DESIGNING FOR ABSOLUTE MOISTURE CONTROL

Co-Authors: R. Mark Nunnelly, PE – President, Nunnelly & Associates, Inc., Birmingham, AL and J. Patrick Fex – President, Fex Environmental Systems, Inc., Gulf Breeze, FL

ABSTRACT

Rarely is an industry trade journal published without at least one article included concerning the topic of Indoor Air Quality (IAQ) or the related Sick Building Syndrome (SBI). Whether the subject of the article is a school, an office building or a public assembly building, chances are the origin of the IAQ problems stem from excessive moisture in the space. Excessive moisture in the space can either come from water damage (leaking pipes or poor integrity of the building envelope), or it comes from uncontrolled, excessive humidity in the space.

This paper pertains to the latter of these two sources. With the air conditioning technologies available, and the published design data available through ASHRAE today, appropriate HVAC systems can be designed to adequately control not just the temperature in the spaces, but also the humidity.

BACKGROUND

"Sick buildings", "indoor air quality problems", "building related illnesses", and on and on the headlines go. Why are we having so many problems in our buildings today? Obviously, there are a number of reasons and causes for these various IAQ related problems, far too many to address in this paper. Hopefully one of the leading causes of these problems can be sufficiently addressed in these next few pages. After reading any number of published articles and studies concerning IAQ problems in buildings, it is obvious that a leading culprit is excessive moisture in the space. This moisture could be from water damage within the building (e.g., leaking pipes, water spills, etc.) or through the building envelope (e.g., improper vapor retarders on the outside walls, leaks in the building envelope, roof leaks, etc.). On the other hand, the moisture damage could be from the lack of moisture control by the building's HVAC system (see Figure 1). And unfortunately, many buildings suffer from a combination of both of these contributors.

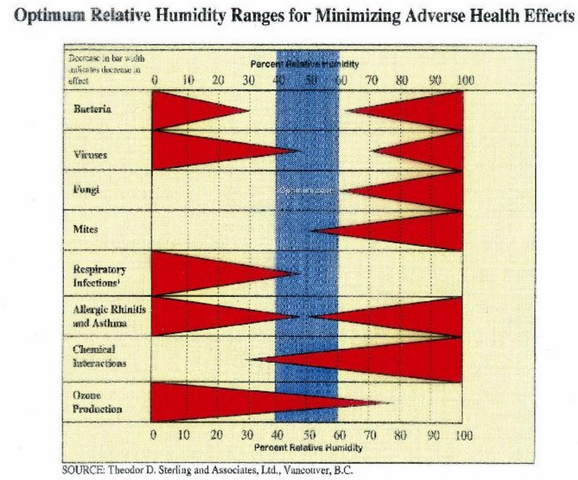

Figure 1. Optimum Humidity Ranges[1]

Discussions concerning the building envelope and/or the pipe leaks will be left for other professionals to cover. The sole focus of this article will be on the proper design and selection of the HVAC system required to achieve and maintain proper *control* of the humidity in the space.

Why the increase in moisture related IAQ problems in our buildings today? It seems that just 10 years ago, we rarely heard of a building with moisture problems. There are obviously a number of causes, but one of the leading causes is sure to be ventilation air. Too much? Not enough? Poor conditioning of the ventilation? Yes to all.

Since the late 1970's, our buildings, as a whole, have been constructed with a much "tighter" envelope. Better insulation, improved vapor barriers, double and triple paned windows, etc. In years gone by, even though our buildings were "leaky" and weren't very energy efficient, they were more "forgiving" and allowed the building to "breathe". Because our buildings are "tighter" now, we are introducing all of the ventilation air for the occupants via some type of HVAC system, or systems. In many cases, the buildings aren't getting sufficient quantities of outside air (i.e., ventilation air). According to a study by the National Institute of Occupational Safety and Health (NIOSH)[2], of the buildings studied that had reported IAQ problems, over half (52%) of the problems were the result of inadequate quantities of ventilation air being introduced into the space[2].

However, simply adding more air into our buildings isn't necessarily the cure either. Especially in geographic regions of our country referred to as "hot and humid" climate zones. We must make certain we are not just trading one problem for another. In much of the country, such as in the hot and humid southeastern United States, proper conditioning of this ventilation air must take place to insure the building will not suffer from excessively high humidity levels. Without a doubt, the majority of the moisture (latent gain) inside a typical building comes from the ventilation air. Therefore, it only makes sense to *actively control* the absolute moisture level of this ventilation air. Far too often, we have only *passively controlled* the moisture (i.e., humidity) within the space as a byproduct of the temperature control of the air (i.e., cooling).

IMPROPER DESIGN CONSIDERATIONS

Let's assume for the sake of discussion, that the summer design condition for a space within a school's classroom is to be 75^0F drybulb and 50% relative humidity (Figure 2). Considering approximately 30 students are to be in the classroom, would you supply air at 75^0F drybulb? Of course not! The supply air would need to be something considerably less than 75^0F drybulb in order to absorb the sensible heat generated by the students, the lights and the solar gain. However, if we are to maintain the space at an absolute moisture level of 55^0F dewpoint (corresponding dewpoint temperature for 75^0Fdb/50%rh, see Figure 2), then why do we often supply air at only 55^0F dewpoint? Do we not need to account for the moisture gain (i.e., latent gain) from the students? Of course we do! When conventional packaged direct expansion (DX) cooling/dehumidification equipment is installed, the average supply air dewpoint is approximately 55^0F. What are we relying on to remove the moisture from the space to achieve the necessary level? And exactly what should this supply moisture level be? This paper will demonstrate how to calculate the absolute moisture level of the supply air in order to *prevent* any moisture-related problems from occurring as a result of the introduction of ventilation air.

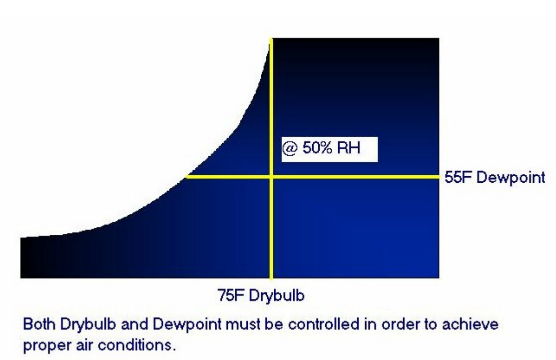

Figure 2. Typical Space Design Condition

LOAD CHARACTERISTICS

When calculating the cooling and dehumidification loads of a building, there are two categories in which all loads can be placed. Either the load is considered an *internal* or *building load* (e.g., people gains, heat from lighting or appliances, solar gain through walls and windows, etc.), or the gain would come from the ventilation air. This latter component of the load is significantly larger than what might have been calculated just over 10 years ago. With revisions in ASHRAE's Standard 62-99, we are in many cases doubling or tripling the amount of fresh air being introduced into our buildings. Unfortunately, in the hot and humid climates, this also means we are doubling or tripling the amount of moisture we are introducing into our buildings. All too often, conventional HVAC systems are installed to include the same technologies that were installed prior to the ventilation rates being increased. Considering the two load types, internal versus external (ventilation), are they coincident loads? Do they follow the same patterns? Of course not! The internal loads of a building are fairly consistent, day-to-day. Generally the same number of lights are on, the same number of occupants are present, the same number of computers are operating, etc. The "skin" load varies, but the changes are gradual due to the thermal storage characteristics of the building materials. On the other hand, the loads associated with the ventilation air (sensible *and* latent loads) can fluctuate significantly throughout the day, unlike the internal loads. This being the case, why should one HVAC system be used to *control* both internal and external loads?

PROPER DESIGN CONSIDERATIONS

By utilizing separate pieces of equipment for the internal loads, and for the treatment of the ventilation air, these loads can be handled independently and accurately. This is called the "Divide and Conquer" method (see Figure 3).

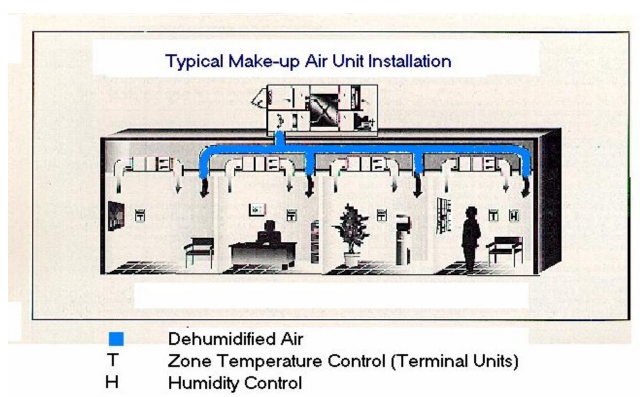

Figure 3. Divide and Conquer[3]

One of the greatest mistakes designers make in sizing these ventilation air conditioners (also referred to as "make-up air units", or "outside air units / outdoor air units") is to ignore the valuable weather data that is available in the ASHRAE *Fundamentals Handbook, 2001*[4]. Past *Fundamentals Handbook* editions (1993 edition and earlier) only included the weather conditions for the occurrences of the Drybulb / Mean Coincident Wetbulb for a number of cities. This condition indicates the peak *sensible* condition of the ventilation air, but does not indicate the peak *latent* condition of the air. The 1997 edition of the *Fundamentals Handbook* for the first time included two other occurrence tables. The second set of tables includes the occurrences of the Wetbulb / Mean Coincident Drybulb and should be used primarily when sizing cooling towers and evaporative coolers. The third set of tables, to be used when sizing ventilation equipment and/or when humidity control is important to the end user, indicates the occurrences of the peak latent ambient condition … Dewpoint / Humidity Ratio and Mean Coincident Drybulb. In other words, we no longer have to guess at the peak latent design conditions. We have historical data available to use for most major cities. See Figures 4 and 5 for a glimpse of the information available in these newer tables.

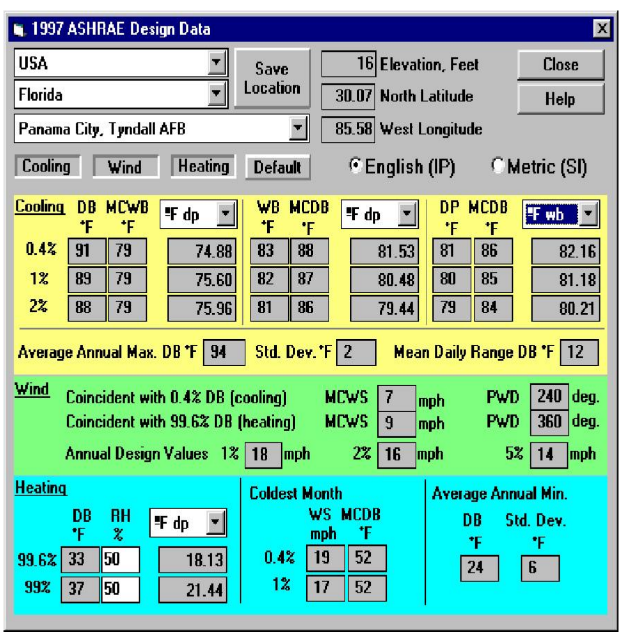

Figure 4. ASHRAE Data, Panama City, FL[3]

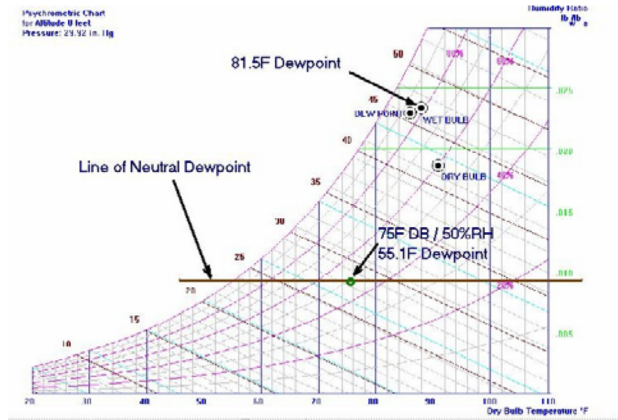

Figure 5. Design Conditions, Panama City, FL (0.4% Occurrence, ASHRAE Condition)[3]

Understanding this new weather data is the first step in accurately designing an outside air conditioning system. For example, considering the following ambient conditions for Panama City, FL, notice the three points indicating the 0.4% occurrence for the three conditions listed earlier (DB/MWB, WB/MDB and DP/HR&MDB). As shown, these three points, occurring the same number of hours per year, have very different total enthalpies and absolute moisture levels (dewpoint temperature and humidity ratios). Understandably, if only the peak sensible ambient condition were to be considered when designing the outside air unit, it would be undersized for the latent load during times of greater moisture levels in the outside air, which represents a significant number of hours during the year. See Figure 6 below.

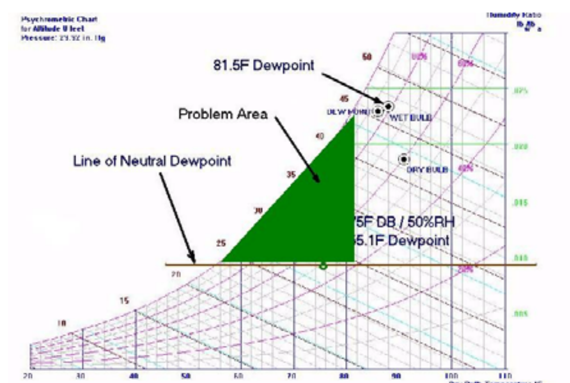

Figure 6. 0.4% ASHRAE Design Data, Panama City, FL with Problematic Area[3]

When conventional design standards and equipment are used to condition the ventilation air, the typical control device is a temperature sensor/thermostat only. When this is done, the system controls the space conditions to a temperature only. This control device is “blind” to the humidity in the space. In other words, when looking at a psychrometric chart, the space is only conditioned to the vertical line represented by the setpoint drybulb temperature (Figure 7). As seen in the figure, the systems with significant amounts of ventilation air will lead to excessive space relative humidity levels at the part load conditions.

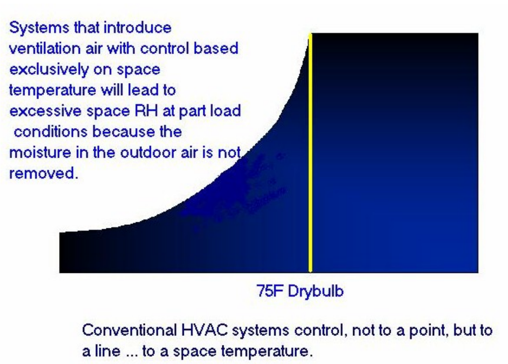

Figure 7. Conventional HVAC Control Logic

CALCULATING SUPPLY DEWPOINT

Understanding that we need to separate the ventilation air conditioning from the internal load conditioning, and that we need to also consider the differing, non-coincident ambient drybulb and dewpoint conditions in our design, we now turn our attention to the condition of the air to be supplied from outside air conditioner. Just as the internal loads are calculated to determine the necessary supply air temperatures in order to maintain the space at design setpoint, 75^0Fdb for example, we should also determine the absolute moisture level of the supply air required to satisfactorily maintain the desired space moisture level, 55^0Fdp for example. Depending on your preference, you may refer to this absolute moisture, or humidity, level in terms of “dewpoint temperature” (Fdp), or “specific humidity” (grains of moisture/pound of dry air), or even “humidity ratio” (pounds of moisture/pounds of dry air). All of these terms are absolutes and can be easily converted from one to another, but the main point is to condition the air to remove the water vapor from the air down to a predetermined level. The mathematical calculation for determining this level is shown in Equation (1) below.

$$\mathbf{W_S = W_R - L_G \ / \ (4840 \ x \ CFM)} \quad \text{Equation (1)}$$

Where:
W_S = Humidity Ratio required of the supply air
W_R = Hum. Ratio of the return air (space condition)
L_G = Latent Gain (either total, or per person)
CFM = Ventilation air requirement (either total, or per person)

Example: At what Humidity Ratio (or dewpoint) must the outside air be supplied in order to maintain the room condition of 75^0F drybulb / 50% relative humidity?

Given:
1) Latent gain of 200 Btu/hr per person
2) Ventilation rate of 20 cfm per person
3) At 75^0Fdb/50%rh, the Humidity Ratio (W_R) = 0.0093 #H_2O/#dry air

Therefore, to determine the necessary supply condition (W_S), we use Equation (1) as follows:

W_S = 0.0093 – 200/(4840 x 20) = <u>0.00723 #H_2O/#dry air</u>

The equivalent moisture level in other terms are: 48.5^0F Dewpoint and also 50.61 grains moisture/# dry air.

In other words, if the designer and/or operator are relying on the conventional DX equipment, with its typical apparatus coil dewpoint temperature of approximately 55^0F to control the moisture, chances are the building will consistently have elevated moisture levels during most of the year in geographical areas that are hot and humid.

Many operators decide after the building’s HVAC system has been commissioned to reduce the amount of ventilation air being introduced into the building. This is done not only in violation of the codes, but without considering the impact this adjustment will have on the intended design. As the following two figures will demonstrate, reducing the amount of ventilation air to the building would require the dewpoint (or humidity ratio) of the air being supplied through the outside air conditioner to be supplied at an even *lower* absolute moisture level in order to overcome the latent gain by the building occupants. For example, if only 15 cfm/person of ventilation air were delivered into the same building that was earlier calculated to require 48.5^0F dewpoint for the 20 cfm/person, air must now be supplied at 45.8^{0}F dewpoint. If the system only supplied 7.5 cfm/person of outside air, it would need to be supplied at 31.9^{0}F dewpoint. See Figures 8 and 9 for graphical representation of these requirements of the reduced ventilation rates.

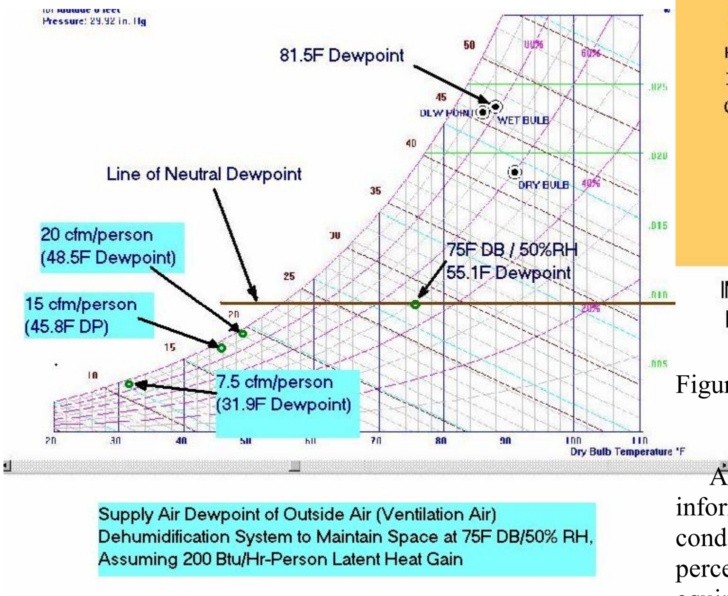

Figure 8. Psychrometric Chart View of Supply Air Dewpoint Requirements[3]

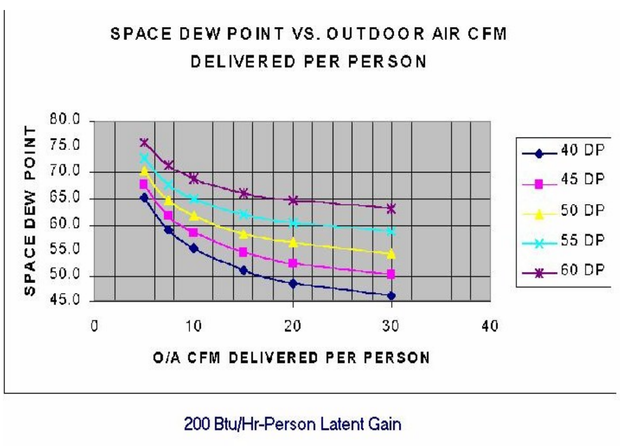

Figure 9. Supply Dewpoint vs. Ventilation Rate[3]

Not only could reducing the ventilation rate into a building likely result in moisture-related problems, but also the probability of infection from airborne infections would increase with inverse proportion of ventilation air. See Figure 10 below.

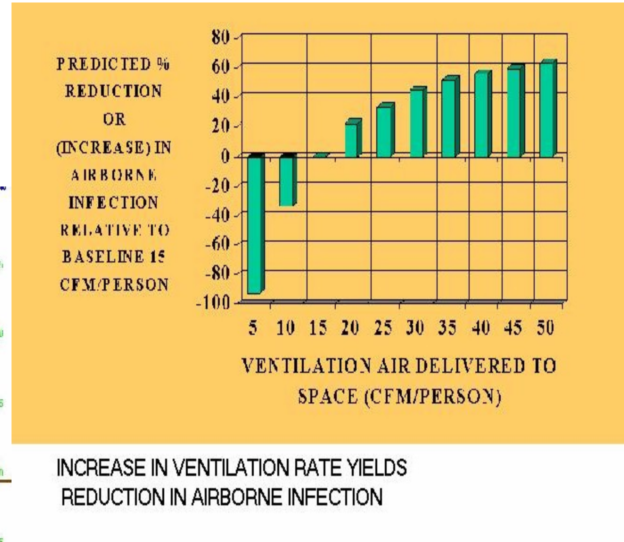

Figure 10. Increase in Ventilation Rate Yields Reduction in Airborne Infections[5]

As can be understood from the aforementioned information and calculations, selecting equipment to condition 100% outside air, or at lease a larger percentage than that required in the past, requires equipment different from the conventional packaged DX equipment atop many of our schools, office buildings and retail shops. Not only should the equipment be capable of delivering air at dewpoint temperatures lower than recognized in the past, but it must be able to deliver these lower dewpoints even at the peak dewpoint ambient design conditions, possibly even from the 81.5^{0}F dewpoint conditions as seen in Panama City, Florida. Furthermore, these OA units must be able to supply these dewpoint temperatures constantly and consistently, not just when the ambient conditions are extreme. In other words, even when the outside dewpoint temperature is fairly mild, say 58^{0}F or so, the OA unit must still be able to deliver the targeted dewpoint. If not, the resulting condition in the space could easily reach the 70% RH range when the space is 75^{0}Fdb. Considering again Figure 1, 70% RH is not within the optimum humidity range and could result in less than healthy conditions if kept at that level for any length of time. Selecting proper equipment for the conditioning of outside air requires equipment with a considerably lower sensible heat ratio (SHR) than conventional equipment, which might be on the order of 0.65 to 0.80. As seen in Figure 11, it would not be unusual for this selected piece of equipment to have a SHR of 0.345.

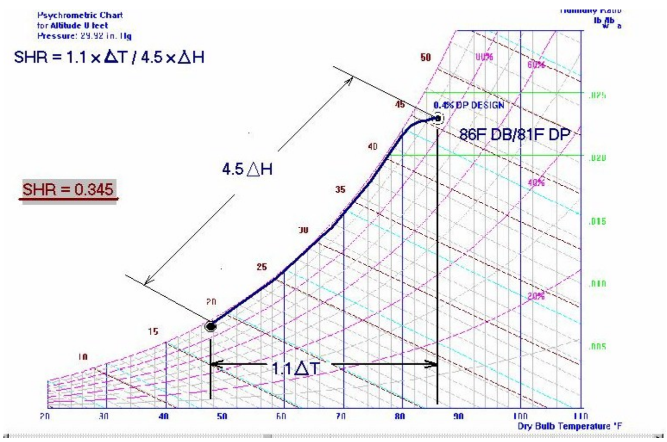

Figure 11. Calculating SHR for OA Equipment[3]

In summary, properly selecting the appropriate equipment for conditioning the ventilation air, and sizing the equipment adequately to supply the necessary dewpoint, or moisture level, our buildings can remain healthy and free of IAQ problems caused by excessive quantities of moisture in the space. With proper attention being paid during the design stages, you can ensure absolute control of space dewpoint by delivering the ASHRAE prescribed quantity of ventilation air at a mathematically determined dewpoint.

BIBLIOGRAHY

1. Theodor Sterling & Associates Study
2. NIOSH Study, ca 1997
3. Keith Dunnavant, Des Champs Laboratories, Inc.
4. ASHRAE Fundamentals Handbook, 2001 Edition, Chapter 26
5. Nardell, Edward A. et. al. – *Airborne Infection: Theoretical Limits of Protection Achievable by Building Ventilation, Feb. 19,1991*