RATIONALE

Chamber settling screen density was increased so uniform velocity is established ahead of the nozzles.

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.

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) |
| $C_{p.a}$ | = | Specific Heat of the Air in $j/(kg*K)$ (Btu/lb/°F) |
| 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 |
| D | = | Diameter in m (inches or feet) |
| DB | = | Dry Bulb in °C (°F) |
| DC | = | Direct Current, amp |
| DN | = | Diameter of Nozzle, $m^2$ ($ft^2$) |
| dn | = | Density of Air at the Nozzle in $kg/m^3$ ($lb/ft^3$) |
| 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) |
| Hg | = | Mercury |
| h | = | Enthalpy (Change in Heat Content), j/kg (Btu/lb) |
| in | = | inches |
| $N_T$ | = | Air Temp DB at Nozzle °C (°F) |
| PD | = | Pressure Differential Pa (psi) |
| $Q_r$ | = | Total Heat Transferred (Total Cooling Capacity), refrigerant side in kW (Btu/hour) |
| $Q_s$ | = | Sensible Heat Transferred (Sensible Cooling Capacity), air side in kW (Btu/hour) |
| $Q_t$ | = | Total Heat Transferred (Total Cooling Capacity), air side in kW (Btu/hour) |
| SCFM | = | Air Volume Flow Rate in Standard Cubic Feet per Minute Based on 1.20 $kg/m^3$ (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) |
| SP | = | Static Pressure, Pa (psig) |
| T | = | Temperature in °C (°F) |
| V | = | Velocity in m/min (feet per minute, FPM) |
| W | = | Air Humidity Ratio in $kg_w/kg_a$ ($lb_w/lb_a$) |
| WB | = | Wet Bulb, °C (°F) |
| WG | = | Water Gauge in mm (inches) |

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    A means of measuring airflow such as described in 5.2.1 through 5.2.3, or any other device of equivalent accuracy that does not interfere with the intent of the test:

5.2.1    Airflow Measurement Chamber (Air Booth)

See Figure 1.

5.2.2    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).

5.2.3    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 "in 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.

VARIABLE EXHAUST SYSTEM

SEE FIGURE 2
SEE FIGURE 3
SEE NOTE 2

AIR

TO PD MANOMETER

$1\frac{1}{2}''$ ± $\frac{1}{4}''$

$1\frac{1}{2}''$ ± $\frac{1}{4}''$

$\frac{1}{2}$ M MIN

$\frac{1}{2}$ M MIN

M

SEE NOTE 1

MP

PLENUM

MP

T2

AIR

BLOWER

TO SP MANOMETER

AIR

EVAP UNIT

T1

SEE NOTE 7 & 8

WB AIR SAMPLE

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 50% or less 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/$A_{xs}$. 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. 3).

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

5.3     Temperature and Humidity Measurement Instrumentation

5.3.1   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).

5.3.2   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.
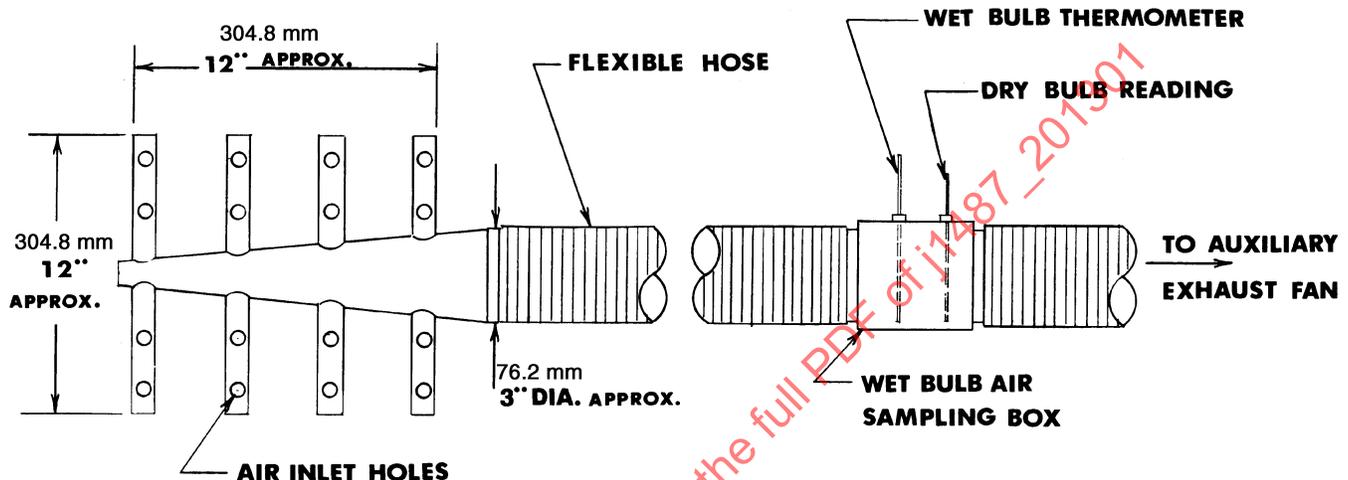


FIGURE 4 - WET BULB SAMPLER

5.3.3   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.

5.3.4   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.

5.3.5   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).

5.4     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).

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

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

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

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

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

5.10 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 32. 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 207 kPa ± 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)

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

(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)

$$CFM = (1086)(A_N)\left(\sqrt{\frac{PD}{dn}}\right)$$

(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)

$$SCFM = (CFM)\left(\frac{dn}{0.075}\right)$$

(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:

$$SCFM = \frac{(CFM) \times (dn) \times (1 + W)}{(0.075)}$$

(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

$$Q_t(Btu/hour) = G(lb_a/hour) \times (\Delta h)(Btu/(lb))$$

(Eq. 5)

$$Q_t(W) = G(kg/s) \times (\Delta h)(j/kg)$$

(Eq. 6)

$$Q_t(W) = 0.2931 \ Q_t(Btu/hour)$$

(Eq. 7)

$$Q_t(Btu/hour) = 3.4118 \ Q_t(W)$$

(Eq. 8)

8.3.4     Sensible Heat Transferred

$$Q_s(Btu/hour) = G(lb_a/hour) \times C_{p.a}(Btu/lb_a/°F) \times (\Delta T)(°F)$$

(Eq. 9)

$$Q_s(W) = G(kg/s) \times C_{p.a}(j/(kg_a*K)) \times (\Delta T)(°C)$$

(Eq. 10)