	1.02755	50.93	86.22
86.22	14.364	1.02752	50.93	86.22

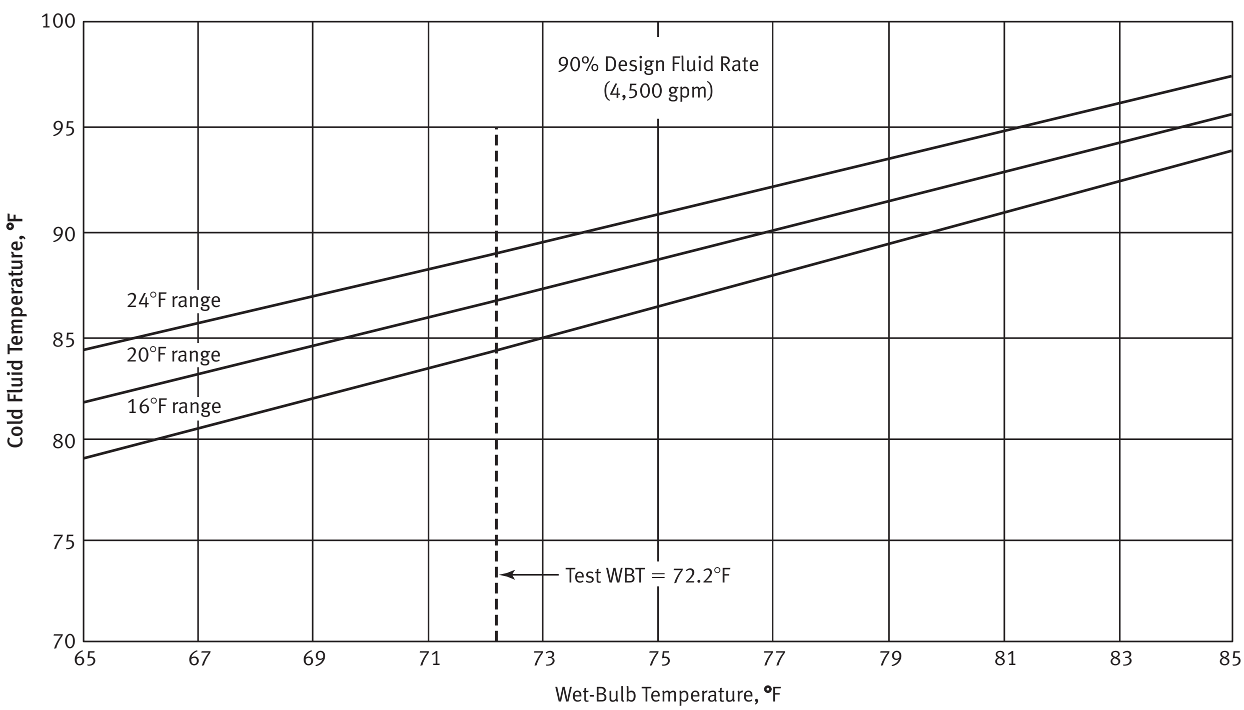


Fig. H1.3-1 Manufacturer's Performance Curve A

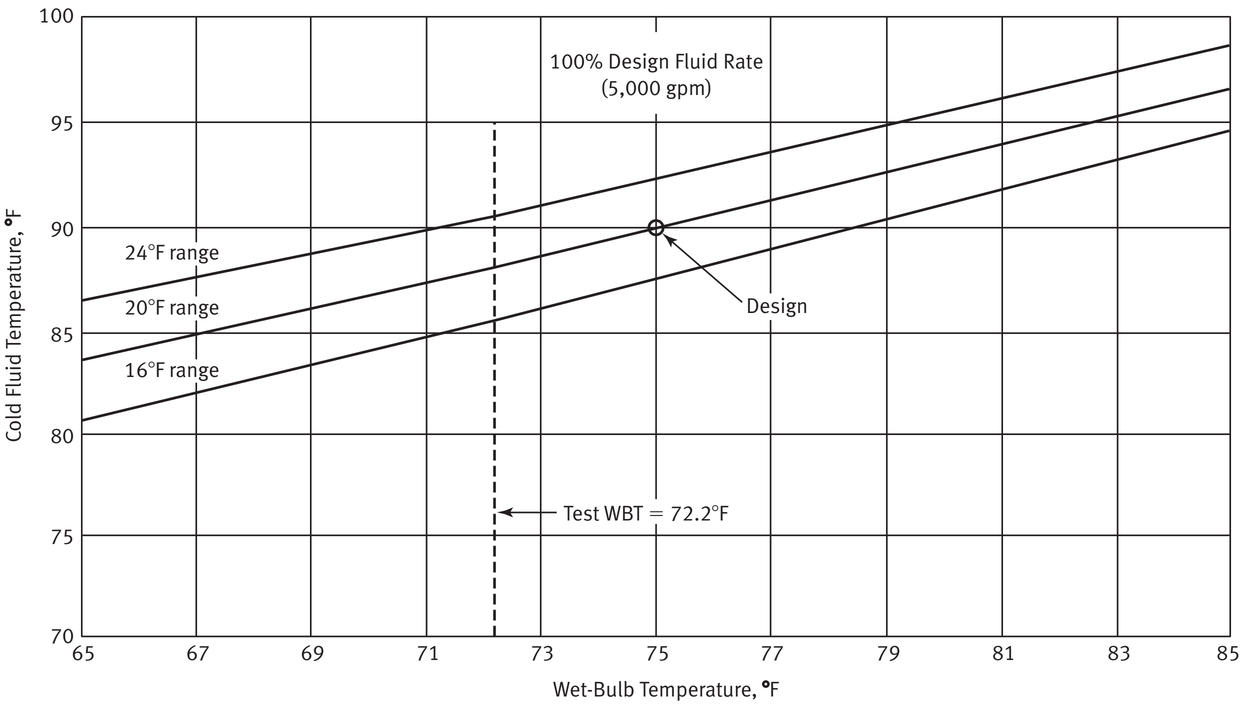


Fig. H1.3-2 Manufacturer's Performance Curve B

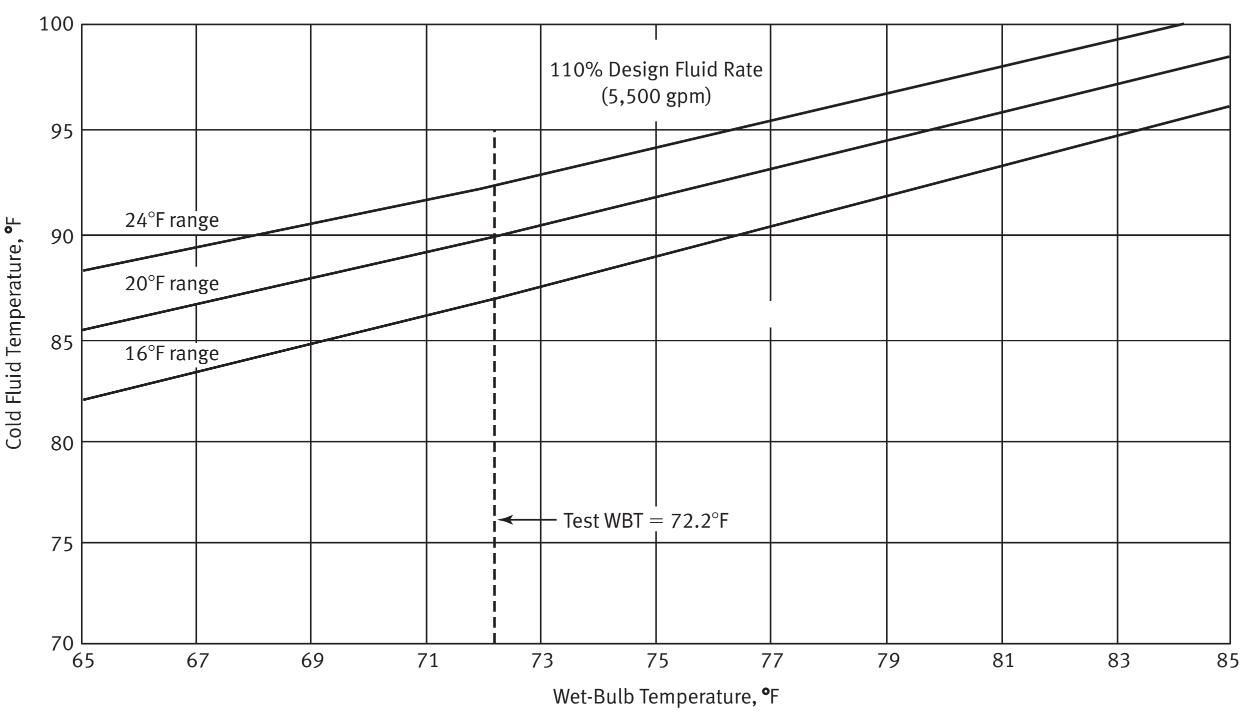


Fig. H1.3-3 Manufacturer's Performance Curve C

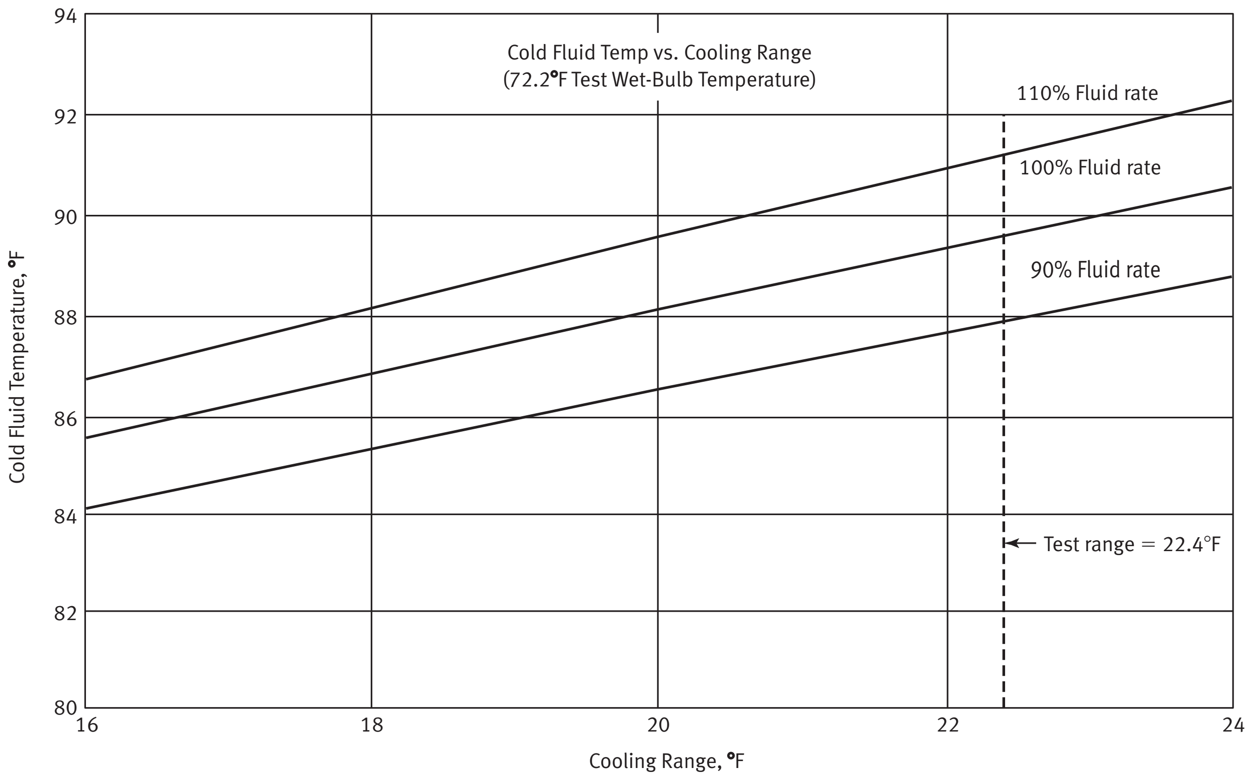


Fig. H1.4 Test Cross Plot No. 1

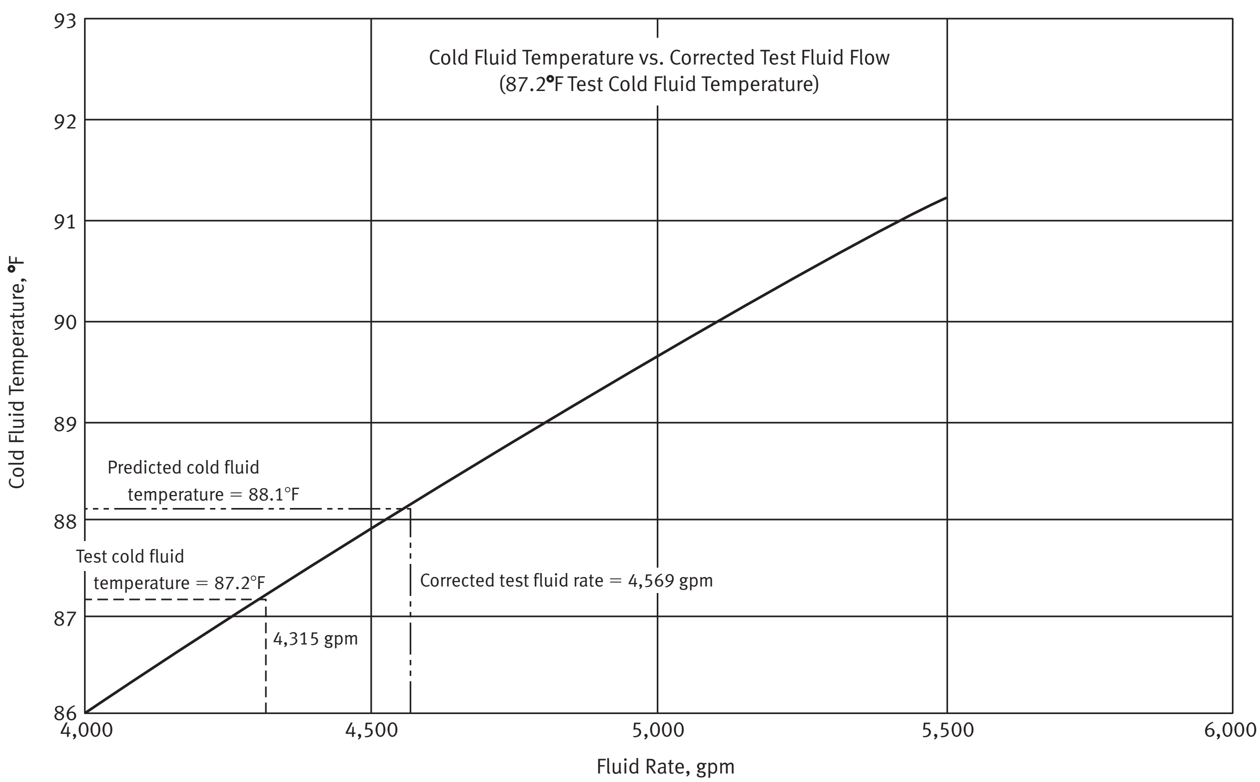


Fig. H1.5 Test Cross Plot No. 2

H3 PREDICTED DESIGN COLD FLUID TEMPERATURE

Construct design cross plot no. 3 (Fig. H3) at the 20°F design range and 75°F design wet-bulb temperature by directly reading the cold fluid temperatures at the three fluid flow rates of the manufacturer's performance curves A through C in Fig. H1.4. No intermediate cross plot is necessary since no interpolation is required at design conditions. In Table H3, the predicted cold fluid temperatures at the respective fluid rates are the basis for design cross plot no. 3.

The predicted cold fluid temperature at the design cooling range, wet-bulb temperature, and fluid rate, based on the tested thermal capability, is obtained by entering the design cross plot no. 3 at a test-compensated flow rate. This test-compensated fluid flow rate is simply the ratio of the design flow rate to the test capability.

$$
\begin{aligned}
\text{Test-compensated fluid flow rate} &= 5{,}000/1.059 \\
&= 4{,}721 \text{ gpm} \\
\text{Predicted design cold fluid temperature} &= 89.15°\text{F} \\
\text{Design cold fluid temperature} &= 90.00°\text{F} \\
\text{Deviation} &= 89.15°\text{F} - 90.00 \\
&= -0.85°\text{F}
\end{aligned}
$$

Table H1.5 Predicted Cold Fluid Temperatures Cross Plot 1

% Design Fluid Rate	Fluid Rate, gpm	Cold Fluid Temperature, °F
90	4,500	87.85
100	5,000	89.60
110	5,500	91.20

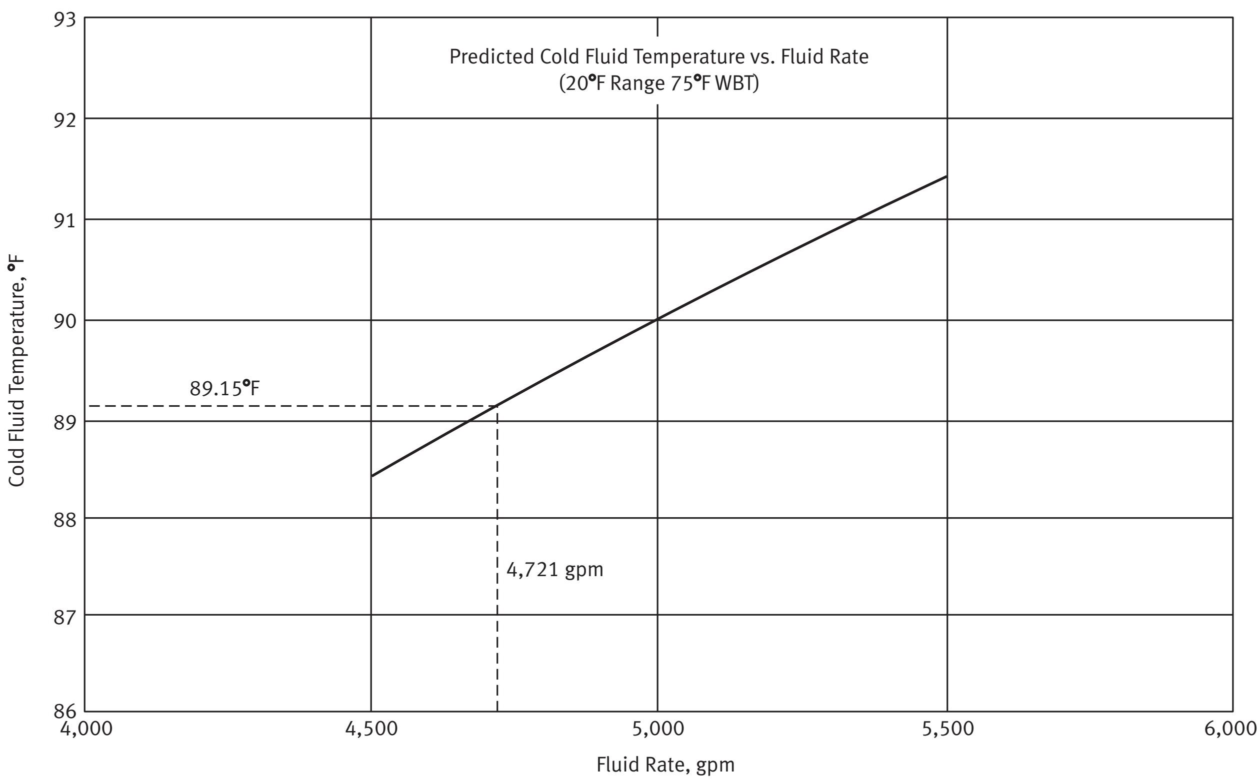


Fig. H3 Design Cross Plot No. 3

Table H3 Predicted Cold Fluid Temperatures Cross Plot 3

% Design Fluid Rate	Fluid Rate, gpm	Cold Fluid Temperature, °F
90	4,500	88.45
100	5,000	90.00
110	5,500	91.42

NONMANDATORY APPENDIX I SAMPLE CALCULATION FOR WET SURFACE AIR-COOLED CONDENSERS (WSACC)

Table I1 Design and Test Conditions

	Design Conditions	Test Conditions
Steam rate, lbm/hr	155,000	148,560
Steam inlet pressure, Hgabs	3.00 in.	2.76 in.
Steam inlet quality	90%	89.85%
Wet-bulb temperature, °F	74.0	70.5
Fan power (per fan), BHP	96.0	98.8
Exponent for fan power corr.	0.250	. . .
Heat rejection, Btu/hr	143.5×10^6	137.5×10^6
Design air rate, acfm	1,639,000	. . .
Design exit air density, lbm.ft^3	0.0705	. . .

The following sample calculations are typically used to evaluate the test data of a wet surface air-cooled condenser (WSACC), using performance curves submitted by the manufacturer in accordance with para. 3-9.

I1 WET SURFACE AIR-COOLED CONDENSER CAPABILITY

Table I1 presents the design conditions and a set of test conditions for a particular WSACC. The procedure for calculation of thermal capability is described in para. 5-10. The test data are compared to the requirements of paras. 3-5, 3-6, 3-7, 3-8, and 4-10 to ensure that they meet the requirements stated therein.

(a) Correct the test fan power to design exit air density (see para. 5-8) for calculation of adjusted test steam rate. The test fan power has already been adjusted for nameplate motor efficiency and power factor according to para. 5-8. The WSACC manufacturer shall also have provided some or all of the following information:

(1) design exit air density, lbm/ft^3

(2) design exit air temperature, °F

(3) design air rate, acfm

If design exit air temperature is provided, the air may be assumed to be saturated. Corresponding values of specific volume and specific humidity may be obtained from psychrometric tables or curves, and the design acfm may then be calculated.

If the design exit air density is provided with the design exit air temperature, the acfm may be calculated using the tables. The test exit air density must be determined by an iterative process, as shown in Table I1.1.

The test exit air density κ_a can be calculated from these data:

$$\kappa_a = 1.03036/14.505 \text{ (at 89.18°F)}$$

$$\kappa_a = 0.07103 \frac{\text{air-vapor mixture, lb}}{\text{air-vapor mixture, ft}^3}$$

Since fan power varies directly with air density for a constant system (constant volume) at constant fan speed and constant blade pitch angle, the test fan power is corrected for the difference between design air density and test air density as follows:

$$\begin{aligned}\text{Corrected test fan power} &= (0.07052/0.07103)\ (98.8)\\ &= 98.09\ HP_t\end{aligned}$$

(b) Determine the adjusted test steam rate according to para. 5-2.2.

$$\begin{aligned}\text{Design fan power} &= 96.0\ HP_d\\ \text{Corrected test fan power} &= 98.09\ HP_t\\ \text{Adjusted test steam rate} &= (148{,}560)\ (96.0/98.09)^{0.25}\\ &= 147{,}762 \text{ lbm/hr}\end{aligned}$$

(c) Determine tower capability from manufacturer's performance curves. The manufacturer will have submitted performance curves A through C at 90%, 100%, and 110% of design steam rate. See Figs. I1.3-1 through I1.3-3.

(d) Construct cross plot no. 1 (Fig. I1.4) from the manufacturer's performance curves by plotting steam inlet quality against steam inlet pressure at the test wet-bulb temperature (70.5°F). In Table I1.4, the predicted steam

Table I1.1

Assumed Exit Air Temperature, °F	Specific Volume, v_{da}, ft^3/lb Dry Air	Humidity Ratio, lb Water/lb Dry Air	Calculated Exit Air Enthalpy, h_2, Btu/lb Dry Air	Corresponding Exit Air Temperature, °F
95.0	14.802	1.03674	55.21	89.48
90.0	14.545	1.03119	54.85	89.22
89.20	14.506	1.03038	54.79	89.18
89.18	14.505	1.03036	54.79	89.18

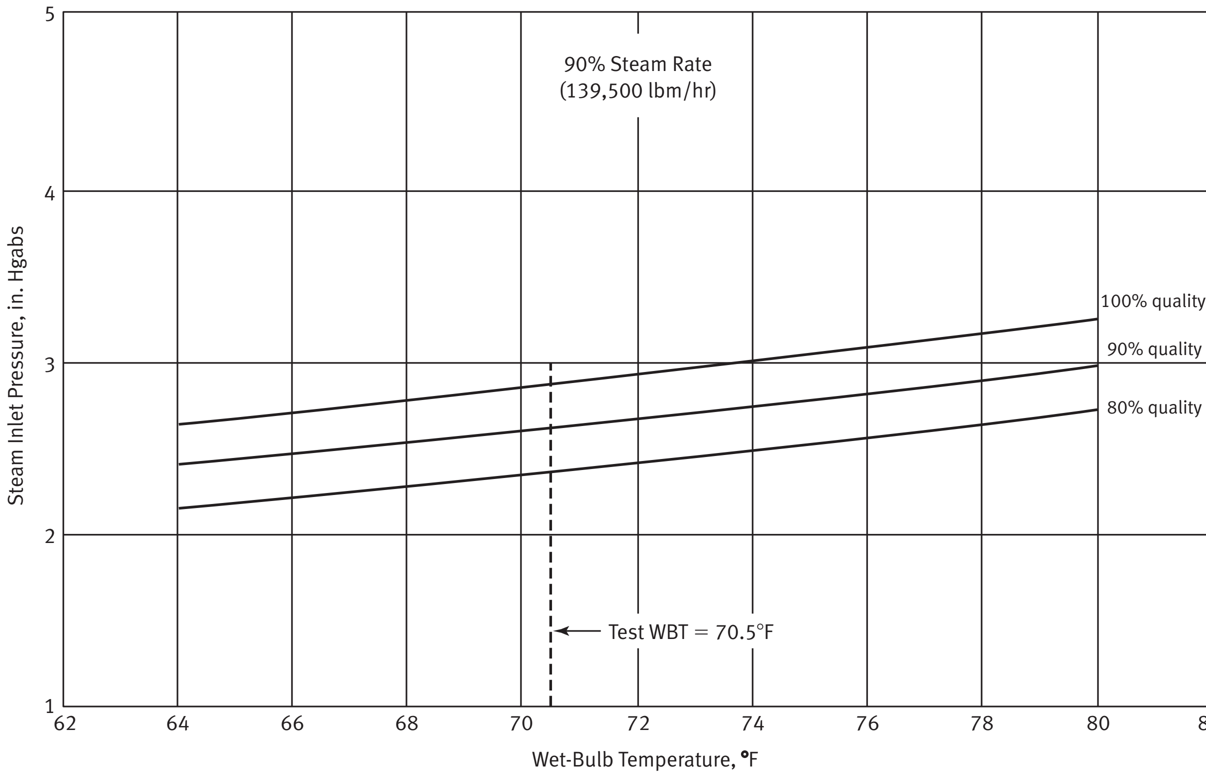


Fig. I1.3-1 Manufacturer's Performance Curve A

inlet pressures are obtained from cross plot no. 1 at the test quality of 89.85%.

(e) Construct test cross plot no. 2 (see Fig. I1.5) by plotting the steam rates against the steam inlet pressures shown in Table I1.4.

(f) The predicted test steam rate is obtained from test cross plot no. 2 by entering the curve at the test steam inlet pressure (2.76 in. Hgabs).

$$\text{Predicted test steam rate} = 148{,}300 \text{ lbm/hr}$$

$$\text{WSACC thermal capability (\%)} = \frac{\text{Adjusted test steam rate}}{\text{Predicted test steam rate}} \times 100$$

$$= \frac{147{,}762}{148{,}300} \times 100$$

$$= 99.64\%$$

I2 PREDICTED TEST STEAM INLET PRESSURE

The predicted steam inlet pressure at the test conditions can be obtained from test cross plot no. 2 by entering the curve at the adjusted test steam rate (147,762 lbm/hr).

Predicted test steam inlet pressure = 2.75 in. Hgabs
Actual test steam inlet pressure = 2.76 in. Hgabs
Deviation = 2.76 in. − 2.75 in.
= 0.01 in. Hgabs

I3 PREDICTED DESIGN STEAM INLET PRESSURE

Construct design cross-plot no. 3 (Fig. I3) at the 90% design steam quality and 74°F design wet-bulb temperature by directly reading the steam inlet pressures at the

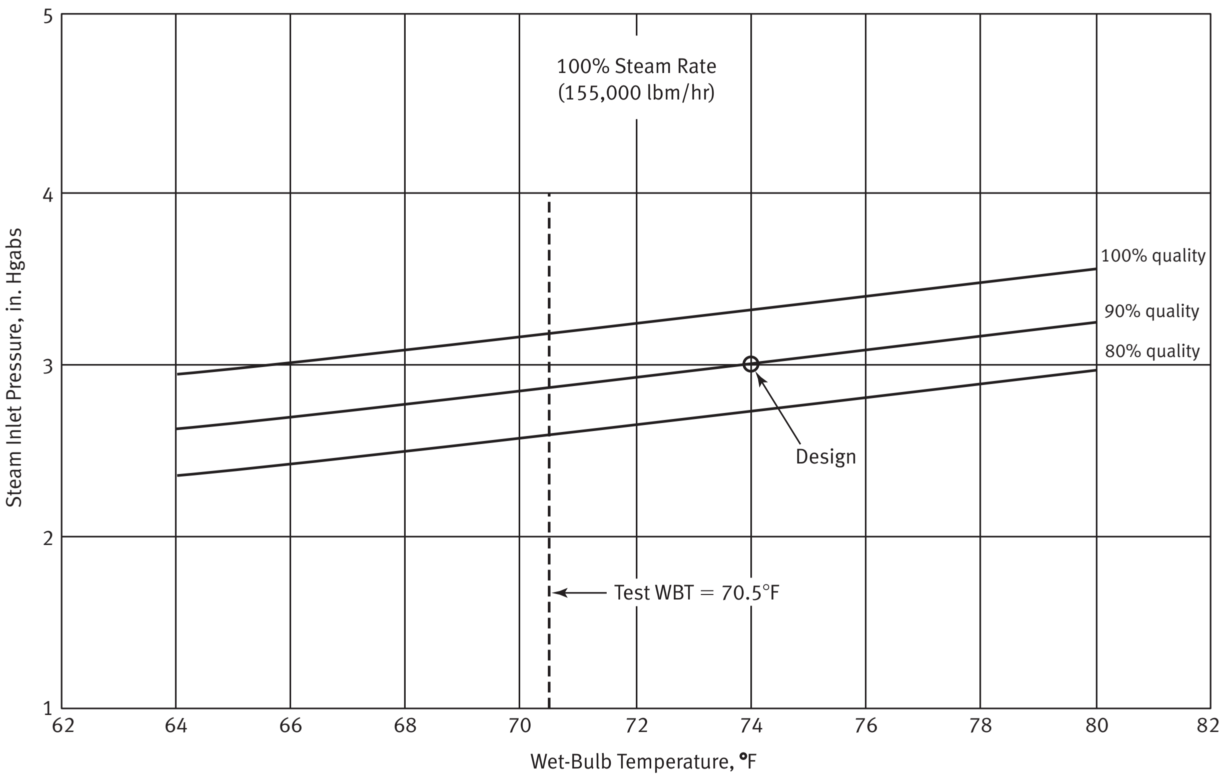


Fig. I1.3-2 Manufacturer's Performance Curve B

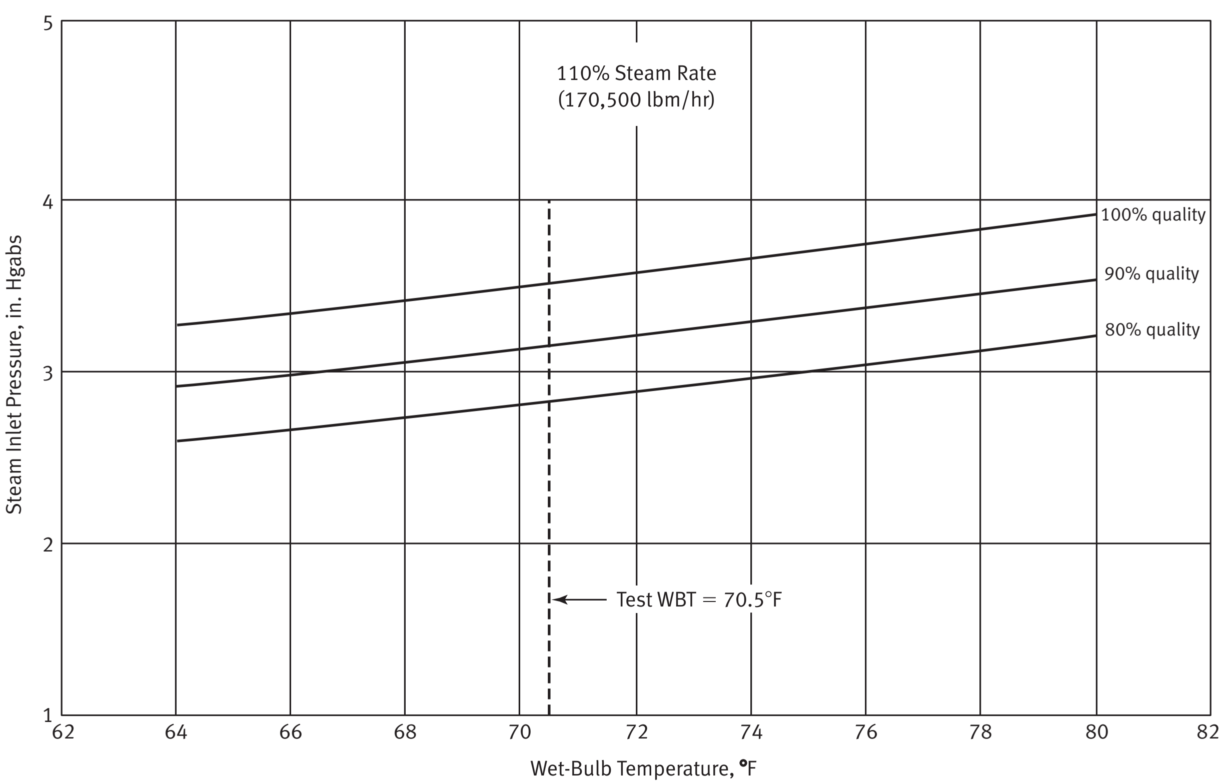


Fig. I1.3-3 Manufacturer's Performance Curve C

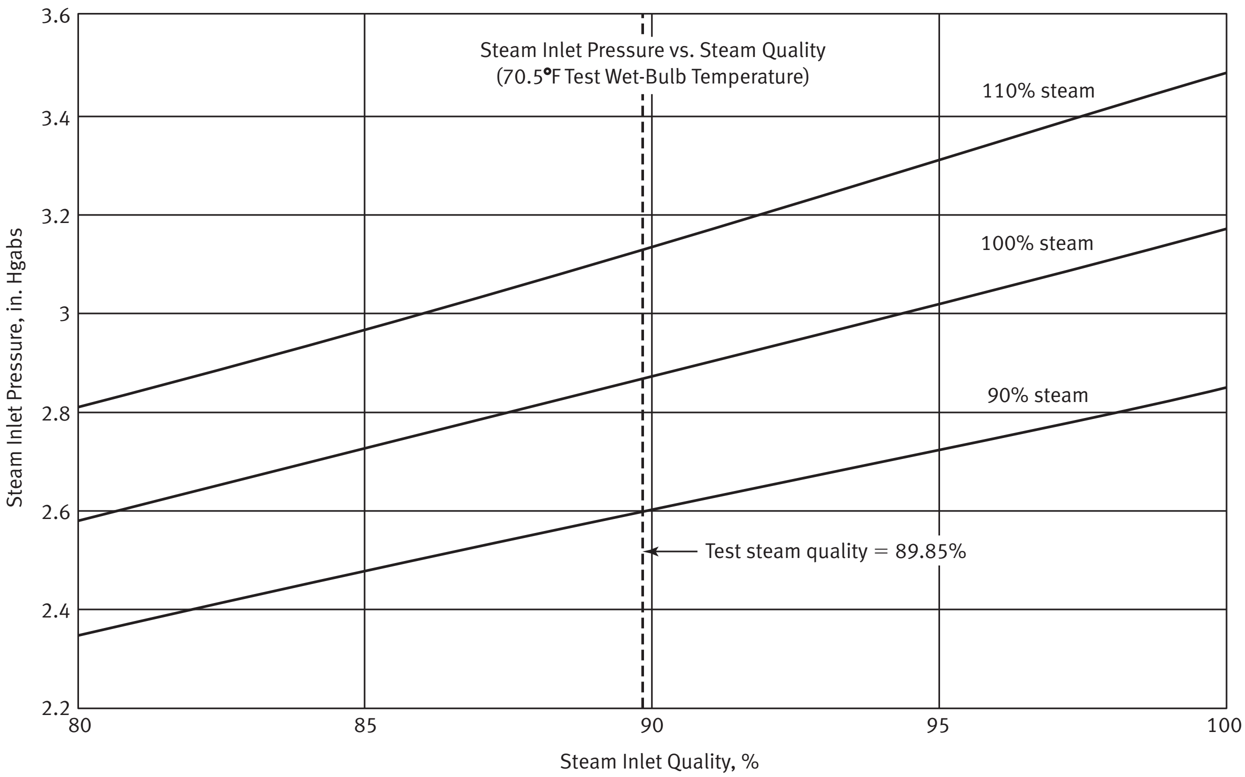


Fig. I1.4 Test Cross Plot No. 1

Table I1.4

% Design Steam Rate	Steam Rate, lbm/hr	Steam Inlet Pressure, in. Hgabs
90	139,500	2.598
100	155,000	2.870
110	170,500	3.125

Table I3

% Design Steam Rate	Steam Rate, lbm/hr	Steam Inlet Pressure, in. Hgabs
90	139,500	2.730
100	155,000	3.000
110	170,500	3.275

three steam rates of the manufacturer's performance curves A through C in Fig. I1.4. No intermediate cross plot is necessary since no interpolation is required at design conditions. In Table I3, the predicted steam inlet pressures at the respective steam rates are the basis for design cross plot no. 3.

The predicted steam inlet pressure at the design steam quality, wet-bulb temperature, and steam rate, based on the tested thermal capability, is obtained by entering design cross plot no. 3 at a test-compensated steam rate. This test-compensated steam rate is simply the ratio of the design steam rate to the test capability.

Test-compensated fluid flow rate = 155,000/0.9964
= 155,564 lbm/hr

Predicted design steam inlet pressure = 3.013 in. Hgabs

Design steam inlet pressure = 3.00 in. Hgabs

Deviation = 3.013 in. − 3.000 in.
= 0.013 in. Hgabs

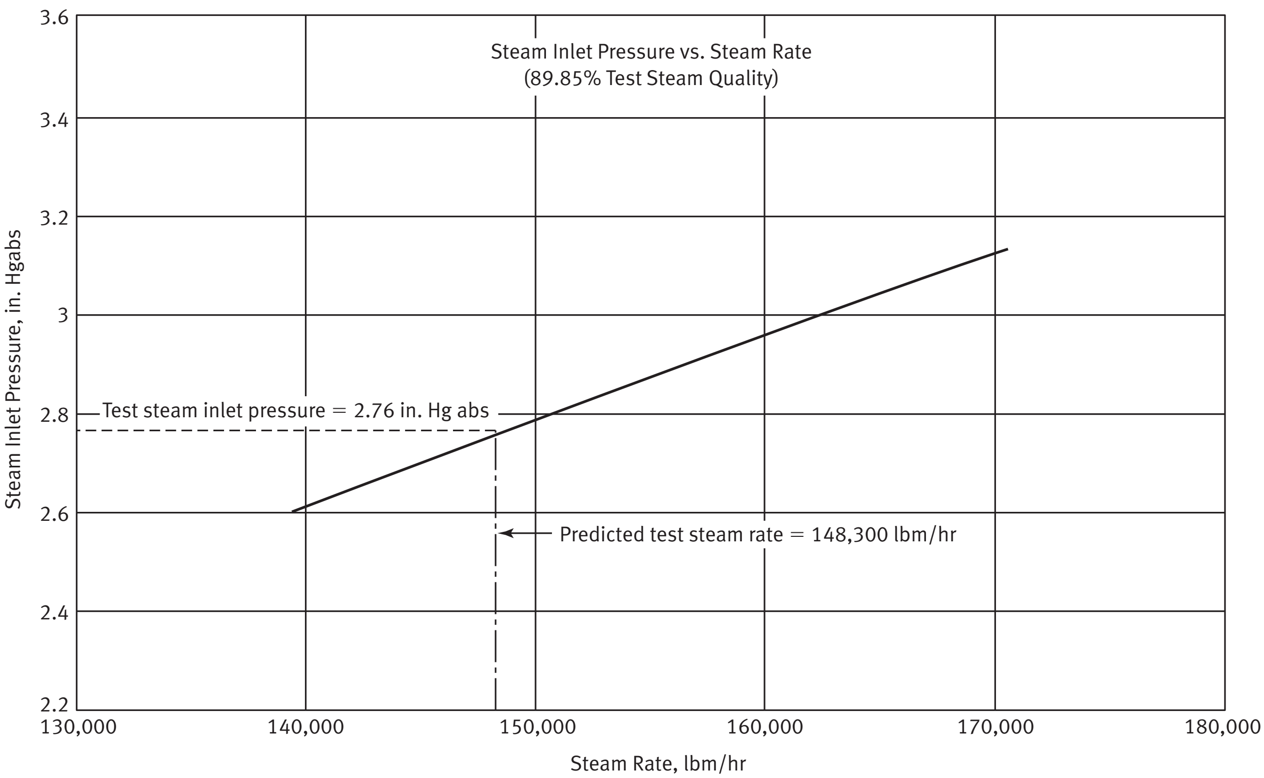


Fig. I1.5 Test Cross Plot No. 2

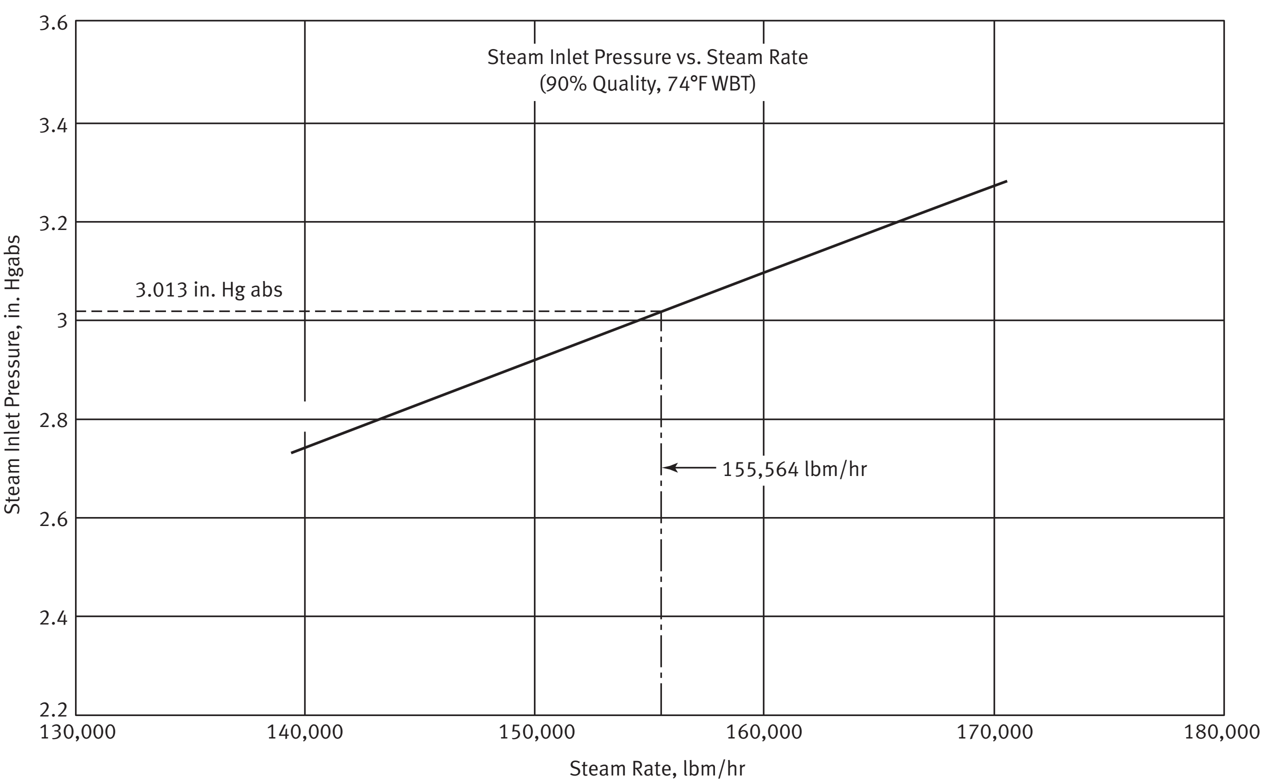
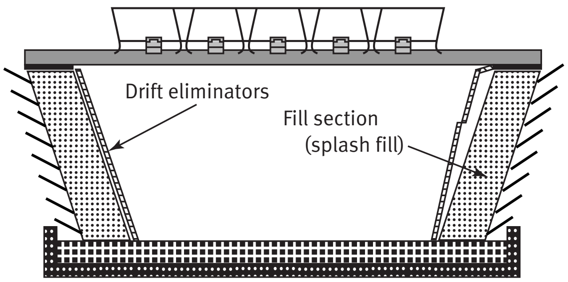


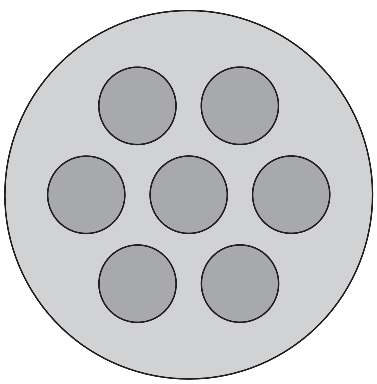
Fig. I3 Design Cross Plot No. 3

NONMANDATORY APPENDIX J
ILLUSTRATIONS OF COOLING EQUIPMENT

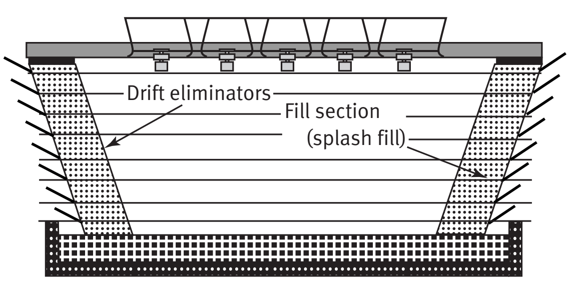
This Appendix contains Figs. J1 through J13.



Side Elevation View

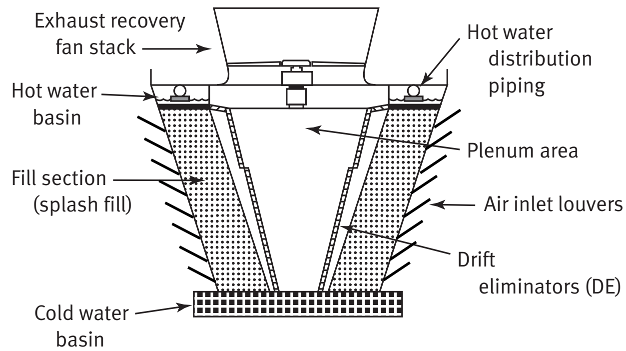


Top View

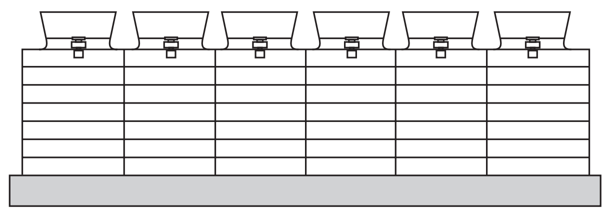


End View

Fig. J1 Mechanical Draft: Round Crossflow Cooling Tower



End Elevation View



Side Elevation View

Fig. J2 Mechanical Draft: Crossflow Cooling Tower

Tower Pumping Head

1. Static head above curb to centerline inlet, ft
2. Inlernal losses
   (a) Friction, ft
   (b) Velocity head, ft
   (c) Static head, ft
3. Total pump head, ft (1+2)

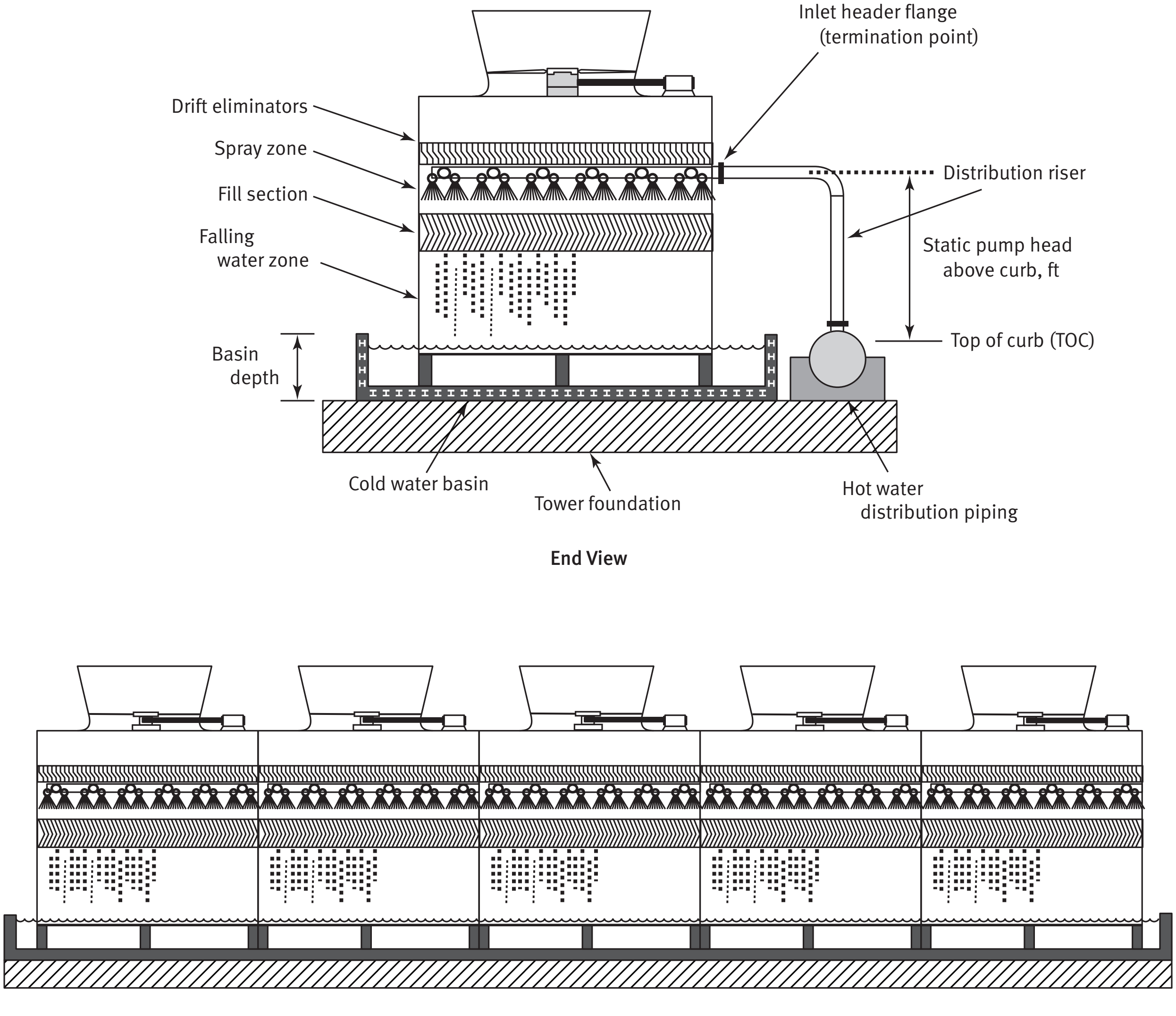


Fig. J3 Mechanical Draft: Counterflow Cooling Tower

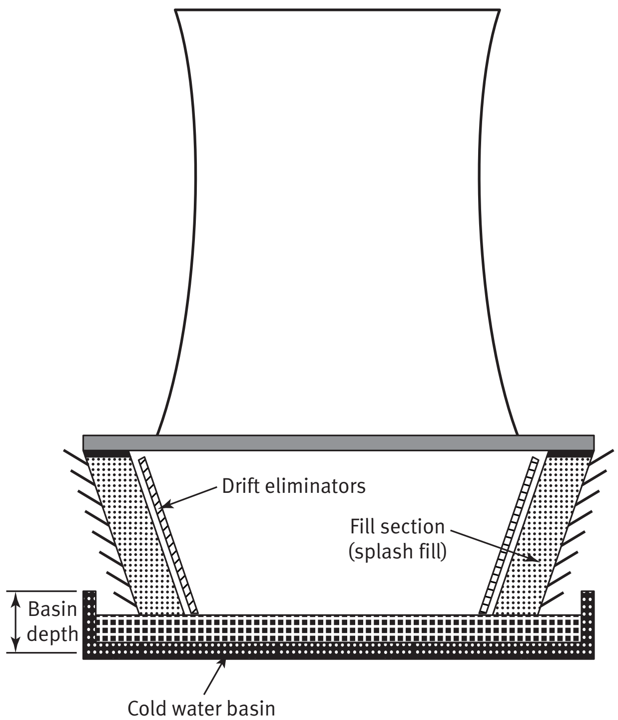


Fig. J4 Natural Draft: Crossflow Cooling Tower

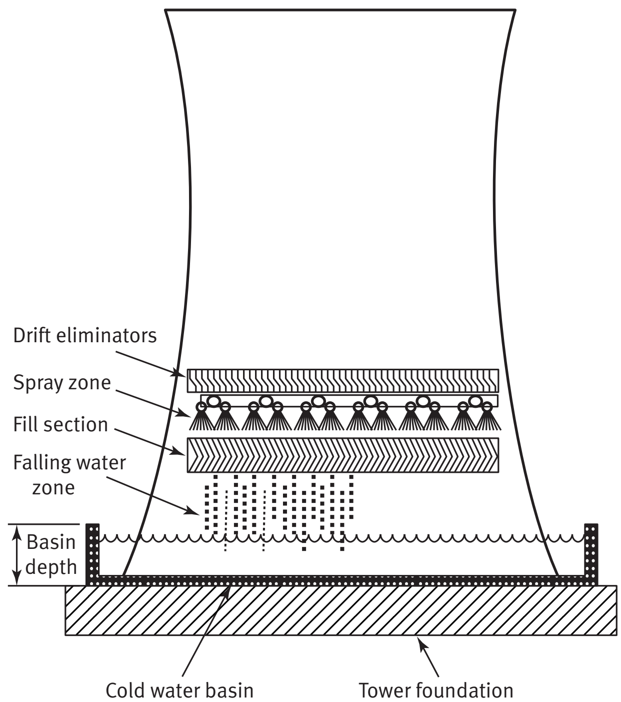
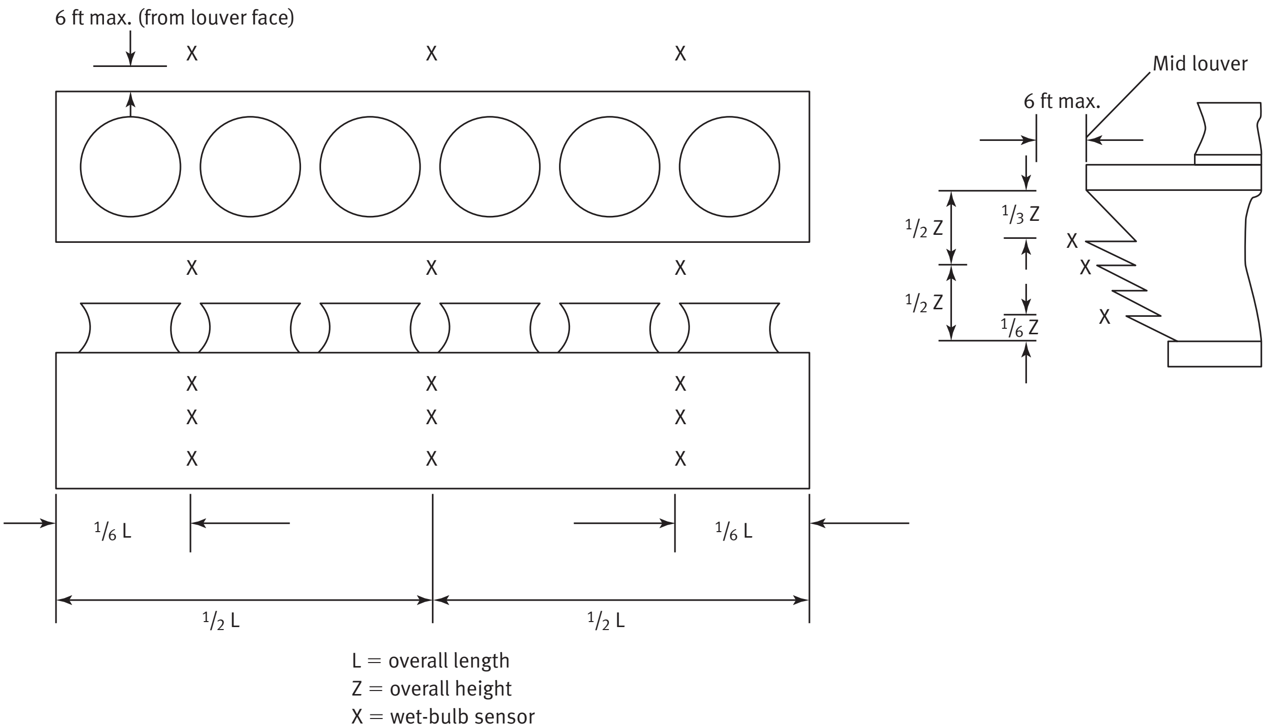
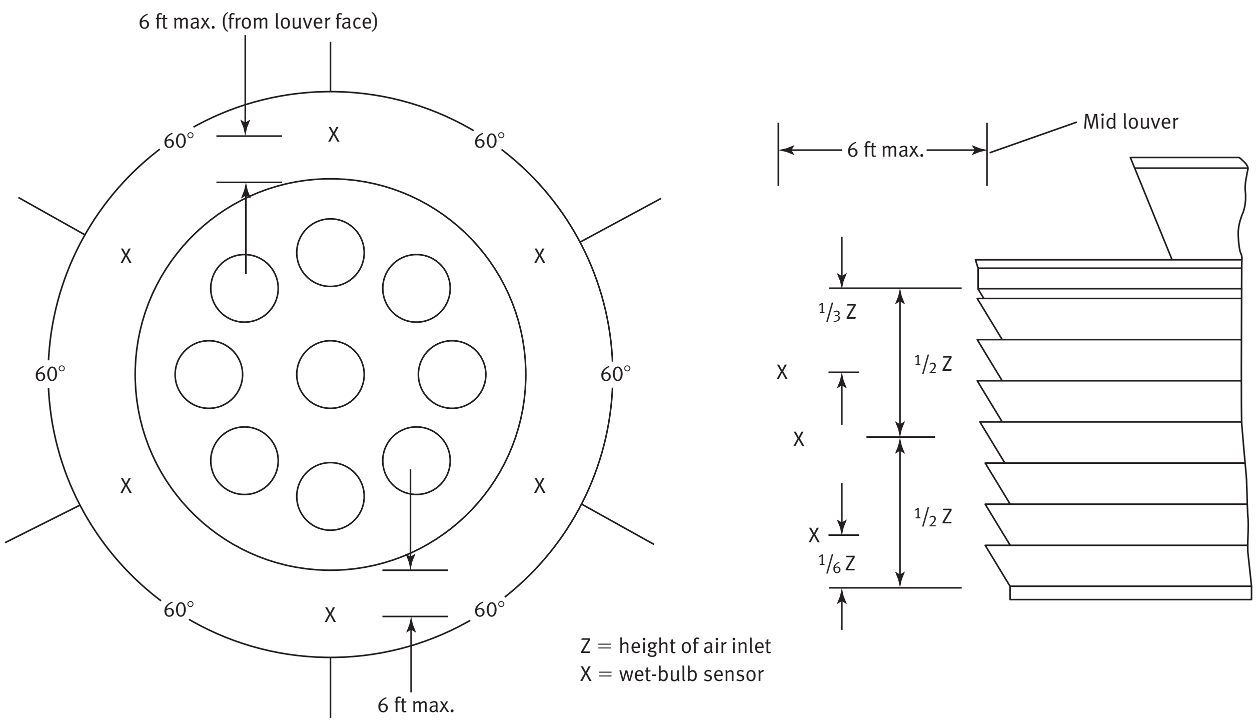


Fig. J5 Natural Draft: Counterflow Cooling Tower

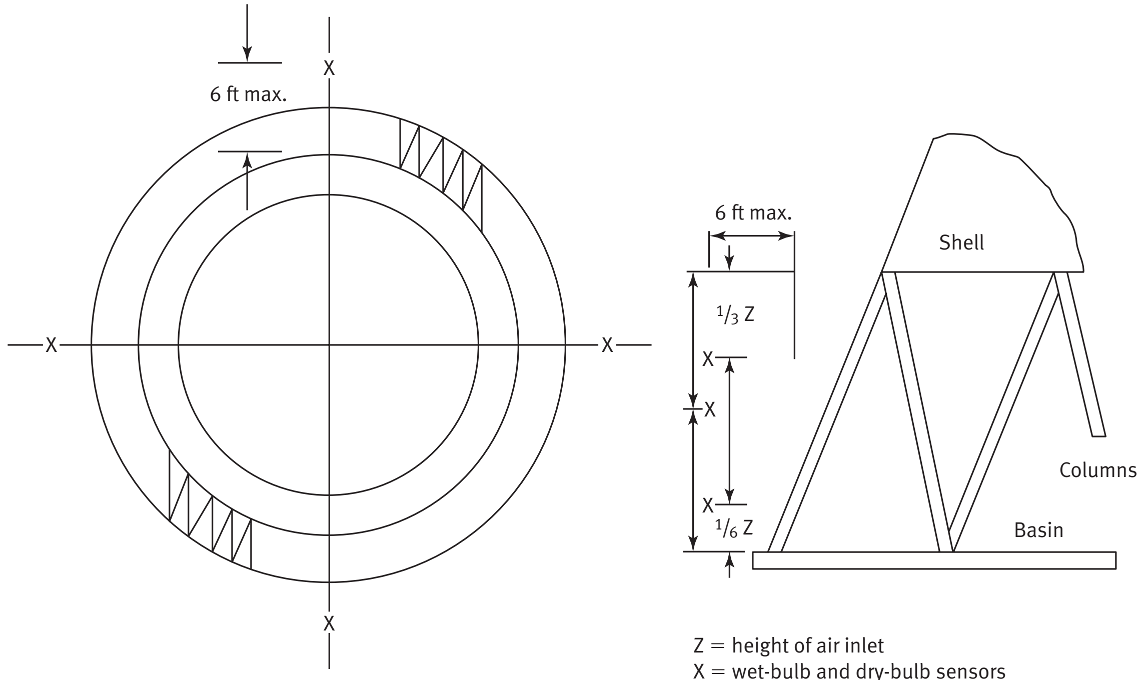


Mechanical Draft — Crossflow, Induced Draft Fans, Rectangular Configuration

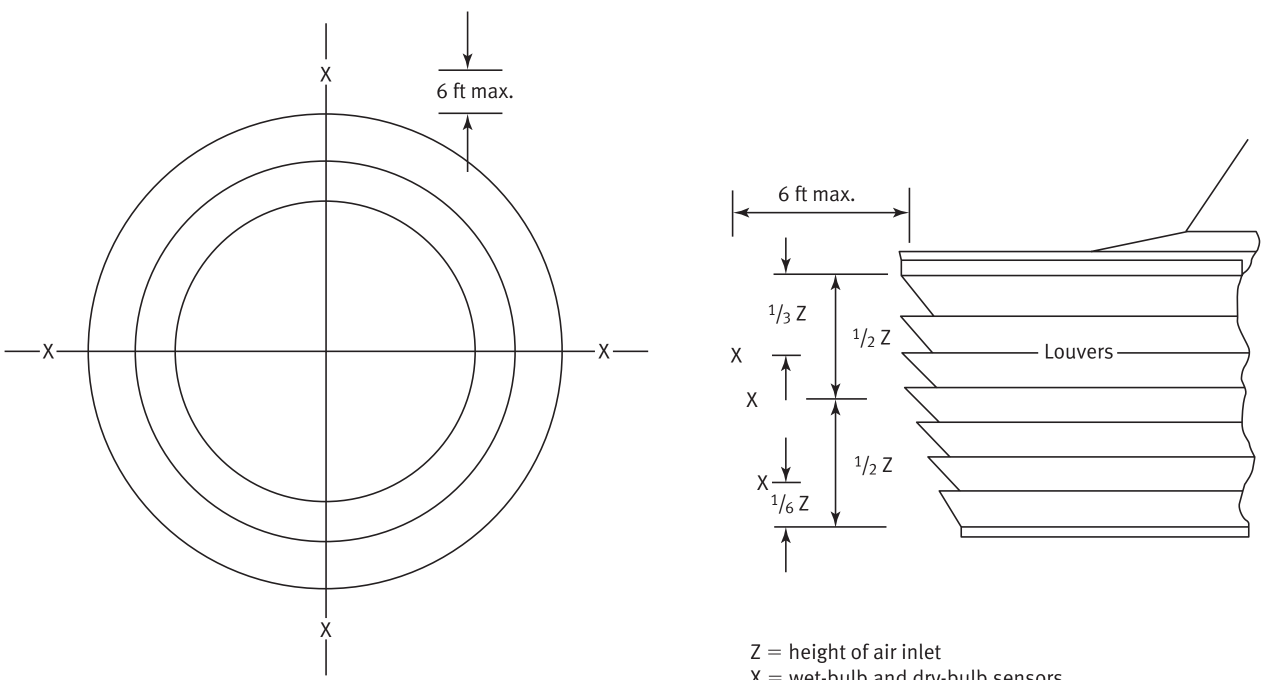


Mechanical Draft — Crossflow, Induced Draft Fans, Circular Configuration

Fig. J6 Air Temperature Sensors: Quantity and Position in Large Towers

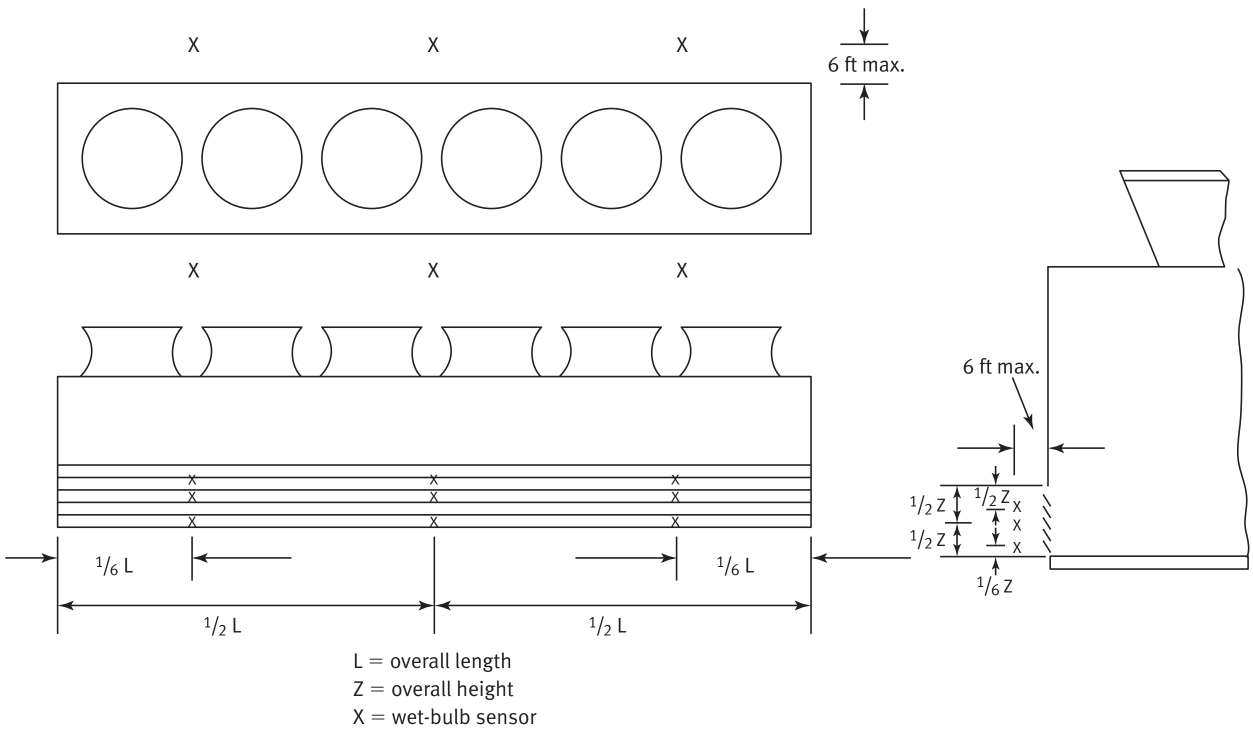


Natural Draft — Counterflow

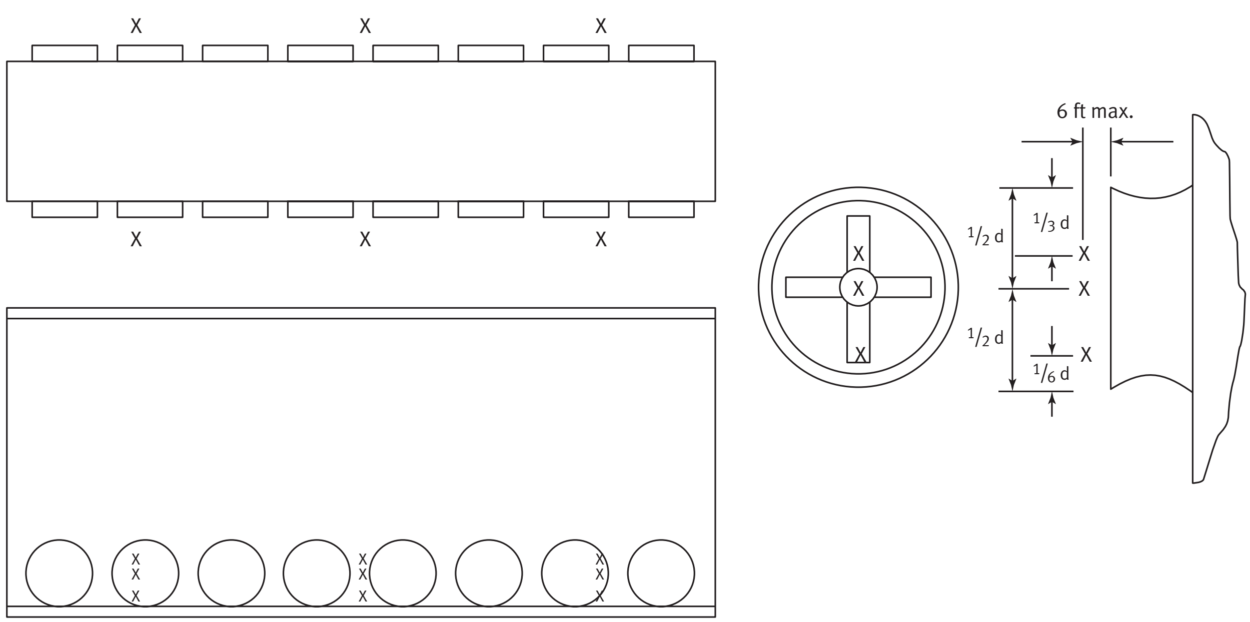


Natural Draft — Crossflow

Fig. J7 Air Temperature Sensors: Quantity and Position in Large Towers



Mechanical Draft — Crossflow, Induced Draft Fans, Rectangular Configuration



Mechanical Draft — Counterflow, Forced Draft Fans, Rectangular Configuration

Fig. J7 Air Temperature Sensors: Quantity and Position in Large Towers (Cont'd)

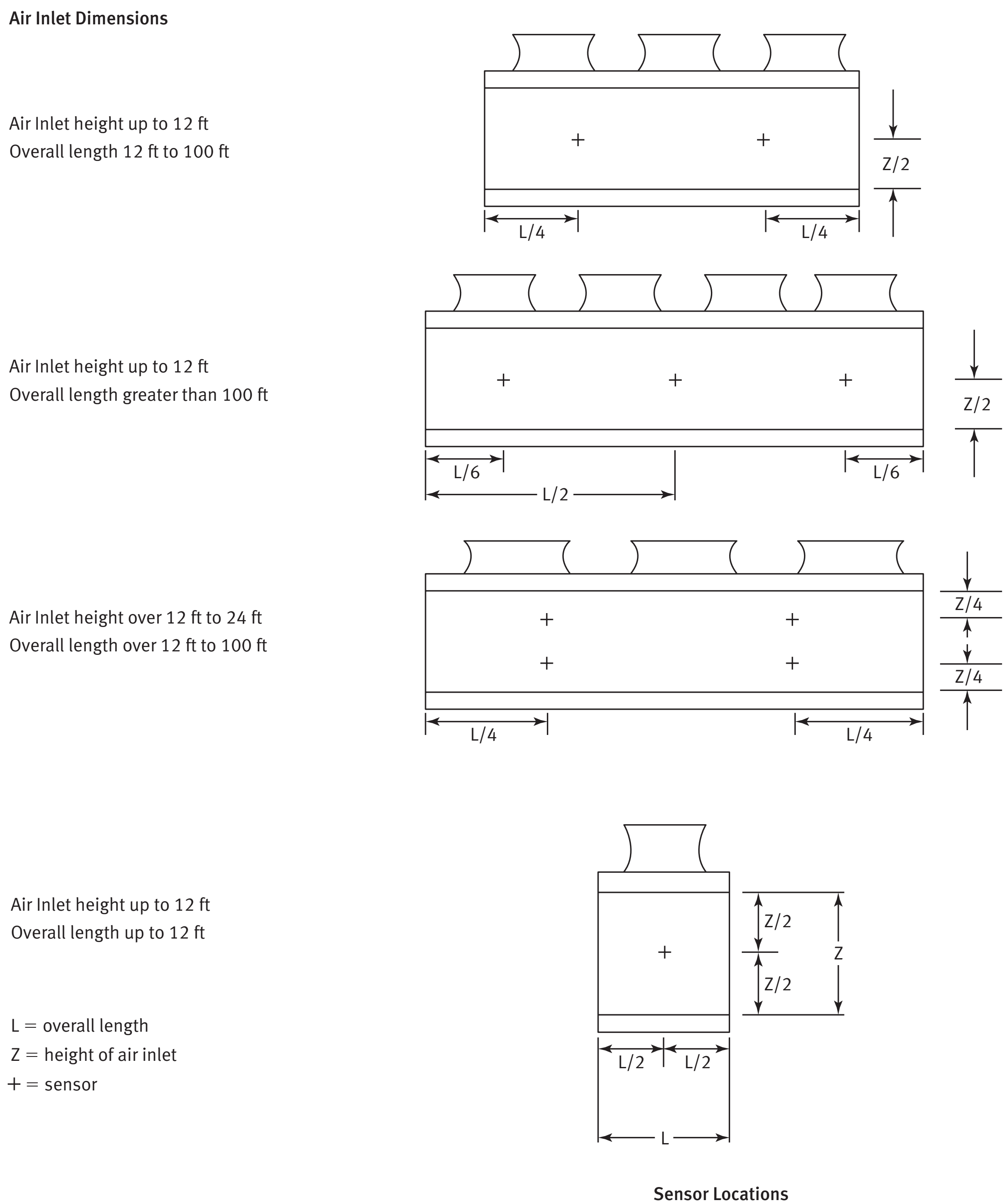
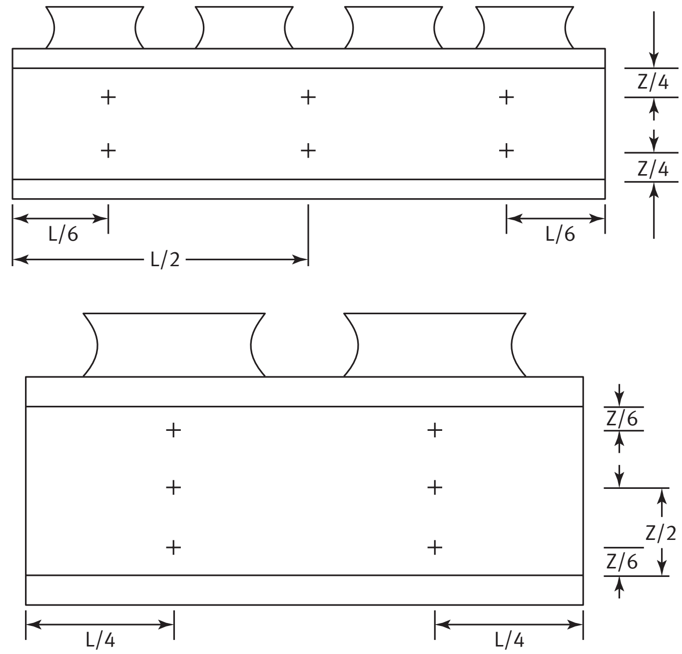


Fig. J8 Air Temperature Sensors: Quantity and Position in Small Towers (Induced Draft: Crossflow and Counterflow)

Air Inlet Dimensions

Air Inlet height over 12 ft to 24 ft
Overall length greater than 100 ft

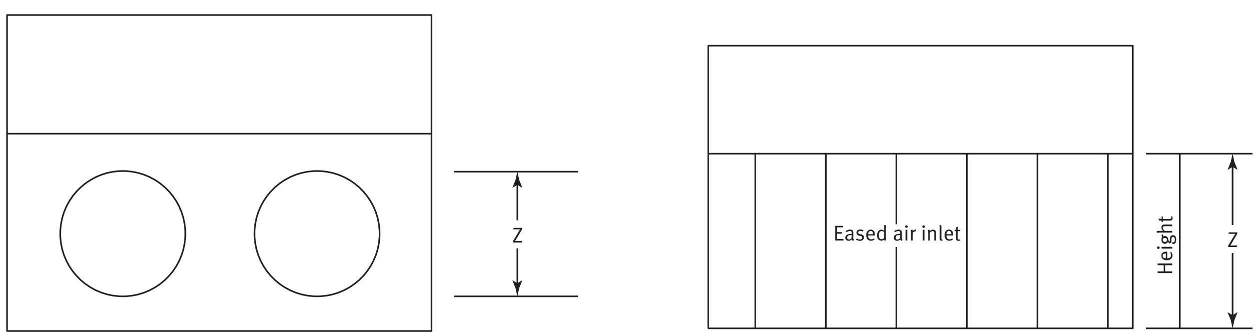
Air Inlet height greater than 24 ft
Overall length up to 75 ft



Sensor Locations

GENERAL NOTES:
(a) For the purpose of this requirement, a small tower is defined as a tower having a total air inlet area of less than 2,000 sq ft for crossflow and 1,000 sq ft for counterflow cooling towers (both faces of double inlet tower).
(b) All sensors shall be positioned no greater than 6 ft from the air inlets.
(c) Sensors are shown for one side; the other side of the double inlet tower is similar.

Fig. J8 Air Temperature Sensors: Quantity and Position in Small Towers (Induced Draft: Crossflow and Counterflow) (Cont'd)



Propeller Fan System **Centrifugal Fan System**

GENERAL NOTES:
(a) For the purpose of this requirement, the fan housing height is defined as the diameter of a propeller fan or the height of the eased air inlet of a centrifugal fan.
(b) All sensors shall be positioned no greater than 6 ft from the fan housing in the flowing air stream.
(c) Sensors are shown for one side; the other side of the double inlet is similar.

Fig. J9 Typical Forced Draft Towers

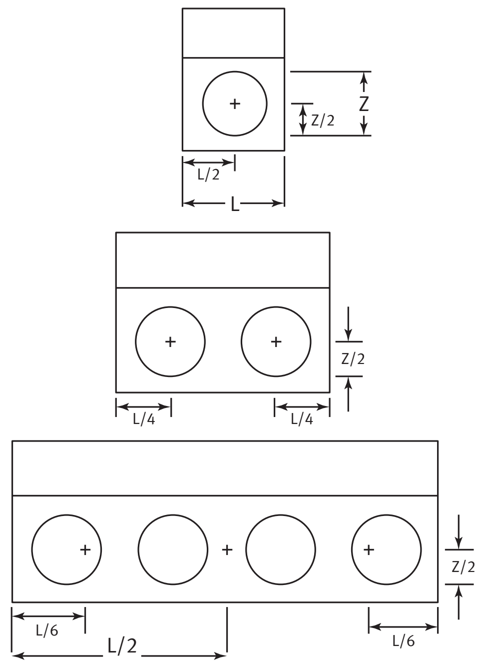
Fan Housing Dimensions

Housing height up to 4 ft
Overall length up to 12 ft

Z = fan housing height
L = overall length
+ = sensor

Housing height up to 4 ft
Overall length over 12 ft to 48 ft

Housing height up to 4 ft
Overall length over 48 ft to 96 ft

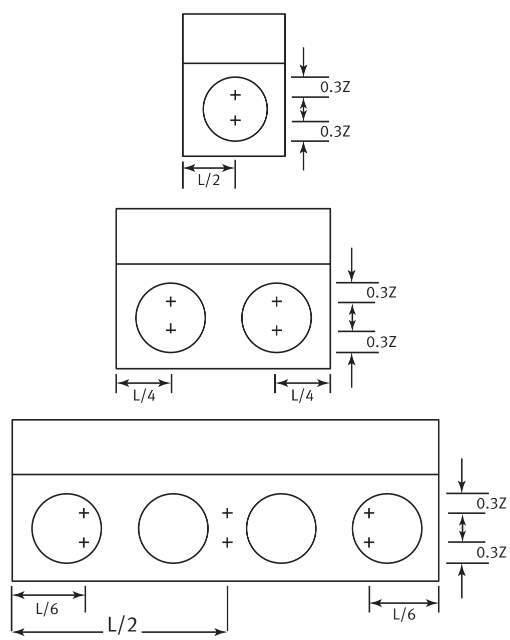


Fan Housing Dimensions

Housing height up to 4 ft
Overall length up to 12 ft

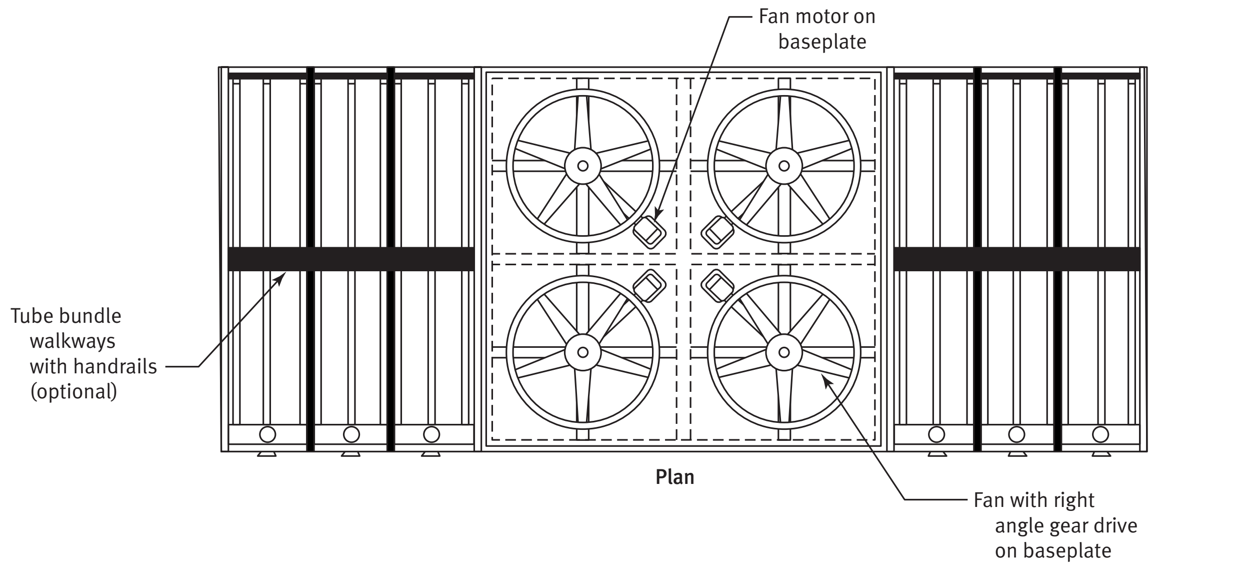
Housing height up to 4 ft
Overall length over 12 ft to 48 ft

Housing height up to 4 ft
Overall length over 48 ft to 96 ft



GENERAL NOTES:
(a) For the purpose of this requirement, the fan housing height is defined as the diameter of a propeller fan or the height of the eased air inlet of a centrifugal fan.
(b) All sensors shall be positioned no greater than 6 ft from the fan housing in the flowing air stream.
(c) Sensors are shown for one side; the other side of the double inlet is similar.

Fig. J10 Air Temperature Sensors: Quantity and Position in Small Towers (Forced Draft: Crossflow and Counterflow)



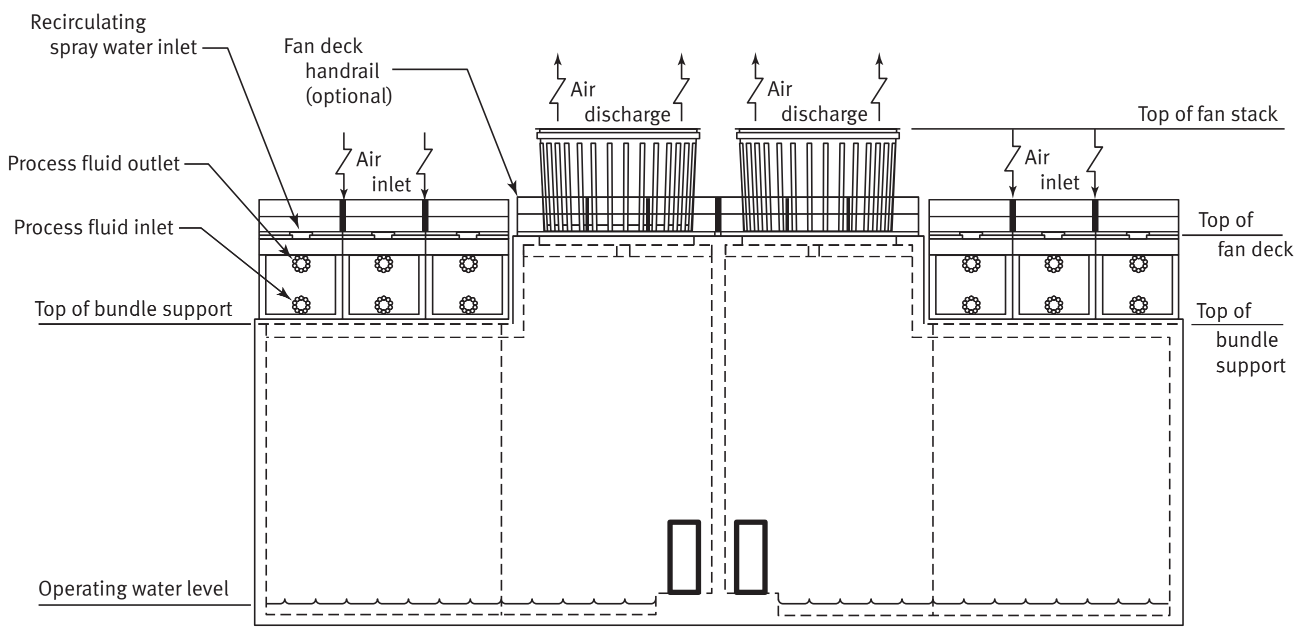


Fig. J11 Closed-Circuit Evaporative Cooler — Parallel Flow Design

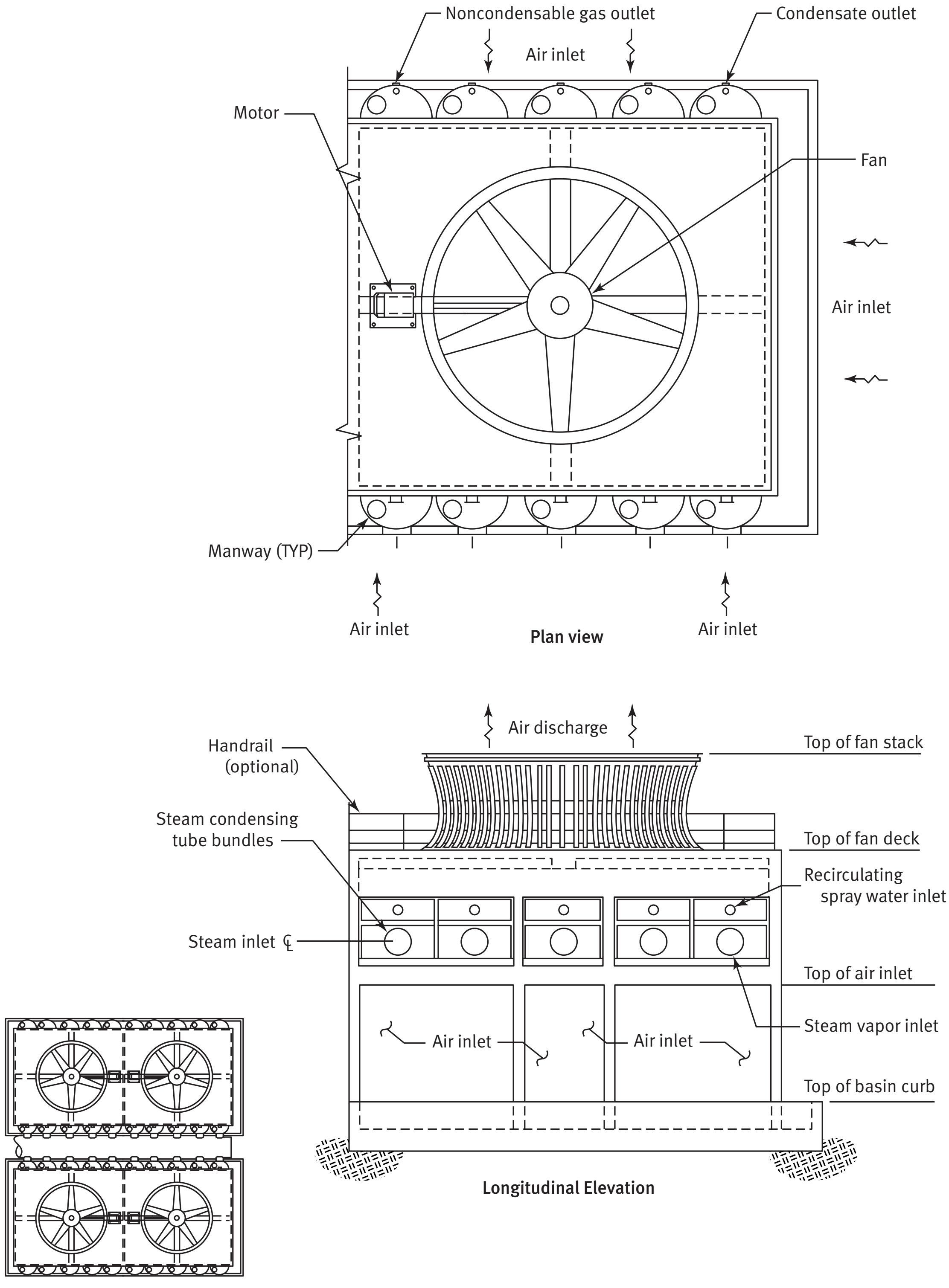


Fig. J12 Closed-Circuit Evaporative Cooler — Counterflow Design

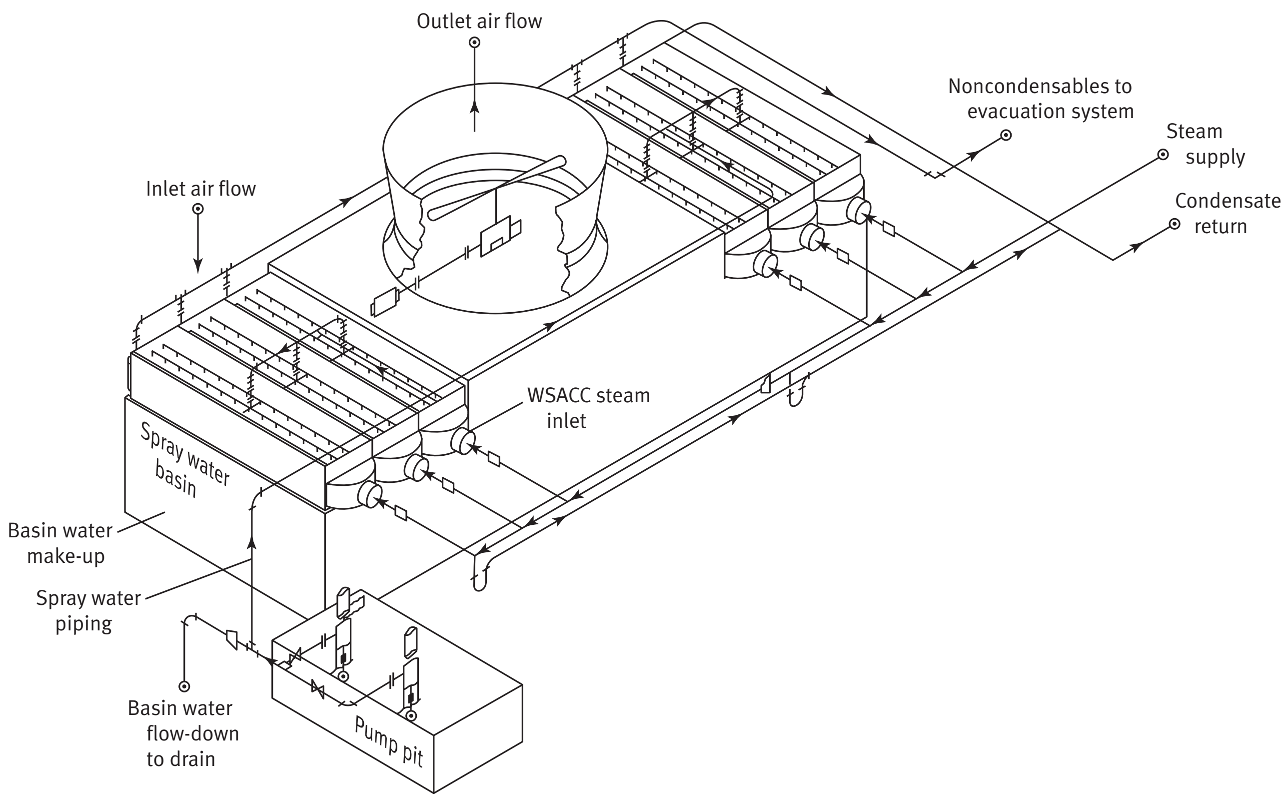


Fig. J13 Wet Surface Air-Cooled Steam Condenser (WSACC)

NONMANDATORY APPENDIX K
OTHER METHODS FOR AIR TEMPERATURE MEASUREMENT

K1 GENERAL

The alternative methods listed below produce test results that are less accurate and less reproducible than those described in Section 4. The Committee recognizes that it is sometimes necessary to perform a test using these methods, so guidance should be offered in their use. If these procedures are used, it must be by mutual agreement between the parties and only after the resulting inaccuracies have been carefully evaluated. These inaccuracies are due to the difference between the true inlet air temperature and that being measured.

For either of the following methods to yield accurate results, the true inlet wet bulb must be substantially the same as that at ground level. This will be true only if the following conditions apply:

(a) The cooling tower is not subject to interference from other cooling towers or other heat sources in the vicinity.

(b) There is no recirculation of the cooling tower plume into the cooling tower inlet. Low wind speed and a vertically rising plume may be taken as indications of no recirculation. Since cooling tower plumes may not be visible, lack of visible recirculation or interference is not sufficient to guarantee that no recirculation or interference exists. This is no stratification of air temperature in the atmosphere.

K1.1 Ambient Test

For mechanical draft cooling towers, the ambient wet-bulb temperatures shall be determined as the arithmetic average of measurements taken at not less than three locations 5 ft (1.5 m) above the basin curb elevation not less than 50 ft (15 m) and not more than 100 ft (30 m) upwind of the equipment, and equally spread along a line substantially bracketing the flow of air to the equipment. Measurements at all locations should be made simultaneously, if possible, or in rapid succession. If the locations specified are not accessible or contain equipment that can affect the measured wet-bulb temperature, alternative locations should be mutually agreed upon. For natural draft cooling tower, wet- and dry-bulb measurements shall be located at the same positions as those specified above.

K1.2 Ground Level Test

Using this methodology, air temperature measurements are made 5 ft (1.5 m) above the basin curb, within 4 ft (1.2 m) of the air inlets. For mechanical draft counterflow cooling towers, the number of psychrometers deployed should be equivalent to that specified in Section 4. Ground level air temperature measurements are not recommended for crossflow mechanical draft cooling towers, due to the significant recirculation that typically occurs.

Ground-level testing is often used for counterflow type natural draft cooling towers, as it is not easy to install psychrometers at several levels.

NONMANDATORY APPENDIX L
REPORTING FORMS

This Appendix contains the following forms:

(a) Form L1, Report of Results of Wet Mechanical and Natural Draft Cooling Towers

(b) Form L2, Report of Results for Closed-Circuit Evaporative Wet Coolers

(c) Form L3, Report of Results of Wet-Dry Cooling Towers

(d) Form L4, Report of Results for Wet Surface Air-Cooled Steam Condensers (WSACC)

Form L1 Report of Results of Wet Mechanical and Natural Draft Cooling Towers

1. General Information

(a) Number of test runs ____________
(b) Duration of test runs ____________hr ____________(s)
(c) Atmospheric pressure ____________in. Hg ____________(Pa)
(d) Wind speed/Gusts ____________/ ____________mph ____________/ ____________(m/s)
(e) Wind direction ____________
(f) Weather ____
(g) Total dissolved solids ____________ppm
(h) Oil content of circulation water ____________ppm
(i) Quantity of water stored in basin ____________lb ____________(kg)
(j) Thermal lag time ____________min ____________(s)

2. Water Flow Rates

(a) Circulating water ____________gpm ____________(m^3/s)
(b) Makeup water ____________gpm ____________(m^3/s)
(c) Blowdown water ____________gpm ____________(m^3/s)

3. Water Temperatures

(a) Hot water temperature ____________°F ____________(°C)
(b) Cold water temperature ____________°F ____________(°C)
(c) Makeup water temperature ____________°F ____________(°C)
(d) Blowdown water temperature ____________°F ____________(°C)
(e) Temperature correction for blowdown and makeup ____________±°F ____________(±°C)
(f) Temperature correction for pumps ____________±°F ____________(±°C)

4. Air Temperatures

(a) Entering wet-bulb temperature ____________°F ____________(°C)
(b) Entering dry-bulb temperature ____________°F ____________(°C)
(c) Ambient wet-bulb temperature ____________°F ____________(°C)
(d) Ambient dry-bulb temperature ____________°F ____________(°C)
(e) Natural draft tower dry-bulb temperature at top of air inlet ____________°F ____________(°C)
(f) Natural draft tower atmospheric temperature gradient ____________°F/ft ____________(°C/m)

5. Tower Pumping Head

(a) Height from centerline of connecting flange to basin curb ____________ft ____________(m)
(b) Head at centerline of connecting flange ____________ft ____________(m)
(c) Velocity head ____________ft ____________(m)

6. Fan and Pumping Information

(a) Motor input ____________kW
(b) Efficiency of motor ____________%
(c) Power output from motor ____________hp ____________(W)
(d) Circulating water pump power ____________ hp ____________ (W)
(e) Circulating water pump efficiency ____________%
(f) Circulating water pressure at test well ____________psig ____________(Pa)

7. Air Flow Rate

(a) Volume rate (dry) ____________ft^3/min ____________(m^3/s)
(b) Mass rate (mixture) ____________lb/hr ____________(kg/s)

8. Exhaust Air Temperatures

(a) Average wet-bulb temperature ____________ °F ____________ (°C)
(b) Average dry-bulb temperature ____________ °F ____________ (°C)

9. Range ____________°F ____________(°C)

10. Approach ____________°F ____________(°C)

11. Evaporation Loss ____________lbm/hr ____________(kg/s)

Form L1 Report of Results of Wet Mechanical and Natural Draft Cooling Towers (Cont'd)

12. Comparison of Performance

(a) Guaranteed water flow rate at test conditions ____________gpm ____________(m^3/s)
(b) Adjusted water flow rate at test conditions ____________gpm ____________(m^3/s)
(c) Tower capability ____________%
(d) Guaranteed cold water temperature (from guaranteed performance curves) ____________°F____________ (°C)
(e) Corrected test cold water temperature ____________°F ____________(°C)
(f) Test uncertainty (capability) ____________±%
(g) Recirculation ____________°F ____________(°C)

13. Comparison of Tower Pumping Heads

(a) Guaranteed tower pumping head at specified water flow ____________ft ____________(m)
(b) Measured tower pumping head corrected to specific flow ____________ft ____________(m)
(c) Difference [Items 13(a) and 13(b)] ____________ft ____________(m)

14. Plume Orientation ____________deg from vertical

Form L2 Report of Results for Closed-Circuit Evaporative Wet Coolers

1. General Information

(a) Number of test runs ____________
(b) Duration of test runs ____________hr ____________(s)
(c) Atmospheric pressure ____________in. Hg ____________(Pa)
(d) Wind speed/Gusts ____________/ ____________mph ____________/ ____________(m/s)
(e) Wind direction ____________
(f) Weather ____
(g) Total dissolved solids of spray water ____________ppm
(h) Oil content of circulation water ____________ppm
(i) Thermal lag time ____________min ____________(s)

2. Flow Rates

(a) Process fluid ____________gpm ____________(m^3/s)
(b) Spray water ____________gpm ____________(m^3/s)
(c) Blowdown water ____________gpm ____________(m^3/s)
(d) Makeup flow, if any ____________gpm ____________(m^3/s)

3. Water Temperatures

(a) Hot fluid temperature ____________°F ____________(°C)
(b) Cold fluid temperature ____________°F ____________(°C)
(c) Spray (basin) water temperature ____________°F ____________(°C)
(d) Makeup water, if any, temperature ____________°F ____________(°C)
(e) Blowdown temperature ____________°F ____________(°C)
(f) Temperature correction for blowdown and makeup ____________±°F ____________(±°C)
(g) Temperature correction for pumps ____________±°F ____________(±°C)

4. Air Temperatures

(a) Entering wet-bulb temperature ____________°F ____________(°C)
(b) Entering dry-bulb temperature ____________°F ____________(°C)
(c) Ambient wet-bulb temperature ____________°F ____________(°C)
(d) Ambient dry-bulb temperature ____________°F ____________(°C)

5. Tower Pumping Head

(a) Height from centerline of connecting flange to basin curb ____________ft ____________(m)
(b) Head at centerline of connecting flange ____________ft ____________(m)
(c) Velocity head ____________ft ____________(m)

6. Fan and Pumping Information

(a) Motor input ____________kW
(b) Efficiency of motor ____________%
(c) Power output from motor ____________hp ____________(W)
(d) Spray water pump power ____________hp ____________(W)
(e) Spray water pump efficiency ____________%
(f) Spray water pressure at test well ____________psig ____________(Pa)

7. Air Flow Rate

(a) Volume rate (dry) ____________ft^3/min ____________(m^3/s)
(b) Mass rate (mixture) ____________lb/hr ____________(kg/s)

8. Exhaust Air Temperatures

(a) Average wet-bulb temperature ____________°F ____________(°C)
(b) Average dry-bulb temperature ____________°F ____________(°C)

9. Process Fluid Range ____________°F ____________(°C)

10. Process Fluid Approach ____________°F ____________(°C)

11. Evaporation Loss ____________lbm/hr ____________(kg/s)

12. Heat Exchanger Pressure Drop ____________psi ____________(Pa)

Form L2 Report of Results for Closed-Circuit Evaporative Wet Coolers (Cont'd)

13. Comparison of Performance

(a) Guaranteed fluid flow rate at test conditions ____________gpm ____________(m^3/s)
(b) Adjusted fluid flow rate at test conditions ____________gpm ____________(m^3/s)
(c) Tower capability ____________%
(d) Guaranteed cold fluid temperature (from guaranteed performance curves) ____________°F ____________(°C)
(e) Corrected test cold fluid temperature ____________°F ____________(°C)
(f) Test uncertainty (capability) ____________±%
(g) Recirculation ____________°F ____________(°C)

14. Comparison of Fan HP and Pump Head

(a) Guaranteed tower pumping head at specified water flow ____________ft ____________(m)
(b) Measured tower pumping head corrected to specific flow ____________ft ____________(m)
(c) Guaranteed fan hp ____________
(d) Measured fan hp ____________

Form L3 Report of Results of Wet-Dry Cooling Towers

1. General Information
(a) Type of test: Thermal Performance ____________, Plume Abatement ____________
(b) Number of test runs ____________
(c) Duration of test runs____________ hr ____________(s)
(d) Atmospheric pressure ____________in. Hg ____________(Pa)
(e) Wind speed/gusts ____________/ ____________mph ____________/ ____________(m/s)
(f) Wind direction ____________
(g) Weather ____
(h) Total dissolved solids of circulation water ____________ppm
(i) Oil content of circulation water ____________ppm
(j) Thermal lag time ____________min ____________(s)

2. Water Flow Rates
(a) Circulating water ____________gpm ____________(m^3/s)
(b) Makeup water ____________gpm ____________(m^3/s)
(c) Blowdown water ____________gpm ____________(m^3/s)

3. Water Temperatures
(a) Hot water temperature ____________°F ____________(°C)
(b) Cold water temperature ____________°F ____________(°C)
(c) Makeup water temperature ____________°F ____________(°C)
(d) Blowdown water temperature ____________°F ____________(°C)
(e) Temperature correction for blowdown and makeup ____________±°F ____________(±°C)
(f) Temperature correction for pumps ____________±°F ____________(±°C)

4. Air Temperatures
(a) Entering wet-bulb temperature __________°F __________(°C)
(b) Entering dry-bulb temperature __________°F __________(°C)
(c) Ambient wet-bulb temperature __________°F __________(°C)
(d) Ambient dry-bulb temperature __________°F __________(°C)

5. Tower Pumping Head
(a) Height from centerline of connecting flange to basin curb ____________ft ____________(m)
(b) Head at centerline of connecting flange ____________ft ____________(m)
(c) Velocity head ____________ft ____________(m)

6. Fan and Pumping Information
(a) Motor input ____________kW
(b) Efficiency of motor ____________%
(c) Power output from motor ____________hp ____________(W)
(d) Circulating water pump power __________hp __________(W)
(e) Circulating water pump efficiency ____________%
(f) Circulating water pressure at test well ____________psig ____________(Pa)

7. Air Flow Rate
(a) Volume rate (dry) ____________ft^3/min ____________(m^3/s)
(b) Mass rate (mixture) ____________lb/hr ____________(kg/s)

8. Exhaust Air Temperatures
(a) Average wet-bulb temperature __________°F __________(°C)
(b) Average dry-bulb temperature __________°F __________(°C)

9. Range ____________°F ____________(°C)
10. Approach ____________°F ____________(°C)
11. Evaporation Loss ____________lbm/hr ____________(kg/s)

Form L3 Report of Results of Wet-Dry Cooling Towers (Cont'd)

12. Comparison of Performance

(a) Guaranteed water flow rate at test conditions ____________gpm ____________(m^3/s)
(b) Adjusted water flow rate at test conditions ____________gpm ____________(m^3/s)
(c) Tower capability ____________%
(d) Guaranteed cold water temperature (from guaranteed performance curves) ____________°F ____________(°C)
(e) Corrected test cold water temperature ____________°F ____________(°C)
(f) Test uncertainty (capability) ____________±%
(g) Recirculation ____________°F ____________(°C)

13. Comparison of Tower Pumping Heads

(a) Guaranteed tower pumping head at specified water flow ____________ft ____________(m)
(b) Measured tower pumping head corrected to specific flow ____________ft ____________(m)
(c) Difference [Items 13(a) and 13(b)] ____________ft ____________(m)

14. Plume Indicator

(a) Visual Inspection: No plume ____________, Wispy ____________, Light____________, Heavy ____________
(b) Measured exhaust relative humidity ____________%
(c) Guaranteed exhaust relative humidity ____________%
(d) Plume indicator ____________% (c)/(b)
(e) Mixing quality coefficient

Form L4 Report of Results for Wet Surface Air-Cooled Steam Condensers (WSACC)

1. General Information
(a) Number of test runs ______
(b) Duration of test runs ______hr ______(s)
(c) Atmospheric pressure ______in. Hg ______(Pa)
(d) Wind speed/Gusts ______/ ______mph ______/ ______(m/s)
(e) Wind direction ______
(f) Weather ______
(g) Total dissolved solids of spray water ______ppm
(h) Oil content of spray water ______ppm
(i) Thermal lag time ______min ______(s)

2. Flow Rates
(a) Steam mass flow rate ______lbm/hr ______(kg/s)
(b) Spray water ______gpm ______(m^3/s)
(c) Blowdown water ______gpm ______(m^3/s)
(d) Makeup flow, if any______ gpm ______(m^3/s)

3. Water Temperatures
(a) Spray (basin) water temperature ______°F ______(°C)
(b) Makeup water, if any, temperature______ °F ______(°C)
(c) Blowdown temperature ______°F ______(°C)
(d) Temperature correction for blowdown and makeup ______±°F ______(±°C)
(e) Temperature correction for pumps ______±°F ______(±°C)

4. Air Temperatures
(a) Entering wet-bulb temperature ______°F ______(°C)
(b) Entering dry-bulb temperature ______°F ______(°C)
(c) Ambient wet-bulb temperature ______°F ______(°C)
(d) Ambient dry-bulb temperature ______°F ______(°C)

5. Tower Pumping Head
(a) Height from centerline of connecting flange to basin curb ______ft ______(m)
(b) Head at centerline of connecting flange ______ft ______(m)
(c) Velocity head ______ft ______(m)

6. Fan and Pumping Information
(a) Motor input ______kW
(b) Efficiency of motor ______%
(c) Tower output from motor ______hp ______(W)
(d) Spray water pump power ______hp ______(W)
(e) Spray water pump efficiency ______%
(f) Spray water pressure at test well ______psig ______(Pa)

7. Air Flow Rate
(a) Volume rate (dry) ______ft^3/min ______(m^3/s)
(b) Mass rate (mixture) ______lb/hr ______(kg/s)

8. Exhaust Air Temperatures
(a) Average wet-bulb temperature ______°F ______(°C)
(b) Average dry-bulb temperature ______°F ______(°C)

9. Heat Load ______BTU/hr ______(Kcal/min)

10. Approach ______°F ______(°C)

11. Evaporation Loss ______lbm/hr ______(kg/s)

12. Steam Quality ______%

13. Condenser Steam Side Pressure ______in. Hga ______(Pa)

Form L4 Report of Results for Wet Surface Air-Cooled Steam Condensers (WSACC) (Cont'd)

14. Noncondensable Gases

Noncondensable gas load ____________SCFM ____________(m^3/s)

15. Comparison of Performance

(a) Guaranteed steam flow rate at test conditions ____________lb/hr ____________(kg/s)
(b) Adjusted steam flow rate at test conditions ____________lb/hr ____________(kg/s)
(c) WSACC capability ____________%
(d) Guaranteed condensing pressure ____________in. Hga ____________Pa
(e) Entering wet bulb temperature ____________°F ____________(°C)
(f) Test uncertainty (capability) ____________±%

16. Comparison of Fan Horsepower and Pump Head

(a) Guaranteed WSACC pumping head at specified water flow ____________ft ____________(m)
(b) Measured WSACC pumping head corrected to specific flow ____________ft ____________(m)
(c) Guaranteed fan hp ____________
(d) Measured fan hp ____________

NONMANDATORY APPENDIX M
REFERENCES

ASHRAE Handbook of Fundamentals, 1981

Publisher: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. (ASHRAE), 1791 Tullie Circle, NE, Atlanta, GA 30329

ASME PTC 19.1, Test Uncertainty, 1995 and 1998 editions

ASME PTC 23, Atmospheric Water Cooling Equipment, 1986 edition

Publisher: The American Society of Mechanical Engineers (ASME International), Three Park Avenue, New York, NY 10016-5990; Order Department: 22 Law Drive, Box 2300, Fairfield, NJ 07007-2300

CTI ATC-105, Acceptance Test Code for Closed Circuit Cooling Towers, 1997

CTI ATC-140, Isokinetic Drift Measurement Test Code for Water Cooling Tower

CTI Cooling Tower Manual, Field Test Handbook, 1983

CTI Bulletin STD-146, Standard Water Flow Measurement

Publisher: Cooling Technology Institute (CTI), 2611 FM 1960 West, Houston, TX 77068-3730

PGT TIN 2001-1640, Uncertainty Analysis and Sample Calculations (internal document), Hennon and Wheeler

Publisher: Power Generation Technologies (PGT), 200 Tech Center Drive, Knoxville, TN 37912

Hennon and Wheeler, "Uncertainty Analysis of Cooling Tower Performance." Paper presented at the American Power Conference, 1996

Laidlaw, I. M. S. and Smart, P. L., "An Evaluation of Fluorescent Dyes for Water Tracing," *Water Resources Research* 13, no. 1 (February 1977):15-33

Turner, D. B. *Workbook of Atmospheric Dispersion Estimates*[1]

Wark, Kenneth and Warner, Cecil F., *Air Pollution: Its Origin and Control*, 3rd ed. Old Tappan, NJ: Addison-Wesley Publishers

[1] Copies may be obtained from National Technical Information Service, U.S. Department of Commerce, 5285 Port Royal Road, Springfield, VA 22151.

Intentionally left blank

Intentionally left blank

ISBN 0-7918-2831-X
9 780791 828311
C05503