PDHonline Course M409 (3 PDH)

HVAC Psychrometric Analysis to Avoid Moisture Problems

Instructor: Fred W. Dougherty, P.E., B.A.E., M.M.E.

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5272 Meadow Estates Drive
Fairfax, VA 22030-6658
Phone & Fax: 703-988-0088
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COURSE CONTENT

1. Scope

This course reviews the application of psychrometrics to the analysis and selection of HVAC cooling systems. At this stage of the design process, heating and cooling loads will have been computed for the project, and an initial selection of heating and cooling apparatus will have been made. Thus, the building room loads and cooling coil capacities are known. Psychrometric analysis is applied to the cooling system because the cooling system is the source of moisture problems and "sick building syndrome". After psychrometric analysis, it may be necessary to repeat the selection process to avoid moisture problems revealed.

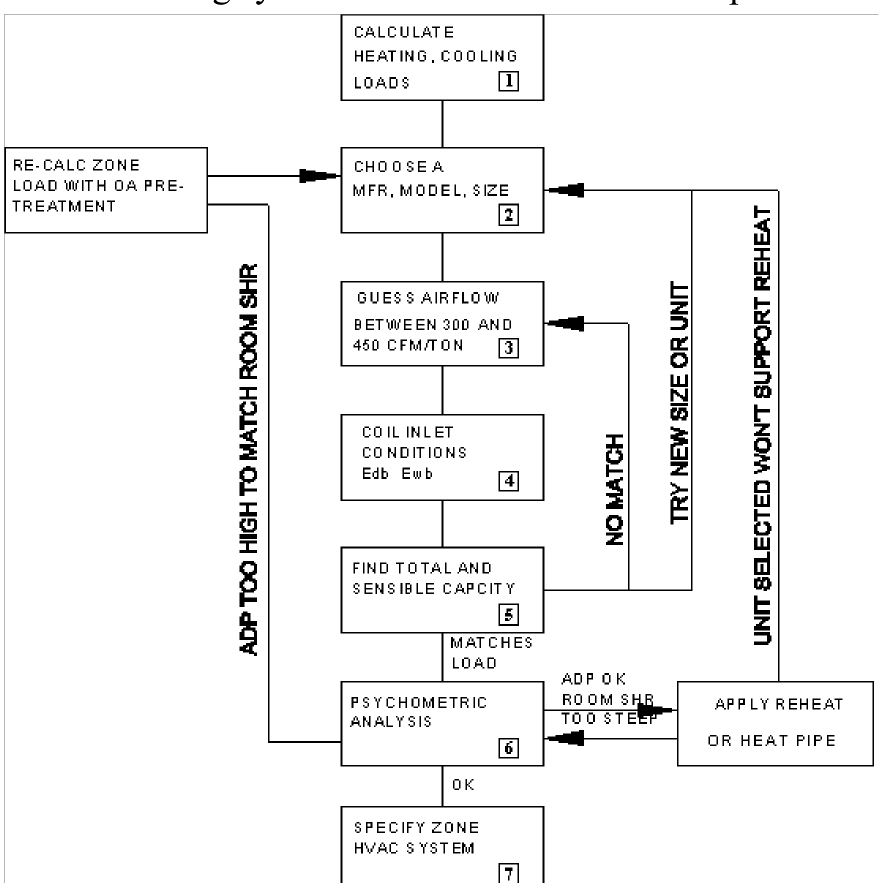


FIGURE 1, HVAC EQUIPMENT SELECTION FLOW CHART

2. The Design Iteration

Psychrometric analysis is only one step in the process of DX cooling system design and equipment selection. This process is shown as a flow chart in figure 1. Steps 1 through 5 follow the process of finding a manufacturer, model, and size to match the total and sensible capacity with the calculated loads. Once a match is found, step 6, psychrometric analysis is required to ensure that the selected equipment can supply conditioned air that will follow the room load process line. How this is done is explained in this course.

3. Definitions

The following terms will be used throughout this course.

A **building** is a roofed and walled structure with controlled environment, built for human occupation and use.

A **ceiling plenum** is a cavity within the pressure envelope that is above a room and that is formed by a dropped lay-in ceiling and floor or roof structure above. Room walls do not necessarily extend above the dropped ceiling to the structure above.

The **pressure envelope** is the primary air barrier of the building, which is sealed to provide the greatest resistance to air leakage from the unconditioned environment. One or more **zones** may be within a building pressure envelope.

Return air is the portion of the supply air that is recirculated to the cooling/heating apparatus after being collected by the return grilles in the zone.

A **return air plenum** is a ceiling plenum with an unobstructed path to an air handler return, and that contains no flammable materials or surfaces.

A **room** is the part of a space bounded by walls and a ceiling that is usually routinely occupied and served by grilles and registers to supply and recirculate or exhaust conditioned air. The term **room** may denote **zone** on the psych chart.

A **space** is a single room, with or without a ceiling plenum.

Supply air is the all of the air delivered by the cooling/heating apparatus to the supply air diffusers in the zone.

The **supply air critical state** is the maximum temperature and dew point of the supply air that will satisfy the zone design temperature and relative humidity at design air flow and cooling load.

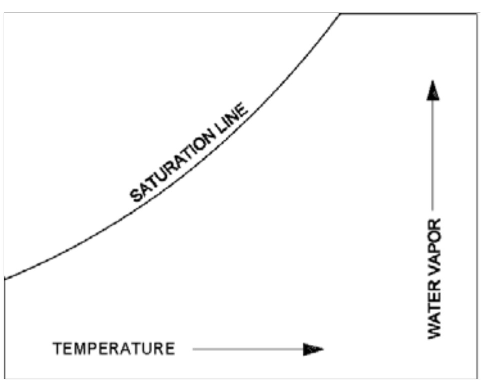
Ventilation air, also called **outdoor air,** is air from outdoors that may be mixed with return air before passing into the cooling/heating apparatus, may be introduced to the apparatus directly before entering a zone, or in certain rare circumstances, may be introduced un-tempered into a zone.

A **zone** is a group of spaces or rooms within the thermal and pressure envelopes which are served by a single air handling system.

4. The Psychrometric Chart

This course is about applying psychrometrics to actual air conditioning projects. It is assumed that the student is already familiar with psychrometric processes. The following brief refresher will hopefully help brush away any cobwebs.

The psychrometric chart maps two primary and six secondary properties of air. The primary properties are dry bulb temperature and water vapor concentration. All of the properties are expressed as concentrations in dry air. The psychrometric chart shown below has two principle axes – the horizontal axis is dry bulb temperature, **tdb,** in ˚C or ˚F, and the vertical axis is water vapor content, usually labeled as humidity ratio, **W,** expressed in IP units as pounds of moisture per pound of dry air (lbs H2O/lb) or grains of moisture per pound of dry air (gr/lb). There are 7000 grains to one pound.



On the saturation line, all of the water vapor in the air has condensed to liquid. There is no stable state to the left of the saturation line.

The six secondary properties are:

wet bulb temperature **twb** – The lowest temperature that can be reached by the evaporation of water only.

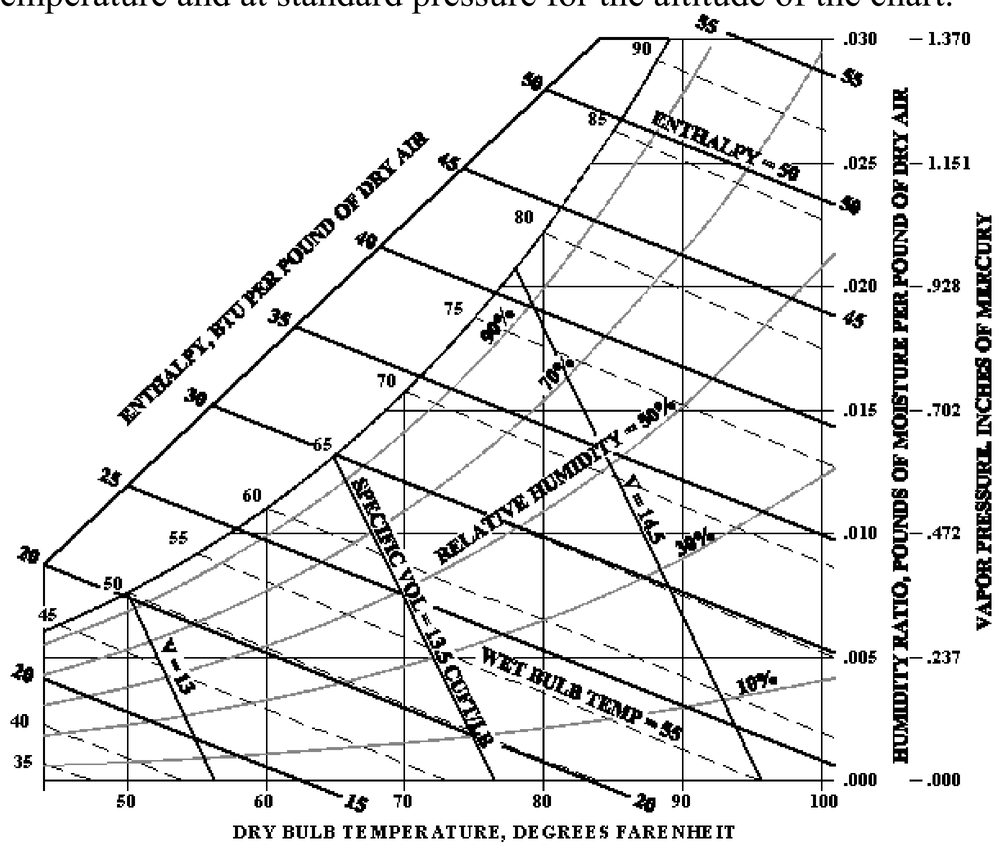
relative humidity, **rh** – the amount of water vapor in air at a given temperature as a percentage of the total amount of water vapor in saturated air at the same dry bulb temperature. Also, the vapor pressure of the air-water vapor mixture as a percentage of the vapor pressure of saturated air at the same dry bulb temperature.

dew point temperature, **tdp** – the temperature to which an air-water vapor mixture must be cooled for the water vapor to condense into liquid water.

enthalpy, **h** – the specific energy of the moist air, expressed in IP units as Btu/lb of dry air.

specific volume, **v** – the total volume of dry air and water vapor in units of cubic feet per pound of dry air (ft^3/lb). This is the reciprocal of the density of the air/water vapor mixture, lb/ft^3.

water vapor pressure, **Vp** - the pressure exerted by the molecules of water vapor in a saturated mixture of water vapor and dry air at a given temperature and at standard pressure for the altitude of the chart.



The Psychrometric Chart, IP Units

Note that these are "secondary" because they are all fully defined by dry bulb temperature and humidity ratio alone. However if any two of the nine properties are known, all of the others can be found by reading from the chart or performing a calculation. Also, dew point and vapor pressure are both defined by horizontal lines that intersect the vertical axis, so that for any humidity ratio (vertical axis), there is a unique value of **tdp** and **Vp**.

The curved line bounding the left of the chart is the saturation line. On the saturation line, the dry bulb temperature is the dew point and wet bulb temperature, and relative humidity is 100%. Thus, on the saturation line:

tdb = **tdp** **tdb** = **twb** and **rh** = 100%

Any point on the psychrometric chart is a **state point** because it defines the thermodynamic state – the temperature and humidity ratio - of the air at that point. If the air at a given state is acted upon by an external process such as heating, mechanical cooling, or humidification it will be changed to another state. The path between the initial and final states is a **process line**.

If any two properties of air are known, then all of the others can be found by reading from the chart or making a computation. For example, if outdoor air temperature and relative humidity are known, say 90 ˚F and 70% rh, that point can be plotted on the chart above, thus fully defining all of the properties of the outdoor air at that point, including wet bulb temperature, dew point, enthalpy, and specific volume.

Air conditioning cooling load is defined in terms of **sensible heat load** and **total heat load**. Likewise, a mechanical cooling coil, DX or chilled water, is defined in terms of **sensible heat capacity** and **total heat capacity**. A change in the **sensible heat** of a mass of air creates a measurable change in its dry bulb temperature, **tdb**. A change in the **latent heat** creates a measurable change in the humidity ratio, **W.** **Total heat** is the sum of the latent heat and sensible heat of a mass of air. A change in the **total heat** of a mass of air is equivalent to a change in its enthalpy, **h**. These changes are quantified by the following formulas:

$$\Delta Q_s = \Delta \mathbf{tdb} * Cc * rho * c_p * \text{ min/hr}$$
$$\Delta Q_l = \Delta \mathbf{W} * Cc * rho * Q_{LV} * \text{min/hr}$$
$$\Delta Q_t = \Delta \mathbf{h} * Cc * rho * \text{min/hr} = \Delta Q_s + \Delta Q_l$$

where (IP units) ΔQ_s is the change in sensible heat of the flowing air (Btu/hr)
ΔQ_l is the change in latent heat of the flowing air (Btu/hr)
ΔQ_t is the change in total heat of the flowing air (Btu/hr)
Cc is the air flow rate (ft^3/min)

rho is the average air density of the flowing air (lb/ft^3)
c_p is the specific heat of air (Btu/lb/˚F)
Q_{LV} is the latent heat of vaporization of water (Btu/lb H2O)

for air at standard conditions:
rho = .075, c_p = .24, Q_{LV} = 1076

$\Delta Q_s = \Delta \mathbf{tdb} * Cc * 1.08$ (1)
$\Delta Q_l = \Delta \mathbf{W} * Cc * 4840$ (2)
$\Delta Q_t = \Delta \mathbf{h} * Cc * 4.5$ (3)

In this course, loads and capacity will be expressed in terms of sensible heat and total heat, because that is how most air conditioner system published performance data is expressed.

Paper psychrometric charts suitable for plotting are often available from HVAC manufacturer's representatives. 50 sheet pads of charts similar to the ones used for this course are available from the ASHRAE bookstore. Also, there are numerous software programs for plotting psychrometric analysis which may be found on the web. This course was developed using the ASHRAE Psychrometric Analysis Program, available from ASHRAE on CD. A blank chart is included at the end of the course that can be printed out for plotting exercises.

5. Cooling System Psychrometric State Points

In this course, psychrometric state points will be identified as follows:

1 – room air
1A – **supply air critical state** point – not a physical point
2 – outdoor air
3A – mixed air entering heat pipe or pre-cool coil, where applicable
3 – mixed air entering cooling coil
4 – air leaving cooling coil
4A – air leaving heat pipe or reheat coil, where applicable

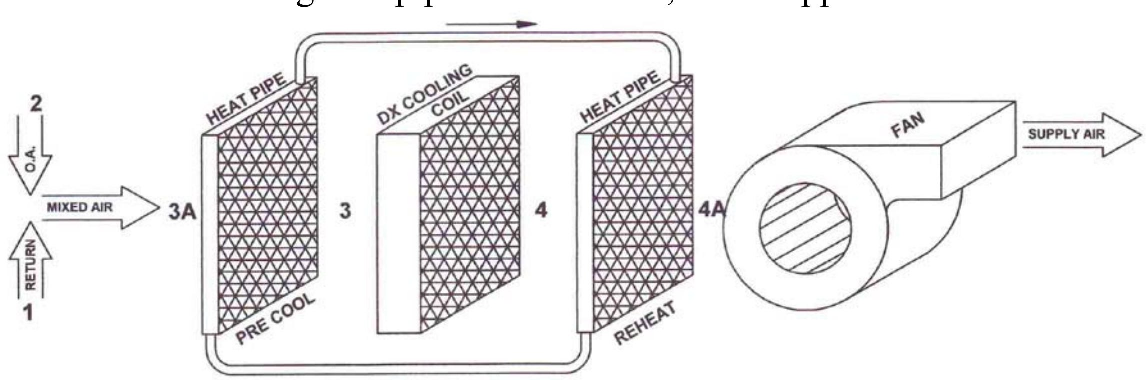


Figure 2 – DX Cooling System Schematic

Figure 2 is a schematic representation of a cooling system, showing the locations of the state points except for point **1A**, which is a calculated, not a physical, point.

Figure 3 is a typical psychrometric chart with important parameters labeled. The cooling design point shown is for a five ton rooftop unit serving a small office zone. (The term "room" can be taken to mean all of the rooms in the zone.) Room air at 76° and 57% relative humidity is mixed with outdoor air at 94° dry bulb and 78° wet bulb. The mixed air passes over a cooling coil, where it is cooled to 59.6° at 91% rh and then delivered to the room (zone) to carry away cooling load heat and moisture.

The most important elements of the psychrometric cycle, from the standpoint of the engineer, are room and coil sensible heat ratio, room dew point, and apparatus dew point. Plotting these on a psych chart, will quickly tell you whether a simple system will do the job, or whether additional components and capabilities will be needed.

Process Lines

The lines on a psychrometric chart we will call "process" lines. Referring to figure 3, line **1-1A** is the "room process line". Point **1A** is the **supply air critical**

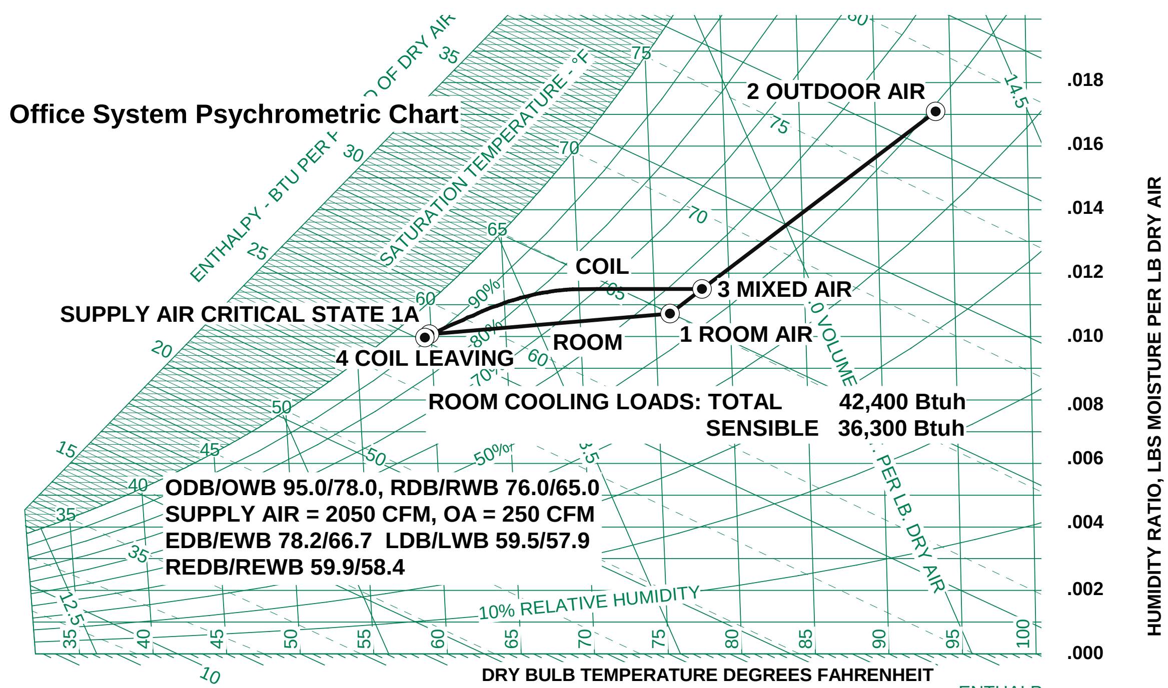


FIGURE 3

5 TON PACKAGED AC, SENSIBLE COOLING CAPACITY 41,960 Btuh, TOTAL COOLING CAPACITY 57,430 Btuh
COIL COOLING LOADS: SENSIBLE 41,300 Btuh, TOTAL 55,600 Btuh

state point and represents the maximum state of the supply air that will satisfy the room load. As the supply air cools and dehumidifies the room, it will follow the slope of the room process line. Thus, the air leaving the cooling apparatus must be at the same or lower temperature and dew point than point **1A** or the desired room temperature and relative humidity cannot be maintained at design conditions. Line **3-4** is the coil process line. Supply air at the flow rate (cfm) of the cooling apparatus enters the coil at state point **3** and leaves the coil at state point **4**, where it is introduced into the room, picks up the sensible and latent load, and then is returned to the cooling apparatus for reprocessing.

For a cooling system to be able to satisfy the design conditions of the room and outdoor air loads, the coil process line must satisfy two conditions. First, the coil leaving air temperature, point **4** or **4A**, must be low enough to significantly dehumidify the air, generally under 60°F. Second, the coil process line must cross the room process line so that the supply air dew point and dry bulb temperature is less than point **1A**. That is, point **4** must be cooler and drier than point **1A**, as shown on Figures 3 and 4. These conditions may be difficult to meet with ordinary DX cooling systems. If either condition is not met, then additional processes may be necessary, such as outdoor air pre-treatment, mixed air pre-cooling, or coil leaving air reheat. These processes must be tailored to cause the supply air, point **4** or **4A**, to end up on or below and cooler than point **1A**.

Plotting Points on the Psychrometric Chart

To do psychrometric analysis, it is necessary to use known data to calculate the unknown points. Referring to Figure 3, the known data are as follows:

odb = **tdb_2** = outdoor air dry bulb temperature, °F
owb = **twb_2** = outdoor air wet bulb temperature, °F
rdb = **tdb_1** = room air dry bulb temperature, °F
rwb = **twb_1** = room air wet bulb temperature, °F
Qps return air plenum sensible heat gain (if any)
Qrs room sensible cooling load, Btu/hr
Qrt room total cooling load, Btu/hr
Coa outdoor air ventilation rate, cfm
Cc supply air - estimated or actual coil air flow rate, cfm
Cr room return (and plenum) air flow rate = **Cc - Coa**
Qscc estimated or actual coil sensible cooling capacity, Btu/hr
Qtcc estimated or actual coil total cooling capacity, Btu/hr

Remember that total cooling load is sensible cooling load plus latent cooling load.

Knowing the parameters listed on page 9, we can find the other points on the chart. Each of the charts presented in this course will show the values of the known parameters, so that the student may compute the state points and plot the process lines himself on a paper chart, or with software.

Note that it is easiest to find dry bulb and wet bulb temperatures on the chart, although it may be necessary to use enthalpy in the calculation procedure. The cooling coil entering state, point **3**, is a mixture of room air and outdoor air, and lies on a line connecting the two states on the chart – see figure 3. The mixed dry bulb temperature is a linear function of the two air flows, outdoor air (**Coa**) and return (**Cr**) as follows:

$$\mathbf{tdb_3 = (t_1 * Cr + t_2 * Coa)/(Cr+Coa)} \qquad (4)$$

The value of $\mathbf{twb_3}$ is most easily found by reading it from the chart at the intersection of $\mathbf{tdb_3}$ and the line connecting points **1** and **2**.

Note that the coil air flow rate is the sum of the air returning from the room or zone and the outdoor air ventilation air flow. All of the coil air flow is delivered to the room (zone), where a portion of it is exhausted or leaks out of the building.

To calculate the dry bulb and wet bulb temperatures for points **1A** and **4** it is necessary to know the enthalpies of points **1** and **3**. These can be looked up on the chart, or found using psychrometric software, as a function (**f**)of dry bulb and wet bulb temperatures (**tdb** and **twb**).

$$\mathbf{h = f(tdb, twb)} \text{ thus } \mathbf{h_1 = f(tdb_1, twb_1)} \text{ and } \mathbf{h_3 = f(tdb_3, twb_3)} \qquad (5)$$

and continuing to find points **1A** and **4**, referring to equations (1) and (3)

$$\mathbf{tdb_{1A} = tdb_1 - Qrs / Cc / 1.08} \qquad (6)$$
$$\mathbf{h_{1A} = h_1 - Qrt / Cc / 4.5} \qquad (7)$$
$$\mathbf{twb_{1A} = f(tdb_{1A}, h_{1A})} \qquad (8)$$

$$\mathbf{tdb_4 = tdb_3 - Qscc / Cc / 1.08} \qquad (9)$$
$$\mathbf{h_4 = h_3 - Qtcc / Cc / 4.5} \qquad (10)$$
$$\mathbf{twb_4 = f(tdb_4, h_4)} \qquad (11)$$

where **Qrs** = room (zone) sensible cooling load
Qrt = room (zone) total cooling load
Qscc = coil sensible cooling capacity, Btu/hr
Qtcc = coil total cooling capacity, Btu/hr
Cc = supply air flow rater

Ceiling Return Air Plenums:

Figure 4 is the same building as represented by Figure 3 but with a return air plenum. For this zone, it is estimated that the plenum will receive half the roof cooling load and 30% of the lighting load. (Guidelines for determining plenum cooling loads are vague, and require application of experience and judgment by the designer.) The effect of plenum heat is to increase the temperature entering the coil, which primarily increases the sensible capacity of the coil. This heat does not appear as "room" load, as it does in Figure 3 which is a zone with no return air plenum. Thus, the room loads are decreased by the load picked up in the plenum, and the combination of increased capacity and reduced room load results in a significant reduction in supply air flow. Note the different values of room and plenum load and supply air flow on Figures 3 and 4.

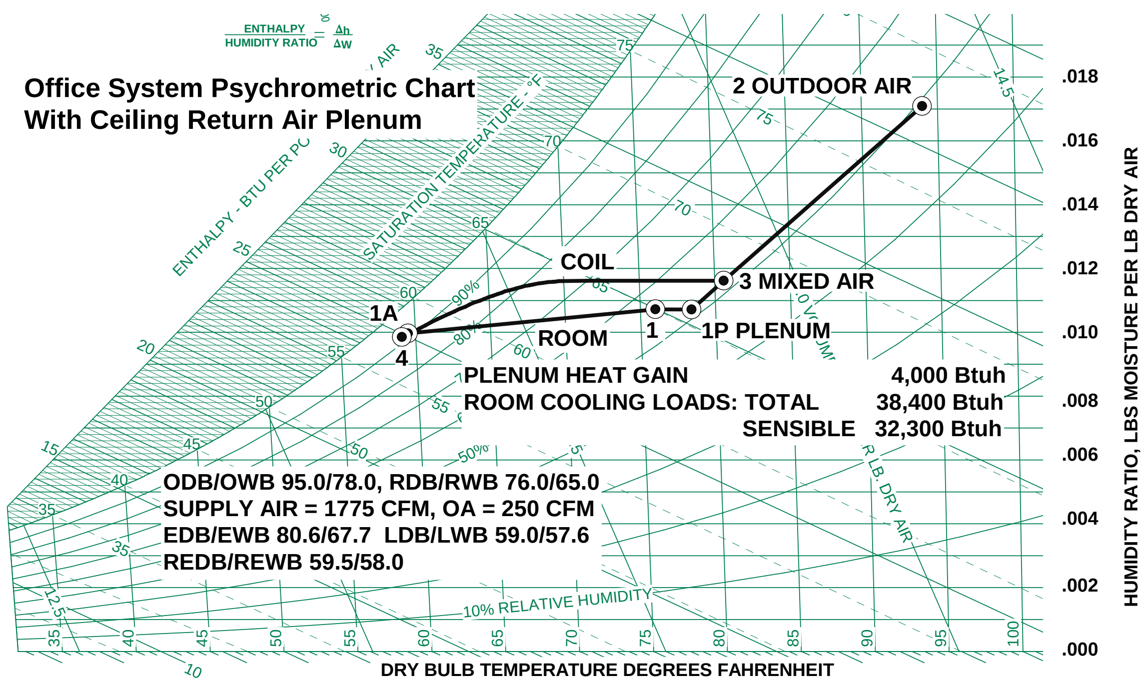


FIGURE 4

5 TON PACKAGED AC, SENSIBLE COOLING CAPACITY 41,960 Btuh, TOTAL COOLING CAPACITY 57,430 Btuh
COIL COOLING LOADS: SENSIBLE 41,190 Btuh, TOTAL 55,700 Btuh

If heat is added to a return air plenum, then the mixed air state point is shifted at constant dew point as shown on figure 4 above. The equations then become:

$$\mathbf{tdb_3 = ((t_1+\Delta tp) * Cr+t_2 * Coa)/(Cr+Coa)} \tag{12}$$
$$\mathbf{\Delta tp = Qps/(Cr*1.1) \;\; and \;\; \Delta tdp_{(1-1P)} = 0} \tag{13}$$

and $\mathbf{twb_3}$ is found by reading it from the chart at the intersection of $\mathbf{tdb_3}$ and the line connecting points **1P** and **2**

Computing state points and process lines for Figure 4

Figure 4 is an opportunity to review the application of the equations given for computing state points and process lines. First, known parameters:

room dry bulb temperature ,	$\mathbf{tdb_1}$ **= 76**°F
room wet bulb temperature,	$\mathbf{twb_1}$ **= 65**°F
outdoor air dry bulb temperature,	$\mathbf{tdb_2}$ **= 95**°F
outdoor air wet bulb temperature,	$\mathbf{twb_2}$ **= 78**°F
room sensible heat load,	**Qrs = 32,300** Btu/hr
room total heat load,	**Qrt = 38,400** Btu/hr
plenum sensible heat load,	**Qps = 4,000 Btu/hr**
cooling coil sensible heat capacity,	**Qscc =42,000** Btu/hr
cooling coil total heat capacity,	**Qtcc = 57,500** Btu/hr
supply air flow,	**Cc = 1,775** cfm
outdoor air flow,	**Coa = 250** cfm

These "known" values have been developed as part of the design process (steps 1 through 5 on the design flow diagram, figure 1) which precedes psychrometric analysis. The source of each is briefly described below:

The outdoor air dry bulb and wet bulb temperatures, $\mathbf{tdb_2}$ and $\mathbf{twb_2}$ are determined from weather data for the particular location and design basis month. The "room" or zone temperatures $\mathbf{tdb_1}$ and $\mathbf{twb_1}$ are comfort conditions set by the designer.

The room cooling loads **Qrs** and **Qrt** are all of the heat and moisture gain within the occupied spaces of the zone (possibly including multiple rooms) which must be removed by the cooling system supply air. They do not include ventilation (outdoor air) loads or plenum heat gain. They do include building envelope loads, (heat gain through walls, ceiling, and glass), internal loads (people, lighting, appliances), and infiltration. Infiltration can be ignored if the building is kept under positive pressure by a surplus of ventilation air over exhaust air.

Ventilation air **Coa** is outdoor air that is mixed with return air and passes over the cooling coil before being introduced into the zone. It is determined either by the requirement to exceed required exhausts, such as restroom or kitchen exhaust, or by indoor air quality standards as defined in ASHRAE Standard 62.1, whichever is greater. The procedure for determining required ventilation air is described in PDH course #M384, **HVAC Ventilation for Indoor Air Quality.**

Coil cooling capacities **Qscc** and **Qtcc** are found from manufacturer's data as a function of air flow **Cc** and coil entering dry bulb and wet bulb temperature. They are selected for a specific cooling system to match as closely as possible the coil cooling loads. This is represented by the iteration loop from step 5 back to step 3 of Figure 1. The coil loads consist of the room or zone loads plus the ventilation load, plus the plenum load. The coil load process line is not shown on a psychrometric chart, but if it were, it would be a line connecting point **3**, which is the coil inlet condition, to point **1A**.

As noted earlier, the room return air flow **Cr** is known because **Coa** and **Cc** are known:

$$\mathbf{Cr = Cc - Coa} = 1{,}525 \text{ cfm}$$

Thus, from equation (13) $\quad \mathbf{\Delta tp = Qps/Cr/1.08} = 2.4^\circ\text{F}$

On the psych chart, pure heating always occurs at constant dew point, because heating never adds or removes moisture from the air. Since plenum heat is pure sensible heat, state point **1P** is plotted at the same dew point as point **1**, plus 2.4˚ dry bulb. Point **1P** is thus plotted at $\mathbf{tdb_{1P}} = 76^\circ\text{F} + 2.4^\circ\text{F} = 78.4^\circ\text{F}$, $\mathbf{tdp_1} = \mathbf{tdp_{1P}} = 59.1^\circ\text{F}$. Knowing the dry bulb temperature and the dew point temperature at point **1P**, all other state parameters, such as wet bulb temperature and enthalpy, can be found either by reading from the psychrometric chart or using a psychrometric software program. Thus $\mathbf{twb_{1P}} = 65.8^\circ\text{F}$.

The mixed dry bulb temperature can now be found using equation (12):

$$\mathbf{tdb_3 = ((tdb_1+\Delta tp) * Cr+tdb_2 * Coa)/(Cr+Coa)} = 80.7^\circ\text{F}$$

As indicated by the discussion of equation (4), the wet bulb temperature at point **3** is located at **tdb_3** and a line connecting point **1P** with point **2**. Most psychrometric software programs will also compute a mixed state point. Thus, reading from the chart, **twb_3** = 67.7˚F.

Using equation (5), we find the enthalpies of points **1** and **3** as a function of dry bulb and wet bulb temperature at points **1** and **3**, needed to compute the required supply air state, point **1A**, and the coil leaving state, point **4**.

$$h_1 = f(tdb_1, twb_1) = 29.98 \text{ Btu/lb} \qquad h_3 = f(tdb_3, twb_3) = 32.09 \text{ Btu/lb}$$

From Eq. (6), (7), (8)

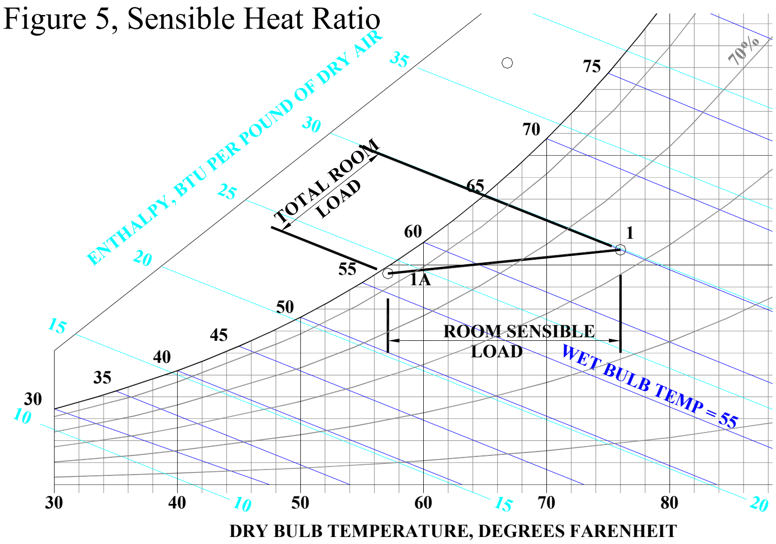
$$tdb_{1A} = tdb_1 - Qrs/Cc/1.08 = 59.5\text{˚F}$$
$$h_{1A} = h_1 - Qrt/Cc/4.5 = 25.17 \text{ Btu/lb}$$
$$twb_{1A} = f(tdb_{1A}, h_{1A}) = 58.0\text{ ˚F}$$

From Eq. (9), (10), (11)

$$tdb_4 = tdb_3 - Qscc/Cc / 1.08 = 59.2\text{˚F}$$
$$h_4 = h_3 - Qtcc/Cc / 4.5 = 24.90 \text{ Btu/lb}$$
$$twb_4 = f(tdb_4, h_4) = 58.0\text{ ˚F}$$

Sensible Heat Ratio

Figure 5, Sensible Heat Ratio



Sensible heat ratio is the ratio of sensible cooling load or capacity to total cooling load or capacity. Room loads include only those loads that originate in the space itself, most notably excluding the load from outdoor air ventilation, duct leakage from or to spaces outside the room, or heat added to a return air plenum.

Sensible loads or capacities are represented on a psychrometric chart by the difference between the high and the low dry bulb temperature of the air stream. Total loads or capacities are represented by the difference between the high and low enthalpy of the air stream. This is shown on Figure 5 for the room loads. It can be seen from this, that the slope of the room process line represents the room sensible heat ratio. Latent loads or capacities are the difference between sensible and total loads or capacities

From formulas (1) and (3) the equations that represent sensible heat ratio are as follows:

$$\mathbf{Qrs = (t_1 - t_{1A})\ \ x\ \ Cc\ \ x\ \ 1.1} \qquad (14)$$
$$\mathbf{Qrt = (h_1 - h_{1A})\ \ x\ \ Cc\ \ x\ \ 4.5} \qquad (15)$$
$$\mathbf{SHR = Qrs/Qrt} \qquad (16)$$

The term "load" applies to the cooling coil only in the context of the total computed loads that have to be removed by the cooling apparatus. On figures 3 and 4, the coil "load" would be represented by a process line from point **3** to point **1A**. The task of the engineer is to select a cooling apparatus with <u>capacity</u> that matches as closely as possible the <u>load</u> represented by the room process line. After he has selected a system, then the capacity of the coil at a particular air flow with entering conditions at state point **3** is represented by the coil process line terminating at point **4**. The convention used here, therefore, is for the sensible heat ratio of the coil to be defined as the sensible heat <u>capacity</u> of the coil divided by its total heat <u>capacity</u>.

Dew Point

The dew point is defined as the temperature at which the water vapor in air will condense. If a room is at a particular humidity ratio, then the dew point of that room will be the dry bulb temperature at the point where a horizontal line from the room state point crosses the saturation line on the psych chart. The apparatus dew point (ADP) is the point where horizontal line from the coil leaving state point, point **4**, crosses the saturation line. It is the dew point of the coil tube and fin surfaces. The bypass factor accounts for the fact that not all of the air passing across the coil comes in contact with a cold tube or fin surface, and is therefore not dehumidified. Bypass factors range from about 4% to as high as 11%.

The sensible and latent cooling capacity of a coil and the bypass factor can be found from manufacturer's tables as a function of the psychrometric conditions of the air entering the coil and the air flow rate. Bypass factor is included in the capacity charts, and may also be shown separately for use in calculating certain processes.

6. Load Variations

There are four basic design point load conditions that the designer will encounter when working with small commercial and institutional projects: 1) moderate occupant density and ventilation air; 2) high occupant density with moderate ventilation air; 3) high ventilation air with moderate occupant density; and 4) high occupant density with high ventilation air. Psychrometric analysis will help determine which of these conditions exists, what special processes will be needed when condition 1) is not met, and how to tailor those processes to match the load.

When the SHR of a building or zone falls below the capabilities of standard air conditioners, then additional design elements will be needed to avoid indoor moisture and mildew problems. These may include pre-treatment of outdoor air, or reheat of supply air to false load the system and cause it to run more. Another option to handle both large quantities of outdoor air and high occupant density is to provide a unit with both hot gas bypass and reheat. The key to these choices when selecting equipment is an understanding of the psychrometrics.

Figures 3 and 4 show how moderate load conditions appear on the psychrometric chart. For the remainder of this course, we will show the psychrometric characteristics of the other basic design point conditions, and how to use the psychrometric chart to select and tailor the appropriate special processes when necessary.

High Occupant Density:

High Occupant density – generally more than 10 occupants per 1000 sf - results in a low room sensible heat ratio because human occupants are a source of latent, or water vapor, cooling load. This is true even if the other primary source of latent load, ventilation air, is moderate.

Figure 6 shows the design point condition of high occupant density with moderate ventilation air. The slope of the room process line is very steep, and the temperature at point **1A** is higher than the dew point at the room design condition. It is clear that the mixed air from point **3** cannot be brought directly to point **1A** because it will not have been cooled enough to remove the moisture indicated by the dew point at point **1A**. (Just because a line can be drawn on a psychrometric chart, there is no guarantee that there is a physical process that can follow that line.)

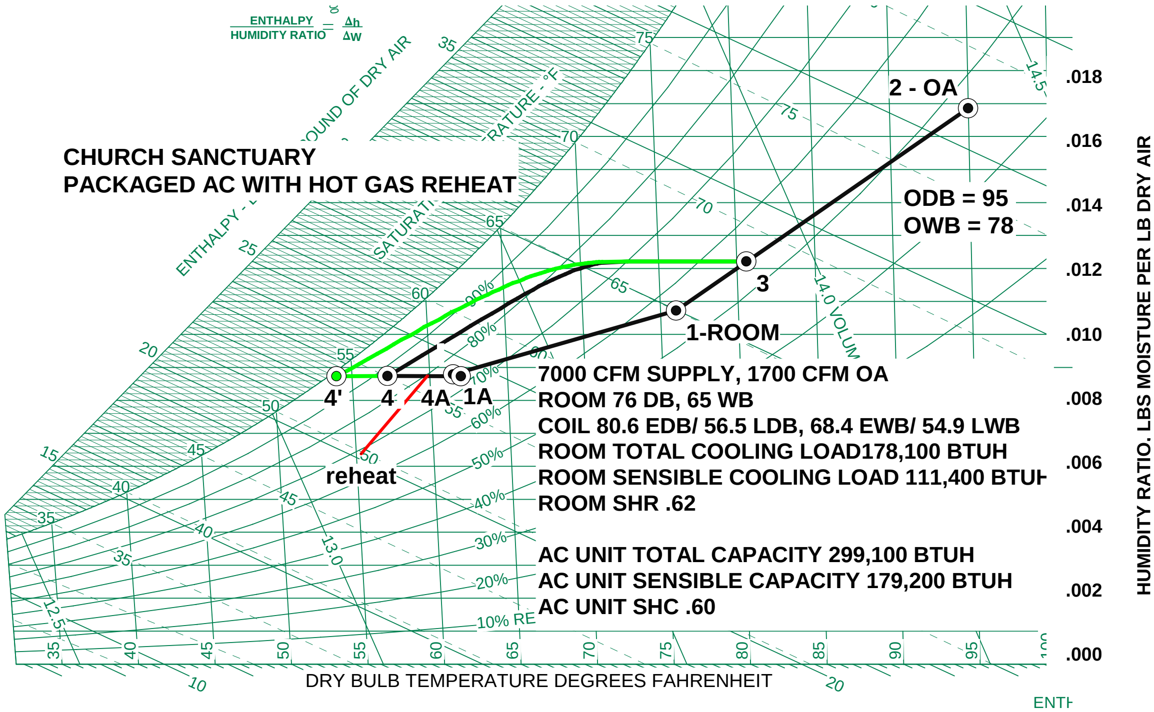


FIGURE 6 - LOW ROOM SHR

The particular project depicted on Figure 6 requires a packaged AC unit. Following steps 1 – 5 of Figure 1, a unit was selected that is large enough to satisfy the mixed air cooling load. Note also that the coil process line **3 – 4** results in a higher than usual temperature, and that the process line does not cross the saturation line when extended. This is because the manufacturer has included in the unit performance a reheat coil that uses the warm liquid leaving the condenser. The green line shows the approximate actual process line for this unit.

The problem with the coil process line for Figure 6 is that the sensible cooling capacity, even with the liquid line reheat, is far too large. Basic air conditioners respond only to zone dry bulb temperature. A low sensible cooling load combined with a low SHR required the ac unit to be over-sized for sensible capacity in order to for point **4** to be below the dew point of point **1A**. Over-sizing for sensible capacity will cause zone air temperature to fall rapidly when the air conditioner is running, and then to rise slowly, resulting in long off cycles for the compressor. The cooling coil de-humidifies only when the compressor is running, so a long off cycle allows zone humidity to rise. This can be devastating in jurisdictions where commercial and institutional air handling equipment is required by code to run continuously during occupied periods, thus actually inducing warm, moist outdoor air during the compressor off cycle.

The solution to this problem is shown on Figure 6. The air leaving the cooling apparatus at point **4** is heated until it intersects the room process line on or near point **1A**. This is accomplished by "hot gas reheat" which is an option available on most light commercial packaged AC units and on built-up split DX systems. The effect of reheat is to allow the AC compressor to run and dehumidify without over-cooling even when the room space temperature is satisfied and the thermostat is not calling for cooling. Reheat is controlled by a zone humidity sensor, and so responds to humidity rise caused by latent heat load.

Another problem when SHR is low, and blowers must run continuously, is that SHR decreases further as cooling load decreases from peak load. At the same time, as outdoor temperature falls, the air conditioner sensible heat capacity increases. Thus, an air conditioner that is satisfactory at peak load on a hot afternoon, may allow indoor humidity to rise unacceptably when outdoor temperatures are low. Because reheat actively controls humidity, it mitigates moisture problems under all load conditions.

High room SHR, more than .8, is generally less of a problem than low SHR because the cooling system will run in response to zone dry bulb temperature, at the same time dehumidifying. As outdoor temperature falls, the system with a high peak SHR will stay within the capabilities of the ac unit to control moisture.

Specialty spaces within a building may be deliberately designed for very high SHR – near 1.0 – both architecturally and mechanically. Large main-frame computer rooms are a good example. These require special air conditioning equipment and design. Help designing for these cases is available from HVAC manufacturer's representatives, and references such as the ASHRAE Handbook.

High Percentage of Outdoor Air

Ventilation air is induced from outdoors to remove odors, VOCs, and bio-aerosols and to inhibit infiltration. By definition, it is heated or cooled and dehumidified by air conditioning systems, and therefore adds sensible and latent cooling load to the equipment. When outdoor air exceeds about 25%, ordinary AC units may not be able to handle the outdoor air loads in addition to the room or zone loads. To solve this problem, it is necessary to pre-treat the outdoor air, removing sensible and latent heat before it passes into the air conditioning cooling coil, or pre-heating the air if the systems are in heating mode. Only the cooling mode is of interest for psychrometric analysis.

The two most common pre-treatment methods are the Energy Recovery Ventilator (ERV) and the Dedicated Outdoor Air (DOA) unit. Equipment is available that combines both methods, and may be applicable for special cases.

Energy Recovery Ventilator

Figure 7 represents a church foyer which requires 400 cfm of ventilation. The zone load is light, even though occupant density is fairly high, the room sensible heat ratio is a reasonable 68%, so reheat or heat pipes are not called for. The coil loads vs the best match that could be found with a DX air conditioner are shown in Table 1.

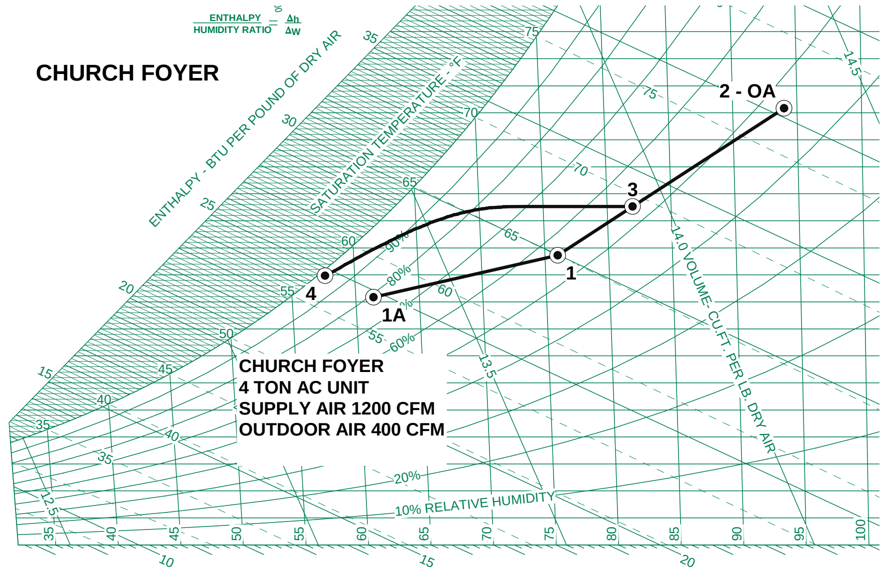


FIGURE 7 – FOYER, 33% OA

Table 1

4 ton ac – 1200 cfm	sensible - Btuh	latent - Btuh	total – Btuh
coil load	27,100	20,200	47,300
coil capacity	32,100	15,200	47,300

The summer design outdoor air distorts the coil load because it brings in 30% more latent heat than sensible heat. Thus, a unit that satisfies the sensible and total coil loads fails to satisfy the latent load, leading to the result shown on Figure 7 with excessive sensible capacity and insufficient latent capacity. It would be possible to select a larger unit – five tons instead of

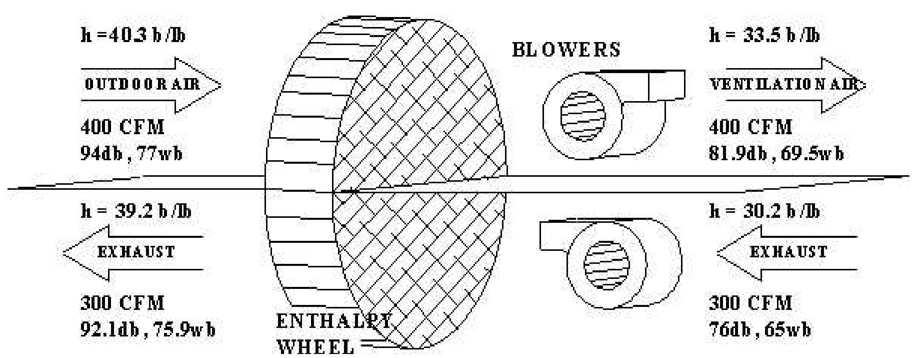


FIGURE 8 – Energy Recovery Ventilator
cross flow neglected, casing and filters not shown

four – and then use reheat to compensate for the excess sensible capacity. However, this would waste energy compared to either of the pre-treatment options available.

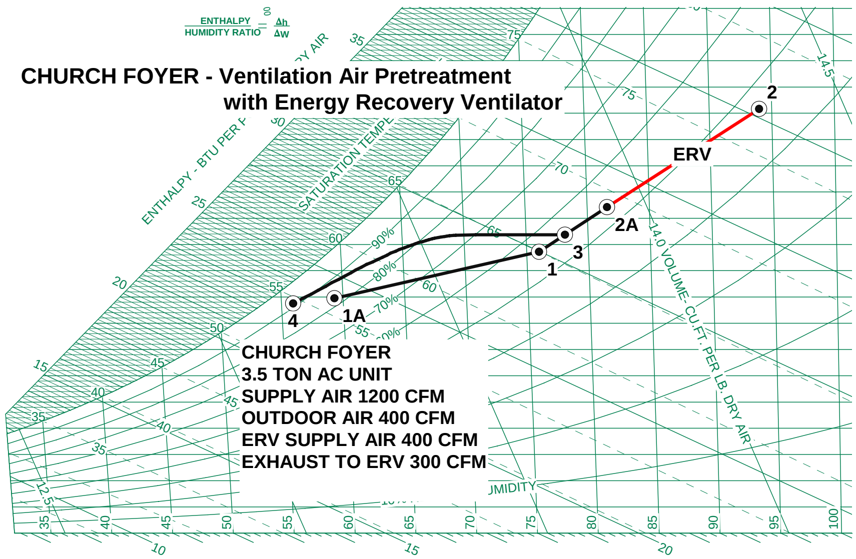


FIGURE 9 – FOYER WITH ERV

Figure 9 shows the same system as Figure 7, but with the ventilation air pre-treated by an energy recovery ventilator. Figure 8 is a schematic of the proposed ERV to be applied. All of the ventilation air and exhaust air from the zone must be processed by the ERV, as shown on figure 8. The schematic shows that the ventilation air when passing through the enthalpy wheel gives up its heat and moisture to the exhaust, which enters the wheel as the cool and dry room air.

The effect of this process is shown by the red process line on Figure 9. Thus, point **2A** becomes the ventilation air condition, shifting the mixed air condition, point **3** to a cooler and dryer condition. Note that the size of the required AC unit is reduced from four to 3.5 tons, and the supply air is reduced from 1200 to 1050 cfm.

It should be noted that the coil sensible capacity shown on Figure 9 remains excessive. In this case, a smaller three ton unit matches the sensible load more closely, but fails again to have adequate latent capacity. The solution to this would be to apply a single-row heat pipe with a modest 3˚ Δt. Heat pipes will be discussed in detail further along in the course, and the student is invited to return to this section and apply a heat pipe to the case shown on Figure 9.

Dedicated Outdoor Air Unit

There are several reasons that energy recovery ventilators may not be usable for a particular project. First, the units are physically large – often larger than the AC unit they serve – and may not fit in the spaces available. Second, all of the exhaust from a zone must be collected and ducted to the ERV. This may not always be practical or possible. Finally, the ERV cannot be used if the exhaust is contaminated, as is the case with kitchen grease hoods and many medical, industrial and laboratory exhausts, all of which present problems of very high volumes of outdoor air.

Small systems design will most often encounter problems with small restaurants having a limited number of seats. Other cases are small buildings or zones having cosmetic, fingerprint or weapons cleaning stations that require side slot hoods with high air flows, but otherwise have little sensible load. For lab and industrial hood design, see the publication Industrial Ventilation – A Manual of Recommended Practice by the American Conference of Governmental Industrial Hygienists. Designs for commercial kitchen hoods are readily available through the hood manufacturers' factory representatives.

Figure 10 shows the initial psychrometric analysis of a small restaurant with a six foot medium load compensating commercial hood. "Compensating" means that part of the required hood exhaust is made up by outdoor air introduced into the kitchen through the hood system. Hoods smaller than 7' rarely can compensate for more than 70% of the exhaust, as is the case for this example. In Figure 10, it is

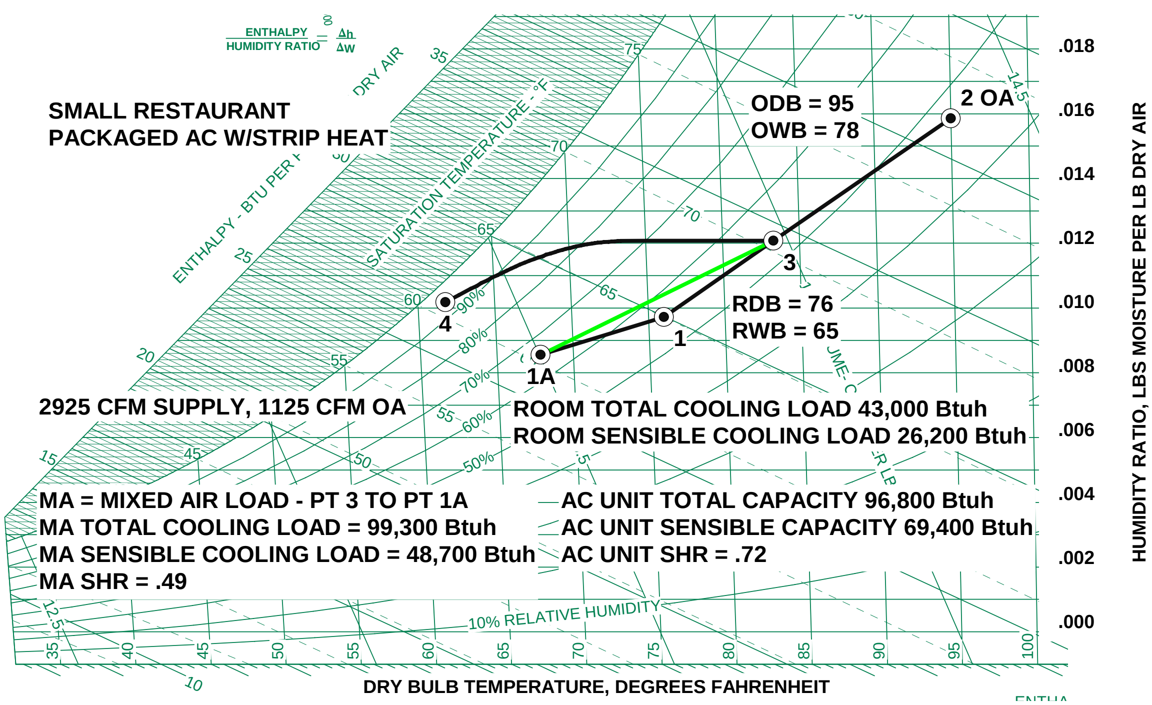


FIGURE 10 – Small Restaurant Initial Selection

assumed that the make-up air is 25% greater than the exhaust, and is introduced through the dining area ac unit to ensure air flow from the dining area into the kitchen. In this case the outdoor air is 40% of the total supply air. (Restaurant ventilation is covered in detail in PDH course #M384, **HVAC Ventilation for Indoor Air Quality.**)

The system depicted on Figure 10 has been selected to satisfy the mixed air total and sensible loads. However, to satisfy the total load, it was necessary to select a system with excessive sensible capacity. The green line from point **3** to point **1A** is the process line for the coil design load. It is clear from Figure 10 that the selected system cannot ever satisfy the room design conditions, even with massive reheat, because point **4** has a higher dew point than point **1A**, which is the maximum dew point that will satisfy the room latent load. Of course, an even larger unit could be tried, but then it would significantly exceed the mixed air sensible load, and massive reheat would be required.

The solution to this situation is to pre-treat the outdoor air before it is introduced into the system. The two most practical ways to do this are with an energy recovery wheel or a dedicated (100%) outdoor air unit. As noted before, the ERV option is never available with a commercial kitchen because the exhaust is contaminated and must be freely exhausted.

The second option is to cool and dehumidify the outdoor air with a separate air conditioner capable of handling 100% outdoor air. This solution is depicted in Figure 11. In general, a 100% outdoor air unit is selected to deliver a dew point to the room or to the zone cooling coil inlet that is lower than the dew point needed to satisfy the room design conditions. In other words, a dew point lower than point **1A** at the room supply air flow rate. In addition, since such units usually include hot gas reheat, the outdoor air can be delivered at a dry bulb temperature near or only slightly lower than the room design dry bulb temperature, thus eliminating the outdoor air load on the space AC cooling coil. In the case shown, however, the outdoor air requirement is so dominant that the entire load is best matched by the 100% outdoor air unit with no return air. The hot gas reheat would be regulated to maintain the room dry bulb set point. Electric or gas heat would be used during cold weather.

Large restaurants will generally require only 25% to 30% outdoor air, and so will

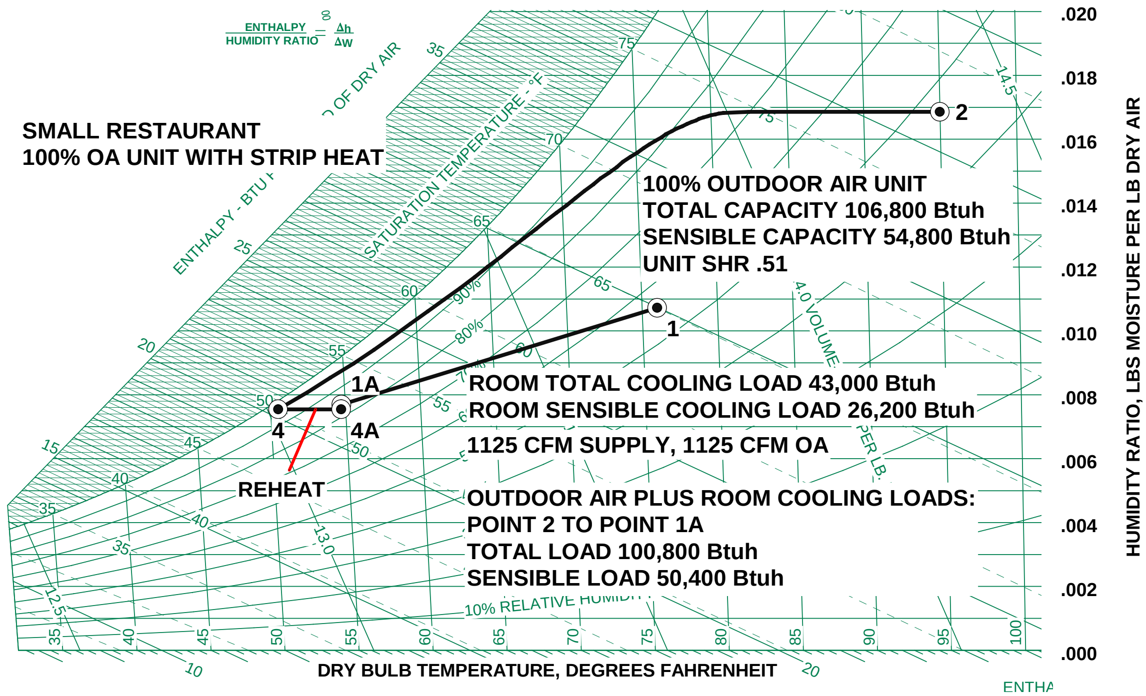


FIGURE 11 – Small Restaurant with 100% Outdoor Air

use systems sized only for the room loads, and will either mix the pre-treated air with the return air from the rooms, or will introduce the pre-treated air directly into the occupied space at the space design state point. Restaurant solutions will usually require the designer to iterate several options to find the optimum match.

The selection and application of outdoor air pretreatment systems is covered in detail in PDH course M384, **HVAC Ventilation for Indoor Air Quality.**

Heat Pipes

Reheat is generally available as a factory option in light commercial packaged units, and is a simple, cost effective solution for conditions of high occupant density or even moderately high outdoor air percentage. However, light commercial split systems under 25 tons generally are not available with hot gas reheat. Other types of reheat are either prohibited by many codes as is the case with electric reheat, or present special maintenance, safety, and installation problems as is the case with natural gas re-heaters located in the supply duct downstream of the air handler. A frequently viable and cost-effective solution may be a heat pipe. A heat pipe is a sealed refrigerant coil that wraps around the cooling coil and transfers sensible heat from the entering stream to the coil leaving stream. See Figure 2, page 7. Because of the characteristics of unitary

DX cooling systems, heat pipes can be extremely effective in reducing the sensible heat ratio of the mixed air process line.

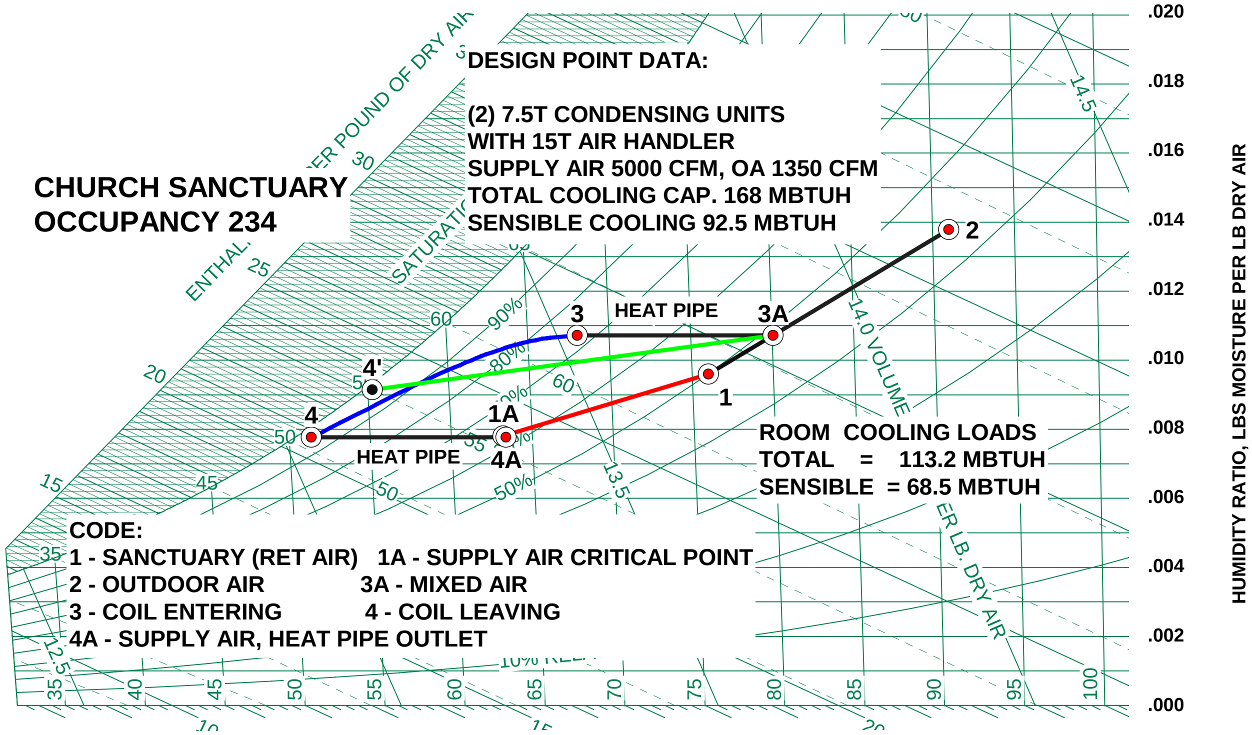


FIGURE 12 – Assembly Occupancy – Heat Pipe

Figure 12 is a chart of the cooling system design point for a church sanctuary served by a 15 ton DX air handler on two 7.5 ton condensing units. Note that the sanctuary (room) process line, **1–1A**, is very steep, and does not intersect the saturation line. The steep slope is caused by the latent load of 234 occupants. Also, the temperature at point **1A**, which is the maximum temperature that will achieve the room design point, is 63˚. There is no DX air conditioning system that can follow this process line, and if there were, the coil leaving temperature of 63˚ would be too high for effective dehumidification.

The solution selected here is a heat pipe, which allows both a low coil leaving temperature (point **4**) and a supply air temperature and dew point near the ideal (point **4A**). A system with hot gas reheat was also considered, but would have required the next larger size unit – 20 tons – and would result in longer run times for dehumidification. Also, reheat is not available as a factory option on split air cooled systems, making it more difficult to include in the system selection.

As shown on Figure 2 (page 7), the heat pipe coils wrap around the evaporator coil of the air conditioning unit. Because refrigerant in an unforced system will flow to the coldest place in the system, refrigerant in the heat pipe flows continuously from the warm entering coil ahead of the evaporator to the cool coil at the exit of the evaporator. This transfers sensible heat from the coil entering air, cooling it, to the coil leaving air, warming it. The designer sets the required delta t (Δt) of the coil by trial and error. The evaporator dry bulb entering temperature ($\mathbf{tdb_3}$) is the mixed temperature at point **3A** minus the heat pipe Δt. Likewise, the heat pipe leaving temperature at point **4A** is the evaporator leaving temperature ($\mathbf{tdb_4}$) plus the heat pipe Δt. In each case, the Δt is applied at constant dew point temperature. The cooling coil total and sensible cooling capacity is taken at the mixed air flow and the state conditions at point **3**.

In the example of Figure 12, the heat pipe Δt is 10˚F, so $\mathbf{tdb_3}$ is about 69˚ at the mixed flow (point **3A**) dew point of about 62˚. Most manufacturers do not publish performance data at coil inlet temperatures below 75˚F, so the designer must obtain them working with the manufacturer's representative.

The green process line shown on Figure 12 represents the unit cooling coil capacity with no heat pipe. Although the coil in this case matches the total load, it has greatly excessive sensible heat capacity, and has a leaving dew point that is too high to satisfy the room humidity design point (insufficient latent capacity) . The heat pipe reduces the sensible heat capacity by pre-cooling the mixed air (point **3A** to **3**). The coil leaving conditions are then brought back to match the room required supply air conditions by reheat. The heat pipe accomplishes this by transferring sensible heat from the warm coil entering air to the coil leaving air. There is no penalty for this, except for somewhat increased pressure drop through the cooling unit. Heat pipe manufacturers call this a "dehumidifying" heat pipe, because as is evident from figure 12, the air passing through the coil is much colder than it would be without the heat pipe.

Heat pipes are not available as standard equipment from ac manufacturers, but must be fitted to the ac unit after purchase, usually at the heat pipe manufacturer's facility. For small commercial applications, heat pipes are usually applied only to split systems. This is because most light commercial packaged units are available with liquid line and/or hot gas reheat, which is less costly than heat pipe installation.

Heat pipe selection and specifications are covered in manuals available from heat pipe manufacturers, such as Heat Pipe Technology, Inc. of Gainesville, Florida.

However, designers are cautioned to work closely with the ac equipment manufacturers' factory representatives and with the heat pipe manufacturers and installers to be certain that the specified heat pipe parameters can be applied to the specified AC equipment.

7. Energy Notes

This course has presented four methods of controlling moisture when psychrometric analysis indicates that the air conditioning system alone will be inadequate. These are reheat, energy recovery ventilator, dedicated outdoor air (DOA) unit, and heat pipe.

In general, as will be discussed below, the energy rank, from lowest to highest energy use, is:

heat pipe.
energy recovery ventilator (ERV)
dedicated outdoor air unit (DOA)
hot gas reheat
reheat with fossil fuel
electric reheat

Reheat vs Heat Pipe

In the case of high occupant density, as shown on Figures 6 (page 17) and 12 (page 24), reheat or a heat pipe are the logical alternatives. Since the ERV and DOA pre-treat ventilation air, they are ineffective when the moisture source is the occupants.

It is obvious that reheat will use more energy than the heat pipe, because the unit must be larger to handle the latent load. However, reheat actively controls humidity in the space, and so may be more effective with applications having highly variable load than the passive control of the heat pipe, which is only effective when the system is responding to heat gain and the compressor is running. This deficiency can be mitigated by specifying an AC unit with capacity reduction capability using unloading or multiple compressors.

High Ventilation Air Flow

Figures 7 (page 19) and 10 (page 21) show the effect of high ventilation air flow. It can be seen that both of these situations can be solved by simply using a larger AC unit with reheat. Although hot gas reheat is "free", what is not free is the extra run time on the AC unit when responding to space humidity while temperature is satisfied. As noted, the reheat approach for the system of Figure 7 would require a 5 ton unit with reheat, while the ERV shown on figure 9 requires only a 3.5 ton unit to satisfy the design load. While the sensible capacity of the unit represented on Figure 9 is still marginally excessive, this could be solved with a modest heat pipe, as previously noted.

The problem depicted by Figure 10 is similar to that of Figure 7, except that an ERV cannot be used because the exhaust is contaminated. As before, it would be possible to increase the size of the AC unit to about 12 tons with reheat to solve the problem. However the solution shown on Figure 8, which is a dedicated outdoor air unit, requires only eight tons capacity and much less reheat. The more reheat that is required, the more the unit will run solely to satisfy room humidity.

Both the ERV and the DOA unit will control humidity caused by high outdoor air under variable load conditions. However, the DOA is a truly active humidity control method, where the ERV will eventually allow room moisture to rise if the compressor off times become very large.

8. Rules

Based on the psychrometrics of the room and coil cooling loads, the engineer can apply the following rules to anticipate the type and complexity of the equipment that will be required.

1. The equipment selected must be able to maintain the conditioned space relative humidity below 70% at all times, and below 60% most of the time.

2. If outdoor air is less than 20% of supply air, and occupant density is less than about 7 persons per 1000 sf, then both room and coil sensible heat ratios will be in the range .65 to .8, which small commercial and residential DX systems can accommodate.

3. If outdoor air is more than 20% of supply air, then coil sensible heat ratio may be less than .65, even as the room ratio remains at .7 or more. In this case, pretreatment of the outdoor air, either by energy recovery or 100% outdoor air unit, may be required.

4. If occupant density is more than about 10 persons per 1000 sf, then both room and coil sensible heat ratio may be less than .65, with coil sensible heat ratio being the lower of the two. In this case, pretreatment may be needed to handle the latent load of the outdoor air and in addition, reheat or heat pipe may be needed to handle the latent load due to occupants. Often, reheat or heat pipe can handle both the outdoor air and the occupant latent loads.

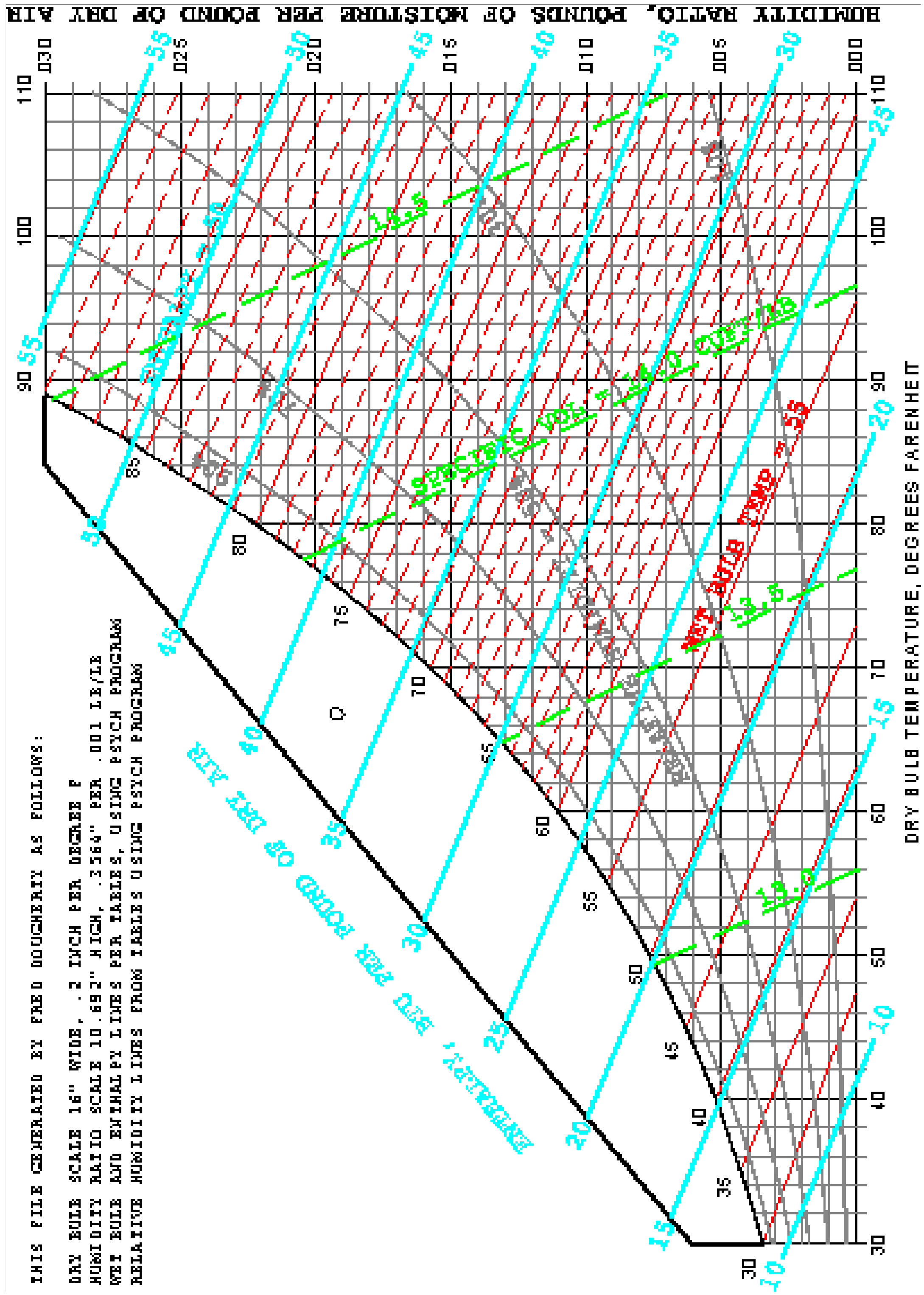
9. Summary

HVAC design is a step-by-step process to find an air conditioning system that will match the calculated cooling coil sensible and total cooling loads. However in addition, the temperature and dew point of the coil leaving air (supply air) must both be below the required maximum to deliver the design temperature and humidity to the occupied zone. Psychrometric analysis is required to determine this maximum, and to compare that to the state of the air leaving the cooling coil.

When the system is ready for analysis, the known parameters will be the state points (dry bulb temperature, wet bulb temperature, enthalpy, etc) of the desired room air and of the outdoor ventilation air. Also known will be the calculated room and cooling coil sensible and total design cooling loads. From these parameters, the mixed air condition entering the coil, the room process line, and the coil process line can be calculated and plotted. The room process line represents the supply air as it enters the room and picks up heat and moisture. The mixed air condition is the state point of the outdoor air mixed with the return air either directly from the room or after picking up heat in a return air plenum. This will be the entering condition to the cooling coil. The cooling coil process line represents the supply air as it passes through the coil and gives up its heat and moisture.

For systems with moderate occupancy and ventilation air, a properly selected air conditioning system will deliver supply air to the room at a temperature and dew point that can satisfy the room design conditions. However, High occupancy or air flow can result in a required maximum supply air temperature and/or dew point that an ordinary cooling coil cannot match. This will be evident with psychrometric analysis, and the designer can use an appropriate strategy to shift the supply air state point as needed to match the room load. This could be reheat, when the coil leaving dew point is acceptable but the leaving temperature is too low, which would cause short run times and moisture build-up. Other options are outdoor air pre-treatment or a heat pipe. These can be plotted to find a solution. In rare cases, it may be necessary to select a different cooling system and begin the iteration process over again.

END


HUMIDITY RATIO, POUNDS OF MOISTURE PER POUND OF DRY AIR
DRY BULB TEMPERATURE, DEGREES FARENHEIT
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WET BULB TEMP = 55
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WET BULB AND ENTHALPY LINES PER TABLES, USING PSYCH PROGRAM
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