A SunCam online continuing education course

HVAC Layout and Design:
Course 2 of 4, Occupant Comfort & Load/Economic Analysis

by

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To properly layout HVAC systems, one must have a full understanding of occupant comfort fundamentals. The first part of Course 2 will include comfort factors and design fundamentals that the HVAC design engineer must keep in mind while layout out systems, locating air devices and equipment. This will lead into a discussion around psychrometric analysis using the psychrometric chart. We will then spend time discussing the psychrometric chart and how it can be used to estimate equipment loads based on ambient conditions, interior conditions and sensible heat ratios.

To further define each zone heating and cooling load requirements, modeling programs require the user to provide input data. I will share my experiences using the Carrier, Hourly Analysis Program and other DOE 2.2/2.3 approved load modeling software packages. I will also briefly discuss the benefit of CFD modeling and provide examples of projects I have been involved with.

The final section of this Course will focus on comparisons between different HVAC equipment and system selections. For preliminary design, this is generally accomplished using economic analysis tools and may be completed using NEUI figures. I will share insights regarding Net Present Value of projects given a building life cycle.

COMFORT FACTORS/PSYCHROMETRICS

As HVAC design engineers, it is our goal to design HVAC equipment and systems that keep occupants comfortable. Many studies have shown that occupant (worker) productivity can be significantly reduced (up to 40%) if they are experiencing discomfort. Lost time at work can result in tremendous costs for Owners as overhead costs represent the largest operating cost for most employers. There is a well-known rule of thumb known as the 3-30-300 rule. The average costs for commercial building utilities is ~$3/s.f., with $30/s.f. in rent (or mortgage) and $300/s.f. in overhead (payroll) per year. Using this rule of thumb, if we have a 35,000 s.f. building, that would represent ~$100K in utility costs, ~$1Mil in rent/mortgage and ~$10Mil in payroll each year. If 15% of the employees are uncomfortable and their productivity decreases by 40%, this represents a cost of $600,000 annually.

Not all of this cost is directly related to the HVAC system design; however, the HVAC design can play a major role. I have seen anywhere from 10% to 25% of productivity can be directly affected by occupant environmental thermal comfort, the perception of noise in the HVAC system, insufficient ventilation and/or inadequate filtration (air quality). Air quality may also contribute to chronic illness. These examples, along with lighting levels, work-station ergonomics, ambient noise and worker ambient activity make up the overall Indoor Environmental Quality (IEQ), and all factors have been shown to directly affect worker productivity.

So, what do employers do with sick employees or employees suffering from "presenteeism"? That productivity loss has to be made up somehow so the company may need to hire more employees, offer tele-commuting options, incentivize employees to work while uncomfortable or simply suffer the productivity reduction. The first part of this section will address several of the IEQ factors related to environmental thermal comfort. We will then discuss ventilation, filtration and removal of contaminants in the workspace.

There are four (4) primary environmental thermal comfort factors that affect occupants of buildings as there is a relationship between the occupant and the space conditions. Environmental comfort factors affect our body's ability to properly regulate our own temperature and comfort. From an occupant perspective, it is all about regulating body heat. These thermal comfort factors are (in order of importance):

1. Conduction/Convection/Radiation
2. Air Movement
3. Humidity
4. Temperature

The typical office worker, seated at a desk and working, generates both sensible and latent heat. The sensible heat, dry bulb temperature change that can be felt, is dissipated through radiation and this dissipation is time delayed based on the clothing the occupant is wearing. Human beings give off a lot of moisture and this amount can vary based on activity level. An average office worker could lose over 1 cup of water per 8-hour workday through perspiration and respiration. This does not include walking around the office, lifting objects, sitting through uncomfortable meetings, etc. Add these factors and an average worker could be closer to 1.5 cups of water per 8-hour workday. The moisture is released through skin or exhalation and evaporates into water vapor, the rate of which is governed by the partial vapor pressure of the occupant environment. The amount of energy given off during evaporation is referred to as the latent heat and affects the wet bulb temperature only.

The sensible heat and latent heat make up the total heat given off by occupants and objects within a given space. The activity level, clothing and person's natural metabolism all affect how much total heat a person gives off. Equipment heat losses are driven by power input and equipment efficiency. For this Course, the amount of energy it takes to reduce dry bulb temperature of an airstream is referred to as sensible load. The amount of energy it takes to condense water vapor back out of the airstream represents the latent load. Together, the sensible and latent load make up the total load.

Examples of occupant heat dissipation values include:

1. Seated at desk, office worker typing on the computer: ~250 Btu of sensible heat and ~225 Btu of latent heat per hour
2. Medium work, like a grocery store employee carrying around boxes and unloading them in the aisles: ~300 Btu of sensible heat and ~450 Btu of latent heat per hour
3. Heavy work, like construction worker performing renovation work inside a conditioned building: ~525 Btu sensible heat and ~925 Btu of latent heat per hour
4. Athlete, similar to a basketball player in a school gymnasium: 710 Btu sensible and 1,090 Btu latent heat per hour

The next few sections will discuss how the environment interacts with occupant's ability to give off latent and sensible heat.

Conduction/Convection Radiation occurs when heat transfers through a substance (conduction through a fixed object) or a fluid (convection, creating movement of air or water) or heat energy is radiated from a solid object as a wavelength (Figure 1).

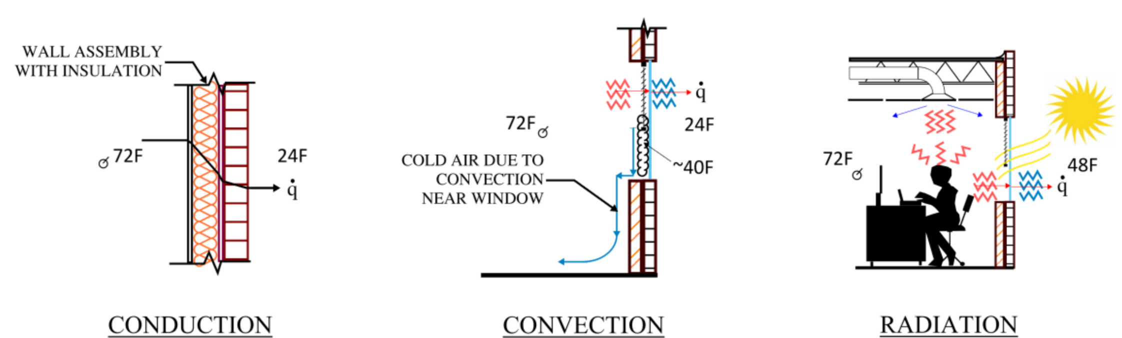


FIGURE 1: EXAMPLE OF CONDUCTION, CONVECTION AND RADIATION

Heat energy will always dissipate to colder surroundings as the molecules attempt to decrease their kinetic energy. Hot surfaces, cold surfaces and sunlight can cause discomfort as heat leaves and comes into our bodies at the same time. Our bodies are trying to regulate temperature and are affected by the environmental condition. The closer we are to a cold surface; the faster heat leaves our bodies. The closer we are to a warm surface; the faster heat joins our bodies. The greater the temperature difference, the faster the rate of gain or loss of energy. All of these factors may cause discomfort.

Conduction results in energy gains and losses through building wall assemblies, mechanical piping, ductwork, etc. The rate of heat transfer through conduction is driven by temperature difference the thermal conductivity of the object itself (e.g., wall assembly). The rate of heat transfer may be minimized with insulation. The Code defines minimum insulation R-Values required for building walls, roofs, slabs and ductwork systems. The R-Value is used to describe a rated performance. In imperial units and SI, R-Value is expressed as:

IP: $(hr\ x\ ft^2\ x\ °\mathrm{F})/Btu$ SI: $m^2\ x\ °\mathrm{C}/W$

Recommended thickness of pipe insulation is given based on the insulation K-Value (or thermal performance). In imperial units and SI, K-Value is expressed as:

IP: $Btu\ x\ in(hr\ x\ ft^2\ x\ °\mathrm{F})$ SI: $W/m\ x\ °\mathrm{C}$

Like R-Value, K-Value is rated based on a mean temperature of the insulation itself which is affected by interior and exterior conditions, wind speed, exposed radiation and thermal conductivity properties of the insulative material itself. To save energy, Architects and Owners may choose to exceed minimum R-Values prescribed in the Code, so the HVAC engineer must be aware of all building materials, wall assemblies, R-Values for materials, etc.

Convection may occur adjacent to cold or warm objects that are experiencing conduction. A cold window, for example, may develop a layer of cold air on the inside surface. If the layer exists in a warm environment and becomes heavy enough, it will begin to fall, forming convective currents of air. Surrounding air is induced into the convective currents forming a cool draft. This may also occur above warm objects as they are heated up in a cold environment. As the heated air rises, ambient air is affected. In practice we use finned tube radiation under large areas of glass to counteract cold drafts. In addition, HVAC design engineers use slotted diffusers above large areas of glass to wash the window with air. This can prevent cold drafts, condensation forming on the window surface during the winter months and warm plumes of air forming when it is warm outside.

Radiant energy can significantly affect comfort levels. Think of sitting outside in the late fall when ambient temperatures are in the low to mid 50's. If the sun is shining and the wind is not blowing, you will likely remove your jacket. Warehouses served by radiant heaters with $\leq$ 1 ACH are typically maintained at 60F setpoint and workers are quite comfortable when radiant heaters are in operation. The floor, walls and internal objects all reach equilibrium temperature and radiate heat as well. When doors are opened for short periods of time, the space quickly recovers.

Air Movement (a.k.a. airflow) can be measured in feet per minute (fpm) velocity. Air movement helps our bodies to regulate temperature. There is no minimum airflow requirement for comfort; however, too much airflow can feel uncomfortable as our bodies struggle to regulate temperature through evaporation. In the winter months, 35 fpm air velocity is the maximum recommended. In the summer, 50 fpm velocity is the maximum recommended. So, what does 50 fpm "feel" like anyway? It depends on the temperature and humidity of the air. For comfort cooling in commercial office buildings, I tell people that 50 fpm feels like a very light breeze on the back of your neck as it is hardly perceptible. 100 fpm on bare skin is perceptible and feels like a calm breeze. 250 fpm is similar to the air disturbance felt when walking briskly passed someone in the hallway. This feels drafty on bare skin.

It is important for the HVAC design engineer to design systems, select air devices/grilles and locate them to provide proper mixing of air in each zone for both heating and cooling modes of operation. In winter months, the humidity of the air is typically low, thereby allowing for more efficient occupant cooling due to the evaporative effect. A velocity of 50 fpm in the winter can be easily perceived and make the room feel drafty. Dumping 55F air on someone's neck at 50 fpm in the summer months is very uncomfortable and will also lead to complaints. Air devices and grilles must be carefully selected to meet the Air Diffusion Performance Index (ADPI) performance factor that keeps everyone in the space comfortable. ADPI is the percentage of points within the occupied zone having a range of effective draft temperatures of -3F to +2F of the average room temperature at a coincident velocity less than 70 fpm. More of this topic will be discussed in Course 3.

If occupants are struggling to maintain a comfortable temperature, increased airflow within conditioned environments may be desired to help increases the evaporative effect. Large spaces may use ceiling fans, destratification fans or HVLS fans to promote increased airflow. In an office environment, occupants may bring in portable, personal fans to increase airflow in their space. It is common to find large, free standing fans in non-conditioned environments such as warehouses and industrial facilities. One must be aware that with high ambient temperatures and very low humidity, increased air movement may actually put stress on the occupant as the human body may stop sweating to conserve moisture. If this occurs, increased air movement has the opposite effect and can warm the occupant.

Humidity levels, or moisture levels within an occupied space also affect comfort. Humidity refers to humidity ratio (W) or the amount of moisture actually present in the air. This is not to be confused with relative humidity (RH). Humidity ratio is the mass of water vapor in a given mass of dry air [grains/lb_{da}] and is also referred to as Specific Humidity. RH is the amount of moisture as compared to the maximum amount of moisture that could exist in a given pound of moist air at the same dry bulb temperature. The same amount of moisture [grains/lb_{da}] could exist in air at different temperatures. With the same amount of moisture at lower temperatures, the RH would be higher than at higher temperatures since warm air has a capacity to hold more moisture as a vapor. As the amount of moisture in a given air mass increases, the dew point temperature also increases. Dew point is defined as the temperature at which saturation occurs. So, if you take a cold jar of pickles (~40F) out of the refrigerator, moisture will begin to condense on the outside of the jar. This occurs because the pickle jar temperature is below the ambient dew point temperature.

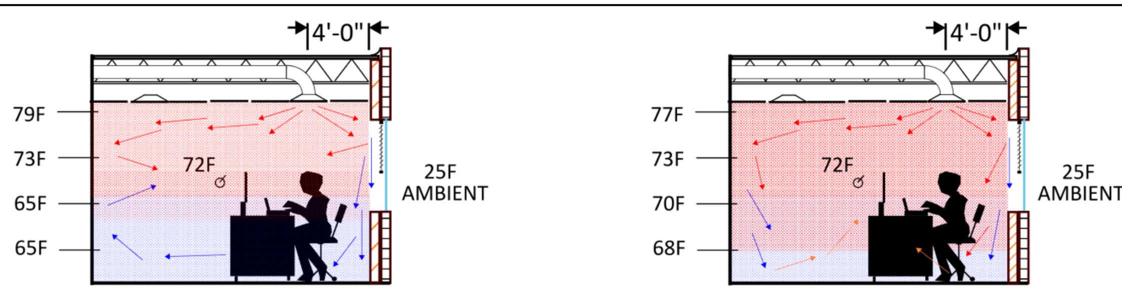
These are important concepts to understand since the human body regulates its temperature by evaporation. The warmer and dryer the ambient air is, the more the human body can evaporate which results in more of a cooling effect. Let us look at an example. If the temperature set point is maintained at 75F and a space has 50% RH, this space would have a humidity ratio (W) of 64.9 grains/lb_{da} and a dew point temperature of $55F_{DP}$. If the dry-bulb thermostat set point is changed to 65F, and no moisture is removed, the space will have 70% RH with dew point of 55F. As will be discussed later in this Course, some amount of moisture will be removed by the HVAC equipment during cooling, but in Commercial buildings the RH will eventually be elevated (likely >65% RH). What does increase in RH do to the human body? It becomes more difficult to evaporate moisture into an environment that has less capacity to hold moisture, so moisture begins to accumulate on the skin. Over time, the occupant begins to feel clammy. The space begins to feel "close" or uncomfortable.

Given the above, ASHRAE 55 has defined both a winter and summer design comfort zones which will be discussed later in this Course. This comfort zone has min/max dry bulb temperatures and min/max dew point temperatures that we as HVAC design engineers should stay within. The minimum dew point temperature is $35F_{DP}$ and maximum is 62_{DP}. Given the dry bulb boundaries of these comfort zones ($67F_{DB}$ to $81F_{DB}$), we end up with RH values of roughly 25% to 60%.

Engineers must be careful designing systems that allow for high dew point temperatures (high amounts of moisture). Mold may develop in areas of the building where moisture can accumulate. This is particularly important in schools and libraries that have books. Mold may also grow inside of walls with failed/damaged moisture barriers as these areas may have more moisture. It is typical for engineers to design systems with maximum 50% RH at design temperatures to avoid these risks.

Temperature is the final thermal comfort factor that affects occupant comfort. The rate of temperature change from set point should not exceed $4F_{DB}$/hr. as occupants will notice the change in temperature more than the temperature itself. Typical HVAC design maintains a +/- $3F_{DB}$ dry bulb temperature swing for cooling and heating applications with +/- 5% RH differential. Pharmaceutical projects, OR's, Data Centers and specialty Labs may require tighter controls such as +/- 1F dry bulb temperature swing and +/- 3% RH. These applications require specialized DX equipment, chilled water, humidification and/or robust controls.

When designing occupied spaces, the Course mentioned selecting air diffusers to meet minimum ADPI performance factors. This is important to prevent the formation of thermal stratification barriers and maintain proper mixing of air within a space. Some level of thermal stratification is typical with HVAC comfort heating and cooling designs. The overall temperature difference from floor to ceiling should not exceed $5F_{DB}$. In cases where thermal barriers exist, temperatures can get $\leq$ 65F at the floor and $\geq$79F at the ceiling creating cold ankles and complaints (Figure 2).



THERMAL STRATIFICATION

PROPER MIXING

FIGURE 2: THERMAL STRATIFICATION AND PROPER MIXING EXAMPLES WITH COOL AMBIENT TEMPERATURES

At 150 fpm, the air device or grille supply air is independent of space temperature. At 100 fpm, the buoyancy of air can overtake supply air being delivered from an air device or grille. At

$\leq$ 75 fpm, warm air buoyancy will cause the air in a space to rise, and the warmer air will hang tight to the ceiling. Heating SA temperatures should not exceed 15F of the occupied design set point temperature as the warm air will tend to "cling" to the ceiling and design velocities should be maintain above 100 fpm to prevent thermal stratification.

Sound Transmission (or acoustical transmission) is the transmission of sound power whereas sound pressure is the receipt of sound power wavelengths. Sound pressure exists over eight frequencies and is experienced by the occupant when they perceive noise in their ear drums. The average person perceives 5-6 of these frequencies and finds the higher frequencies more of an annoyance. This sound pressure is measured in decibels (dB's) and dB's will decrease as a person gets farther away from the sound pressure source. The perceived noise is almost double for every 10 dB increase.

When evaluating equipment noise, sound pressure is measured 5' from the noise generating equipment and a resultant A-weighted dBA is calculated by limiting sound pressure in the lower, inaudible frequencies. Sound pressure can be cumulative and damage the human ear drum over lengthy exposures. Sound power waves may also bounce around closed spaces, further increasing the resultant dBA in a given space. In addition, sound power waves may be absorbed by other equipment, wall coverings and/or insulation.

When evaluating spaces, a dBA meter must be used to measure the actual accumulated sound pressure at various points within the space. OSHA sets a limit of 90 dBA for 8-hour exposure. In most cases, mechanical room equipment is designed for a maximum of 85 dBA to account for accumulation of noises. Some examples of dBA include: 30 dBA = Whisper; 50 dBA = Normal Conversation; 80 dBA = Ringing Phone – Occupants must speak loudly to be understood

In the HVAC industry, sound power radiated from air devices and grilles is rated using Noise Criteria (NC). NC describes the indoor noise perceived in an unoccupied space when airflow passes through an air device or grille and enters the space. In HVAC design, we generally select air devices for maximum NC 25 (30 dBA) with maximum airflow. The rated performance of air devices is typically reduced by 10 NC to account for absorption of sound within the space and will be noted on the performance data sheets.

The acceptable NC varies based on the market sector. For example, busy, public areas and labs generally accept higher NC levels from the air devices and grilles due to background noise (white noise) essentially "masking" the HVAC equipment noise. The air devices for these spaces can be selected for NC 35 (40 dBA). Air devices for restaurants can be selected up to NC 40 (50 dBA). In some circumstances, white noise may be introduced to help mask operation of HVAC systems, so NC selection criteria needs to be coordinated with the design team.

Exhaust and supply fans are rated using the sones, another unit of loudness. When fans are located near occupied spaces, they are generally selected with < 10 sones (~70 dBA). The quieter the building, the lower the sones. For example, library exhaust fans and supply fans may be selected with ~4 sones (60 dBA). The final selection depends on location and proximity to occupants.

Higher frequency sound power generated from RTU, or AHU supply fans can be easily attenuated in the HVAC ductwork. Low frequency sound power requires more work to attenuate and requires introduction of mass. Both high frequency and low frequency sound power tends to "break-out" of the ductwork anytime airflow changes direction. An elbow fitting located near the mechanical room, for example, may experience "break-out" noise due to fan noise and airflow noise within the ductwork. If the elbow fitting were located above occupants, they might hear the noise so mass would need to be introduced between the elbow fitting and the occupants. This mass may consist of dry-wall, batt insulation or a combination of both. For stable HVAC system fan(s), high frequency and low frequency sound power will generally be dissipated with three changes of directions. This may not be the case if the HVAC fan(s) are experiencing surge or stall conditions.

The HVAC primary and terminal equipment may generate mechanical noise or airflow noise (usually caused when airflow is restricted). This noise may be perceived from the equipment itself (radiated), equipment inlet (radiated), equipment outlet (discharge) or may propagate down the ductwork and "break-out" several feet down the system. HVAC design engineers need to be aware of the sound power generated by all equipment and select the equipment within acceptable dBA limits to reduce occupant complaints. Duct mounted sound attenuators may be used; however, these devices generate pressure drop. Pressure drop must be overcome by the system fans which takes more energy.

Careful evaluation must be made, and attenuation applied appropriately to optimize HVAC equipment and system first costs as well as operating costs. I generally design systems that do not have white noise with maximum 30-40 NC when located above occupied spaces. Mechanical rooms and roof mounted equipment may be designed at 50-60 dBA when adjacent to or above occupants and 70-80 dBA when removed from occupants. Course 3 will provide example duct design techniques to mitigate propagation of or generation of sound in ductwork systems. Duct design varies based on the application as each market sector has its own acceptable level of NC.

Air Quality is key to maintaining healthy buildings and adequate IEQ. Building materials used in all market sectors contain volatile organic compounds (VOC's) such as benzene, ethylene glycol, formaldehyde, methylene chloride to name a few. Occupants (people and in some cases pets) also emit large quantities of VOC's and contaminates including CO2, dead skin flakes, bacteria and viruses. In today's market, buildings of all types are being constructed with very few leaks to save energy, control migration of moisture and maintain positive pressurization to the outside world. The contaminants are essentially trapped inside our buildings. There are basically two options to deal with these contaminants including dilution through ventilation with outside air (OA) and filtration. Some contaminants may also be destroyed/reduced using ultraviolet radiation or ionization.

Outside air contains breathable air that occupants require. The Code outlines specific requirements to ensure the OA is suitable for ventilation. The Code also defines amounts of natural or mechanical Ventilation Air based on market sector, use/function, square footage, occupant density, etc. Ventilation effectiveness needs to be considered based on the type of HVAC system being designed. Healthcare projects may require additional ventilation air based on FGI or ASHRAE/ASHE-170 recommendations. These calculated airflows must be compared to ICC Code requirements and the more stringent design criteria applied to the HVAC design. The Pharmaceutical industry requires ventilation air for pressurization and cleanliness. Again, a comparison must be made with the ICC Code requirements and the more stringent design criteria applied.

For VAV systems using central station AHU's or VAV RTU's, the ICC Codes currently allow for reduction of OA percentage as compared to critical zone requirements. The basic premise is that OA requirements for certain spaces may be higher than others. Since the air is circulated and mixed, OA may not be "used" in all spaces where it is delivered. This "unused" OA may be re-distributed to all spaces and OA intake to the primary equipment reduced. A detailed ventilation outdoor intake calculation example is included later in this Course.

Air that is brought in from outside or circulated around the building must be filtered to protect equipment coils, heat exchangers and fans. A Minimum Efficiency Reporting Value of MERV 5 disposable filter is the minimum recommended filter efficiency for intake sections to remove insects, cottonwood and other large, airborne contaminants. Some make-up air equipment comes with washable filters for this purpose. Most commercial buildings now utilize 2” thick MERV 8 filters which used to be referred to as the 30% Efficient filter. Manufacturing techniques have developed MERV 13 filters in 1” thickness. These filters can fit in most Residential and Light Commercial equipment. The industry even offers a combination MERV8/MERV13 pre-filter for some applications.

The healthcare industry requires a MERV13 or MERV14 final filter to serve patient areas (depending on the application) with terminal HEPA filters for OR’s and procedure rooms. The Food industry requires a minimum MERV13 final filtration. The Pharmaceutical industry typically includes a MERV14 final filter with terminal HEPA or ULPA filters serving all Grade spaces. In all cases, the OA is pre-filtered at the primary equipment.

The Code references ASHRAE 62.1, Ventilation for Acceptable Indoor Air Quality. Within ASHRAE 62.1, there is a procedure known as the Indoor Air Quality Procedure (IAQP) which is accepted by many AHJ’s. This calculation allows HVAC design engineers to reduce the amount of OA if the airborne contaminants can be reduced by some other means. Known as air purification, these technologies include the use of activated carbon filtration, potassium permanganate media, electro-static filters, ultraviolet germicidal irradiation (UVGI), non-ozone producing ionization (e.g., Global Plasma Solutions), chemical bed scrubbers, etc. Some of these technologies may need to be used to clean ventilation air being brought into the building in certain municipalities.

Psychrometric analysis requires a basic understanding of the psychrometric chart. Developed in the early 1900’s, this chart shows properties of air at a constant pressure including dry bulb temperature (F_{DB}), wet bulb temperature (F_{WB}), relative humidity (%RH), humidity ratio (grains/lb_{DA}), dewpoint temperature (F_{DP}), specific enthalpy (h) and specific volume ($\dot{v}$). The units for enthalpy (h) is Btu/lb_{AIR} and the units for specific volume (S.V.) is ft^3/lb_{AIR}.

Dry bulb temperature is the dry thermometer temperature that we use to control almost every HVAC system. Dry bulb temperature is not affected by the amount of moisture in the air. Use of dry bulb temperature control is interesting considering it is the fourth comfort factor and yet we control the majority of HVAC systems with it. When HVAC first evolved, dry bulb temperature control was the most economical method and we, as an industry, evolved around it so it is still used to this day.

Wet bulb temperature is the temperature of evaporation of water from a wet wick in air or in an airstream. This temperature is dependent upon the amount of moisture in the air and, as with humans, the more moisture in the air, the higher the wet bulb temperature because evaporation is hindered. Relative humidity was defined previously as the amount of moisture as compared to the maximum amount of moisture that could exist in a given pound of moist air at the same dry bulb temperature. Humidity ratio (W) is the ratio of mass of water vapor to the mass of dry air and can also be expressed as lb_V/lb_{DA}. In the HVAC industry, the moisture content is referred to as grains of moisture and the following equations are used to calculate moisture in air:

$$7{,}000\ Grains\ =\ 1\ lb\ of\ Water$$

$$W\ x\ 7{,}000 = Grains\ of\ Moisture$$

$$W = 0.622\ x\ \left(\frac{p_v}{p_b - p_v}\right)\ [lb_V/lb_{DA}]$$

$p_{b=Barometric\ Pressure}\ [psia]$

$p_{v=Partial\ Vapor\ Pressure}\ [psia]$

Dew point is the surface temperature at which condensation occurs. Enthalpy is the sum of two components of the energy balance equation: Internal Energy + Flow Work Energy [$Btuh/lb_{AIR}$]. And finally, specific volume is the volume per unit mass of dry air [ft^3/lb_{AIR}]. If any of these two data points are known, the remaining data points can be identified using the psychrometric chart. Below is an example of the Trane Psychrometric Chart and we will identify each of these components on the chart (Figure 3).

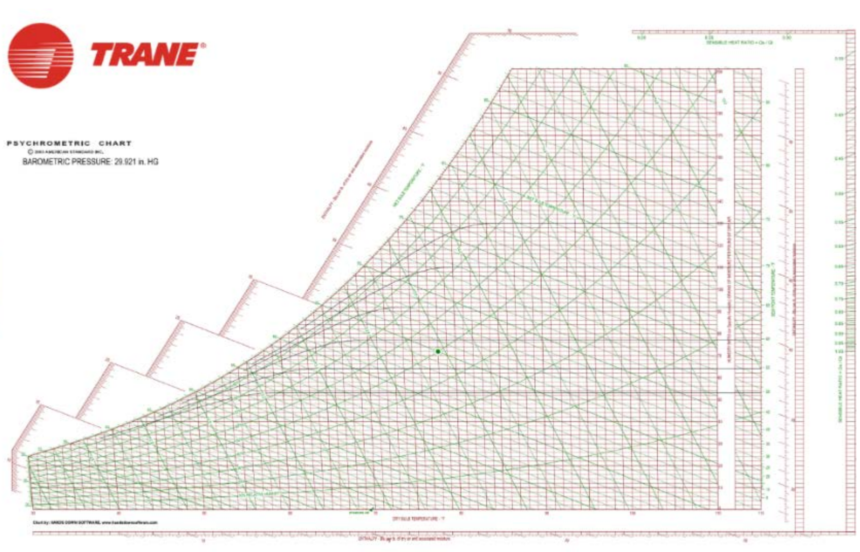


FIGURE 3: PSYCHROMETRIC CHART COURTESY OF THE TRANE COMPANY

Below is an enlarged view of the dry bulb temperatures (F_{DB}) along the bottom of the chart. Note that the vertical lines are not perfectly vertical (Figure 4).

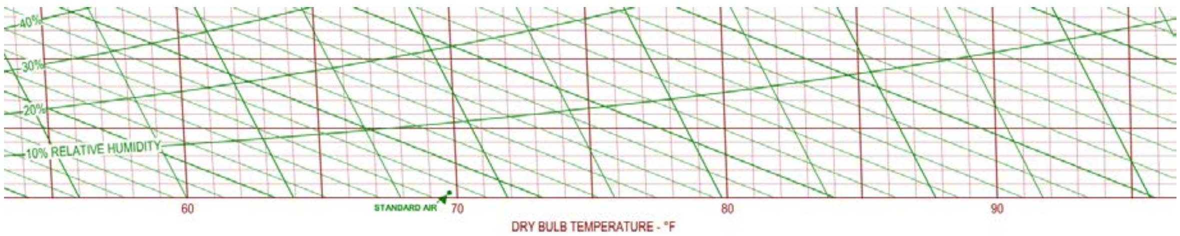


FIGURE 4: PSYCHROMETRIC CHART DRY BULB TEMPERATURES (F_{DB})

Below is an enlarged view of the wet bulb temperatures (F_{WB}) and enthalpy (h). I have circled some of the wet bulb temperatures and drawn a red line to the enthalpy (Figure 5). There is a correlation between wet bulb temperatures and enthalpy. In this case, a $56F_{WB}$ correlates to 23.84 Btuh/lb_{DA} enthalpy. Note that the wet bulb temperature lines intersect with the dry bulb temperature lines. The condition of the air is defined at these points of intersection in F_{DB}, F_{WB} and enthalpy (h). Also note the percentage lines on Figure 5 broken up into 10% increments. These are the %RH lines. And finally, notice the 12.8, 13.0, 13.2, 13.4 numbers and lines. These are the specific volume (S.V.) lines in ft^3/lb_{AIR}. The curved line where the red circles are located is referred to as the Saturation Line. At these points on the chart, air is completely saturated with moisture that is condensing out of the air.

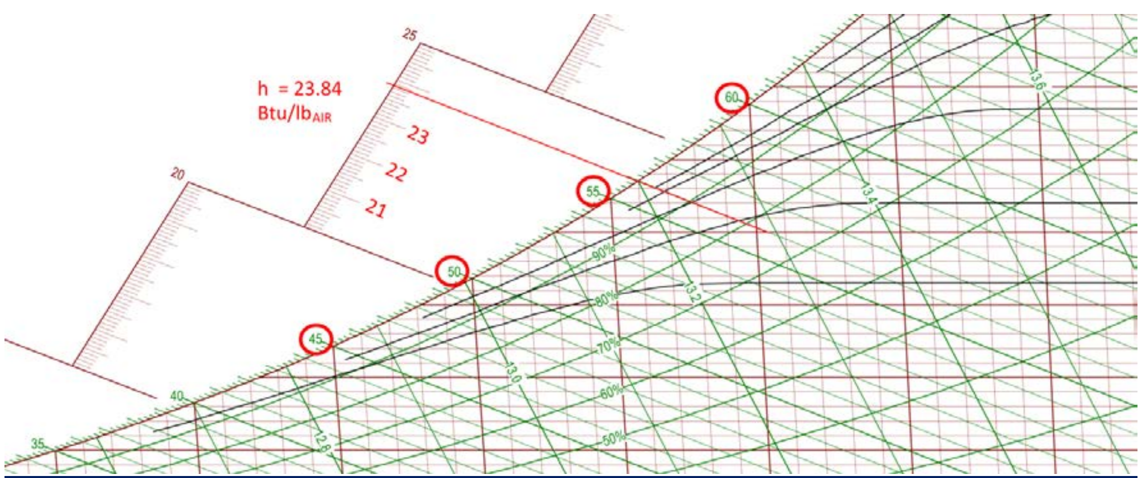


FIGURE 5: PSYCHROMETRIC CHART WET BULB TEMPERATURES (F_{WB})

You may be asking yourself, “how did he get 23.84 Btuh/lb_{AIR} for the enthalpy number?” I happen to own a wet bulb to enthalpy conversion chart published in 1958 that provides these numbers. This was given to me by my original mentor who was ~70 years old when I started in this business in 1995. Needless to say, these concepts have been around a long time, but it is important to fully understand these fundamentals and revisit them from time to time. This is particularly true when dealing with low humidity spaces, coolers, freezers and spaces that must be maintained with very tight HVAC control.

Below is an enlarged view of the Humidity Ratio (grains/lb_{DA}), dewpoint temperature (F_{DP}) and a concept we discussed in Course 1, Sensible Heat Ratio (SHR) along the right side of the chart (Figure 6). The SHR is defined as the Sensible Load divided by the Total Load and will be discussed below.

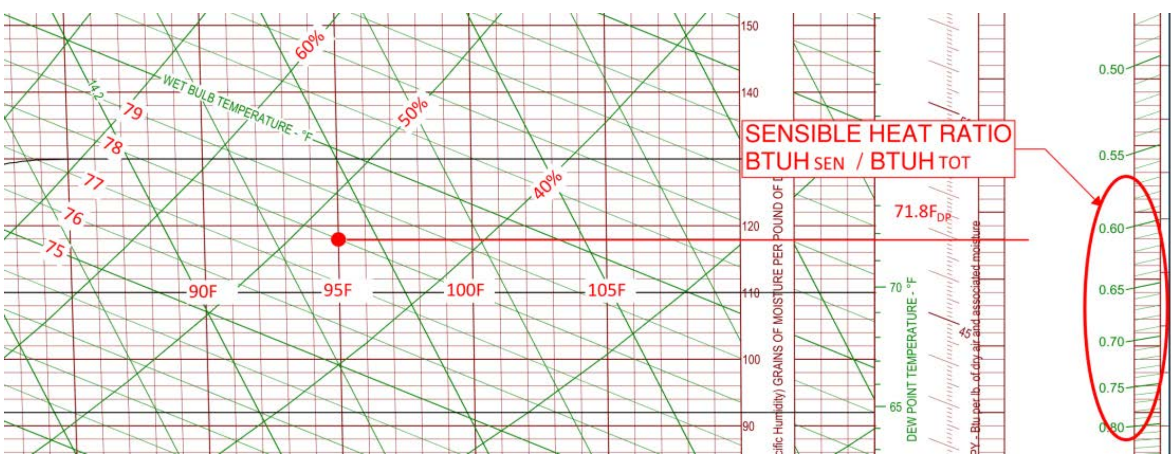


FIGURE 6: PSYCHROMETRIC CHART HUMIDITY RATIO, F AND SHR

I took the liberty of identifying F_{DB}, F_{WB} and %RH values as well as a particular point on the chart (red dot): $95F_{DB}/78F_{WB}$. By drawing a line horizontally, one can graphically determine the moisture content of 118 grains/lb_{DA} and dew point temperature of $71.8F_{DP}$. At this point the RH is ~48%.

In general terms, the condition of air becomes more humid as one moves up on the chart and less humid as one moves down the chart. These latent changes correlate to %RH, F_{WB}, h, grains/lb_{DA} and F_{DP}. From a sensible standpoint, it is cooler to the left side of the chart and warmer to the right (Figure 7).

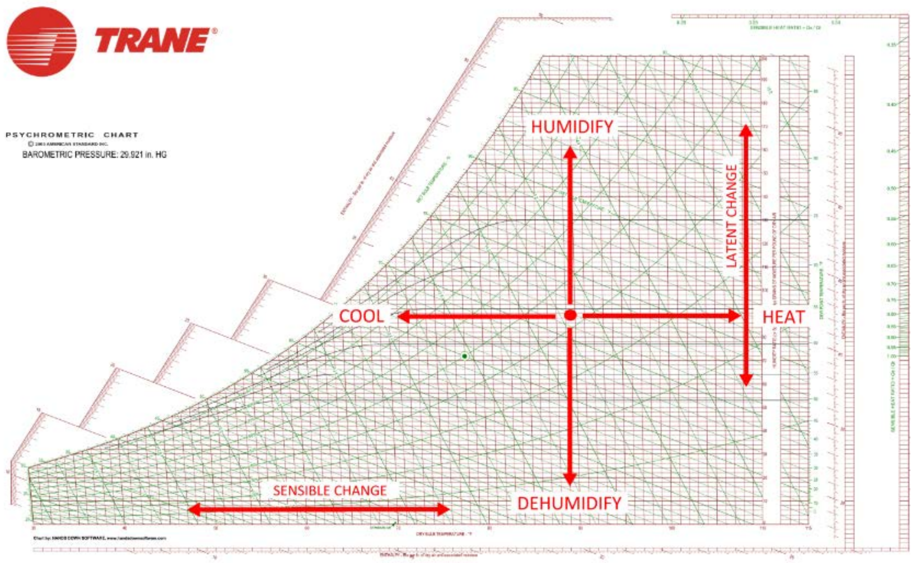


FIGURE 7: MOVEMENT ON THE PSYCHROMETRIC CHART

As one can determine, the far left side of the chart may be "cooler" from a dry-bulb perspective, but the %RH increases as the air becomes more saturated. High %RH and low F_{DB} result in uncomfortable conditions. Fortunately, the typical building generates sufficient sensible reheat to reduce the %RH. These concepts will be further discussed in this Course. For now, this would be a good time to grab your favorite psychrometric chart and become familiar with all of the values it can provide. These are also available on-line for free download.

Before we use the chart to determine equipment loads, let us revisit some equations introduced in Course 1:

Equation (1): $Sensible\ Heat[Btuh] = SA\ Airflow[CFM]\ x\ 1.085\ x\ \Delta T[°\mathrm{F}]$

Equation (2): $Total\ Capacity[Btuh] = SA\ Airflow[CFM]\ x\ 4.5\ x\ \Delta\mathrm{Enthalpy}[\mathrm{Btu}/lb_{AIR}]$

In Equation (1), the 1.085 is calculated follows:

Equation (3):$1.085 = \rho\ x\ c_p\ x\ 60\ [min/hr]$ where:

$\rho = density\ of\ air = 0.072\ lb/ft^3$ and

$c_p = specific\ heat = 0.241\ Btu./lb°\mathrm{F}$

In Equation (2), the 4.5 is calculated as follows:

Equation (4): $4.5 = \rho\ x\ 60\ [min/hr]$ where:
$\rho = density\ of\ air = 0.072\ lb/ft^3$

We have now discovered that the enthalpy can be determined graphically by defining the dry bulb temperature (F_{DB}) and wet bulb temperature (F_{WB}). If we can calculate the mixed air entering condition, we will understand the cooling entering air temperature (EAT). This information, coupled with the total and sensible heat load of the space and SHR will allow us to determine the required cooling airflow using Equation (2). This is an iterative process, so we will start with an assumed CFM, and see if the LAT values meet our space conditioning needs.

Course 1 included load estimates that may be used during early design. The second section of this Course will provide additional load estimating tools including manual load estimating and additional CFM/s.f. check figures that can be used to verify calculations. For now, let's look at a simple example to better understand OA mixing calculations and determine required leaving air temperatures (LAT's).

For the cooling example, let's go back to the 25,000 s.f. spec office building with limited conference rooms. Based on 400 s.f./ton and 90% load diversity, we assumed a nominal 60 ton RTU with ~22,000 CFM SA and 17% OA (3,740 CFM). Let's assume 2.5" external static pressure (ESP) which will be discussed further in Course 3. The external static pressure is required to select the supply fan and return/exhaust fan and represents frictional losses downstream of the unit supply and upstream of the unit return/exhaust connection.

With OA condition of $95F_{DB}/78F_{WB}$ and RA condition of $75F_{DB}/62.5F_{WB}$, let's calculate the mixed air condition graphically (Figure 8). This is a graphical exercise so the enthalpy numbers, wet bulb temperatures, measurements, etc. may not be exact. As you will discover in the HVAC world, we many times deal in approximate values with engineering judgement applied.

For a more finite analysis, adiabatic mixing calculations can be carried out and/or psychrometric analysis programs may be downloaded from the internet that provide these figures. These programs are typically free of charge and give the HVAC design engineer capability of determining air conditions, plotting conditions electronically on the psychrometric chart and printing results. The following charts in this Course were generated manually by marking up the chart in ".pdf" form.

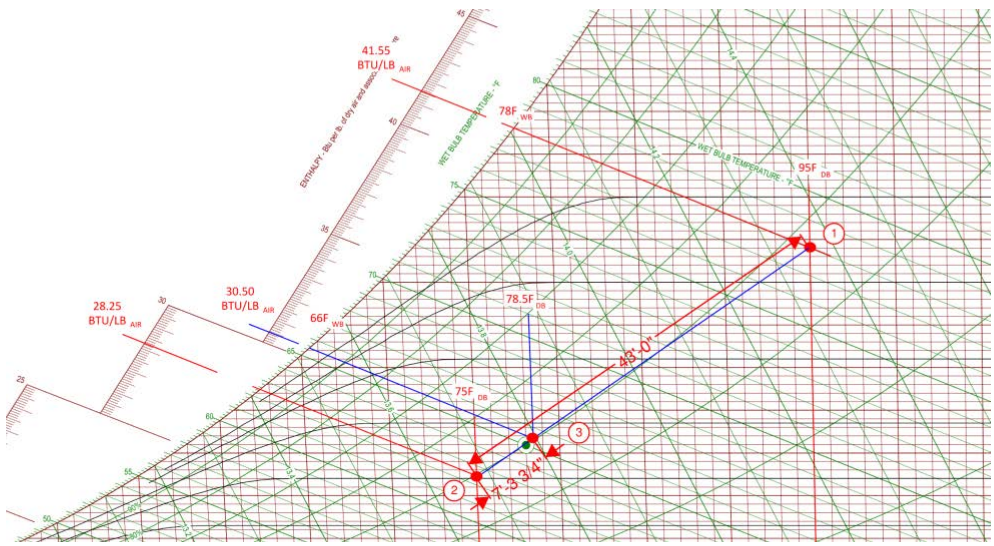


FIGURE 8: MIXED AIR TEMPERATURE CALCULATION

Step 1: Plot the $95F_{DB}/78F_{WB}$ OA condition on the psychrometric chart (Point #1)
Step 2: Plot the $75F_{DB}/62.5F_{WB}$ RA condition on the psychrometric chart (Point #2)
Step 3: Measure the distance between the two points using any scale you wish (3/32" shown)
Step 4: Calculate 17% of your measurement: $0.17\ x\ 43' - 0" = \sim 7' - 3\ 3/4"$
Step 5: Measure ~7'-3 3/4" from the RA condition (this represents 17% OA)
Step 6: Plot the intersection dry bulb and wet bulb temperature at this point (Point #3)
Step 7: Identify the mixed air enthalpy value: 30.50 Btu/lb_{AIR}

Let's assume that a block load estimate, including ventilation air, has been completed by hand, and we have developed a total load of 858,445 $Btuh_{TOT}$ and sensible load of 623,222 $Btuh_{SEN}$. These values can also be expressed in MBH by simply dividing the total Btuh by 1,000 (858.4 MBH_{TOT} and 623.2 MBH_{SENS} respectively). The 858.4 MBH_{TOT} represents total connected load with no load diversity.

For VAV systems, load diversity can be taken into account when buildings experience differing solar load, if occupants move from one place to another within the building, if there are transient occupants that come and go throughout the day or a combination of all three. For VAV applications of this nature, it is common to apply an 85% to 90% overall system diversity factor. Since we are in early design, let's apply a 90% diversity factor to arrive at 772.2 MBH_{TOT} and 560.9 MBH_{SEN}. This assumes the ventilation load is also diversified through use of the Demand

Control Ventilation strategies which will be discussed later in this Course. Plugging in the assumed CFM and Using Equation (2), we can now determine our Δh.

Equation (2): $772{,}200[Btuh] = 22{,}000[CFM]\ x\ 4.5\ x\ \Delta\text{Enthalpy}[\text{Btu}/lb_{AIR}]$

$$\Delta h = 7.8\ \text{Btu}/lb_{AIR}$$

Based on this airflow, we can calculate the enthalpy condition leaving the equipment cooling coil of 22.70 Btu/lb_{AIR} (30.50-7.8) and determine our correlating leaving wet bulb temperature of 54.0F_{WB}. The coil leaving air condition is nearly at saturation as the coil has accomplished sensible and latent cooling, the dew point of the airstream has been reduced considerably. This reduction in dew point has reduced enthalpy as moisture is condensed and removed. It is assumed that the calculated load above <u>does not</u> include the fan heat load, or any system reheat loads. The next step is to determine the sensible cooling accomplished by the RTU. Using the same 22,000 CFM and Equation (1), we come up with:

Equation (1): $560{,}900[Btuh] = 22{,}000[CFM]\ x\ 1.085\ x\ \Delta T[°\text{F}]$

$$\Delta T = 23.5°\text{F}$$

We can now establish our unit coil LAT condition As Point # 4 (Figure 9):

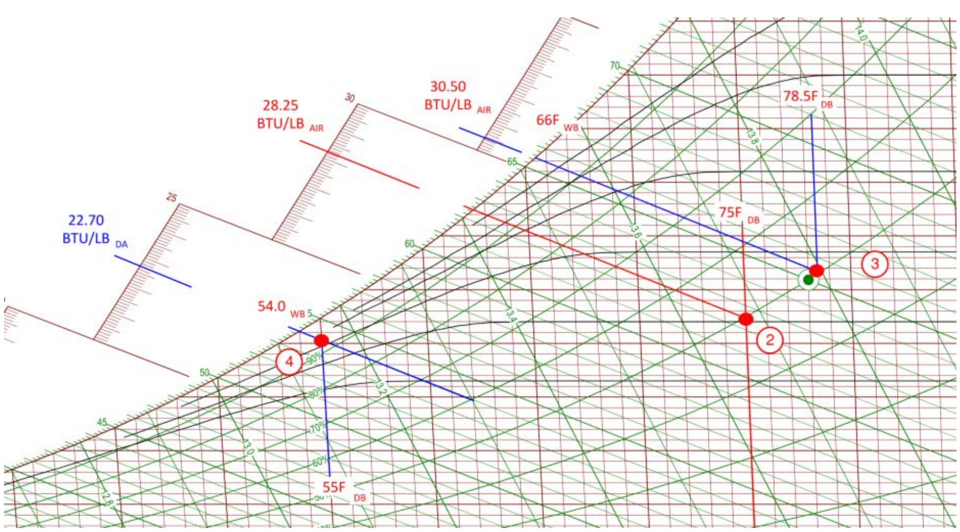


FIGURE 9: LEAVING AIR TEMPERATURE

The amount of moisture removed from the airstream (gpm) may be important to know. This can be calculated using Equation (5):

$$GPM = Airflow\ [CFM]\ x\ (1/S.V.)\left[\frac{lb_{DA}}{ft^3}\right] x\ \Delta W \left[\frac{\text{grains}}{lb_{DA}}\right] \text{x}\ \frac{1}{7{,}000} \left[\frac{lb_{WATER}}{\text{grains}}\right] \text{x}\ 0.12\ [\frac{\text{gal}}{lb_{WATER}}]$$

Using the chart, we can find that W = $71.5\left[\frac{grains}{lb_{DA}}\right] - 60\left[\frac{grains}{lb_{DA}}\right] = 11.5\left[\frac{grains}{lb_{DA}}\right]$

With S.V. = 13.3, we can solve for GPM of 0.33 GPM. That equates to ~20 gallons of water per hour if the unit were to operate continuously. This includes moisture removed from the space and incoming ventilation air. Off course, there are many factors to consider, but on a design day, it would be safe to estimate ~10 gallons of water per hour. One can easily calculate the amount of moisture removed from the space and ventilation separately by understanding the air conditions and airflow amounts of each. Remember that the mixed air including ventilation air is taken from Point #3 to Point 4 for the purpose of these calculations. The space alone is taken from Point #2 to Point #4.

Let's continue with the psychrometric chart and plot the reheat and space latent and sensible loads. To complete this, we must assume ~3F_{DB} of fan reheat and ~1.5F_{DB} increase of temperature due to ductwork heat gain. This gives us a combined reheat of ~4.5 F_{DB} which we will plot on the chart. Notice I am using the approximate symbol (~) frequently. These figures are not set in stone for any VAV project as the fan speed is constantly changing, for ducted return, the plenum temperature may be different during different parts of the day and the amount of airflow being distributed varies constantly in VAV systems. When we perform these preliminary calculations, we assume the fan is operating 100% brake horsepower, the OA is open to maximum ventilation position and plenum is at maximum assumed temperature. In operation, the VAV system will constantly change, and we are assuming worst case for preliminary design.

Another important note, the design temperature of 95F_{DB}/78F_{WB} is a conservative value corresponding to a summer ambient condition that, on average, is only exceeded ~50 hours a year. Many designers may utilize a less conservative 93F_{DB}/76F_{WB} ambient, summer design condition; however, in early design, I like to remain conservative as these preliminary calculations may lead to equipment selections and project budgets. Remember that the budgeting phase is the most important phase of any project.

We must also define the sensible heat ratio (SHR) based on the calculated sensible and total loads in the space. Going back to our loads, we had assumed 772.2 MBH_{TOT} and 560.9 MBH_{SEN} which included the ventilation load and 90% diversity. The SHR is defined by the space loads. In our case, let's assume the space load is 534.1 MBH_{TOT} and 460.9 MBH_{SENS} with no diversity. This makes the SHR=0.86. Please note the green dot with green circle around it. We will use this point on the chart to graphically represent our sensible heat ratio line by finding the calculated SHR on the right hand side of the chart and connecting that point to the green dot with a line (Figure 10):

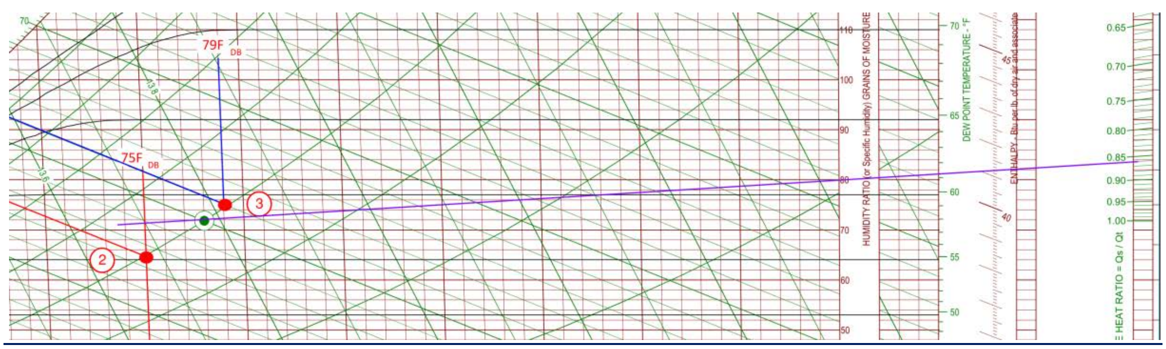


FIGURE 10: SENSIBLE HEAT RATIO LINE CREATION

Now, shift this line down to your return air point on the chart without changing the slope (Figure 11):

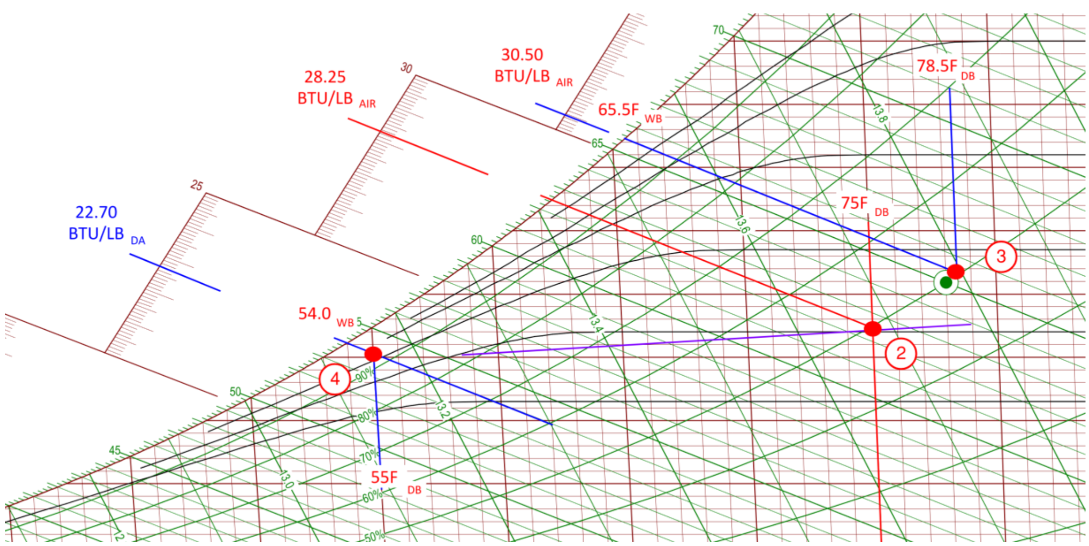


FIGURE 11: SENSIBLE HEAT RATIO FOR SPACE

Next, we apply the reheat and see if the air entering the space intersects with the SHR line. We have already recognized ~3F_{DB} increase due to fan heat and ~1.5F_{DB} increase due to duct heat gain (Point 5). Let's plot these on the chart to see if the two lines intersect (Figure 12):

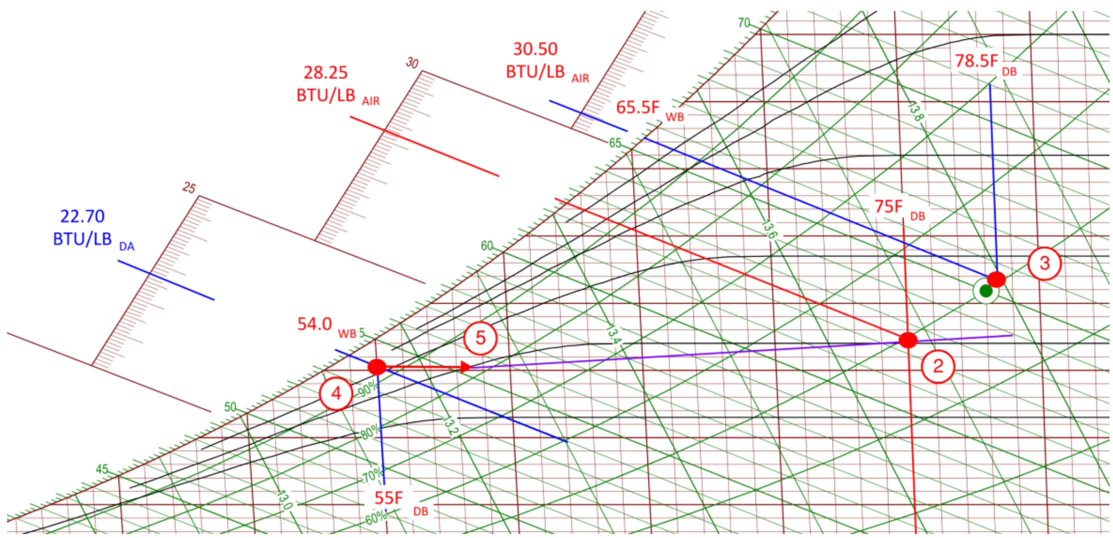


FIGURE 12: SENSIBLE HEAT GAIN OF AIRSTREAM

As you can see, the two lines do intersect, although many times they do not intersect, and the HVAC design engineer must change the airflow assumption. As stated before, this is an iterative process. Going through this exercise allowed us to establish the RTU requirements for summer cooling including:

1. Total Capacity(with 90% Diversity): 772.2 MBH_{TOT} (64.4 tons)
2. Sensible Capacity(with 90% Diversity): 560.9 MBH_{SEN}
3. Mixed EAT: 78.5F_{DB} / 65.5F_{WB}
4. Coil LAT: 55F_{DB} / 54F_{WB}
5. SA Airflow: 22,000 CFM @ ~2.5" External Static Pressure (ESP)
6. OA Airflow, 3,740 CFM @ 95F_{DB} / 78F_{WB}
7. RA Airflow Condition: 75F_{DB} / 62.5F_{WB}

These criteria would allow a manufacturer to run a selection and develop a good budget price. The team may decide to carry a nominal 70 ton VAV RTU for early budgeting until which time a detailed load analysis can be performed using DOE 2.2/2.3 approved software. At the end of the day, we will likely end up with a 60 ton VAV RTU given our design conditions.

If the load has not been calculated, the chart may also be used to determine loads using an assumed CFM value and sensible reheat. The fan reheat is calculated based on brake-horsepower (BHp) which is calculated based on CFM (standard altitudes):

$$BHp = \frac{Airflow[CFM]x\ Total\ Static\ Pressure\ ["\ W.C.]}{6356\ x\ Fan\ Efficiency}$$

I usually use a 0.80 for fan efficiency (CFM values $\leq$ 5,000; use 0.50), and for Light Commercial and Commercial applications I estimate TSP based on CFM as follows:

0-1,000 CFM:	2.0	10,001-15,000 CFM:	5.0
1,001-5,000 CFM:	2.5	15,001-20,000 CFM:	5.5
5,001-7,500 CFM:	3.5	20,001-25,000 CFM:	5.5
7501-10,000 CFM:	4.5	$\geq$ 25,000 CFM:	6.0

These work out to the following BHp values:

1,000 CFM:	0.6	15,000 CFM:	14.7
5,000 CFM:	3.9	20,000 CFM:	21.6
7,500 CFM:	5.2	25,000 CFM:	27.0
10,000 CFM:	8.8		

Given these motor Bhp values, we can calculate the ΔT values using Equation (1) as we know the CFM and can convert Hp to Btuh (1Hp = 2,544 Btuh). The values for direct drive motors in the airstream are (approximately):

1,000 CFM:	1.5F	15,000 CFM:	2.3F
5,000 CFM:	1.8F	20,000 CFM:	2.5F
7,500 CFM:	1.6F	25,000 CFM:	2.5F
10,000 CFM:	2.1F		

For Light Commercial and Commercial buildings, I perform CFM take-offs depending on the space use and function. If the spaces are not defined, I fall back to standard CFM/s.f. numbers discussed in Course 1. For healthcare projects, the SA CFM's are defined by FGI or ASHRAE/ASHE-170 recommendations and for Pharmaceutical projects, CFM's are defined by cGMP recommended ACH calculations discussed in Course 1. Below are some example CFM/s.f. numbers I have used over the years for general areas not governed by Codes, Standard or Guidelines. I have zones located on the exterior of the building (with glass) broken out separately as "exterior":

1. Combination Interior/Exterior Open Office Space: 1.2 CFM/s.f.
2. Exterior Office Space: 1.5 CFM/s.f.
3. Interior Office Space: 0.8 CFM/s.f.
4. Exterior Conference Rooms: 2.0 CFM/s.f.
5. Interior Conference Rooms: 1.5 CFM/s.f.
6. Exterior Break Rooms: 2.5 CFM/s.f.
7. Interior Break Rooms: 2.0 CFM/s.f.
8. Corridors: 0.5 CFM/s.f.
9. Exterior Waiting Rooms: 2.5 CFM/s.f.
10. Reception Areas: 2.0 CFM/s.f.
11. Small Restrooms (≤ 3 fixtures): Exhaust Only (w/ Transfer Air)
12. Large Restrooms (> 3 fixtures): 0.5 CFM/s.f. + Exhaust
13. Electric, IT Rooms: Exhaust Only or Separate Units
14. Storage/Warehouse: 1.0 CFM/s.f.

These numbers as well as the CFM/s.f. numbers outlined in Course 1 can be used as good engineering checks when evaluating loads. Once the CFM's are developed for a particular project, the desired set point is defined and ventilation airflow estimated (based on the Code), mixed air calculations can be carried out. By plotting an assumed SHR line with fan and ductwork reheat assumptions, the coil LAT temperatures can be identified. This will allow the HVAC design engineer to "back into" the primary equipment loads by either calculating them using Equations (1) and (2) or plotting them on the psychrometric chart. The risk with this approach is the SHR assumption as the SHR will vary based on space load conditions. Conservative values can be assumed based on the application to secure preliminary equipment selections and equipment budgets including the following examples:

1. Light Commercial and Commercial Offices: 0.80
2. Conference/Meeting Halls: 0.75
3. Churches: 0.70
4. School Classrooms: 0.78
5. Pharmaceutical Production Spaces: 0.85

Gaining a full understanding of psychrometric analysis is a great tool for preliminary HVAC design. It is important to understand the final design set point, SHR and sensible reheat to quickly ascertain the required leaving wet bulb condition. This data will drive equipment system selection and development of accurate project budgets. For example, commercial DX equipment is typically designed for $80F_{DB}$ / $67F_{WB}$ MAT entering the coil and $55F_{DB}$ / $54F_{WB}$ coil LAT. If you are designing an OR that must maintain $65F_{DB}$ / $53F_{WB}$(45%RH), then chilled water would likely be the better choice. On the other hand, there are custom DX options that may achieve this set point condition provided the space can tolerate +/- $3F_{DB}$ and +/-5% RH.

These fundamentals can also be applied to more complex models that may involve adiabatic mixing calculations mentioned previously (Equation 6):

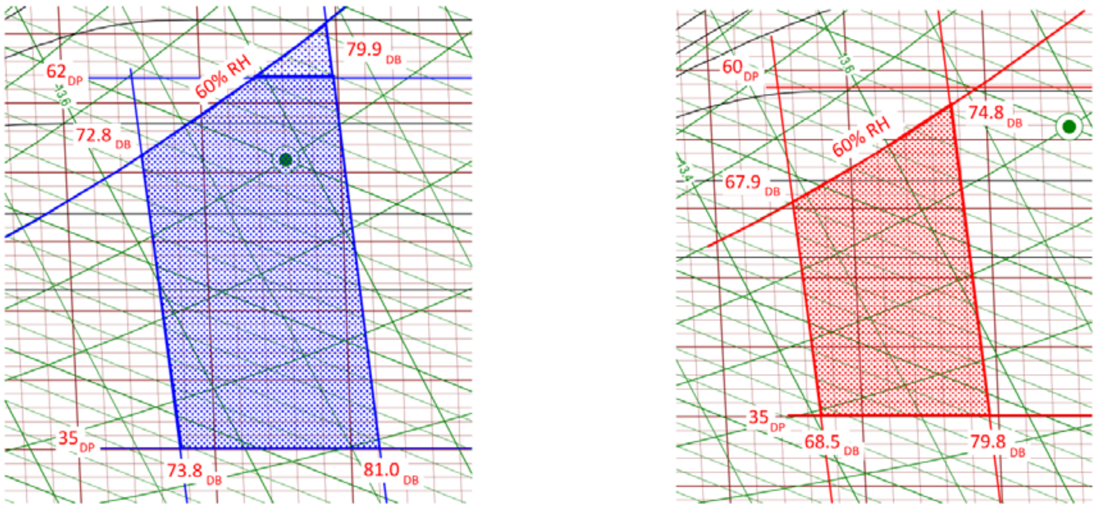
Equation (6): $h_3 = \frac{(m_{a1})(h_1) + (m_{a2})(h_2)}{m_{a1} + m_{a2}}$ where: $m = \frac{CFM}{S.V.}$

This equation is most helpful when performing calculations associated with humidification of air, combined heating and humidification of air, factoring in coil bypass, cooling moist air with spray washes, desiccant dehumidification (passive and active), passive sensible reheat, etc. These specific topics are outside of this Course, but the HVAC design engineer may use this equation to calculate mixed air conditions.

Ultimately, the HVAC design engineer must be aware of one fundamental fact: for cooling applications, if the leaving coil Dew Point is above the desired space Dew Point, proper dehumidification will not occur, and the space set point cannot be met.

I was asked to get involved with a project whereby the engineer of record had designed a 100% OA make-up air unit (MAU) cooling coil to deliver $65F_{DB}$. The MAU served terminal FCU's in the space with set point temperature of $75F_{DB}$. Overnight it got down to $55F_{DB}$ and when systems were started in the morning, it was raining outside with $72F_{DB}$. Nearly every surface in the building began to sweat because supply air to the space was $65F_{DB}$ and $64.5F_{WB}$! Perhaps the FCU's would have eventually dried out the space with sufficient sensible load from the space, but this process may have taken several hours.

It is imperative that HVAC equipment and systems be designed to maintain set point temperatures within the ASHRAE 55 summer and winter comfort zones to avoid complaints and possible growth of mold. These must be studied on a case by case basis and must include economizer operation when required by Code. During economizer (free-cooling) operation the outside air conditions can be obtained from NOAA data to ensure upper and lower limits are specified within these same comfort zones. Below are boundaries I have developed typical of an office environment, based on typical clothing worn during each season and airflow velocities described earlier in this Course (Figure 13).



SUMMER COMFORT ZONE

WINTER COMFORT ZONE

FIGURE 13: HVAC COMFORT ZONES

LOAD INPUT/OUTPUT

During preliminary design (PA/PD) and Schematic Design (SD), it is common to perform hand load or block load calculations to better define HVAC equipment and system design options. When performing any load, one must consider heat gains and losses due to conduction, radiation, internal loads (e.g., lights, people, equipment), ventilation load and equipment loads. Assumptions can be made with regards to wall , roof and floor assembly construction R-Values based on Code minimum requirements. During early design, the HVAC design engineer should be collaborating with the Architect to define these design elements and determine window square footage for each façade. Coupled with the internal use and function for each space, hand calculations can be carried out to determine sensible and latent loads. Based on the project size, it might be just as efficient to enter these spaces into a DOE 2.2/2.3 approved load modeling software. Several software packages also offer block load simulation models that allow for system comparisons.

We will start by revisiting a simplified procedure for developing loads manually for a typical office building. This simplified procedure will only consider conductive heat gains/losses and radiation through windows. This fundamental knowledge will then be expounded upon and transferred to common inputs used with simulation software programs.

As stated above, heat is transferred through wall, roof and floor assemblies through conduction. The energy transfer ($\dot{q}$) expressed in Btuh can be calculated using the following Equation (7) which is based on the wall assembly u-value where $u_{value} = 1/R_{value}$. Note that air films and convective heat exchange are not included in this equation:

Equation (7): $\dot{q}[Btuh] = u[\frac{Btuh}{h\ x\ ft^2\ x\ °\text{F}}]\ x\ A[ft^2] x\ \Delta T[°\text{F}]$

For example, if we have a 250' by 100' office building that is 13' tall, the total roof area would be 25,000 s.f. The building is oriented such that the 250' is running east/west. With a light colored, membrane roof with R-Value of 35, interior temperature of 72F and exterior temperature of 95F, one could calculate 16,429 $\text{Btuh}_{\text{SENS}}$. With this same example, we will assume 20% of the wall surface area is covered with glass leaving us with ~7,280 s.f. of wall area. The R-15 walls would experience 11,163 $\text{Btuh}_{\text{SENS}}$ of conductive heat gain. We will not consider radiation on the walls and will assume the floor slab is R-10 with below grade temperature equal to interior space so no heat transfer will occur.

Now we must account for windows which will experience both conduction and radiation heat gain. Windows are required by Code to be rated in accordance with ANSI/NFRC 100/200. In our example, we will use R-3 windows, which would have conductive heat gain of 13,955 $\text{Btuh}_{\text{SENS}}$. So far, we have established 41,547 $\text{Btuh}_{\text{SENS}}$ (41.5MBH_{SENS}) total conductive heat gain for this example.

For this Course, we will calculate the radiation heat transfer using the following Equation (8):

Equation (8): $\dot{q}[Btuh] = A[ft^2]\ x\ (E_{bn} + E_{dn}) \left[\frac{Btu}{h\ x\ ft^2}\right] x\ SHGC\ x\ IAC$

The "A" is glass area, the Solar Heat Gain Coefficient (SHGC) essentially measures how well the glass blocks heat from the sun's radiation and "IAC" refers to internal/external shading efficiencies. "$E_{bn} + E_{dn}$" represent values for a clear sky irradiance at noon on a design day found from NOAA weather data charts, NREL publications and/or there are calculators on the market that can calculate these figures. For our example, we will use a SHGC value of 0.75. We will assume moderate internal shading (blinds are half-way open) for the south/west facades, no internal shading for the north/east facades and no external shading (e.g., overhead sunscreens, building projections, etc.). The interior shading (IAC) will have a value of 0.5 for this example which will be applied to south and west facades only.

Clear sky irradiance is actually made up of three components including: 1.) the direct sunlight beams generated by the sun's disc shape in the sky (beams); 2.) the background radiation as sunlight bounces around the atmosphere and finally makes it to the window (diffuse); and 3.) direct sunlight that bounces off of the ground to the window . We will neglect direct sunlight that bounces off the ground for this example and consider direct sunlight beams (E_{bn}) for the south and west facades only. For our example, we will use 250 and 40 for our E_{bn} and E_{dn} values, respectively. This allows us to calculate radiation through our windows on the east, north facades (using E_{dn} only) as 27,300 Btuh and the south, west facades (using E_{bn} and E_{dn}) as 98,963 Btuh.

Interior loads are made up of infiltration, people, lighting, normal plug loads (e.g., copy machines, computers) and intermittent equipment loads (e.g., Electric Rooms, hospital MRI scanners). For our example, we will assume the following:

1. 0.05 CFM/s.f. infiltration through north/west walls (300 CFM) :
    a. Equation (1): 13,020 $Btuh_{SENS}$
    b. Equation (2) – 13,020 $Btuh_{SENS}$ = 13,035 $Btuh_{LAT}$
2. One person for every 150 s.f. = 167 people:
    a. 250 Btuh/person sensible load = 41,750 $Btuh_{SENS}$
    b. 225 Btuh/person latent load = 37,575 $Btuh_{LAT}$
3. 1.0 W/s.f. lighting load (w/ Ballast) = 20,000W = 85,300 $Btuh_{SENS}$
4. 0.25W/s.f. task lighting load = 6,250W = 21,325 $Btuh_{SENS}$
5. 0.5 W/s.f. plug load = 12,500 W = 42,650 $Btuh_{SENS}$
6. We have some special areas of the building to consider including:
    a. Two (2) 180 s.f. electric rooms @ 15W/s.f. (ea.) = 18,425 $Btuh_{SENS}$
    b. Two (2) 300 s.f. break rooms:
        i. 5W/s.f. sensible load @ 1,500W(ea.) = 10,236 $Btuh_{SENS}$
        ii. 2,000 $Btuh_{LAT}$ (ea.) = 4,000 $Btuh_{LAT}$
    c. Two (2) 180 s.f. copy/print areas @ 7.5W/s.f. (ea.) = 18,425 $Btuh_{SENS}$

If we add convective heat gain, radiation heat gain and internal loads, we can now calculate the internal cooling loads for the building: 400.5 MBH_{SENS} + 54.6 MBH_{LAT} = 455.4 MBH_{TOT}. As HVAC design engineers, we would typically include a 20% safety factor for hand heating/cooling calculations at this stage in design making the figures 546.2 MBH_{TOT}, 480.6 MBH_{SENS} and 65.5 MBH_{LAT}. The heating loads are calculated based on wall/fenestration losses and infiltration only as we typically do not take into account internal heat gain or radiation heat gain. The total heat load for this example (+20% safety factor), with 72F interior temperature and 0F exterior temperature is 28.1 MBH_{SENS} for infiltration, 61.7 MBH_{SENS} for the roof, 41.9 MBH_{SENS} for walls and 52.4 MBH_{SENS} for windows. We typically do not designate "SENS" for heating Btuh calculations so the total heating load for the building would be 184.2 MBH.

Now we must consider the ventilation and understand where the HVAC equipment fan brake horsepower (BHp) loads will be accounted for. Ventilation load is calculated using Tables and equations in IMC Chapter 4. For our building, we have several different spaces to consider including:

1. Open Office Space (15,000 s.f.): 5 CFM/Occupant + 0.06 CFM/s.f.
    a. Assume 167 people could be in offices = 1,735 CFM
2. Conference Rooms (3,000 s.f.): 5 CFM/Occupant + 0.06 CFM/s.f.
    a. Assume 60 people could be in conference rooms = 480 CFM
3. Break Rooms (600 s.f.): 7.5 CFM/Occupant + 0.18 CFM/s.f.
    a. Assume 25 people could be in break rooms = 296 CFM
4. Corridors (3,000 s.f.): 0 CFM/Occupant + 0.06 CFM/s.f. = 180 CFM
    a. Assume 25 people could be in corridors
5. Electric, IT Rooms, Dead Wall Space (3,400 s.f.) = 0 CFM

This totals 2,616 CFM of Ventilation Air. We must consider the 0.8 zone air distribution effectiveness (E_z) which brings our total number up to ~3,270 CFM. Since we are looking at a VAV system, we must utilize the multiple-zone, recirculating systems equations and apply the appropriate system ventilation efficiency (E_v) and distribution effectiveness (E_z) for each occupancy category based on the primary outdoor air fraction. For now, we can use the following primary airflow assumptions:

1. Open Office Space: 15,000 s.f. @ 1.2 CFM/s.f. = 18,000 CFM (E_v = 1.0)
2. Conference Rooms: 3,000 s.f @ 1.5CFM/s.f. = 4,500 CFM (E_v = 1.0)
3. Break Rooms: 600 s.f @ 2.0 CFM/s.f. = 1,200 CFM (E_v = 0.9)
4. Corridors: 3,000 s.f. @ 0.5 CFM/s.f. = 1,500 CFM (E_v = 1.0)
5. Electric, IT Rooms: Exhaust Only or Separate Units

And we can calculate the Occupant Diversity: $D = \frac{167}{267} = 0.63$

This makes our uncorrected outdoor air intake:

$$\mathrm{V}_{ou} = (D\ x \sum_{all\ zones}^{1} (Occupants\ x\ \frac{CFM}{Occupant})) + \sum_{all\ zones}^{1} (s.f\ x\ \frac{CFM}{s.f.})$$

$$V_{ou} = \left(0.63\ x\ (1044 + 375 + 234)\right) + 1{,}710 = 2{,}737 \text{ CFM}$$

Now we can apply our maximum E_v derived from our maximum Z_p of 0.9 to get 3,041 CFM.

Once this is completed for preliminary design, I typically put a safety factor of ~20% on the OA figure bringing us up to 3,650 CFM. Occupancies may change during the course of early design so it is important to ensure the HVAC equipment used in early project budgeting can handle increased levels of OA. Note that I rounded up the OA to 3,650 CFM (15% OA). During detailed design, this exercise must also be carried out for heating operation to determine the maximum amount of outside air as a percentage of heating supply airflow. These calculations will help determine the final primary unit heating load and humidifier load that may be required. In this example, by keeping the Break Room constant volume and slightly elevating the Conference Room heating CFM, we can keep our max $Z_p \leq 0.35$ which gives us an E_v value of 0.8 to get 3,421 CFM which is within our assumed value of 3,650 CFM.

It is important to understand the Code that applies to your application. The above description was based on IMC 2015. I recently completed a project utilizing IMC 2003 which has a different minimum occupancy densities, CFM's of OA and different calculation method for determining ventilation air requirements associated with VAV systems. In addition, healthcare projects require increased levels of ventilation air depending on the space use and function. All requirements must be analyzed, and the worst case ventilation air included. Work with your AHJ to determine final occupancy numbers associated with high density spaces such as Conference Rooms, Training Rooms, Classrooms, etc.. In many cases, the AHJ will allow you to apply diversity or reduce occupant counts due to absenteeism, etc.

When calculating final ventilation CFM's, we must also consider the exhaust/make-up air needs of the building and building pressurization:

1. Ten (10) toilet fixtures on the project @ 75 CFM EA (ea.) = 750 CFM
2. Break Room Exhaust @ 275 CFM (ea.) = 550 CFM
3. SA Ductwork leakage @ 0.01CFM/s.f. = 250 CFM
4. Building pressurization @ 75 CFM/exterior door (x 8) 600 CFM

We have to ensure that the final ventilation intake air meets the exhaust/make-up air and pressurization requirement (2,150 CFM in this case) and allows for minimum 10% of OA of relief for TAB efforts. In our case, the ventilation air minimum is 2,365 CFM which is exceeded by our assumed value of 3,650 CFM. Going back to the psychrometric chart and assuming $95F_{DB}/78F_{WB}$ ambient conditions with $55F_{DB}/54F_{WB}$ coil LAT's, we can calculate the loads associated with cooling 3,650 CFM of OA to saturation. With a $\Delta h = 19.30\ Btu/lb_{AIR}$, the total Ventilation load is 317.0 MBH_{TOT}. With a $\Delta T = 40F$, the sensible load = 158.4 MBH_{SENS}. This brings the unit total load up to 863.2 MBH_{TOT} and 639.0 MBH_{SENS} which does not include fan heat or load diversity. The total ventilation heating loads are calculated after SA CFM is defined and minimum heating airflows established for each VAV box (usually figure 40% for this stage of design). Remember that a comparison must be made with normal heating operation and morning warm-up associated with VAV RTU's that operate in setback mode.

Besides the compressor power consumed during cooling operation, the fan BHp loads can make up the most significant part of the building's HVAC cooling energy consumption. When defining the HVAC equipment needs, it is very important to clarify whether these loads are to be included in the equipment selections. For packaged HVAC projects, the HVAC design engineer typically defines the ventilation load, space load and then defines the HVAC cooling requirements leaving the coil. The packaged HVAC equipment is usually rated based on air conditions leaving the unit, and this may vary with manufacturer. It is imperative the equipment schedules clarify this such that overall equipment capacity is scheduled properly.

Now that we have defined sensible and latent loads for each space, defined the overall % OA and estimated SA CFM's for each zone type, we can apply the psychrometric analysis to ensure we are meeting the loads for each zone type based on our assume CFM's. This will help us to define the equipment selection(s) used during preliminary budgeting.

HVAC Load Analysis Software can be utilized as we complete the SD phase of design. The building is typically defined, which allows the HVAC design engineer to begin development of the space by space, HVAC load analysis. For this effort, final wall assemblies, roof assemblies, windows, etc. are input into the load modeling software along with internal loads. I have used EnergyPro, the Carrier Hourly Analysis Program (HAP) and Trane Trace. I am most familiar with the HAP program; however, all of the programs have the same basic input requirements. Each space is defined with a unique name which may include the Room name. It is helpful to also include the primary equipment tag if there are multiple pieces of primary HVAC equipment on the project. This makes it easier to assign particular rooms to primary equipment, zones, etc.

The units are defined (IP or S/I), the project location is defined, and simulation weather data loaded in as this allows for economic analysis to take place. It is important to choose a project location that best matches your project's weather patterns. Weather data is average, hour by hour data measured by weather stations across the world. These figures are updated as the world climate changes, and regional trends are modified. For example, the RH in Pheonix, AZ was increased after the influx of population as there was more moisture in the ambient air (perhaps due to watering lawns).

Building occupancy schedules must be defined for proper economic analyses to occur. Schedules may be set up for weekdays, weekends, holidays, etc. and include occupant density during different times of the day. For example, if an office building opens at 6:00 AM, only 10% of the occupants may be present. At 8:00 AM, the building begins to load up with occupants (perhaps 80%) with 100% occupancy by 9:00 AM. During lunch, occupants may leave for an hour. The occupancy may diminish as the workday wraps up with 90% after 5:00 PM and 60% after 6:00 PM, etc. The same strategy applies to weekends and holidays.

Different occupancy schedules may be established for lighting, people occupancy, equipment plug loads, miscellaneous latent loads, etc. When calculating the maximum cooling load, most programs assume worst case occupancy, lighting, plug loads, etc. The HVAC equipment and fans are also defined to operate on their own schedules. It may be advantageous, for example, to cycle off chillers during certain peak energy hours to reduce demand charges.

The ventilation strategy being utilized must be defined (Sum of the Spaces, ASHRAE 62.1-2015, ASHRAE 62.1-2017, etc.) and the project libraries set up including each wall type, partition type, roof type, window size and type, shading factors, etc. Unlike the block load analysis described above, the roof assembly must be defined as high albedo (light-colored roof) or low albedo (dark-colored roof) as these modeling programs consider absorptivity of radiant energy. The walls consider this factor and surface air film factors related to convection. There are several options for lighting, equipment, people activity level and other inputs. One must understand the building construction, the HVAC design air return strategy (plenum return or ducted return), building use and building function to properly input these values. Let's consider typical input values for each category.

Roof Assemblies come in various configurations. Most modeling programs come with standard templates that can be used, or custom roof assemblies can be constructed based off of the architectural drawings. It is important to understand the roof insulation system and whether or not the steel slopes. If the steel does not slope, the roof insulation system likely tapers to provide roof slope. Insulation thicknesses will be different across the entire roof assembly, so one must determine the minimum thickness such that the R-Value can be calculated.

Membrane roof systems being installed today are typically high albedo to reduce absorption of sunlight. Dark roofs absorb heat which is conducted into the plenum or above ceiling space. Dark roofs can also reach temperatures 10F-15F above normal ambient design temperatures affecting air-cooled condenser performance and putting undo wear on HVAC compressors. The HVAC design engineer must also understand what is being installed on the roof. Vegetated green roofs can decrease roof temperatures through evapotranspiration whereas solar panels can increase roof temperatures. These factors must be considered and input properly.

Sloped roofs must be included with pitch defined. The angle of the sun changes which affects how the roof absorbs the sun's energy.

Walls, like roofs come in various configurations. Walls are generally exterior, vertical design elements that separate the building from the exterior environment. Walls can be below grade or above grade. Each wall assembly must be modeled and accounted for separately. For storefront applications, the "wall" can be aluminum mullions that support glass. It is up to the engineer to closely study the architectural drawings and details to understand where the vapor barriers exist, where insulation exists and understand all interior/exterior treatments. In most cases, the wall assembly R-value is derated based on stud locations and spacing where batt insulation does not exist. The engineer must investigate and confirm this is accounted for during load modeling.

For existing buildings undergoing renovations, wall data may be difficult to ascertain. If existing architectural drawings do not exist, wall thicknesses can provide clues to the wall assembly make-up. It may be necessary to explore existing walls by dismantling interior finishes.

Windows may include reflective coatings, inert gas between panes, etc. As mentioned earlier in this Course, for new construction, windows are now required to comply with NFRC test standards and must carry a performance rating which includes R-values, SHGC values, etc. The architectural drawings will specify this data as well as window sizes such that they can be entered into the load modeling software. Windows can be entered as a function of total square footage in a space or individually by section. I prefer to enter them individually to account for additional mullions and frames.

Windows may include recessed installation which can be accounted for when defining them in the load modeling software. Sunscreens or projects may also exist above and around windows. These design elements create additional shading which should be included in the model. Internal shading factors should also be included as these significantly reduce radiation heat load.

Storefront glass is handled as a window system that may include separate mullions. I input the mullions as a wall type and include individual storefront sections on each façade. Storefront glass may also have overhangs or other shading to consider. Skylights are typically input with the roof or as separate design elements. These may be operable or fixed and must be rated by the NFRC test standards. The Code limits the number of skylights that can be used for certain applications although they are gaining popularity in Warehouse applications.

Existing window data can be found from (American Society of Heating, Refrigeration and Air-Conditioning Engineers) ASHRAE Tables or other on-line sources. It is not uncommon to find single pane glass on older buildings.

Partitions are interior vertical and horizontal separations between spaces. For partitions, if both spaces are conditioned, the partition load is zero, but if an adjacent space is unconditioned (e.g., storage room, ceiling cavity), the conductive heat transfer must be considered. In these cases, the adjacent space is generally defined with minimum and maximum temperatures based on ambient conditions. I generally use 50F for minimum temperature of ceiling cavities when it is 0F outside and 85F for maximum temperature of ceiling cavities when it is 95F outside. Adjacent storage or warehouse spaces vary depending on heating and ventilation designs.

Non-Heated Floor Slabs are in direct contact with soil temperatures and in commercial buildings are installed with a minimum R-value edge insulation. The insulation depth is typically 24" and again the HVAC design engineer must study and understand the details.

Lighting is a significant load in any building. Commercial buildings are transitioning to use of LED lights which conduct significantly less heat to the space. These changes are being reflected in the new energy Codes and affect our building designs. Coupled with decreased plug loads, commercial building SHR's are decreasing making it more important to perform proper psychrometric analyses. The design engineer needs to understand these heat loads on a space by space basis, the light venting requirements, where lighting heat is dissipated (above ceiling and below ceiling) and method of light control. Daylighting is also becoming more popular in which case the HVAC system must be able to turn down thereby increasing the SHR even more when daylighting is in use.

The IECC is steadily decreasing allowable lighting power densities. Office spaces utilizing recessed fluorescent fixtures may be 0.8W/s.f. to the space and ballast heat may be an additional 20% of that (+0.16W/s.f.) to the above ceiling cavity or plenum. For recessed LED lighting, I have used 0.45W/s.f. to the space with 60% of that (+0.27W/s.f.) to the above ceiling cavity or plenum. For plenum designs, the heat gets back to the RTU, for non-plenum designs, the heat is released into the ceiling cavity whereby temperatures can be elevated during warm summer months. For Pharmaceutical, Lab and other projects utilizing interstitial or occupied mezzanine spaces, the heat needs removed while protecting the fixtures. Ventilation is very important for recessed light fixtures as heat will accumulate and potentially damage the fixture.

Exposed or surface mount fixtures put all of their heat into the space. It is important to understand the lighting choice for each application and if wall sconces or accent lighting will be used. For example, Pharmaceutical applications may utilize exposed, tear-drop LED lights with very high lighting density in process spaces and may use recessed, LED or fluorescent lighting in packaging areas. Production/office areas may have recessed lighting with task lighting at each cubicle.

Plug Loads have also steadily decreased over the years. Technology is becoming more compact and laptops more prevalent. Offices and office areas now utilize small, flat screen monitors and many building owners are eliminating portable fans and heaters to help control costs. What used to be 1.0W/s.f. has quickly turned into 0.5W/s.f. plug load. Additional equipment such as copy machines, printers, plotters, etc. need to be located and each piece of equipment analyzed for heat dissipation. Break rooms and small kitchens require additional scrutiny with equipment in each space being input as W/s.f. or Btuh/s.f. Electrical panel rooms may require conditioning via the "house" VAV system, separate DX split system or ventilation cooling via exhaust fan and transfer air. In some cases, there are small transformers located in these rooms that can release ~2.5% of their KVA rating to the space. I generally use 15W/s.f. and then add transformers separately.

IT Closets that have a power density $\leq$ 20 W/s.f. or a total load $\leq$ 10 KW are typically conditioned with a separate DX split system as sensitive computer equipment requires cooling during unoccupied hours and is generally designed for 65F-68F set point. For rooms housing computer or electronic equipment exceeding these values, a separate Computer Room Air-Conditioner (CRAC) unit is provided. Computer equipment generally gives off 80% of their input power as heat. IT Racks can be given a 0.7 diversity factor based on total load output. So, if I have a computer room with two (2) 40A breakers at 240/3/60, I will calculate total KW input using Equation (9):

Equation (9): $KW = \frac{80[A] x\ 240\{V\} x 1.72\ x\ 0.85[p.f]}{1000} = 28.1KW$

I would then take 27.1 KW x 0.8 x 0.7 to get 15.72 KW output. If I have an 18' x 12' room for the two IT racks, my equipment (or machine) input would be 75 W/s.f. This does not include partition loads, lighting loads, UPS or other electrical panels that exist in this space. In this example, the room will be classified as a computer room even if it exists within a large office building. The HVAC equipment must then meet testing requirements stipulated in the Code.

For Labs, Healthcare, Pharmaceutical, Food & Beverage, Industrial and some Warehouse applications, I generally procure equipment lists showing input electrical requirements. Manufacturers may provide equipment efficiencies such that heat output can be calculated and in some cases, the heat output in Btuh is provided. This information is then tabulated to develop a space by space W/s.f. number. I've encountered Industrial Panel Room (I/O Rooms) supporting the Site Process Control System (PCS) with sensible loads exceeding 150W/s.f. so this analysis is very important.

Occupant Load was described in the first part of this Course including descriptions of heat loads. Most load modeling programs include default sensible and latent Btuh values given a defined activity level. Activity level is important as is the occupant density. Minimum densities are defined in the Code, however Training Rooms, large Conference Spaces, etc. may have increased levels of occupancy. The HVAC design engineer must include these occupants such that sufficient sensible and latent loads can be accounted for. Buildings that include these spaces typically have larger diversity applied when sizing primary equipment. It is important to understand the Owner's planned usage for the facility. I was involved with a building which included a large pre-function/museum space, meetings rooms, board room and office space. I learned that every space in the building could be 100% occupied as they planned to bring visitors in periodically.

A common question I receive is load output data that indicates fractional amounts of occupants in each space (e.g., 0.68 people in a 100 s.f. office). For small offices, I generally round up to the nearest person; however, larger offices may have transient loads so fractional occupants are completely acceptable. As we will discover in Course 3, the CFM required to cool "0.4 of a person" (~6 CFM) is generally accounted for in the final space CFM selection. The real affect is on the amount of outside air required for all of the spaces when dealing with large facilities.

Miscellaneous Latent Load can be generated in kitchens, break rooms, wash-rooms and various other spaces with food, water and/or steam. Dishwashers and autoclaves can intermittently generate steam which may or may not be exhausted. Commercial kitchens have capture hoods for the dishwashing equipment and typically do not have hoods above the prep sinks or drying racks. I generally include 2,000 $Btuh_{LAT}$ for break rooms, 2,000 $Btuh_{LAT}$ for each cook top and 2,000 $Btuh_{LAT}$ per wash sink and drying rack in commercial kitchens. Although dishwasher hoods do remove large portions of steam, I typically include 500 $Btuh_{LAT}$ per machine.

For process steam loads such as laundries, sterilizers, autoclaves, etc., the manufacturer's data will have water consumptions which can be used to calculate latent load via the steam tables. A rough estimate I have used is GPM x 500 = PPH Steam and 1 PPH = 969 Btuh and then take those times 85% to get latent heat of vaporization. I usually assume 2% escapes to the room and the rest either condenses within the machine or gets captured with exhaust. So, an autoclave using 3 GPM of water would produce some ~ 23,256 $Btuh_{LAT}$ for a short period of time during its run cycle. Larger equipment typically requires secondary exhaust to capture steam before it mixes with the space.

Infiltration can be a significant load and create comfort issues if not addressed during design. In general, buildings are positively pressurized during occupied hours, however airflow may infiltrate building envelopes in heavy winds, airflow may enter through doors as they open and shut, etc. This becomes a larger issue in Hi-Rise Commercial applications and Warehouses. For design input, I typically assume infiltration of 0.03 CFM/s.f. to 0.05 CFM/s.f. for facades (exposed walls) experiencing prevailing winds. I also assume infiltration of 100 CFM through each man-door for vestibules and egress only doors for tall buildings. For Warehouse projects, infiltration could be up to 400 CFM per door (dock position). For unoccupied hours, when the HVAC fans are off and OA dampers shut, infiltration may increase to 0.10 CFM/s.f. for all facades.

Moisture can infiltrate buildings and rooms opposite of airflow depending upon the vapor pressure differential. I was involved with a project requiring a space to be maintained at $68F_{DB}$ and 7% RH. The adjacent spaces were at $68F_{DB}$ and 25% RH, so moisture infiltration had to be calculated by hand based upon the permeance construction materials, openings between the two spaces, ductwork leakage, etc. Since the space condition was outside of typical selection program parameters, the additional space loads and HVAC unit serving this space were all calculated manually.

HVAC System and Equipment inputs include a complete description of each primary piece of equipment serving the building with their respective equipment tags. Some systems, such as chilled beam, may be modeled as a similar piece of equipment (e.g., fan coil units) depending on the software package used. These inputs may include:

1. Equipment type:
   a. Central Station AHU's (applied equipment)
   b. Packaged RTU, DX Split systems
   c. Terminal FCU's, Chilled Beam, etc.
2. Equipment accessories:
   a. Economizer, Total Enthalpy Wheels
   b. Coil Configurations, Locations and Types
   c. Sensible Heat Reclaim Devices
   d. Hot Gas (or Other) Reheat
   e. SA, EA, RA Fan Data, TSP, Efficiency
3. Distribution Ductwork information
   a. Assume Heat Gain, Leakage
   b. Ducted Return Air or Plenum
   c. If Plenum, What % of Total Wall is Plenum

If VAV application, the spaces are grouped into Zones and assigned to primary equipment. Each zone heating/cooling set point is defined, direct exhaust, diversity factor and Air Terminal units defined in terms of reheat, minimum airflows, etc. Large zones associated with VAV systems are assigned a means to accomplish Demand Control Ventilation (DCV) as required by Code. DCV utilizes space Occupancy Sensors and CO2 Sensors which measure the amount of CO2 (in ppm) thereby predicting occupant numbers. The multiple space equation is constantly calculated in the Building Automation System (BAS) based on the number of occupants in these spaces. This allows for reduction of outside air into the building as measured by an OA airflow meter. For economic analysis, these inputs allow the program to reduce OA ventilation based on load diversity and occupancies of these zones.

Equipment sizing data and safety factors are input. For load analysis, I typically input 5% or 10% safety factor depending on complexity of the project, project type, etc. The safety factor may be limited by jurisdiction or state so local Codes and ordinances should be consulted. In some cases, spare capacity may be required for future phases, etc. It is important that the DX equipment not be over-sized or be provided with proper staging controls if spare capacity is required.

For DX applications, it is common for HVAC units to short cycle when the set point temperature is lowered and there is insufficient sensible load in the space. DX equipment requires ~10-15 minutes of operation to remove moisture. If the DX equipment is staging on/off every 5 minutes, the moisture will condense on the cooling coil and never drip off, rather, it will re-evaporate into the airstream. Over time, this will build humidity and moisture in the occupied space. If the equipment is not supplied with a means to dehumidify the space, this can lead to growth of mold in the building.

For chilled water and hot water applications, plant information is input and applied equipment and systems (AHU's, FCU's etc.) assigned. Each chilled water and hot water plant must include system configuration, boiler information, chiller information, pump data (hp/GPM), fluid type, cooling tower information. Boiler efficiencies may be input based on capacity demand. Chiller "maps" may be entered which include varying KW/ton energy consumptions based on various condenser water temperatures. For economic analyses, it is important to include all pump and miscellaneous loads including transfer fans, exhaust fans, chemical feed pumps, electric humidifiers, etc.

Output Data will provide a space by space required cooling and heating airflow. I will round each CFM up to the nearest increment of 25to develop my design CFM's for each space. For example, if the output report has required cooling CFM at 134 for an office, I would round this up to 150 CFM. Similarly, if a conference room shows 378 CFM, I would round this up to 400 CFM. Keep in mind that, for office applications, the Test and Balance Contractor only needs balance to +/- 10% of design airflow. It is possible that the final CFM for the office could be 135CFM and the conference room 360 CFM. If the load input has a 5% safety factor, the conference room still falls within design CFM for the worst case, maximum cooling load.

The output data will also provide primary equipment capacity data required to meet the heating and cooling loads. The capacities include CFM's, entering air temperatures and leaving air temperatures that need to be studied very closely. For VAV applications, this is good data to have as the system diversity is already considered at the system level. It is important to still select equipment with supply and exhaust fans that can handle the connected CFM as this will be required during test and balancing activities.

Ventilation data is also included in the output data on a space by space and system level. For multiple space, VAV applications, the software has already calculated the multiple space equations and determined the primary equipment capacity based on worst case minimum ventilation over all of the load profiles and occupancies. For example, a VAV unit operating at 70% design airflow may experience higher entering air dewpoint conditions requiring more capacity to meet the required LAT's.

For healthcare applications requiring compliance with FGI or ASHRAE/ASHE-170 minimum outside air, supply and exhaust airflow amounts, I generally run the airflow analysis separately via spreadsheet. This data is then compared to the load output to determine worst-case outside air, supply and exhaust airflow amounts. The final AHU or RTU selections can be made based on the spreadsheet data. Large hospitals utilizing central station AHU's or custom AHU's typically include minimum 10% spare capacity for primary equipment. This is to account for possible changes that may occur in the future (e.g., converting patient exam areas into a Physical Therapy Suite). For economic analyses, the total space ACH or ventilation amount can be altered in the input data to better simulate the final equipment selection. Increases to BHp can also be captured as miscellaneous load or static pressure requirements increased to account for additional loads.

For Pharmaceutical applications requiring high ACH rates, it is typical to select equipment based off of the spreadsheet data which may include space and equipment sensible/latent loads. Ventilation loads must always be compared to required outside air for pressurization to ensure compliance with the Code. Data may be input into a load modeling software depending upon the project permit requirements. Large-scale manufacturing facilities utilizing custom AHU's typically include minimum 20% spare capacity for primary equipment to account for future equipment and process changes. It is common to include fan wheel width control to allow for appropriate fan selections or utilize multiple fan arrays to allow for future capacity.

It has been my experience that active desiccant units used in Pharmaceutical, Food & Beverage and specific lab applications require separate analysis. These airstreams are typically mixed with return air from the space thereby altering mixed air conditions. When this equipment is applied, hand calculations are generally required to determine final EAT temperatures that can be used in final equipment selections.

As discussed in Course 1, Refrigerated Warehouses are modeled using specialty software programs that take into account product loading, product in-bound temperatures, redundancy to account for defrost cycles, etc. These programs should be applied to any Warehouse maintained below $60F_{DB}$ and 60% RH. Conditioned Warehouses above this temperature can be modeled using standard load modeling software; however, product loading and recovery times can be difficult to calculate. Generally speaking, these applications require 4-6ACH with good mixing between aisles which can be accomplished with destratification fans, air turnover units or HVLS fans in some cases. Heat and Vent Warehouse loads are typically calculated manually using a spreadsheet. The office areas located within or adjacent to these large Warehouses are modeled using load modeling software.

One must always review the psychrometric chart output data to ensure appropriate input assumptions have been made and the output will meet the zone heating and cooling needs. As with any program, the output is based on the input data, so the engineering checks described above and in Course 1 can be applied to verify the final output design is in line with normal standard of care.

The use of computational fluid dynamics (CFD) is becoming more popular with performance based designs. I have been involved with CFD designs associated with Pharmaceutical clean rooms. The room geometry is entered into the software along with interior furniture, equipment, finishes, HVAC SA, RA and EA air devices/grilles and data entered for HVAC discharge, return and exhaust velocities and airflows. Given all of this data, the program will predict airflow vectors throughout the space. Each vector will include airflow velocity, airflow direction and can predict temperature zones within the space. A temperature “map” can be developed showing potential for hot zones, cold zones and thermal barriers. Areas of turbulence can be predicted based on these static conditions. I was also involved with a project utilizing CFD analysis to predict Smoke Evacuation in a four-story atrium. The model considered heat/smoke generation from a fuel source and removal of heat/smoke plumes given a make-up air source, exhaust locations and space geometry.

The CFD models are powerful tools and are shown to be accurate given static situations (i.e. the only thing moving is the air). The CFD models can predict changes to airflow and temperature patterns given occupants standing within the room. In the above examples, CFD analysis was used to predict proper velocity within a Pharmaceutical clean room showing unidirectional airflow and acceptable velocity at a work surface as well as sustained velocities within the space to prevent accumulation of contaminants. In the other example, CFD analysis was also shown to predict smoke evacuation of a large space with no occupants. In both cases, smoke testing was planned to occur such that the CFD analysis could be validated. This was carried out with the Atrium and the CFD model was shown to be valid.

In the healthcare industry, there have been studies to show modeled airflow patterns and temperature gradients are valid with occupants in the room and out of the room. This was accomplished using temperature sensors, airflow sensors, etc. in an OR room. All of this data was then compared to the CFD model which showed a high level of accuracy when occupants were standing still. When occupants move within a space, obviously the data is skewed, however this model proved proper unidirectional flow of air above the operating arena given the HVAC SA diffuser layout.

CFD analysis can be used in commercial design to verify good airflow mixing within occupied spaces, verify air device/grille positions and determine if airflow velocities will be sufficient to condition the space in VAV applications. HVAC SA, RA and EA air devices/grille positions can be changed, or selections modified to enhance mixing. CFD can also be used with natural ventilation systems to validate ventilation is reaching all occupants within the space. In addition, CFD analysis can predict propagation of particles, viruses, etc. in occupied spaces by using time lapse models.

CFD analysis has many uses including predictive analysis, design enhancement and verification of existing systems. This technology will continue to evolve and grow within the HVAC industry. There has already been software integration with Virtual Design and Construction tools, and I predict this trend will continue.

ECONOMIC ANALYSIS

Course 1 described various HVAC system and equipment options that may be discussed and evaluated during early design. That Course also emphasized the importance of proper financial analysis and budgeting during early design. As with load calculations, economic analysis may be carried out using hand calculations, by using spreadsheets and/or by utilizing load modeling and economic software. The HVAC design engineer must be well versed in equipment and system options, have an understanding of first cost (equipment and installation), ongoing utility costs, maintenance costs, operations costs and replacement costs associated with each piece of equipment. If the project is financed, project payments and compounding interest must be considered. When analyzing these options, we typically take all costs back to a Net Present Value in today’s dollars over a set term such that informed decisions can be made. Typically, we present comparative models analyzing HVAC equipment and system options (e.g., Chilled Beam with DOAS vs. High Performance VAV)

Utility costs are the first step to completing these calculations. In the case of renovation, the cost of electricity, natural gas and water may be estimated from existing utility bills. For new construction projects, this data can be procured from utility companies based on the project size and type. Data is also available from the Energy Information Administration (EIA). The utility company rate plans may be structured with base fees, on-peak and off-peak rates. The rate plans will likely include a rachet clause which increases the rate beyond a particular consumption during peak hours. For hand calculations, I typically use a flat rate during occupied hours to perform a comparative analysis. An example in the Midwest might be \$0.08/kWh or \$1.10/Therm. Water rates are defined by the utility provider or municipality. For consumed water, chemical costs must also be considered. For example, it may cost \$2.0/1,000 gallons of water and an additional \$2.2/1,000 gallons of water in chemicals for a combined total of \$4.4/1,000 gallons.

System run hours can be estimated based on building use. For an office building that experiences unoccupied mode, we may consider cooling will operate 7 months of the year during occupied hours: $\frac{8{,}760\ hours\ per\ year}{12\ months}\ x\ 7\ months\ x\ \frac{10\ hours}{24\ hours} = \sim 2{,}130\frac{hours}{yr}$

…and heating may operate during occupied hours (during occupied hours): $\sim 1{,}521\frac{\text{hours}}{\text{yr}}$.

As a simple example, we might consider a system that consumes 1.3 kW/ton of cooling @ \$0.08/kWh and has a cooling load of 500 tons. For this example, we would have an annual cooling bill of \$110,760. By increasing the efficiency of the cooling system to 1.1 kW/ton, we can save the Owner \$17,040/yr. If it costs +\$75,000 for this upgrade, we have a simple payback of 4.5 years which does not include increased cost for maintenance, operation or disposal.

We may also consider simple comparisons for cooling Light Commercial and small Commercial buildings including these assumptions:

- Single Zone RTU	\$15.00/s.f.	2.0 kW/Ton
- RTU VAV w/ Electric Reheat	\$20.00/s.f.	2.2 kW/Ton
- RTU VAV with Hydronic Reheat	\$25.00/s.f.	1.9 kW/Ton
- Air-Cooled Chilled Water w/ AHU's and Hydronic Heat	\$35.00/s.f.	1.7 kW/Ton
- Water-Cooled Chilled Water w/ AHU's and Hydronic Heat	\$45.00/s.f.	1.1 kW/Ton

Using the above figures for a 25,000 sq. ft. office building using ~60-tons (70-tons with no diversity) of cooling 2,130 hours/yr. and we get the following annual energy costs:

- Single Zone RTU	$375K	$23,856/yr.
- RTU VAV w/ Electric Reheat	$500K	$22,493/yr.
- RTU VAV with Hydronic Reheat	$625K	$19,426/yr.
- Air-Cooled Chilled Water w/ AHU's and Hydronic Heat	$875K	$17,381/yr.
- Water-Cooled Chilled Water w/ AHU's and Hydronic Heat	$1,125K	$11,246/yr.

This data leads us to the following conclusions for this cooling comparison:

1. The Single Zone RTU option seems to be the most economical choice
2. The RTU VAV w/ Electric Reheat option is more costly, does not have a good payback (<$1,365/yr.) and does add more zone control
3. The remaining option energy paybacks are very low making increased project costs prohibitive

These simple hand calculations work well for individual pieces of equipment or simple system comparisons; however, it is easier to use spreadsheets for more complicated calculations including discount rates, escalation and net present value of money. The discount rate is defined as the interest rate used to discount future value of money so that monies can be represented in today's value (net present value). If the increased capital cost has a 6% discount rate (i), we consider an annual utility escalation of 7%, and we consider increased service contracts and maintenance at $5,000/year with an annual escalation of 5% over a 20 year term (t), we can calculate the Net Present Value (NPV) of this proposed change. The simple interest equation used for year over year cost escalations is shown below as Equation (10): $A = P(1 + rt)$

where: A = the accrued amount year over year (principal + interest)
P = Principal Amount; r = rate of interest; t = term (we are using 20, 1 year terms)

The equation for NPV is shown below as Equation (11) and this must be calculated for each period:

Equation (11): $\text{NPV}=\frac{R_t}{(1+i)^t}$ where R_t = Cash outflow or cash inflow.

Using Equation (10), we find that our accumulated energy savings over 20 years (annual periods) is 698,553 (which includes utility escalation costs) and our cost of maintenance is $165,330 (which includes maintenance escalation costs). Using Equation (11), we find the accumulated savings in NPV = -$265,678. So, with an initial investment of $75,000, the change will make ~$190K over 20 years as represented in today's dollar value. By tracking cumulative cash flow by annual period, these numbers show that the actual pay-back period is a little over 5 years. Cumulative cash flow looks at all the savings and expenses on a year by year basis.

For system comparisons, these calculations can also be used with Net or Site Energy Intensity Index (NEUI/EUI) figures expressed in kBtuh/s.f./year. This data can be converted to kWh/s.f./year and used in a comparative analysis. For example, an office building utilizing constant volume, DX RTU's may have a Site EUI of 21 kWh/s.f./year. High performance VAV may have 15 kWh/s.f./year and ground-source, geothermal heat pumps (GSHP) may have a Site EUI of 9 kWh/s.f./year. Installation costs, maintenance costs, etc. can be developed for each system type and comparisons developed in NPV.

For all comparative analyses, local utility rebate programs should be considered to offset initial capital costs and utility rate rebates should be factored into ongoing utility costs. In many cases, the process can be started with spreadsheet calculations. I was involved with a Data Center Design whereby the Utility provided nearly $250,000 in rebates for more efficient HVAC systems.

In the above office building example comparing the use of both constant volume, DX RTU's and High performance VAV with ground-source, geothermal heat pumps, we found that the GSHP had about a 19.8 year simple payback and an actual +25 year payback. This was due in large part to the cost of a new bore field and low cost of electricity for this particular project. (typically, GSHP projects pay for themselves in 7-10 years). This example proves that calculations should be carried out on a job by job, system by system basis.

For early design, several load simulation software packages offer system level comparisons and analyses. These are based on block load data appropriate for early PA/PD or SD design phase. The building shape, orientation, R-Values, fenestration percentages, etc. can be changed with different systems to understand the energy impact. Some Owners may request that full energy analyses be completed using load simulation software tools based on space by space input. These comparisons require additional inputs including utility costs, building power factors and source energy generating efficiency. Many load modeling software packages include standard EIA utility data including demand charges.

Inputs are also available for all escalation percentages, cost of capital, discount rate and analysis period. Installation and equipment costs are input for each system as well as maintenance, service contracts, disposal costs, etc. If all systems and equipment have been entered properly, the output data can include electrical and natural gas annual consumption, energy intensity, etc. The HVAC engineer should be familiar with executing these analyses and be prepared to present results as Owners and design teams evaluate different decisions. In addition, complete energy Code evaluations need to occur with each system to ensure scope is complete. Course 1 mentioned ComCheck which is an excellent resource for ensuring Energy Code Compliance.

SUMMARY

This Course described key comfort factors that the HVAC design engineer should consider during equipment and system design. These included (in order of importance):

1. Conduction/Convection/Radiation
2. Air Movement
3. Humidity
4. Temperature

In addition to these thermal factors, we discussed other factors associated with Indoor Environmental Quality (IEQ) including the perception of noise in the HVAC system, air quality, lighting levels, work-station ergonomics, ambient noise and worker ambient activity.

This transitioned into a review of psychrometric analysis fundamentals that can be applied to more complicated scenarios. It is important that the HVAC design engineer fully understand the specific application such that proper leaving coil dew point temperatures can be achieved with the HVAC equipment.

This Course was intended to review these design fundamentals and familiarize the reader with Load Input/Output techniques. Economic analysis is a useful tool in developing Owner recommendations such that final HVAC equipment and system selections can be made. After a complete evaluation occurs using system selection criteria outlined in Course 1, and a final system selection is made, the knowledge obtained in this Course will allow for actual HVAC layout and design discussed in Course 3.