

## Air Water Vapor Mixtures: Psychrometrics

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c 1996, 2010

### Introduction

To establish proper comfort conditions within a building space, the designer must consider the air temperature and the amount of water vapor in the air. The latter is important because too high a level of water vapor can lead to uncomfortable feeling, sweat does not evaporate fast and people feel too wet and “clammy”. If the water vapor is too low, people suffer from dryness in their respiratory systems, skin dries out and is itchy and there can be a build up of static electricity, which can cause uncomfortable shocks to people and also disrupt electrical equipment.

Evaporation of water into the air can serve to lower the air temperature. Cooling towers are used in conjunction with air conditioners to produce a higher COP. The condenser of the air conditioner is in contact with cooling tower water, which is at a lower temperature than the outside ambient air.

We will need to define several new terms to deal with air-water vapor mixtures, sometimes referred to as psychrometrics. In addition, mass and energy balances will be established for both the air and the water. A graphical technique will be introduced which aids in visualizing the processes for the air-water vapor mixture.

### Mixtures

When air and water vapor exist alone at moderate or low pressures, always the case for our conditions, they closely approximate ideal gases. In that case the relationship between pressure, temperature and mass can be expressed as,

$$p_a V = n_a \bar{R} T = m_a R_a T \quad (1)$$

for the air and

$$p_v V = n_v \bar{R} T = m_v R_v T \quad (2)$$

for the water vapor. In these equations, the subscript a refers to the air and v to the water vapor, n is the number of moles, m the mass, T the absolute temperature, in deg Kelvin or Rankine,  $\bar{R}$  is the universal gas constant and  $R_a$  and  $R_v$  are the ratios of  $\bar{R}$  to their respective molecular weights. At low pressure, if their and water vapor are mixed together the interaction between the molecules of the two gases is modest so that the mixture can be considered an ideal gas,

$$pV = (n_a + n_v)\bar{R}T \quad (3)$$

where  $p$  is the total pressure of the mixture. Adding equations 1 and 2 and comparing to equation 3 it can be seen that the total pressure in the mixture is just the sum of the pressure of each gas existing alone at the same temperature and in the same total volume, Dalton's Law,

$$p = p_a + p_v \quad (4)$$

When the gases are mixed together  $p_a$  and  $p_v$  are referred to as the partial pressure of air and water vapor, respectively.

The total enthalpy is the sum of the enthalpy of the air and water vapor,

$$H = H_a + H_v = m_a h_a + m_v h_v \quad (5)$$

Dividing the total enthalpy of the mixture by the mass of the air (note, this is not the total mixture mass), we obtain the enthalpy of the mixture per unit mass of dry air which will be written as  $h$  without any subscript,

$$m_a \left( \frac{H}{m_a} \right) = m_a(h) = m_a h_a + m_v h_v \quad (6)$$

Although this seems an odd choice just now, it will become evident why this is a convenient choice. Dividing equation 6 by the mass of the air in the mixture,

$$h = h_a + \left( \frac{m_v}{m_a} \right) h_v = h_a + \omega h_v \quad (7)$$

Note the new enthalpy of the mixture we just defined,  $h$ , is the sum of the specific enthalpy of the air and the product of the specific enthalpy of the water vapor and the ratio of the mass of water vapor to the mass of air in the mixture. The air mass is sometimes called dry air to remind the reader that only the air without the water vapor mass is being considered. Note, be careful when you use  $h$  since different tables can use different states for the zero values of  $h_v$  and  $h_a$  which are added together to get  $h$  of the mixture.

### Humidity ratio or Specific Humidity

The ratio of mass of water vapor to mass of air in a mixture is referred to as the specific humidity or the humidity ratio with the symbol  $\omega$  (no this is not the humidity the goofy weathermen are reporting). The advantage in dealing with the specific humidity is that *if the amount of water vapor in the air remains unchanged, the specific humidity remains constant*. The specific humidity is expressed in grams of moisture (water vapor) per kilogram of dry air. Alternatively it is given as  $\text{lbs}_v / \text{lbs}_a$  or grains of water vapor per pound of dry air where 7000 grains is one

pound mass.

The internal energy of the mixture can also be defined in a similar form as the enthalpy,

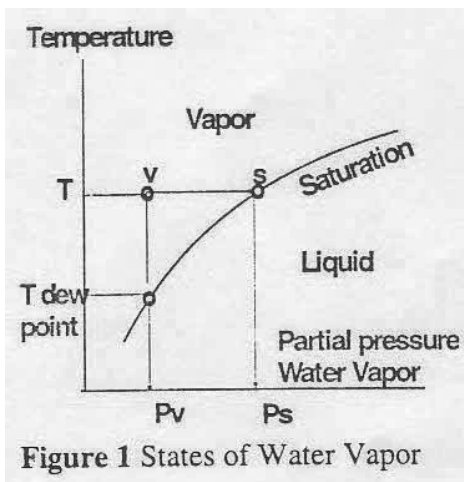
$$u = u_a + \left( \frac{m_v}{m_a} \right) u_v = u_a + \omega u_v \quad (8)$$

but since we will be considering open systems in steady state, we won't spend much time on this.

### Alternate definitions of humidity

Specific humidity we defined above will be useful in writing energy balances and the like but it does not give a good feel for the degree of moisture in the air. For that there is the relative humidity,  $\phi$ . For ideal gases, this is the ratio of the partial pressure of the water vapor in the mixture,  $p_v$ , to the partial pressure of water vapor in a mixture saturated with water vapor at the same temperature,  $T_s$

$$\phi = \frac{P_v}{P_s} \quad (9)$$



**Figure 1** States of Water Vapor

The relationship of  $p_v$  and  $p_s$  can be best seen by referring to the saturation curve of water, Figure 1. Note: this curve only involves the water which is in the mixture, not the air. The state of the water vapor in the mixture is shown by the point v at temperature T and partial pressure  $p_v$ . In general this state will be in the superheated vapor region, away from the saturation line. If additional water vapor is added to the mixture while maintaining temperature constant at T, the largest amount which the mixture could contain is given by point s on the saturation curve between liquid water and water vapor. If more water vapor beyond s is added, the additional water vapor would condense out since the excess vapor pressure would not be in equilibrium with liquid water at this

temperature.

The specific humidity and relative humidity can be related by (see for example, McQuiston and Parker)

$$\phi = \frac{\omega}{0.622} \frac{P_a}{p_s} \quad (10)$$

Note,  $\omega$  must be expressed as kg water vapor/kg of dry air or lb water vapor/lb dry air in equ.10.

Another term frequently used is the dew point temperature,  $T_d$ .  $T_d$  is found by cooling the mixture of air and water vapor, at constant total pressure, until liquid water just begins to condense. The temperature at which condensation first occurs is the dew point temperature. It is also shown on Figure 1. Note that in this case the cooling of the mixture takes place without adding or subtracting any water vapor from the air, i.e., at a constant specific humidity. When the relative humidity is high, near 100%, or the dew point temperature is close to the air temperature, the air is close to being saturated with water vapor. When this happens in the summer you had better look for an air-conditioned space.

#### Example

A mixture of water vapor and air at sea level has a specific humidity of 0.013 kg water vapor per kg of dry air.

The temperature is 25°C. What is the relative humidity and enthalpy of the mixture?

#### Solution

From saturation tables of water find  $P_s$  for water corresponding to 25 °C,  $p_s=3.17$  kPa  
 $h_v=2547$  kJ/kg for saturated water vapor (If the water vapor is an ideal gas,  $h_v(T)$  is equal to  $h_{v, \text{sat}}$  at the same temperature).

The total pressure of air and water vapor is approximately 101 kPa. At this point  $P_v$  is unknown, but it is generally one to two orders of magnitude less than  $P_a$ . As a first guess take  $P_a$  as 101 kPa.

The related humidity, from Equ. 10 is,

$$\phi = \frac{\omega}{0.622} \frac{P_a}{P_s} = \frac{(0.013)}{(0.622)} \frac{101}{3.17} = 0.666 \approx 67\%$$



To be more precise we can now calculate  $P_v$  from  $\phi$  ( $\phi = P_v/P_s$ ) and correct our guess for  $P_a$ .

To find the mixture enthalpy, we need the water vapor enthalpy,  $h_v$ , found above and  $h_a$  the enthalpy of the dry air. Taking  $h_a$  as zero at 25°C,

$$h_a(25^\circ\text{C}) = C_p(25 - 0) = 1.005(25) = 25.13 \frac{\text{kJ}}{\text{kg}}$$

$$h = h_a + \omega h_v = 25.13 + (0.013) 2547 =$$
$$h = 25.13 + 33.11 = 58.24$$

Note that in the mixture the magnitude of the air enthalpy, referenced to air at 0°C, and the water vapor enthalpy, referenced to liquid water at 0°C, is about the same. Therefore, changes in air temperature of about 10°C and evaporation of water increasing relative humidity by fifty percentage points lead to the same order of magnitude change in the mixture enthalpy. Both effects are important in an energy balance for the air-water vapor mixture.

#### Measurement of the State of Mixture

The state of the air-water vapor mixture can be defined by knowing the temperature, the ratio of mass of water vapor to air in the mixture and the total pressure in the mixture. In our consideration the total pressure is at or close to (the building may be slightly pressurized) the local barometric pressure. This does make some difference between say Boston and Denver but generally sea level pressures are used. The temperature (called the dry bulb temperature in this instance) can be found by using a conventional thermometer, thermocouple or thermistor. The latter two are electronic devices whose voltage output is proportional to the temperature.

The most tricky measurement is the amount of water vapor in the air. In earlier days, use was made of a wet bulb thermometer whose bulb was kept in contact with a wet wick. Some water would evaporate from the wick, lowering the thermometer reading below the dry bulb temperature. The difference in reading between the wet and dry bulb thermometers can then be used to calculate the humidity of the air. The lower the relative humidity, the more evaporation ensues and the lower the wet bulb temperature. In recent times, the humidity is determined from the capacitance measurement of a thin film whose electrical properties are found to vary with water vapor content. There is also an automated instrument which has a cooling and heating element. The temperature of a mirror is adjusted until condensate is first detected; the measured mirror temperature at this condition is the dew point temperature.

### Graphical Representation

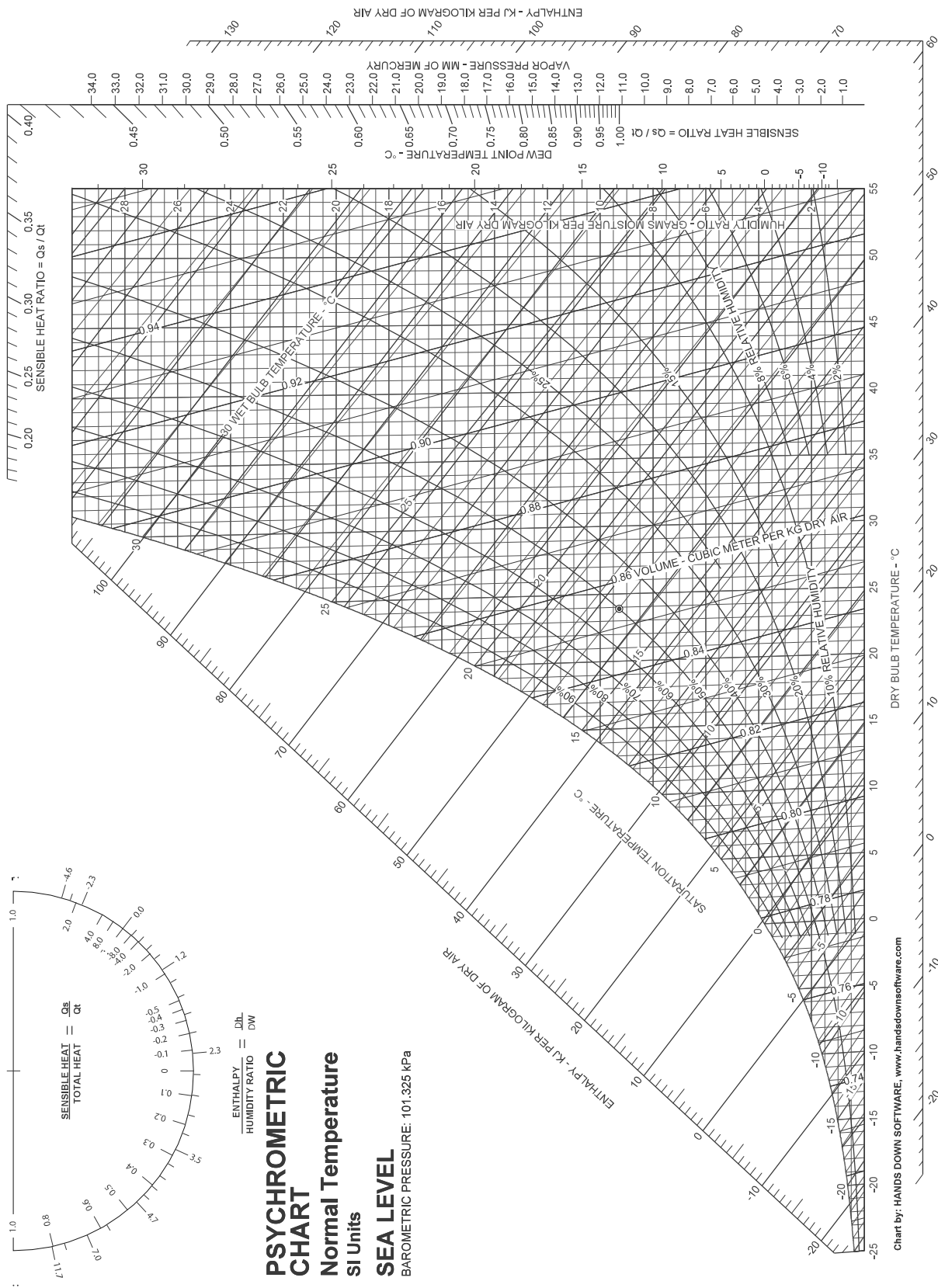
The dry bulb temperature and amount of water vapor in a moist air mixture can be graphically shown on a psychrometric chart. Such a chart is very useful to visualize the changes undergone by a mixture of air and water vapor as the air is conditioned. The chart is also useful in determining the amount of energy required for a specific conditioning of the air. It is strongly suggested that you become familiar with the chart.

Figure 2 shows the psychrometric chart for sea level in SI units. Similar charts are available for selected altitudes and in English units. The right hand vertical scale gives the humidity ratio or specific humidity of the mixture. Note that in most situations encountered in buildings practice the mass of water vapor is much less than the mass of air in the mixture. Typical values of  $\omega$  are 0 to 20 grams of water vapor per kilogram of air. Thus, the mass of water vapor is only at most one to two percent of the mass of air.

The horizontal axis is the dry bulb temperature. There are parallel slanted straight lines starting from the upper left which are lines of constant mixture enthalpy per mass of dry air defined by equ. 7. To keep the enthalpy lines straight and parallel, it turns out that lines of constant dry bulb temperature deviate somewhat from vertical.

The chart also shows other information. Curves of constant relative humidity,  $\phi$ , are shown as are lines of constant wet bulb temperature. Thus, if the dry bulb temperature and relative humidity are known, the state of the mixture can be located on the chart and the corresponding value of the mixture enthalpy and humidity ratio read from the chart.

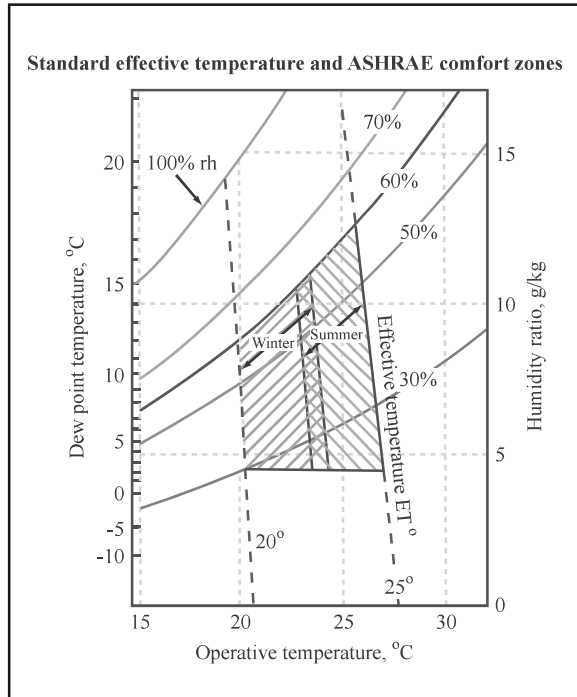
The left hand border represents saturation conditions of the mixture, 100% relative humidity.



ASHRAE Psychrometric Chart No. 1 removed due to copyright restrictions. This image was created by Bob Hanna, can be found on Wikipedia, and is in the public domain.

## Comfort Conditions

The psychrometric chart can also be used to show typical comfort conditions for indoor environments. Shown on Figure 3 are comfort conditions for individuals doing light duties such reading while seated wearing typical summer or winter clothing. These are obtained by



subjecting volunteers to different temperature and humidity conditions and tabulating their vote on their sensation of comfort. There are factors other than dry bulb temperature and humidity which are important to comfort: such as air velocity, radiant energy, and temperature gradient. For our purposes at this time we will take the operative temperature in Figure 3 as the dry bulb temperature. If the occupants are doing more intensive activities, such as working out in a gym, the conditions for comfort will change and a new chart analogous to Figure 3 can be produced.

Figure by MIT OpenCourseWare.

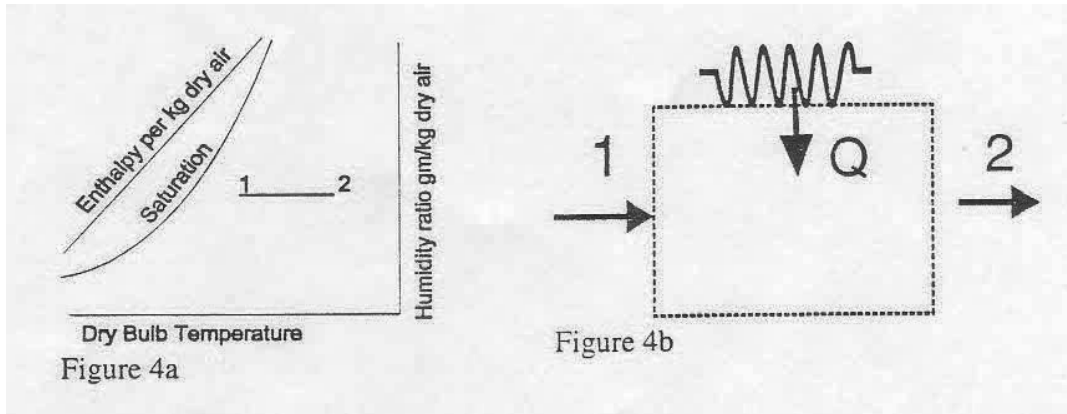
## Conditioning Processes

The purpose of heating, cooling and ventilation systems within buildings is to provide proper ventilation and comfort conditions for all of the occupants of a building. Ventilation is concerned with supplying enough fresh air to keep the concentration of pollutants such as CO<sub>2</sub> in the building within a safe level. In terms of the comfort conditions shown on Figure 3, the aim is to maintain the temperature and humidity conditions throughout the building within the comfort bands shown. This requires heating or cooling the air and in some instances changing the amount of water vapor in the air. There are a number of techniques which can be used to accomplish this. We will start with the most straightforward ones.

## Heating

When the air dry bulb temperature is below the comfort zone, such as state 1 in Figure 4a, the temperature can be raised by forcing the air over a heat exchanger surface. On the other side of the surface, e.g. tubes, is a fluid at a higher temperature. The heat transfer from the hot fluid increases the energy of the air. Taking a control volume around the heat exchanger as shown in Figure. 4b, and assuming steady state operation we can write first a mass balance for the air,

$$\dot{m}_1 = \dot{m}_2 \quad (11)$$



In this process, we will write the mass flow rate of dry air without the subscript a. There isn't any water vapor added to the air so the mass balance for the water becomes,

$$\dot{m}_w = \dot{m}_1 \omega_1 = \dot{m}_2 \omega_2 \quad (12)$$

Combining this with eqn. 11,

$$\omega_1 = \omega_2 \quad (13)$$

Thus, the humidity ratio remains constant during the heating. We can also write an energy balance for the control volume,

$$\dot{Q} = \dot{m}_2 h_{a2} + \dot{m}_{v2} h_{v2} - (\dot{m}_1 h_{a1} + \dot{m}_{v1} h_{v1}) \quad (14)$$

which can be rewritten as,

$$\dot{Q} = \dot{m}_2 (h_{a2} + \omega_2 h_{v2}) - \dot{m}_1 (h_{a1} + \omega_1 h_{v1}) \quad (15)$$

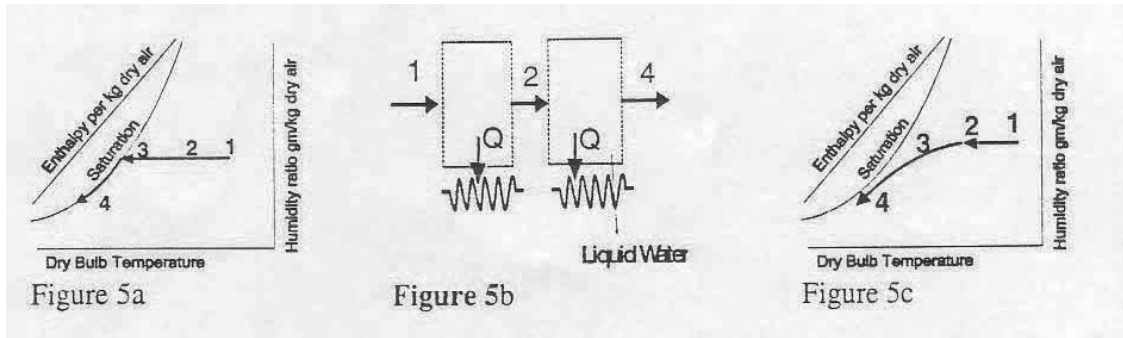
Using the enthalpy of the mixture per unit mass of the dry air, eqn 15 becomes,

$$\dot{Q} = \dot{m}_2 h_2 - \dot{m}_1 h_1 \quad (16)$$

Now you can see the utility of the way h was defined for the mixture.

All of the enthalpy change for both the air and the water vapor is associated with temperature increase and each could be given as  $c_p \Delta T$ . The heat transfer in this process is referred to as sensible heat.

The heating process is represented on the psychrometric diagram as a horizontal line from 1 to



2 since  $\omega$  remains constant. Note that the relative humidity decreases in this process as the dry bulb temperature increases.

To reduce the dry bulb temperature of a mixture of air and water the mixture is passed over heat exchanger tube containing cold fluid inside. The cooling process without any addition or subtraction of water vapor is shown on Figure 5a. The equations for the mass and energy balance are exactly the same as the heating case.

### Dehumidification

If the air has too high a moisture content and  $\omega$  must be reduced, the process gets a little trickier. The heat exchanger tube must be at a temperature below the dew point for the air water vapor mixture. When that occurs, some of the water vapor will condense as liquid on the tubes. The liquid water drains off the tubes by gravity and is collected below and disposed of. In the idealized case, the process is shown as 2-3-4 on Figure 5a. First the mixture is cooled to state 3 on the saturation curve without loss of water vapor. As it is cooled further, it moves down the saturation curve to state 4 which has a lower humidity ratio and a lower dry bulb temperature.

In the real process shown on Figure 5c, the condensation begins before the saturation curve is reached since the tube wall temperature is normally lower than the dew point temperature.

For this process conservation of air mass yields,

$$\dot{m}_2 = \dot{m}_4 \quad (17)$$

while conservation of water must balance the water vapor carried into the exchanger with the sum of the water vapor and liquid water leaving at steady state,

$$\dot{m}_{v2} = \dot{m}_2 \omega_2 = \dot{m}_{v4} + \dot{m}_{l4} = \dot{m}_4 \omega_4 + \dot{m}_l \quad (18)$$

The energy balance between states 2 and 4 must also include the stream of liquid leaving,

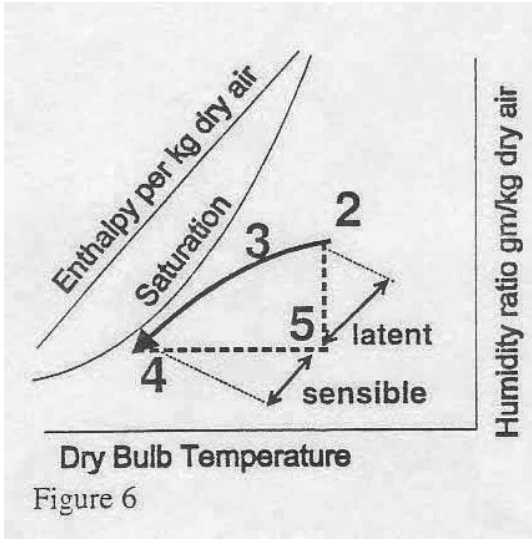
$$\dot{Q}_{2-4} = \dot{m}_4 h_{a4} + \dot{m}_{v4} h_{v4} + \dot{m}_l h_l - (\dot{m}_2 h_{a2} + \dot{m}_v 2 h_{v1}) \quad (19)$$

Since the enthalpy of the air and water leaving the exchanger is lower than that entering,  $Q_{2-4}$  is negative, the mixture is cooled. The energy balance can be rearranged by use of the enthalpy of the air-water vapor mixture again,

$$\dot{Q}_{2-4} = \dot{m}_4 (h_4) + \dot{m}_l h_l - \dot{m}_2 h_2 = \dot{m}_2 (h_4 - h_2) + \dot{m}_l h_l \quad (20)$$

Note that in this case the rate of cooling,  $\dot{Q}$ , involves the change of mixture enthalpy as well as the enthalpy of the leaving water stream. In general the magnitude of the latter term is small and it can be neglected.

$$(\dot{Q}_{3-4})_{sensible} = \dot{m}(h_4 - h_5) = \dot{m}\Delta h_{sensible} \quad (21)$$

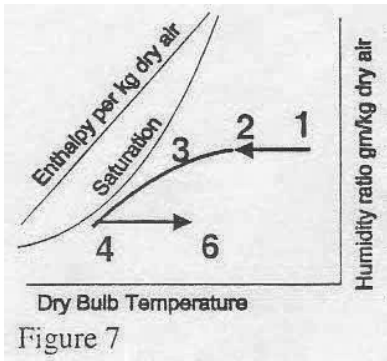


In theory the process from state 2-3-4 could be replaced first by a reduction in water vapor at constant dry bulb temperature, process 2-5, Figure 6, followed by a reduction in dry bulb temperature at constant water vapor content, process 5-4. The latter process is one of pure sensible heat,

While the process 2-5 has no sensible heat since  $\Delta T$  remains constant. In process 2-5 all the heat is associated with the phase change of water from vapor to liquid and this heat is called the latent heat,

$$(\dot{Q}_{3-4})_{latent} = \dot{m}(\omega_2 - \omega_4) h_{lg} = \dot{m}\Delta h_{latent} \quad (22)$$

For the actual process 2-3-4,  $\Delta h_{latent} / \Delta h_{sensible}$ , given in eqns 22 and 21, is the ratio of latent to sensible heat for the process.



We have gone to all the work of dehumidifying the air in the process 2-3-4 above. After all this, when we compare state 4 to the comfort conditions shown in Figure 3 we usually find that the dry bulb temperature is too low for comfort. To achieve comfort conditions, we must reheat the mixture from state 4 to a higher dry bulb temperature as shown in Figure 7. Notice after expending energy to run a chiller or air conditioner to provide cooling to dehumidify the air from state 2 to 4 now we must supply more energy to reheat the air.

Example

For the cooling, dehumidification and reheat process shown on the following figure, use the psychrometric chart to show how an additional heat exchanger can be used to reduce both the cooling energy and reheat energy (no, this is not violating some sacred law, it's just a use of smarts to save energy albeit with the need for an additional heat exchanger).

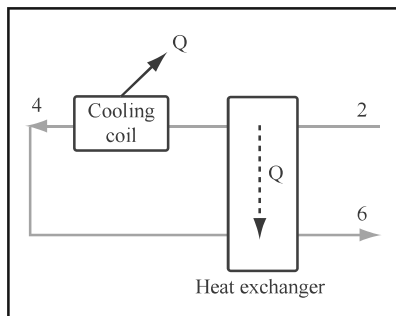


Figure by MIT OpenCourseWare.

Solution:

Looking at the process on the psych chart, notice that temperature  $T_4$  is lower than  $T_2$ . If we put a heat exchanger in system, we can transfer heat from the hotter mixture at  $T_2$  to the colder mixture at  $T_4$ . This reduces the amount of cooling we must supply as well as the amount of heat for the reheat process.



### The Condition Line

Now that we've found out how to produce air at any given dry bulb temperature and humidity ratio that we like, let's put it to work maintaining a comfortable interior space. We must consider the amount of sensible and latent energy given off by occupants and equipment in a building space.

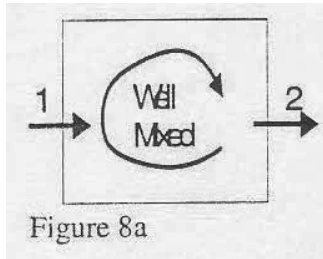


Figure 8a

Consider the space shown in Figure 8a. In most systems air is brought in through a diffuser at a high enough velocity to set up a well mixed condition. Everywhere in the space, the dry bulb and humidity of the air is the same. The heat and moisture transfer from the occupants and equipment to the air can be represented as heat and moisture transfer to a control volume covering all of the air in the space. Table 1 below gives some typical values of sensible and latent heat from people. If the inlet dry air flow rate

is  $\dot{m}$  and the inlet mixture enthalpy and humidity ratio are respectively  $h_1$  and  $\omega_1$ , respectively, then the increase of the mixture enthalpy is given as,

$$\dot{m}(h_2 - h_1) = \dot{Q}_T \quad (23)$$

where  $\dot{Q}_T$  represents the total, latent plus sensible, energy transfer rate to the air. Similarly the increase in the water vapor can be determined by,

$$(\dot{m}_{v2} - \dot{m}_{v1}) h_{fg} = \dot{m}(\omega_2 - \omega_1) h_{fg} = \dot{Q}_L \quad (24)$$

where  $\dot{Q}_L$  represents the latent energy transfer associated with the evaporation of water added to the air.

Taking the ratio of equation 24 to 23,

$$\frac{\omega_2 - \omega_1}{h_2 - h_1} = \frac{\dot{Q}_L}{\dot{Q}_T} \quad (25)$$

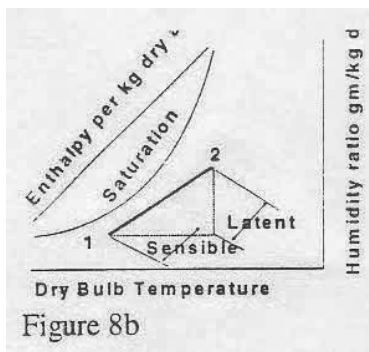
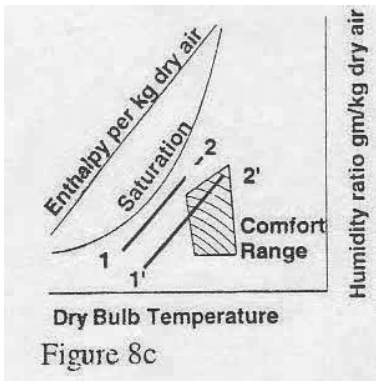


Figure 8b

We can see that the airflow rate into the room falls out and the ratio of  $\Delta\omega/\Delta h$  is determined by the ratio of latent to total energy input. That is, the state of the well mixed air in the room, state 2, must be on the line, called the condition line, with slope  $\Delta\omega/\Delta h$  which passes through the inlet state 1 as shown on Figure 8b.



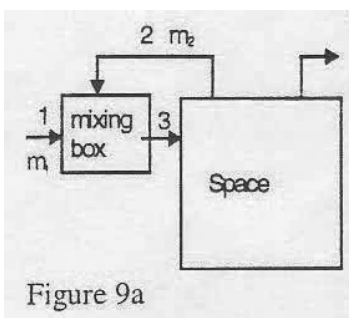
If we want to make sure the room falls within the comfort conditions, then state 1 must be located so that the line 1-2 passes through the range of comfort on the psychrometric diagram. For example, Figure 8c shows a typical condition line and summer comfort zone. With the original state of inlet air to the space, labeled 1, the condition line does not pass through the comfort zone no matter what airflow rate is used. The inlet air must be altered to state 1' to achieve comfortable conditions.

Table 1  
Rates of Heat Gain from Occupants in Conditioned Spaces

	Sensible (W)	Latent (W)
Seated, light work (e.g. MIT student)	70	45
Moderate dancing (BU student)	90	160
Athletics (Harvard student)	210	315

Taken from ASHRAE Handbook of Fundamentals, p 26.8, 1993

In a building having a common supply of conditioned air and rooms housing different activities with different ratios  $\Delta\omega/\Delta h$ , achieving comfortable conditions in all rooms may be difficult.



#### Adiabatic Mixing

To conserve fan energy and duct size in some buildings, the temperature of the cold air in the supply system of a building can be lowered (but at the cost of a decreased COP for the chiller). To prevent cold drafts on occupants near the diffuser outlet we can use a mixing box which mixes warmer room air with the cold supply air, see Figure 9. The question is: how to determine the dry bulb temperature and humidity of the mixture leaving the box? In most cases the box is small enough

so that heat transfer from the external surface of the box is negligible. We can write a mass balance for the air entering and leaving the box at steady state,

$$\dot{m}_3 = \dot{m}_1 + \dot{m}_2 \quad (26a)$$

and a mass balance for the water vapor,

$$\omega_3 \dot{m}_3 = \dot{m}_1 \omega_1 + \dot{m}_2 \omega_2 \quad (26b)$$

which can be rewritten as,

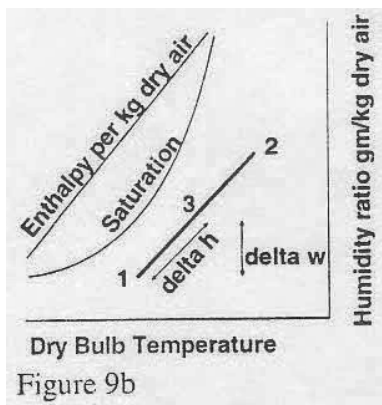
$$\omega_3 = \frac{\dot{m}_1}{\dot{m}_3} \omega_1 + \frac{\dot{m}_2}{\dot{m}_3} \omega_2 \quad (26c)$$

Similarly an energy balance using the mixture enthalpies becomes,

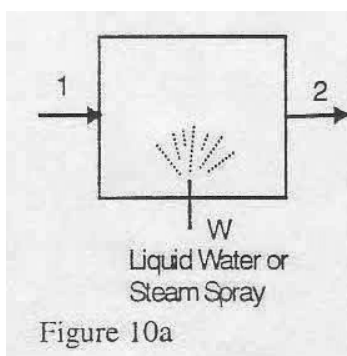
$$\dot{m}_3 h_3 = \dot{m}_1 h_1 + \dot{m}_2 h_2 \quad (27a)$$

which can be written as,

$$h_3 = \frac{\dot{m}_1}{\dot{m}_3} h_1 + \frac{\dot{m}_2}{\dot{m}_3} h_2 \quad (27b)$$



When the supply air flow  $\dot{m}_1$  is equal to the room air to the box,  $\dot{m}_2$ , it is easy to see that the enthalpy of the mixture at state 3 is the average of  $h_1$  and  $h_2$ , while  $\omega_3$  is the average of  $\omega_1$  and  $\omega_2$ . State 3 shown of the figure is half way along a straight line between 1 and 2. For any other mixture proportions it can be shown that state 3 is always on the straight line between 1 and 2 with its location set by the proportion of cold air to recycled air in the mixing box.



Adiabatic Water Addition to Air Stream

In some instances, the air entering a building has too small an amount of water vapor. Water or steam sprays are used to add moisture. In earlier times, people would put out pans of liquid water. Water vapor evaporating from the surface would mix with the air passing over. Sometimes this technique can be used in hot dry climates to supplement or replace the conventional air conditioning system.

We will assume that the spray mixes with the air stream in a small mixing box which is adiabatic, with no heat transfer from the outside surface of the box. See Fig. 10a. Just as before we can write conservation of mass for the steady state process,

$$\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_1 \quad (28)$$

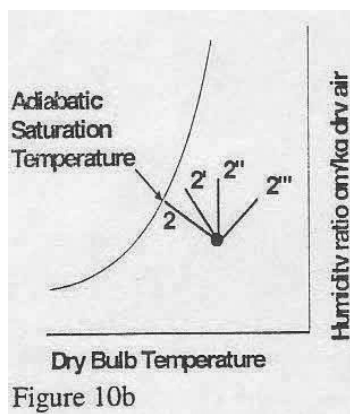
$$\dot{m}_w = \omega_2 \dot{m}_{a2} - \omega_1 \dot{m}_{a1} = (\omega_2 - \omega_1) \dot{m}_1 \quad (29)$$

and the energy balance becomes,

$$\dot{m}_1 h_1 + \dot{m}_w h_w = \dot{m}_2 h_2 \quad (30)$$

Note this holds whether the spray is liquid water or steam. Dividing both sides of eqn. 30 by  $\dot{m}_{a1}$ , we can find an expression for  $h_2$ ,

$$h_2 = (\omega_2 - \omega_1) h_w + h_1 \quad (31)$$



When the spray is a cold liquid, the enthalpy of the liquid stream is small compared to that of the air mixture and as a good first approximation, the enthalpy of the air mixture leaving the mixing box is equal to the enthalpy of the mixture entering. On the psychrometric chart the process line 1 to 2 moves at constant  $h$  from a low humidity ratio and high dry bulb temperature to a final state at higher humidity and lower dry bulb temperature. The energy necessary to evaporate the liquid water comes from the air, causing a lower temperature. In a way this could be termed free cooling, but the price is water consumption.

When hot liquid or steam with progressively higher enthalpy is sprayed into the air, eqn. 31 indicates that the enthalpy of the air leaving the mixing box will increase. Figure 10b illustrates this. When the enthalpy of the steam in the spray is the same as the enthalpy of the water vapor in the mixture entering, the leaving mixture is at the same temperature as the entering mixture.

### Adiabatic Saturation Temperature

If the water spray in Figure 10a is sufficiently large, the mixture leaving at state 2 is saturated. The temperature,  $T_2$ , of the mixture leaving the mixing box is called the adiabatic saturation temperature. This provides another means of measuring the unknown moisture content of a mixture at, say, state 1. Rather than using a spray, the air at state 1 is passed over a long liquid water surface. If the surface area is large enough, the leaving stream, at state 2 is saturated. By measuring the dry bulb temperature of states 1 and 2 and knowing state 2 is saturated with water vapor, an energy balance or the psychrometric chart will allow us to find the humidity ratio of state 1. Thus, a mixture of air and water vapor with a given dry bulb temperature and humidity ratio has a unique value of adiabatic saturation temperature.

When a closed amount of liquid is used in the adiabatic saturation apparatus, the liquid temperature,  $T_w$ , approaches the outlet temperature  $T_2$  so  $h_w$  in equ. 30 can be accurately determined.

Since the liquid enthalpy is a small component of equation 31, lines of constant enthalpy are approximately parallel to lines of constant adiabatic saturation temperature.

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Fall 2008

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