2023 Standard for

Performance Rating of Forced-circulation Air-cooling and Air-heating Coils

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ICS Code: 23.120

Note:

This standard supersedes AHRI Standard 410-2001 with Addenda 1, 2, and 3.

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Intent

This standard is intended for the guidance of the industry, including manufacturers, engineers, installers, contractors, and users.

Review and Amendment

This standard is subject to review and amendment as technology advances.

2023 Edition

This edition of AHRI Standard 410 (SI/I-P)*, Performance Rating of Forced-circulation Air-cooling and Air-heating Coils*, was prepared by Air Cooling and Air Heating Coils (ACHC) Standards Technical Committee (STC). The standard was approved by the Applied Standards Subcommittee in May 2023.

Origin and Development of AHRI Standard 410

The initial publications were ARI Standard 410-1956, *Forced Circulation Air-Cooling Coils* and ARI Standard 411-1956, *Methods of Testing and Rating Forced Circulation Air-Cooling and Air-Heating Coils*. Subsequent revisions were:

ARI Standard 410-1958, *Forced Circulation Air-Cooling Coils*

ARI Standard 410-1964, *Forced Circulation Air-Cooling Coils*

ARI Standard 410-1972, *Standard for Forced Circulation Air-Cooling and Air-Heating Coils*

ARI Standard 410-1981, *Standard for Forced Circulation Air-Cooling and Air-Heating Coils*

ARI Standard 410-1987, *Standard for Forced Circulation Air-Cooling and Air-Heating Coils*

ARI Standard 410-1991, *Standard for Forced Circulation Air-Cooling and Air-Heating Coils*

AHRI Standard 410-2001, *Forced-circulation Air-cooling and Air-heating Coils*

AHRI Standard 410-2001 (with Addendum 1), *Forced-circulation Air-cooling and Air-heating Coils*

AHRI Standard 410-2001 (with Addenda 1 and 2), *Forced-circulation Air-cooling and Air-heating Coils*

AHRI Standard 410-2001 (SI/I-P) (with Addenda 1, 2 and 3) *Forced-circulation Air-cooling and Air-heating Coils*

Summary of Changes

AHRI Standard 410-2023 (SI/I-P) contains the following update(s) to the previous edition:

- Convert to a joint unit (SI/I-P) standard
- Update schematic figures to include joint units
- Enhance 100+ equations, including those used to determine coil performance

Committee Personnel

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Air Cooling and Air Heating Coils Standards Technical Committee Scope:

The Air-Cooling and Air-Heating Coils (ACHC) Standards Technical Committee is responsible for the development and maintenance of AHRI standards and guidelines pertaining to Forced-circulation Air-cooling and Air-heating Coils.

This STC applies to Forced-Circulation Air-Cooling Coils under non-frosting conditions and Forced-Circulation Air-Cooling and Air-Heating Coils only intended for:

- Field installation (built-up systems and duct-mounted coils); and
- Use in central station air-handling units.

Coils in other applications and products are out of scope.

Product definitions are as defined within AHRI Standard 410.

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Applied Standards Subcommittee Scope:

The scope of the Applied Standards Subcommittee is standards and guidelines related to the end products that are part of the AHRI Applied Industry Sector. (The definition of and list of products associated with each sector are found on the AHRI website at http://www.ahrinet.org.)

This list represents the membership at the time the Standards Technical Committee and Standards Subcommittee were balloted on the final text of this edition. Since that time, changes in the membership may have occurred. Membership on these committees shall not in and of itself constitute an endorsement by the committee members or their employers of any document developed by the committee on which the member serves.

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PERFORMANCE RATING OF FORCED-CIRCULATION AIR-COOLING AND AIR-HEATING COILS

Section 1. Purpose

This standard establishes definitions, classifications, test requirements, rating requirements, minimum data requirements for *published ratings*, operating requirements, marking and nameplate data, and conformance conditions for *forced-circulation air-cooling and air-heating coils*.

Section 2. Scope

2.1   Scope

This standard applies to *forced-circulation air-cooling and air-heating coils* as defined in Section 3.

2.2   Exclusions

This standard does not apply to:

- *Forced-circulation air-cooling coils* under frosting conditions
- *Microchannel coils*
- *Bare tube coils*
- Air-cooling *refrigerant evaporator coils* for direct-expansion *refrigerant* systems with *refrigerant* liquid flow controlled by:
  - o   Capillary tube assemblies
  - o   Restrictor orifices
- Air-cooling *refrigerant evaporator coils* for flooded *refrigerant* systems
- Air-heating *refrigerant* condenser coils for *refrigerant* systems
- Indoor or outdoor *refrigerant* coils for heat pump *refrigerant* systems
- *Forced-circulation air-cooling and air-heating coils* for airside fluids other than air
- Non-round tube coils

Section 3. Definitions

All terms in this document shall follow the standard industry definitions in the ASHRAE Terminology website unless otherwise defined in this section.

3.1   Expression of Provisions

Terms that provide clear distinctions between requirements, recommendations, permissions, options, and capabilities.

3.1.1   "Can" or "cannot"

Express an option or capability.

3.1.2   "May"

Signifies a permission expressed by the document.

3.1.3   "Must"

Indication of unavoidable situations and does not mean that an external constraint referred to is a requirement of the document.

3.1.4 **"Shall" or "shall not"**

Indication of mandatory requirements to strictly conform to the standard and where deviation is not permitted.

3.1.5 **"Should" or "should not"**

Indication of recommendations rather than requirements. In the negative form, a recommendation is the expression of potential choices or courses of action that is not preferred but not prohibited.

3.2 **Standard Specific Definitions**

3.2.1 **Aqueous Glycol**

Refers only to aqueous ethylene glycol and aqueous propylene glycol solutions. The thermodynamic and thermophysical properties of *aqueous glycol* are found in the latest edition of the *ASHRAE Handbook – Fundamentals* or from the glycol's manufacturer.

3.2.2 **Azeotropic**

A *blend* containing two or more *refrigerants* whose equilibrium vapor and liquid-phase compositions are the same at a given pressure.

3.2.3 **U.S. Standard Atmosphere at Sea Level**

Atmospheric air pressure at sea level (altitude, Z, equal to 0 m (0 ft)), $P_{b\ STD}$, is 101.325 kPa (29.921 in Hg) abs.

3.2.4 **Job Conditions**

The barometric pressure expressed as $P_{b\ JOB}$, as function of altitude, Z, that is calculated using Equation 1 for SI units and Equation 2 for IP units. This equation is accurate for altitude, Z, from -305 m (-1000 ft) to 9150 m (30000 ft).

$$P_{b\ JOB} = 101.325(1 - 2.2558 \text{ x } 10^{-6}\ Z)^{5.2559} \qquad\qquad 1$$

$$P_{b\ JOB} = 29.921(1 - 6.8754 \text{ x } 10^{-6}\ Z)^{5.2559} \qquad\qquad 2$$

3.2.5 **Blend**

A *refrigerant* consisting of a mixture of two or more different chemical compounds that can be used individually as *refrigerants* for other applications. *Refrigerant* mixtures of two or more different chemical compounds are *azeotropic*, *near azeotropic*, or *zeotropic*.

3.2.6 **Bubble Point**

The liquid saturation temperature of a *refrigerant* at the specified pressure and is the temperature where a liquid *refrigerant* first begins to boil.

3.2.7 **Circuiting**

The individual *tube passes* in a coil that can be interconnected by return bends to form the serpentine arrangement of multi-pass tube circuits. Coils can be used with different circuit arrangements and combinations.

3.2.8 **Forced-circulation Air-cooling and Air-heating Coil**

A heat exchanger, with or without extended surfaces, where tubeside fluid is circulated for the purpose of total cooling (sensible cooling plus latent cooling) or heating of a forced-circulation air stream.

3.2.8.1 **Bare Tube Coil**

A coil without *fins*.

3.2.8.2   **Coil Line**

A product offering of coils with the following in common:

- Tubeside fluid (*single-phase liquid*, steam, or *refrigerant*)
- Tube size, spacing, and arrangement (parallel or staggered)
- Tube internal construction
- *Fin* configuration (not spacing)

3.2.8.3   **Forced-circulation Air-cooling Coil**

A heat exchanger, with or without extended surfaces, where either cold *single-phase liquid* or *refrigerant* is circulated, for the purpose of total cooling (sensible cooling plus latent cooling) of a forced-circulation air stream.

Note: This can be referred to as an air-cooling coil.

3.2.8.4   **Forced-circulation Air-heating Coil (Air-heating Coil)**

A heat exchanger, with or without extended surfaces, where either hot *single-phase liquid* or steam is circulated for the purpose of heating of a forced-circulation air stream.

3.2.8.5   **Integral Face-and-bypass Coil**

A coil (can be with steam as the tubeside fluid) that has damper(s) built in that controls the amount of airflow that bypasses the heat transfer surface.

3.2.8.6   **Microchannel Coil**

A coil (can be aluminum) with multiple flat tubes containing small channels (microchannels) where a tubeside fluid (that can be *refrigerant*) flows. Angled and louvered *fins* are located in between the flat tubes. Coil *circuiting* is accomplished by placing baffles in the distribution manifolds to feed the *refrigerant* through the flat tubes.

3.2.8.7   **Refrigerant Evaporator Coil**

An air-cooling coil using *refrigerant* as the tubeside fluid for direct-expansion *refrigerant* systems with *refrigerant* liquid flow controlled by an expansion valve. This coil can be described as a DX evaporator coil (direct-expansion evaporator coil).

3.2.8.8   **Single-phase Liquid Coil**

An air-cooling coil or *air-heating coil* using *single-phase liquid* as the tubeside fluid.

3.2.8.9   **Steam Coil**

An *air-heating coil* using steam as the tubeside fluid.

3.2.8.9.1   **Distributing Tube**

A *steam coil* with double tube (steam distributing inner tube-in-steam condensing outer tube) design as shown in Figure 1. The outer tubes can have *fins* on the airside. Each outer tube has a smaller inner tube with perforations that distribute steam evenly through the entire length of the outer tube.
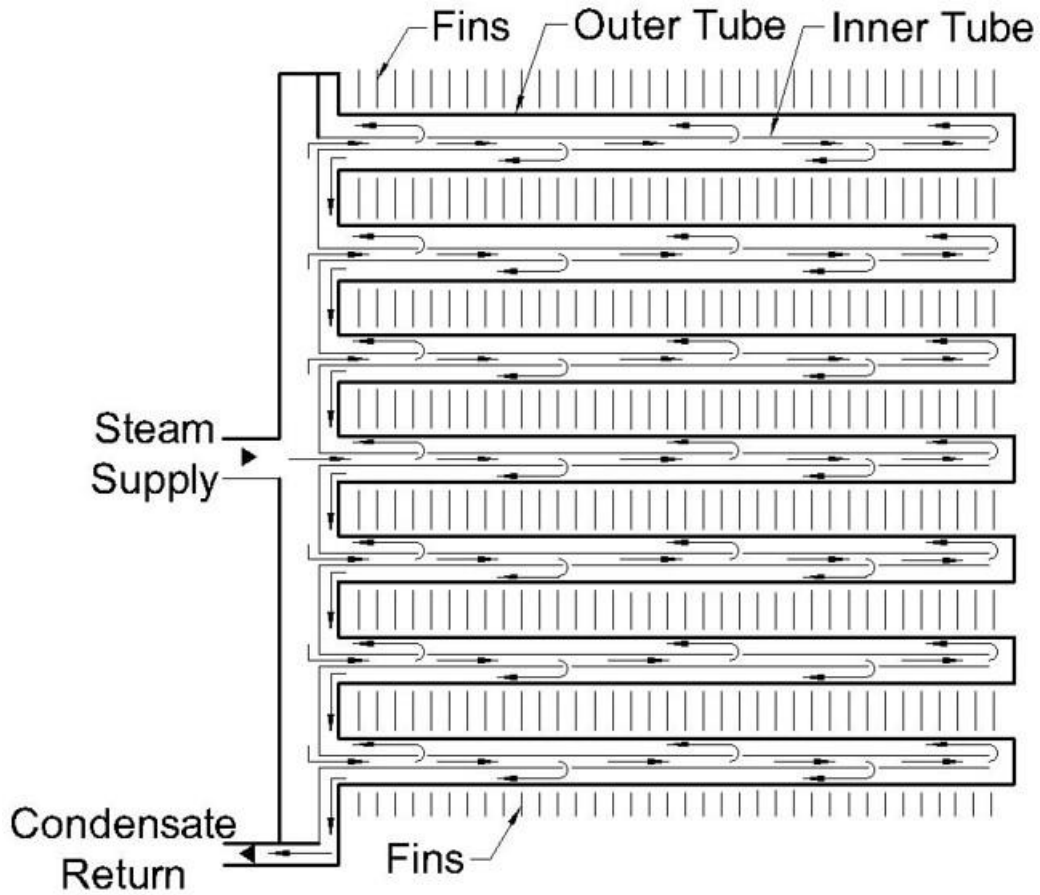
Figure 1 Distributing Tube Steam Coil with Inner Steam Distributing Tubes

3.2.8.9.2   **Single Tube**

Refer to Figure 2. The tubes can have *fins* on the airside. For multiple circuit one-pass coils, the steam supply header and condensate return header are at opposite ends of the coil.
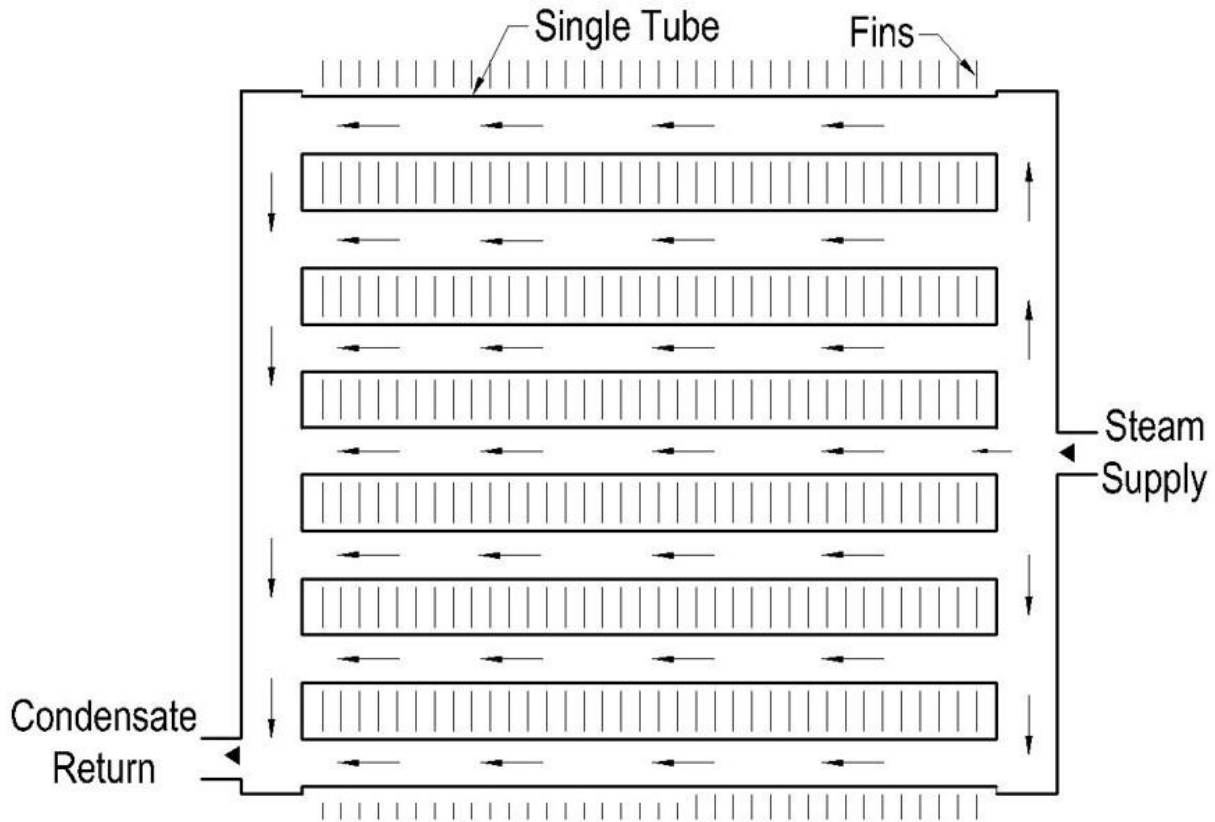
Figure 2 Single Tube Steam Coil

3.2.8.10   Test Coil

A coil that has *laboratory tests* performed on it.

3.2.9   Cooling Capacity

The total capacity associated with the change in air enthalpy that includes both the *latent capacity* and *sensible capacity* expressed in kW (Btu/h).

3.2.9.1   Latent Capacity

Capacity associated with a change in humidity ratio.

3.2.9.2   Sensible Capacity

Capacity associated with a change in dry-bulb temperature.

3.2.10   Dew Point

The vapor saturation temperature of a *refrigerant* at the specified pressure which is the temperature where the last drop of liquid *refrigerant* boils.

3.2.11   Equivalent Annular Fin

Representation of a non-circular fin with a surface area of same heat transfer effectiveness, such as fins between staggered tube arrangement (hexagonal fin surface) or fins between parallel tube arrangement (rectangular fin surface).

3.2.12   Equivalent Length

Total length of straight tubes and return bends in a single circuit in the coil.

3.2.13   Fin

Thin piece of material (can be metal) mechanically or metallurgically bonded to a pipe, tube, or other surface to increase the heat transfer area.

3.2.14   Fin Configuration

3.2.14.1   Spiral Fin

Continuous strip of metal helically wound and attached to a tube.

3.2.14.2   Crimped Spiral Fin

Spiral fin of constant thickness whose *fin* surface adjacent to tube is radially crimped. The crimped geometry results in an annular *fin* of constant cross-sectional area for heat flow that is defined by Equation 129.

3.2.14.3   Smooth Spiral Fin

Spiral fin with a smooth *fin* surface whose *fin* geometry results in an annular *fin* of constant cross-sectional area for heat flow that is defined by Equation 126.

3.2.14.4   Plate Fin on Individually-finned Tube

Square or rectangular *fin* with one hole or one *fin* collar on an individually-finned tube

3.2.14.5   Continuous Plate Fin

Square or rectangular *fin* with holes or *fin* collars to receive multiple tubes arranged in a staggered or parallel tube arrangement.

3.2.14.6   Configured Plate Fin

Plate *fin* with non-flat *fin* surfaces (corrugated, wavy, louvered, lanced).

3.2.14.7   Flat Plate Fin

Plate *fin* with flat *fin* surfaces.

3.2.15   Fin Pitch

Number of transverse *fins* per unit length of tube that can be used for coils rated with I-P units. An example is 12 *fins*/in (fpi).

3.2.16   Fin Spacing

Distance between two successive transverse *fins* on a tube and can be used for coils rated with SI units. An example is 2.0 mm/*fin*.

3.2.17   Headers

A manifold extending from the coil that distributes fluid entering the tubes or collects fluid leaving the tubes or both.

3.2.18   Heating Capacity

The capacity associated with the change in dry-bulb temperature expressed in kW (Btu/h).

3.2.19   Internally Grooved Tubes

Tubes with internally enhanced tube surface used to increase turbulence of tubeside fluids, and a smooth external tube surface.

3.2.20   Laboratory Tests

Tests conducted per ANSI/ASHRAE Standard 33-2016 on representative coils to determine basic heat transfer and pressure drop characteristics that are used in developing *published ratings*.

3.2.21   Maximum Temperature Glide

The difference between the saturated liquid temperature (*bubble point*) and the saturated vapor temperature (*dew point*) for the as-formulated *blend* composition at constant pressure.

3.2.22 **Near Azeotropic**

A *zeotropic blend* with a small *temperature glide* that can be disregarded without consequential error in analysis for a specific application.

3.2.23 **Published Ratings**

A statement of the assigned values of those performance characteristics, under stated *rating conditions*, where a unit can be chosen to fit the application. These values apply to all units of the same nominal size and type (identification) produced by the same manufacturer. This includes the rating of all performance characteristics shown on the unit or published in specifications, advertising or other literature controlled by the manufacturer, at stated *rating conditions*.

3.2.23.1 **Application Ratings**

A rating based on tests performed at *rating conditions* other than *standard rating conditions*.

3.2.23.1 **Rating Conditions**

Any set of operating conditions where a single level of performance results and causes only that level of performance to occur.

3.2.23.2 **Standard Rating**

A rating based on tests performed at *standard rating conditions*.

3.2.23.3 **Standard Rating Conditions**

Rating conditions used as the basis of comparison for performance characteristics. Refer to Table 1 in Section 5.

3.2.24 **Refrigerant**

A two-phase working fluid for the vapor compression cycle that enters the expansion device as a liquid, enters the air-cooling coil as a two-phase fluid, absorbs heat from the airstream by the process of evaporation, and leaves the coil as a superheated gas.

3.2.25 **Single-phase Liquid**

Aqueous glycol or water in the liquid phase.

3.2.26 **Smooth Tubes**

Tubes with smooth internal and external tube surfaces.

3.2.27 **Standard Air**

Air weighing 1.2 kg dry air/m^3 (0.075 lbm dry air/ft^3) that approximates dry air at 21.1°C (70.0°F) and at a standard barometric pressure of 101.325 kPa (29.921 in Hg) abs.

3.2.28 **Standard Coil Orientation**

The standard coil position is that of horizontal tubes and vertical coil face with horizontal airflow.

3.2.29 **Standard Water**

Water weighing 998.9 kg/m^3 (62.361 lbm/ft^3) that approximates water density at 15.6°C (60.0°F) and at a standard barometric pressure of 101.325 kPa (29.921 in Hg) abs.

3.2.30 **Temperature Glide**

The absolute value of the difference between the starting and ending temperatures of a phase change process by a *refrigerant* within a component of a refrigerating system, exclusive of any subcooling or superheating. This term can describe condensation or evaporation of a *zeotropic blend*.

3.2.31 **Test Series**

Multiple individual tests performed on the same *test coil*.

3.2.32 Tube Pass

A *single tube* that runs the length of the coil and crosses the airflow.

3.2.33 Turbulators

Mechanical devices (inserts) inside tubes used to increase turbulence of fluids. Turbulated tube external tube surface is smooth.

3.2.34 Zeotropic

Blends comprising multiple components of different volatilities that, when used in refrigeration cycles, change volumetric composition and saturation temperatures as *blends* evaporate (boil) or condense at constant pressure.

Section 4. Assumptions

4.1 Appendix D Reference

Please refer to Appendix D to locate information about assumptions, classifications, dimensions, fin/coil geometry, and fin efficiency.

Section 5. Test Requirements

5.1 Method for Laboratory Tests of Testing for Rating

Forced-circulation air-cooling and air-heating coils shall be tested in accordance with ANSI/ASHRAE Standard 33-2016 and within the bounds of Table 1, except as outlined in Section 5.1.1 and Section 5.1.2.

5.1.1 Redundant Measurements

Duplicate instrumentation can be utilized to verify the accuracy of the *single-phase liquid* measurements. For *integral face-and-bypass coils* that utilize a *single-phase liquid*, duplicate instrumentation shall be used. For any test performed using duplicate instrumentation, the following shall apply.

1) Instrumentation shall meet the accuracy and location requirements as spelled out in ANSI/ASHRAE Standard 33-2016.

2) Temperature measurements between instruments shall be within ±0.11°C (±0.2°F) for the same *single-phase liquid* temperature measurement (such as an outlet).

3) Mass flow rate measurements between instruments shall be within ±1.0%.

4) If this method is used and the tolerance requirements have been met, the average of the two *single-phase liquid* measurements shall be used for reporting purposes.

5.1.2 Refrigerant Temperature Measuring Instruments

The temperature of *refrigerants* shall be measured by inserting the temperature measuring instrument into a liquid-filled well inserted in the conduit. the temperature measuring instrument shall not be inserted directly into the *refrigerant*.

Refer to Table 1 to see the range of standard rating conditions.

P_b, barometric pressure shall be between 30.139 and 104.979 kPa (8.9 and 31.0 in Hg) abs for all coils.

Maximum P_s, average air absolute static pressure at coil shall be 110.316 kPa (16.0 psia) for all coils.

Table 1 Range of Standard Rating Conditions (SI/IP)

Item		Air-cooling Coils			Air-heating Coils		
		Refrigerant	Cold Water	Cold *Aqueous Glycol*	Steam	Hot Water	Hot *Aqueous Glycol*
Standard Dry Air Face Velocity for Dry Surface Coils (Sensible Only)	Standard m/s	0.51 to 7.62	0.51 to 7.62	0.51 to 7.62	0.51 to 7.62	0.51 to 7.62	0.51 to 7.62
	Standard ft/min	100 to 1500	100 to 1500	100 to 1500	100 to 1500	100 to 1500	100 to 1500
Standard Dry Air Face Velocity for Wet Surface Coils (Air-cooling and Dehumidifying)	Standard m/s	0.51 to 4.06	0.51 to 4.06	0.51 to 4.06	—	—	—
	Standard ft/min	100 to 800	100 to 800	100 to 800	—	—	—
Entering Air Dry-bulb Temperature	°C	18.3 to 48.9	18.3 to 48.9	18.3 to 48.9	-28.9 to 37.8	-17.8 to 37.8	-28.9 to 37.8
	°F	65.0 to 120.0	65.0 to 120.0	65.0 to 120.0	-20.0 to 100.0	0.0 to 100.0	-20.0 to 100.0
Maximum Entering Air Wet-bulb Temperature	°C	29.4	29.4	29.4	—	—	—
	°F	85.0	85.0	85.0	—	—	—
Mean *Single-phase Liquid* Velocity Inside Tubes[1,2]	mean m/s	—	0.15 to 2.44	0.15 to 2.44	—	0.15 to 2.44	0.15 to 2.44
	mean ft/min	—	0.5 to 8.0	0.5 to 8.0	—	0.5 to 8.0	0.5 to 8.0
Entering *Single-phase Liquid* Temperature	°C	—	1.7 to 32.2	-17.8 to 32.2	—	37.8 to 121.1	37.8 to 121.1
	°F	—	35.0 to 90.0	0.0 to 90.0	—	100.0 to 250.0	100.0 to 250.0
Saturated *Refrigerant* Vapor Temperature Leaving Coil Suction Header	°C	-1.1 to 12.8	—	—	—	—	—
	°F	30.0 to 55.0	—	—	—	—	—
Minimum *Refrigerant* Vapor Superheat Leaving Coil Suction Header	°C	3.3	—	—	—	—	—
	°F	6.0	—	—	—	—	—
Steam Pressure at Coil Inlet	kPag	—	—	—	13.79 to 1723.69	—	—
	psig	—	—	—	2.0 to 250.0	—	—

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Item		Air-cooling Coils			Air-heating Coils		
		Refrigerant	Cold Water	Cold *Aqueous Glycol*	Steam	Hot Water	Hot *Aqueous Glycol*
Maximum Steam Superheat at Coil Inlet	°C	—	—	—	27.8	—	—
	°F	—	—	—	50.0	—	—
Aqueous Glycol Concentration by Mass	%	—	—	10 to 60	—	—	10 to 60
Minimum *Fin* Surface Temperature	°C	> 0.0	> 0.0	> 0.0	—	—	—
	°F	> 32.0	> 32.0	> 32.0	—	—	—
Minimum Internal Tube Wall Surface Temperature	°C	—	> 0.0	> *Aqueous Glycol* freezing point	> 0.0	> 0.0	> *Aqueous Glycol* freezing point
	°F	—	> 32.0	> *Aqueous Glycol* freezing point	> 32.0	> 32.0	> *Aqueous Glycol* freezing point
Tubeside Fouling Factor Allowance	m²·K/W	0.00000	0.00000 to 0.00018	0.00000 to 0.00018	0.00000	0.00000 to 0.00018	0.00000 to 0.00018
	h·ft²·°F/Btu	0.00000	0.00000 to 0.00100	0.00000 to 0.00100	0.00000	0.00000 to 0.00100	0.00000 to 0.00100

Notes:
1) Predicted performance and actual performance of water coils operating at mean *single-phase liquid* velocities inside tubes below the minimum listed is expected to show variations exceeding currently accepted tolerances for the following reasons:
    a. Application of coils at low mean *single-phase liquid* velocity inside tubes can lead to fouling and can lead to air entrapment.
    b. Differences in coil design/type affect the variation in low Re_L heat transfer coefficient.
2) On lower limit, Re_L shall exceed 700 at t_{Lm}.

5.2   Test Coils and Test Parameters

Each *coil line* rated in accordance with this Standard shall have separate *laboratory tests* conducted using representative *test coils*.

5.2.1   Required Laboratory Tests

Refer to Table 2 to see the required laboratory tests.

5.2.1.1   Thermal Resistance (*R*)

The overall thermal resistance, *R*, can be determined by test or either of the following procedures:

1)   For coils with *smooth tubes* R_{aD} is determined from *steam coil* tests, assuming a steam-side heat transfer coefficient, fv, of 11356 W/(m^2·K) (2000 Btu/(h·ft^2·°F)). One-row *steam coil* tests shall be used to determine R_{aD} for one-row hot water coils. One-row or two-row *steam coil* tests shall be used to determine R_{aD} for two-row hot water coils. For coils with *turbulators* or *internally grooved tubes*, R_{aD} cannot be determined from *steam coil* tests.

2) R_{aD} is determined from cold water sensible only air-cooling coil tests. One-row cold water sensible only air-cooling coil tests shall be used to determine R_{aD} for one-row hot water coils. One-row or two-row cold water sensible only air-cooling coil tests shall be used to determine R_{aD} for two-row hot water coils. Three or more-row cold water sensible only air-cooling coil tests shall be used to determine R_{aD} for three or more-row hot water coils.

For procedures in Section 5.2.1.1(1) and Section 5.2.1.1(2) isothermal water pressure drop tests shall be conducted per Section 5.6.2.

5.2.1.2   Hot Water Two-row Laboratory Tests

The dry surface air film thermal resistance, R_{aD}, can be determined as in Section 5.2.1.1 except that two-row coils shall be used.

5.2.1.3   All Cooling Test Series Requirement

A complete *test series* shall be run with each *refrigerant* with tests required per Section 5.4.5.

5.2.1.4   All Cooling Tests with Four or More Rows

If a *coil line* has four or more rows, a test shall be performed on a coil with at least four rows. If the *coil line* only has up to three rows, the three-row coil shall be tested.

5.2.1.5   Aqueous Glycol Tests

These coils shall have separate *laboratory tests* using the same coil as used for water dry surface coil tests unless a water coil with *smooth tubes* of identical tube spacing and *fin* configuration has already been tested to determine the air film thermal resistance, R_a, and air pressure drop as functions of standard dry air face velocity and *fin spacing*.

Table 2 Required Laboratory Tests

Coil Type	One-row	Two-row		Three-row or More
		In-Line Tubes or *Flat Plate Fins*	All Other Configurations	
Distributing Tube Steam	Test	Test	No Test[1]	No Test[1]
Single Tube Steam	Test[2]	Test[2]	No Test[1]	No Test[1]
Hot Water	Test	Test	Test	Test
All Cooling	Test[3]	Test	Test[3]	Test
Aqueous Glycol	Test	Test	Test	Test

Notes:

- Where "No Test" is indicated, the manufacturer can perform tests to establish performance factors, in which case the requirements in Section 5.2.1 and notes from Table 2 do not apply.

1. Steam ratings can be calculated using data from one-row tests of the same coil type.

2. If ratings are determined for *distributing tube steam coil* of the same surface geometry, the same ratings can be used for *single tube steam coil* and additional tests are not required.

3. A complete set of tests is not required, provided that the air film heat transfer coefficient, f_a, as determined from a cold water sensible only air-cooling coil *test series*, is not greater than 102.5% of those tested in accordance with Section 5.2.1.4. If this agreement exists for a one-row coil, a test is not required for a two-row coil.

5.2.2   Physical Coil Requirements

The manufacturer shall confirm that the *test coils* used for *laboratory tests* are representative of current production materials and manufacturing methods. The *test coils* shall comply with the following requirements:

5.2.2.1   Coil Face Area

0.186 m² to 1.86 m² (2 ft² to 20 ft²).

5.2.2.2   Rows

In accordance with Table 2.

5.2.2.3   Fin Spacings

The maximum and minimum *fin spacings* or *fin pitch* shall be tested. If the difference between the maximum and minimum *fin spacing* exceeds 3 mm/*fin* or 8 *fins*/in, an intermediate spacing shall be tested.

5.2.2.4   Fin Thickness and Material

The *test coil* shall be constructed with aluminum *fins* at the minimum *fin* thickness if aluminum *fins* are offered. If aluminum *fins* are not offered, test each material type at the minimum *fin* thickness. For changes from *test coil*, see Section 5.7.1 and Section 5.7.2.

5.2.3   Coil Circuiting Guidance

Circuiting guidance for *test coils* is as follows:

5.2.3.1   Steam Coils

Circuiting shall provide representative tube loadings and distribution.

5.2.3.2   Single-phase Liquid Coils

Circuiting shall be designed to match the intended flow arrangement, such as crossflow or thermal cross-counterflow for one-row and two-row coils and thermal counterflow for three or more row coils.

Note: When determining airside ratings, the *test coil* should be designed with the smallest number of circuits that use up as much tubeside pressure drop in the *test coil*) within flow capabilities of the test facility (system pressure drop versus flow curve). This results in a higher *single-phase liquid* tubeside velocity that has a lower tubeside resistance, and minimizes uncertainty of the tubeside resistance on the air film thermal resistance.

Note: When determining tubeside ratings, the *test coil* should be designed with the number of circuits that allows the coil to be tested over the required *single-phase liquid* tubeside fluid velocity range with resulting tubeside fluid pressure drop values that are within flow capabilities of the test facility (system pressure drop versus flow capability).

5.2.3.3   Refrigerant Evaporator Coils

Circuiting for each of the two required circuit lengths shall estimate the intended flow arrangement, such as thermal parallel flow or thermal counterflow for three or more row coils.

5.3   Rating Variables to be Determined by Laboratory Tests for Coils with Smooth Tubes

Information contained in this section is for testing coils with *smooth tubes*. Refer to Section 5.5 for testing requirements of coils with *turbulators* or *internally grooved tubes*.

5.3.1   Rating Variables

Laboratory tests are designed such that heat transfer and pressure drop variables The variables to be determined for coils with *smooth tubes*, as further described in Section 5.4, include the following:

5.3.1.1   Steam Coils

Overall thermal resistance, air pressure drop, and steam pressure drop.

5.3.1.2   Single-phase Liquid Coils

Air film thermal resistance for dry and wet surface conditions, air pressure drop for dry and wet surface conditions, and *single-phase liquid* pressure drop.

5.3.1.3   Refrigerant Evaporator Coils

Refrigerant film thermal resistance and *refrigerant* pressure drop for each *refrigerant* to be rated.

5.3.2   Range of Heat Transfer Variables

The range of heat transfer variables that can have ratings applied shall be limited strictly to the range included in the *laboratory tests* as described under Section 5.4. Values shall not be extrapolated outside the range covered in the *laboratory tests* as described under Section 5.4 except for the following:

1) Initial air-to-tubeside fluid temperature difference, *ITD*, for all coils.
2) Inlet steam pressure, p_{v1}, for *steam coils*.
3) Saturated *refrigerant* vapor temperature leaving coil suction header, t_{r2g}, for *refrigerant evaporator coil*s.
4) *Single-phase liquid* velocity inside tubes, V_L, for *single-phase liquid coils*.
5) *Single-phase liquid* temperatures, t_L, for *single-phase liquid coils*.
6) *Single-phase liquid* concentrations, x_L, for *aqueous glycol* coils.

5.3.3   Range of Heat Transfer Variables

The heat transfer variables for the coil applications covered by this standard shall be evaluated for their effect on thermal performance by conducting *laboratory tests* and are described in Section 5.4.

5.4   **Minimum Requirements for Laboratory Tests for Coils with Smooth Tubes**

Information contained in this section is for testing coils with *smooth tubes*. Refer to Section 5.5 for testing requirements of coils with *turbulators* or *internally grooved tubes*.

5.4.1   **Air Velocity**

All of the following *test series* for specific coil applications, except under Section 5.5.2.3, Section 5.4.5.3, Section 5.5.3, Section 5.5.2.4, and Section 5.6.2 shall be made with at least four different standard dry air face velocities, covering the complete rating range in equally spaced velocity increments on a logarithmic scale.

5.4.2   **Air Pressure Drop**

The coil air pressure drop for all dry and wet surface tests shall be recorded per ANSI/ASHRAE Standard 33-2016.

5.4.3   **Steam Coils**

The *steam coils test series* determines the variation in the overall thermal resistance, R, with the standard dry air face velocity, V_a, and to determine the steam pressure drop through the coil, Δp_v.

5.4.3.1   **Steam Pressure**

For any test, the inlet steam pressure shall be 13.8 to 69.0 kPag (2 to 10 psig) with an inlet steam superheat as specified in ANSI/ASHRAE Standard 33-2016.

5.4.3.2   **Air Temperature**

The entering air dry-bulb temperature shall not exceed 37.8°C (100°F).

Note: The conditions for determining ratings of *steam coils* should be:

t_{1db} =   Entering air dry-bulb temperature = 15.6°C (60°F).

p_{v1} =   Entering steam pressure = 34.5 kPag (5 psig).

5.4.4   **Single-phase Liquid Coils**

5.4.4.1   **Tubeside Fluid**

Water shall be used, if obtainable, as the tubeside fluid for all *single-phase liquid* tests in order to simplify the testing and data reduction procedure. If water is not obtainable an *aqueous glycol* shall be used.

5.4.4.2   **Tubeside Fluid Velocity**

For any test with *single-phase liquid coils* except under Section 5.5.2.3, Section 5.5.2.4, and Section 5.6.2, one *single-phase liquid* velocity inside tubes shall be used in the range from 0.91 to 1.83 m/s (3 to 6 ft/s).

5.4.4.3   **Air-heating Coils**

Hot *single-phase liquid air-heating coils* shall be subjected to a single *test series* for the purpose of determining the variation in the dry surface air film heat transfer coefficient, f_a, and air pressure drop, Δp_{sD}, with standard dry air face velocity, V_a (see Section 5.4.1, Section 5.4.2, and Section 5.4.4.1.). Alternatively, airside performance data from *steam coils* or sensible only air-cooling coils shall be used (see Table 2). If any hot *single-phase liquid air-heating coils* are tested, the temperature limitations in Section 5.6.1 apply and the tests shall be the same as those required in Section 5.5.2.2.

Note: The conditions for determining airside ratings of hot *single-phase liquid air-heating coils* should be:

t_{1db} =   Entering air dry-bulb temperature = 15.6°C (60°F).

t_{L1} =   Entering *single-phase liquid* temperature = 82.2°C (180°F).

Re$_L$	=	*Single-phase liquid* Reynolds Number ≥ 10000 (turbulent flow regime).
t_{L1} - t_{L2}	=	*Single-phase liquid* temperature rise ≥ 2.8°C (≥ 5°F) at minimum air velocity.

5.4.4.4   **Sensible Only Air-cooling Coils**

For coils with *smooth tubes*, the *single-phase liquid* film heat transfer coefficient, f_L, is initially known and shall be calculated using equations in Section 6.2.3.3.1.2. For coils with *smooth tubes*, the *single-phase liquid* Darcy-Weisbach friction factor, f'_{DWL}, is initially known and shall be calculated using equations in Section 6.2.3.3.1.2. Therefore, only a single *test series* is required for the purpose of determining the variation in the dry surface air film heat transfer coefficient, f_a, and air pressure drop, Δp_{sD}, with standard dry air face velocity, V_a (see Section 5.4.1, Section 5.4.2, and Section 5.4.4.1.).

To assure completely dry airside surface, the entering *single-phase liquid* temperature, t_{L1}, for all tests shall be at least 1.7°C (3°F) greater than the entering air *dew point* temperature, t_{1dp}.

Note: The conditions for determining airside ratings of dry surface *single-phase liquid* sensible only air-cooling coils should be:

t_{1db}	=	Entering air dry-bulb temperature = 35°C (95°F).
t_{1wb}	=	Entering air wet-bulb temperature = 17.8°C (64°F).
t_{L1}	=	Entering *single-phase liquid* temperature = 7.2°C (45°F) (1.7°C [3°F]) above entering air *dew point* temperature).
Re$_L$	=	*Single-phase liquid* Reynolds Number ≥ 10000 (turbulent flow regime).
t_{L2} - t_{L1}	=	*Single-phase liquid* temperature rise ≥ 2.8°C (≥ 5°F) at minimum air velocity.

5.4.4.5   **Air-cooling and Dehumidifying Coils**

Two *test series* shall be conducted.

5.4.4.5.1   **Dry Surface Tests**

If the *test coil* has not been previously tested per the requirements of Section 5.4.4.4, then those tests shall comprise the first *test series*. If the exact *test coil* has been tested per Section 5.4.4.4, then additional dry surface tests are not required.

5.4.4.5.2   **Wet Surface Tests**

The second *test series* shall be conducted on the exact coil used for the procedures in Section 5.4.4.5.1. The wet surface *test series* determines the completely wet surface air film heat transfer coefficients and air pressure drop.

5.4.4.5.2.1   **Surface Condition**

To determine air film heat transfer coefficients, f_a, and air pressure drop under completely wet surface conditions, each test shall be conducted with the entire airside surface actively condensing moisture. This operating condition exists when the airside surface temperature at all locations throughout the coil is below the entering air *dew point* temperature ($t_{s1} < t_{1dp}$).

5.4.4.5.2.2   Range of Variables

For each wet surface test, the range in each of the following design variables shall fall within the limits listed below:

t_{L1} =   Entering *single-phase liquid* temperature: 1.7° to 12.8°C (35.0° to 55.0°F).

$t_{1db} - t_{1wb}$ =   Entering wet-bulb depression $\geq$ 3.3°C ($\geq$ 6.0°F).

q_s/q_t =   Air sensible heat ratio ≤ 0.75.

$t_{2wb} - t_{L1} \geq 2.8$°C ($\geq 5.0$°F).

Note: The conditions for determining airside ratings of wet surface *single-phase liquid* air-cooling and dehumidifying coils should be:

t_{1db} = Entering air dry-bulb temperature = 26.7°C (80°F).

t_{1wb} = Entering air wet-bulb temperature = 21.7°C (71°F).

t_{L1} = Entering *single-phase liquid* temperature = 5.6°C (42°F) (13.9°C (25°F) below entering air *dew point* temperature).

Re_L = *Single-phase liquid* Reynolds Number $\geq$ 10000 (turbulent flow regime).

$t_{L2} - t_{L1}$ = *Single-phase liquid* temperature rise ≥ 2.8°C ($\geq$ 5°F) at minimum air velocity.

5.4.5   Refrigerant Evaporator Coils

Complete testing, as outlined below, shall be conducted to establish ratings for each *refrigerant* and tube diameter. Three series of tests shall be run on the exact same prototype coil or coils.

5.4.5.1   Dry Surface Tests

If the *test coil* has not been previously tested per the requirements of Section 5.4.4.4, then those tests shall comprise the first *test series*. If the exact *test coil* has been tested per Section 5.4.4.4, then additional dry surface tests are not required.

5.4.5.2   Wet Surface Tests

If the *test coil* has not been previously tested per the requirements of Section 0, then those tests shall comprise the second *test series*. If the exact *test coil* has been tested per Section 5.4.4.5.2, then additional wet surface tests are not required.

5.4.5.3   Refrigerant Evaporator Coil Tests

The third *refrigerant evaporator coil test series* determines the *refrigerant* evaporating film heat transfer coefficients, f_r, as a function of *refrigerant* loading rate per tube circuit, q_t/N_c, and the *refrigerant* pressure drops through the coil tube circuits, Δp_{rc}, as a function of *refrigerant* mass flow rate per tube circuit, w_r/N_c. Using *refrigerant*, the third *test series* shall be conducted with at least two different lengths of *refrigerant* circuits for a given coil with the following *circuiting* arrangements:

5.4.5.3.1   Circuiting 1

A circuit whose length is not less than 50% of the maximum equivalent length used in the line of rated coils.

5.4.5.3.2   Circuiting 2

A circuit whose length is not greater than 33% of the length of *circuiting* 1 (Section 5.4.5.3.1).

5.4.5.3.3   Minimum Number of Tests Per Circuiting Arrangement

For each of the two *refrigerant* circuit lengths, a minimum of four tests are required. These four tests are to cover the complete range of *refrigerant* loading per circuit, q_t/N_c, in equal increments of capacity on a logarithmic scale, and shall be conducted with a constant liquid temperature in the range of 42.2° to 44.4°C (108°F to 112°F) entering the control device and with a maximum of 12.8°C (55°F) saturated *refrigerant* vapor temperature leaving coil suction header.

5.4.5.3.4   Evaluation

The above eight tests shall serve to evaluate the effects of *refrigerant* loading per circuit on both the coil circuit saturated *refrigerant* pressure drop and the *refrigerant* evaporating film heat transfer coefficient.

5.4.5.3.5   Air Velocity

For any test, the standard dry air face velocity, V_a, shall be between 0.51 to 4.06 standard m/s (100 to 800 standard ft/min).

5.4.5.3.6   Surface Condition

The surface shall be operated either completely dry ($t_{r2g} > t_{1dp} + 1.7°C$ (3°F)) or completely wet ($t_{s1} < t_{1dp}$) to simplify the data reduction procedure.

5.4.5.3.7   Range of Variables

For each wet surface test, the range in each of the following design variables shall fall within the limits listed below:

t_{r2g}	=	Saturated *refrigerant* vapor temperature leaving coil suction Header: 1.7° to 12.8°C (35.0° to 55.0°F).
t_{1db} - t_{1wb}	=	Entering wet-bulb depression $\geq 3.3°C$ ($\geq 6.0°F$).
q_s /q_t	=	Air sensible heat ratio ≤ 0.75.

$t_{2wb} - t_{r1} \geq 2.8°C$ ($\geq 5.0°F$) for counterflow coils.

$t_{2wb} - t_{r2g} \geq 2.8°C$ ($\geq 5.0°F$) for parallel flow coils.

$t_{1db} - t_{r2g} - \Delta t_{r2g} \geq 2.8°C$ ($\geq 5.0°F$).

5.4.5.3.8   Refrigerant Vapor Superheat Leaving Coil Suction Header

Refrigerant vapor superheat leaving coil suction header, for any test, shall be maintained per ANSI/ASHRAE Standard 33-2016 and shall not be less than 3.3°C (6°F).

Note: The conditions for determining tubeside ratings of dry surface *refrigerant* evaporator sensible only air-cooling coils should be:

t_{1db}	=	Entering air dry-bulb temperature = 35°C (95°F).
t_{1wb}	=	Entering air wet-bulb temperature = 17.8°C (64°F).
t_{r0}	=	Subcooled *refrigerant* liquid temperature entering liquid control device = 43.3°C (110°F).
t_{r2g}	=	Saturated *refrigerant* vapor temperature leaving coil suction header = 7.2° to 12.8°C (45° to 55°F) (minimum 1.7°C (3°F) above entering air *dew point* temperature).
Δt_{r2g}	=	*Refrigerant* vapor superheat leaving coil suction header = 5.6°C (10°F).

Note: The conditions for determining tubeside ratings of wet surface *refrigerant* evaporator air-cooling and dehumidifying coils should be:

t_{1db}	=	Entering air dry-bulb temperature = 26.7°C (80°F).
t_{1wb}	=	Entering air wet-bulb temperature = 21.7°C (71°F).
t_{r0}	=	Subcooled *refrigerant* liquid temperature entering liquid control device = 43.3°C (110°F).
t_{r2g}	=	Saturated *refrigerant* vapor temperature leaving coil suction header = 1.7° to 12.8°C (35° to 55°F).
Δt_{r2g}	=	*Refrigerant* vapor superheat leaving coil suction header = 5.6°C (10°F).

5.5   **Minimum Requirements for Laboratory Tests for Coils with Turbulators or Internally Grooved Tubes**

Information contained in this section is for *testing coils* with *turbulators* or *internally grooved tubes*. These tests determine the heat transfer and pressure drop characteristics of the non-smooth tubes since the information shown in Figure 3 is not valid for non-smooth tubes. Except for the use of non-smooth tubes, the *test coils* used in these *test series* shall be identical in design and arrangement to the *test coils* in Section 5.4.3 through Section 5.4.5.

5.5.1   **Steam Coils**

Tests shall be conducted as described in Section 5.4.3. The overall thermal resistance, R, and steam pressure drop through the coil, Δp_v, determined from this *test series* shall be used for rating coils with non-smooth tubes.

5.5.2   **Single-phase Liquid Coils**

Tests shall be conducted as described in Section 5.5.2.1.

5.5.2.1   **Air-heating Coils, Sensible Only Air-cooling Coils, and Air-cooling and Dehumidifying Coils**

In these individual tests, both the tubeside fluid and air film heat transfer coefficients, as a function of the respective fluid mass flow rate, are unknown. Two *test series* are required for this coil design. An *air-heating coil* or sensible only air-cooling coil shall be used for the testing. The temperature limitations in Section 5.6.1 apply for *air-heating coils*.

To assure completely dry airside surface, the entering *single-phase liquid* temperature, t_{L1}, for all tests shall be at least 1.7°C (3°F) greater than the entering air *dew point* temperature, t_{1dp}.

5.5.2.2   **Test Series Number 1**

A single *test series*, as described in Section 5.4.4.4 shall first be conducted on a coil whose design and arrangement are the same in all respects to the non-smooth design except that *smooth tubes* are used. This *test series* determines the variation in the dry surface air film heat transfer coefficient, f_{aD}, with standard dry air face velocity, V_a (see Sections 5.4.1, Section 5.4.2, and Section 5.4.4.1).

5.5.2.3   **Test Series Number 2**

A single *test series* shall be conducted on a coil whose design and arrangement are the same as the rated design including internal tube geometry. This *test series* determines the tubeside *single-phase liquid* heat transfer and pressure drop performance data.

A minimum of eleven tests shall be conducted to define the Colburn-j heat transfer factor, j_L and Darcy-Weisbach friction factor, f'_{DWL}, similar to that shown in Figure 3. Four test points shall be at Re_L below 2100, three test points shall be at Re_L between 2100 and 7000, and four test points shall be at Re_L above 7000. All tests shall be conducted at one standard dry air face velocity between 1.02 to 4.06 standard m/s (200 and 800 standard ft/min).

Note: A high standard dry air face velocity and close *fin spacing* should be used for accuracy reasons.

These tests shall be conducted using a base coil with L_s/D_i ratio between 30 and 90. Ratings for the range of Re_L between 700 and 7000 shall be based upon the L_s/D_i ratio to the -0.33333 power in the laminar flow region.

For each of the test points above, the *single-phase liquid* pressure drop shall be recorded to verify the Darcy-Weisbach friction factor f'_{DWL} determined from the isothermal pressure drop test (see Section 5.5.2.4).

A second *test series* may be conducted on a non-smooth tube coil whose design and arrangement are the same as the first coil tested in *test series* Number 2, but with a different straight tube length per pass, L_s. This *test series* enable the determination of the L_s/D_i exponent in the laminar region ($Re_L < 2100$). A minimum of seven tests shall be conducted to define the Colburn-j heat transfer factor, j_L, and Darcy-Weisbach friction factor, f'_{DWL}, to that shown in Figure 3. Four test points shall be at Re_L below 2100 and three test points shall be at Re_L between 2100 and 7000. The L_s/D_i exponent determined from the two *test coils* in the laminar region ($Re_L < 2100$) may then be used for rating purposes. The last three points ($2100 \leq Re_L < 7000$) are used to define the transition region.

5.5.2.4 Isothermal Pressure Drop for Single-phase Liquid Coils

For each *test coil* tested in Section 5.5.2.3, the *single-phase liquid* pressure drop through the coil shall be determined under isothermal operating conditions. Conduct at least four tests at Re_L below 1000, at least four tests at Re_L between 1000 and 4000, and at least four tests at Re_L above 4000. Testing shall be conducted to cover the complete rating range of Re_L in equally spaced increments on a logarithmic scale for each of the three ranges described above. For each of the test points above, the *single-phase liquid* pressure drop shall be recorded to enable determination of the Darcy-Weisbach friction factor, f'_{DWL}.

5.5.3 Refrigerant Evaporator Coils

Tests requirements are identical to those in Section 5.4.5.1 and Section 5.4.5.2 except that the *test coil* for Section 5.4.5.3 shall use non-smooth tubes, but otherwise have the same design and arrangement as the coil used in Section 5.4.5.1 and Section 5.4.5.2.

5.6 Requirements for Optional Laboratory Tests

If a manufacturer performs optional *laboratory tests*, beyond those required in Table 2, testing shall be conducted in accordance with the provisions of this section.

5.6.1 Single-phase Liquid Air-heating Coils

The difference between the entering *single-phase liquid* temperature, t_{L1}, and the entering air dry-bulb temperature, t_{1db}, shall be at least 26.7°C (80°F), and the *single-phase liquid* and airside conditions shall be within the range of Table 1.

5.6.2 Isothermal Single-phase Liquid Pressure Drop

For each *single-phase liquid coil* tested, the *single-phase liquid* pressure drop through the coil shall be determined under isothermal operating conditions. Testing shall be conducted with at least four different *single-phase liquid* velocities, covering the complete rating range of *single-phase liquid* velocity inside tubes in equally spaced increments on a logarithmic scale. The results of this test shall be used with the Darcy-Weisbach friction factor for the tube in the *test coils* to verify manufacturer established *single-phase liquid* pressure drop of tubeside coil attachments, $\varDelta p_{L\ ATT}$.

5.7   Changes to Established Coil Line Ratings

5.7.1   Changes Requiring Calculated Verification of Ratings (No Additional Laboratory Tests)

After establishing the original *standard ratings*, one or more of the following changes can be made or offered as an option in a *coil line* without changing *published ratings*, provided that the calculated influence of any or all of these changes does not reduce the capacity to less than 97.5% of the corresponding *standard ratings* and does not increase the calculated pressure drop to more than 105.0% of the corresponding *standard ratings*. The next version of *published ratings* shall reflect the calculated influence of these changes.

1)   Copper *fin* thickness can be decreased up to 30% below aluminum *fin* thickness for dry surface coils.

2)   *Fin* thickness increase.

3)   Tube wall thickness.

4)   Tube material, limited to materials listed in Table 5.

If the calculated capacity is less than 97.5% or the calculated pressure drop is greater than 105.0%, new ratings shall be published.

5.7.2   Changes Requiring Additional Laboratory Tests

After establishing the original *standard ratings*, one or more of the following changes can be made or offered as an option in a *coil line* provided a sensible only air-cooling coil or *air-heating coil test series* of four standard dry air face velocities is run and the test capacities are not less than 97.5% of the corresponding *standard ratings*:

1)   *Fin* material other than copper or aluminum

2)   Method of *fin*-to-tube bonding

3)   Tube material other than as provided in Table 5

4)   *Fin* thickness decrease

If the test capacities are less than 97.5% of *standard ratings*, or if other changes such as tube outside diameter, tube spacing, *fin* configuration, tube internal surface, or tube arrangement are made, a complete set of *laboratory tests* shall be run, and *published ratings* shall be changed accordingly. If the air pressure drops are greater than 105% of *standard ratings*, a series of air pressure drop tests shall be run and ratings shall be published accordingly.

Section 6. Rating Requirements

6.1   Ratings

Ratings for *forced-circulation air-cooling and air-heating coils* consist of *standard ratings* used in the selection or application of coils. Ratings can be given for a range of conditions encountered, so that the rating at any selected condition can be selected directly or by interpolation.

1)   Ratings shall include performance characteristics produced under specified conditions or means for calculating specific coil requirements.

2)   Ratings shall be presented in the form of curves, tables, charts, or automated rating/selection computer procedures.

3)   Ratings shall be developed using basic performance characteristics obtained from *laboratory tests* in accordance with this standard.

6.2   Heat Transfer Equations for Laboratory Test Reduction and for Ratings

6.2.1   Initial Test Data Reduction Procedure

The initial test data reduction procedure and calculations for the determination of the parameters listed below are given in ANSI/ASHRAE Standard 33-2016 for *air-heating coils*, sensible only air-cooling coils, and air-cooling and dehumidifying coils.

1) Both airside and tubeside fluid flow rates

2) The average or rated sensible or total heat transfer capacity, or both

3) Entering and leaving airside and tubeside fluid conditions

4) Both airside and tubeside fluid pressure drops

6.2.2   Total Metal Thermal Resistance of Fin and Tube Assembly for All Applications

The total metal thermal resistance, R_m, to heat flow through the external *fin* metal and the prime tube wall metal is calculated by Equation 3.

$$R_m = R_f + R_t \qquad\qquad 3$$

The constant tube metal thermal resistance, R_t, is calculated by Equation 4.

$$R_t = \frac{BD_i}{2k_3 k_t}\left(\ln\frac{D_o}{D_i}\right) \qquad\qquad 4$$

Calculations of the variable *fin* metal thermal resistance, R_f, shall be based on the *fin* efficiencies, ϕ, as developed in *Efficiency of Extended Surface*. This data by Gardner for annular shaped *fins* with both constant *fin* thickness and constant *fin* cross-sectional area designs are shown, respectively, in Figure 29 and Figure 32. *Fin* efficiencies shall be determined mathematically or graphically as detailed in Section 1.

For non-circular shaped *fins*, the *fin* segmentation method, described in *The Resistance to Heat Flow Through Finned Tubing and The Efficiency and Thermal Resistance of Annular Fins*, can be used to compute the variable *fin* metal thermal resistance, R_f. The *fin* efficiency of the individual *fin* segments shall be based on data by Gardner from Figure 29 and Figure 32. *Fin* efficiencies shall be determined mathematically or graphically as detailed in Section 1.

The variable *fin* metal thermal resistance, R_f, based on total external surface effectiveness, η, is calculated by Equation 5 for dry surfaces and by Equation 6 for wet surfaces. For wet surfaces, the term, m''/c_p, is determined from Equation 7. The influence of m'' on R_{aW} as shown in Equation 6 is based on a derivation by Brown in *Theory of Moist Air Heat Exchangers* that was applied to wet coil theory by Ware and Hacha in *Heat Transfer From Humid Air to Fin and Tube Extended Surface Cooling Coils* to determine the implicit effect on R_f.

$$R_{fD} = \left(\frac{1-\eta}{\eta}\right)(R_{aD}) \qquad\qquad 5$$

$$R_{fW} = \left(\frac{1-\eta}{\eta}\right)\left(R_{aW}\frac{c_p}{m''}\right) \qquad\qquad 6$$

$$\frac{m''}{c_p} = \frac{\frac{dh}{dt}\text{ at }t_s}{c_p} = \frac{\text{Slope of the saturated air temperature enthalpy curve at the coil surface temperature},\, t_s}{\text{air-water vapor mixture specific heat at constant pressure},\, c_p} \qquad 7$$

The total external surface effectiveness, η, is calculated by Equation 8.

$$\eta = \frac{\phi A_s + A_p}{A_o} \qquad\qquad 8$$

6.2.3   Air-heating Coils and Sensible Only Air-cooling Coils (Dry Surface Sensible Heat Transfer Equations)

Equations relating the average or rated *sensible capacity*, q_s, to both air and tubeside fluid, by a heat balance, are given in applicable sections in ANSI/ASHRAE Standard 33-2016.

6.2.3.1 **Sensible Capacity**

The identical *sensible capacity*, q_s, corresponding to this heat balance, in terms of the overall heat transfer rate between air and tubeside fluids for a counterflow *circuiting* arrangement (refer to Section 6.5.3 for other *circuiting* arrangements) is calculated by Equation 9.

$$q_s = \frac{A_o \Delta t_m}{R}$$
<div align="right">9</div>

6.2.3.2 **Overall Thermal Resistance**

For clean surfaces, the overall thermal resistance between air and tubeside fluids is calculated by Equations 10 and 11.

$$R = R_{aD} + R_m + R_L (\text{or } R_v \text{ or } R_r)$$
<div align="right">10</div>

$$R_{aD} + R_m = \frac{R_{aD}}{\eta} + R_t$$
<div align="right">11</div>

6.2.3.3 **Tubeside Fluid Film Thermal Resistance**

6.2.3.3.1 **Tubeside Single-phase Liquid Film Thermal Resistance**

The *single-phase liquid* film thermal resistance, R_L, is calculated by Equation 12.

$$R_L = \frac{B}{f_L}$$
<div align="right">12</div>

6.2.3.3.1.1 **Tubeside Single-phase Liquid Fouling Factor Allowance**

For cold and hot *single-phase liquid coil laboratory test* data reduction $R_{ffa} = 0.00000$. For ratings, manufacturer can choose to use tubeside fouling factor allowances within the range of Table 1. See AHRI Guideline E for values.

6.2.3.3.1.2 **Tubeside Single-phase Liquid Film Heat Transfer Coefficient for Coils With Smooth Tubes**

Calculate the tubeside *single-phase liquid* film heat transfer coefficient, f_L using the following equations in the sequence shown.

All *single-phase liquid* properties shall be evaluated at the mean *single-phase liquid* bulk temperature inside tubes, t_{Lm}, except as indicated.

Calculate the mean *single-phase liquid* velocity inside tubes, V_L, at mean *single-phase liquid* density, ρ_L, using Equation 13.

$$V_L = \frac{w_L}{k_{10} \rho_L A_{ix}}$$
<div align="right">13</div>

Calculate the standard *single-phase liquid* velocity inside tubes, $V_{L\ STD}$, at *standard water* density, $\rho_{L\ STD}$, using Equation 14.

$$V_{LSTD} = \frac{w_L}{k_{10} \rho_{L\ STD} A_{ix}}$$
<div align="right">14</div>

Calculate the tubeside *single-phase liquid* mass velocity, G_L, using Equation 15.

$$G_L = k_{10}\rho_L V_L = k_{10}\rho_{L\,STD} V_{L\,STD} = \frac{w_L}{A_{ixt}} \qquad 15$$

Calculate the tubeside *single-phase liquid* Reynolds Number, Re_L using Equation 16.

$$Re_L = \frac{D_i G_L}{k_3 \mu_L} \qquad 16$$

Calculate the tubeside *single-phase liquid* Colburn-*j* heat transfer factor, j_L, using Equation 17 with the Table 3 curve fit coefficients corresponding to the appropriate Reynolds number range. The Equation 17 correlation is shown graphically in Figure 3. The laminar region ($Re_L < 2100$) is based upon the Seider-Tate correlation for laminar flow. The transition region ($2100 \leq Re_L < 10000$) is mathematically blended to connect laminar flow to turbulent flow. The turbulent region ($10000 \leq Re_L$) is based upon the Dittus-Boelter correlation for turbulent flow. The Equation 17 correlation and Table 3 curve fit coefficients are only for *single-phase liquid* flow inside *smooth tubes*.

$$\ln(j_L) = J1 + J2\ln(Re_L) + J3\,(\ln(Re_L))^2 + [J4 + J5\ln(Re_L) + J6\,(\ln(Re_L))^2] \cdot \ln(L_s/D_i) \qquad 17$$

**Table 3 Colburn-j Heat Transfer Factor Curve Fit Coefficients Inside
Smooth Tubes for Single-phase Liquid Coils**

	J1	J2	J3	J4	J5	J6
$Re_L < 2100$	0.620576	-0.66666	0.0	-0.33333	0.0	0.0
$2100 \leq Re_L < 10000$	23.9147	-6.2352	0.329875	-11.359	2.46100	-0.13330
$10000 \leq Re_L$	-5.2036	0.073562	-0.01184	0.0	0.0	0.0

Calculate the tubeside *single-phase liquid* Prandtl number, Pr_L, using Equation 18.

$$Pr_L = \frac{c_{pL}\mu_L}{k_L} \qquad 18$$

The tubeside Colburn-*j* heat transfer factor is defined by Equation 19.

$$j_L = St_L \, Pr_L^{2/3} \left(\frac{\mu_{Ltw}}{\mu_L}\right)^{0.14} \qquad 19$$

Rearranging the Equation 19 tubeside Colburn-*j* heat transfer factor, calculate the tubeside *single-phase liquid* Stanton number using Equation 20.

$$St_L = \frac{j_L}{Pr_L^{2/3} \left(\frac{\mu_{Ltw}}{\mu_L}\right)^{0.14}} \qquad 20$$

The tubeside *single-phase liquid* absolute viscosity, μ_{Ltw}, at tubeside mean tube wall surface temperature, t_{Ltw}, is determined by an iterative solution. The mean tube wall surface temperature, for *single-phase liquid* sensible only air-cooling coils is defined by Equation 21. The mean tube wall surface temperature, for *single-phase liquid air-heating coils* is defined by Equation 22.

$$t_{Ltw} = t_{Lm} + \frac{R_L}{R}\Delta t_m \text{ for } \textit{single-phase liquid} \text{ sensible only air-cooling coils} \qquad 21$$

$$t_{Ltw} = t_{Lm} - \frac{R_L}{R}\Delta t_m \text{ for } \textit{single-phase liquid air-heating coils} \qquad 22$$

The tubeside *single-phase liquid* Stanton number is defined by Equation 23.

$$St_L = \frac{f_L}{c_{pL}G_L} \qquad 23$$

Rearranging the Equation 23 tubeside *single-phase liquid* Stanton number, calculate the tubeside *single-phase liquid* film heat transfer coefficient, f_L using Equation 24.

$$f_L = St_L c_{pL} G_L \qquad 24$$

j$_L$ - Colburn-j Heat Transfer Factor Inside Smooth Tubes for Single-phase Liquid Coils
f$_{DWL}$ - Darcy-Weisbach Friction Factor Inside Smooth Tubes for Single-phase Liquid Coils

Re$_L$ - Reynolds Number
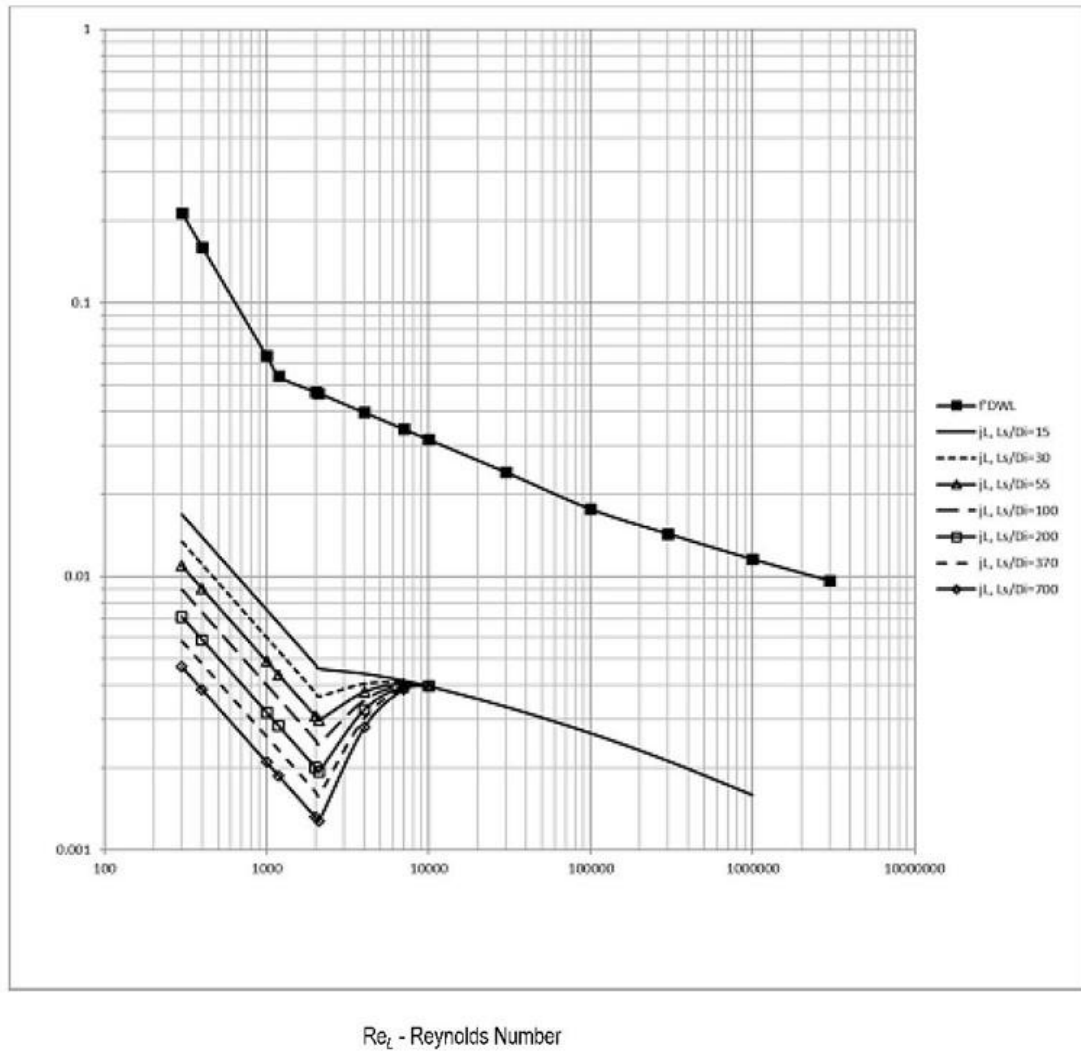
Figure 3. Colburn-*j* Heat Transfer Factor and Darcy-Weisbach
Friction Factor Inside Smooth Tubes for Single-phase Liquid Coils

6.2.3.3.1.3   Tubeside Single-phase Liquid Film Heat Transfer Coefficient for Coils with Turbulators or Internally Grooved Tubes

The tubeside *single-phase liquid* film heat transfer coefficient, f_L is determined by test as described in Section 5.5.

Note   The tubeside *single-phase liquid* Colburn-j heat transfer factors from tests of non-smooth tubes should have curve fit constants determined using the Equation 17 correlation. This enables the user to determine the tubeside *single-phase liquid* film heat transfer coefficient, f_L, for non-smooth tube coils using the methodology detailed in Section 6.2.3.3.1.2, but with curve fit coefficients determined from Section 5.5 tests for non-smooth tubes.

25

6.2.3.3.2   Tubeside Steam Condensing Film Thermal Resistance

For *steam coils* used to determine R_{aD} for hot *single-phase liquid coils* per Table 2, Section 5.2.1.1(1), the tubeside steam condensing film heat transfer coefficient, f_v, is 11,356 W/(m²·°C) (2000 Btu/(h-ft²-°F)). Calculate the tubeside steam condensing film thermal resistance, R_v, using Equation 25.

$$R_v = \frac{B}{f_v}$$ 25

6.2.3.3.3   Tubeside Refrigerant Evaporating Film Thermal Resistance

For *refrigerant evaporator coils*, the tubeside *refrigerant* evaporating film heat transfer coefficient, f_r, is determined by test as described in Sections 5.4.5 and 5.5.3. Calculate the tubeside *refrigerant* evaporating film thermal resistance using Equation 26.

$$R_r = \frac{B}{f_r}$$ 26

6.2.3.4   Logarithmic Mean Temperature Difference

For counterflow cold *single-phase liquid* sensible only air-cooling coils, calculate the logarithmic mean temperature difference, Δt_{lm} = LMTD, using Equation 27. An illustrative thermal diagram for these coils is shown in Figure 4. This figure illustrates the condition where the coil surface is operating completely dry ($t_{s2} > t_{1dp}$) and the temperature conditions of fluids and *fin* surface.

$$\Delta t_m = \frac{(t_{1db} - t_{L2}) - (t_{2db} - t_{L1})}{ln\left(\frac{t_{1db} - t_{L2}}{t_{2db} - t_{L1}}\right)}$$ 27
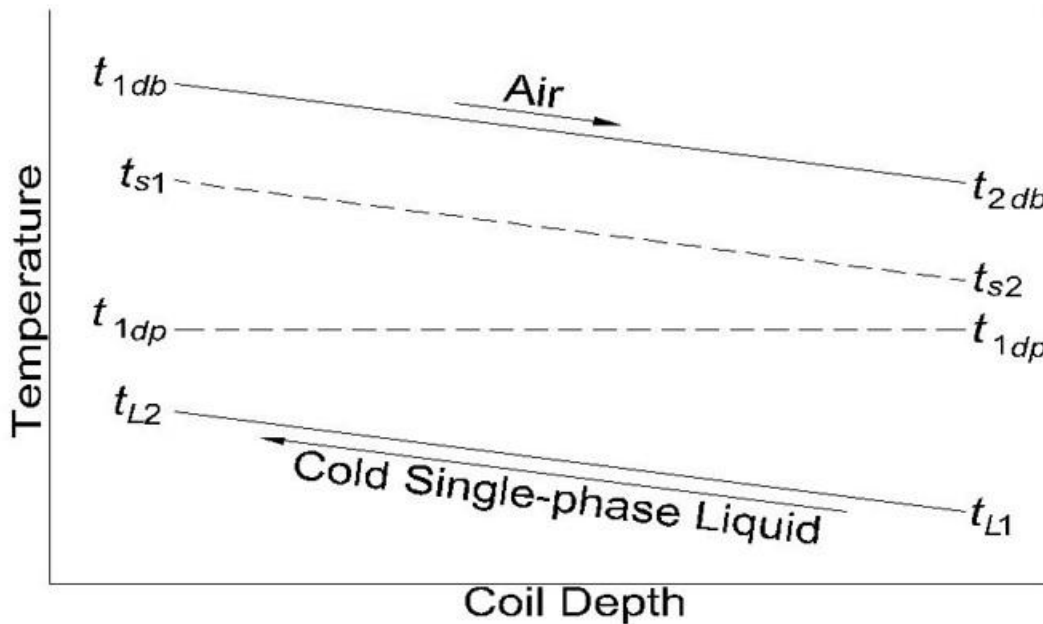
**Figure 4 Thermal Diagram of Counterflow Cold Single-phase Liquid
Sensible Only Air-cooling Coils**

For counterflow hot *single-phase liquid air-heating coils*, calculate the logarithmic mean temperature difference, $\Delta t_{lm} = $ LMTD, using Equation 28. An illustrative thermal diagram for these coils is shown in Figure 5. This figure illustrates the temperature conditions of fluids.

$$\Delta t_m = \frac{(t_{L2} - t_{1db}) - (t_{L1} - t_{2db})}{\ln\left(\frac{t_{L2} - t_{1db}}{t_{L1} - t_{2db}}\right)} \qquad\qquad 28$$

Figure 5 Thermal Diagram of Counterflow Hot Single-phase Liquid Air-heating Coils

For thermal counterflow (physical parallelflow) *refrigerant* evaporator sensible only air-cooling coils, calculate the logarithmic mean temperature difference, $\Delta t_{lm} = LMTD$, using Equation 29. An illustrative thermal diagram for these coils is shown in Figure 6. This figure illustrates the condition where the coil surface is operating completely dry ($t_{s2} > t_{1dp}$), and the tempe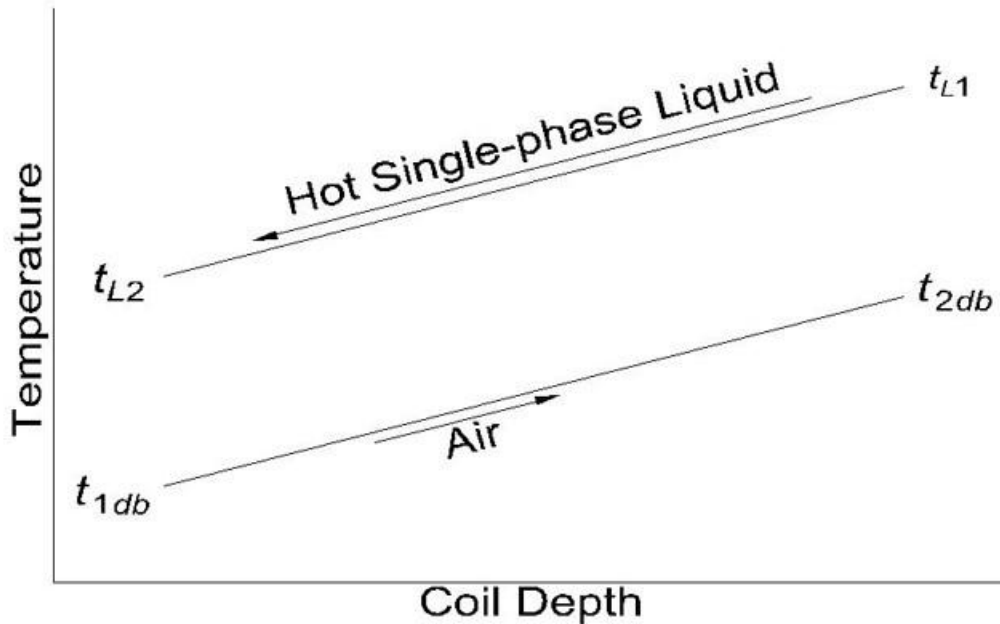rature conditions of fluids and *fin* surface. For *zeotropic refrigerants* with large *maximum temperature glides*, for example R-407C, this figure only applies when the *refrigerant* temperature decreases through the tube circuits.

$$\Delta t_m = \frac{(t_{1db} - t_{r1}) - (t_{2db} - t_{rc2g})}{\ln\left(\frac{t_{1db} - t_{r1}}{t_{2db} - t_{rc2g}}\right)} \qquad\qquad 29$$

27
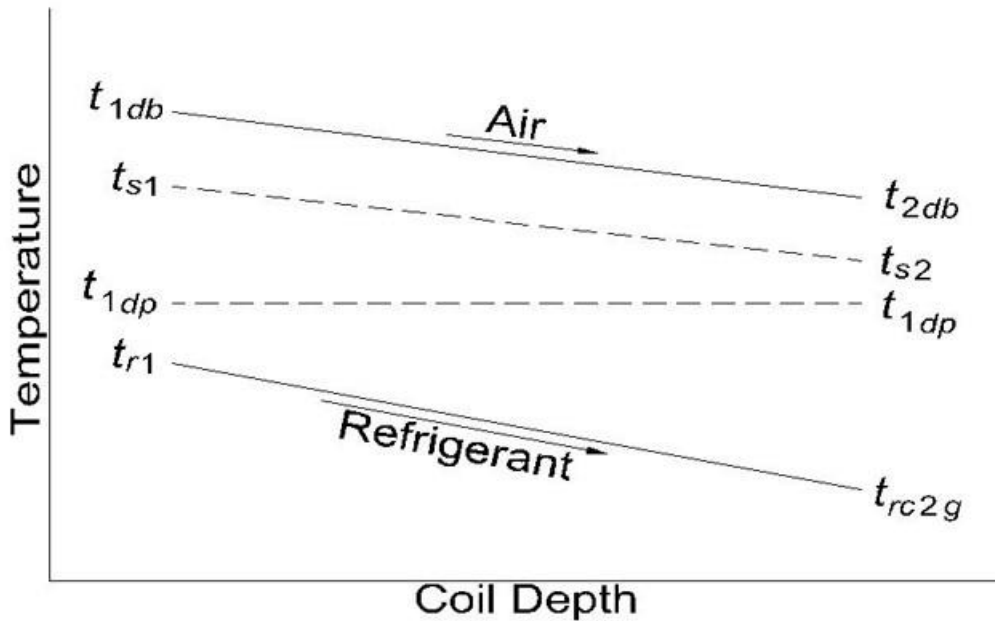
**Figure 6 Thermal Diagram of Counterflow (Physical Parallelflow)
Refrigerant Evaporator Sensible Only Air-cooling Coils**

For thermal parallelflow (physical counterflow) *refrigerant* evaporator sensible only air-cooling coils, calculate the logarithmic mean temperature difference, $\Delta t_{lm} = LMTD$, using Equation 30. An illustrative thermal diagram for these coils is shown in Figure 7. This figure illustrates the condition where the coil surface is operating completely dry ($t_{s2} > t_{1dp}$), and the temperature conditions of fluids and *fin* surface. For *zeotropic refrigerants* with large *maximum temperature glides*, for example R-407C, this figure only applies when the *refrigerant* temperature decreases through the tube circuits.

$$\Delta t_m = \frac{(t_{1db} - t_{rc2g}) - (t_{2db} - t_{r1})}{\ln\left(\frac{t_{1db} - t_{rc2g}}{t_{2db} - t_{r1}}\right)}$$

<div align="right">30</div>
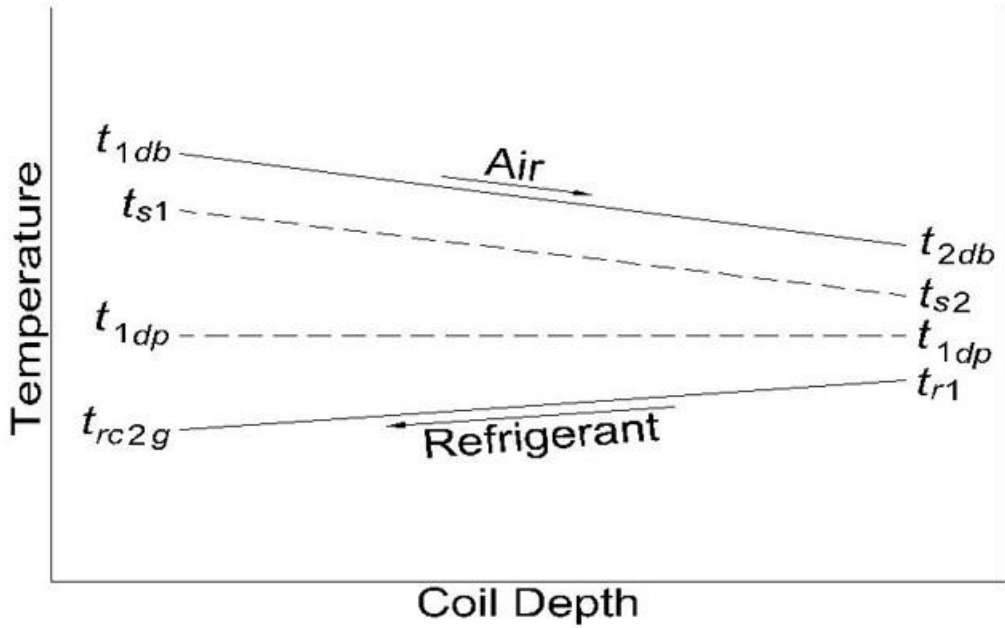
**Figure 7 Thermal Diagram of Parallelflow (Physical Counterflow)
Refrigerant Evaporator Sensible Only Air-cooling Coils**

For *single tube steam coils*, calculate the logarithmic mean temperature difference, $\Delta t_{lm} = LMTD$, using Equation 31. An illustrative thermal diagram for these coils is shown in Figure 8. This figure illustrates the temperature conditions of fluids.

$$\Delta t_m = \frac{(t_{2db} - t_{1db})}{\ln\left(\frac{t_{vmg} - t_{1db}}{t_{vmg} - t_{2db}}\right)}$$

31
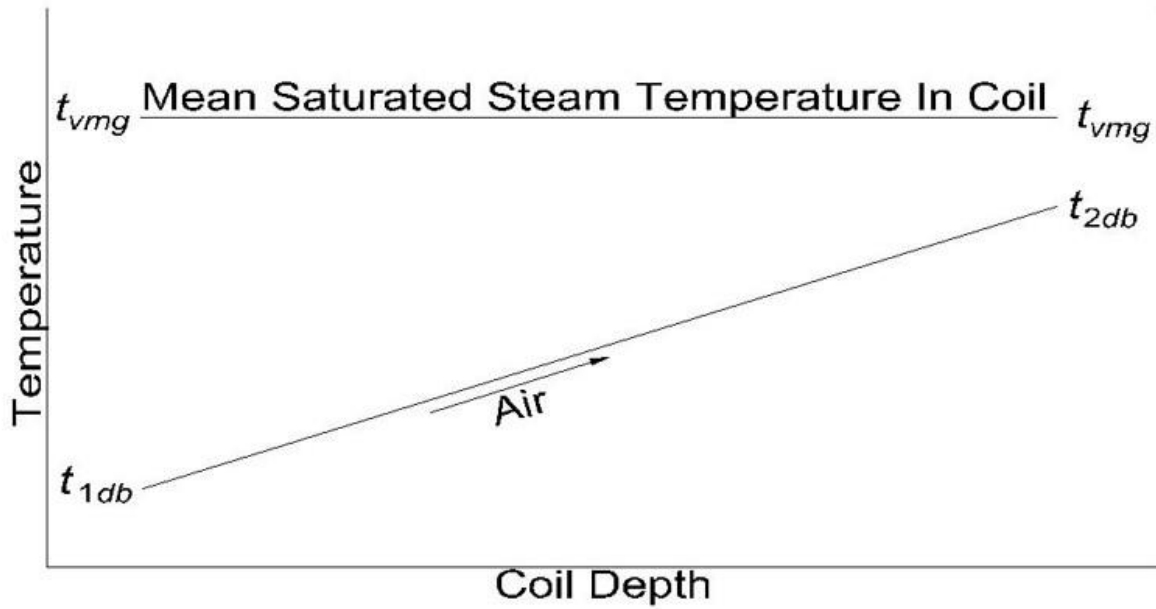
Figure 8 Thermal Diagram of Single Tube Steam Coils

For *distributing tube steam coils*, calculate the logarithmic mean temperature difference, $\Delta t_{lm} = LMTD$, using Equation 32. An illustrative thermal diagram for these coils is shown in Figure 9. This figure illustrates the temperature conditions of fluids.

$$\Delta t_m = \frac{(t_{2db} - t_{1db})}{\ln\left(\frac{t_{v2g} - t_{1db}}{t_{v2g} - t_{2db}}\right)} \qquad\qquad 32$$

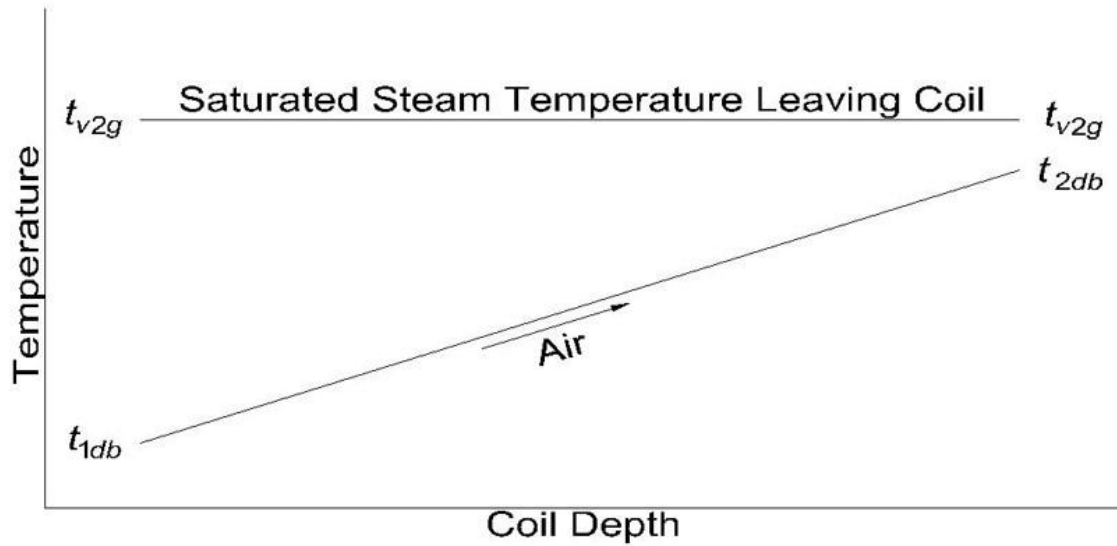**Figure 9 Thermal Diagram of Distributing Tube Steam Coils**

For other tube *circuiting* arrangements see Section 6.5.

6.2.3.5   **Determination of Air Pressure Drop Through Coil**

Calculate the air pressure drop for *air-heating coils* and sensible only air-cooling coils as follows.

The air pressure drop through coil at standard dry air density is calculated using Equation 33.

$$\Delta p_{a\,STD} = (DPSD)N_r \tag{33}$$

For example, if a coil has a DPSD of 0.125 in H_2O/row, the $\Delta p_{a\,STD}$ for a six-row coil can be calculated using Equation 33 as follows:

$$\Delta p_{a\,STD} = (DPSD)N_r = (0.125)6 = 0.75 \text{ in } H_2O$$

The air pressure drop through coil at job (actual) conditions is calculated using Equation 34 and Equation 35 or Equation 36 where the average air absolute pressure at coil, P_s, is in Hg abs. See Appendix C for additional information on converting between actual and standard conditions. To simplify calculations, the average air absolute pressure at coil, P_s, can be defined as the air barometric pressure at *job conditions*, $P_{b\,JOB}$.

$$\Delta p_{a\,JOB} = F_{aD}\Delta p_{a\,STD} \tag{34}$$

For *single-phase liquid air-heating coils* and sensible only air-cooling coils, calculate F_{aD} using Equation 35.

$$F_{aD} = \rho_{a\,STD}\left(\frac{k_{12}\big(k_5 + 0.5(t_{1db} + t_{2db})\big)}{P_s k_1 k_{11}}\right)\frac{\left(1 + \dfrac{W_1}{k_9}\right)}{(1 + W_1)} \tag{35}$$

For *steam coils*, calculate F_{aD} using Equation 36.

$$F_{aD} = \rho_{a\,STD}\left(\frac{k_{12}(k_5 + t_{2db})}{P_s k_1 k_{11}}\right)\frac{\left(1 + \dfrac{W_1}{k_9}\right)}{(1 + W_1)} \tag{36}$$

6.2.3.6   **Tubeside Fluid Pressure Drop Through Coil**

6.2.3.6.1   **Single-phase Liquid Pressure Drop Through Coil**

Calculate the *single-phase liquid* pressure drop through coil, Δp_L, using Equation 37.

$$\Delta p_L = \Delta p_{Lt} + \Delta p_{L\,ATT} \tag{37}$$

Calculate the *single-phase liquid* head loss through coil at average fluid density, h_L, using Equation 38.

$$h_L = \frac{k_1 \Delta p_L}{\rho_{Lm}} \tag{38}$$

6.2.3.6.1.1   **Single-phase Liquid Pressure Drop of Tubeside Coil Attachments**

The manufacturer shall provide manufacturer established *single-phase liquid* pressure drop of tubeside coil attachments, $\Delta p_{L\,ATT}$.

6.2.3.6.1.2 Single-phase Liquid Pressure Drop Through Smooth Straight Tubes

Calculate the *single-phase liquid* pressure drop through smooth straight tubes, Δp_{Lt}, using the following equations in the sequence shown. Alternatively, a manufacturer's test data can be used to establish *single-phase liquid* Darcy-Weisbach friction factor expression for flow inside smooth straight tubes.

All *single-phase liquid* properties shall be evaluated at the mean *single-phase liquid* bulk temperature inside tubes, t_{Lm}, except as indicated.

Calculate the mean *single-phase liquid* velocity inside tubes, V_L, at mean *single-phase liquid* density, ρ_L, using Equation 13.

Calculate the standard *single-phase liquid* velocity inside tubes, $V_{L\ STD}$, at *standard water* density, $\rho_{L\ STD}$, using Equation 14.

Calculate the *single-phase liquid* mass velocity, G_L, using Equation 15.

Calculate the *single-phase liquid* Reynolds number, Re_L using Equation 16.

Calculate the *single-phase liquid* Darcy-Weisbach friction factor, f'_{DWL}, using Equation 39, Equation 40, or Equation 41 corresponding to the Reynolds number calculated from Equation 16. Equation 39, Equation 40, and Equation 41 are shown graphically in Figure 3. Equation 39 is the Poiseuille correlation for laminar flow with $Re_L \leq 1187$. Equation 40 is the Blasius correlation for transition and turbulent flow with $1187 \leq Re_L < 100000$. The lower Re limit in Equation 40 is extended down and intersects the laminar flow equation, making it conservative over the transition flow region ($2100 \leq Re_L < 4000$). Equation 41 is for turbulent flow with $100000 \leq Re_L < 3000000$. Equation 39, Equation 40, and Equation 41 are only valid for *single-phase liquid* flow inside *smooth tubes*.

$$f'_{DWL} = 64/Re_L \text{ for } Re_L \leq 1187 \qquad\qquad 39$$

$$f'_{DWL} = 0.3164/Re_L^{0.25} \text{ for } 1187 < Re_L < 100000 \qquad\qquad 40$$

$$f'_{DWL} = 0.0032 + 0.221/Re_L^{0.237} \text{ for } 100000 \leq Re_L \leq 3000000 \qquad 41$$

Alternatively, calculate the *single-phase liquid* Darcy-Weisbach friction factor, f'DWL, using Equation 42 with the Table 4 curve fit coefficients corresponding to the appropriate Reynolds number range. The Equation 42 correlation and Table 4 curve fit coefficients are the same as Equation 39, Equation 40, and Equation 41, but utilize one curve fit equation for programming simplicity.

$$\ln(f'_{DWL} + DWL4) = DWL1 + DWL2\ln(Re_L) + DWL3(\ln(Re_L))^2 \qquad 42$$

Table 4 Darcy-Weisbach Friction Factor Curve Fit Coefficients
Inside Smooth Tubes for Single-phase Liquid Coils

	DWL1	DWL2	DWL3	DWL4
ReL ≤ 1187	4.1589	-1.0	0.0	0.0
1187 < ReL < 100000	-1.1507	-0.25	0.0	0.0
100000 ≤ ReL ≤ 3000000	-1.5096	-0.237	0.0	0.0032

The Darcy-Weisbach equation for flow inside tubes is defined by Equation 43.

$$\Delta p = \rho f \frac{L}{D} \frac{V^2}{2g}$$ 43

Calculate the *single-phase liquid* pressure drop through smooth straight tubes Δp_{Lt}, using Equation 44 that was derived by substituting AHRI Standard 410 nomenclature into Equation 43.

$$\Delta p_{Lt} = \rho_{Lm} f'_{DWL} \frac{N_p L_s}{D_{it}} \frac{V_t^2}{2g}$$ 44

6.2.3.6.1.3 **Single-phase Liquid Pressure Drop Through Straight Tubes for Coils with Turbulators or Internally Grooved Tubes**

The *single-phase liquid* Darcy-Weisbach friction factor, f'_{DWL}, is determined by test as described in Section 5.5.

Note: The tubeside *single-phase liquid* Darcy-Weisbach friction factors from tests of non-smooth tubes should have curve fit constants determined using the Equation 42 correlation. This enables the user to determine the tubeside *single-phase liquid* Darcy-Weisbach friction factor, f'_{DWL}, for non-smooth tube coils using the methodology detailed in Section 6.2.3.6.1.2, but with curve fit coefficients determined from Section 5.5 tests for non-smooth tubes.

6.2.3.6.2 **Steam Pressure Drop Through Tube Circuits**

Calculate the steam pressure drop through coil, Δp_v, using Equation 45.

$$\Delta p_v = \Delta p_{vLe} + \Delta p_{v\ ATT}$$ 45

The manufacturer shall provide manufacturer established steam pressure drop of tubeside coil attachments, $\Delta p_{v\ ATT}$.

6.2.3.6.3 **Refrigerant Pressure Drop Through Tube Circuits**

Calculate the *refrigerant* pressure drop through coil, Δp_r, using Equation 46.

$$\Delta p_r = \Delta p_{rLe} + \Delta p_{r\ ATT}$$ 46

The manufacturer shall provide manufacturer established *refrigerant* pressure drop of tubeside coil attachments, $\Delta p_{r\ ATT}$.

6.2.4   Air-cooling and Dehumidifying Coils (Wet Surface Total Heat Transfer Equations)

The dual potential method used in this standard to calculate wet surface coil performance is, with modifications, per the basic theory as presented by McElgin and Wiley in *Calculation of Coil Surface Areas for Air Cooling and Dehumidification* and outlined in BCMI Code TS-4044 *Proposed Commercial Standard For Rating and Testing Air Cooling Coils Using Non-volatile Refrigerants*. Anderson presents a brief history of air-cooling coil rating standard development in air-cooling and dehumidifying coil performance based on ARI Industrial Standard 410-64. Anderson and ASHRAE *Handbook – HVAC Systems and Equipment*, Chapter 23 *Air-cooling and Dehumidifying Coils* define in detail the dual-potential method used in this standard for determining wet surface coil performance.

Note: Other investigators (Ware and Hacha in *Heat Transfer From Humid Air to Fin and Tube Extended Surface Cooling Coils*, Goodman in *Performance of Coils For Dehumidifying Air*, and Wile in *Air Cooling Coil Performance*) have developed rating methods by converting the basic dual potential method to an equal single potential method. These single potential methods are not as accurate as the AHRI Standard 410 dual potential method but are included as informative references on the topic.

6.2.4.1   Sensible Heat Ratio

The ratio of airside sensible-to-total capacity is calculated using Equation 47.

$$\frac{q_s}{q_t} = \frac{c_p(t_{1db} - t_{2db})}{h_1 - h_2} \qquad\qquad 47$$

The ratio, q_s/q_t, is used as an index to define the procedure required for calculating ratings as follows.

1)   If $q_s/q_t < 0.95$, the coil is wet and the equations listed in the remainder of Section 6.2.4 shall be used.

2)   If $q_s/q_t \geq 0.95$, the coil is dry, and the dry surface sensible heat transfer equations listed under Section 6.2.3. shall be used.

6.2.4.2   Total Heat Capacity or Total External Surface Area Requirements

Depending upon operating conditions, the coil airside surface can operate either completely wet or a portion of the coil can operate with dry surface. For the case where all surface is completely wet, all surface temperatures, t_s, of the coil are below the entering air dew-point temperature, t_{1dp}. For the case where the surface temperatures, t_s, of a part of the coil are above the entering air dew-point temperature, t_{1dp}, this portion of the coil external surface area, A_D, operates dry with the remainder of the coil external surface area, A_W, wet or actively condensing moisture. For this latter case, the coil external surface area requirements are separately calculated for the dry and wet parts of the coil.

For the wet part of the coil, the wet external surface area, A_W, or the corresponding total heat capacity, q_{tW}, is calculated by using the mean air enthalpy difference between the air stream and that corresponding to the coil surface temperature.

Equations relating the average or rated total capacity, q_t, to both air and tubeside fluid, by a heat balance, are given in applicable sections of ANSI/ASHRAE Standard 33-2016. The identical total capacity, q_t, corresponding to this heat balance, for the case where a portion of the coil external surface area operates dry is calculated using Equation 48, Equation 49, and Equation 50.

$$q_t = q_{tD} + q_{tW} \qquad\qquad 48$$

$$q_{tD} = \frac{A_D \Delta t_m}{R} \qquad\qquad 49$$

$$q_{tW} = \frac{A_W \Delta h_m}{c_p R_{aW}} \qquad\qquad 50$$

The q_{tD} term in Equation 48 is omitted if the entire airside surface is actively condensing moisture.

For counterflow cold *single-phase liquid* air-cooling and dehumidifying coils, calculate the logarithmic mean temperature difference, $\Delta t_{lm} = LMTD$, using Equation 51 and the logarithmic mean enthalpy difference, $\Delta h_{lm} = LMED$, using Equation 52. An illustrative thermal diagram for these coils is shown in Figure 10. This figure illustrates the condition where a part of the coil surface is operating dry, the air enthalpies, the temperature conditions of fluids and *fin* surface, and the dry-wet boundary conditions.

$$\Delta t_m = \frac{(t_{1db} - t_{L2}) - (t_{Bdb} - t_{LB})}{\ln\left(\frac{t_{1db} - t_{L2}}{t_{Bdb} - t_{LB}}\right)} \qquad\qquad 51$$

$$\Delta h_m = \frac{(h_B - h_{sB}) - (h_2 - h_{s2})}{\ln\left(\frac{h_B - h_{sB}}{h_2 - h_{s2}}\right)} \qquad\qquad 52$$
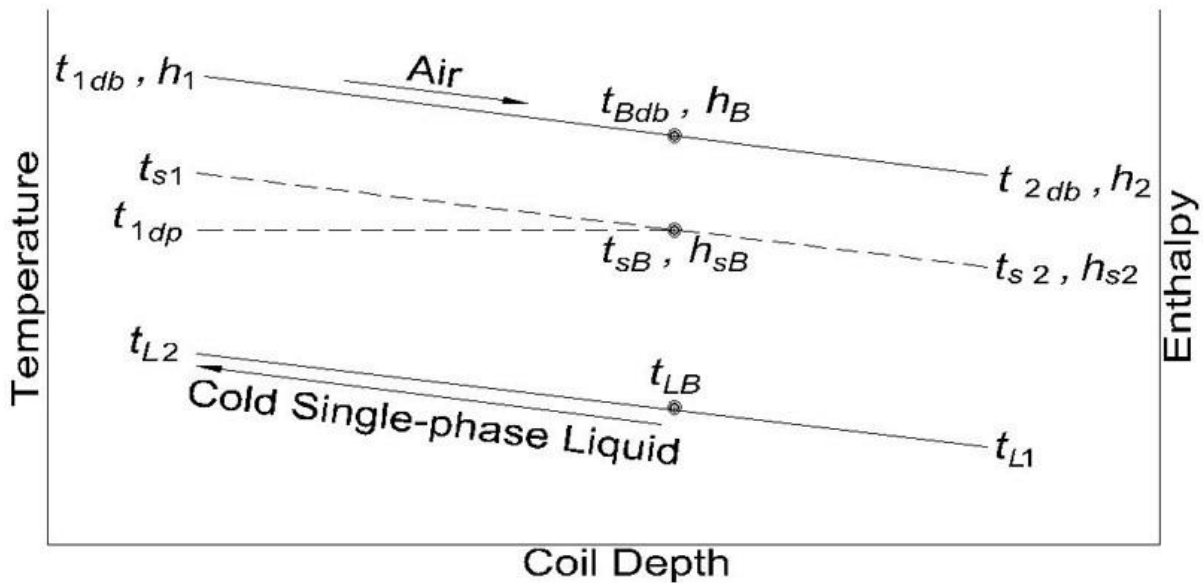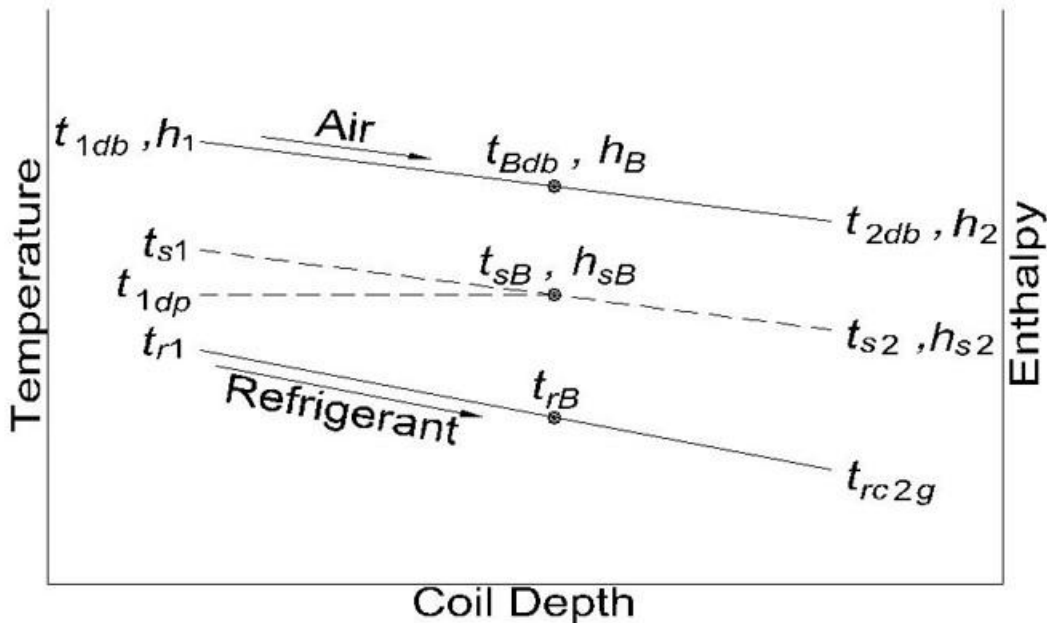

Figure 10 Thermal Diagram of Counterflow Cold Single-phase Liquid Air-cooling and Dehumidifying Coils

For thermal counterflow (physical parallelflow) *refrigerant* evaporator air-cooling and dehumidifying coils, calculate the logarithmic mean temperature difference, $\Delta t_{lm} = LMTD$, using Equation 53 and the logarithmic mean enthalpy difference, $\Delta hlm = LMED$, using Equation 54. An illustrative thermal diagram for these coils is shown in Figure 11. This figure illustrates the condition where a part of the coil surface is operating dry, the air enthalpies, the temperature conditions of fluids and *fin* surface, and the dry-wet boundary conditions. For *zeotropic refrigerants* with large *maximum temperature glides*, for example R-407C, this figure only applies when the *refrigerant* temperature decreases through the tube circuits.

$$\Delta t_m = \frac{(t_{1db} - t_{r1}) - (t_{Bdb} - t_{rB})}{\ln\left(\frac{t_{1db} - t_{r1}}{t_{Bdb} - t_{rB}}\right)} \tag{53}$$

$$\Delta h_m = \frac{(h_B - h_{sB}) - (h_2 - h_{s2})}{\ln\left(\frac{h_B - h_{sB}}{h_2 - h_{s2}}\right)} \tag{54}$$

Figure 11 Thermal Diagram of Counterflow (Physical Parallelflow) Refrigerant Evaporator Air-cooling and Dehumidifying Coils

For thermal parallelflow (physical counterflow) *refrigerant* evaporator air-cooling and dehumidifying coils, calculate the logarithmic mean temperature difference, $\Delta t_{lm} = LMTD$, using Equation 55 and the logarithmic mean enthalpy difference, $\Delta h_{lm} = LMED$, using Equation 56. An illustrative thermal diagram for these coils is shown in Figure 12. This figure illustrates the condition where a part of the coil surface is operating dry, the air enthalpies, the temperature conditions of fluids and *fin* surface, and the dry-wet boundary conditions. For *zeotropic refrigerants* with large *maximum temperature glides*, for example R-407C, this figure only applies when the *refrigerant* temperature decreases through the tube circuits.

$$\Delta t_m = \frac{(t_{1db} - t_{rc2g}) - (t_{Bdb} - t_{rB})}{\ln\left(\dfrac{t_{1db} - t_{rc2g}}{t_{Bdb} - t_{rB}}\right)}$$ 55

$$\Delta h_m = \frac{(h_B - h_{sB}) - (h_2 - h_{s2})}{\ln\left(\dfrac{h_B - h_{sB}}{h_2 - h_{s2}}\right)}$$ 56

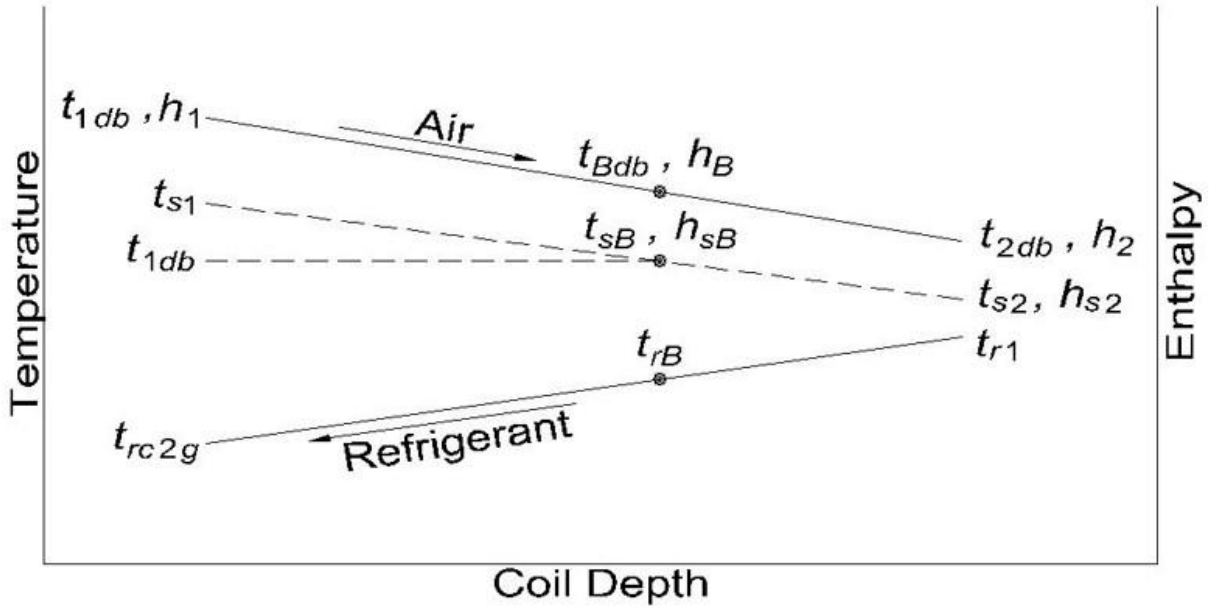**Figure 12 Thermal Diagram of Parallelflow (Physical Counterflow)
Refrigerant Evaporator Air-cooling and Dehumidifying Coils**

6.2.4.3   The Coil Characteristic

The coil characteristic in terms of the individual thermal resistances is calculated using Equation 57 for *single-phase liquid coils* and using Equation 58 for *refrigerant evaporator coils*.

$$C = \frac{R_m + R_L}{c_p R_{aW}}$$ 57

$$C = \frac{R_m + R_r}{c_p R_{aW}}$$ 58

For any point condition within the wet surface region, such as the terminal differences, the coil characteristic is used to obtain the correct division between the airside enthalpy difference, $h - h_s$, and tubeside temperature difference, $t_s - t_L$, using Equation 59 for *single-phase liquid coils* or tubeside temperature difference, $t_s - t_r$, using Equation 60 for *refrigerant evaporator coils*.

$$C = \frac{t_s - t_L}{h - h_s} \qquad\qquad 59$$

$$C = \frac{t_s - t_r}{h - h_s} \qquad\qquad 60$$

C is calculated from Equation 57 or Equation 58 using known values of R_{aW}, R_m and R_L, or R_r. Then with known values of h and t_L or t_r at a given position within the wet surface region, the corresponding values of t_s and h_s can be exactly calculated from Equation 59 or Equation 60 by trial and error using air enthalpy tables.

6.2.4.4   **Dry-wet Boundary Determination for Partially Wet Surface**

Under operating conditions where a portion of the coil is operating wet, the boundary condition between the dry and wet surface regions is established by calculating the air stream enthalpy, h_B, at this point.

For counterflow *single-phase liquid coils*, use Equation 61 to calculate the air enthalpy at the dry-wet boundary.

$$h_B = \frac{t_{1dp} - t_{L2} + yh_1 + Ch_{1dp}}{C + y} \qquad\qquad 61$$

For thermal counterflow (physical parallelflow) *refrigerant evaporator coils*, use Equation 62 to calculate the air enthalpy at the dry-wet boundary.

$$h_B = \frac{t_{1dp} - t_{r1} + yh_1 + Ch_{1dp}}{C + y} \qquad\qquad 62$$

For thermal parallelflow (physical counterflow) *refrigerant evaporator coils*, use Equation 63 to calculate the air enthalpy at the dry-wet boundary.

$$h_B = \frac{t_{1dp} - t_{rc2g} - yh_1 + Ch_{1dp}}{C - y} \, for \, C > y \qquad\qquad 63$$

To calculate the ratio of tubeside fluid temperature rise to air enthalpy drop, y, use Equation 64 for *single-phase liquid coils* and Equation 65 for *refrigerant evaporator coils*.

$$y = \frac{t_{L2} - t_{L1}}{h_1 - h_2} \qquad\qquad 64$$

$$y = \frac{t_{r1} - t_{rc2g}}{h_1 - h_2} \qquad\qquad 65$$

If $h_B \geq h_1$, the entire coil external surface area is wet and $A_D = 0$. Only the wet external surface area, A_W, is calculated for this condition. If $h_B < h_1$, part of the external surface is operating dry. For this condition, the dry, A_D, and wet, A_W, external surface areas are separately calculated.

The air dry-bulb temperature at the dry-wet boundary is calculated by Equation 66.

$$t_{Bdb} = t_{1db} - \left(\frac{h_1 - h_B}{c_p}\right) \qquad\qquad 66$$

The total capacity for the dry surface region is calculated by Equation 67.

$$q_{tD} = k_7 w_a (h_1 - h_B)$$ <div style="float:right">67</div>

The total capacity for the wet surface region is calculated by Equation 68.

$$q_{tW} = q_t - q_{tD}$$ <div style="float:right">68</div>

For counterflow *single-phase liquid coils*, use Equation 69 to calculate the cold *single-phase liquid* temperature at the dry-wet boundary.

$$t_{LB} = t_{L2} - y(h_1 - h_B)$$ <div style="float:right">69</div>

For thermal counterflow (physical parallelflow) *refrigerant evaporator coils*, use Equation 70 to calculate the *refrigerant* temperature at the dry-wet boundary. For *refrigerant evaporator coils* with large amounts of glide, for example R-407C, the thermal flow arrangement only applies when the *refrigerant* temperature decreases through the tube circuits.

$$t_{rB} = t_{r1} - y(h_1 - h_B)$$ <div style="float:right">70</div>

For thermal parallelflow (physical counterflow) *refrigerant evaporator coils*, use Equation 71 to calculate the *refrigerant* temperature at the dry-wet boundary. For *refrigerant evaporator coils* with large amounts of glide, for example R-407C, the thermal flow arrangement only applies when the *refrigerant* temperature decreases through the tube circuits.

$$t_{rB} = t_{rc2g} + y(h_1 - h_B)$$ <div style="float:right">71</div>

6.2.4.5   **Determination of Leaving Air Dry-Bulb Temperature**

If the effective coil surface temperature is greater than or equal to the entering air *dew point* temperature ($t_{\bar{s}} \geq t_{1dp}$), the coil surface is dry and the leaving air dry-bulb temperature shall be calculated using the method described in Section 6.2.3.

6.2.4.5.1   **For Completely Wet Coil Surface**

The leaving air dry-bulb temperature, t_{2db}, shall be calculated using Equation 72 through Equation 75.

$$t_{2db} = t_{\bar{s}} + (t_{1db} - t_{\bar{s}})\, e^{-c}$$ <div style="float:right">72</div>

$$c = \frac{A_o}{k_7 c_p w_a R_{aD}}$$ <div style="float:right">73</div>

$$e^{-c} = \frac{h_2 - h_{\bar{s}}}{h_1 - h_{\bar{s}}} = \frac{t_{2db} - t_{\bar{s}}}{t_{1db} - t_{\bar{s}}}$$ <div style="float:right">74</div>

The saturated air enthalpy, $h_{\bar{s}}$, corresponding to the effective coil surface temperature, $t_{\bar{s}}$, is calculated by Equation 75.

$$h_{\bar{s}} = h_1 - \left(\frac{h_1 - h_2}{1 - e^{-c}}\right)$$ <div style="float:right">75</div>

6.2.4.5.2   **For Partially Wet Coil Surface**

The leaving air dry-bulb temperature, t_{2db}, shall be calculated using Equation 76 through Equation 79.

$$t_{2db} = t_{\bar{s}} + (t_{Bdb} - t_{\bar{s}})\, e^{-c} \qquad\qquad 76$$

$$c = \frac{A_W}{k_7 c_p w_a R_{aD}} \qquad\qquad 77$$

$$e^{-c} = \frac{h_2 - h_{\bar{s}}}{h_B - h_{\bar{s}}} = \frac{t_{2db} - t_{\bar{s}}}{t_{Bdb} - t_{\bar{s}}} \qquad\qquad 78$$

The saturated air enthalpy, $h_{\bar{s}}$, corresponding to the effective coil surface temperature, $t_{\bar{s}}$, is calculated in Equation 79.

$$h_{\bar{s}} = h_B - \left(\frac{h_B - h_2}{1 - e^{-c}}\right) \qquad\qquad 79$$

6.2.4.6   **Determination of Air Pressure Drop Through Coil.**

6.2.4.6.1   **Air Pressure Drop Through Coil at Standard Dry Air Density**

The air pressure drop for partially wet air-cooling and dehumidifying coils shall be determined by either of two methods. While either method is can be used, the first is more conservative.

6.2.4.6.1.1   **Option 1**

Treat as though the entire coil is wet and use the wet surface air pressure drop data (as was the case for previous versions of the standard), as shown in Equation 80.

$$\Delta p_{a\,STD} = (DPSW)N_r \qquad\qquad 80$$

For example, if $DPSW$ is 0.125 in H_2O/row, $\Delta p_{a\,STD}$ for a six-row coil can be calculated using Equation 80 as follows:

$$\Delta p_{a\,STD} = (DPSW)N_r = (0.125)6 = 0.75 \text{ in } H_2O$$

6.2.4.6.1.2   **Option 2**

Use the dry surface air pressure drop data for the dry portion of the coil and use the wet surface air pressure drop data for the wet portion of the coil, as shown in Equation 81.

$$\Delta p_{a\,STD} = \frac{\left(A_D(DPSD) + A_W(DPSW)\right)N_r}{A_o} \qquad\qquad 81$$

For example, if a coil has a dry external surface area, A_D, of 36 ft^2, $DPSD$ of 0.10 in H_2O/row, wet external surface area A_W, of 12 ft^2, $DPSW$ of 0.15 in H_2O/row, and a total external surface area, A_o, of 48 ft^2, the $\Delta p_{a\,STD}$ for a six-row coil can be calculated using Equation 81 as follows:

$$\Delta p_{a\,STD} = \frac{\left(A_D(DPSD) + A_W(DPSW)\right)N_r}{A_o} = \frac{(36(0.10) + 12(0.15))6}{48} = \frac{(3.6 + 1.8)6}{48} = \frac{32.4}{48} = 0.68 \text{ in } H_2O$$

6.2.4.6.2   Air Pressure Drop Through Coil at Job Air Conditions

Calculate the coil air pressure drop at job (actual) conditions using Equation 82 through Equation 84 where the average air absolute pressure at coil, P_s, is in Hg abs. See Appendix C for further information on converting between actual and standard conditions. To simplify calculations, P_s can be defined as the air barometric pressure at *job conditions*, $P_{b\,JOB}$.

$$\Delta p_{a\,JOB} = \frac{(F_{aD}(A_D\Delta p_{sD}) + F_{aW}(A_W\Delta p_{sW}))N_r}{A_o} \qquad 82$$

$$F_{aD} = \rho_{aSTD}\left(\frac{k_{12}\big(k_5 + 0.5(t_{1db} + t_{2db})\big)}{P_s k_1 k_{11}}\right)\left(\frac{1 + \dfrac{W_1}{k_9}}{1 + W_1}\right) \qquad 83$$

$$F_{aW} = \rho_{aSTD}\left(\frac{k_{12}\big(k_5 + 0.5(t_{1db} + t_{2db})\big)}{P_s k_1 k_{11}}\right)\left(\frac{1 + 0.5(\dfrac{W_1 + W_2}{k_9})}{1 + 0.5(W_1 + W_2)}\right) \qquad 84$$

6.2.4.7   Determination of Sensible and Total Capacities

The airside Sensible, q_s, and total, q_t, capacities are calculated from applicable Equations in ANSI/ASHRAE Standard 33-2016.

6.3   Reduction of Laboratory Test Data to Determine Parameters for Ratings

6.3.1   Coil Physical Data Calculations

The procedure for calculation of coil physical data and *fin* efficiency is based on analysis of *laboratory tests*, preparation of ratings, and equations in Section 4.

6.3.2   Calculation of Total Metal Thermal Resistance of Fin and Tube Assembly

The calculation procedure for establishing the total metal thermal resistance, R_m, is based on equations listed under Section 6.2.2. Experimental data are not required for these calculations. Metal thermal conductivities are contained in Table 5.

R_m consists of two metal resistances in series:

1) The variable *fin* metal thermal resistance, R_f, of the external *fin* metal based on total external surface effectiveness for either dry or wet surface.

2) The constant tube metal thermal resistance, R_t, of the prime tube wall metal. Depending upon tube design, R_t can or cannot be negligible.

Table 5 Metal Thermal Conductivities

Material	Temperature	Temperature	Thermal Conductivity, *k*	Thermal Conductivity, *k*
	°C	°F	W/(m. °C)	Btu·ft/(h·ft²·°F)
Aluminum Alloy 1100 Temper O	25	77	222.1	128.3
Aluminum Alloy 3003 Temper O	25	77	193.3	111.7
Aluminum Alloy 3003 Temper H18	25	77	154.4	89.2
Aluminum Alloy 7072 Temper O	25	77	222.1	128.3
Copper (C11000)	20	68	391.1	226.0
Copper (C12200)	20	68	339.2	196.0
Red Brass (85-15%, C23000)	20	68	159.2	92.0
Cupronickel (90-10%, C70600)	20	68	45.0	26.0
Cupronickel (70-30%, C71500)	20	68	29.4	17.0
Admiralty (C44300, C44400, C44500)	20	68	110.8	64.0
Steel-Carbon (SAE 1020)	100	212	51.9	30.0
Stainless Steel 304, 304L, 316, 316L	100	212	16.3	9.4
Stainless Steel 410 and 420	100	212	24.9	14.4
Stainless Steel 347 and 321	100	212	16.1	9.3

Note:

- Copper Development Association, International Nickel Co., and The Aluminum Association are the sources used to compile this information.

  An illustrative plot of (R_{aD} + R_{mD}) vs. R_{aD} per Equation 85 is shown in Figure 13 and is for dry surface application only. This plot is used for analysis of either laboratory test results or ratings of dry surface coils.

$$R_{aD} + R_{mD} = \frac{R_{aD}}{\eta} + R_t \qquad\qquad 85$$
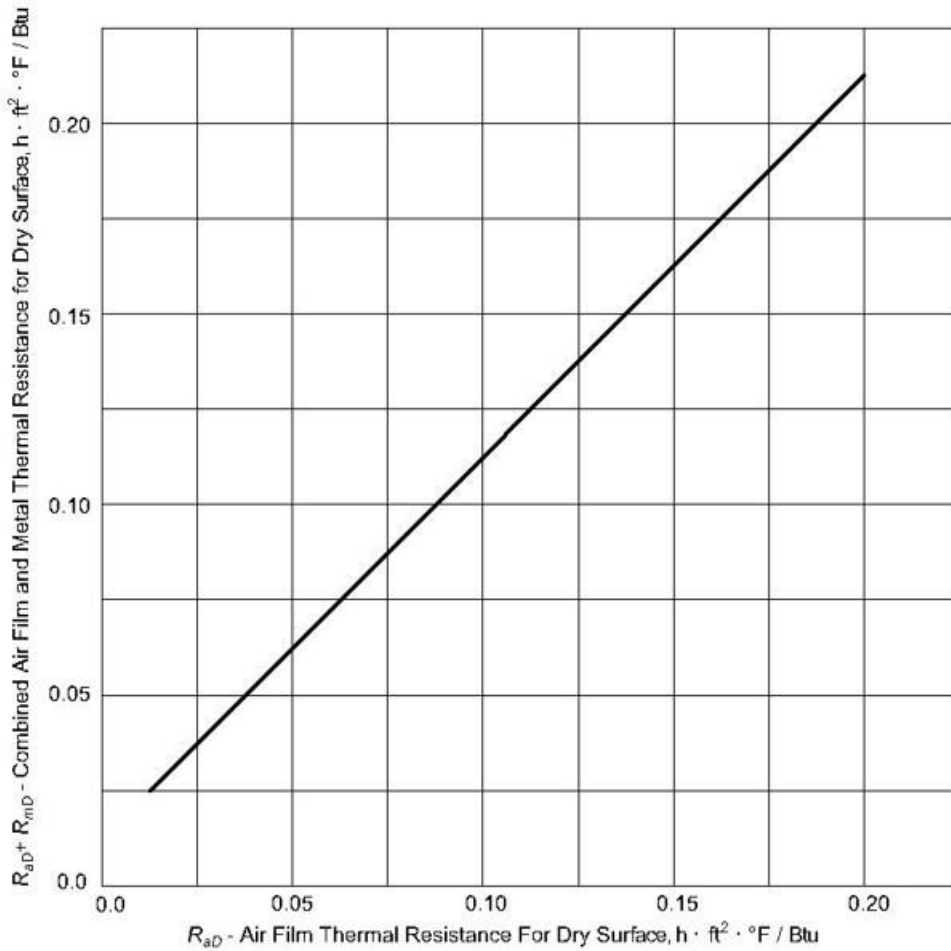
**Figure 13 Combined Air Film and Metal Thermal Resistance for Dry
Surface versus Air Film Thermal Resistance for Dry Surface (IP)**

An illustrative plot of R_{mD} vs. f_{aD} for dry surface coils (Equation 86) and R_{mW} vs. f_{aW} for wet surface coils (Equation 87) is shown in Figure 14. The data illustrated in Figure 14 are used to determine R_{mD} and R_{mW} in either the analysis of laboratory test results or for ratings of all coil types.

$$f_a = f_{aD} = \frac{1}{R_{aD}} \qquad\qquad 86$$

$$f_a = f_{aW} = \left(\frac{1}{R_{aW}}\right)\left(\frac{m"}{c_p}\right) \qquad\qquad 87$$

For wet surface coils, $m"/c_p$ is determined using Equation 7.

$$\frac{m"}{c_p} = \frac{\frac{dh}{dt} \text{ at } t_s}{c_p} = \frac{\text{Slope of the saturated air temperature enthalpy curve at the coil surface temperature, } t_s}{\text{air-water vapor mixture specific heat at constant pressure, } c_p} \qquad 7$$
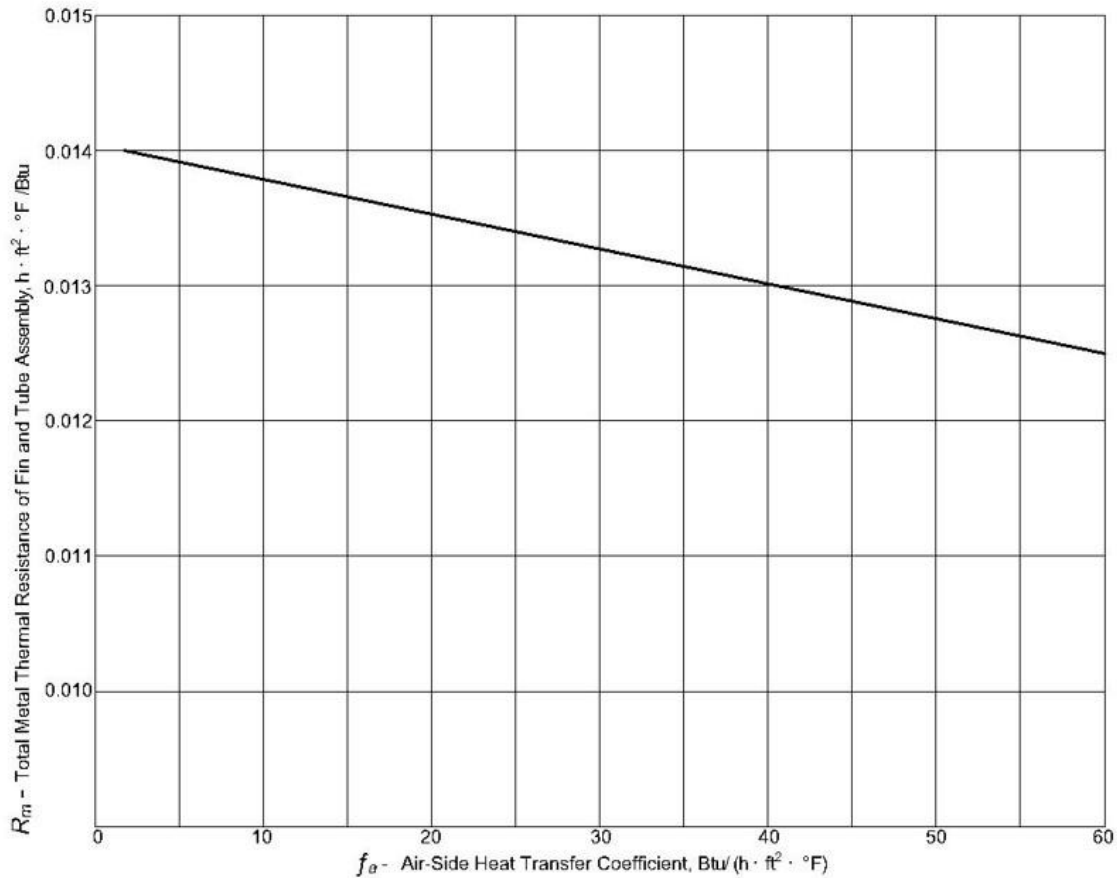
**Figure 14 Total Metal Thermal Resistance of Fin and Tube Assembly
Based on Total External Surface Effectiveness (IP)**

6.3.3 Steam Coils

The calculation procedure for determining the performance factors for *steam coil* ratings is derived from *laboratory tests*. This analysis is based on the applicable heat transfer equations under Section 6.2.3. This procedure includes parameters, based on experimental results, used to determine coil air pressure drop. The steam pressure drop inside tubes is determined by the calculation procedure.

From analysis of *laboratory tests* on *steam coils* the following plots of experimental data are used for ratings as illustrated in Figure 15:

R vs. V_a

$\dfrac{\Delta p_{sD}}{N_r}$ vs. V_a

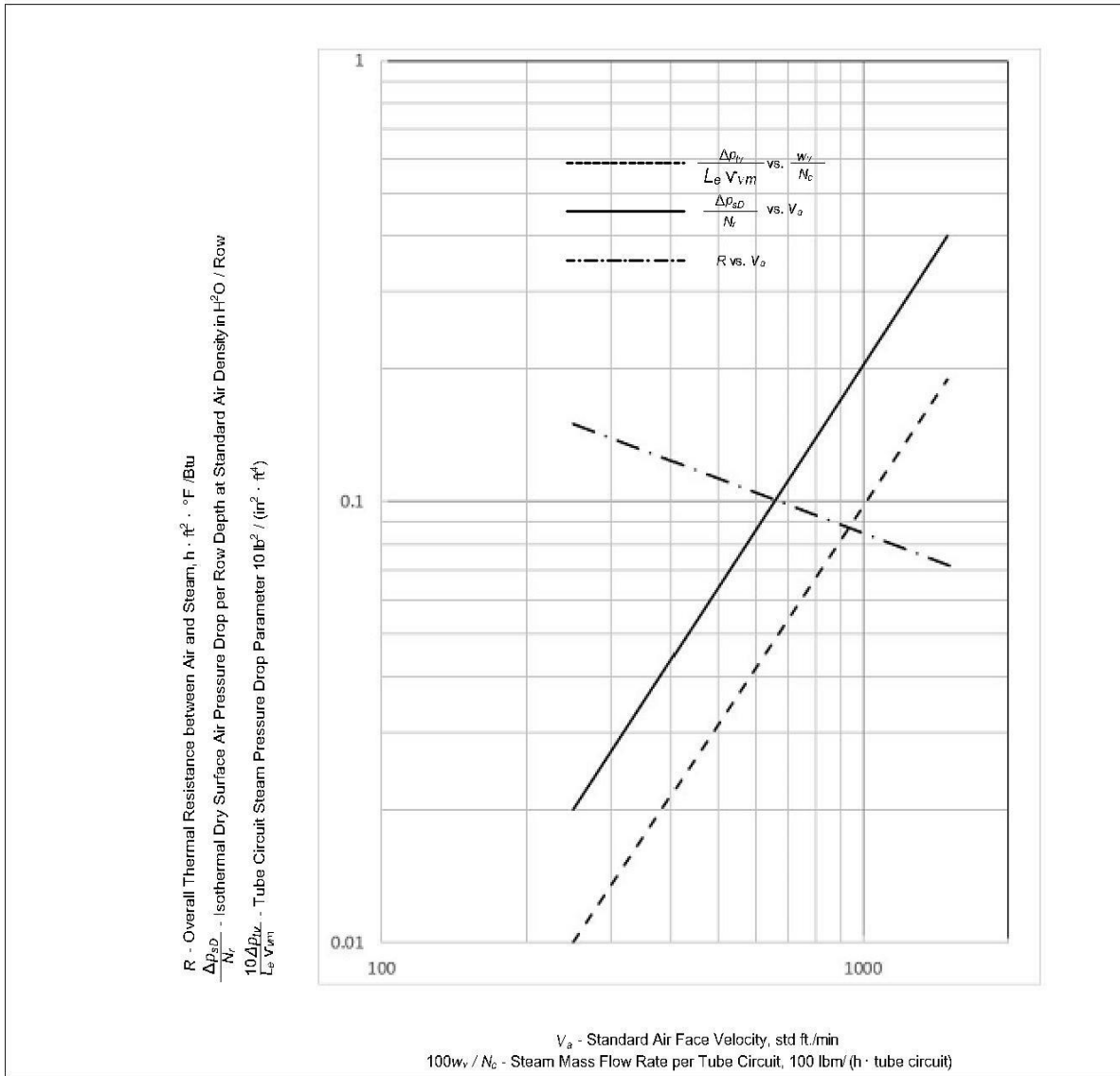$\dfrac{\Delta p_{tv}}{L_e \upsilon_{vmg}}$ vs. $\dfrac{w_v}{N_c}$

Figure 15 Rating Data for Steam Coils (I-P)

6.3.4   Cold Single-phase Liquid Sensible Only Air-cooling Coils and Hot Single-phase Liquid Air-heating Coils

The calculation procedure for determining the performance factors for cold *single-phase liquid* sensible only air-cooling coil and hot *single-phase liquid air-heating coil* ratings is derived from *laboratory tests.* This analysis is based on the applicable heat transfer equations under Section 6.2.3. This procedure includes parameters, based on experimental results, used to determine coil air pressure drop. The *single-phase liquid* pressure drop inside tubes is determined by the calculation procedure

From analysis of *laboratory tests* on cold *single-phase liquid* sensible only air-cooling coils or hot *single-phase liquid air-heating coils*, or both , the following plots of experimental data are used for ratings as illustrated in Figure 16 along with tubeside *single-phase liquid* performance (Darcy-Weisbach friction factor and Colburn-j heat transfer factor versus Reynolds number) for the appropriate tube surface.

R_{aD} vs. V_a

$(R_{aD} + R_{mD})$ vs. V_a

$\dfrac{\Delta p_{sD}}{N_r}$ vs. V_a

For coils with *smooth tubes*: f'_{DWL} vs. Re_L and j_L vs. Re_L (refer to Figure 3).

For coils with *turbulators* or *internally grooved tubes*: f'_{DWL} vs. Re_L and j_L vs. Re_L per Section 5.5 (refer to Figure 3).
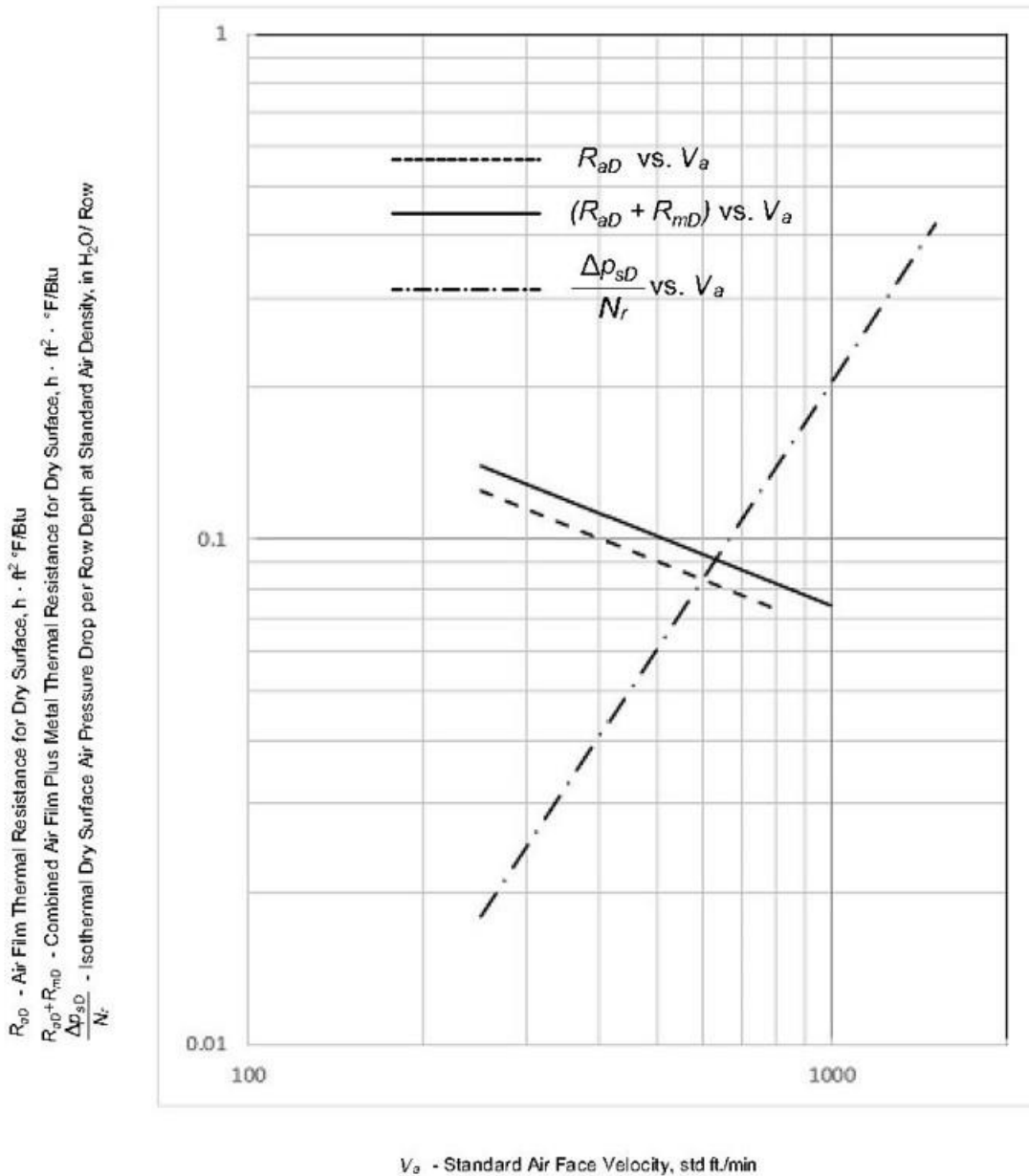
Figure 16 Rating Data for Cold Single-phase Liquid Sensible Only Air-cooling Coils and Hot Single-phase Liquid Air-heating Coils (IP)

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6.3.5 **Cold Single-phase Liquid Air-cooling and Dehumidifying Coils**

The calculation procedure for determining the performance factors for cold *single-phase liquid* air-cooling and dehumidifying coil ratings, derived from *laboratory tests,* is based on the applicable heat transfer equations under Section 6.2.4. Separate procedures are included for conditions where the coil surface is either completely wet or completely dry.

From analysis of *laboratory tests* on cold *single-phase liquid* air-cooling and dehumidifying coils, the following plots of experimental data are used for ratings as illustrated in Figure 17 along with tubeside *single-phase liquid* performance (Darcy-Weisbach friction factor and Colburn-j heat transfer factor versus Reynolds number) for the appropriate tube surface.

R_{aW} vs. V_a

$\dfrac{\Delta p_{sW}}{N_r}$ vs. V_a

For coils with *smooth tubes*: f'_{DWL} vs. Re_L and j_L vs. Re_L (see Figure 3).

For coils with *turbulators* or *internally grooved tubes*: f'_{DWL} vs. Re_L and j_L vs. Re_L per Section 5.5 (see Figure 3).

Figure 17 shows the following data from Figure 16 (see Section 6.3.4) based on dry surface tests that are applicable for partially wet surface operation:

R_{aD} vs. V_a

$\dfrac{\Delta p_{sD}}{N_r}$ vs. V_a
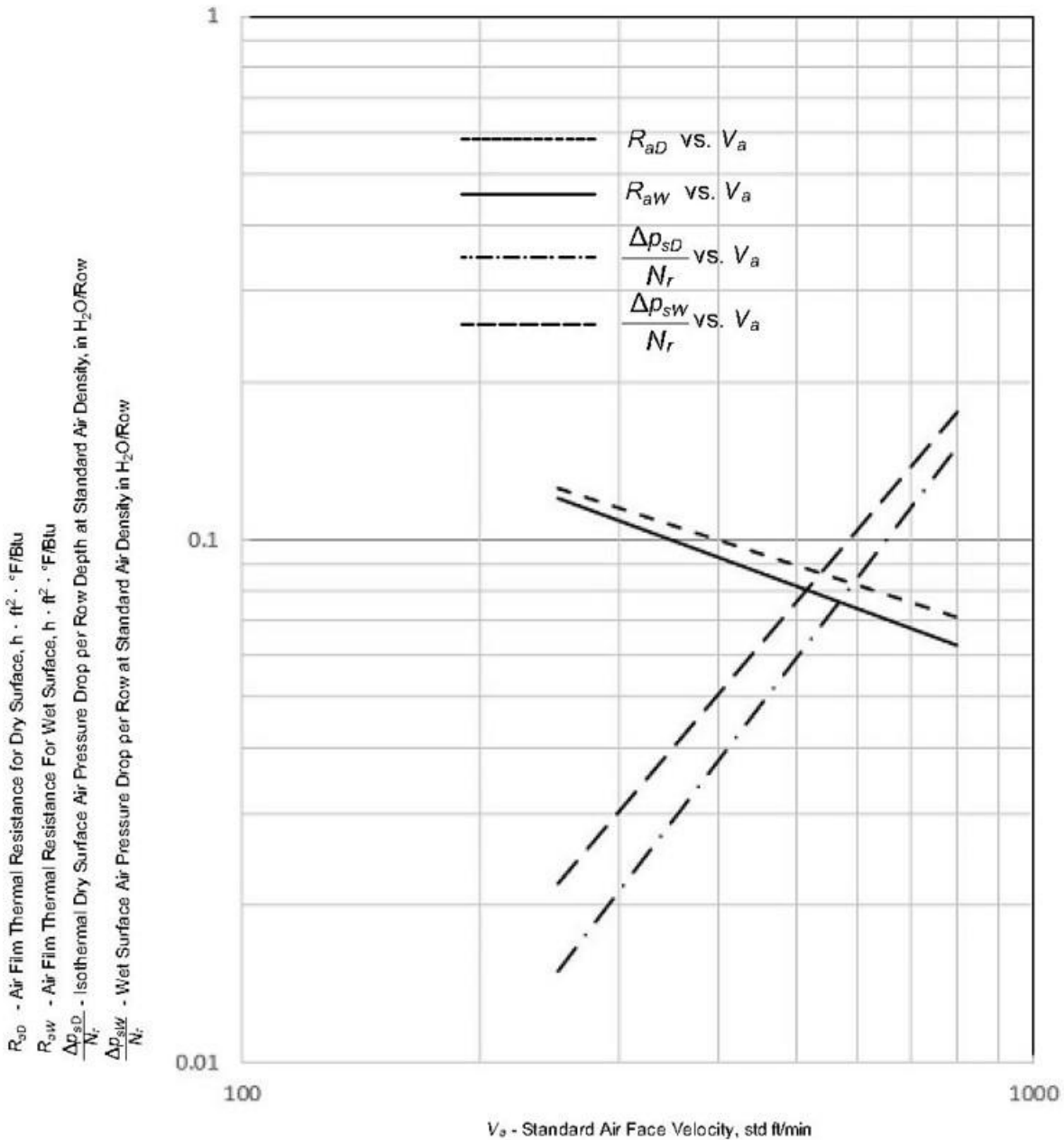
Figure 17 Rating Data for Cold Single-phase Liquid Air-cooling and Dehumidifying Coils (IP)

6.3.6   Refrigerant Evaporator Air-cooling and Dehumidifying Coils

The calculation procedure for determining the performance factors for *refrigerant* evaporator air-cooling and dehumidifying coil ratings is derived from *laboratory tests*. This analysis is based on the applicable heat transfer equations in Section 6.2.4. Separate procedures are included for conditions where the coil surface is either completely wet or completely dry.

From analysis of *laboratory tests* on *refrigerant* evaporator air-cooling and dehumidifying coils as, the following plots of experimental data are used for ratings as illustrated in Figure 18.

$$\frac{1}{f_r} \quad \text{vs.} \quad \frac{q_t}{N_c}$$

$$\frac{\Delta p_{rc}}{L_e v_{rc2g}} \quad \text{vs.} \quad \frac{w_r}{N_c}$$

Figure 18 shows the following data from Figure 17 determined from water coil tests and previous calculations.

$$\frac{\Delta p_{sD}}{N_r} \quad \text{vs.} \quad V_a$$

$$\frac{\Delta p_{sW}}{N_r} \quad \text{vs.} \quad V_a$$

$$R_{aD} \quad \text{vs.} \quad V_a$$

$$R_{aW} \quad \text{vs.} \quad V_a$$
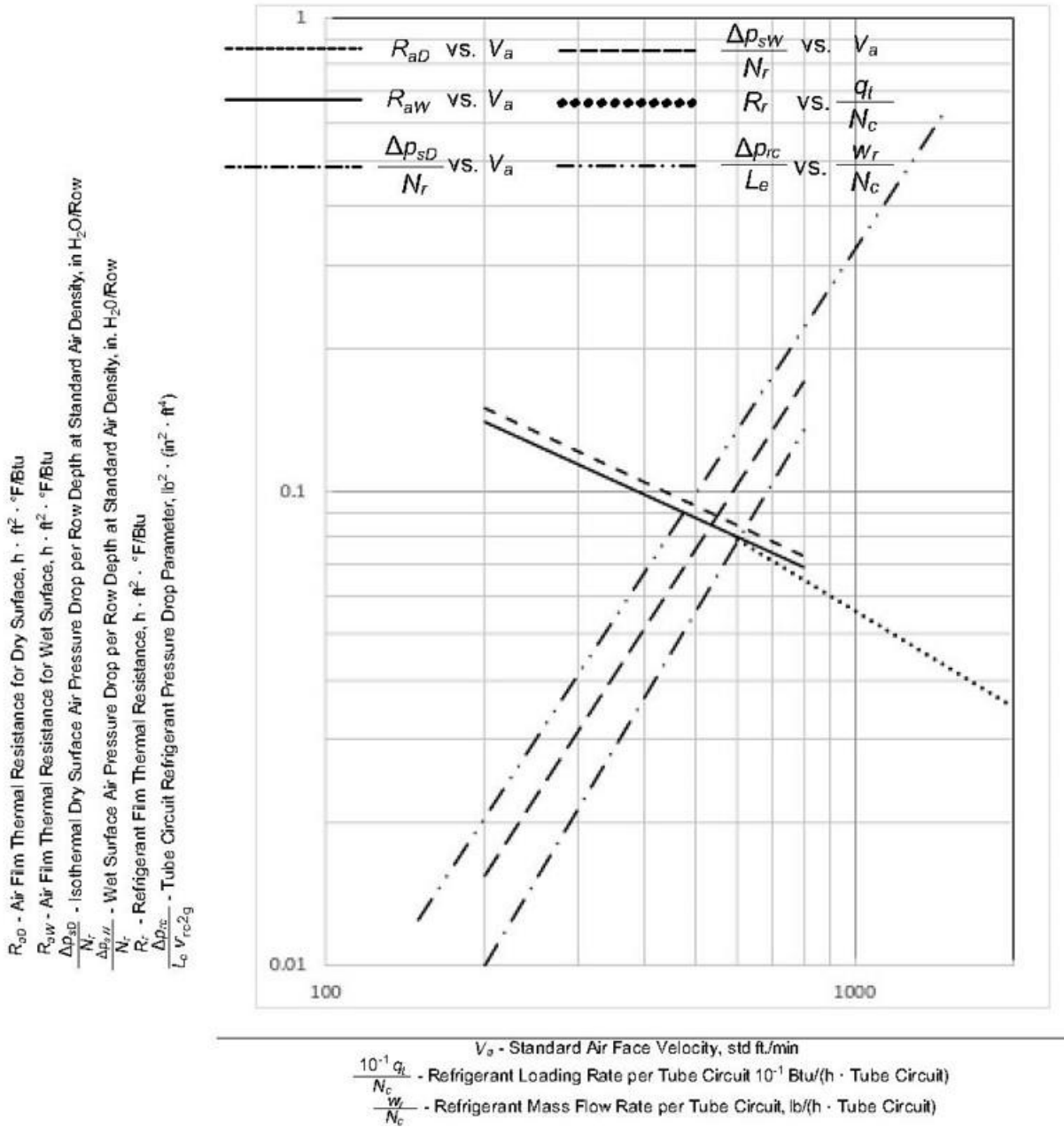
**Figure 18 Rating Data for Refrigerant Evaporator Air-cooling and
Dehumidifying Coils (IP)**

6.4   Standard Ratings

6.4.1   Tolerances

Standard ratings shall indicate that any coil selected at random has a total capacity, when tested, not less than 95% of the published total capacity of the coil. Published values of air pressure drop through coil at standard dry air density, shall not be exceeded by more than 10%, or 12 Pa (0.05 in H_2O), whichever is greater. Published values of *single-phase liquid* head loss through coil, shall not be exceeded by more than 10%, or 1.0 ft of *single-phase liquid*, whichever is greater.

6.4.2 **Computations**

Computations for *standard ratings* shall be based on heat transfer coefficients and coil characteristics obtained by *laboratory tests*.

In this standard, expressions for logarithmic mean effective differences, Δt_{lm} and Δh_{lm}, are only illustrated for the case of thermal counterflow between the air and tubeside fluids, as defined under Sections 6.2.3.4 and 6.2.4.2 except for *refrigerants* where thermal parallel flow is shown in Figure 7 and Figure 12. The manufacturer shall include appropriate allowances to these logarithmic mean effective differences for those coil designs where the tube *circuiting* arrangement causes deviations from the thermal counterflow relationships described in this standard.

In publishing *standard ratings*, the manufacturer shall include appropriate allowances for the effects of tubeside fluid pressure drop, orientations other than *standard coil orientation*, and condensate accumulation, on coil capacity and air pressure drop.

6.4.3 **Steam Coils**

Standard ratings for determining either number of rows, N_r, requirements or *sensible capacity*, q_s, for specific *job conditions* can be obtained by use of the following data:

1) Performance factors as illustrated in Figure 15.

2) Applicable heat transfer equations in Section 6.2.3.

3) Manufacturer established steam pressure drop of tubeside coil attachments ($\Delta p_{v\,ATT}$).

6.4.4 **Cold Single-phase Liquid Sensible Only Air-cooling Coils and Hot Single-phase Liquid Air-heating Coils**

This method of rating the sensible only air-cooling coils is for application where the air sensible heat ratio, $q_s/q_t \geq 0.95$. For conditions where, $q_s/q_t < 0.95$, use the wet surface total heat method described in Section 6.4.5. Cold *single-phase liquid* sensible only air-cooling coils and hot *single-phase liquid air-heating coils* can be rated.

Standard ratings for determining either number of rows, N_r, requirements or *sensible capacity*, q_s, for specific *job conditions* can be obtained by use of the following data:

1) Performance factors as illustrated in Figure 16.

2) For coils with *smooth tubes*: f'_{DWL} vs. Re_L and j_L vs. Re_L (see Figure 3).

3) For coils with *turbulators* or *internally grooved tubes*: f'_{DWL} vs. Re_L and j_L vs. Re_L per Section 5.5 (see Figure 3).

4) Applicable heat transfer equations in Section 6.2.3.

5) Manufacturer established *single-phase liquid* pressure drop of tubeside coil attachments ($\Delta p_{L\,ATT}$).

6.4.5 **Cold Single-phase Liquid Air-cooling and Dehumidifying Coils**

This method of rating is for application where the air sensible heat ratio, $q_s/q_t < 0.95$. For conditions where $q_s/q_t \geq 0.95$, use the dry surface sensible heat method described in Section 6.4.4. Cold *single-phase liquid* air-cooling and dehumidifying coils can be rated.

Standard ratings for determining either total external surface area, A_o, requirements or total heat capacity, q_t, for specific *job conditions* can be obtained by use of the following data:

1) Performance factors as illustrated in Figure 17.

2) For coils with *smooth tubes*: f'_{DWL} vs. Re_L and j_L vs. Re_L (Figure 3).

3) For coils with *turbulators* or *internally grooved tubes*: f'_{DWL} vs. Re_L and j_L vs. Re_L per Section 5.5 (refer to Figure 3).

4) Applicable heat transfer equations in Sections 6.2.3 and 6.2.4.

5) m''/c_P for wet surface.

6) R_m for dry and wet surface as illustrated in Figure 14.

7) Manufacturer established *single-phase liquid* pressure drop of tubeside coil attachments ($\Delta p_{L\ ATT}$).

6.4.6 Refrigerant Evaporator Sensible Only Air-cooling Coils

This method of rating the sensible only air-cooling coils is for application where the air sensible heat ratio, $q_s/q_t \geq 0.95$. For conditions where, $q_s/q_t < 0.95$, use the wet surface total heat method described in Section 6.4.7. *Refrigerant* evaporator sensible only air-cooling coils can be rated.

Standard ratings for determining either number of rows, N_r, requirements or *sensible capacity*, q_s, for specific *job conditions* can be obtained by using the dry surface sensible heat transfer method described in Section 6.4.4 except for the following change of data:

1) Use the plot for $1/f_r$ as illustrated in Figure 18.

2) Use the plot for $\Delta p_r/(L_e v_{rc2g})$ as illustrated in Figure 18.

3) Use manufacturer established *refrigerant* pressure drop of tubeside coil attachments ($\Delta p_{r\ ATT}$).

6.4.7 Refrigerant Evaporator Air-cooling and Dehumidifying Coils

This method of rating is for application where the air sensible heat ratio, $q_s/q_t < 0.95$. For conditions where $q_s/q_t \geq 0.95$, use the dry surface sensible heat method described in Section 6.4.6. *Refrigerant* evaporator air-cooling and dehumidifying coils can be rated.

Standard ratings for determining either total external surface area, A_o, requirements or total heat capacity, q_t, for specific *job conditions* can be obtained by use of the following data:

1) Performance factors as illustrated in Figure 18.

2) Applicable heat transfer equations in Section 6.2.3 and Section 6.2.4.

3) m''/c_p for wet surface.

4) R_m for dry and wet surface as illustrated in Figure 14.

5) Manufacturer established *refrigerant* pressure drop of tubeside coil attachments ($\Delta p_{r\ ATT}$).

6.5 Airside Effectiveness

Airside effectiveness is used for determining the *sensible capacity*, q_s, for specific *job conditions* using the dry surface sensible heat transfer method described in Section 6.4.3, Section 6.4.4, and Section 6.4.6.

Calculate the initial air-to-tubeside fluid temperature difference, ITD, the air dry-bulb temperature difference, Δt_{adb}, and the air-to-fluid heat capacity ratio, M, using equations for the appropriate coil type as follows:

6.5.1 Single Tube Steam Coils

Equation 88, Equation 89, and Equation 90 are used to calculate *single tube steam coils*.

$$ITD = t_{v1g} - t_{1db} \qquad\qquad 88$$

$$\Delta t_{adb} = t_{a2db} - t_{a1db} \qquad\qquad 89$$

$$M = 0 \qquad\qquad 90$$

6.5.2 Distributing Tube Steam Coils

Equation 91, Equation 89, and Equation 90 are used to calculate *distributing tube steam coils*.

$$ITD = t_{v2g} - t_{1db} \qquad\qquad 91$$

$$\Delta t_{adb} = t_{a2db} - t_{a1db} \qquad\qquad 89$$

$$M = 0 \qquad\qquad 90$$

6.5.3   **Hot Single-phase Liquid Air-heating Coils**

Equation 92, Equation 89, and Equation 93 are used to calculate hot *single-phase liquid* air heating coils.

$$ITD = t_{L1} - t_{1db} \tag{92}$$

$$\Delta t_{adb} = t_{a2db} - t_{a1db} \tag{89}$$

$$M = \frac{k_{13}c_p Q_{a\,STD}}{w_L c_{pL}} = \frac{(t_{L1} - t_{L2})}{\Delta t_{adb}} \tag{93}$$

6.5.4   **Cold Single-phase Liquid Sensible Only Air-cooling Coils**

Equation 94, Equation 95, and Equation 96 are used to calculate cold *single-phase liquid* sensible only air-cooling coils.

$$ITD = t_{a1db} - t_{L1} \tag{94}$$

$$\Delta t_{adb} = t_{a1db} - t_{a2db} \tag{95}$$

$$M = \frac{k_{13}c_p Q_{a\,STD}}{w_L c_{pL}} = \frac{(t_{L2} - t_{L1})}{\Delta t_{adb}} \tag{96}$$

6.5.5   **Refrigerant Evaporator Sensible Only Air-cooling Coils**

Equation 97, Equation 95, and Equation 98 are used to calculate *refrigerant* evaporator sensible only air-cooling coils.

$$ITD = t_{a1db} - t_{rc2g}\# \tag{97}$$

$$\Delta t_{adb} = t_{a1db} - t_{a2db} \tag{95}$$

$$M = \frac{(t_{r1} - t_{rc2g}\#)}{\Delta t_{adb}} \tag{98}$$

Where # indicates that $t_{rc2g} < t_{r1}$, that requires that the temperature change due to *refrigerant* pressure drop is greater than that due to glide.

Calculate the sensible heat transfer exponent, NTU, using Equation 99.

$$NTU = \frac{A_o}{(k_{13}c_p Q_{a\,STD}R)} = \frac{\Delta t_{adb}}{\Delta t_m} \tag{99}$$

Calculate the airside effectiveness, ε, for flow configuration as follows:

6.5.5.1   **Airside Effectiveness for All Coil Types**

Equation 100 is used to calculate airside effectiveness for all coil types.

$$\varepsilon = \frac{\Delta t_{adb}}{ITD} \tag{100}$$

6.5.5.2   **Crossflow Airside Effectiveness**

Equation 101 and Equation 102 are used to calculate crossflow airside effectiveness. Crossflow airside effectiveness is shown graphically on Figure 19.

For $M > 0$: $\varepsilon = \dfrac{1 - e^{-M\left(1 - e^{-NTU}\right)}}{M}$ <div style="float:right">101</div>

For $M = 0$: $\varepsilon = 1 - e^{-NTU}$ <div style="float:right">102</div>

6.5.5.3 Thermal Two-pass Cross-counterflow Airside Effectiveness

Equation 103 and Equation 104 are used to calculate thermal two-pass cross-counterflow airside effectiveness. Thermal two-pass cross-counterflow airside effectiveness is shown graphically on Figure 20.

For $M > 0$: $\varepsilon = \dfrac{1}{M}\left(1 - \left(\dfrac{e^{-M\left(1 - e^{(-NTU/2)}\right)}}{\cosh\left(M\left(1 - e^{(-NTU/2)}\right)\right) + e^{(-NTU/2)}\sinh\left(M\left(1 - e^{(-NTU/2)}\right)\right)}\right)\right)$ <div style="float:right">103</div>

For $M = 0$: $\varepsilon = 1 - e^{-NTU}$ <div style="float:right">104</div>

6.5.5.4 Thermal Counterflow Airside Effectiveness

Equation 105, Equation 106, and Equation 104 are used to calculate thermal counterflow airside effectiveness. Thermal counterflow airside effectiveness is shown graphically on Figure 21.

For $M \neq 1$: $\varepsilon = \dfrac{1 - e^{-NTU(1-M)}}{1 - M\,e^{-NTU(1-M)}}$ <div style="float:right">105</div>

For $M = 1$: $\varepsilon = \dfrac{NTU}{NTU + 1}$ <div style="float:right">106</div>

For $M = 0$: $\varepsilon = 1 - e^{-NTU}$ <div style="float:right">104</div>

6.5.5.5 Thermal Parallelflow Airside Effectiveness

Equation 107 and Equation 104 are used to calculate thermal parallelflow airside effectiveness. Thermal parallelflow airside effectiveness is shown graphically on Figure 22.

For $M > 0$: $\varepsilon = \dfrac{1 - e^{-NTU(1+M)}}{1 + M}$ <div style="float:right">107</div>

For $M = 0$: $\varepsilon = 1 - e^{-NTU}$ <div style="float:right">104</div>

The mean temperature difference, Δt_{m}, in the NTU expression is defined as the log mean temperature difference for counterflow, $\Delta t_{lm\ counterflow}$, multiplied by an appropriate correction factor, F_{LMTD}, for the flow arrangement see Equation 108.

$\Delta t_{m} = F_{LMTD}\Delta t_{lm\ counterflow}$ <div style="float:right">108</div>
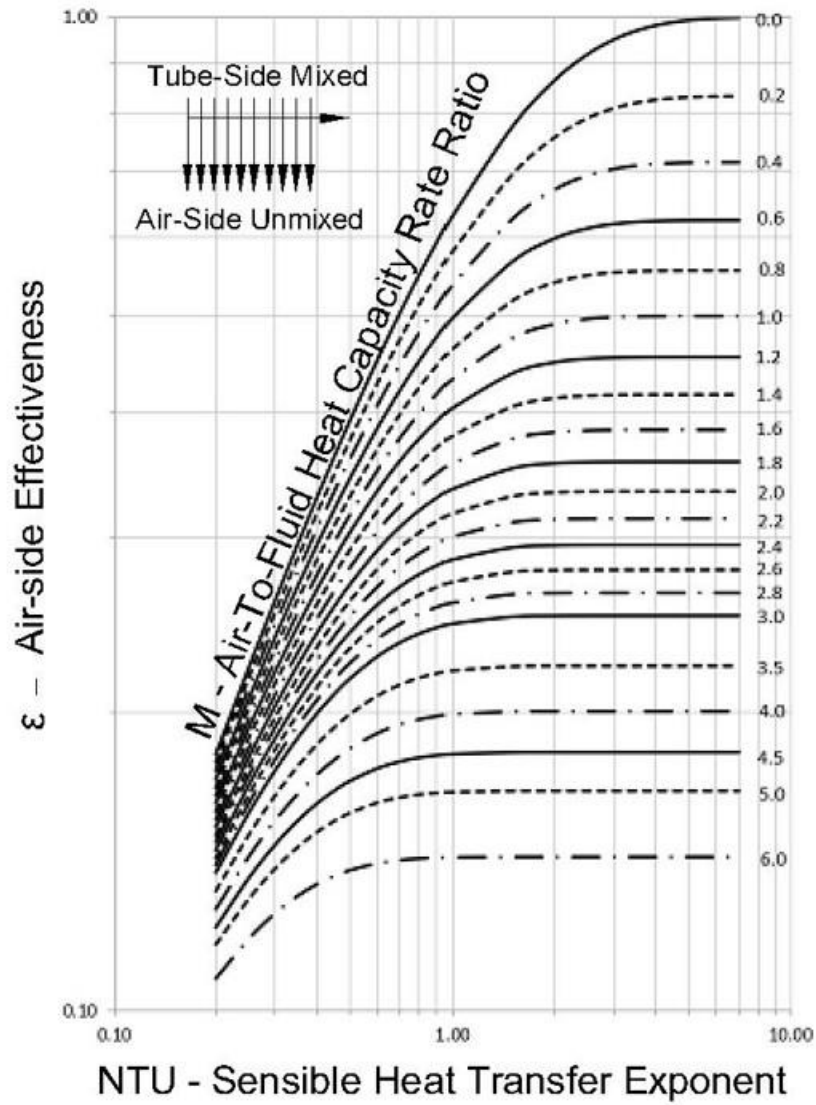
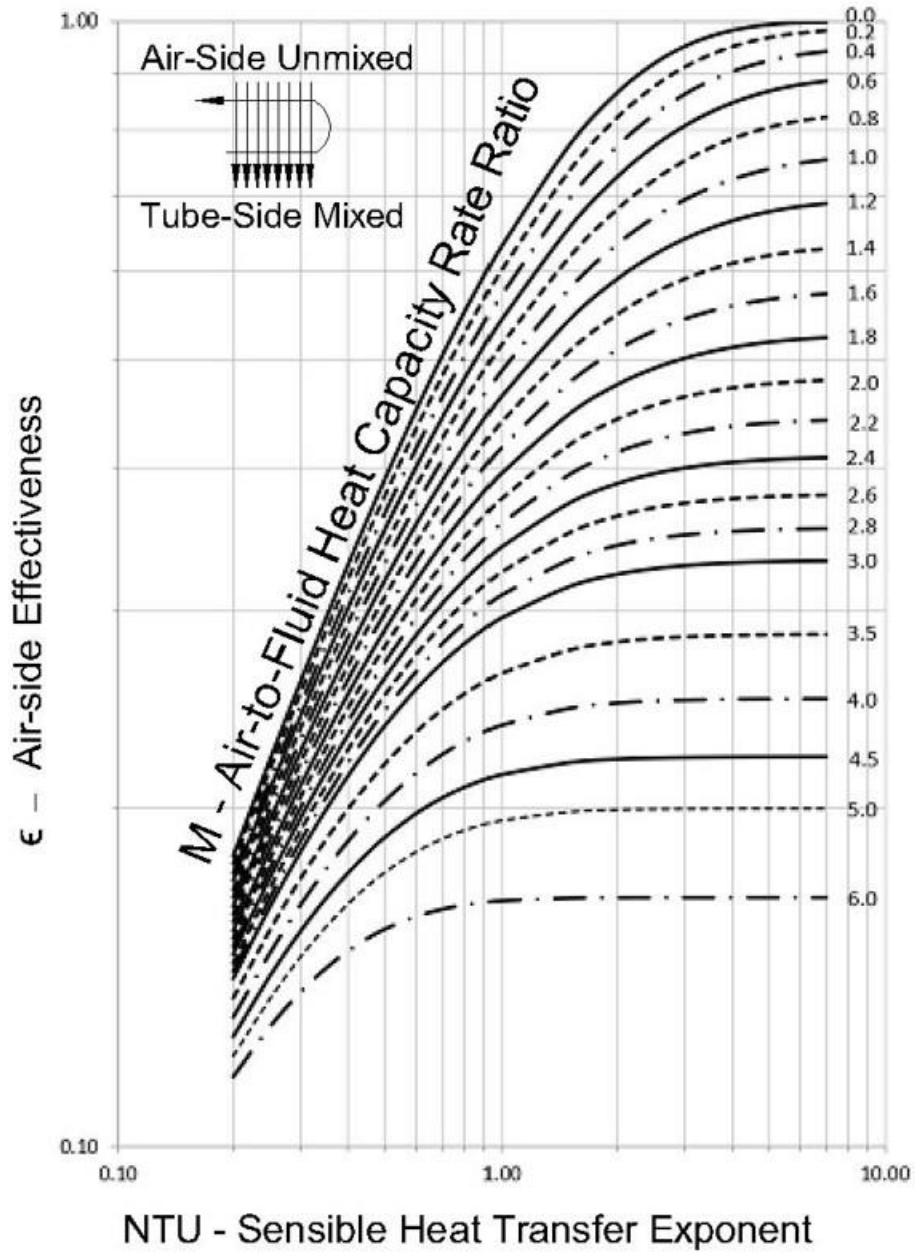**Figure 19 Crossflow Airside Effectiveness**

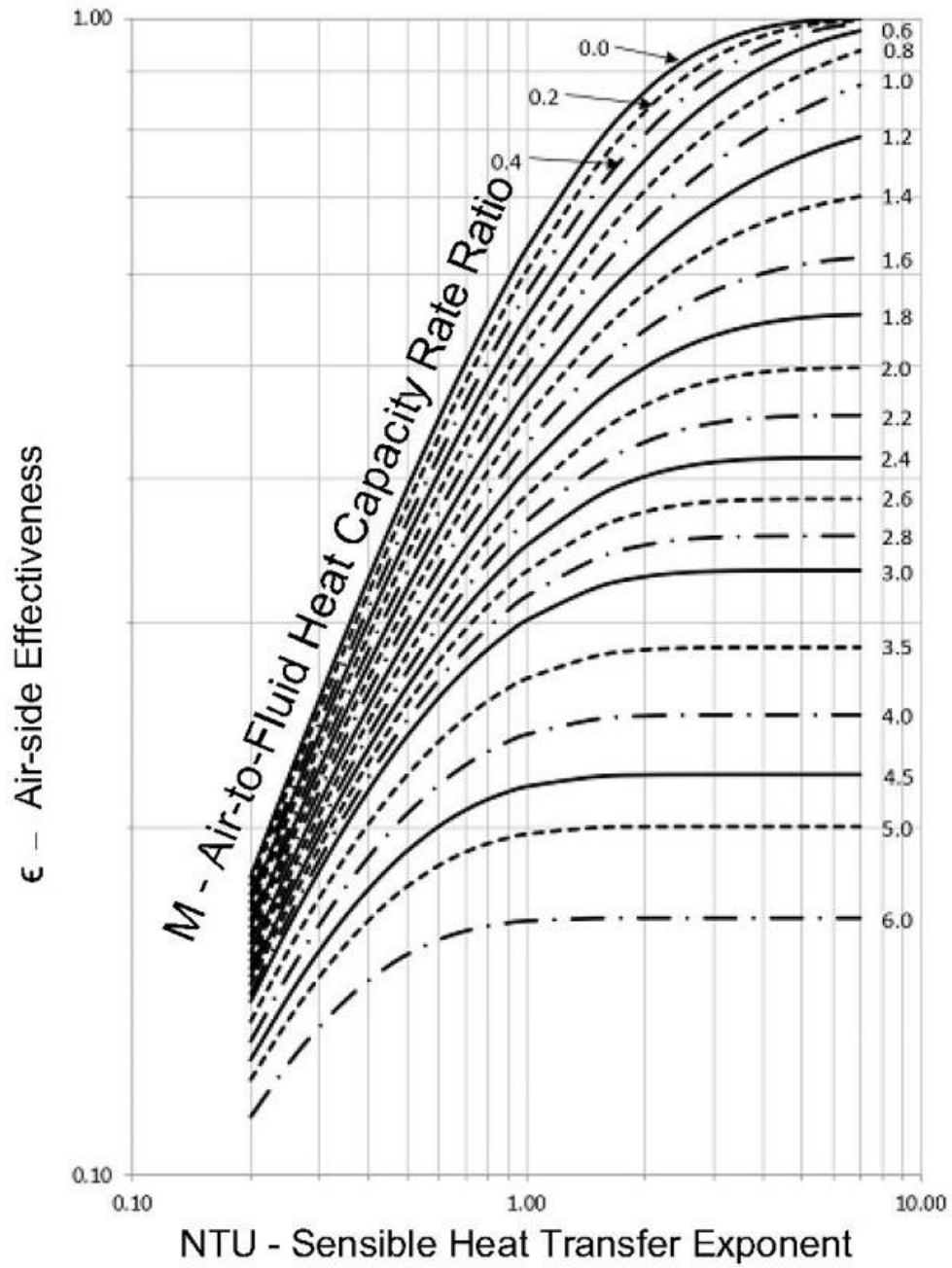**Figure 20 Thermal Two-pass Cross-counterflow Airside Effectiveness**

Figure 21 Thermal Counterflow Airside Effectiveness
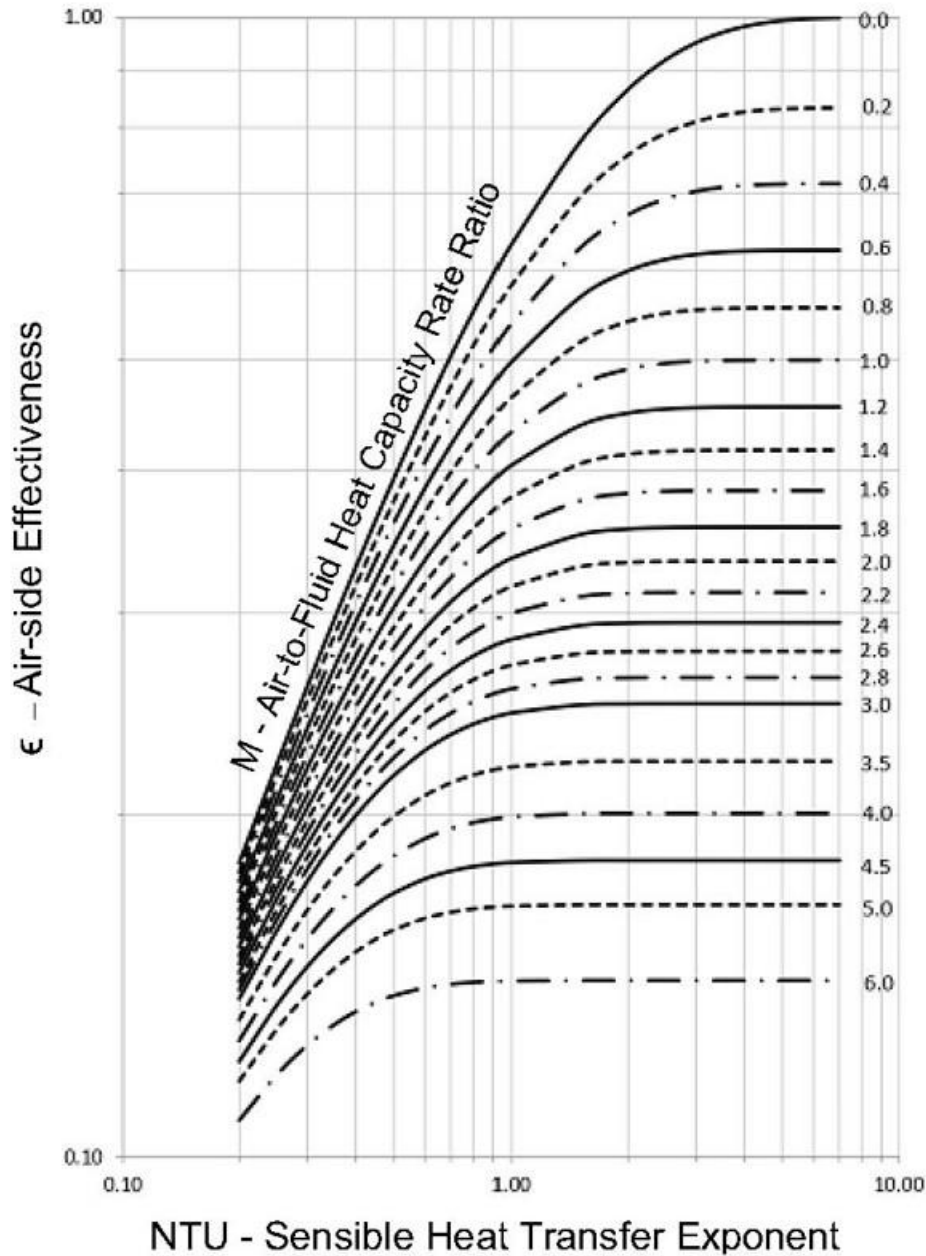
Figure 22 Thermal Parallelflow Airside Effectiveness

Section 7. Minimum Data Requirements for Published Ratings

7.1   Minimum Data Requirements for Published Ratings

As a minimum, *published ratings* shall include all the applicable information outlined below by coil type. Appendix C and Appendix D contain information on converting between actual and standard measurements. The *published ratings* shall be at standard barometric pressure, $P_{b\ STD}$, of 101.325 kPa (29.921 in Hg abs = 14.696 psia), and can be shown at other barometric pressures, providing these are clearly identified.

7.2   Single-phase Liquid Coil Ratings

1)   Manufacturer's name

2)   Model, size, or type, or all three

3)   *Published rating* name and version number/date (such as software name and version as well as the date the selection was run)

4) P_b, barometric pressure, kPa (psia), or equal elevation m (ft), or both

5)   Standard dry air volumetric flow rate, standard m^3/s (scfm) or (actual) entering air volumetric flow rate, m^3/s (acfm), or both

6)   Standard dry air face velocity, standard, m/s (standard ft/min) or (actual) entering air face velocity, m/s (fpm), or both

7)   Air pressure drop through coil at standard dry air density, kPa (in H$_2$O) (2 decimal points)

8)   Air pressure drop through coil at *job conditions* (actual air density), if applicable, kPa (in H$_2$O)

9)   Entering air dry-bulb temperature, °C (°F)

10) Entering air wet-bulb temperature (dehumidifying coils only), °C (°F)

11) Leaving air dry-bulb temperature, °C (°F)

12) Leaving air wet-bulb temperature (dehumidifying coils only), °C (°F)

13) Heating/cooling tubeside *single-phase liquid* name

14) Heating/cooling tubeside *single-phase liquid* concentration by mass (*aqueous glycol* coil only), %

15) Standard *single-phase liquid* volumetric flow rate, standard L/s (sgpm) or (actual) entering *single-phase liquid* volumetric flow rate, L/s (gpm), or both

16) Mean *single-phase liquid* velocity inside tubes, at mean *single-phase liquid* density, m/s (ft/s)

17) Entering *single-phase liquid* temperature, °C (°F)

18) Leaving *single-phase liquid* temperature, °C (°F)

19) Tubeside *single-phase liquid* fouling factor allowance, m^2·°C/W (h·ft^2·°F/Btu)

20) *Single-phase liquid* head loss through coil (including tubeside coil attachments) at mean *single-phase liquid* density, m (ft) of *single-phase liquid* (1 decimal point) or *single-phase liquid* pressure drop through coil (including tubeside coil attachments), kPa (psi) (1 decimal point), or both

21) Total heating/*cooling capacity*, W (Btu/h) or kW (MBH) (rounded to the nearest 30 W (100 Btu/h))

22) Sensible *cooling capacity*, W (Btu/h) or kW (MBH) (rounded to the nearest 30 W (100 Btu/h))

7.3   Steam Coil Ratings

1)   Manufacturer's name

2)   Model, size, or type, or both size and type

3)   *Published rating* name and version number/date (for example software name and version as well as the date the selection was run)

4) P_b, barometric pressure, psia or equal elevation ft, or both

5)   Standard dry air volumetric flow rate, standard m^3/s (scfm) or (actual) entering air volumetric flow rate, m^3/s (acfm), or both

6)   Standard dry air face velocity, standard m/s (ft/min) or (actual) entering air face velocity, m/s (fpm), or both

7)   Air pressure drop through coil at standard dry air density, Pa (in H$_2$O [2 decimal points])

8)   Air pressure drop through coil at *job conditions* (actual air density), if applicable, Pa (in H$_2$O)

9)   Entering air dry-bulb temperature, °C (°F)

10) Leaving air dry-bulb temperature, °C (°F)

11) Steam pressure at coil inlet, kPa (psig)

12) Steam superheat at coil inlet, °C (°F)

13) *Heating capacity*, W (Btu/h) or kW (MBH) (0 decimal point or 1 decimal point)

7.4 Refrigerant Evaporator Coil Ratings

1) Manufacturer's name

2) Model, size or type, or size and type

3) *Published rating* name and version number/date (for example software name and version as well as the date the selection was run)

4) P_b, barometric pressure, kPa (psia) or equal elevation, m (ft), or both

5) Standard dry air volumetric flow rate, standard m^3/s (scfm) or (actual) entering air volumetric flow rate, m^3/s (acfm), or both

6) Standard dry air face velocity, standard m/s (ft/min) or (actual) entering air face velocity, m/s (fpm), or both

7) Air pressure drop through coil at standard dry air density, Pa (in H_2O) (2 decimal points)

8) Air pressure drop through coil at *job conditions* (actual air density), if applicable, Pa (in H_2O)

9) Entering air dry-bulb temperature, °C (°F)

10) Entering air wet-bulb temperature (dehumidifying coils only), °C (°F)

11) Leaving air dry-bulb temperature, °C (°F)

12) Leaving air wet-bulb temperature (dehumidifying coils only), °C (°F)

13) *Refrigerant* name or designation

14) Saturated *refrigerant* vapor temperature leaving coil suction Header, °C (°F)

15) *Refrigerant* vapor superheat leaving coil suction Header, °C (°F)

16) Subcooled *refrigerant* liquid temperature entering liquid control device, °C (°F)

17) Total *cooling capacity*, W (Btu/h) or kW (MBH) (0 decimal or 1 decimal)

18) Sensible *cooling capacity*, W (Btu/h) or kW (MBH) (0 decimal or 1 decimal)

Section 8. Marking and Nameplate Data

As a minimum, nameplate shall display the manufacturer's name and identify designation, such as model or type.

Section 9. Conformance Conditions

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

Section 10. Symbols and Units

Symbols and units are dimensionless unless otherwise indicated.

10.1 Letter Symbols

A =   Area, m^2 (ft^2)

A_i =   Total internal surface area, m^2 (ft^2)

A_{ix} =   Total internal cross-sectional fluid flow area, m^2 (ft^2)

A_o =   Total external surface area, m^2 (ft^2)

A_p	=	Primary external surface area, m^2 (ft^2)
A_s	=	Secondary external surface area (net *fin* area), m^2 (ft^2)
B	=	Ratio of total external surface area to the total internal surface area, A_o/A_i
C	=	Coil characteristic, kg dry air. °C/kJ (lbm dry air·°F/Btu)
c	=	Leaving air dry-bulb temperature calculation exponent
c_p	=	Specific heat at constant pressure of air-water vapor mixture, $1.005 + 1.859W$ kJ/(kg dry air°C) (0.240 + 0.444W, Btu/(lbm dry air·°F)). To simplify calculation and rating procedures, a constant value of $c_p = 1.017$ (0.243) can be used for air-cooling coil calculations and a constant value of $c_p = 1.009$ (0.241) can be used for *air-heating coil* calculations.
c_{pL}	=	Specific heat at constant pressure of *single-phase liquid*, kJ/(kg·°C) [Btu/(lbm·°F)]
D	=	Diameter, mm (in)
$DPSD$	=	Dry surface air pressure drop per row depth at standard dry air density (for a rated dry air face velocity and *fin* surface as illustrated in Figure 15, Figure 16, Figure 17, and Figure 18), Pa/row (in H_2O/row)
$DPSW$	=	Wet surface air pressure drop per row depth at standard dry air density (for a rated dry air face velocity and *fin* surface as illustrated in Figure 17 and Figure 18), Pa/row (in H_2O/row)
DWL	=	Curve fit coefficients for Darcy-Weisbach friction factor inside tubes for *single-phase liquid coils* (see Table 4 for coils with *smooth tubes* and Section 6.2.3.6.1.3 for coils with *turbulators* or *internally grooved tubes*).
F_a	=	Air pressure drop correction factor
F_{LMTD}	=	LMTD correction factor to determine MTD
f	=	Heat transfer coefficient (airside is referred to total external surface area, A_o; all others are referred to total internal surface area, A_i), W/(m^2·°C) (Btu/(h·ft^2·°F))
f'_{DWL}	=	Darcy-Weisbach friction factor inside tubes for *single-phase liquid coils*
$f_a \eta$	=	Effective film coefficient, W/(m^2·°C) (Btu/(h·ft^2·°F))
G	=	Mass flux or mass velocity kg/(s·m^2) (lbm/(h·ft^2))
H	=	Coil face height , mm (in.)
h	=	Enthalpy, kJ/kg (Btu/lbm) (when applied to air, kJ/kg dry air (Btu/lbm dry air))
h_L	=	Head loss at average *single-phase liquid* density, m of *single-phase liquid* (ft of *single-phase liquid*)
Δh	=	Enthalpy difference, kJ/kg dry air (Btu/lbm dry air)
I_n	=	Modified bessel function of the first kind, order n
ITD	=	Initial air-to-tubeside fluid temperature difference, °C (°F)
j_L	=	Colburn-*j* heat transfer factor inside tubes for *single-phase liquid coils*
J	=	Curve fit coefficients for Colburn-*j* heat transfer factor inside tubes for *single-phase liquid coils* (see Table 3 for coils with *smooth tubes* and Section 6.2.3.3.1.3 for coils with *turbulators* or *internally grooved tubes*).
K_n	=	Modified bessel function of the second kind, order n
k_1	=	10^6 mm^2/m^2 (144 $in.^2/ft^2$)
k_3	=	1000 mm/m (12 in./ft)
k_5	=	conversion factor between °C and K, 273.15 (conversion factor between °F and R, 459.67)
k_7	=	1000 J/kJ (60 minutes/1 hour)
k_9	=	0.622, correction factor for relative density of air-water vapor mixture (see ASHRAE Standard 33)

k_{10}	=	1 (3600 seconds/1 hour)
k_{11}	=	0.001, conversion factor between kPa and Pa ($_{0.49116}$, conversion factor between in Hg abs and psia = (14.696 psia/29.921 in Hg abs))
k_{12}	=	Gas constant for dry air, 287.06 J/(kg dry air-K) (53.350 ft·lbf/(lbm dry air-R))
k_{13}	=	1.2 kg dry air/m³ (4.5 (lbm dry air/ft³)(s/min) = (0.075 lbm dry air/ft³)(60s/min))
k	=	Material thermal conductivity, W/(m·°C) (Btu·ft/(h·ft²·°F)) (see [Table 5](#))
k_f	=	*Fin* material thermal conductivity, W/(m·°C) (Btu·ft/(h·ft²·°F)
L	=	Length, mm (in.)
L_e	=	Total equivalent length of tube circuit, m (ft)
L_{eb}	=	Equivalent length of tube circuit per return bend, mm (in).
$LMED$	=	Logarithmic mean enthalpy difference = Δh_{lm}
$LMTD$	=	Logarithmic mean temperature difference = Δt_{lm}
m''	=	Slope of saturated air temperature-enthalpy curve at the coil surface temperature, t_s, kJ/(kg. °C (Btu/(lbm dry air·°F))
M	=	Air-to-fluid heat capacity rate ratio
MTD	=	Mean temperature difference = Δt_m
N	=	Number of
NTU	=	Number of heat transfer units = sensible heat transfer exponent
P	=	Absolute pressure, kPa abs (psia or in Hg abs)
$P_{b\ STD}$	=	Standard atmospheric pressure, kPa abs (29.921 in Hg abs)
p	=	Gage pressure, kPa gage (psi or in Hg or in H₂O)
Δp	=	Difference in pressure, kPa (psi or in H₂O)
Δp_{ATT}	=	Manufacturer established pressure drop of tubeside coil attachments (connections, *headers*, tube entrances and exits, equivalent lengths of return bends, tube bushings, tube orifices, and any other tube circuit components), psi
Pr	=	Prandtl Number
$Q_{a\ STD}$	=	Standard dry air volumetric flow rate (standard dry air density = 1.2 kg dry air/m³ (0.075 lbm dry air/ft³)) that approximates dry air density at 21.1°C (70°F) and 101.325 kPa (14.696 psia), standard m³/s (scfm)
$Q_{L\ STD}$	=	Standard *single-phase liquid* volumetric flow rate (*standard water* density = 998.9 kg/m³ (62.361 lbm/ft³)), standard L/s (sgpm)
q	=	Heat transfer capacity, W (Btu/h) (1 MBH = 1000 Btu/h)
R	=	Thermal resistance, referred to total external surface area, A_o, m². °C/W (h·ft²·°F/Btu)
Re	=	Reynolds number
s_{hf}	=	*Continuous plate fin* hole spacing across coil face, mm/*fin* hole (in./*fin* hole)
s_{hr}	=	*Continuous plate fin* hole spacing in direction of airflow, mm/*fin* hole (in./*fin* hole)
s_{tf}	=	Tube spacing across coil face, mm/tube (in./tube)
s_{tr}	=	Tube spacing in direction of airflow, mm/tube (in./tube)
St	=	Stanton Number
t	=	Temperature, °C (°F)
Δt	=	Temperature difference, °C (°F)

U	=	Overall heat transfer coefficient, referred to the total external surface area, A_o, W/(m²·°C) (Btu/(h·ft²·°F))
U_b	=	Expression for determining *fin* efficiency
U_e	=	Expression for determining *fin* efficiency
V_a	=	Standard dry air face velocity, standard m/s (standard ft/min)
V_L	=	Mean *single-phase liquid* velocity inside tubes at mean *single-phase liquid* density, m/s (ft/s)
$V_{L\,STD}$	=	Standard *single-phase liquid* velocity inside tubes at *standard water* density, standard m/s (standard ft/s)
W	=	Humidity ratio of air-water vapor mixture, kg water vapor/kg dry air (lbm water vapor/lbm dry air)
w	=	Height of equivalent annular *fin*, x_e-x_b, mm (in) (when subscript is not used), mass flow rate (when used with subscript), for air kg dry air/ s (lbm dry air/min), for *single-phase liquid*, *refrigerant* and steam – kg/s (lbm/h)
x	=	Radius of annular (spiral) or equivalent annular (non-circular) *fins*, mm (in)
x_b	=	*Fin* root radius, mm (in.)
x_e	=	*Fin* tip radius, mm (in.)
x_g	=	Concentration by mass of *aqueous glycol*, %
x_r	=	Mass fraction of *refrigerant* vapor
Y	=	*Fin* thickness, mm (in)
y	=	Ratio of fluid temperature rise to air enthalpy drop, kg dry air. °C/kJ (lbm dry air·°F/Btu)
Z	=	Altitude relative to sea level for barometric pressure, m (ft)
β	=	Expression for determining *fin* efficiency
ε	=	Airside effectiveness
η	=	Total external surface effectiveness
ϕ	=	*Fin* efficiency
ρ	=	Density, kg/m³ (lbm/ft³)
μ	=	Absolute or dynamic viscosity at mean bulk temperature, mPa-s (lbm/(ft·h))
μ_{tw}	=	Absolute or dynamic viscosity at mean internal tube wall temperature, mPa-s (lbm/(ft·h))
υ	=	Specific volume, m³/kg (ft³/lbm)

10.2   Subscripts

10.2.1   Numerical subscripts:

0	=	Refers to conditions entering liquid control device
1	=	Refers to conditions entering coil
2	=	Refers to conditions leaving coil

10.2.2   Letter Subscripts Used to Identify Letter Symbols

a	=	Airside
B	=	Conditions at dry-wet boundary
b	=	Barometric (when used with symbol, *P*)
	=	*Fin* root of annular (spiral) or equal annular (non-circular) *fins* (when used with symbol, *x*)
	=	Return bends (when used with symbol, *N*)

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c	=	Assumed value (when used with symbol, *A*)
	=	Plate *fin* external collar (when used with symbol, *L*)
	=	Tube circuits (when used with symbol, *N*)
D	=	Dry surface
d	=	Plate *fin* depth in direction of airflow (when used with symbol, *L*)
	=	At outer edge of *spiral fin* at D_f (when used with symbol, *Y*)
db	=	Dry-bulb
dp	=	*Dew point* (when used with symbol, *t*)
	=	Saturated air at t_{1dp} (when used with symbol, *h*)
e	=	Outside edge of *spiral fin* (when used with symbol, *Y*)
	=	Outside, or equal annular area of non-circular *fin* or of annular or *spiral fin* (when used with symbol, *x*)
f	=	Across Coil face (when used with symbol, *s*)
	=	Coil face (when used with symbol, *A*)
	=	Outside diameter of *spiral fin* (when used with symbol, *D*)
	=	*Fins* in net finned tube length (when used with symbol, *N*)
	=	Plate *fin* length perpendicular to direction of tubes exposed to the airflow (when used with symbol, *L*)
	=	Plate *fins* of constant thickness (when used with symbol, *Y*)
	=	Saturated liquid (when used with symbols, h_r and t_r)
ffa	=	Tubeside fouling factor allowance for *single-phase liquid coil* ratings
g	=	Saturated vapor (when used with symbols h_r, t_r, v_r, t_v, and v_v)
h	=	Holes in plate *fin*
i	=	Inside tube
ih	=	Intermediate *headers*
JOB	=	*Job conditions* (actual air pressure and density)
lm	=	Logarithmic mean difference (when used with Δh or Δt)
L	=	*Single-phase liquid* (such as water, *aqueous gylcol*)
m	=	Mean or average (can be combined with other subscripts)
	=	Mean temperature difference when used with Δt
	=	Metal (when used with symbol, *R*)
n	=	*Crimped spiral fin* neutral diameter (when used with symbol, *D*)
	=	*Crimped spiral fin* thickness at neutral diameter (when used with symbol, *Y*)
	=	Modified bessel function order
o	=	Outside tube (when used with symbol, *D*)
p	=	*Tube passes* per tube circuit
r	=	Coil rows in direction of airflow (when used with symbols, *N*)
	=	*Fin* root for *spiral fins* with constant metal area for heat flow (when used with symbol, *Y*)
	=	*Refrigerant* (all other symbols)
rc	=	*Refrigerant evaporator coil* circuit
rh	=	*Refrigerant evaporator coil* suction header

s	=	Coil surface (when used with symbol, t)
	=	Saturated air at coil surface temperature, t_s, (when used with symbol, h)
	=	Sensible (when used with symbol, q)
	=	Straight tube (when used with symbol, L)
	=	Static (when used with symbol, P)
$\bar{s}$	=	Effective coil surface (when used with symbol, t)
	=	Saturated air at effective coil surface temperature, $t_{\bar{s}}$ (when used with symbol, h)
sD	=	Isothermal dry surface airside, at standard dry air density
sW	=	Wet surface airside, at standard dry air density
t	=	Net finned tube (when used with symbol, L)
	=	Total (when used with symbol, q)
	=	Tube (when used with symbols h_L, k, R, N and s)
tw	=	Tube wall
v	=	Steam
W	=	Wet surface
w	=	Water
wb	=	Wet-bulb
x	=	Cross section

Where letter subscript does not follow c_p, h or t these symbols designate air-water vapor mixture properties.

10.3 Superscripts

The power that a number or symbol is raised is indicated by numerical superscripts.

Section 11. Reference Information

11.1 Reference Properties

The properties of fluids shall be obtained from the sources listed below:

11.1.1 Air

The thermodynamic properties of air-water vapor mixture shall be obtained from the latest edition of the *ASHRAE Handbook – Fundamentals*.

11.1.2 Water and Steam

The thermodynamic properties of water and steam shall be obtained from the latest edition of *Steam Tables: Thermodynamic Properties of Water Including Vapor, Liquid, and Steam*. The thermophysical properties of water shall be obtained from the latest edition of the *ASHRAE Handbook – Fundamentals*.

11.1.3 Aqueous Glycol

The thermodynamic and thermophysical properties of *aqueous glycol* shall be obtained from the latest edition of the *ASHRAE Handbook – Fundamentals* or the glycol manufacturer.

11.1.4 Refrigerants

The thermodynamic properties of *refrigerants* shall be obtained from the latest edition of the *ASHRAE Handbook – Fundamentals*, NIST REFPROP, or the *refrigerant* manufacturer.

11.2 Conversion Factors

For conversion factors between I-P and SI units of measure, see Table 1 of ANSI/ASHRAE Standard 33-2016 and Table 6 in this standard.

Table 6 Conversion Factors

Item	I-P	SI	Conversion Factor SI x Factor =I-P	Conversion Factor I-P x Factor = SI
Absolute or Dynamic Viscosity	lbf/(h·ft)	mPa·s	2.4191	0.41338
Density	lbm/ft^3	kg/m^3	16.0184	0.06243
Heat Transfer Capacity	Btu/h	kW	3412.142	0.000293
Heat Transfer Coefficient	Btu/(h·ft^2·°F)	W/(m^2·°C)	0.1762	5.678
Mass Flow Rate	lbm/h	kg/s	7936.647	0.000126
Mass Velocity	lbm/(h·ft^2)	kg/(m^2·s)	737.354	0.0013562
Pressure	In H$_2$O	Pa	0.00402	248.84
Pressure	In Hg	kPa	0.2953	3.38639
Pressure	psi	kPa	0.14503	6.8948
Pressure Drop Parameter	lb^2/(in^2·ft^4)	kPa·kg/m^4	0.002760	362.35
Temperature	°F	°C	°C x 9/5 + 32	(°F - 32) x 5/9
Thermal Conductivity	Btu·ft/(h·ft^2·°F)	W/(m·°C)	0.5778	1.7307
Thermal Resistance	h·ft^2·°F/Btu	m^2·°C/W	5.6782	0.17611
Velocity	fpm	m/s	196.850	0.0051
Velocity	ft/s	m/s	3.2808	0.3048
Volumetric Flow Rate	cfm	m^3/s	0.000472	2118.88
Liquid Flow Rate	gpm	L/s	15.85	0.06309

11.3   Significant Figures

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

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

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

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

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

APPENDIX A. REFERENCES – NORMATIVE

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

A.1.   ANSI/ASHRAE Standard 33-2016, *Methods of Testing Forced Circulation Air Cooling and Air Heating Coils*, 2015, ASHRAE, 180 Technology Parkway NW, Peachtree Corners, U.S.A.

A.2.   AHRI Guideline E-1997, *Fouling Factors: A Survey of Their Application in Today's Air-Conditioning and Refrigeration Industry*, 1997, Air Conditioning, Heating and Refrigeration Institute, 2111 Wilson Blvd., Suite 500, Arlington, VA 22201, U.S.A.

A.3.   *ASHRAE Handbook – Fundamentals*, Chapter 1 Psychrometrics, Chapter 30 Thermophysical Properties of Refrigerants, and Chapter 31 Physical Properties of Secondary Coolants (Brines)*, 2017, ASHRAE, 180 Technology Parkway NW, Peachtree Corners, U.S.A.

A.4.   ASHRAE Terminology. ASHRAE. Accessed November 10, 2022. https://www.ashrae.org/technicalresources/free-resources/ashrae-terminology.

A.5.   Blast Coil Manufacturers Institute, *Proposed Commercial Standard for Rating and Testing Air Cooling Coils Using Non-Volatile Refrigerant*, 1945, BCMI Code TS-4044.

A.6.   Brown, Gosta, *Theory of Moist Air Heat Exchangers*, 1954, Transactions of the Royal Institute of Technology, Stockholm, Sweden, Pages 12-15, Nr 77.

A.7.   Carrier, Willis H. & Anderson Samuel W., *The Resistance to Heat Flow Through Finned Tubing*, 1944, ASHVE Transactions, Pages 117-152, Vol. 50, ASHRAE, 180 Technology Parkway NW, Peachtree Corners, U.S.A.

A.8.   Copper Development Association, Inc., *Standards Handbook, Part 2-Alloy Data*, 1985, 260 Madison Ave., New York, NY 10016, U.S.A.

A.9.   Gardner, Karl A., *Efficiency of Extended Surface*, 1945, ASME Transactions, Pages 621-631, Vol. 67, American Society of Mechanical Engineers, Three Park Ave., New York, NY 10016, U.S.A.

A.10.   International Nickel Co. Inc., *Properties of Some Metals and Alloys*, 1982, Suffern, NY 10901, U.S.A.

A.11.   Kays, William M., London, Alexander Louis, & Johnson D.W., *Gas Turbine Plant Heat Exchangers*, 1951, American Society of Mechanical Engineers, Three Park Ave., New York, NY 10016, U.S.A.

A.12.   Keenan, J.H., Keyes, F.G., Hill, P.G., & Moore, J.G., *Thermodynamic Properties of Water Including Vapor, Liquid, and Solid Phase*s, 1969 I-P version, John Wiley & Sons, Inc.

A.13.   Keenan, J.H., Keyes, F.G., Hill, P.G., & Moore, J.G., *Thermodynamic Properties of Water Including Vapor, Liquid, and Solid Phases*, 1978 SI version, John Wiley & Sons, Inc.

A.14.   McElgin, John & Wiley, D.C., *Calculation of Coil Surface Areas for Air Cooling and Dehumidification*, March 1940, Heating, Piping and Air Conditioning, Pages 195-201.

A.15.   Rich, Donald G., *The Efficiency and Thermal Resistance of Annular Fins*, 1966, Proceedings of the Third International Heat Transfer Conference, Vol. III, Pages 281-289, American Institute of Chemical Engineers, 345 East 47th Street, New York, NY 10017, U.S.A.

A.16.   Stevens, R.A., Fernandez, J., & Woolf, J.R., *Mean Temperature Difference in One-, Two- and Three-Pass Crossflow Heat Exchangers*, 1957, ASME Transactions, Pages 287-297, Vol. 79, American Society of Mechanical Engineers, Three Park Ave., New York, NY 10016, U.S.A.

A.17.   Ware, Chester D. & Hacha, Thomas H., *Heat Transfer From Humid Air to Fin and Tube Extended Surface Cooling Coils*, 1960, ASME Paper No. 60-HT-17, American Society of Mechanical Engineers, Three Park Ave., New York, NY 10016, U.S.A. Tables.

APPENDIX B. REFERENCES – INFORMATIVE

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

B.1. *ASHRAE Handbook – HVAC Systems and Equipment*, Chapter 23 Air-cooling and Dehumidifying Coils and Chapter 27 Air-Heating Coils, 2020, ASHRAE, 180 Technology Parkway NW, Peachtree Corners, U.S.A.

B.2. Anderson, Samuel W., *Air-cooling and Dehumidifying Coil Performance Based on ARI Industrial Standard 410-64*, Papers Presented at the Symposium on Heat and Mass Transfer To Extended Surfaces at the Semiannual Meeting of ASHRAE January 27-30, 1969, Chicago IL, 1970, Pages 22-28, ASHRAE, 180 Technology Parkway NW, Peachtree Corners, U.S.A.

B.3. Goodman, William., *Performance of Coils for Dehumidifying Air*, 1938 & 1939, Heating, Piping and Air Conditioning, Vol. 10 (Nov.-Dec. 1938), Vol. 11 (Jan.-May 1939).

B.4. Wile, D.D., *Air Cooling Coil Performance*, July 1953, Refrigerating Engineering, Pages 727-732, 794, 796, Vol. 61. ASHRAE, 180 Technology Parkway NW, Peachtree Corners, U.S.A.

B.5. The Aluminum Association. "The Aluminum Association," 2021. https://www.aluminum.org/.

B.6. Copper Development Association Inc. "Copper Development Association Inc.," 2023. https://www.copper.org/.

APPENDIX C. VOLUMETRIC FLOW RATES OF AIR AND SINGLE-PHASE LIQUID – NORMATIVE

This appendix provides the equations necessary for volumetric flow rates of air and *single-phase liquid*.

The entering air-water vapor mixture volumetric flow rate, $Q_{a1\,Air\text{-}Water\,Vapor\,Mixture}$, is calculated by Equation 109.

$$Q_{a1\,Air-Water\,Vapor\,Mixture} = Q_{a\,STD\,Dry\,Air}\frac{\rho_{a\,STD}}{\rho_{a1\,Dry\,Air}} \qquad\qquad 109$$

The entering *single-phase liquid* volumetric flow rate, Q_{L1}, is calculated by Equation 110.

$$Q_{L1} = Q_{L\,STD}\frac{\rho_{L\,STD}}{\rho_{L1}} \qquad\qquad 110$$

The leaving *single-phase liquid* volumetric flow rate, Q_{L2}, is calculated by Equation 111.

$$Q_{L2} = Q_{L\,STD}\frac{\rho_{L\,STD}}{\rho_{L2}} \qquad\qquad 111$$

APPENDIX D. ASSUMPTIONS, CLASSIFICATIONS, DIMENSIONS, FIN GEOMETRY, FIN EFFICIENCY, AND COIL GEOMETRY – NORMATIVE

D.1. Assumptions

The bulk model equation set included in this standard provides for idealized calculation methods for sizing and rating coils. The calculation equations used throughout the standard shall follow the assumptions below unless otherwise specified by the manufacturer.

D.1.1. Coil Physical Characteristics

D.1.1.1. Tube Circuits

All tube circuits have the same number of identical tubes and the same flow length.

D.1.1.2. Tube Circuiting

All tube circuits have the same tube-to-tube pattern in relation to the airflow.

D.1.1.3. Tube Arrangement

The row-to-row tube layout, referenced to the tube ends, is repeatable such that addition of an additional row continues a regular pattern of center-to-center tube distances between respective tubes in adjacent rows and respective tubes in the same row.

D.1.1.4. Fin Profile

The fins are of identical patterns, identical thickness and spaced uniformly.

D.2. Fluid Flow Characteristics

D.2.1.1. Air Dry-Bulb Temperature Profile

The air dry-bulb temperature at any location on the inlet (air on) face of the coil is equal to that at any other respective location on the inlet face.

D.2.1.2. Air Face Velocity Profile

The air face velocity at any location on the inlet (air on) face of the coil is equal to that at any other respective location on the inlet face.

D.2.1.3. Air Wet-Bulb Temperature Profile

The air wet-bulb temperature at any location on the inlet (air on) face of the coil is equal to that at any other respective location on the inlet face.

D.2.1.4. Fluid Mass Flow Rate in All Tube Circuits

The fluid mass flow rate inside any tube circuit is equal to that of any other respective tube circuit. Ideally, this results in identical fluid velocity in all circuits.

D.2.1.5. Fluid Inlet Temperature in All Tube Circuits

The fluid inlet temperature inside any tube circuit is equal to that of any other respective tube circuit.

D.3. Coil Classifications by Tube Arrangement and Fin Configuration

D.3.1.1. Staggered Tubes with:

1) Continuous flat plate fins

2) Continuous configured plate fins

3) Crimped spiral fins

4) Smooth spiral fins

5)   Flat plate fins on individually-finned tube

6)   Configured plate fins on individually-finned tube
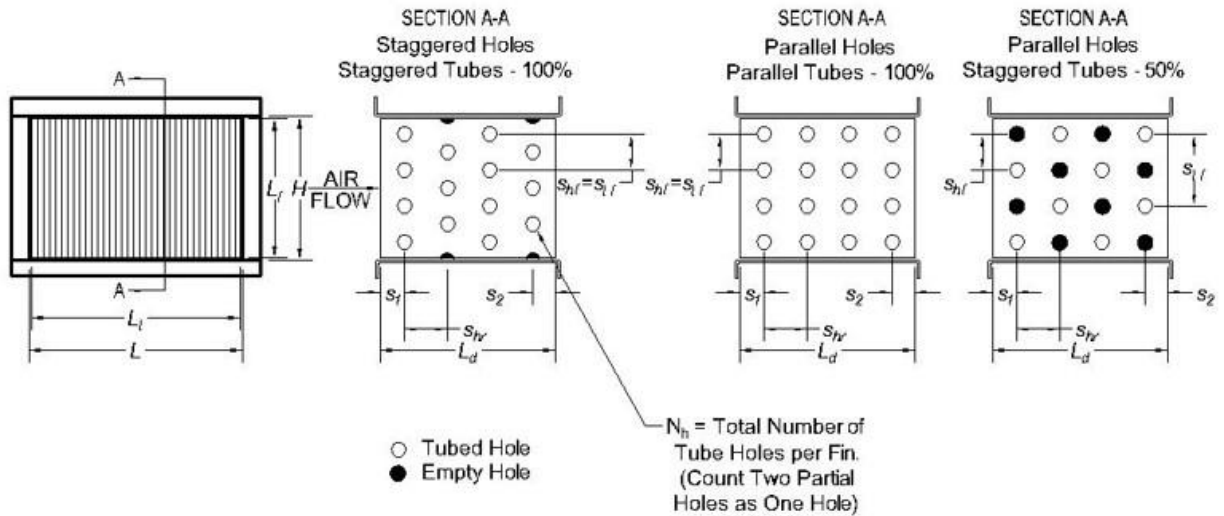
D.3.1.2.   Parallel (In-line) Tubes with:

1)   Continuous *flat plate fins*

2)   Continuous *configured plate fins*

3)   *Crimped spiral fins*

4)   *Smooth spiral fins*

5)   *Flat plate fins* on individually-finned tube

6)   *Configured plate fins* on individually-finned tube

D.4.   Coil Assembly Dimensions

In [Figure 23](#), [Figure 24,](#) and [Figure 25,](#) H applies as shown whether channel flanges are turned inward or outward. Where an option is offered in the measurement of any dimension, the same basis shall be used to determine rating data as used in the evaluation of test results.

Dimensions L_f and L_d for a *configured plate fin* are determined, at the option of the manufacturer, from the blank *fin* sheet size prior to forming the configuration that does not provide edge trimming is performed after forming or from the finished *fin* size after forming.

Staggered tubes and parallel (in-line) tubes with continuous *flat plate fins* or continuous *configured plate fins* are shown in [Figure 23](#).
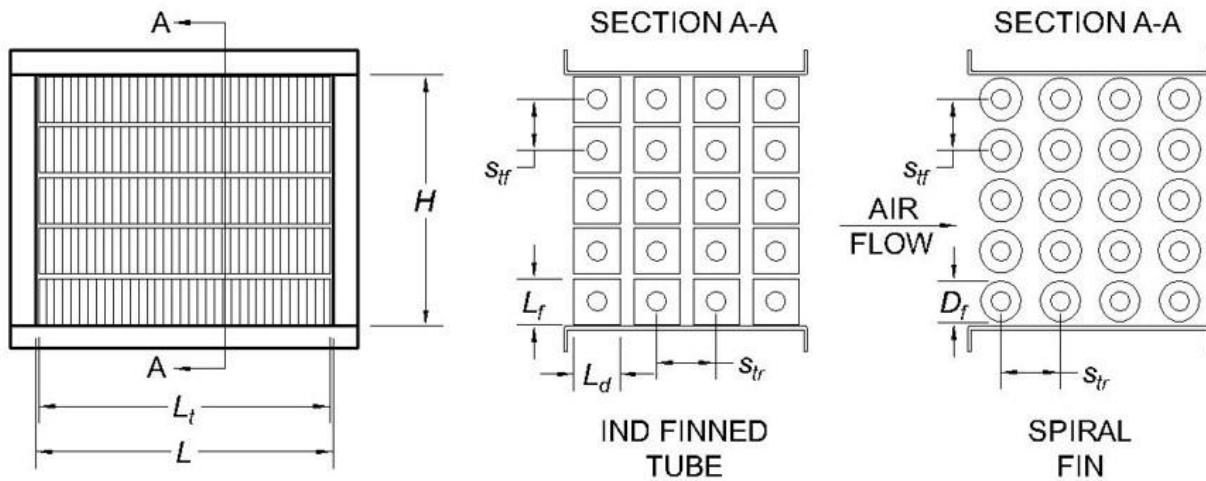

**Figure 23 Staggered Tubes and Parallel (In-line) Tubes with
Continuous Flat Plate Fins or Continuous Configured Plate Fins**

Staggered tubes with *smooth spiral fins* or *crimped spiral fins* or with *flat plate fins* or *configured plate fins* on individually-finned tube are shown in [Figure 24](#) . Air baffles shown are optional and *H* can be the distance between channels as shown in [Figure 23.](#)
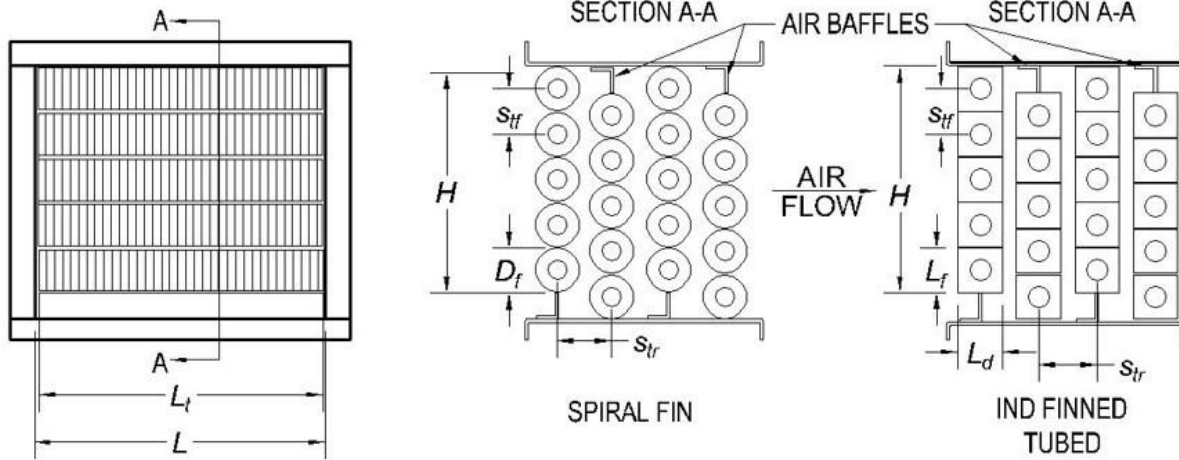
Figure 24 Staggered Tubes with Smooth Spiral Fins or Crimped Spiral Fins or with Flat Plate Fins or Configured Plate Fins on Individually-finned Tube

Parallel (in-line) tubes with smooth or *crimped spiral fins* or with *flat plate fins* or *configured plate fins* on individually-finned tube are shown in Figure 25.

Figure 25 Parallel (In-line) Tubes with Smooth Spiral Fins or Crimped Spiral Fins or with Flat Plate Fins or Configured Plate Fins on Individually-finned Tube

D.5.   Fin-Tube Assembly Dimensions and Methods for Determining Fin Efficiency
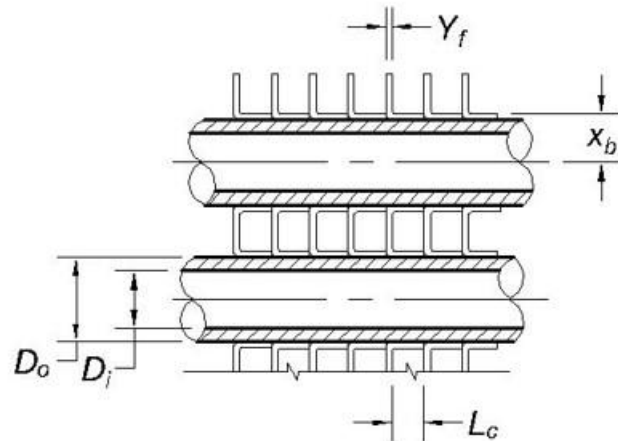
Fin efficiency shall be used to determine variable *fin* metal thermal resistance as detailed in Section 6.3

D.5.1.1.   Plate Fins of Constant Thickness

D.5.1.1.1.   Fin-Tube Assembly Dimensions and Equivalent Annular Fin Geometry

Plate *fins* with collars touching adjacent *fins* are shown in Figure 26.

Figure 26 Plate Fins with Collars Touching Adjacent Fins

To calculate the equivalent annular *fin* outside radius from tube centerline, x_e, and the equivalent annular *fin* root radius from centerline, x_b, for *continuous plate fins* with collars touching adjacent *fins*, use dimensions from Figure 23 and Figure 26 in Equation 112 and Equation 113.
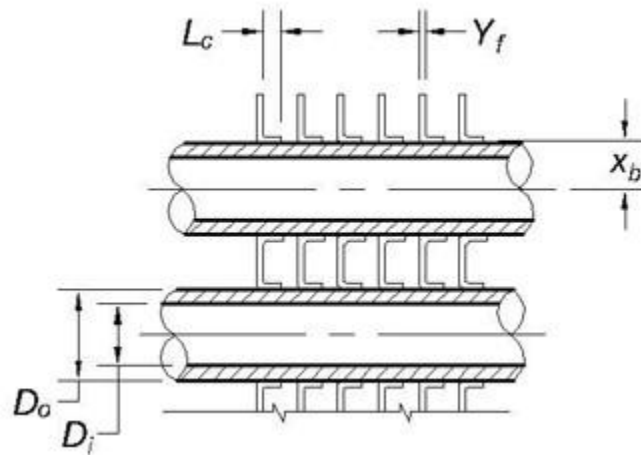
$$x_e = \left(\frac{L_f L_d}{\pi N_t}\right)^{0.5}$$ 112

$$x_b = \frac{D_o + 2Y_f}{2}$$ 113

To calculate the equivalent annular *fin* outside radius from tube centerline, x_e, and the equivalent annular *fin* root radius from centerline, x_b, for plate *fins* with collars touching adjacent *fins* on individually-finned tube, use dimensions from Figure 24 and Figure 25 or and Figure 26 in Equation 114 and Equation 113.

$$x_e = \left(\frac{L_f L_d}{\pi}\right)^{0.5}$$ 114

$$x_b = \frac{D_o + 2Y_f}{2}$$ 113

Plate *fins* with collars not touching adjacent *fins* are shown in Figure 27.

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Figure 27 Plate Fins with Collars not Touching Adjacent Fins

To calculate the equivalent annular *fin* outside radius from tube centerline, x_e, and the equivalent annular *fin* root radius from centerline, x_b, for *continuous plate fins* with collars not touching adjacent *fins*, use dimensions from Figure 23 and Figure 27 in Equation 112 and Equation 115.

$$x_e = \left(\frac{L_f L_d}{\pi N_t}\right)^{0.5} \hspace{6cm} 112$$
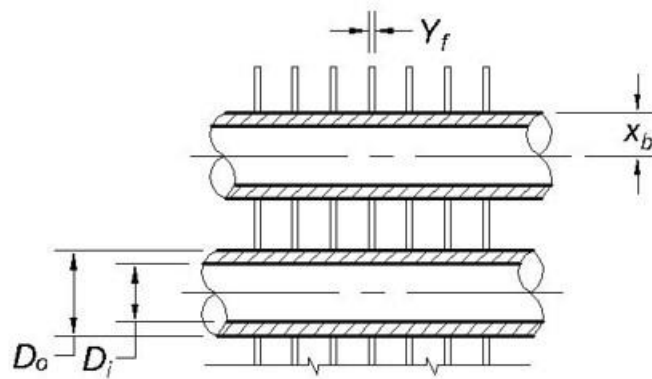
$$x_b = \frac{D_o + Y_f}{2} \hspace{6cm} 115$$

To calculate the equivalent annular *fin* outside radius from tube centerline, x_e, and the equivalent annular *fin* root radius from centerline, x_b, for plate *fins* with collars not touching adjacent *fins* on individually-finned tube, use dimensions from Figure 24 or Figure 25 and Figure 27 in Equation 114 and Equation 115.

$$x_e = \left(\frac{L_f L_d}{\pi}\right)^{0.5} \hspace{6cm} 114$$

$$x_b = \frac{D_o + Y_f}{2} \hspace{6cm} 115$$

Plate *fins* without collars are shown in Figure 28.

Figure 28 Plate Fins without Collars

To calculate the equivalent annular *fin* outside radius from tube centerline, x_e, and the equivalent annular *fin* root radius from centerline, x_b, for *continuous plate fins* without collars, use dimensions from Figure 23 and Figure 28 in Equation 112 and Equation 116.

$$x_e = \left(\frac{L_f L_d}{\pi N_t}\right)^{0.5} \qquad\qquad 112$$

$$x_b = \frac{D_o}{2} \qquad\qquad 116$$

To calculate the equivalent annular *fin* outside radius from tube centerline, x_e, and the equivalent annular *fin* root radius from centerline, x_b, for plate *fins* without collars on individually-finned tube, use dimensions from Figure 24 and Figure 28 in Equation 114 and Equation 116

$$x_e = \left(\frac{L_f L_d}{\pi}\right)^{0.5} \qquad\qquad 114$$

$$x_b = \frac{D_o}{2} \qquad\qquad 116$$

D.5.1.1.2. Fin Efficiency of Continuous Plate Fins and Plate Fins on Individually-Finned Tube (Efficiency of Equivalent Annular Fins of Constant Thickness)

Fin efficiency, ϕ, shall be determined mathematically or graphically.

D.5.1.1.2.1. Mathematical Solution

Calculate *fin* efficiency using the equivalent annular *fin* geometry equations corresponding to fin-tube assembly and Equation 117 through Equation 121.

$$w = x_e - x_b \qquad\qquad 117$$

$$U_b = \frac{w\sqrt{2f_a/\left(k_3 k_f Y_f\right)}}{(x_e/x_b - 1)}$$

<div align="right">118</div>

$$U_e = U_b(x_e/x_b)$$

<div align="right">119</div>

$$\beta_1 = \frac{I_1(U_e)}{K_1(U_e)}$$

<div align="right">120</div>

$$\phi = \frac{2}{U_b(1 - (U_e/U_b)^2)}\left(\frac{I_1(U_b) - \beta_1 K_1(U_b)}{I_0(U_b) + \beta_1 K_0(U_b)}\right)$$

<div align="right">121</div>

D.5.1.1.2.2. Graphical Solution

Determine values for Equations 117, 122 and 123 using the equivalent annular *fin* geometry equations corresponding to fin-tube assembly. Using the values from Equation 122 and Equation 123 as input, graphically determine the *fin* efficiency from Figure 29.

$$w = x_e - x_b$$

<div align="right">117</div>

$$\textit{Fin Efficiency X-Axis Parameter} = w\sqrt{2f_a/\left(k_3 k_f Y_f\right)}$$

<div align="right">122</div>

$$\textit{Fin Efficiency Fin Tip Radius/Fin Root Radius Ratio} = x_e/x_b$$
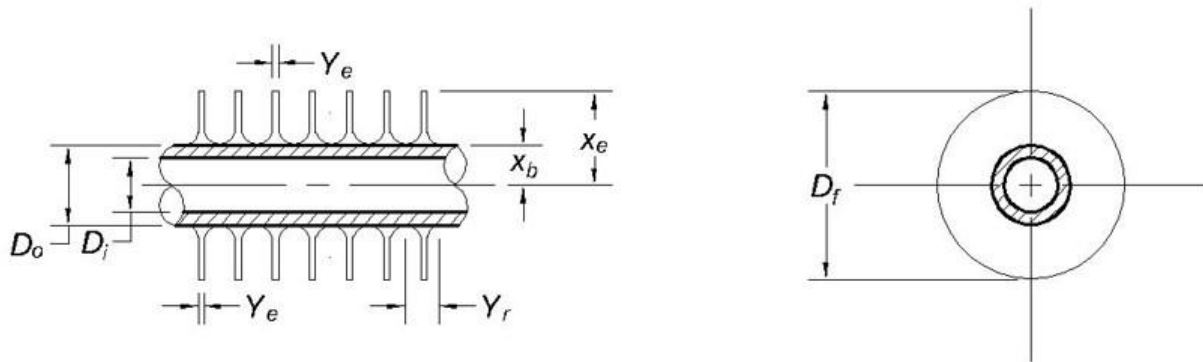
<div align="right">123</div>
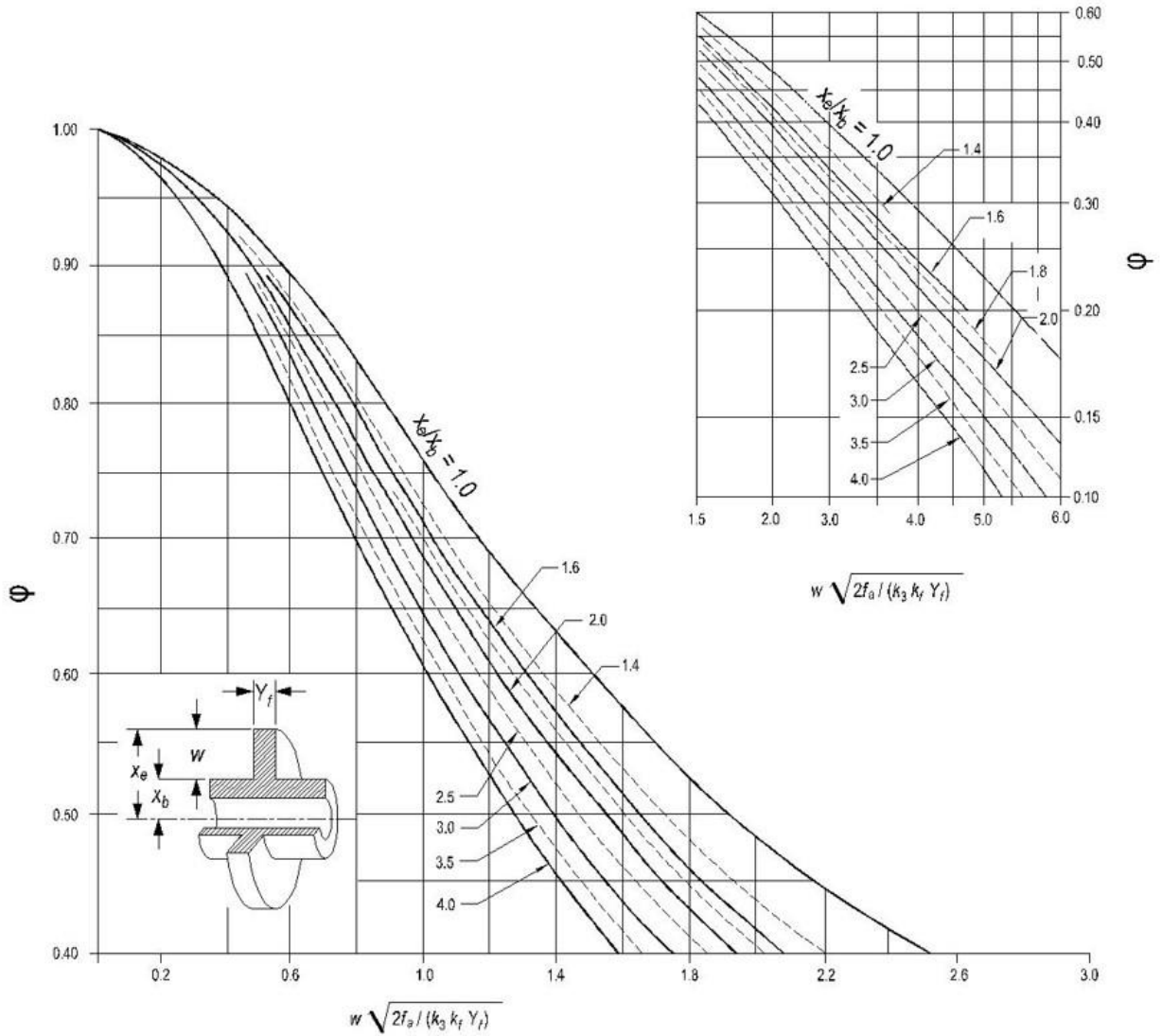
**Figure 29 Efficiency of Annular Fins of Constant Thickness
(Plate Fins)**

D.5.1.2.   Spiral Fins of Constant Cross-Sectional Area for Heat Flow

D.5.1.2.1.   Fin-Tube Assembly Dimensions and Annular Fin Geometry

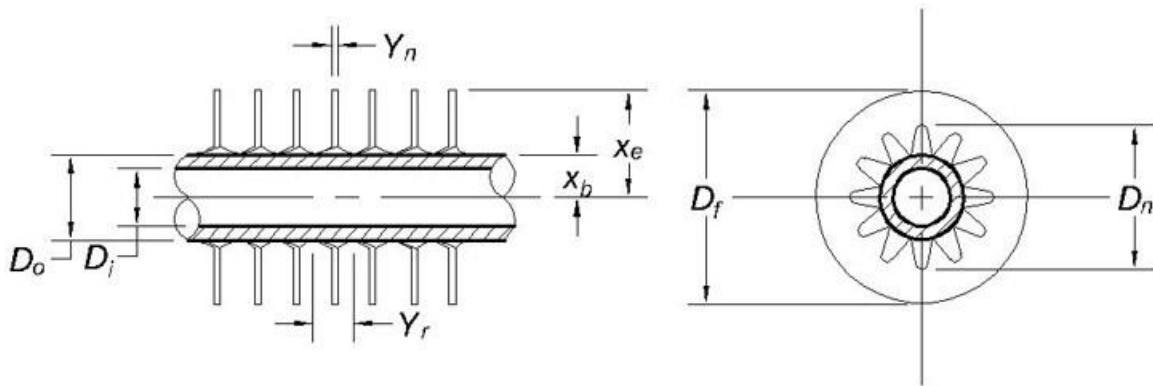*Smooth spiral fins* are shown in Figure 30.

Figure 30 Smooth Spiral Fins

To calculate the annular *fin* outside radius from tube centerline, x_e, and the annular *fin* root radius from centerline, x_b, and the *fin* thickness at root, Y_r, for *smooth spiral fins*, use dimensions from Figure 30 and Equations 124, 125 and 126.

$$x_e = \frac{D_f}{2} \tag{124}$$

$$x_b = \frac{D_o}{2} \tag{125}$$

$$Y_r = \frac{Y_e D_f}{D_o} \tag{126}$$

Crimped spiral fins are shown in Figure 31.

Figure 31 Crimped Spiral Fins

To calculate the annular *fin* outside radius from tube centerline, x_e, the annular *fin* root radius from centerline, x_b, and the *fin* thickness at root, Y_r, for *crimped spiral fins*, use dimensions as shown on Figure 31 and Equations 127, 128, and 129.

$$x_e = \frac{D_f}{2} \qquad\qquad 127$$

$$x_b = \frac{D_o}{2} \qquad\qquad 128$$

$$Y_r = \frac{Y_n D_n}{D_o} \qquad\qquad 129$$

D.5.1.2.2. **Fin Efficiency of Smooth Spiral Fins and Crimped Spiral Fins (Annular Fins of Constant Cross-Sectional Area for Heat Flow)**

Fin efficiency, ϕ, shall be determined mathematically or graphically.

D.5.1.2.2.1. Mathematical Solution

Calculate *fin* efficiency using the annular *fin* geometry equations corresponding to fin-tube assembly and Equation 117 and Equation 130 through Equation 133.

$$w = x_e - x_b \qquad\qquad 117$$

$$U_b = \frac{2w\sqrt{2f_a/(k_3 k_f Y_r)}}{3(x_e/x_b - 1)} \qquad\qquad 130$$

$$U_e = U_b(x_e/x_b)^{3/2} \qquad\qquad 131$$

$$\beta = \frac{-I_{-2/3}(U_e)}{I_{2/3}(U_e)} \qquad\qquad 132$$

$$\phi = \frac{4}{3U_b(1 - (U_e/U_b)^{4/3})}\left(\frac{I_{-2/3}(U_b) + \beta I_{2/3}(U_b)}{I_{1/3}(U_b) + \beta I_{-1/3}(U_b)}\right) \qquad 133$$

D.5.1.2.2.2. Graphical Solution

Determine values for Equation 117, Equation 134, and Equation 123 using the annular *fin* geometry equations corresponding to fin-tube assembly. Using the values from Equation 134 and Equation 123 as input, graphically determine the *fin* efficiency from Figure 32.

$$w = x_e - x_b \qquad\qquad 117$$

$$\textit{Fin Efficiency X-Axis Parameter} = w\sqrt{2f_a/(k_3 k_f Y_r)} \qquad 134$$

$$\textit{Fin Efficiency Fin Tip Radius/Fin Root Radius Ratio} = x_e/x_b \qquad 123$$
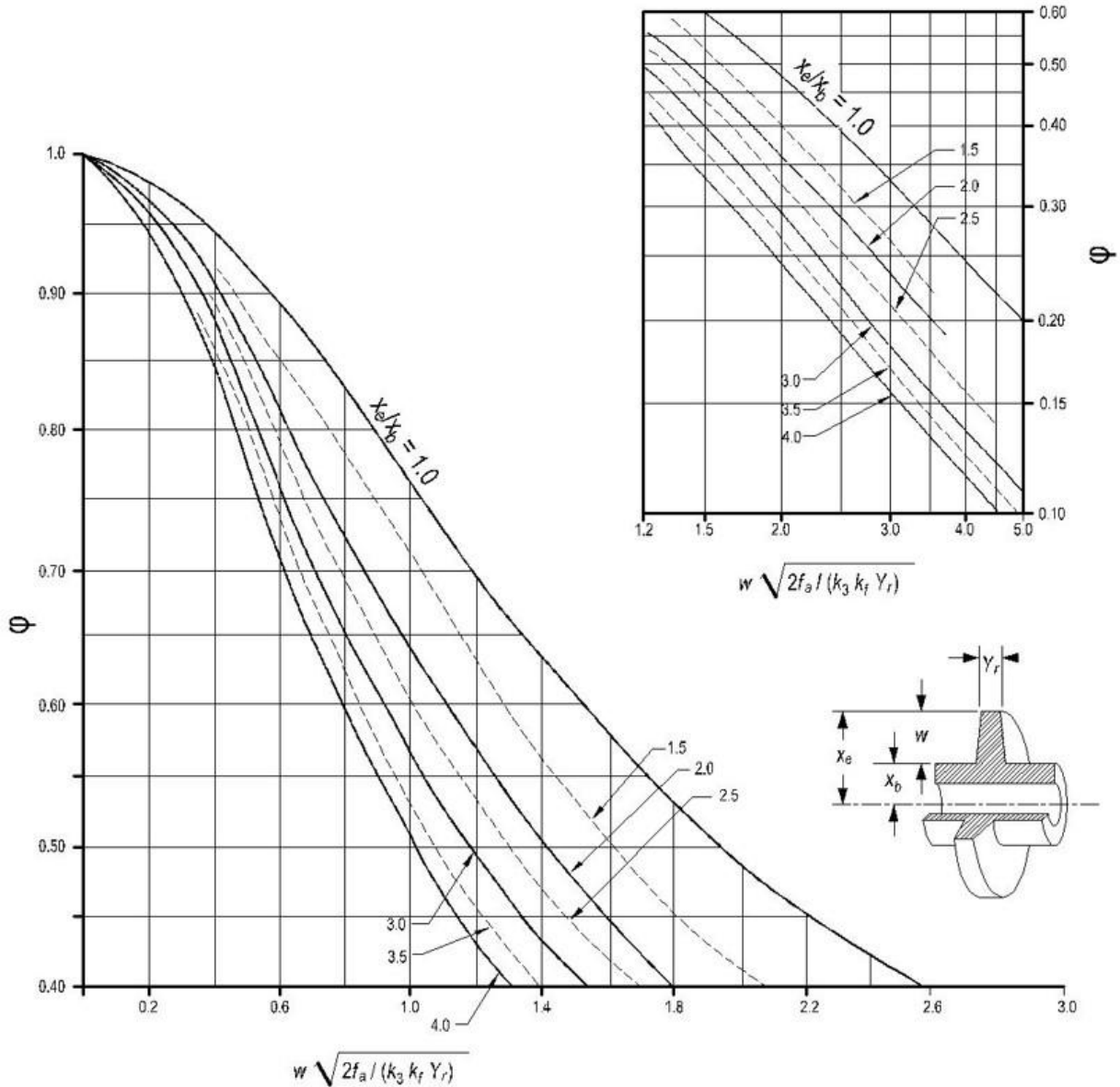
**Figure 32 Efficiency of Annular Fins of Constant Cross-Sectional
Area for Heat Flow (Spiral Fins)**

D.6.   Coil Geometry

> **D.6.1.   Calculate the Primary and Secondary External Surface Area**
>
>> Calculate the primary external surface area, A_p, and the secondary external surface area, A_s, by *tube arrangement* and *fin* configuration as follows:
>>
>> **D.6.1.1.   Continuous Plate Fins**
>>
>>> To calculate A_p for *continuous plate fins* for staggered tube and parallel tube arrangements, use Equation 135 or Equation 136 (these equations are the same). The primary external surface area is the exposed external tube area (if any) plus the external *fin* collar area, if used.

$$A_p = \frac{\pi D_o \left(L_t - N_f \left(L_c + Y_f \right) \right) N_t}{k_1} + \frac{\pi (D_o + 2Y_f) L_c N_f N_t}{k_1} \qquad 135$$

Rearranging Equation 135 results in Equation 136.

$$A_p = \frac{\pi N_t}{k_1} \left(D_o L_t - N_f Y_f (D_o - 2L_c) \right) \qquad 136$$

To calculate A_s for continuous *flat plate fins* for staggered tube and parallel tube arrangements, use Equation 137. The secondary external surface area is the sum of the areas of the *fin* sheets minus the areas of the tube holes and excludes any *fin* collar areas.

$$A_s = N_f \left(\frac{2L_f L_d}{k_1} - \frac{\pi N_h (D_o + 2Y_f)^2}{2k_1} + \frac{\pi (D_o + 2Y_f)(N_h - N_t)L_c}{k_1} \right) \qquad 137$$

To calculate A_s for continuous *configurated plate fins* for staggered tube and parallel tube arrangements, use Equation 137. Area dimensions L_f and L_d are already determined at the option of the manufacturer, from the blank *fin* sheet size prior to forming the configuration. The formation of the configuration does not provide edge trimming. Edge trimming is performed after forming, or from the finished *fin* size after forming.

D.6.1.2. Smooth Spiral Fins

To calculate A_p and A_s for *smooth spiral fins* for staggered tube and parallel tube arrangements, use Equation 138 and Equation 139. The primary external surface area is the total external tube surface area minus the area under the *fins* corresponding to the *fin* root thickness. The secondary external surface area is the exposed lateral and outside *fin* edge areas, calculated based on an individual annular *fin* that neglects the helical *fin pitch*.

$$A_p = \frac{\pi D_o N_t}{k_1} \left(L_t - N_f Y_r \right) \qquad 138$$

$$A_s = \frac{\pi N_f N_t}{2k_1} \left(D_f^2 - D_o^2 + 2D_f Y_e \right) \qquad 139$$

D.6.1.3. Crimped Spiral Fins

To calculate A_p and A_s for *crimped spiral fins* for staggered tube and parallel tube arrangements, use Equation 140 and Equation 141. The primary external surface area is the total external tube surface area minus the area under the *fins* corresponding to the *fin* root thickness. The secondary external surface area is the exposed lateral and outside *fin* edge areas, calculated based on an individual annular *fin* that neglects the helical *fin pitch*. For *crimped spiral fins*, the exposed lateral area is composed of the actual developed area of the crimped *fin* portion plus any smooth annular area at the outer extremities of the *fin*.

$$A_p = \frac{\pi D_o N_t}{k_1} \left(L_t - N_f Y_r \right) \qquad 140$$

$$A_s = \frac{\pi N_f N_t}{2k_1} \left(D_n (D_n - D_o) + \frac{D_f^2 - D_n^2}{2} + D_f Y_e \right) \qquad 141$$

D.6.1.4. **Plate Fins on Individually-finned Staggered Tube and Parallel (in-line) Tube Arrangements**

To calculate A_p and A_s for plate *fins* on individually-finned staggered tube and parallel (in-line) tube arrangements, use Equation 142 and Equation 143.

$$A_p = \frac{\pi N_t}{k_1}(D_o + 2Y_f)(L_t - N_f Y_f)$$

142

$$A_s = \frac{N_t N_f}{k_1}\left(2L_f L_d - \frac{\pi(D_o + 2Y_f)^2}{2} + (L_f + L_d)2Y_f\right)$$

143

D.6.2. **All Coil Types**

For all coil types, use Equation 144 through Equation 151 to determine the face area, A_f; the total external surface area, A_o; the total internal surface area, A_i; the ratio of the total external surface area to the total internal surface area, B; the total internal cross-sectional fluid flow area, A_{ix}; the number of *tube passes* per tube circuit, N_p; the number of return bends, N_b; and the total equivalent length of the tube circuit, L_e:

$$A_f = \frac{HL}{k_1}$$

144

$$A_o = A_s + A_p$$

145

$$A_i = \frac{\pi D_i N_t L_t}{k_1}$$

146

$$B = \frac{A_o}{A_i}$$

147

$$A_{ix} = \frac{\pi}{4k_1}D_i^2 N_c$$

148

$$N_p = \frac{N_t}{N_c}$$

149

$$N_b = N_p - N_{ih} - 1$$

150

$$L_e = \frac{(L_s N_p + L_{eb} N_b)}{k_3}$$

151

APPENDIX E. MASS AND VOLUMETRIC FLOW RATES OF AIR – INFORMATIVE

This appendix provides the informative equations for mass and volumetric flow rates of air.

The standard dry air volumetric flow rate, $Q_{a\,STD\,Dry\,Air}$, is calculated by Equation 152.

$$Q_{a\,STD\,Dry\,Air} = \frac{w_{a\,Dry\,Air}}{\rho_{a\,STD}} \qquad\qquad 152$$

The entering air-water vapor mixture mass flow rate, $w_{a1\,Air-Water\,Vapor\,Mixture}$, is calculated by Equation 153.

$$w_{a1\,Air-Water\,Vapor\,Mixture} = w_{a\,Dry\,Air}(1 + W_1) \qquad\qquad 153$$

The entering standard air-water vapor mixture volumetric flow rate, $Q_{a1\,STD\,Air-Water\,Vapor\,Mixture}$, is calculated by Equation 154 or Equation 155.

$$Q_{a1\,STD\,Air-Water\,Vapor\,Mixture} = \frac{w_{a1\,Air-Water\,Vapor\,Mixture}}{\rho_{a\,STD}} \qquad\qquad 154$$

$$Q_{a1\,STD\,Air-Water\,Vapor\,Mixture} = Q_{a\,STD\,Dry\,Air}(1 + W_1) \qquad\qquad 155$$

The entering air-water vapor mixture volumetric flow rate, $Q_{a1\,Air-Water\,Vapor\,Mixture}$, is calculated by Equation 156 or Equation 157.

$$Q_{a1\,Air-Water\,Vapor\,Mixture} = Q_{a\,STD\,Dry\,Air}\frac{\rho_{a\,STD}}{\rho_{a1\,Dry\,Air}} \qquad\qquad 156$$

$$Q_{a1\,Air-Water\,Vapor\,Mixture} = Q_{a1\,STD\,Air-Water\,Vapor\,Mixture}\frac{\rho_{a\,STD}}{\rho_{a1\,Air-Water\,Vapor\,Mixture}} \qquad\qquad 157$$

The leaving air-water vapor mixture mass flow rate, $w_{a2\,Air-Water\,Vapor\,Mixture}$, is calculated by Equation 158.

$$w_{a2\,Air-Water\,Vapor\,Mixture} = w_{a\,Dry\,Air}(1 + W_2) \qquad\qquad 158$$

The leaving standard air-water vapor mixture volumetric flow rate, $Q_{a2\,STD\,Air-Water\,Vapor\,Mixture}$, is calculated by Equation 159 or Equation 160.

$$Q_{a2\,STD\,Air-Water\,Vapor\,Mixture} = \frac{w_{a2\,Air-Water\,Vapor\,Mixture}}{\rho_{a\,STD}} \qquad\qquad 159$$

$$Q_{a2\,STD\,Air-Water\,Vapor\,Mixture} = Q_{a\,STD\,Dry\,Air}(1 + W_2) \qquad\qquad 160$$

The leaving air-water vapor mixture volumetric flow rate, $Q_{a2\,Air-Water\,Vapor\,Mixture}$, is calculated by Equation 161 or Equation 162.

$$Q_{a2\,Air-Water\,Vapor\,Mixture} = Q_{a\,STD\,Dry\,Air}\frac{\rho_{a\,STD}}{\rho_{a2\,Dry\,Air}} \qquad\qquad 161$$

$$Q_{a2\,Air-Water\,Vapor\,Mixture} = Q_{a2\,STD\,Air-Water\,Vapor\,Mixture}\frac{\rho_{a\,STD}}{\rho_{a2\,Air-Water\,Vapor\,Mixture}} \qquad\qquad 162$$

APPENDIX F. RATIONALE AND DETAILS FOR ALLOWING THE CHANGES IN A COIL LINE – INFORMATIVE

F.1.   Purpose

This appendix provides the rationale and details for allowing the changes in a *coil line* described in Section 5.7.1. and the calculations chosen that demonstrate the calculated capacity is less than 97.5% or the calculated pressure drop is greater than 105.0%.

F.2.   Copper Fin Thickness

Section 5.7.1 (1) states that copper *fin* thickness can be decreased up to 30% below aluminum *Fin* thickness for dry surface coils. Calculations of the variable *fin* metal thermal resistance, R_f, are based on the *fin* efficiencies, ɸ, as developed in *Efficiency of Extended Surface*. The variable *fin* metal thermal resistance, R_f, based on total external surface effectiveness, η, is calculated by Equation 5 for dry surfaces and by Equation 6 for wet surfaces. The total external surface effectiveness, η, is calculated by Equation 8. The product of *fin* metal thermal conductivity times *fin* thickness ($k_f\,Y_f$) is an input to these equations. For a given *fin* surface configuration, if this product is a constant, the resulting *fin* efficiency, total external surface effectiveness, and *fin* material thermal resistance is constant. For dry surface coils, if the original *standard ratings* are established with aluminum *fins*, these ratings can be used to predict the performance of copper *fin* coils as long as the copper *fin* thickness reduction results in a product $(k_f\,Y_f)_{Cu}$ that is greater than the corresponding aluminum *fin* product $(k_f\,Y_f)_{Al}$ with a safety factor. This is only valid for dry surface aluminum and copper fin coils due to the *fin* surface wettability of aluminum compared to copper is different. For a coil with the same *fin* configuration, the wet surface air pressure drop of copper *fin* coils is much higher than the wet surface air pressure drop of aluminum *fin* coils. Equation 163 and Equation 164 are used to determine the safety factors to validate the maximum copper *fin* thickness.

$$(k_fY_f)_{Cu} \geq (k_fY_f)_{Al}\,Safety\,Factor \tag{163}$$

$$\frac{(Y_f)_{Cu}}{(Y_f)_{A;}} \geq \frac{(k_f)_{Al}}{(k_f)_{Cu}}\,Safety\,Factor \tag{164}$$

Using values for aluminum and copper *fins*, derive the Section 5.7.1 (1) copper *fin* thickness be decreased up to 30% below aluminum *fin* thickness for dry surface coils rule. Using Table 5, the *fin* material thermal conductivity of A91100 Temper O aluminum *fins* is 128.3 Btu·ft/(h·ft^2·°F) and for C11000 Temper OS025 copper *fins* is 226.0 Btu·ft/(h·ft^2·°F). Using a 22.8% safety factor, this results in a maximum copper *fin* thickness reduction of 30% (1-0.7 = 30%), as shown below:

$$\frac{(Y_f)_{Cu}}{(Y_f)_{Al}} \geq \frac{(k_f)_{Cu}}{(k_f)_{Al}}\,SF = \frac{128.3}{226.0}\,SF = 0.57SF = 0.57 * 1.228 = 0.7$$

$$(Y_f)_{Cu} \geq 0.7(Y_f)_{Al}$$

F.3.   Fin Thickness Increase

Refer to Section 5.7.1 (2). As the *fin* thickness increases, the coil capacity increases due to a combination of two reasons:

1) The variable *fin* metal resistance, R_f, decreases as the *fin* thickness increases. The variable *fin* metal thermal resistance based on total external surface effectiveness, η, is calculated by Equation 5 for dry surfaces and by Equation 6 for wet surfaces. The total external surface effectiveness, η, is calculated by Equation 8. The product of *fin* metal thermal conductivity times *fin* thickness ($k_f Y_f$) is an input to these equations. For a given *fin* surface configuration, when the *fin* thickness increases, the *fin* efficiency and total external surface effectiveness increases, and this causes the *fin* material thermal resistance to decrease.

2) The air film heat transfer coefficient increases as the *fin* thickness increases. As the *fin* thickness increases, the minimum airside cross-sectional flow area decreases, resulting in higher local air velocity through the *fins*. This higher local air velocity results in a higher air film heat transfer coefficient. For a given standard dry air face velocity (fixed dry air mass flow rate), the air film heat transfer coefficient increases as the *fin* thickness increases, so using ratings of thinner *fins* to predict performance of coils with thicker *fins* is conservative from a coil capacity perspective. However, the air pressure drop increases as the *fin* thickness increases. Coils with increased *fin* thickness do not need to be tested if the air pressure drop is not more than 105% of the corresponding *standard ratings*.

F.4. Tube Wall Thickness

Refer to Section 5.7.1 (3). For a given outside diameter, the tube wall thickness dictates the inside diameter of the tube, and the effect of this on the overall coil performance is accurately captured by:

1) The constant tube metal thermal resistance, R_t, (Equation 4).

2) The tubeside performance (heat transfer coefficient and pressure drop) of *single-phase liquid* smooth tube coils.

3) The tubeside performance (heat transfer coefficient and pressure drop) of *single-phase liquid* non-smooth tube coils.

4) The tubeside performance (heat transfer coefficient and pressure drop) of steam condensing in smooth tube coils.

5) The tubeside performance of *refrigerant* evaporating (heat transfer coefficient and pressure drop) in smooth tube and non-smooth tube coils.

For a given tubeside fluid, tube outside diameter, and tube internal surface, the tube wall thickness can be changed from that used to determine *standard ratings*, and tube wall thickness can be used to accurately predict coil performance without additional coil tests.

F.5. Tube Material, Limited to Materials

Refer to Section 5.7.1 (4) and Table 5. With any change of material (as with any change of method of *fin*-to-tube bonding during manufacture), the *fin*-to-tube contact resistance (can be called *fin*-to-tube bond resistance) can vary and can depend on temperature. The manufacturer is responsible for including appropriate allowances for these effects.