TRANE

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Psychrometry

A Trane Air Conditioning Clinic

The Trane Company believes that it is incumbent on manufacturers to serve the industry by regularly disseminating information gathered through laboratory research, testing programs, and field experience.

The Trane Air Conditioning Clinic series is one means of knowledge sharing. It’s intended to acquaint a nontechnical audience with various fundamental aspects of heating, ventilating, and air conditioning.

We’ve taken special care to make the clinic as uncommercial and straightforward as possible. Illustrations of Trane products only appear in cases where they help convey the message contained in the accompanying text.

This particular clinic introduces the concept of **psychrometry**, the science concerned with the physical laws that govern air – water mixtures.

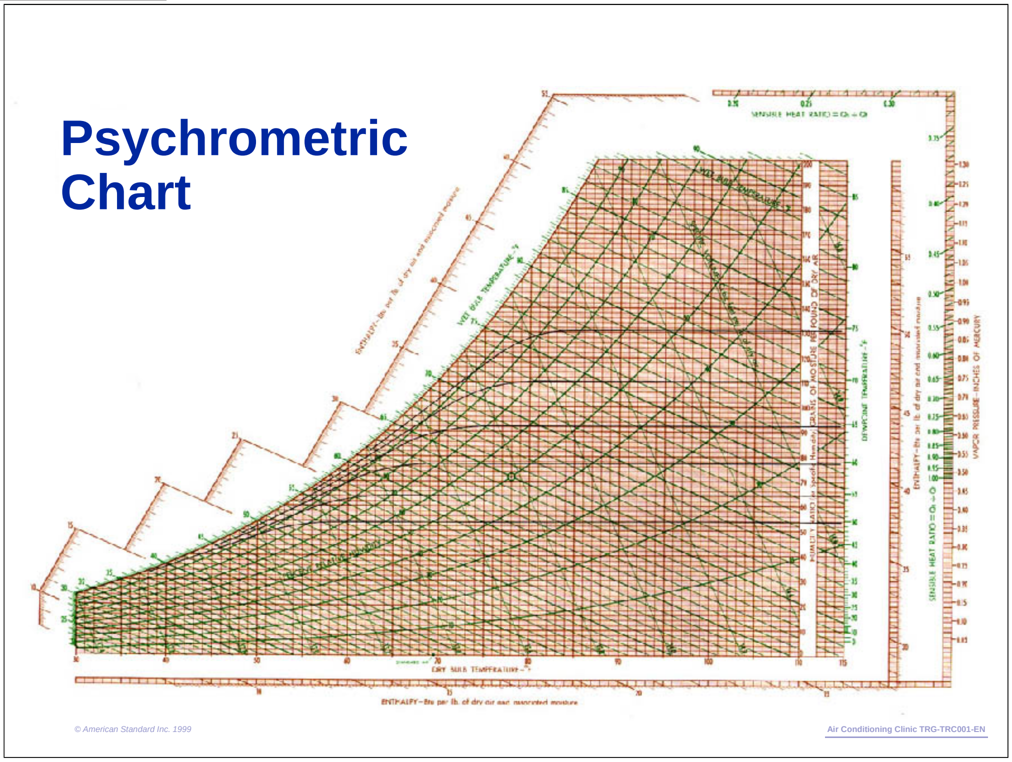
Psychrometry

period one

The Psychrometric Chart

Psychrometry is the science dealing with the physical laws of air – water mixtures.

When designing an air conditioning system, the temperature and moisture content of the air to be conditioned, and the same properties of the air needed to produce the desired air conditioning effect, must be known. Once these properties are known, the air conditioning task can be determined. This analysis can be performed using the **psychrometric chart**. The psychrometric chart graphically displays several physical properties of air over a broad range of conditions. Knowing the relationship of these air properties aids the task of air conditioning system design and analysis.



Properties of Air

At first glance, the psychrometric chart appears to be an imposing network of lines. When properly used, however, it provides valuable information about the properties of air. During this session, the psychrometric chart and its use in solving many air conditioning problems will be explained.

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Properties of Air

- Dry-bulb temperature
- Wet-bulb temperature
- Dew-point temperature
- Relative humidity
- Humidity ratio

The psychrometric chart contains five physical properties to describe the characteristics of air:

- Dry-bulb temperature
- Wet-bulb temperature
- Dew-point temperature
- Relative humidity
- Humidity ratio

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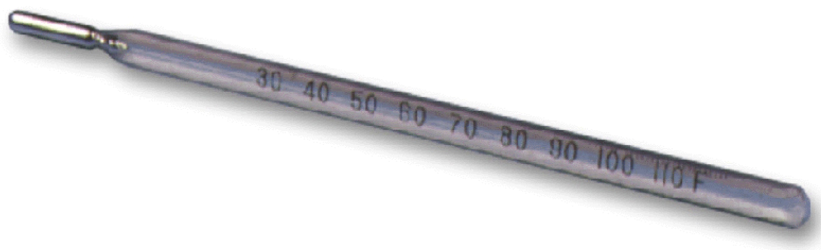
Dry-Bulb Thermometer



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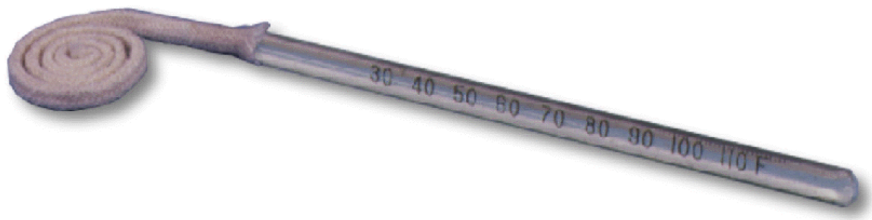
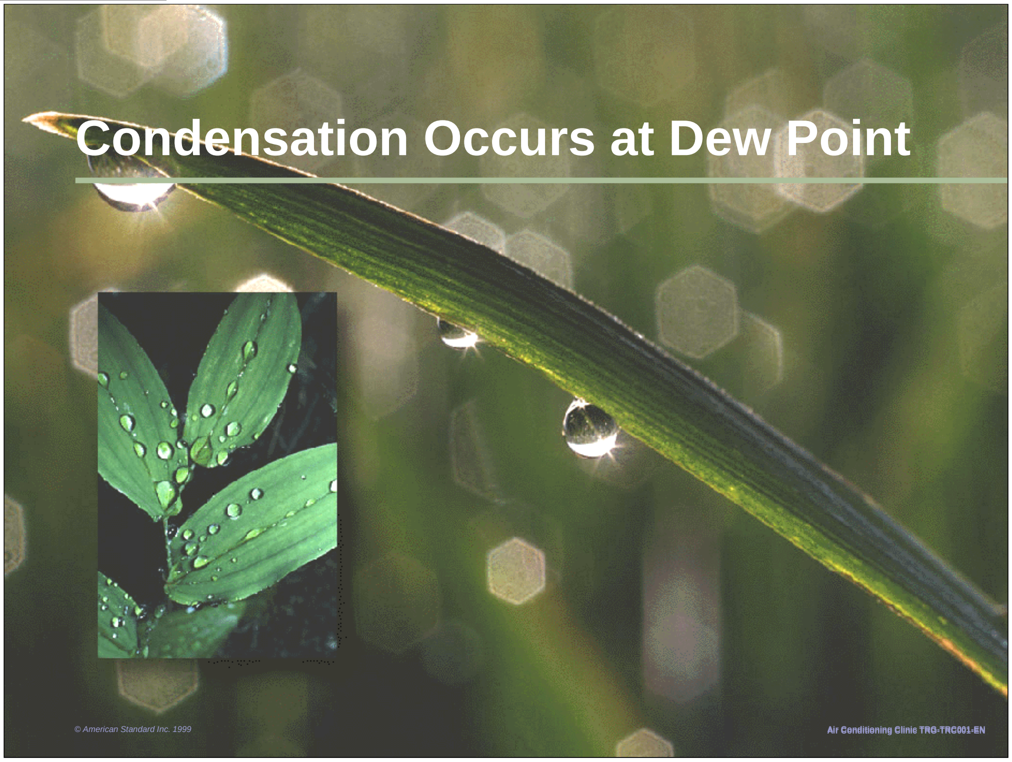
Dry-bulb temperatures are read from an ordinary thermometer that has a dry bulb.

Wet-Bulb Thermometer



Wet-bulb temperatures are read from a thermometer whose bulb is covered by a wet wick. The difference between the wet-bulb temperature and the dry-bulb temperature is caused by the cooling effect produced by the evaporation of moisture from the wick. This evaporation effect reduces the temperature of the bulb and, therefore, the thermometer reading.

Consequently, the difference between dry-bulb and wet-bulb temperature readings is a measure of the dryness of air. The drier the air, the greater the difference between the dry-bulb and wet-bulb readings.



The third property, **dew-point temperature**, is the temperature at which moisture leaves the air and condenses on objects, just as dew forms on grass and plant leaves.

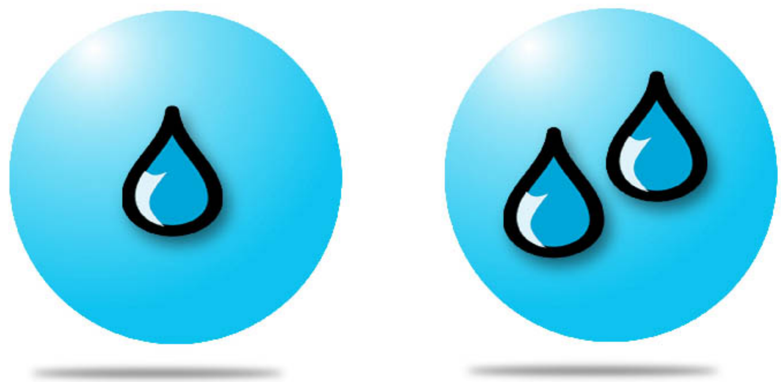
When the dry-bulb, wet-bulb, and dew-point temperatures are the same, the air is **saturated**. It can hold no more moisture. When air is at a saturated condition, moisture entering the air displaces moisture within the air. The displaced moisture leaves the air in the form of fine droplets. When this condition occurs in nature, it is called fog.

Relative Humidity

Relative Humidity = $\dfrac{\text{Amount of moisture that a given amount of air is holding}}{\text{Amount of moisture that a given amount of air can hold}}$

The fourth property, **relative humidity**, is a comparison of the amount of moisture that a given amount of air *is* holding, to the amount of moisture that the same amount of air *can* hold, at the same dry-bulb temperature.

Relative Humidity



Relative humidity is expressed as a percentage. For example, if the relative humidity of the air is 50%, it contains one-half the amount of moisture possible at the existing dry-bulb temperature.

Humidity Ratio

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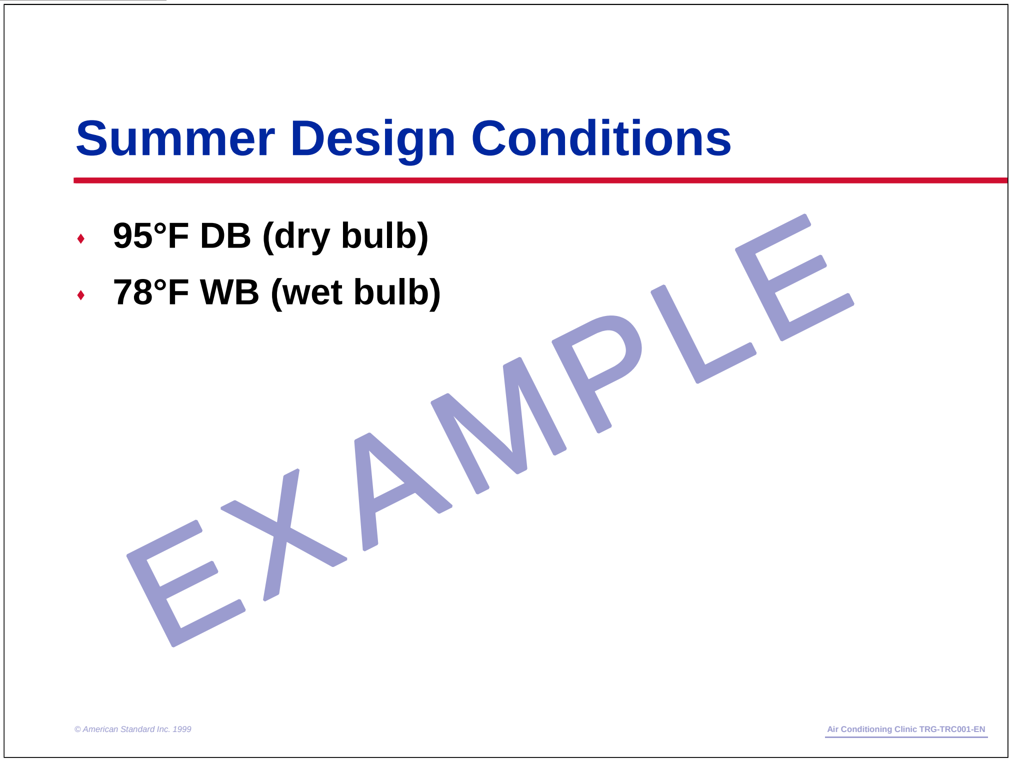
Finally, **humidity ratio** describes the actual weight of water in an air – water vapor mixture. In other words, if one pound of air were wrung completely dry, comparing the weight of the water to the weight of the dry air would yield its humidity ratio.

Humidity ratio can be expressed as pounds of moisture per pound of dry air, or as grains of moisture per pound of dry air. There are 7000 grains of water in a pound. To appreciate the magnitude of these units of measurement, at sea level one pound of 70°F air occupies approximately 13.5 cubic feet, and one grain of water in that air weighs about two-thousandths (0.002) of an ounce.

Properties of Air

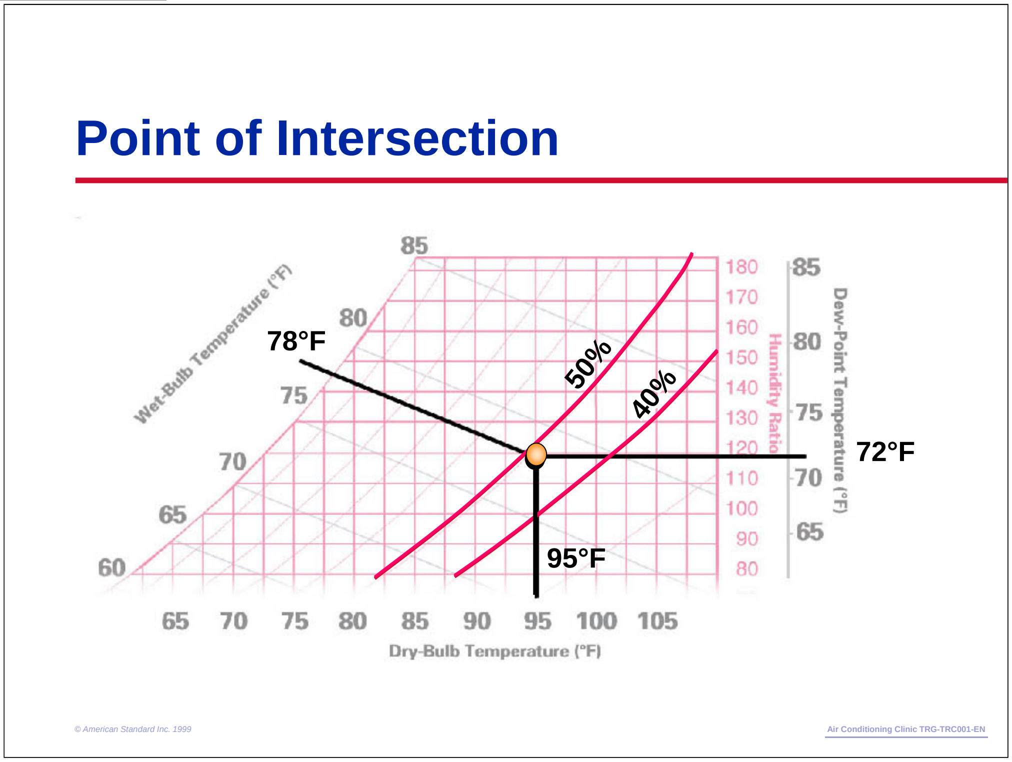
- Dry-bulb temperature
- Wet-bulb temperature
- Dew-point temperature
- Relative humidity
- Humidity ratio

When any two of these five properties of air are known, the other three can be quickly determined from the psychrometric chart.



For example, let's assume that the summer design conditions are 95°F dry bulb and 78°F wet bulb.

What is the relative humidity, humidity ratio, and dew point?



Only one point on the psychrometric chart represents air with both of these conditions. This point is located where the vertical 95°F dry-bulb (DB) and diagonal 78°F wet-bulb (WB) temperature lines intersect.

From this intersection, the remaining three air properties can be read from the chart. Both the dew-point and humidity-ratio lines are horizontal and the values are shown on the right side of the chart. In this example, the humidity ratio is about 118 grains of moisture per pound of dry air and the dew-point temperature is approximately 72°F.

Notice that the point of intersection falls between two relative humidity curves: 40% and 50%. By interpolation, the relative humidity at this condition is approximately 47%.

Properties of Saturated Air

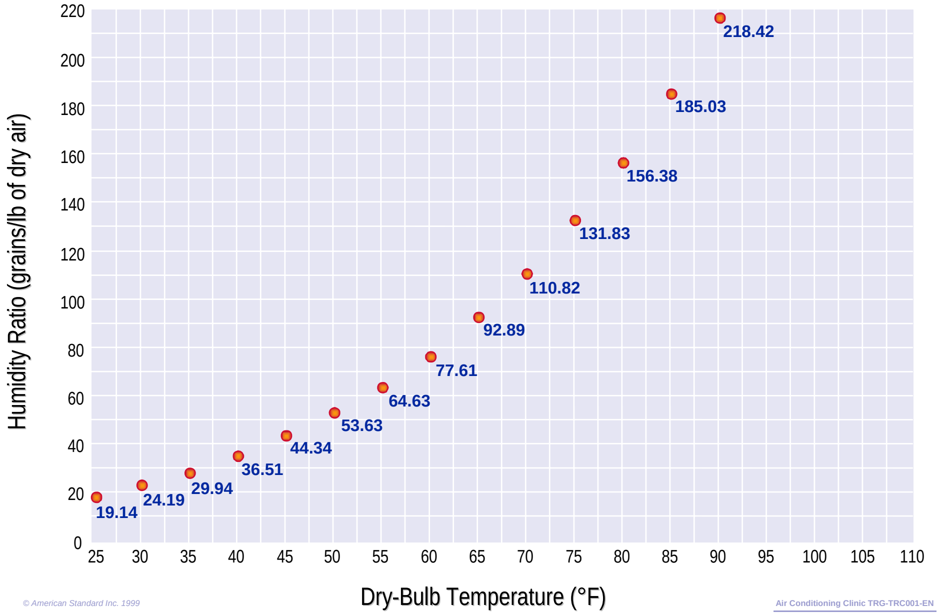
dry-bulb temp.	humidity ratio	dry-bulb temp.	humidity ratio
25°F	19.14	60°F	77.61
30°F	24.19	65°F	92.89
35°F	29.94	70°F	110.82
40°F	36.51	75°F	131.83
45°F	44.34	80°F	156.38
50°F	53.63	85°F	185.03
55°F	64.63	90°F	218.42

Constructing a Simple Psychrometric Chart

To better understand the psychrometric chart and show why the lines intersect as they do, we will construct a simple chart.

The amount of moisture contained in saturated air depends on dry-bulb temperature. This table shows the maximum amount of water vapor that one pound of dry air can hold at various dry-bulb temperatures. For example, at 25°F, one pound of dry air can absorb and hold 19.14 grains of water; at 30°F it can absorb 24.19 grains; at 35°F it can absorb 29.94 grains; and so on. Each of these conditions is a **saturation point**.

Plotting Saturation Points



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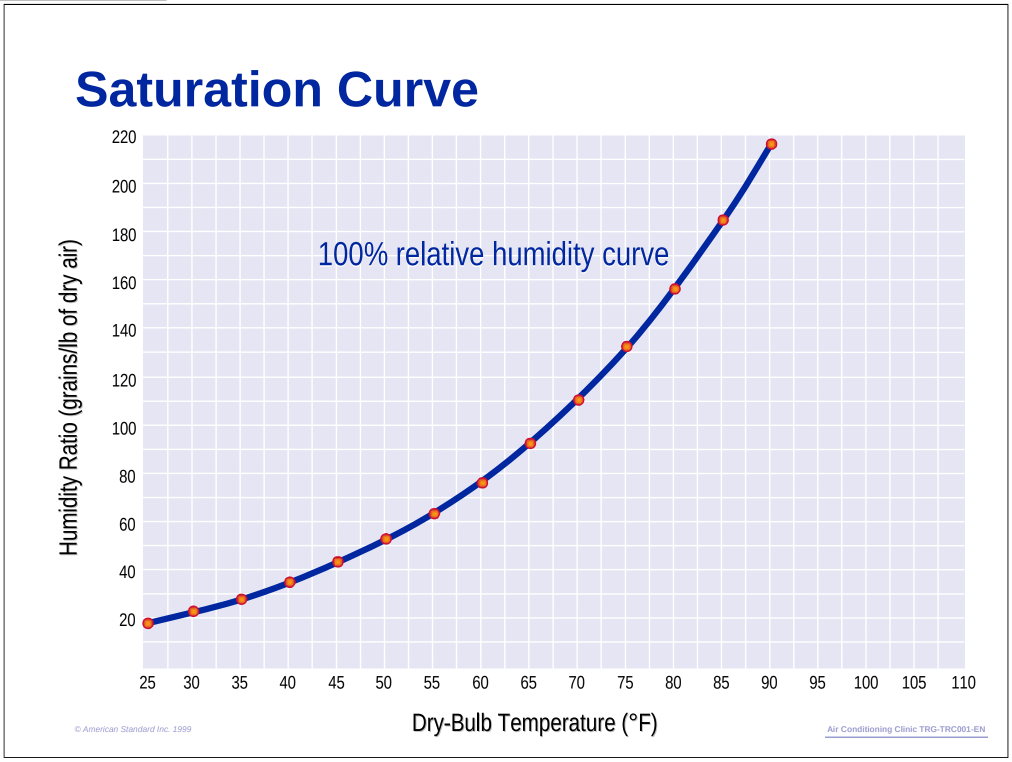
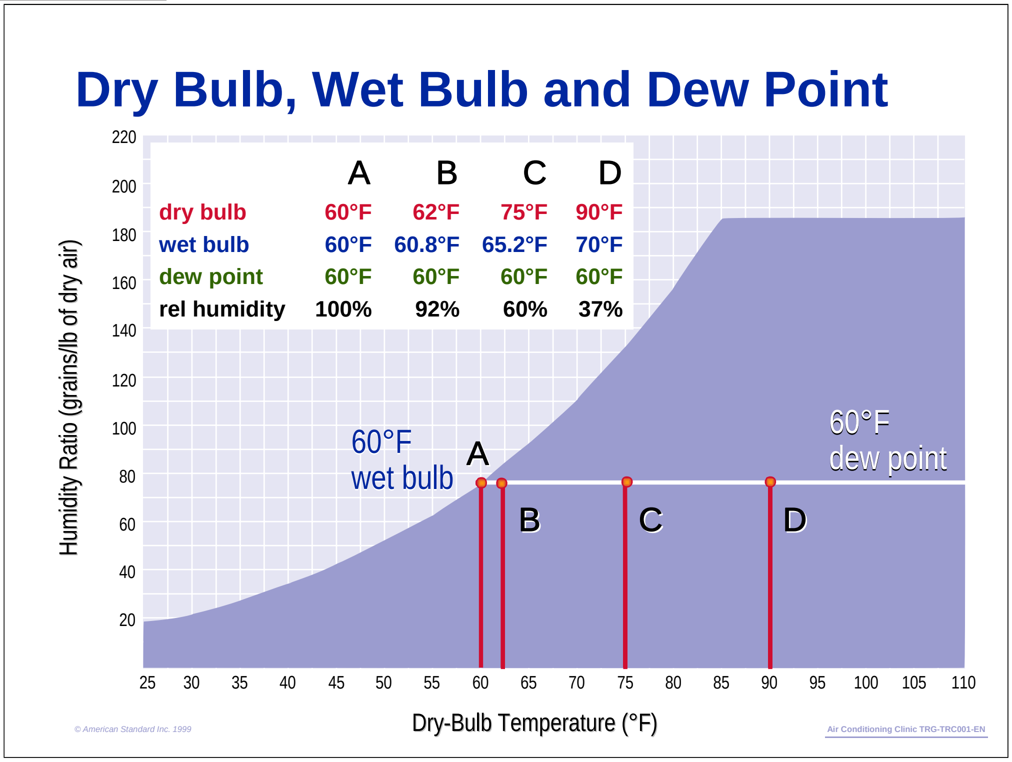
These saturation points can be plotted on a chart with dry-bulb temperature along the horizontal axis and humidity ratio along the vertical axis.

When several saturation points are plotted, the curve created resembles the relative humidity curves of the psychrometric chart.



In fact, this curve forms the 100% relative-humidity curve or **saturation curve**. All points on this curve represent the moisture content that constitutes complete saturation of air at the various dry-bulb temperatures.



Another fact about saturated air should be discussed before we proceed. Assume a volume of moist air has the initial conditions indicated in column D of the table. The air has a 90°F dry-bulb temperature and a 60°F dew point. A wet-bulb thermometer shows the wet-bulb temperature to be 70°F. From a moisture-content table, the relative humidity of the air is approximately 37%.

With no change in the moisture content of this volume of air, the table shows the progressive change that occurs as the air cools.

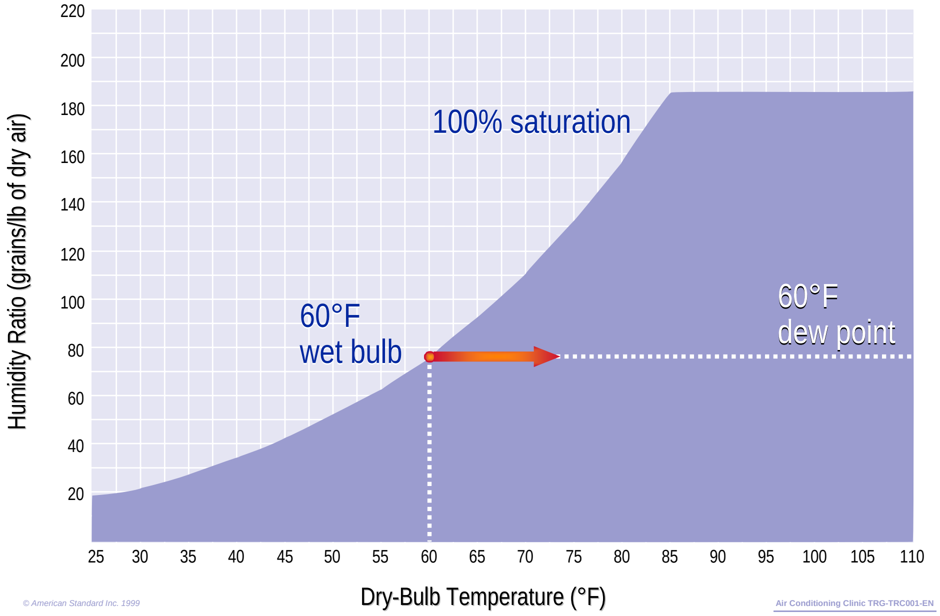
Point C: As the dry-bulb temperature drops from 90°F to 75°F, the wet-bulb temperature drops from 70°F to 65.2°F, yet the dew point remains the same at 60°F. The relative humidity rises from 37% to 60%.

Point B: When the dry-bulb temperature reaches 62°F, the wet-bulb temperature is about 60.8°F, the dew point remains constant at 60°F, and the relative humidity is 92%.

Point A: Finally, when the dry-bulb temperature reaches the 60°F dew-point temperature, the wet bulb cannot be reduced any lower because evaporation can no longer occur: the air is saturated and contains all the moisture it can hold. The relative humidity is now 100%.

At any point on a 100% relative-humidity curve, the three air temperatures—dry bulb, wet bulb and dew point—are equal.

Adding Sensible Heat



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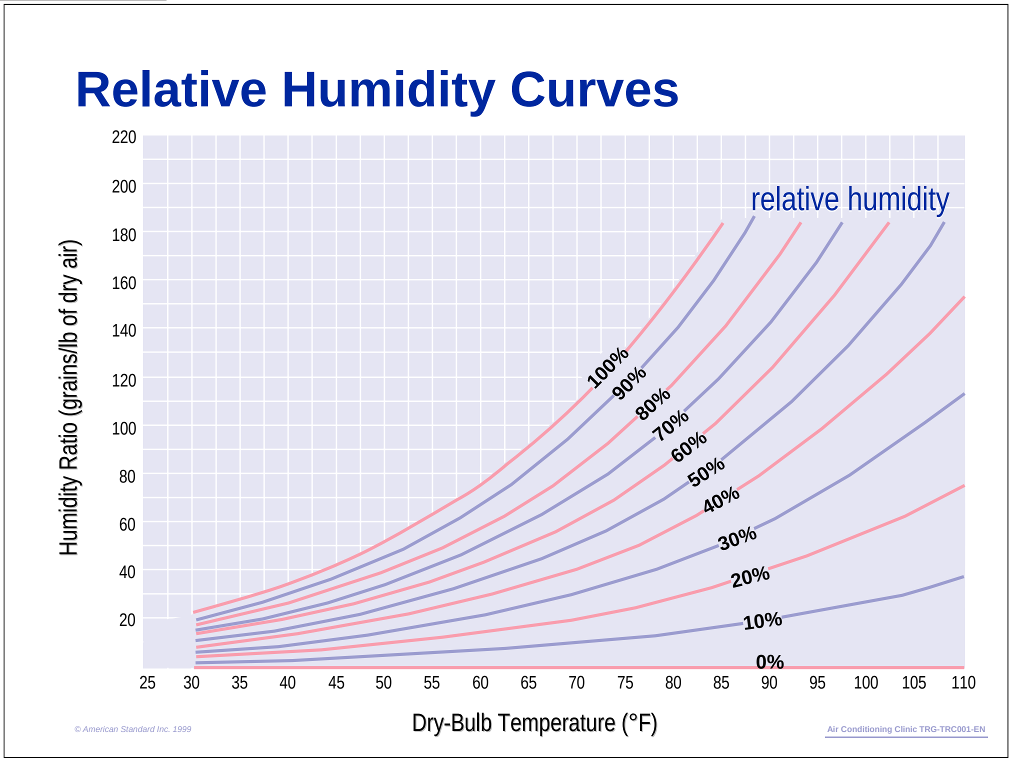
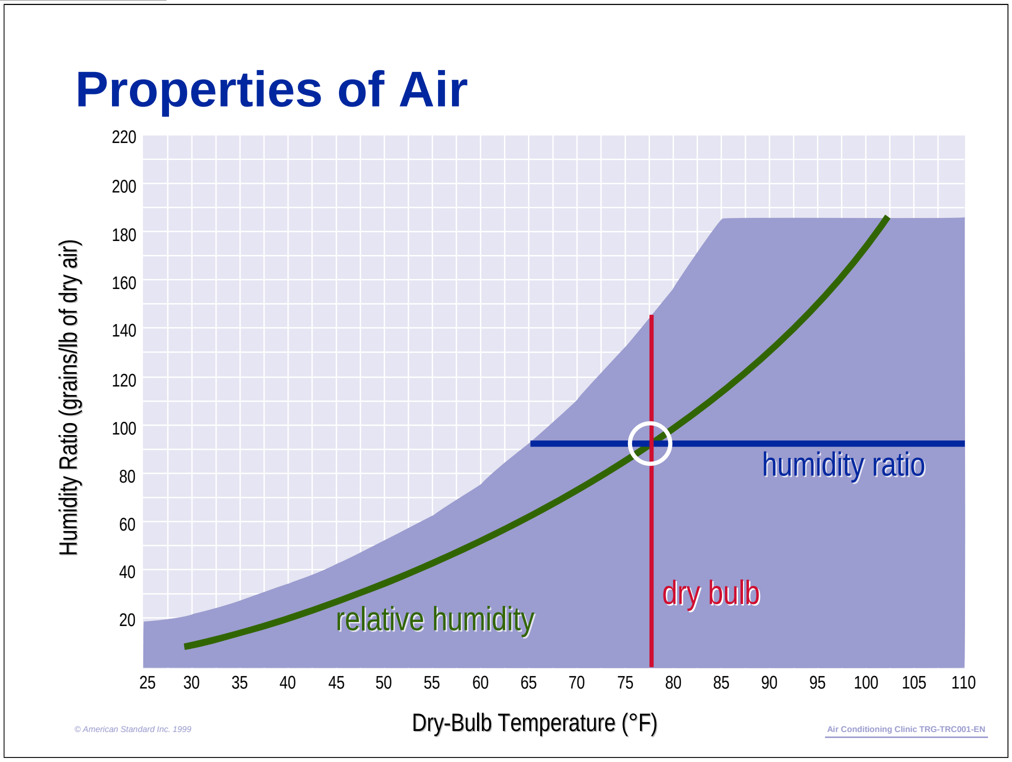
Additionally, the dew-point temperature does not change as the dry-bulb temperature changes, provided that the moisture content of the air remains the same. Merely heating the air does not change its moisture content. Therefore, as the air is heated, its condition will move horizontally along a constant humidity-ratio line.

In this example, heating 60°F saturated air moves the air condition along a horizontal humidity-ratio line that corresponds to a constant 77.56 grains of moisture per pound of dry air.

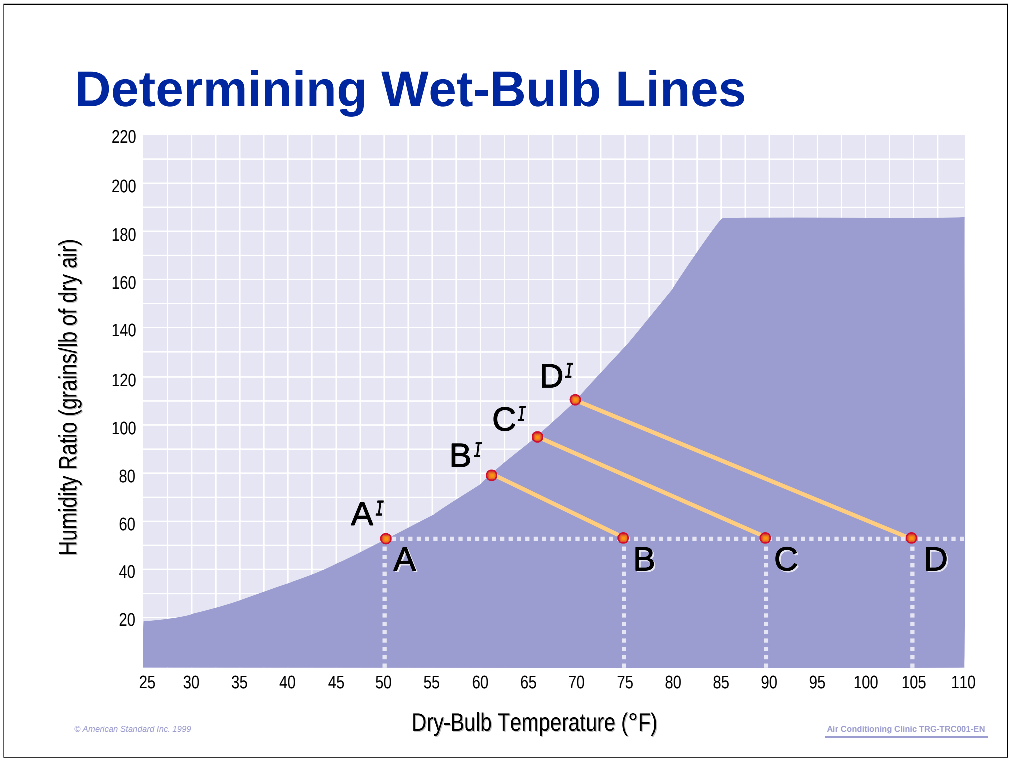
Horizontal lines can be drawn from each saturation point across to the right side of the chart. A horizontal line can be provided for each humidity-ratio value.



Additional curves can be added to the chart to represent relative humidity conditions that are less than 100%. The curves shown are at 10% intervals and represent humidity conditions ranging from completely saturated air to completely dry air. When air is completely dry, its relative humidity cannot change with temperature. The 0% condition is therefore represented by the horizontal axis of the chart.



This basic chart now shows three air conditions: dry-bulb temperature (vertical lines), humidity ratio (horizontal lines) and relative humidity (diagonally curved lines).

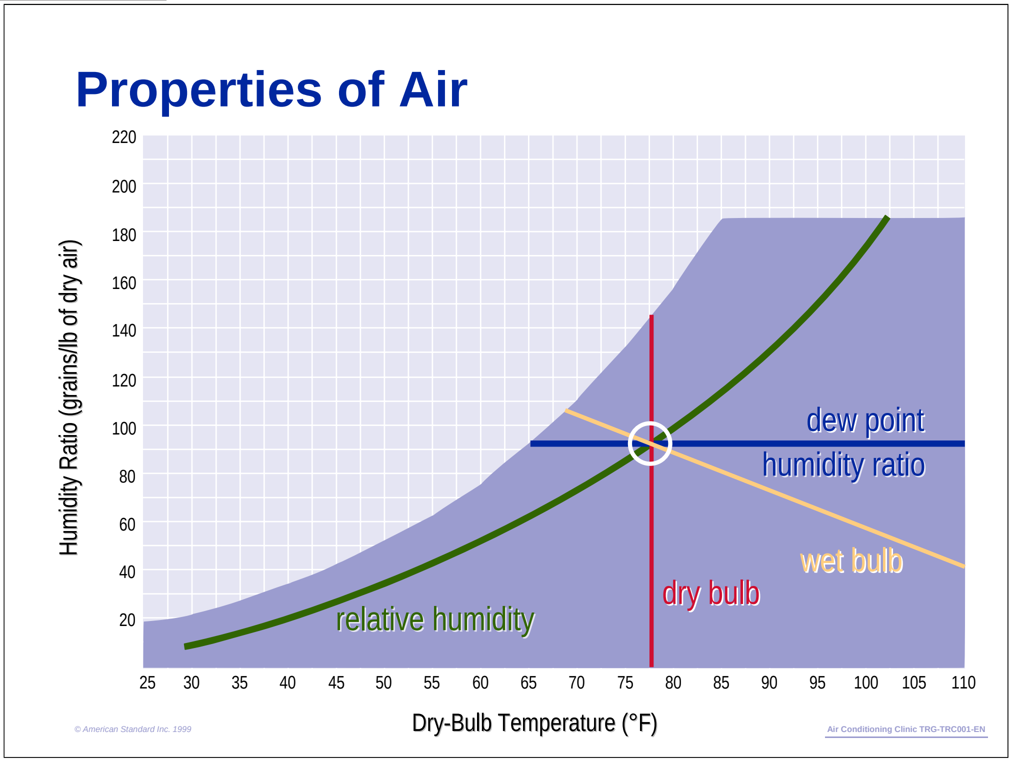


To complete this basic chart, the wet-bulb temperature lines must be added.

Once again, at a saturated condition the wet-bulb, dry-bulb and dew-point temperatures are equal. Therefore, the wet-bulb temperature lines start at the saturation curve.

To observe what happens to wet-bulb temperatures when air is heated, start with saturated air at 50°F dry bulb. At this condition, the air has a moisture content of approximately 54 grains per pound as shown by A. If the temperature of this air is increased to 75°F dry bulb without changing its moisture content, the air condition will move along the constant humidity ratio line (54 grains/lb) to B. The wet-bulb temperature of this warmed air is approximately 60.1°F.

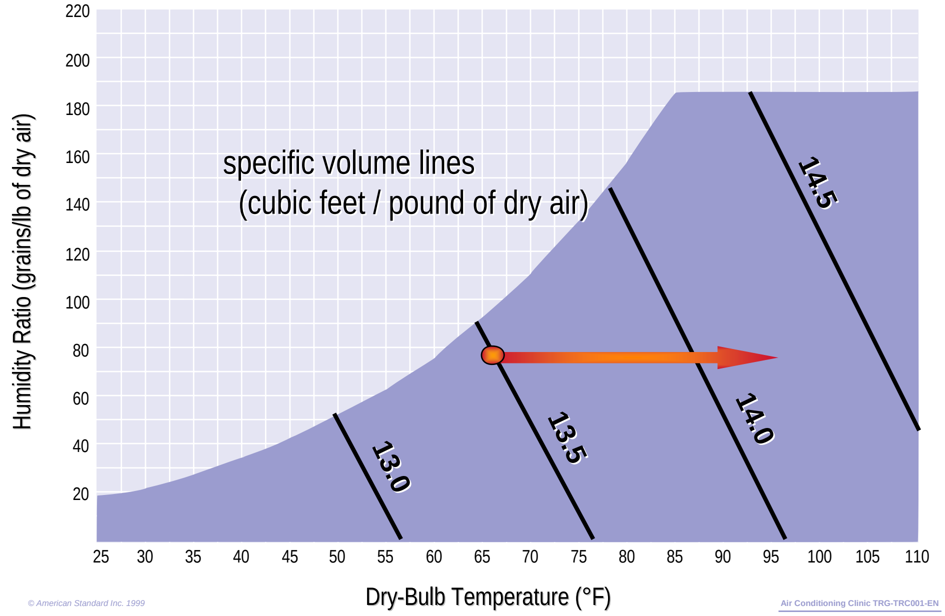
A line drawn from B to a point on the saturation curve that represents 60.1°F saturation temperature (B') gives an indication of the direction the wet-bulb temperature lines will run. By taking numerous wet-bulb readings under different conditions, the wet-bulb temperature lines can be added to the chart.



The psychrometric chart now defines these five properties of air: dry-bulb temperature (vertical lines), humidity ratio and dew-point temperature (horizontal lines), relative humidity (curved lines) and wet-bulb temperature (diagonal lines).

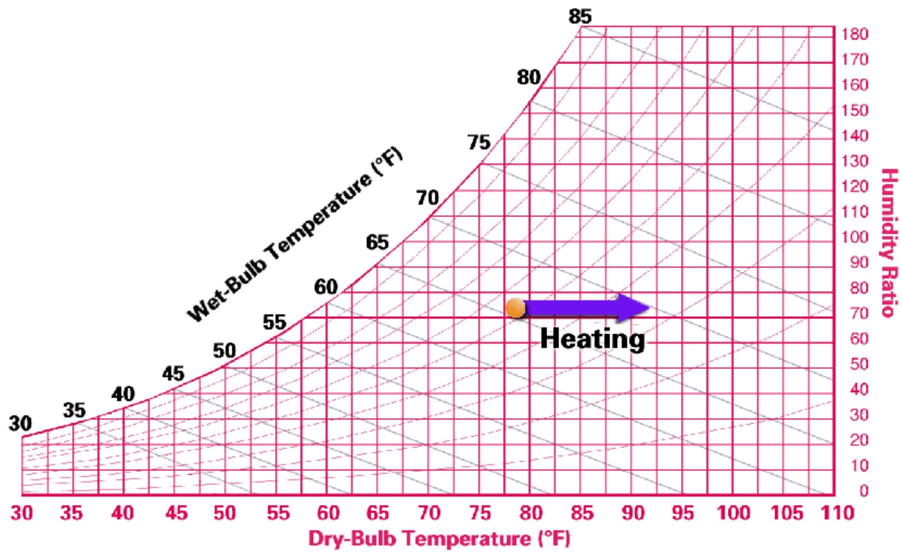
Remember: if any two of these five air conditions are known, the other three can be found on the psychrometric chart by locating the point of intersection of the two known conditions.

Specific Volume



There is one more property of air that is displayed on the psychrometric chart—specific volume. **Specific volume** is defined as the volume of one pound of dry air at a specific temperature and pressure. As one pound of air is heated it occupies more space—the specific volume increases.

Effect of Adding Sensible Heat



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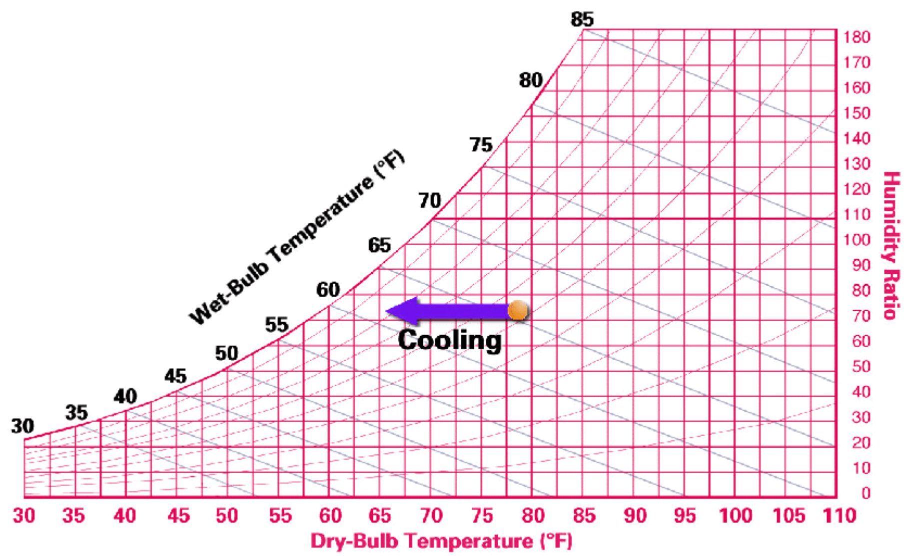
Effect of Sensible Heat and Moisture Changes

When either the sensible heat content or the moisture content of air changes, the point on the psychrometric chart that represents the original air condition moves to a position that represents the new condition of temperature and/or humidity.

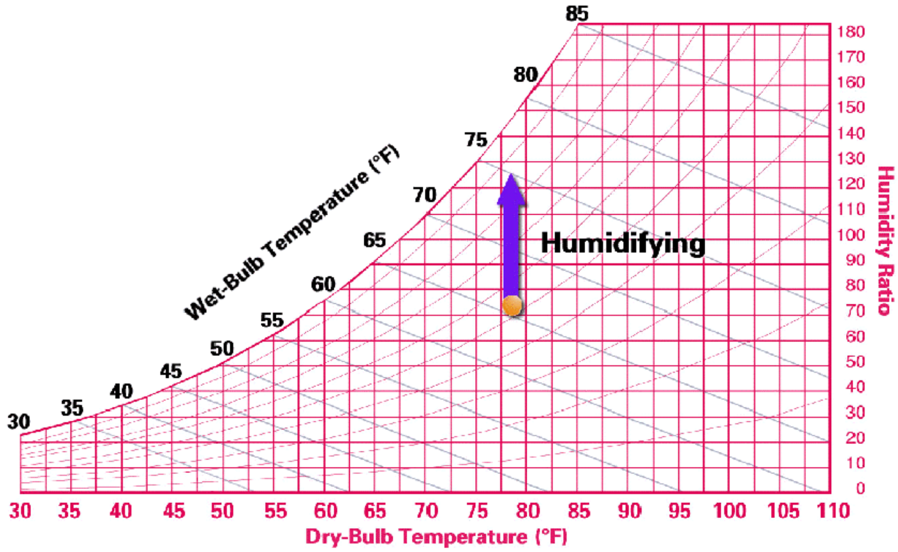
For example, if sensible heat is added to air, the air condition moves horizontally to the right.

Effect of Removing Sensible Heat



Conversely, if sensible heat is removed from air, the air condition moves horizontally to the left. As long as the moisture content of the air remains unchanged, the humidity ratio remains the same. Therefore, this movement follows the horizontal humidity-ratio lines.

Effect of Adding Moisture



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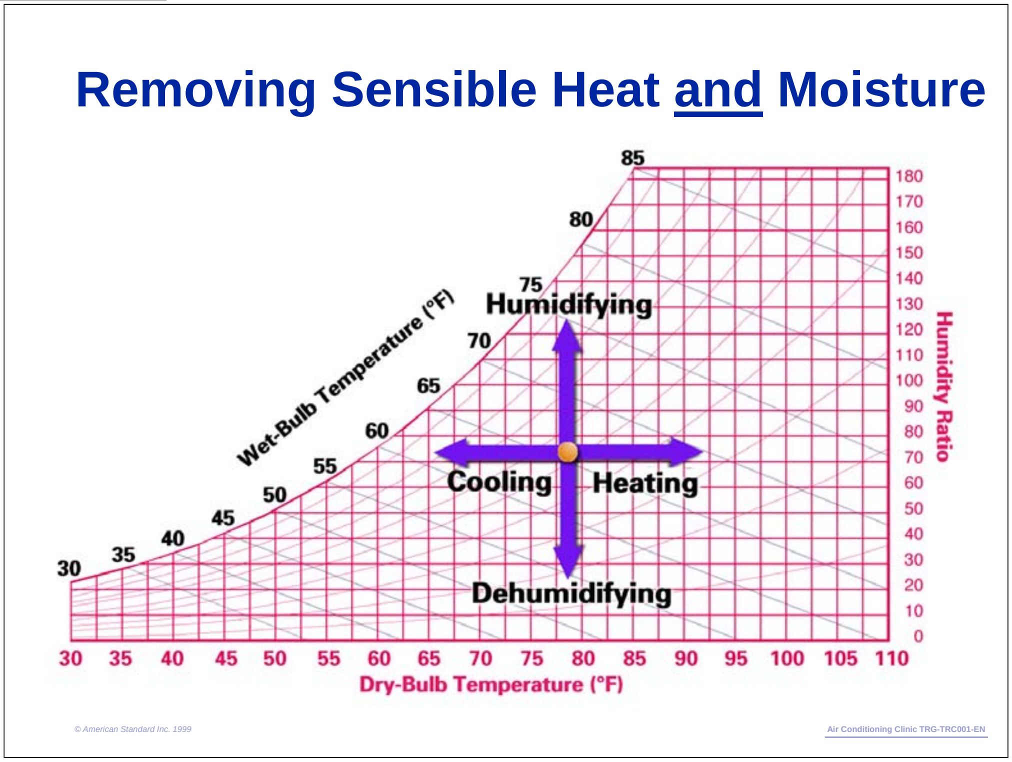
On the other hand, if moisture is added to air without changing the dry-bulb temperature, the air condition moves upward along a dry-bulb temperature line.

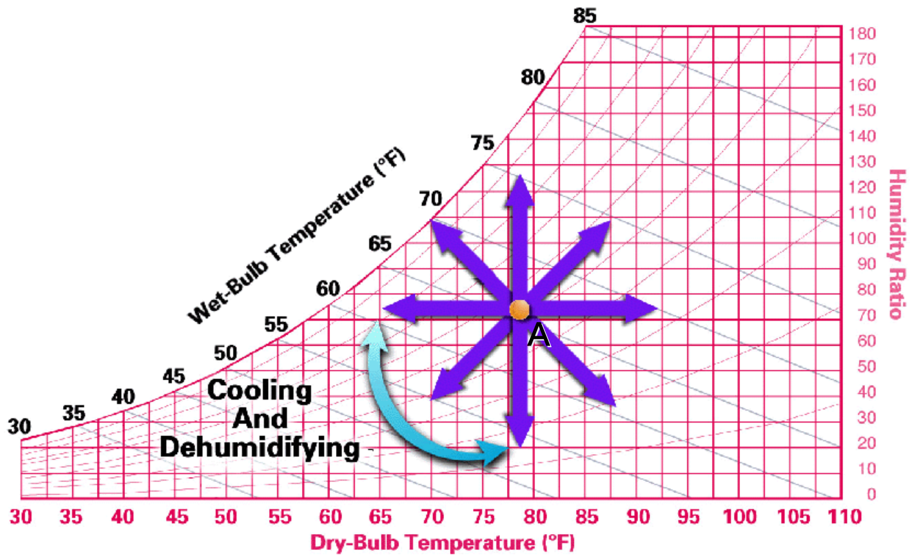
Effect of Removing Moisture

Finally, if moisture is removed from the air without changing its dry-bulb temperature, the air condition moves downward along a dry-bulb temperature line.



Put all of these changes together on one chart and they show the direction the air condition will move when the dry-bulb temperature or moisture content is altered.

Removing Sensible Heat and Moisture



In actual practice, however, both the dry-bulb temperature and moisture content of the air generally change simultaneously. When this happens, the resulting air conditions move from A at some angle. The exact angle and direction depend upon the proportions of sensible and latent heat added or removed. **Sensible heat** causes a change in the air's dry-bulb temperature with no change in moisture content. **Latent heat** causes a change in the air's moisture content with no change in dry-bulb temperature.

To provide summer comfort, air is cooled and dehumidified, moving the air condition downward and to the left, resulting in a lower dry-bulb temperature and a lower moisture content.

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Psychrometry

period two

Air Mixtures

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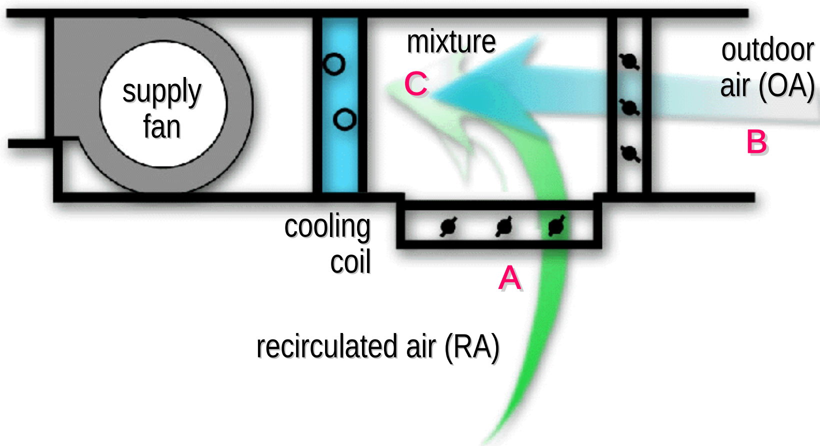
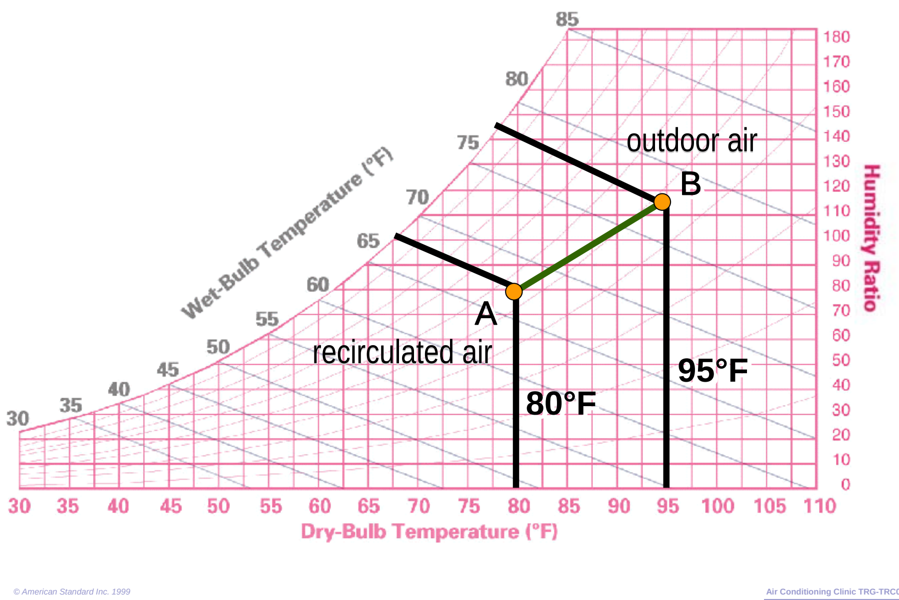
Before an air conditioning problem can be analyzed on the psychrometric chart, the conditions of the air to be cooled or heated must be known.

Determining Entering Air Conditions



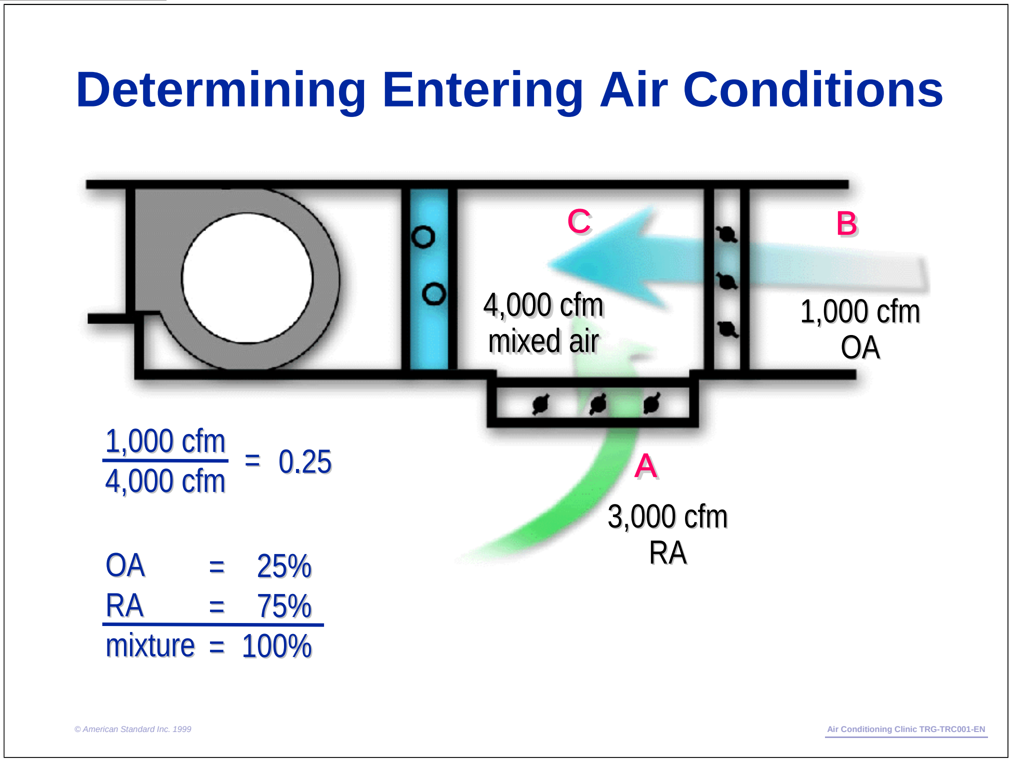
The air entering the cooling coil may be 100% recirculated (A), 100% outdoor (B), or a mixture of the two (C).

Determining Entering Air Conditions



If outdoor air (B) is mixed with recirculated air (A), the conditions of the resulting mixture are found somewhere on a straight line connecting the two points.

If the mixture is half and half, this condition falls on the midpoint of the line between A and B. If more than half of the mixture is recirculated air (A), the condition of the mixture will fall closer to A than to B.



In this example, 1,000 cfm of outdoor air (OA) is mixed with 3,000 cfm of recirculated air (RA) for a total supply airflow of 4,000 cfm.

First, the percentage of outdoor air within the mixture is determined. This is done by dividing the outdoor air quantity by the total air quantity.

$$\frac{1{,}000\ \text{cfm}}{4{,}000\ \text{cfm}} = 0.25$$

The outdoor air quantity in this example constitutes 25% of the mixture, while the recirculated air makes up the remaining 75%.

The next step is to determine the dry-bulb temperature of the air mixture.

Determining Entering Air Conditions

$$
\begin{aligned}
95°F \times 0.25 &= 23.75°F \\
80°F \times 0.75 &= 60.00°F \\
\hline
\text{mixture} &= 83.75°F
\end{aligned}
$$

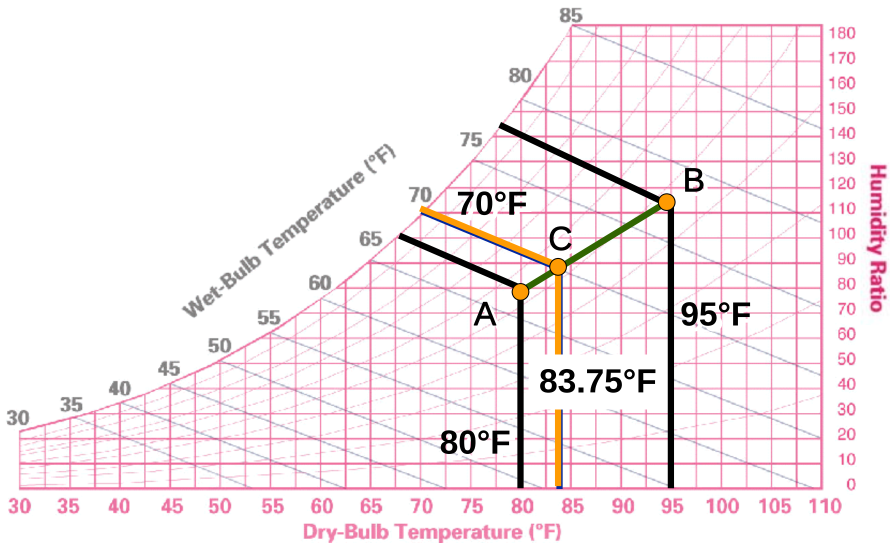
This is done by multiplying the dry-bulb temperature of each air condition byits percentage and summing the results.

For example, if the outdoor dry-bulb temperature is 95°F and it represents 25% of the air mixture, it contributes 23.75°F (or 0.25 × 95) to the dry-bulb temperature of the air mixture.

Similarly, if the dry-bulb temperature of the recirculated air is 80°F, it contributes 60°F (or 0.75 × 80) to the dry-bulb temperature of the air mixture.

The sum of 23.75°F and 60°F equals 83.75°F, the resulting dry-bulb temperature of this air mixture.

Determining Entering Air Conditions



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Returning to the psychrometric chart, point C, at which the 83.75°F dry-bulb temperature falls on the line from A to B, represents the conditions of the air mixture: 83.75°F DB and approximately 70°F WB.

Because the recirculated air quantity constitutes a larger percentage (75%) of the mixture, the mixed-air condition (C) is much nearer to the indoor design condition (A) than to the outdoor design condition (B).

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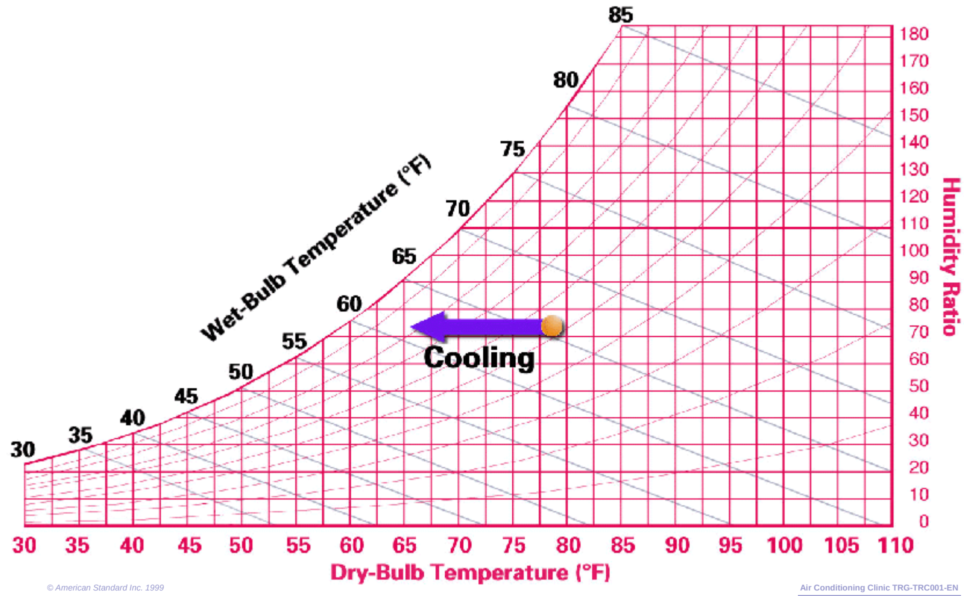
Psychrometry

period three

Sensible Heat Ratio

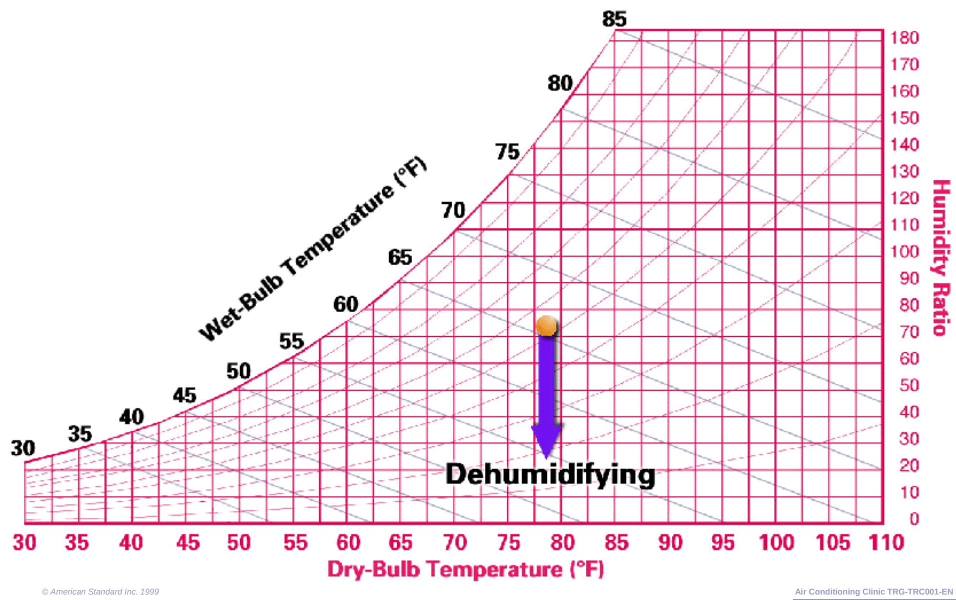
This period is devoted to understanding the term **sensible heat ratio** and how it is represented on the psychrometric chart. The ratio of sensible heat gain to total heat gain is one of the most important factors affecting air conditioning system requirements.

Effect of Removing Sensible Heat



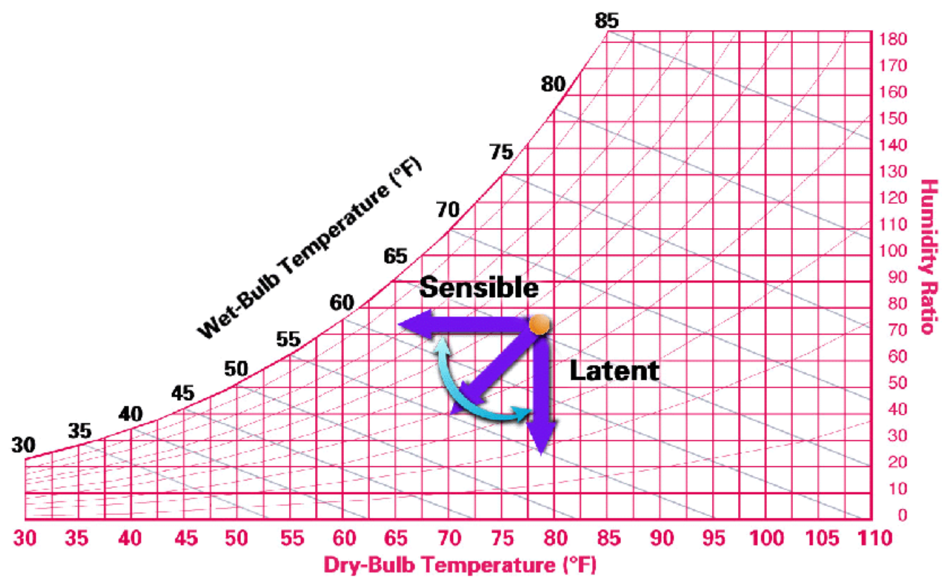
If only sensible heat is removed from the air, the line representing this change moves from the original condition horizontally to the left. This results in a lower dry-bulb temperature, while the moisture content (the humidity ratio) remains constant.

Effect of Removing Latent Heat



Conversely, if only latent heat is removed, the line moves vertically downward along a constant dry-bulb temperature line. This results in a lower moisture content or humidity ratio.

Removing Sensible and Latent Heat



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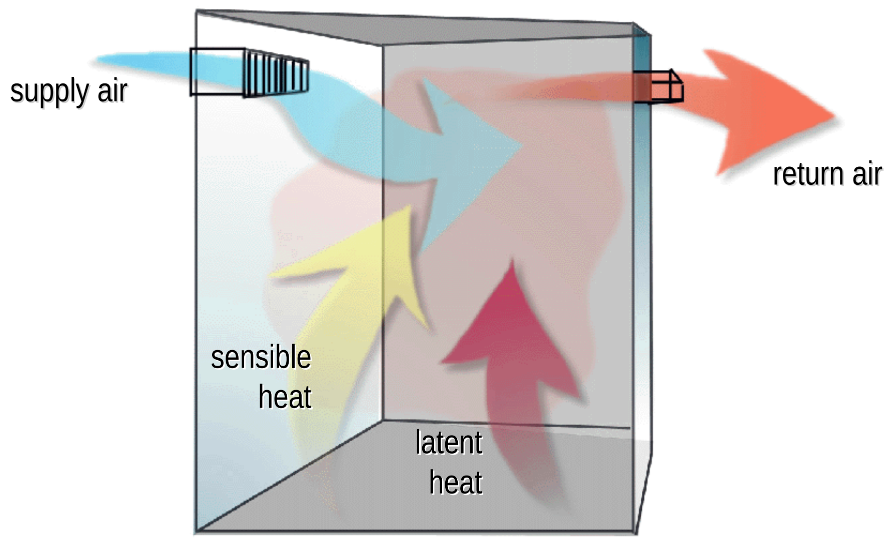
If both sensible and latent heat are removed from the air, the resulting air condition will be to the left and below the initial condition. The proportions of sensible and latent heat removed will determine the exact direction the resulting air condition follows.

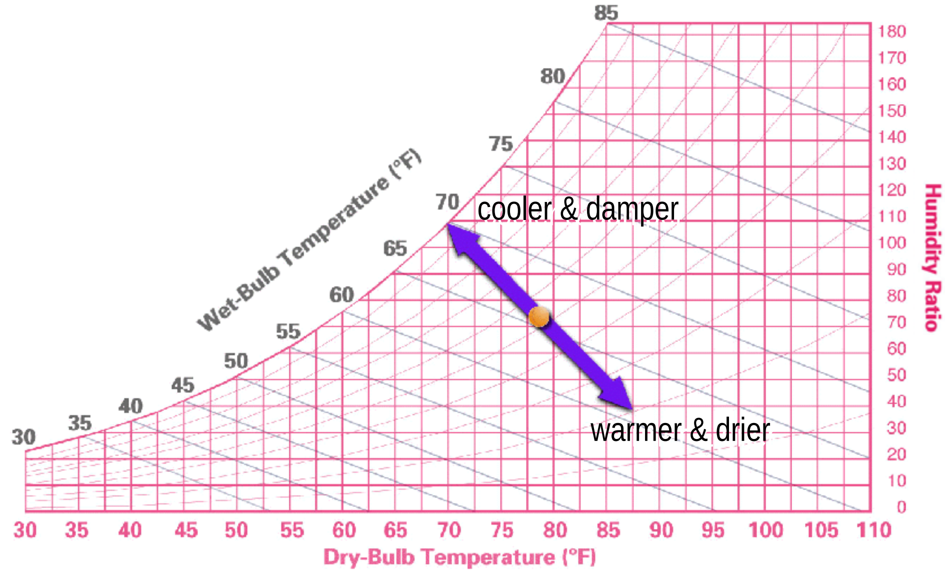
Heat and Moisture Transfer



Imagine conditioned supply air as a sponge. As it enters a room, it absorbs heat and moisture. The amount of heat and moisture absorbed depends on the temperature and humidity of the supply air. This "sponge," the supply air, must be cool enough to pick up the room's excess sensible heat gain and dry enough to pick up the room's excess latent heat (i.e. moisture.)

Therefore, the excess sensible and latent heat in the room determine the required dry-bulb and wet-bulb temperatures of that supply air.

Removing Sensible and Latent Heat

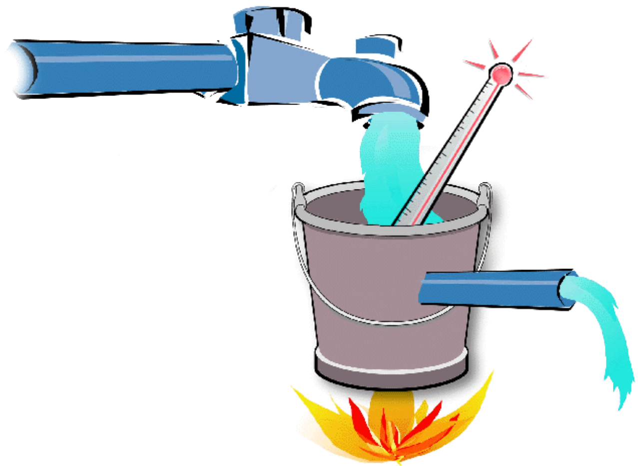


When the required amount of sensible and latent heat are not properly removed from the room, the desired room conditions cannot be maintained. For example, if too much sensible heat and not enough latent heat are removed, the room feels cold and damp. On the psychrometric chart, room conditions move up and to the left.

On the other hand, if too much latent heat but not enough sensible heat is removed, the room feels warm and dry. On the psychrometric chart, room conditions move down and to the right.

Therefore, the conditions of the supply air must be controlled accurately to ensure that both sensible and latent heat are removed in the proper proportions. There are several combinations of dry-bulb and wet-bulb temperatures that will produce the desired room conditions. Each of these combinations requires a different quantity of air.

Heat Gain



This relationship between the conditions and quantity of the supply air can be described using the analogy of maintaining a constant temperature within a container of water.

In this illustration, the container of water is capable of absorbing heat. The amount of heat it absorbs is called **heat gain**. To maintain the water temperature at a constant 75°F, any heat gain must be offset by mixing cool water with the water already in the container.

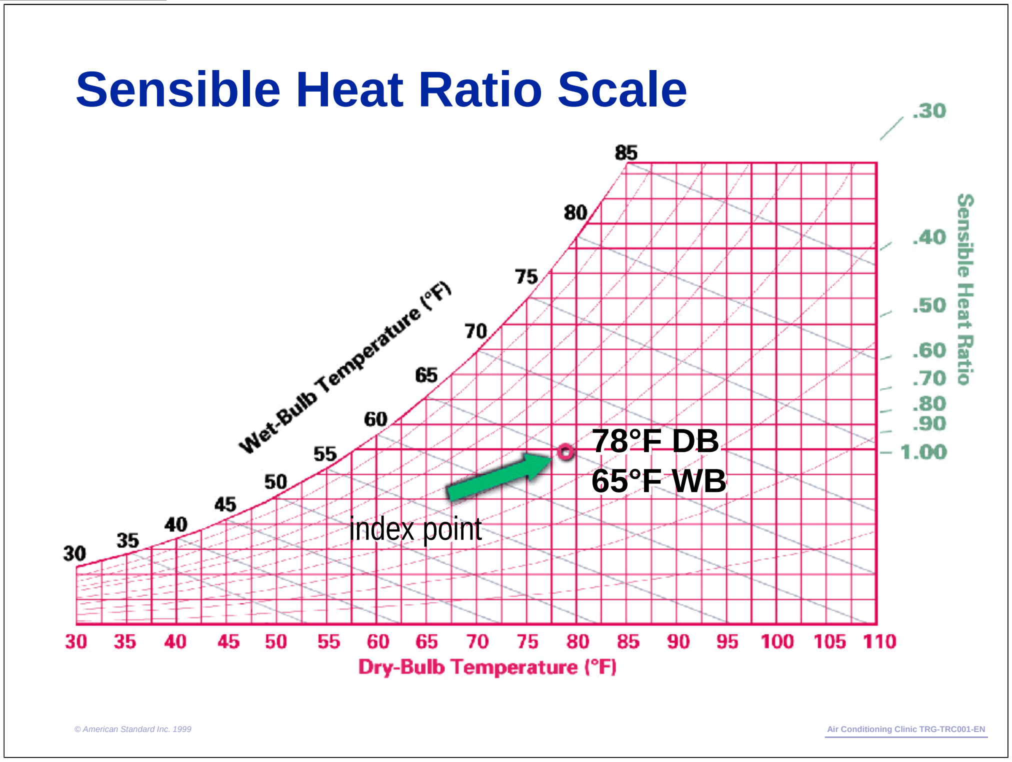
The rate at which this cool water is added to the container is determined by its temperature. For a given water temperature there is a certain flow rate—measured in gallons per minute (gpm)—that will offset the heat gain and maintain the desired temperature in the container. If the water is warm, a higher flow rate is required than if the water is very cold.

Sensible Heat Ratio (SHR)

$$\text{SHR} = \frac{\text{Sensible Heat Gain}}{\text{Sensible Heat Gain + Latent Heat Gain}}$$

The sensible heat ratio, abbreviated as SHR, refers to the comparison of sensible heat gain to total heat gain (sensible heat plus latent heat). Once this ratio is known, an SHR line can be drawn on the psychrometric chart.

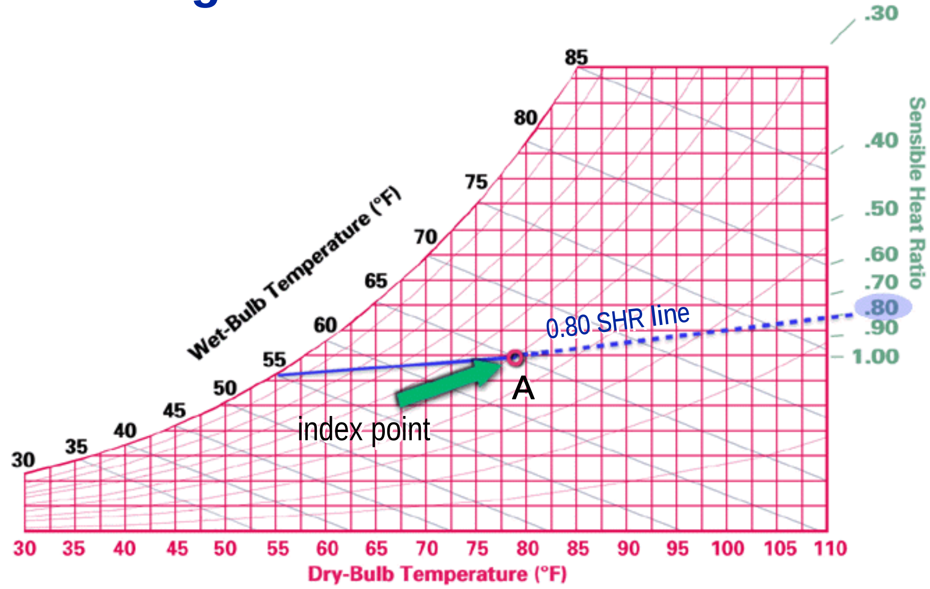
$$\text{Sensible Heat Ratio (SHR)} = \frac{\text{Sensible Heat Gain}}{\text{Sensible Heat Gain} + \text{Latent Heat Gain}}$$



A scale around the right and top edges of the chart gives the SHR values. Also, there is an **index point** in the middle of the chart at the 78°F DB and 65°F WB condition.

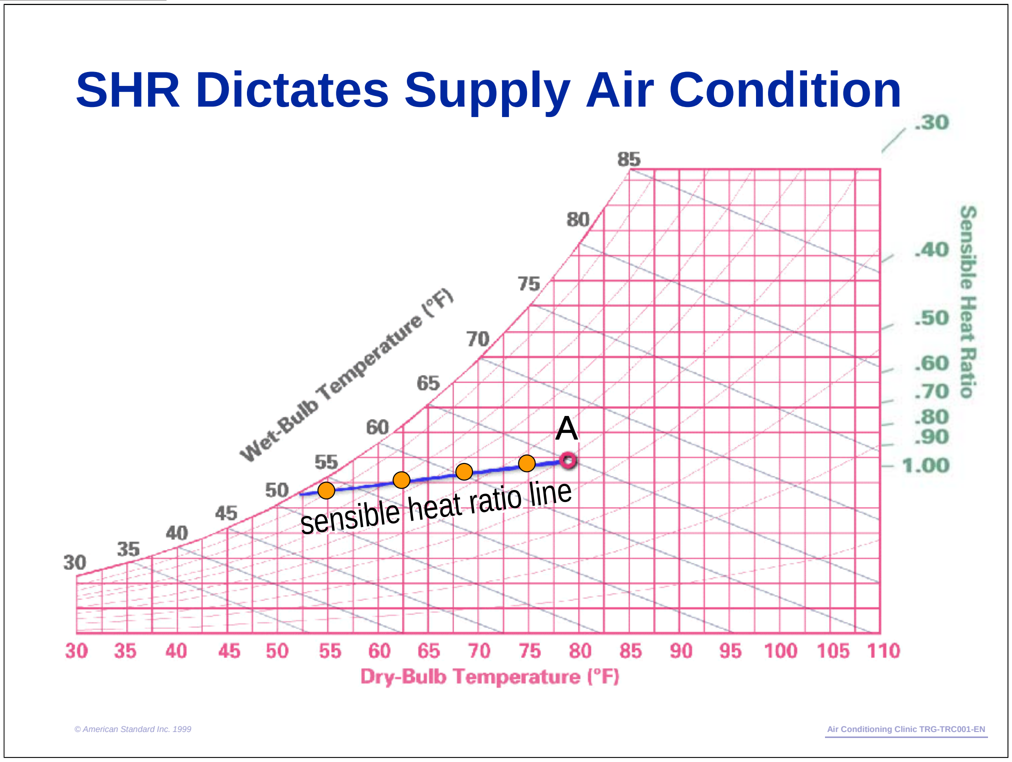
Using a straight edge, a sensible heat ratio line can be drawn by aligning the appropriate SHR value on the scale with the index point.

Drawing an SHR Line



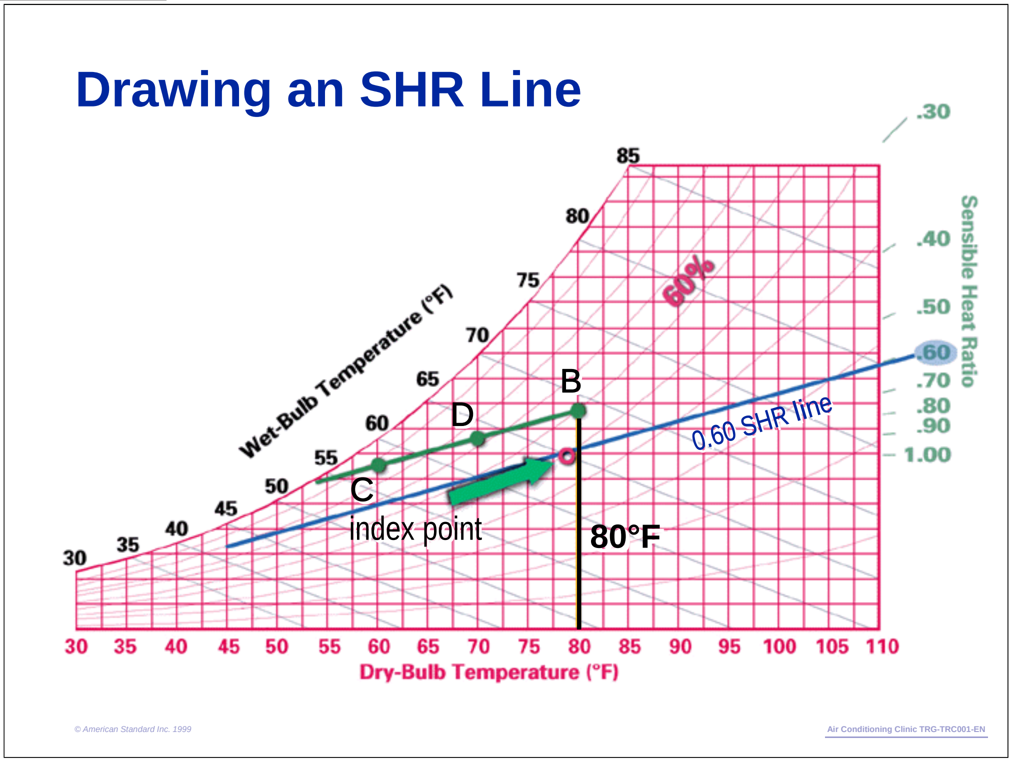
Assume that room design conditions (A) are 78°F DB and 65°F WB, and that the sensible heat ratio is calculated as 0.80. That is, sensible heat gain represents 80% of the total (sensible plus latent) heat gain.

The SHR line is found by aligning the index point with the 0.80 marking on the sensible-heat-ratio scale and drawing a line from the index point to the saturation curve.



Supply air with any combination of dry-bulb and wet-bulb temperatures that falls on this line will be able to absorb the room's sensible and latent heat in the correct proportions needed to maintain the desired room conditions (A = 78°F DB, 65°F WB).

Each of these combinations, however, requires a different quantity of air to do the task. Recall the analogy with the container of water. If the supply air is warm, a higher quantity of air is required than if the supply air is cold.



Sensible-heat-ratio lines for other conditions are drawn in the following manner. Assume that room design conditions are 80°F DB and 60% RH, and that the SHR is calculated as 0.60.

First, line up the index point with the 0.60 marking on the sensible-heat-ratio scale and draw a line. Next, draw a second line, parallel to the first, through the point (B) that represents the design room conditions. This is the 0.60 SHR line for a room at 80°F DB and 60% RH.

Supply air at C (60°F DB and 58°F WB) will maintain the desired room conditions, as will supply air at D (70°F DB and 64°F WB). To do so, each of these combinations will require a different quantity of air.

Psychrometry

period four

Air Quantity

Next, we will determine the flow rate of supply air necessary to maintain a given set of design room conditions.

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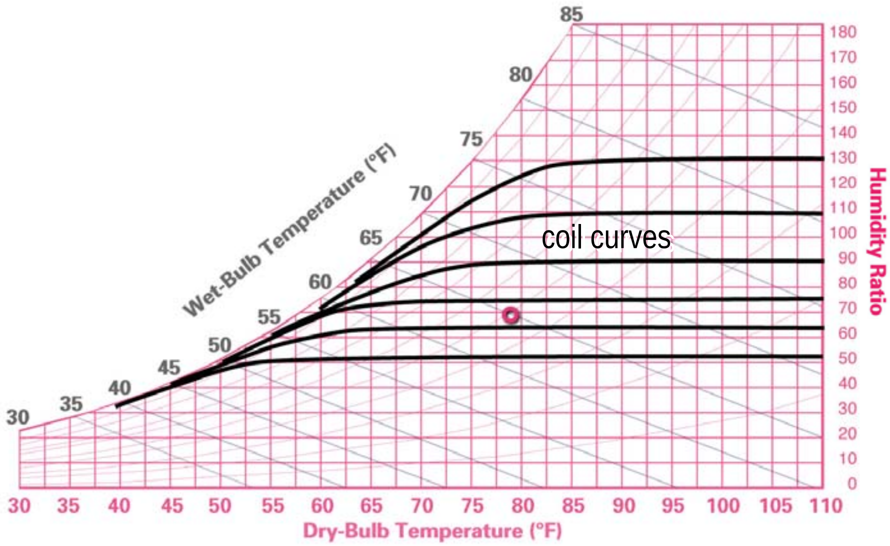
Coil Curves



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Before proceeding, one more set of curves on the psychrometric chart must be identified. These curved lines represent the changes in dry-bulb and wet-bulb temperatures as air passes through a “typical” cooling coil. These are commonly referred to as **coil curves**; they depict approximate coil performance. Exact coil performance depends on the actual coil geometry and can be precisely determined by coil selection software.

These curves were established from hundreds of laboratory tests of various coil geometries at different air and coolant temperatures. They let you determine leaving-coil conditions and postpone coil selection until the final design. The use of these coil curves will be discussed later.

Determining Supply Airflow

STEP 1: Calculate the sensible heat ratio (SHR)

80,000 Btu/hr sensible heat gain

20,000 Btu/hr latent heat gain

$$\text{SHR} = \frac{80{,}000 \text{ Btu/hr}}{100{,}000 \text{ Btu/hr}} = 0.80$$

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To demonstrate how the required supply airflow is determined, assume that a room's sensible heat gain is 80,000 Btu/hr and its latent heat gain is 20,000 Btu/hr. First, divide the sensible heat gain by the total heat gain. The resulting sensible heat ratio (SHR) is 0.80.

Determining Supply Airflow

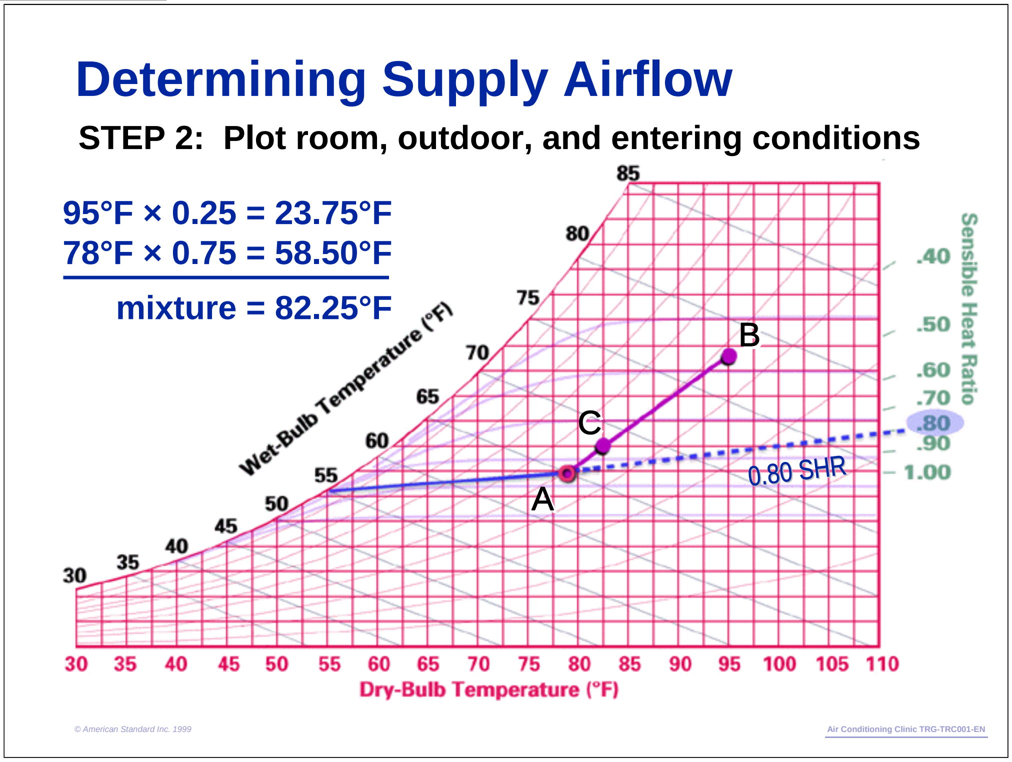
room — 78°F DB, 50% RH

outdoor air (OA) — 95°F DB, 78°F WB

ventilation — 25% OA

The second step is to determine the entering air conditions. Design room air is 78°F DB, 50% RH; design outdoor air is 95°F DB, 78°F WB.

Twenty-five percent (25%) outdoor air is required for ventilation purposes.



Plot the outdoor air B (95°F DB, 78°F WB) and indoor air A (78°F DB, 65°F WB) conditions on the psychrometric chart. Then calculate the mixed-air conditions using the method learned in Period 2.

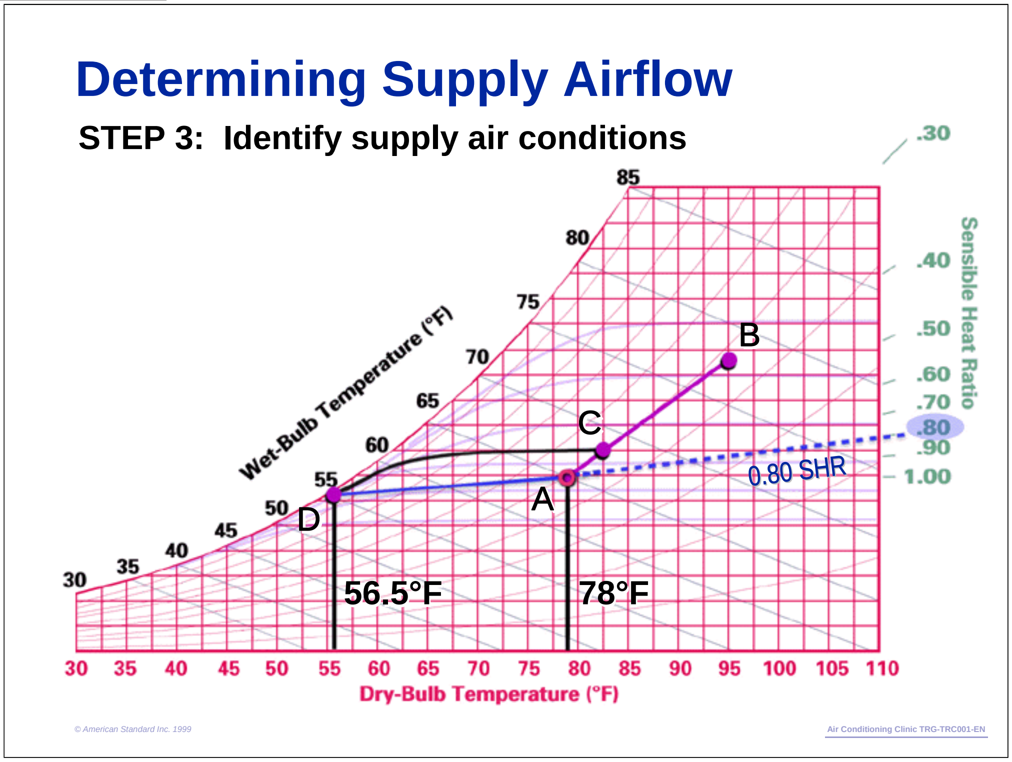
95°F x 0.25 = 23.75°F

78°F x 0.75 = 58.50°F

Mixed-Air Temperature = 23.75°F + 58.50°F = 82.25°F

Locate the mixed air conditions C on the psychrometric chart (82.25°F DB, 68.6°F WB).

Next, establish the SHR line by aligning the 0.80 mark on the scale with the index point and drawing a line through both points to the saturation curve. In this case, the room design conditions and the index point are the same (A = 78°F DB, 65°F WB).



The third step is to determine the required supply air conditions. This is where the coil curves are used. Using the curvature of the nearest coil line as a guide, draw a curve from the mixed-air condition C until it intersects the SHR line.

Point D, at which this curve crosses the SHR line, represents the supply air condition that will absorb the room's sensible and latent heat in the correct proportions needed to maintain the desired room conditions. Here, this supply air condition is found to be 56.5°F DB and 55.2°F WB.

Determining Supply Airflow

STEP 4: Solve the supply airflow equation

$$\text{Supply Airflow} = \frac{\text{Sensible Heat Gain}}{1.085 \times (\text{Room DB} - \text{Supply DB})}$$

With the supply air conditions known, the next step is to calculate the specific quantity of air (cfm or cubic feet per minute) needed to satisfy the room heat gains. The required supply airflow is determined using the following formula, where the sensible heat gain is expressed in Btu/hr and the two temperatures are in °F.

$$\text{Supply Airflow (cfm)} = \frac{\text{Sensible Heat Gain}}{1.085 \times (\text{Room DB - Supply DB})}$$

Realize that 1.085 is not a constant! It is the product of density, the specific heat of air and the conversion factor of 60 minutes per hour. These properties of air at “standard” conditions (69.5°F DB dry air at sea level) result in the value 1.085. Air at other conditions and elevations will cause this factor to change.

Density = 0.075 lb/cu ft

Specific Heat = 0.24 Btu/lb°F

0.075 x 0.24 x 60 min/hr = 1.085

Determining Supply Airflow

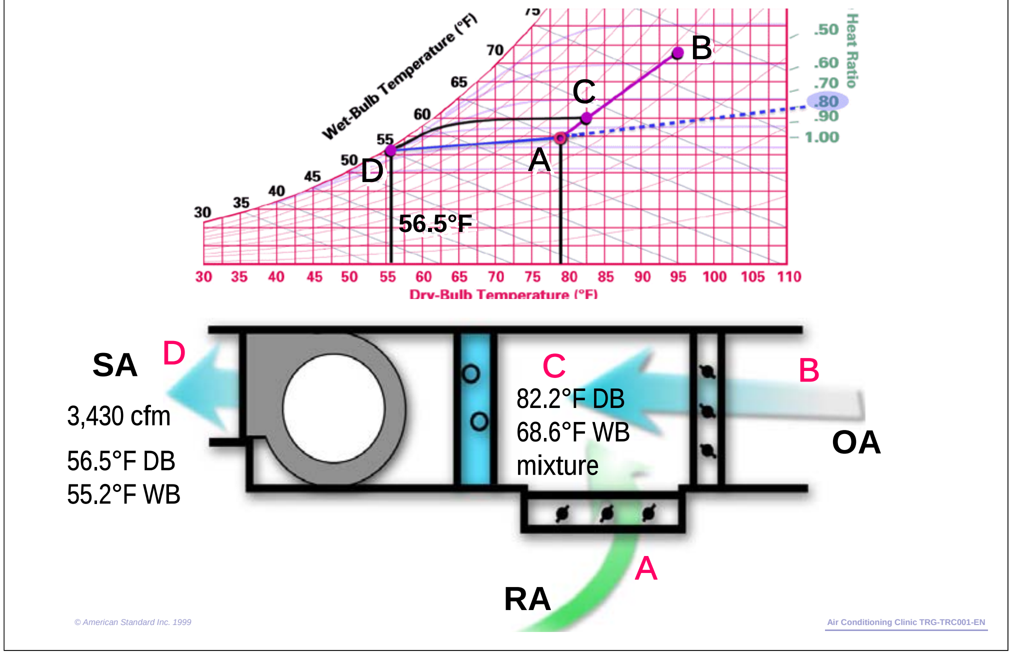
STEP 4: Solve the supply airflow equation

$$\frac{80{,}000 \text{ Btu/hr}}{1.085 \times (78°F - 56.5°F)} = 3{,}430 \text{ cfm}$$

For this example, the supply airflow is calculated as follows:

$$\text{Supply Airflow (cfm)} = \frac{80{,}000 \text{ Btu/hr}}{1.085 \times (75°F - 56.5°F)} = 3{,}430 \text{ cfm}$$

Determining Supply Airflow



The cooling coil must cool and dehumidify 3,430 cfm of air from the entering condition C to the supply air condition D to maintain the desired room conditions.

Arbitrarily Using 55°F Supply Air

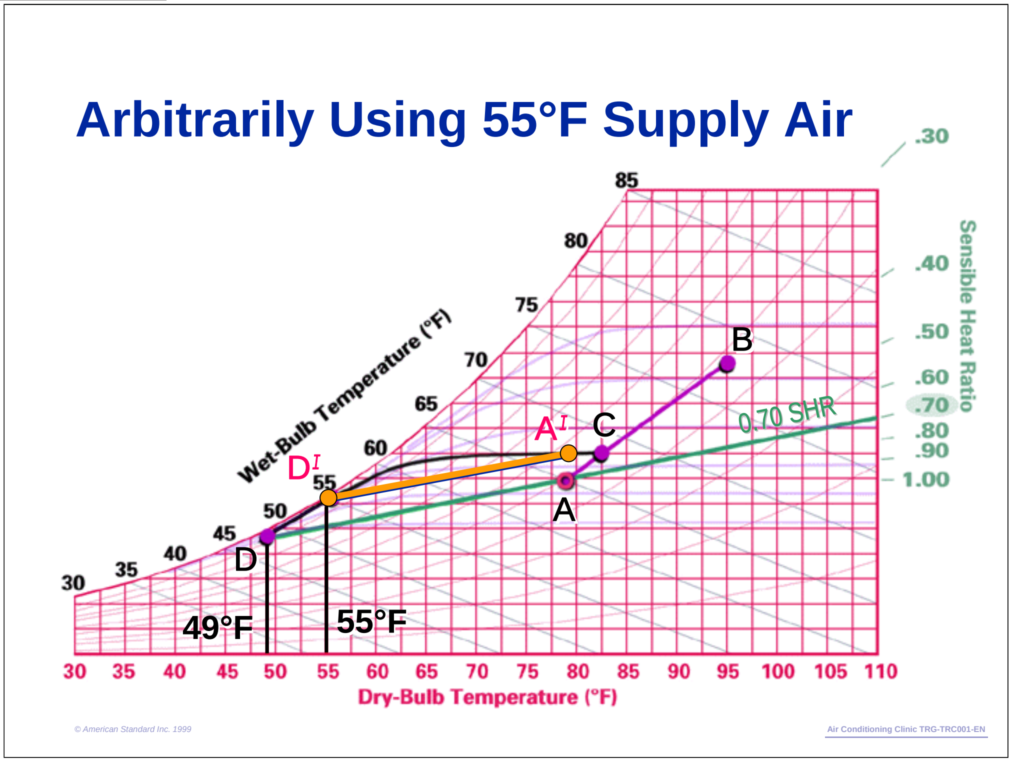
47,000 Btu/hr sensible heat gain

20,000 Btu/hr latent heat gain

$$SHR = \frac{47{,}000 \text{ Btu/hr}}{67{,}000 \text{ Btu/hr}} = 0.70$$

Some designers prefer to set the supply air temperature at 55°F or use a 20°F temperature differential (Room DB – Supply DB) without regard for the actual sensible heat ratio of the room.

Using our same example, let's examine how this has the potential for creating a problem. Assume that the building design changes to use a much-higher-quality glass that will reduce the sensible portion of the design load from 80,000 Btu/hr to 47,000 Btu/hr. This reduces the SHR to 0.70.



Plotting this new SHR line on the psychrometric chart, we find that the SHR line crosses the coil curve at approximately 49°F DB (D).

If the system is arbitrarily designed with a 55°F supply air temperature D', the resulting room conditions will fall on the 0.70 SHR line drawn through D'. The resulting room conditions A' will be 78°F DB, 57% RH. This arbitrary design practice results in a higher room relative humidity than desired.

Psychrometry

period five

Tons of Refrigeration

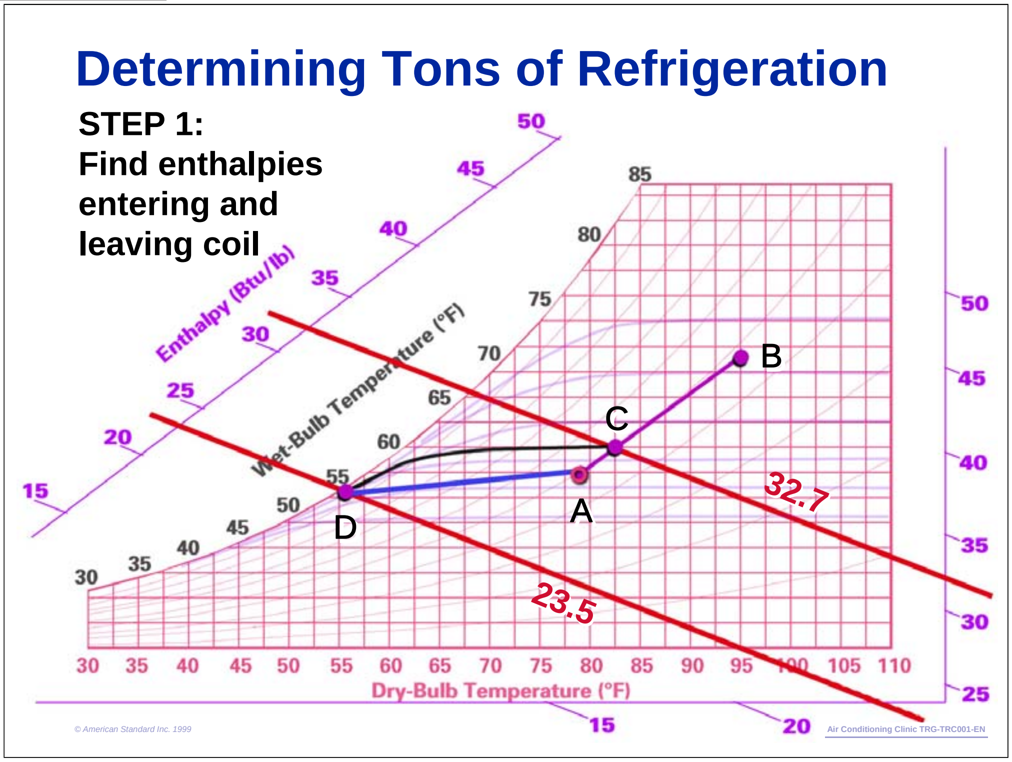
The psychrometric chart can also be used to determine the total load on the refrigeration equipment, expressed in Btu per hour or **tons of refrigeration**. One ton equals 12,000 Btu/hr.

What is Enthalpy?

The total heat energy in one pound of air (Btu/lb) at its present condition.

Enthalpy (*h*) = Sensible Heat + Latent Heat

Another property of air, enthalpy, must now be defined. **Enthalpy** describes the total amount of heat energy, both sensible and latent, in one pound of air at its present condition. It is expressed in Btu per pound of dry air (Btu/lb). When displayed in formulas, enthalpy is usually designated as *h*.



Using the previous example for calculating supply airflow, the first step is to determine the enthalpies of the air entering and leaving the cooling coil. This is accomplished by lining up three points on the chart, including the entering-air condition and identical points on the two enthalpy scales—one each on the left and right edges of the psychrometric chart.

Using this method, the enthalpy of the mixed air entering the coil C is found to be 32.7 Btu/lb. Similarly, the enthalpy of the supply air leaving the coil D is found to be 23.5 Btu/lb.

Determining Tons of Refrigeration

STEP 2: Solve the total refrigeration load equation

$$\text{Refrigeration Load (Btu/hr)} = 4.5 \times \text{Supply Airflow} \times (h_1 - h_2)$$

h_1 = enthalpy of air entering coil (Btu/lb)
h_2 = enthalpy of air leaving coil (Btu/lb)

The total refrigeration load, in terms of Btu per hour, is then calculated using the following formula, where the supply airflow is expressed in cfm, h_1 is the entering-air enthalpy in Btu/lb, and h_2 is the leaving-air enthalpy in Btu/lb.

$$\text{Refrigeration Load (Btu/hr)} = 4.5 \times \text{Supply Airflow} \times (h_1 - h_2)$$

Realize that 4.5 is not a constant! It is the product of density of air and the conversion factor of 60 minutes per hour. The density of air at "standard" conditions (69.5°F DB dry air at sea level) results in the value 4.5. Air at other conditions and elevations will cause this factor to change.

Density = 0.075 lb/cu ft

0.075 x 60 min/hr = 4.5

Determining Tons of Refrigeration

STEP 2: Solve the total refrigeration load equation

$$4.5 \times 3{,}430 \text{ cfm} \times (32.7 - 23.5) = 142{,}000 \text{ Btu/hr}$$

$$\frac{142{,}000 \text{ Btu/hr}}{12{,}000 \text{ Btu/hr/ton}} = 11.8 \text{ tons of refrigeration}$$

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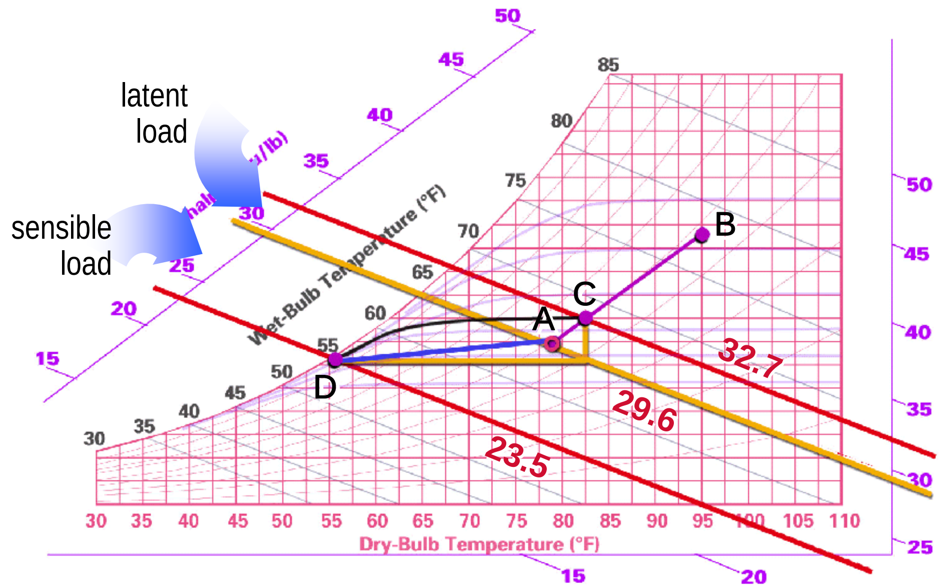
Using the supply airflow calculated during Period 4 and the enthalpy values read from the psychrometric chart:

$$\text{Refrigeration Load (Btu/hr)} = 4.5 \times 3{,}430 \text{ cfm} \times (32.7 - 23.5) = 142{,}000 \text{ Btu/hr}$$

Converting to the more common units of tons:

$$\frac{142{,}000 \text{ Btu/hr}}{12{,}000 \text{ Btu/hr/ton}} = 11.8 \text{ tons of refrigeration}$$

Sensible and Latent Coil Loads



The psychrometric chart can also be used to determine the sensible and latent components of the coil's refrigeration load.

First, draw a right triangle though the coil entering and leaving air conditions. The vertical leg represents the amount of moisture removed by the coil, i.e., latent load, and the horizontal leg represents the amount of change in dry-bulb temperature through the coil, i.e., sensible load.

Sensible and Latent Coil Loads

4.5 × 3,430 cfm × (29.6 – 23.5) = 94,150 Btu/hr

7.8 tons of refrigeration (sensible)

4.5 × 3,430 cfm × (32.7 – 29.6) = 47,850 Btu/hr

4.0 tons of refrigeration (latent)

By determining the enthalpy values for these three points, the same equation can be used to calculate both the sensible and the latent portions of the coil's refrigeration load.

$$\text{Sensible Refrigeration Load (Btu/hr)} = 4.5 \times 3{,}430\text{ cfm} \times (29.6 - 23.5) = 94{,}150\text{ Btu/hr}$$

$$\frac{94{,}150\text{ Btu/hr}}{12{,}000\text{ Btu/hr/ton}} = 7.8\text{ tons of refrigeration}$$

$$\text{Latent Refrigeration Load (Btu/hr)} = 4.5 \times 3{,}430\text{ cfm} \times (32.7 - 29.6) = 47{,}850\text{ Btu/hr}$$

$$\frac{47{,}850\text{ Btu/hr}}{12{,}000\text{ Btu/hr/ton}} = 4.0\text{ tons of refrigeration}$$

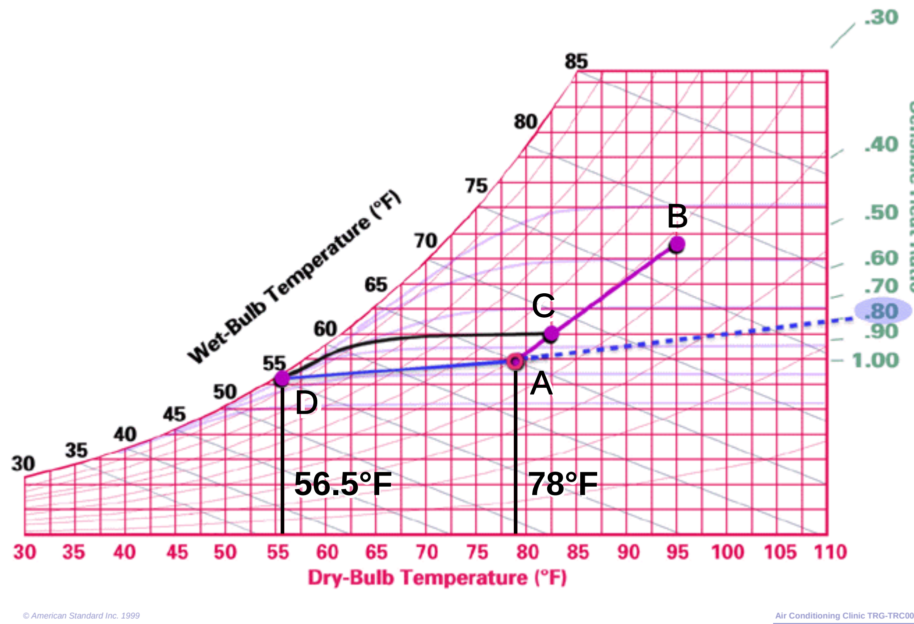
Psychrometry

period six

Psychrometric Analyses

Now we will look at a few ways that the psychrometric chart can help us analyze air conditioning systems. For simplicity, we will limit our examples to systems serving a single zone.

SHR at Full-Load Conditions



In the previous example, the sensible heat ratio was based on **full load or design load conditions**. It must be understood that the sensible portion of total heat gain is particularly subject to change throughout the day, causing the ratio of sensible to total heat gain to change.

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SHR Changes with Room Load

Full-Load SHR:

$$\frac{80{,}000 \text{ Btu/hr}}{80{,}000 \text{ Btu/hr} + 20{,}000 \text{ Btu/hr}} = 0.80$$

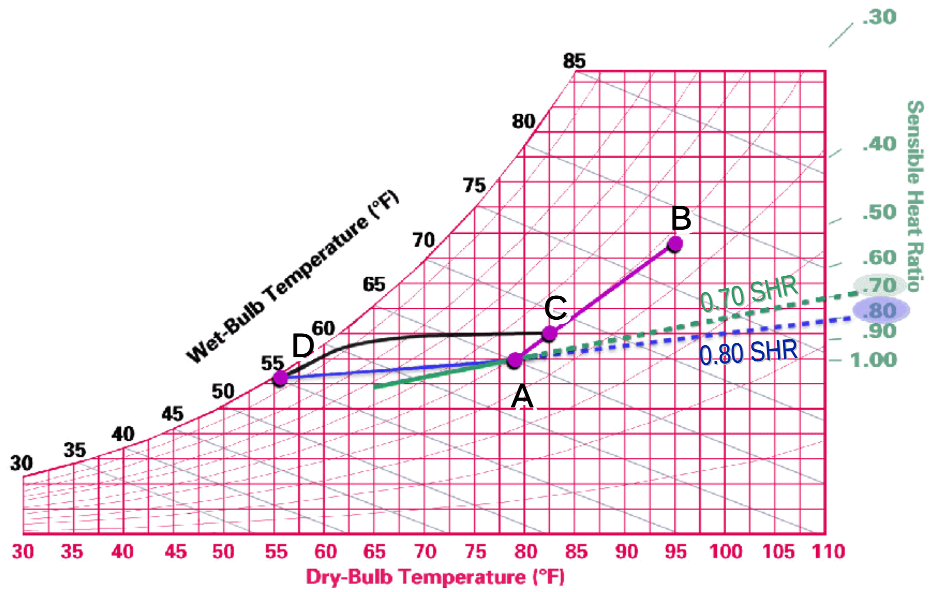
Part-Load SHR:

$$\frac{47{,}000 \text{ Btu/hr}}{47{,}000 \text{ Btu/hr} + 20{,}000 \text{ Btu/hr}} = 0.70$$

For example, assume that at full load the room is subject to an 80,000 Btu/hr sensible heat gain and a 20,000 Btu/hr latent heat gain. The full-load sensible heat ratio is 0.80.

At other times during the day, clouds block the sun and reduce the solar heat gain, and some of the lights are turned off. This reduces the room's sensible heat gain from 80,000 Btu/hr to 47,000 Btu/hr. The room's latent heat gain originates primarily from people. Assuming that the occupancy of the room remains constant, the latent heat gain is still 20,000 Btu/hr and the part-load sensible heat ratio becomes 0.70.

SHR at Part-Load Conditions



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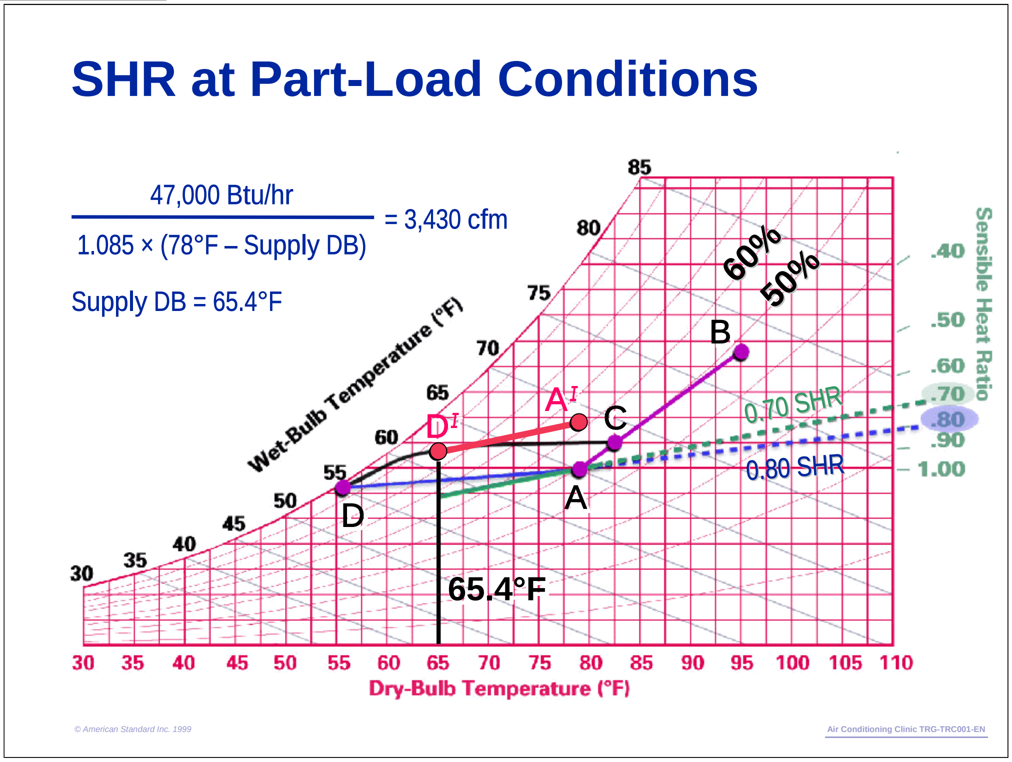
To maintain the design room conditions A for this part-load sensible heat ratio, a different supply air condition—one that falls on the 0.70 SHR line—and a different airflow are required. But suppose the system in this example was designed to deliver a constant quantity of air and vary its supply temperature to meet the changing loads.



In response to the reduction in room sensible heat gain, the coil capacity is throttled, raising the supply air temperature from D to D' to balance the new room sensible heat gain. This new supply air temperature is dictated by the equation:

$$\text{Supply Airflow (cfm)} = \frac{\text{Sensible Heat Gain}}{1.085 \times (\text{Room DB - Supply DB})}$$

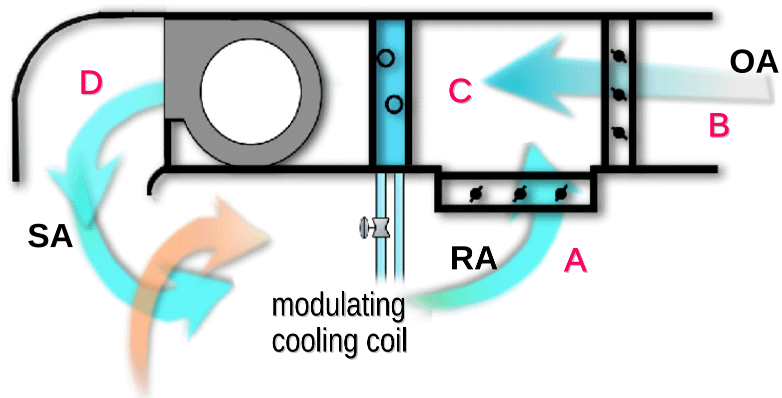
Since the supply airflow and the desired room dry-bulb temperature are constant, the only variable that responds to this change in sensible heat gain is supply air temperature.

$$\frac{47{,}000\ \text{Btu/hr}}{1.085 \times (78°\text{F - Supply DB})} = 3{,}430\ \text{cfm}$$

$$\text{Supply DB} = 65.4°\text{F}$$

This new supply air temperature D' is delivered in sufficient quantity to absorb the room's sensible heat gain, but it does not fall on the part-load SHR line and is not dry enough to completely absorb the latent heat gain. When the conditioned air enters the room, it mixes with room air along the 0.70 sensible heat ratio line from D' to A'. The resulting room condition A', where the SHR line intersects the room dry-bulb temperature line (78°F), shows that the relative humidity increased to 61%.

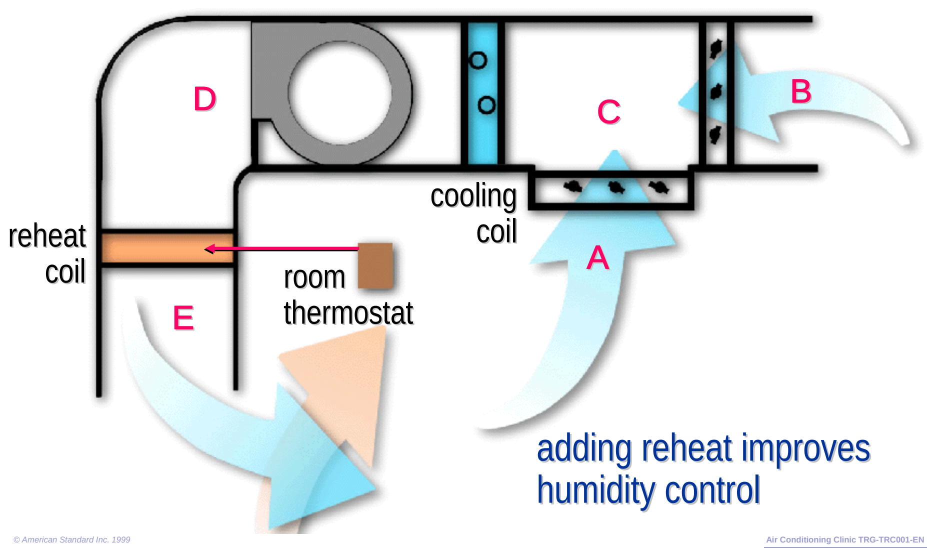
Constant-Volume System



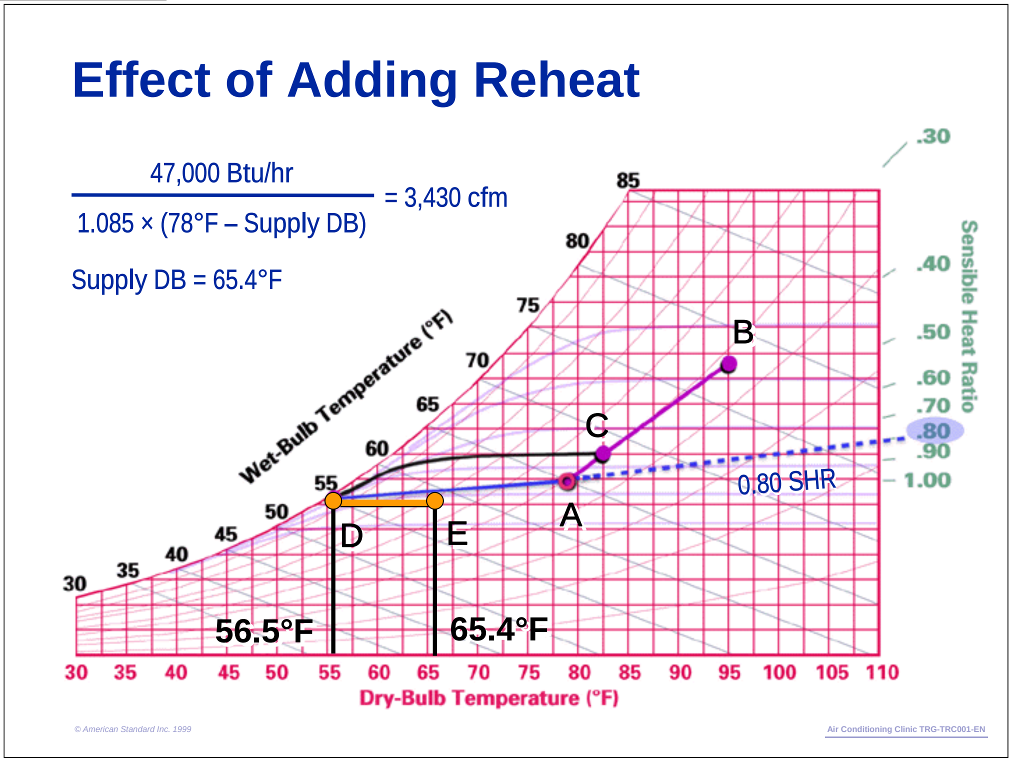
constant quantity of variable-temperature air

This is the manner in which a constant-volume, variable-temperature system with a modulating coil performs. It provides a constant quantity of air to the room and responds to part-load conditions by varying the supply air temperature. This is performed by modulating the flow of the cooling fluid through the coil, typically using a 2-way or 3-way control valve controlled by a thermostat that senses the room dry-bulb temperature. Such a system can provide good dry-bulb temperature control. As the sensible heat ratio changes from full load, however, it may lose control of the room relative humidity.

Constant-Volume with Reheat



One method of improving the constant-volume system's ability to control room humidity is to reheat the supply air. In this example, reheat is provided by a heating coil located downstream of the air handler. This reheat coil is controlled by a thermostat sensing the room dry-bulb temperature, while the cooling coil is controlled to provide a constant leaving-air temperature.



Using the part-load conditions from the previous example, the room's sensible heat gain is reduced from 80,000 Btu/hr to 47,000 Btu/hr while the latent heat gain remains the same.

Sensing the reduction in dry-bulb temperature due to the lower sensible heat gain, the room thermostat assumes control of the reheat coil. The cooling coil is controlled to provide a constant supply air temperature (D = 56.5°F DB), while the reheat coil is controlled to add just enough heat to the supply air to offset the reduction in room sensible-heat gain.

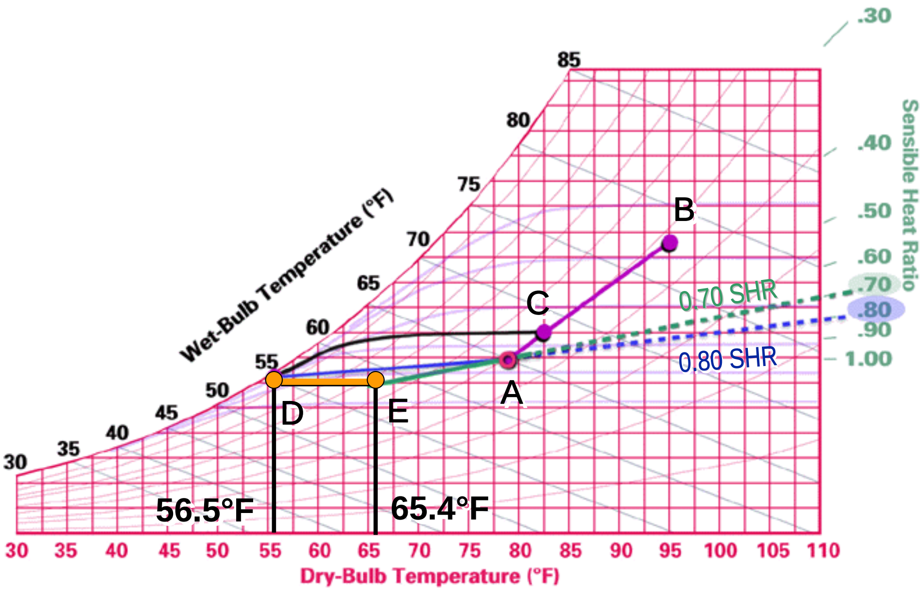
Since the supply airflow is constant and the desired room dry-bulb temperature and sensible heat gain are known, we can calculate the required "re-heated" supply air temperature:

$$\frac{47{,}000\ \text{Btu/hr}}{1.085 \times (78^\circ\text{F} - \text{Supply DB})} = 3{,}430\ \text{cfm}$$

$$\text{Supply DB} = 65.4^\circ\text{F}$$

Since the supply air leaving the coil is sensibly heated (i.e. no moisture is added or removed), it moves horizontally along a constant humidity-ratio line from D to E. The resulting supply air conditions are 65.4°F DB, 58.9°F WB.

Effect of Adding Reheat



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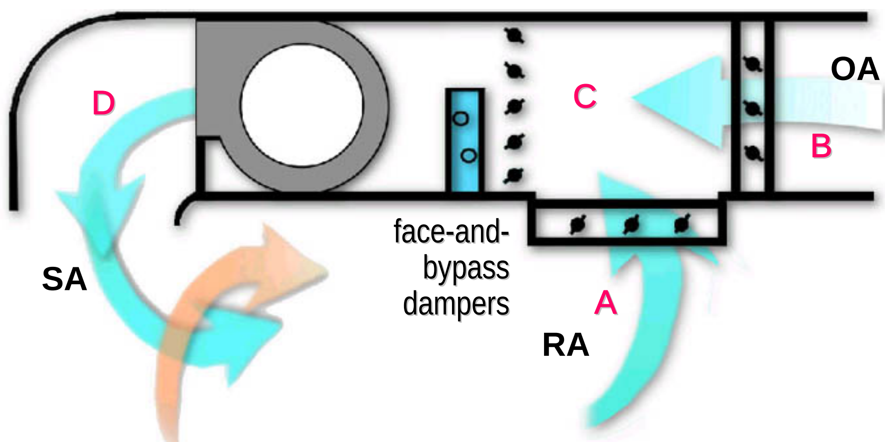
This supply air mixes with room air along the part-load 0.70 SHR line from E to A, arriving at the desired room conditions A. Now, if the room's latent heat gain were also changed, the resulting room conditions would not fall exactly on A, but on the appropriate SHR line that runs through E.

Adding reheat permits better room humidity control at various part-load conditions while maintaining room dry-bulb temperature control. Realize, however, that this system uses more energy than the previous constant-volume system with a modulating cooling coil: it constantly cools the supply air to 56.5°F, then reheats the air as necessary when the building sensible load drops.

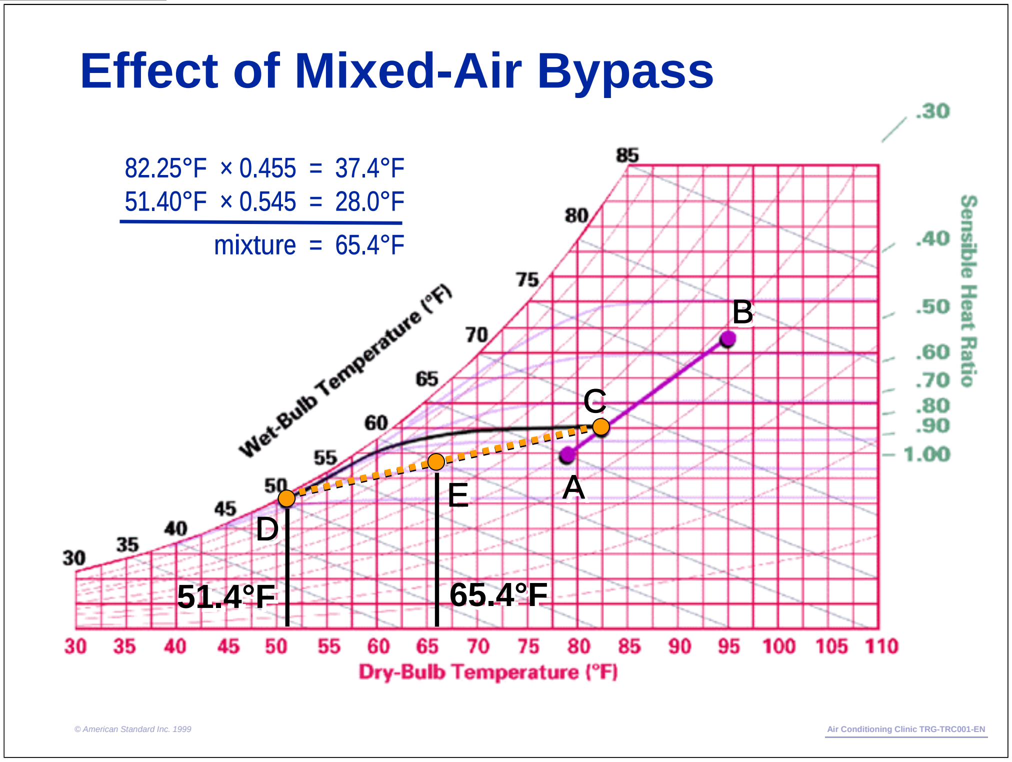
Mixed-Air Bypass



mixed-air bypass improves humidity control

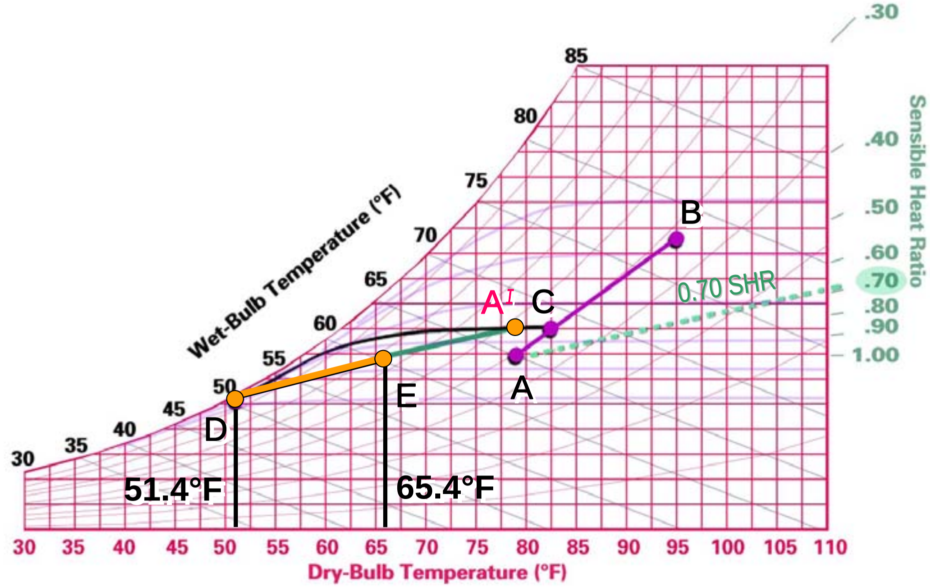
Another method of improving the constant-volume system's ability to control room humidity is to bypass mixed air around the cooling coil. In this example, face-and-bypass dampers are placed in front of the cooling coil and used to vary the portion of the supply air that actually passes through the coil, thus varying the supply air temperature as the two airstreams mix downstream of the air handler.

The face-and-bypass dampers are controlled by the room dry-bulb thermostat. The cooling coil is allowed to “run wild,” causing the air that does pass through it to be cooled more at partial airflows.



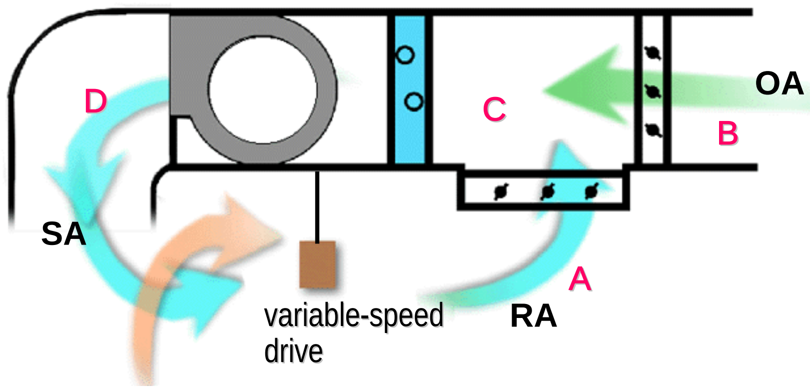
At our example part-load conditions, the room thermostat assumes control of the face-and-bypass dampers, which reduces the amount of air passing through the cooling coil. Since the coil is now “running wild,” the reduced airflow through the coil (1,870 cfm) is cooled and dehumidified more than at full load (D). When the conditioned air mixes with the bypass air (1,560 cfm), the required supply air condition (3,430 cfm at E) results.

Effect of Mixed-Air Bypass



This supply air E mixes with room air along the part-load 0.70 SHR line, arriving at the resulting new room conditions A'. While the quantity and temperature of supply air are suitable to absorb the room's sensible heat gain, they are unable to completely absorb the latent heat gain. The result is a shift in room conditions from the design point A to 78°F DB, 58% RH (A').

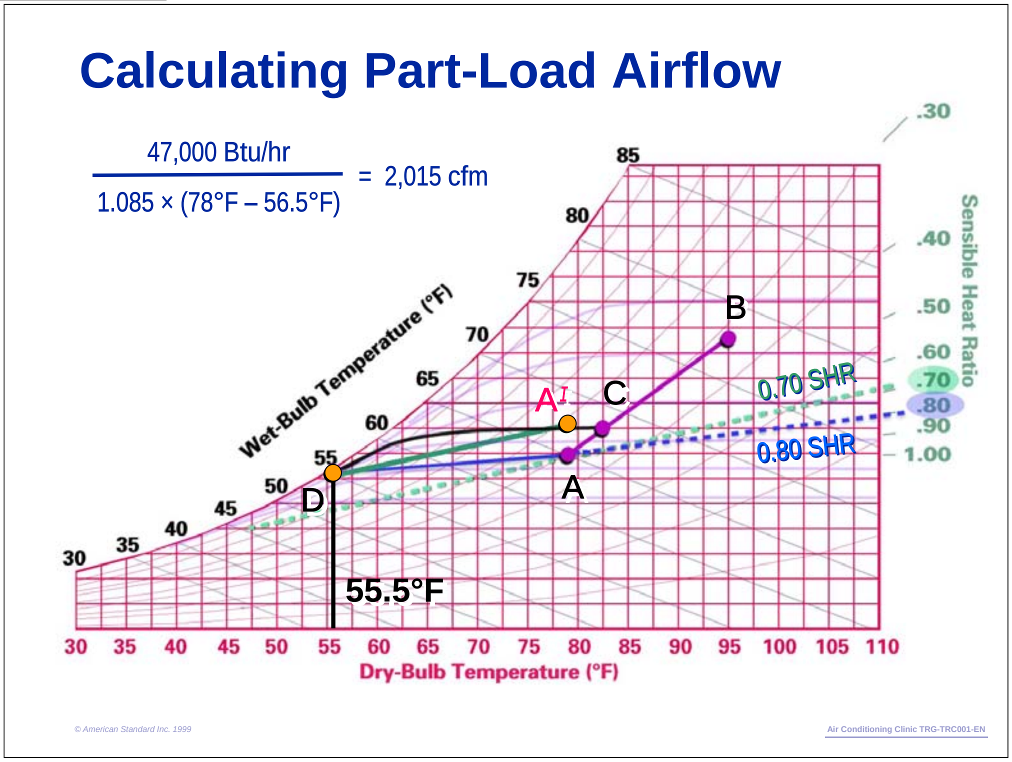
Variable-Air-Volume (VAV) System



variable quantity of constant-temperature air

The final method of part-load control we will analyze is to vary the supply airflow to the room.

Let’s look at the same example again, this time using a simple variable-air-volume (VAV) system. This system responds to part-load conditions by supplying a variable quantity of constant-temperature air. At full load, this system looks the same on the psychrometric chart as the constant-volume system—it supplies 3,430 cfm of 56.5°F air to the room.



At part load, when the SHR of the room is reduced from 0.80 to 0.70, the VAV system responds by reducing the quantity of 56.5°F air supplied to the room to match the reduced sensible heat gain. The part-load sensible heat gain of 47,000 Btu/hr and the constant supply air temperature, 56.5°F DB, are used to determine the required part-load air quantity.

$$\text{Supply Airflow (cfm)} = \frac{47{,}000\ \text{Btu/hr}}{1.085 \times (78°\text{F} - 56.5°\text{F})} = 2{,}015\ \text{cfm}$$

When the conditioned supply air D enters the room, it mixes with room air along the part load SHR line from D to A'. This quantity and temperature of supply air are suitable to absorb the room's sensible heat gain, but are unable to completely absorb the latent heat gain. The result is a shift in the room conditions from the design point A to 78°F DB and 59% RH (A'). While the simple VAV system does a better job of controlling room humidity than the simple constant-volume system, it is still unable to maintain the desired condition of 50% RH.

To more accurately determine the final room conditions, you would cycle through the psychrometric chart again. First, use the new room condition A' to calculate the mixed-air entering conditions. Then use the coil curves to find the condition of the 56.5°F dry-bulb supply air. Finally, draw the SHR line through this new supply air condition to find the resulting room conditions. Repeating this process a few times allows the room condition to converge and be equal to the condition used in the previous iteration.

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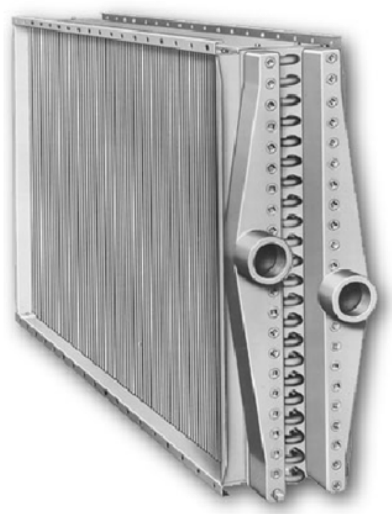
Software Tools



The psychrometric chart is a visual tool that helps designers find solutions to many common HVAC problems by plotting conditions on the chart.

Today, many of these same problems can be quickly solved by computers, which can often eliminate the need for a graphical solution altogether. Still, a basic understanding of psychrometric principles is required to use these tools, and is fundamental to the science of air conditioning.

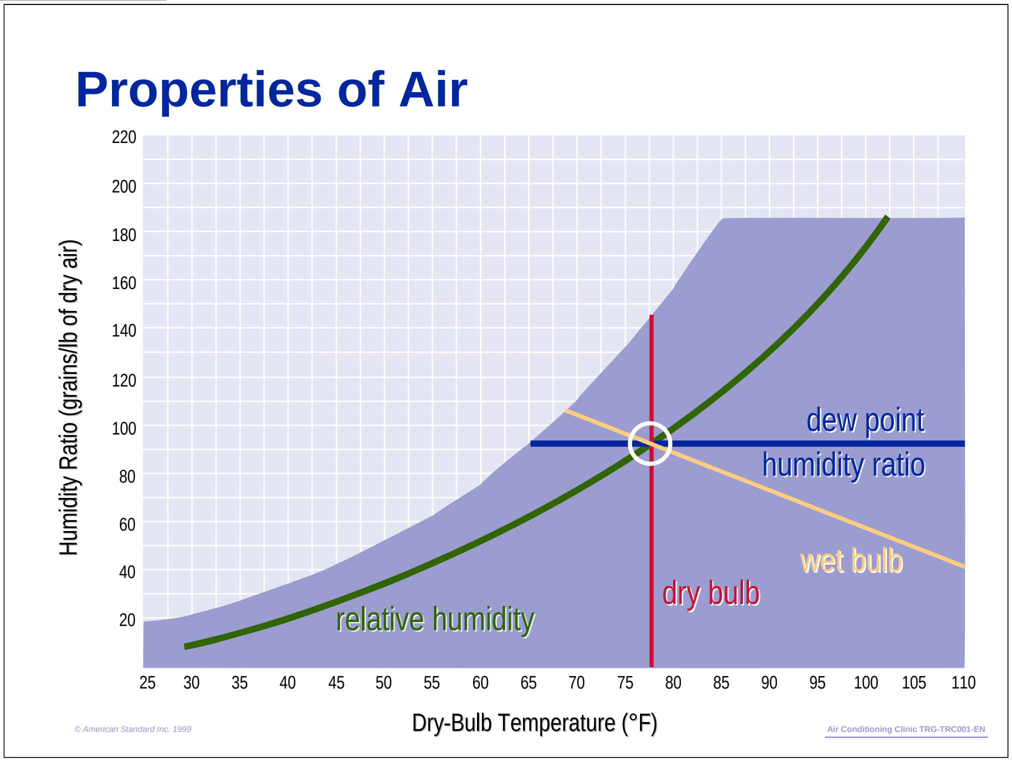
Instead of relying solely on the “typical” coil curves printed on the psychrometric chart, many manufacturers provide computerized coil selection programs to determine the actual performance of specific coils. Software tools are also available to assist you in performing psychrometric calculations, such as determining the properties of air at a given set of conditions, and finding the conditions that result when two air streams are mixed.

Psychrometry

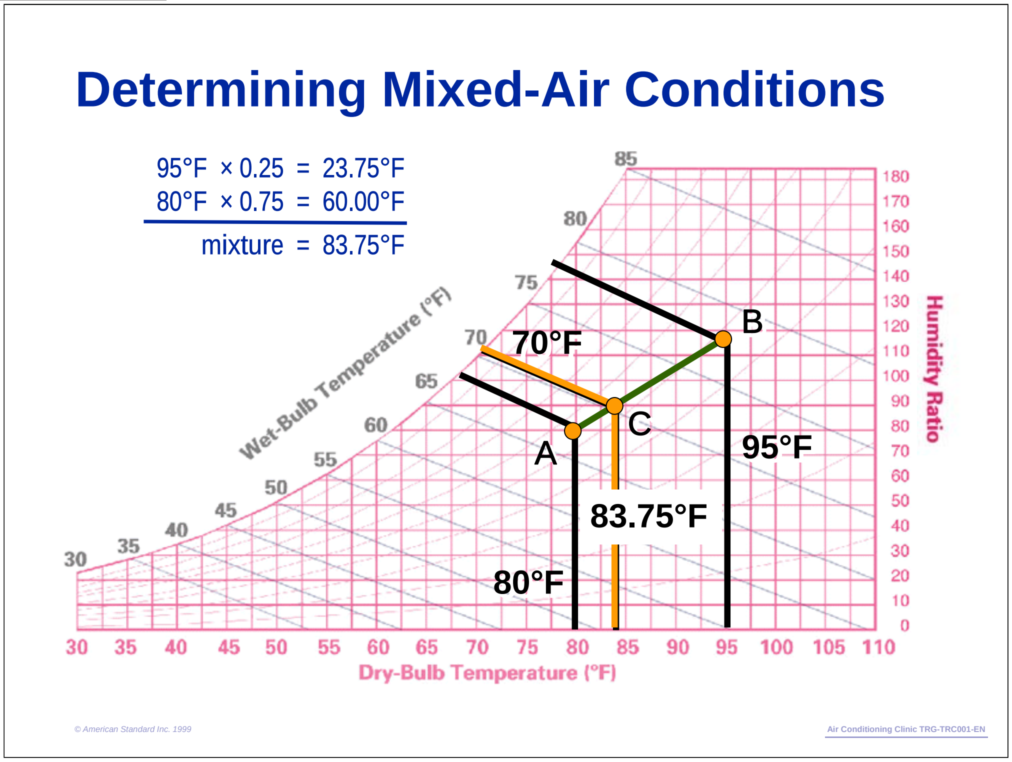
period seven

Review

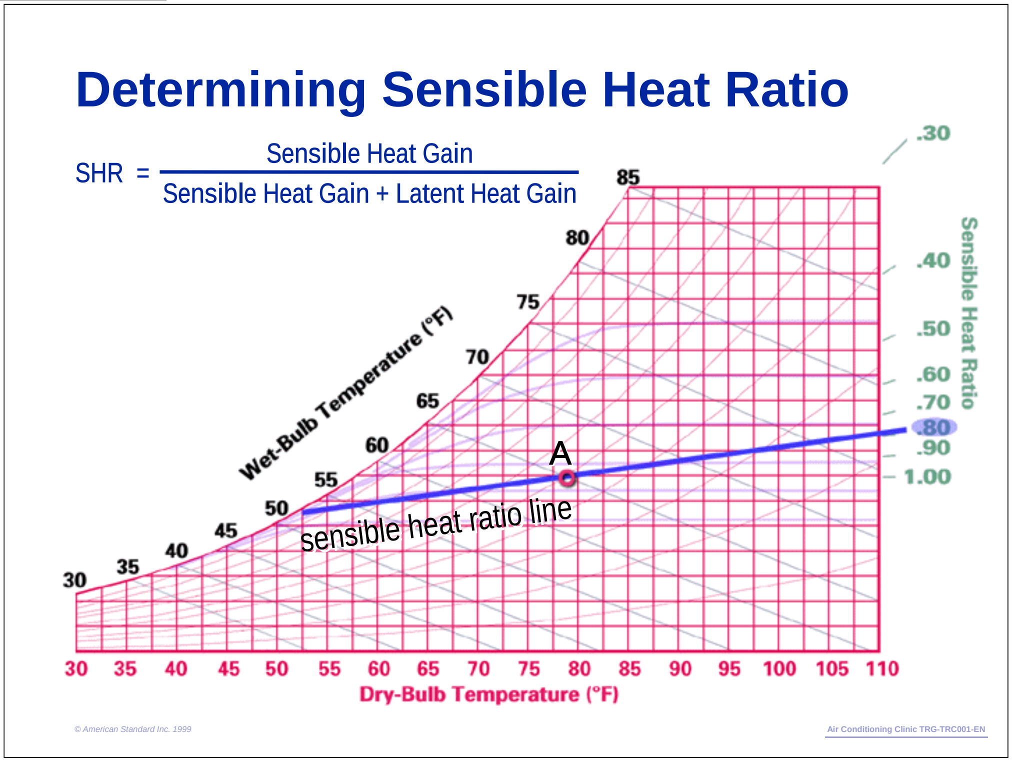
Let’s review some of the main concepts from this clinic on psychrometry.



The lines of the psychrometric chart represent five physical properties of air: **dry bulb**, **wet bulb**, **dew point**, **humidity ratio**, and **relative humidity**. If any two of these properties are known, the remaining properties can be determined from the chart.



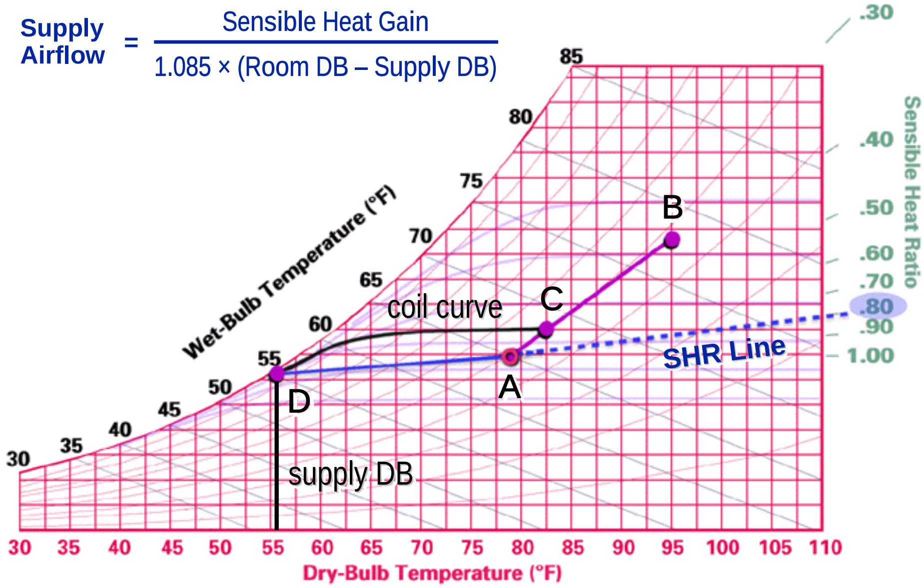
In Period Two, a method was discussed to determine the resulting properties of an air mixture. By plotting the conditions of the **outdoor air** and **recirculated air**, and using the percentage of outdoor air, the resulting condition of the air mixture was calculated and plotted on a straight line from A to B connecting the two air conditions.



In Period Three, the **ratio of sensible heat gain to total heat gain** was discussed. With the aid of the sensible heat ratio scale on the chart, an SHR line was drawn. It was also shown that any combination of air conditions that fall on this line will maintain the desired room conditions A. Each set of conditions requires a different supply airflow.

Determining Coil Leaving Conditions

$$\text{Supply Airflow} = \frac{\text{Sensible Heat Gain}}{1.085 \times (\text{Room DB} - \text{Supply DB})}$$



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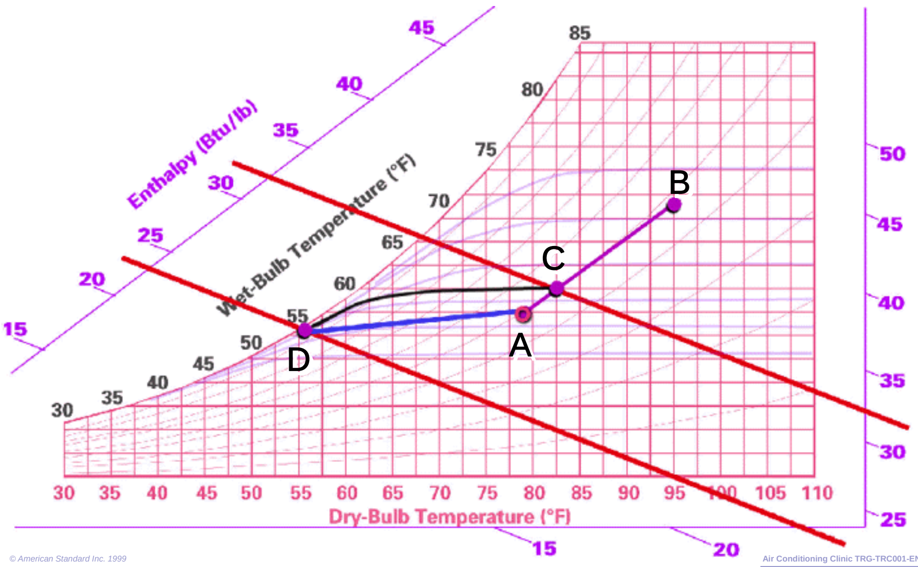
After determining the entering air conditions for the coil and the slope of the SHR line, the **coil curves** were used to find the required supply air conditions. This point (D) was established by the intersection of the coil curve and the SHR line.

By knowing the design room conditions A and the required supply air conditions D, the corresponding **supply airflow** could be calculated.

Determining Tons of Refrigeration

Refrigeration Load = 4.5 × Supply Airflow × ($h_1 - h_2$)



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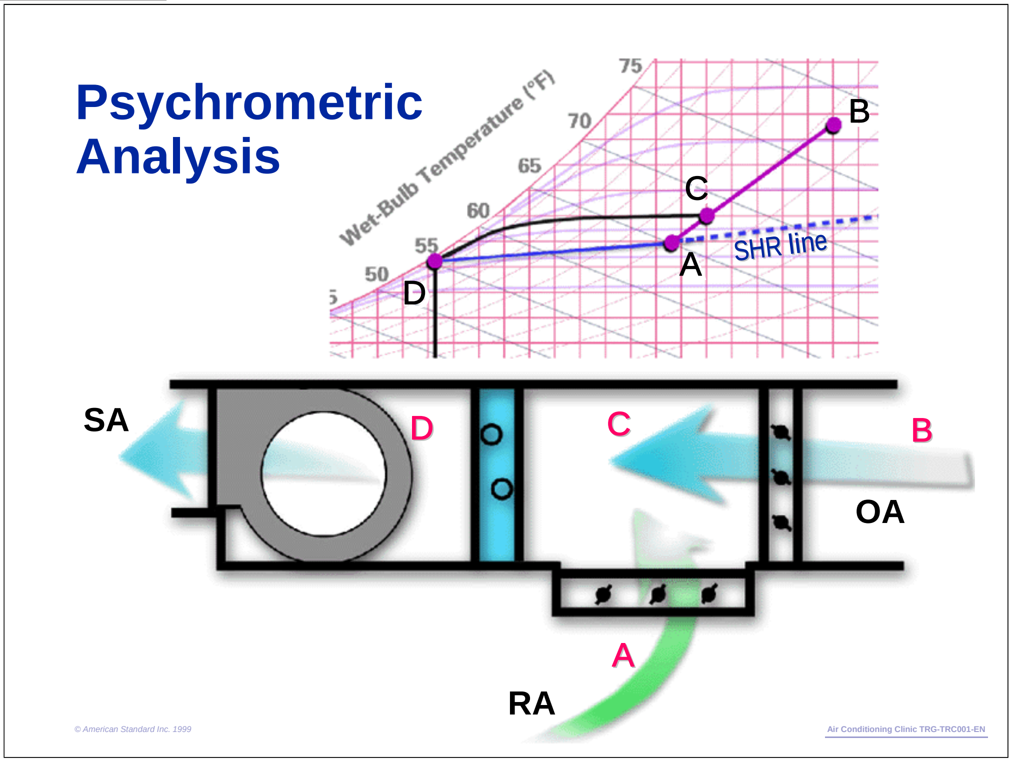
After the entering (C) and leaving (D) coil conditions were established, the **enthalpies** for each were read from the psychrometric chart. These enthalpy values and the previously calculated supply airflow were used to determine the **refrigeration load** in Btu/hr or tons.



The resulting psychrometric chart plot represents the changes that a volume of air undergoes as it travels through a typical air conditioning system.

In this illustration, recirculated air A is mixed with outdoor air B, producing a mixed air condition C.

This air mixture passes through the cooling and dehumidifying coil, with the changes in dry-bulb temperature and humidity ratio represented by the coil curve from C to D.

This supply air D enters the room and mixes with the room air along the SHR line from D to A, absorbing the room's sensible and latent heat gains, to maintain the room at desired conditions A.

Again, for this specific supply air condition, a specific airflow is required to maintain the desired room conditions.

For more information, refer to the following references:

- Trane Air Conditioning Manual
- ASHRAE Handbook—Fundamentals
- Fundamentals of Thermodynamics and Psychrometrics, ASHRAE self-directed learning course
- Psychrometrics: Theory and Practice, ASHRAE

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