SAE International™ SURFACE VEHICLE RECOMMENDED PRACTICE

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Rating Air-Conditioner Evaporator Air Delivery and Cooling Capacities

1. ***Scope***—The purpose of this SAE Recommended Practice is to establish uniform test procedures for measuring and rating air delivery and cooling capacity of truck and off-road self-propelled work machines used in earth moving, agriculture, and forestry air-conditioner evaporator assemblies. It is the intent to measure only the actual cooling capacity of the evaporator. It is not the intent of this document to rate and compare the performance of the total vehicle air-conditioning system.

1.1 This procedure is designed to provide truck and off-road self-propelled work machines used in earth moving, agriculture, and forestry manufacturers and air conditioning system suppliers with a cost-effective, standardized test method and calculations for measuring and rating air delivery and cooling capacity of truck cab air-conditioner evaporator assemblies. This procedure relates to HFC-134a (R-134a) refrigerant system.

2. ***References***—There are no referenced publications specified herein.

3. ***Definitions***

3.1 **Evaporator Assembly—**The evaporator assembly as defined consists of a coil (heat transfer surface), the means of forcing air over the heat transfer surface into the cab, and the complete enclosure (air conditioning unit) to be furnished for the installation.

3.2 **Refrigerant Control—**The device furnished with the evaporator assembly to regulate the flow of refrigerant into the coil. It is to be the control specified and included with the evaporator assembly being tested.

3.3 **Evaporator Assembly Capacity**

3.3.1 AIR DELIVERY RATE—The actual rate of airflow SCFM (standard air volume flow rate as specified in 8.1) for wet coil conditions. The air delivery is to be measured during the Capacity Rating Test in Section 7.

3.3.2 COOLING CAPACITY (AIR SIDE)—The amount of heat absorbed from the air flowing through the evaporator in W, kW (Btu/hour), as specified in 8.3 and 8.4.

3.3.3 COOLING CAPACITY (REFRIGERANT SIDE)—The amount of heat absorbed by refrigerant flowing through the evaporator tubes in W, kW (Btu/hour), as specified in Section 9.

3.4 **Dry Bulb Temperature—**Air temperature in °C (°F) as read from a standard thermometer or other appropriate temperature measuring device.

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3.5 Wet Bulb Temperature—Air temperature in °C (°F) essentially equal to that read from a wet bulb thermometer, or one whose sensing bulb is covered by a water-wetted wick located in the moving air stream.

3.6 Relative Humidity—Ratio of the amount of moisture (in mol-fraction) in the air to the maximum amount the air can hold at the same temperature and pressure.

3.7 Humidity Ratio—Ratio of the mass of water vapor to the mass of dry air kg_w/kg_a (lb_w/lb_a).

3.8 Dew Point (Saturation) Temperature—Air temperature in °C (°F) at which moisture begins to condense out as the air is cooled at constant pressure.

3.9 Total Heat (Enthalpy)—The total heat content (sensible and latent) in the air j/kg_a (Btu/lb_a) equal to the sum of the individual partial enthalpies of the dry air and water vapor.

3.10 Sensible Heat—The amount of heat associated with a change in the dry bulb temperature of the air.

3.11 Latent Heat—The amount of heat required to change the state of a substance. Specifically, it is the heat released as water vapor condenses out of moist air, and also the heat associated with the phase change of certain heat transfer fluids (volatile refrigerants, steam, etc.).

3.12 Discharge Line—High pressure (inlet line) that carries liquid Refrigerant to the expansion device.

3.13 Suction Line—Low pressure (outlet line) that carries evaporated (gaseous) Refrigerant to the compressor.

3.14 Subcooling—The degrees of temperature below the saturation temperature (based on the inlet pressure of expansion device) of the liquid Refrigerant °C (°F).

3.15 Superheat—The degrees of temperature above the saturation temperature (based on the outlet pressure of the evaporator) of the vaporized Refrigerant °C (°F).

3.16 Expansion Device—Valve or fixed orifice in the Refrigerant circuit whose purpose is to meter refrigerant into the evaporator inducing a large pressure drop causing a change of state.

3.17 Compressor—Pumps low pressure Refrigerant vapor out of the evaporator by suction, raises its pressure, and then pumps it, under high pressure, into the condenser.

3.18 Condenser—Removes heat from the entering high pressure, high temperature de-superheated Refrigerant vapor changing it to a high pressure, high temperature liquid.

3.19 Evaporator—Removes unwanted heat from the air by the boiling of liquid Refrigerant in the evaporator coil.

3.20 Psychrometric Chart—Graphical presentation of moist air properties.

3.21 Resistance Temperature Device (RTD)—used for precision measurement of refrigerant temperature for purpose of capacity calculations.

4. Symbols and Units

A = Area in m^2 (ft^2)
A_N = Area of Nozzle in m^2 (ft^2)
BP = Barometric Pressure (Absolute) in Pa (in of Hg)
CFM = Air Volume Flow Rate in m^3/min (Cubic Feet per Minute, CFM) at the Air Density Calculated for the Conditions Existing at the Air Measurement Nozzle
DN = Diameter of Nozzle, m^2 (ft^2)
T = Temperature in °C(°F)
DB = Dry Bulb in °C (°F)
DC = Direct Current, amp
dn = Density of Air at the Nozzle in kg/m^3 (lb/ft^3)
Hg = Mercury
in = inches
N_T = Air Temp DB at Nozzle °C(°F)
PD = Pressure Differential Pa (psi)
SCFM = Air Volume Flow Rate in Standard Cubic Feet per Minute Based on1.20 kg/m^3 (0.075 lb/ft^3) for Dry Air at 21 °C (70 °F) and 101.04 kPa (29.92 in Hg) (see detailed explanation in 8.1.2)
V = Velocity in m/min (feet per minute, FPM)
SP = Static Pressure, Pa (psig)
WB = Wet Bulb, °C (°F)
WG = Water Gauge in mm (inches)
W = Air Humidity Ratio in kg_w/kg_a (lb_w/lb_a)
G = Mass Flow Rate of the Air (kg/s, sometimes kg/hour (lb/hour)) kg_a/s, kg_{mix}/s, kg_w/s; (lb_a/hour, lb_{mix}/hour, and lb_w/hour) = air mass flow of the dry air, moist air - air-water vapor mixture, and water vapor
G_r = Mass Flow Rate of the Refrigerant in kg/hour (lb/min)
Q_t = Total Heat Transferred (Total Cooling Capacity), air side in kW (Btu/hour)
Q_s = Sensible Heat Transferred (Sensible Cooling Capacity), air side in kW (Btu/hour)
Q_r = Total Heat Transferred (Total Cooling Capacity), refrigerant side in kW (Btu/hour)
h = Enthalpy (Change in Heat Content), j/kg (Btu/lb)
$C_{p.a}$ = Specific Heat of the Air in j/(kg*K) (Btu/lb/° F)
D = Diameter in m (inches or feet)

5. Test Equipment

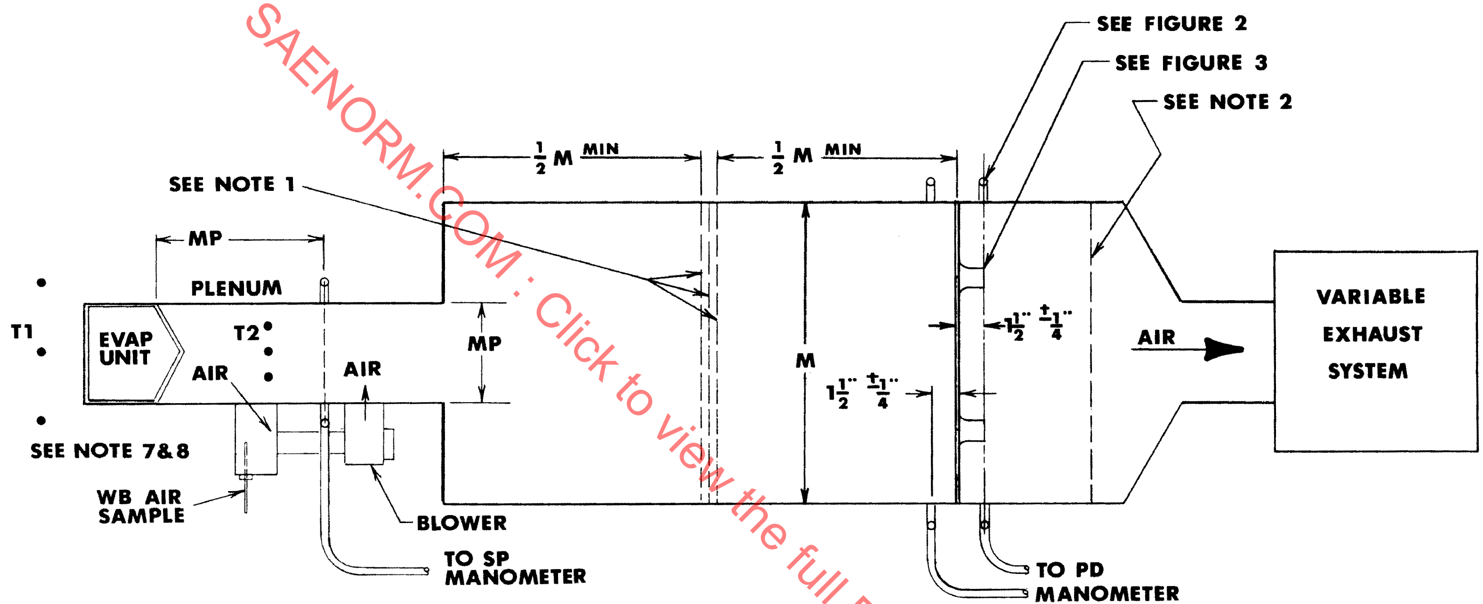
5.1 **Test Room (Calorimeter Room)—**A room in which specified ambient test conditions can be maintained to supply a controlled ambient for the evaporator assembly with constant inlet DB and WB as specified in 6.4.

5.2 **Airflow Measurement Chamber (Air Booth)—**See Figure 1.

5.3 **Nozzle—**See Figure 3.

The nozzle throat shall be measured on four diameters at 45 degree intervals, to an accuracy of 0.001 D and shall not deviate more than 0.002 D from the mean. Measurements shall also be made on the inlet of the parallel section which may be +0.002 D greater than the exit, but no less. The nozzle surface must be smooth with surface waves no greater than 0.001 D peak-to-peak. Recommended construction is spun aluminum, and the dimension D can be calculated from 8.3 based on anticipated air flow measurement requirements.

NOTE—Minimum acceptable nozzle PD shall be 99.5 Pa (0.4 in WG).



1. Chamber settling means shall be provided consisting of one perforated plate 30 to 40% open area followed by at least ten orifice diameters with two screens of 55 to 65% open area at least ten mesh lengths apart, or other means to provide substantially uniform velocity distribution ahead of nozzles. Maximum velocity at any point in a plane midway between the nozzle plate and the settling means, shall not exceed 122 m/min (400 FPM).
2. One perforated plate or screen of 50% maximum open area is to be placed downstream of nozzles as shown. There shall be a minimum clearance between the nozzle discharge and the perforated plate of 2–½ DN of the largest nozzle.
3. Chamber cross sectional area shall be such that V_M is less than 122 m/min (400 FPM) where $V_M = CFM/A_M$. Chamber cross sectional shape may be circular or rectangular. M is the diameter of a round chamber or the equivalent diameter of a rectangular chamber with sides a and b: $M = \sqrt{4av/\pi}$.
4. If more than one nozzle, they shall be located as symmetrically as possible.
5. Minimum distance from center line of nozzle to wall of chamber shall be 1.5 DN.
6. Minimum distance between centers of any two nozzles in simultaneous use shall be 3 DN of the larger nozzle.
7. T_1 represents the air inlet DB and WB temperature averaging grid (see Fig. 4).
8. T_2 represents the air outlet WB temperature averaging grid (see 7.3.13).
9. SP and PD are to be measured with four connections 90 deg apart manifolded into a piezometer ring. Static taps shall be 0.318 cm ID (⅛ in) or less, and are to be straight and normal to the surface for a distance of two diameters. No surface irregularities shall be in the vicinity of the static holes (see Fig. 2).
10. The minimum acceptable reading for nozzle PD shall be 99.5 Pa WG (0.4 in) (see Fig. 3).

FIGURE 1—AIR MEASUREMENT CHAMBER

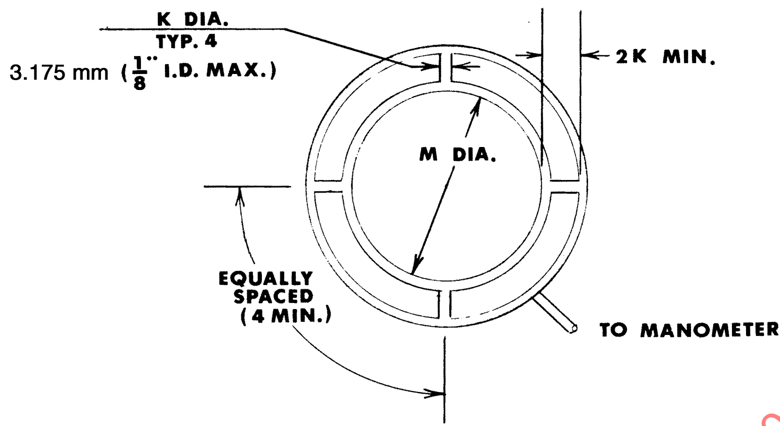


FIGURE 2—PIEZOMETER RING

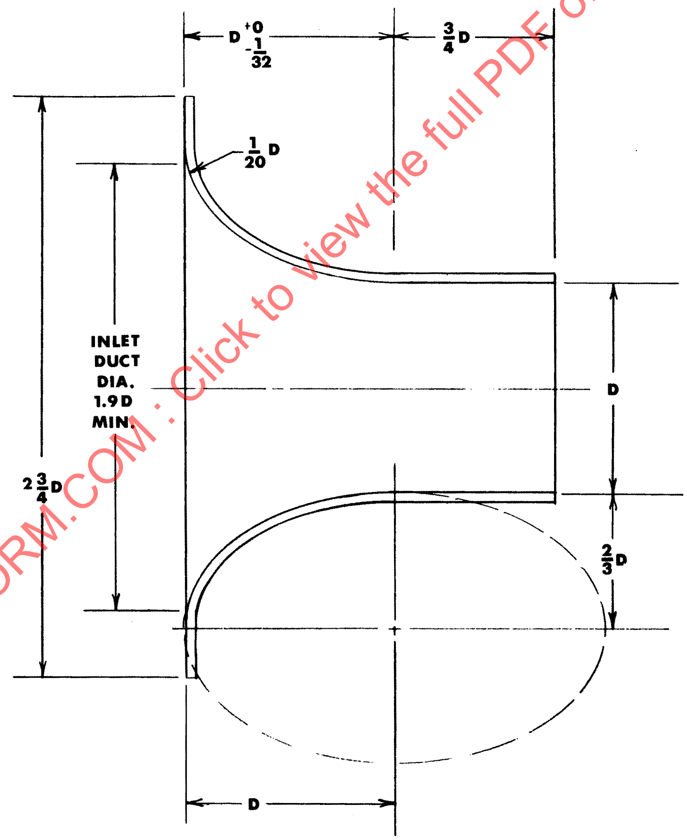


FIGURE 3—NOZZLE

5.4 Inclined Manometers—Inclined manometers used in conjunction with 5.2 shall have a scale expansion factor of at least 10 [distance between 0 to 249 Pa (0 and 1 " WG) to be at least 25.4 cm (10 in)] with scale divisions of 2.49 Pa (0.01 in WG) maximum for pressures below 498 Pa (2 in WG) and 12.4 Pa (0.05 in WG) maximum scale divisions for pressures above 498 Pa (2 in WG). Micromanometers may be used in place of the inclined manometers.

5.5 Temperature and Humidity Measurement Instrumentation

5.5.1 Air dry bulb temperatures are to be measured with an accuracy of ±0.3 °C (±0.5 °F). The temperature measurements may be made with precision thermometers, thermocouples, or RTD's with the measurement device having a minimum resolution and certified accuracy of ±0.3 °C (±0.5 °F).

5.5.2 Wet bulb temperatures are to be measured with a certified accuracy and minimum resolution of ±0.1 °C (±0.2 °F). The wet bulb temperatures are to be made with precision wet bulb thermometers and air sampling device (shown in Figure 4) to obtain an average air sample and provide a minimum velocity of 305 m/min (1000 FPM) over the bulb.

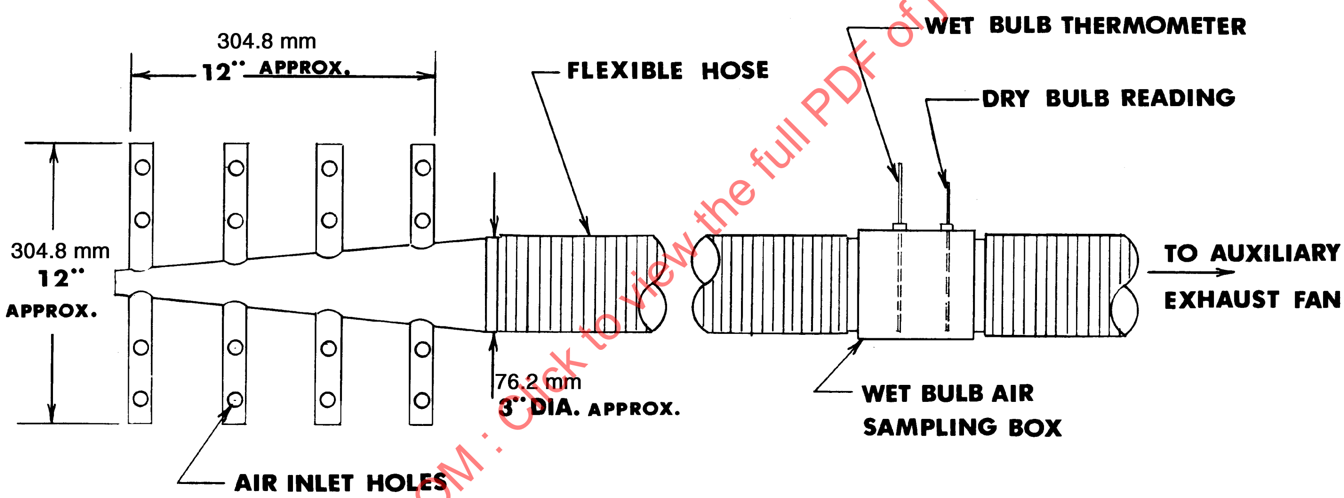


FIGURE 4—WET BULB SAMPLER

5.5.3 Dew Point Temperatures are to be measured with a certified accuracy and minimum resolution of ±0.1 °C (±0.2 °F). The dew point temperatures are to be made with precision chilled mirror or other similar device.

5.5.4 Relative Humidity (RH) is to be measured with a certified accuracy of ≤±2% of the rate. The RH is to be made with precision relative humidity measuring device.

5.5.5 Refrigerant Temperatures are to be measured with a certified accuracy of ±0.1 °C (±0.2 °F). The refrigerant temperatures are to be made with precision RTD's that immerse into appropriate refrigerant line by means of temperature tap. RTD's have to be installed on the liquid and suction lines as close as possible to the evaporator core but not further than 15.4 cm (6 in) from evaporator (expansion device).

5.6 Refrigerant pressure gauges or pressure transducers with a minimum resolution and certified accuracy of ±3.45 kPa (±0.5 psi) for evaporator suction pressure measurements and ±6.89 kPa (±1.0 psi) for the pressure measurement at the inlet to the refrigerant control per 3.2. Pressure measuring devices have to be installed on the liquid and suction lines by means of pressure tap as close as possible to the evaporator core but not further than 15.4 cm (6 in) from evaporator (expansion device).

5.7 Condenser of sufficient capacity to maintain the conditions specified in 6.5.

5.8 Compressor to be of sufficient capacity to maintain the conditions as specified in 6.6.

5.9 DC voltmeter with a minimum resolution and certified accuracy of ±0.1 V.

5.10 DC ammeter with a minimum resolution and certified accuracy of ±0.2 amp.

5.11 Mercury barometer with vernier scale to provide ±67.5 Pa (±0.02 in Hg) minimum resolution.

5.12 Refrigerant Mass Flow Rate are to be measured with a minimum resolution and certified accuracy of ±0.15% of the rate by means of precision refrigerant flow meter (for example, MicroMotion mass flow meter, model Elite)

6. Test Conditions

6.1 The evaporator blower is to be operating on high speed, the fresh air inlet control is to be in fresh air mode, and the outlet air control is to be set to the air-conditioning mode.

6.2 The voltage input shall be the nominal voltage ±0.1 V measured at the blower motor leads (at a maximum of 15.4 cm (6 in) from the motor); i.e., 12.0 V ± 0.1 for 12.0 V systems. Referencing the voltage to the motor leads is necessary to eliminate the variations in input voltage caused by different wire harness sizes and other variable voltage losses between the vehicle power source and the blower motor.

6.3 The evaporator assembly outlet air is to discharge to the airflow measurement chamber plenum, and the variable speed exhaust fan is to be adjusted to maintain 0 Pa (0 in WG) SP through 498 Pa (2.0 in WG) SP at 124 Pa (0.50 in) increments in the plenum with 12.0 V input to the evaporator blower motor.

6.4 The ambient air in the test room is to be maintained at 26.7 °C (80 °F) DB ± 0.6 °C (±1 °F) and 19.4 °C WB (67 °F) ± 0.3 °C (±0.5 °F), which is 50% ± 1.5% of RH and 14.4 °C (58 °F) ± 0.6 °C (±1 °F) of Dew Point Temperature. This is the air supply to the evaporator assembly inlet.

6.5 The condenser capacity is to be adjusted to maintain refrigerant subcooling of 5 °C to 7 °C (~9 °F to 12 °F) at 1380 kPa ± 34 kPa (200 psig ± 5 psig) at the refrigerant control in 3.2. The refrigerant pressure and temperature are to be measured as close as possible but not further than 15.4 cm (6 in) from the refrigerant control inlet .

6.6 The compressor capacity (speed) is to be adjusted to maintain an average of 207kPa ± 3.47 kPa (30 psig ± 0.5 psi) at the evaporator suction line outlet. The pressure is to be as close as possible but not further than 15.4 cm (6 in) from the evaporator suction line outlet.

6.7 With evaporator assembly enclosures containing a heater core that requires no water valve shutoff, a water temperature of 82.2 °C (180 °F) at the core inlet should be maintained with a flow rate of 816.6 kg/hour (30 lb/min).

7. Test Procedure

7.1 One very important condition that must be satisfied during the testing process when measurements are taken is that the air/refrigerant heat transfer balance should be within 3%. During vehicle HVAC system operations, the airside heat transfer is balanced on the tubeside of the coil with an equal amount of heat being either absorbed or given up by the fluid flowing through the tubes. So, the measuring procedure should begin only after stable air/refrigerant sides heat transfer balance has been achieved. Comparison of this heat transfer balance shall be made on the basis of air and refrigerant side capacity calculations as specified in 8.3 or 8.4 and Section 9.

NOTE—If evaporator assembly enclosures containing a heater core that requires no water valve shutoff (see 6.7), the heat absorbed from the hot coolant should also be considered in the heat balance calculations.

7.2 Data shall be recorded after the above test conditions and heat transfer balance have stabilized for a minimum of 20 min. In addition to the stabilized conditions defined in 6.1 to 6.6, the measured air delivery shall not decrease by more than 2% during the stabilization period or the time shall be extended until such stabilization is obtained. A minimum of seven sets of readings shall be recorded with a minimum of 5 min between sets.

NOTE—Remember the A/C loop is a dynamic system, i.e., changing one condition has an effect on everything else. Therefore, when changing any part of the system wait several minutes for all conditions to stabilize again before readjusting.

7.3 The following information shall be recorded for each of the sets of data described in 7.2.

7.3.1 Input voltage.

7.3.2 Current consumption of evaporator blower motor(s).

7.3.3 Static pressure in plenum.

7.3.4 Nozzle pressure drop.

7.3.5 Refrigerant pressure at flow control inlet.

7.3.6 Refrigerant temperature at flow control inlet.

7.3.7 Refrigerant pressure at evaporator suction line outlet.

7.3.8 Refrigerant temperature at evaporator suction line outlet.

7.3.9 Refrigerant mass flow rate.

7.3.10 Dry bulb inlet temperature to the evaporator.

7.3.11 Wet bulb inlet temperature to the evaporator.

7.3.12 Dry bulb temperature at the nozzle.

7.3.13 Wet bulb outlet temperature, or dew point temperature, or relative humidity from the evaporator. Note that the outlet temperatures or relative humidity is not to be measured across the face area of the coil and must be measured downstream of the blower. One of the following locations is to be used for measuring the outlet WB temperature:

7.3.14 A wet bulb thermometer, chilled mirror or other measuring device may be inserted directly into the main branch of the outlet duct at the cross section of the duct where it has been established that all further ducting and extensions are confined within the cab. The wet bulb, dew point, or RH reading is to be an average of three locations at this cross section. A minimum of 304 m/min (1000 FPM) air velocity is required at each of the three locations.

7.3.14.1 A wet bulb, dew point, or RH reading may be obtained from an air sample from the tunnel plenum obtained with the air sampling device described in Figure 4.

7.3.15 Barometric pressure is to be recorded at the start and at the completion of the test.

8. Air Side Calculations

8.1 The calculations are to be based on the averages obtained for the sets of data in 7.2.

8.1.1 NOZZLE AIR DENSITY (dn).

$$dn = \frac{(BP)(1.326)}{459.7 + N_T}, \quad lb/ft^3 \qquad \text{(Eq. 1)}$$

where:

$1\ lb/ft^3 = 16.02\ kg/m^3$
N_T = Air Temperature DB at the Nozzle in °F
BP = Absolute Barometric Pressure in of Hg.

8.1.2 AIR DELIVERY RATE (CFM)

$$CFM = (1086)(A_N)\left(\sqrt{\frac{PD}{dn}}\right) \qquad \text{(Eq. 2)}$$

where:

1 CFM = 1.7 m^3/hour = 0.028333 m^3/min
CFM is actual air volume flow rate, ft^3/min
A_N = Area of the Nozzle in ft^2
PD = Pressure Drop Across the Nozzle in WG
dn = Nozzle Air Density specific volume, lb_a/ft^3 per 8.1.1

The Equation 2 calculates the actual air volume flow rate (ACFM or CFM). Most air conditioning equipment, however, is rated on the basis of standard air CFM or SCFM. Standard air is a concept that was established in order to maintain uniformity in the testing, rating, and application of this equipment. Its use permits relatively simple calculation procedures for determining the performance of coils, fans and other products. The coil performance ratings in this document are based on standard air CFM.

At T_a = 20.7 °C (69.4 °F) and P_{atm} = 760 mm Hg (29.921 in Hg) only:

1 CFM = 2.048 kg/hour = 0.00569 kg/s
1 kg/hour = 0.829 m^3/hour = 0.488 CFM
1 CFM = 1SCFM

By definition, standard air has a density of 0.075 lb/ft^3 = 1.201385 kg/m^3. At sea level pressure (29.921 in Hg), this density corresponds to that of dry air at a temperature of 20.7 °C (69.4 °F). The corresponding temperature for moist air is higher and depends on the actual moisture content. For practical purposes, 21.1 °C (70 °F) is the generally accepted base temperature.

8.2 Standard Airflow (SCFM)—Published Air Delivery Rating. (Note that this is a wet coil rating obtained in 7.2)

$$SCFM = (CFM)\left(\frac{dn}{0.075}\right) \qquad \text{(Eq. 3)}$$

For process cooling and other applications where significant levels of moisture are involved, the following formulas may be used to convert the air flow rate to SCFM:

$$SCFM = \frac{(CFM) \times (dn) \times (1 + W)}{(0.075)} \qquad \text{(Eq. 4)}$$

where,

W is humidity ratio, lb_w/lb_a
0.075 is density of standard air, lb_{mixt}/ft^3

Note that the (1 + W) term = lb_{mixt}/lb_a (ratio of the mass of the moist air – air-water vapor mixture to the mass of the dry air). For comfort conditioning applications, this term (1 + W) is usually ignored for simplicity, since the humidity ratio is a relatively small number. However, as temperature and humidity increase, W becomes increasingly significant and must be included to maintain accuracy. This is especially true for many process cooling applications where humidity ratios of 0.5 or higher are common. The error involved in neglecting the (1 + W) term is directly proportional to the humidity ratio itself. For example, ignoring a value of W = 0.01 results in a 1% error, when W = 0.1, the error = 10%, etc.

8.3 Cooling Capacity kW (Btu/hour), Wetbulb (Dewpoint) and Drybulb Temperatures Measurement Method—Published Capacity Rating.

8.3.1 HEAT TRANSFER PROCESS—The amount of heat transfer Q W, kW (Btu/hour) that takes place as an air stream passes through a cooling coil is the product of the mass flow rate of the air G, kg/hour (lb/min) and the change in its heat content h j/kg (Btu/lb). During a cooling and dehumidifying process, the airstream undergoes changes in both sensible and latent heat contents. Sensible heat is the heat associated with the change in the air dry bulb temperature. Latent heat is that amount released by the water vapor as it condenses. Sensible and latent heats are usually combined and expressed as total heat, or enthalpy (h).

A change in enthalpy includes changes in both sensible and latent heats, but it does not account for the small amount of heat in the condensed water that has left the air stream. However, for some applications, the enthalpy of the condensed water is very small in comparison to the total enthalpy change, and as such it is neglected for simplicity.

When the airstream is heated or cooled without dehumidification, no latent heat transfer takes place. In this case, the heat transfer is said to be all sensible, or total heat transfer is sensible heat transfer.

The airside heat transfer is balanced on the tubeside of the coil with an equal amount of heat being either absorbed or given up by the fluid flowing through the tubes. Again, the heat transferred is the product of the fluid flow rate G, kg/hour or kg/s (lb/hour) and a corresponding change in fluid enthalpy h j/kg (Btu/lb). The fluid enthalpy change may be due to a temperature change only (for single phase fluids such as water, glycol, oil, etc.), a phase change (evaporating or condensing refrigerants, steam, etc.) or a combination of both.

8.3.2 CALCULATIONS WITHIN NORMAL COMFORT CONDITIONING RANGE—In the normal comfort conditioning range, when the humidity ratio has not been included in the SCFM derivation, the following formulas are generally used for heating and cooling coil calculations.

8.3.3 TOTAL HEAT TRANSFERRED

$$Q_t(\text{Btu/ hour}) = G(\text{lb}_a/\text{ hour}) \times (\Delta h)(\text{Btu/ (lb)}) \quad \text{(Eq. 5)}$$

$$Q_t(\text{W}) = G(\text{kg/ s}) \times (\Delta h)(\text{j/ kg}) \quad \text{(Eq. 6)}$$

$$Q_t(\text{W}) = 0.2931\ Q_t(\text{Btu/ hour}) \quad \text{(Eq. 7)}$$

$$Q_t(\text{Btu/ hour}) = 3.4118\ Q_t(\text{W}) \quad \text{(Eq. 8)}$$

8.3.4 SENSIBLE HEAT TRANSFERRED

$$Q_s(\text{Btu/ hour}) = G(\text{lb}_a/\text{ hour}) \times C_{p.a}(\text{Btu/ lb}_a/\ ^\circ F) \times (\Delta T)(^\circ F) \quad \text{(Eq. 9)}$$

$$Q_s(W) = G(\text{kg/ s}) \times C_{p.a}(\text{j/ }(\text{kg}_a{*}K)) \times (\Delta T)(^\circ C) \quad \text{(Eq. 10)}$$

where:

G is Air mass flow rate, lb/hour or kg/hour of dry air
$\Delta h = h_1 - h_2$ is Enthalpy change of the air, j/kg (Btu/lb)
h_1 = Total Heat Content (Enthalpy) of the inlet air to the evaporator
h_2 = Total Heat Content (Enthalpy) of the outlet air in j/kg (Btu/lb) of dry air.
$C_{p.a}$ is Specific heat of the air, j/(kga*K) (Btu/lba/ °F)
DT $=T_1 - T_2$ is Dry Bulb Temperature change of the air, °C (°F)
T_1 = Air temperature at the inlet to the evaporator
T_2 = Air temperature at the outlet

The average inlet and outlet WB temperatures or other moist air measured parameters obtained in 7.3.12 and 7.3.14 are to be referred to the any of the available Psychrometric Tables (for example, see Table 1 at the end of this procedure), Psychrometric Chart, or variety of available software for air psychometric properties calculations to obtain the values of h_1 and h_2. Note that the rating is to be based on the actual conditions.

Psychrometric charts and tables usually present enthalpy, specific volume and other moist air properties on a DRY AIR basis rather than the actual air-water vapor mixture. This convention was adopted because it eliminates having to work with a varied air mass flow rate. During a dehumidifying process, the total mass flow rate of the air-water vapor mixture changes as the vapor condenses out of the air stream. Because of this vapor loss, the mass of the mixture leaving the process is less than that which entered. However, the amount of DRY AIR in the air stream remains constant throughout the process. Using air properties expressed in terms of this dry air portion enables us to use a constant air mass flow rate, resulting in much simpler calculations.

Proper conversion of SCFM to lb_a/hour depends on how the SCFM was derived originally. If the humidity ratio was not included in the derivation, the conversion is:

$$G(\text{lb}_a/\text{ hour}) = (0.075) \times (60) \times (\text{SCFM}) = (4.5) \times (\text{SCFM}) \quad \text{(Eq. 11)}$$

where:

4.5 is (0.075 lb/ft^3) x (60 min/hour)

As a result, Total Heat Transferred or Total Cooling Capacity can be calculated as follows:

$$Q_t(\text{Btu/ hour}) = (4.5) \times (\text{SCFM}) \times (\Delta h); \quad \text{(Eq. 12)}$$

$$Q_t(W) = 0.2931 \times Q_{t,}(\text{Btu/ hour}); \text{ or} \quad \text{(Eq. 13)}$$

$$Q_t(\text{Btu/ hour}) = 3.4118 \times Q_{t,}(W) \quad \text{(Eq. 14)}$$

8.3.5 SENSIBLE CAPACITY

$$Q_s(\text{Btu} / \text{hour}) = (1.09) \times (\text{SCFM}) \times (\Delta T) \qquad \text{(Eq. 15)}$$

where:

$1.09 = (0.075\ \text{lb/ ft}^3) \times (0.242\ \text{Btu/lb/°F}) \times (60\ \text{min/hr})$
$\Delta h = h_1 - h_2$ is Enthalpy change of the air, Btu/lb,
$\Delta T = T_1 - T_2$ is Dry Bulb Temperature change of the air, °C or °F

Although not exact, Equations 5 and 9) provide reasonable accuracy for the range of application.

8.3.6 CALCULATIONS OUTSIDE THE NORMAL COMFORT CONDITIONING RANGE—If the humidity ratio was included, Equation 11 becomes:

$$G(\text{lb}_a / \text{hour}) = \frac{(4.5) \times (\text{SCFM})}{1 + W} \qquad \text{(Eq. 16)}$$

1 kg/hour = 0.03675 lb/min = 2.205 lb/hour,
1 lb/hour = 0.0167 lb/min = 0.04537 kg/hour

As a result the formulas for total (Q_t) and sensible(Q_s) capacity calculations, when SCFM has been derived using the humidity ratio (Equation 16), can be presented as follows:

$$Q_t(\text{BTU} / \text{hour}) = \frac{(4.5) \times (\text{SCFM}) \times (\Delta h)}{(1 + W)} \qquad \text{(Eq. 17)}$$

$$Q_s(\text{BTU} / \text{hour}) = \frac{(4.5) \times (C_{p.a.}) \times (\text{SCFM}) \times (\Delta T)}{(1 + W)} \qquad \text{(Eq. 18)}$$

where,

$C_{p.a.}$ is Specific heat of the air, Btu/lb_a/°F.

The specific heat of moist air may be obtained from:

$$C_{p.a.} = 0.240 + (0.444 \times W) \qquad \text{(Eq. 19)}$$

where:

0.240 is Specific heat of dry air
0.444 is Specific heat of water vapor
W is Humidity ratio, lb_w/lb_a.

NOTE—When the humidity ratio changes significantly from the entering to the leaving air condition, the specific heat should be based on an average value for W.

8.4 Cooling Capacity, kW (Btu/hour), Condensate Collection Method—This very reliable method involves weighing the condensate collected off the core over a given time interval. Using the Equations 5 and 9 the air mass flow (G) across the core can be read in lb/min or kg/hour. For enthalpy h_1 determination should be used the same method as in 8.3: locate where the dry bulb temperature (from sampling device) and the wet bulb (from sampling device) or dewpoint temperature intersect on a psychrometric chart. Then follow that point over to the left with a straight edge to find the corresponding enthalpy value (h_1) at those conditions. Also, follow the intersection point over to the right to determine how many grains of moisture are going into the core (moisture in).