Dehumidification Issues Of Standard 62-1989

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HVAC equipment manufacturers, specifiers and end users are only beginning to address the series of issues promulgated by the increased outside air requirements in *ANSI/ASHRAE Standard 62-1989: Ventilation for Acceptable Indoor Air Quality*.[1] These issues include:

1) A twofold to fourfold increase in outside air requirements for many commercial building applications (compared to the 1981 version of the standard) that has cascaded into building codes during the early to mid-1990s.[2]

2) The growing recognition by IAQ experts of the harmful effects of elevated indoor relative humidity (rh) on the building and occupants, and Standard 62 recommended practice that building spaces in humid climates be maintained between 30% and 60%.[3]

3) The introduction of new ASHRAE weather data identifying design dew point temperatures (peak outside air humidity conditions) in the *1997 ASHRAE Handbook—Fundamentals* for specifiers to use to size and select HVAC equipment.[4, 5]

In response to these issues, air handling units are supplying large percentages of outside air (% OA) in numerous commercial building types. The large humidity loads from outside air in non-arid climates can now be readily recognized and quantified at design conditions. To mitigate or nullify these additional weather loads, outdoor air preconditioning technologies are promoted in combination with conventional HVAC operations downstream as a means of delivering the required fresh air and controlling humidity indoors.[6, 7, 8] This article discusses the individual issues further, illustrates a simple methodology for identifying building types susceptible to humidity control problems and highlights detailed procedures for evaluating dehumidification capabilities of preconditioning equipment solutions.

Outside Air Requirements

In the commercial sector today, IAQ equates directly to outside air quantity, as prescribed by Standard 62-1989. During the early 1990s, these ventilation rates were adopted by all three major model building codes.[9, 10, 11] In turn, those revised model codes were accepted into many state and local codes by the mid-1990s. In response to the appearance of sick building syndrome in the 1980s, Standard 62-1989 prescribed a twofold to fourfold increase in outside air requirements for many more applications than those outlined in the 1981 standard. For example, non-smoking area outside air requirements changed from 5 to 10 cfm/person (2 to 5 L/s) for retail spaces, from 7 to 15 cfm/person (3 to 7 L/s) for auditoriums, from 7 to 20 cfm/person (3 to 9 L/s) for meeting rooms and from 5 to 20 cfm/person (2 to 9 L/s) for offices.

Humidity Control Requirements

Increased outside air volumes can result in periods of increased indoor humidity levels in non-arid climates. Many examples of the harmful effects on humans and buildings of elevated humidity levels have been documented.[12, 13] As a result, it was proposed that revisions to Standard 62-1989 include a requirement that building spaces in humid climates average ≤60% rh for occupied periods (≤70% rh for unoccupied periods). While this requirement is not currently advanced for public review, the 1989 Standard does include a recommendation for maintaining rh between 30% and 60%. Good practice in humid climates dictates that this requirement be met to prevent the problems that occur at higher humidity levels.

As manufacturers and specifiers react to the magnitudes of moisture present in outside air and to the threat posed to IAQ by uncontrolled indoor humidity, alternative HVAC equipment solutions to precondition outside air will be increasingly utilized to isolate and solve this problem.

Ultimately, the end users in these market segments will determine to what extent indoor humidity must be controlled by placing a value on improved humidity control and balancing it

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against the operating and first cost of the required HVAC equipment. This "economic" decision by the end user will be driven by prevailing market forces including IAQ standards, codes, practices and enforcement; perceived and experienced liability; employee health, safety and productivity policies; other occupant health and comfort issues; and building aesthetics and maintenance concerns.

Design Data for Moisture Loads

Moisture loads present in outside air are finally receiving long overdue recognition in the 1997 edition of the *ASHRAE Handbook—Fundamentals*. The 1993 Handbook (Chapter 24) contains cooling design dry bulb temperatures (1%, 2.5% and 5% summer conditions; replaced by 0.4%, 1% and 2% annual conditions in 1997).[14] The 1997 edition (Chapter 26) also contains the design dew point temperature and design humidity ratio. This design dew point has been the long overlooked "other peak cooling condition." In fact, in non-arid climates, the cooling load resulting from outside air is larger at the design dew point than at the design dry bulb as shown in *Figure 1* for three cities. (The authors have chosen to use the more familiar 2.5% summer design data, as at the time of acceptance of this article the new *1997 ASHRAE Handbook—Fundamentals* was just distributed with the less familiar annual conditions.)

Figure 1 further illustrates the breakdown of sensible (temperature reduction) and latent (moisture removal) cooling loads due to outside air. Even at design dry bulb conditions (peak temperature reduction cooling load), outside air loads are on the order of 50% sensible/50% latent (sensible heat ratio, or SHR, of 0.5). The loads become predominately latent, falling to around 0.2 SHR, at the design dew point conditions (peak moisture removal cooling load).

It is the relationship between % OA and the resulting SHR of the cooling load (for the entire building or specific air handling units in that building) that forms the premise for preconditioning OA. Conventional cooling equipment characteristically matches up well with cooling loads with SHR's of 0.75 or higher. At 0% OA, commercial buildings have cooling loads with SHRs typically approaching 0.9 but possibly as low as 0.75 (with high occupancy activities per square foot). *Figure 2* shows that as % OA increases from 0 to 100%, the design cooling load SHR decreases at a modest slope for design dry bulb conditions and at a steep slope for design dew point. As *Figure 2* illustrates, for Atlanta, it is imperative that designers consider the design dew point when evaluating cooling equipment sizing and selection, otherwise traditional design dry bulb analysis will not reveal the significant mismatches between conventional equipment performance and cooling load SHRs.

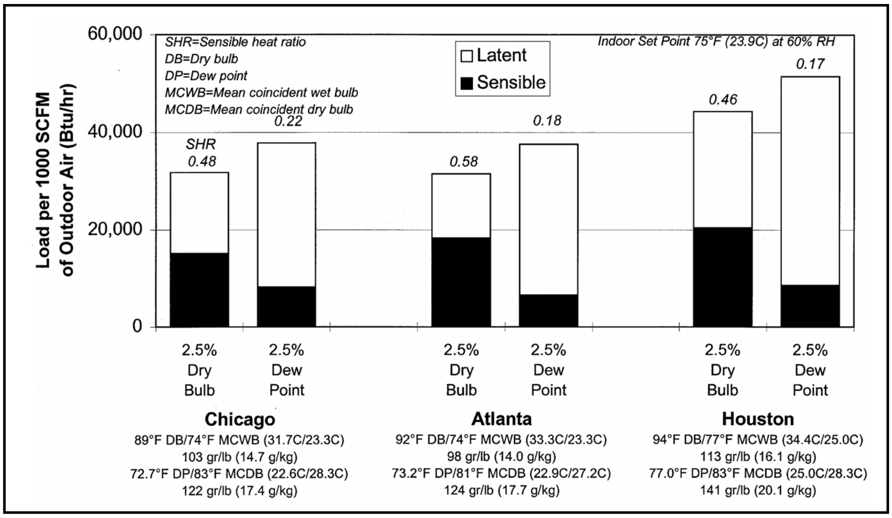

Figure 1: Outside air loads at design conditions.

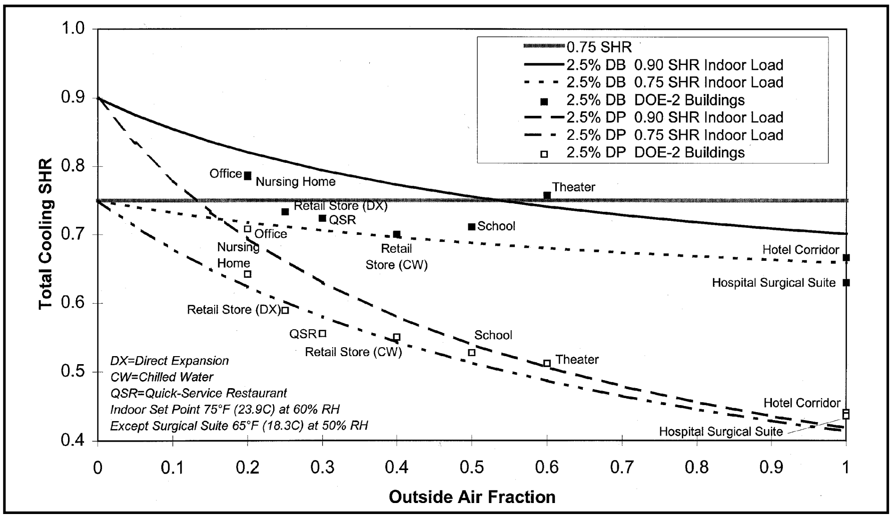

Figure 2: Total cooling SHR at design conditions for Atlanta.

These mismatches occur whenever the generic design load curves fall below the 0.75 SHR line. Under these circumstances, one can anticipate excess moisture loads that cannot be wholly removed by conventional equipment and which allow relative humidity levels to climb above 60% in the building space. Standard 62-1989 recommends that building spaces be maintained at 60% rh or lower. Higher rh levels can result in microbial activity, including mold and mildew growth, with resultant damage to building materials/furnishings and allergic reactions in humans.

A select group of commercial and institutional building types were modeled in the DOE-2.1E analysis program and the individual building points plotted in *Figure 2* for the design dry bulb and dew point conditions (points identify modeled building cooling load SHR at the Standard 62-1989 prescribed % OA for that building). Note that the modeled design conditions for each building type generally fall within the generic design load curves described earlier. Applications with the highest outside air fractions, such as hospital operating room air handlers, hotel corridor makeup air systems, theaters and schools, will be the most difficult to satisfy with conventional HVAC systems. As a result, these applications will be the first to see the use of alternative ventilation air treatment technologies. This trend is already manifesting in the marketplace. For example, manufacturers are successfully specifying alternative systems, using desiccant dehumidifiers or enthalpy exchangers, into a number of schools and hospitals throughout the southern United States.

Alternative Treatment Technologies

A wide variety of equipment configurations are available for controlling humidity. Four of the more commonly used system configurations were modeled to compare the humidity control performance and energy costs for different types of ventilation air treatment:[15]

Baseline, a standard DX reheat system sized to meet the design dry bulb temperature requirement in the space with no special consideration for humidity control.

Enhanced Reheat, a DX reheat system with increased cooling capacity to meet the additional cooling load required to overcool the supply air for humidity control plus a "free" source of reheat, such as a heat pipe, hot gas reheat, or condenser heat (which represents an ideal lower limit on operating costs, neglecting additional operating costs due to increased fan power, etc.).

Enthalpy Wheel, the enhanced reheat system (with oversized cooling capacity and "free" reheat) with a 70% effective sensible and latent heat exchanger between the relief air and outdoor air.

Desiccant, the baseline system with a desiccant wheel to remove latent heat from the outdoor air, producing hot and dry air which is then partially cooled using a sensible heat exchanger (to the relief air) with the remaining cooling provided by a DX coil (*Figure 3*).

These systems were evaluated using DOE-2.1E to determine humidity control performance and the resulting energy costs of achieving improved humidity control. *Figure 4* compares the energy costs for increasing levels of humidity control for a movie theater using Atlanta, TMY weather data. The 11,000 ft^2 (1022 m^2) theater was modeled as one theater of a multiplex served by DX rooftop units with a baseline system capacity of 75 tons (264 kW).

Figure 4 plots annual space conditioning energy cost vs. the number of occupied hours that the space relative humidity is greater than 60%. The dot represents the performance of the baseline system operating with no active humidity control—the system controls for space dry bulb set point only.

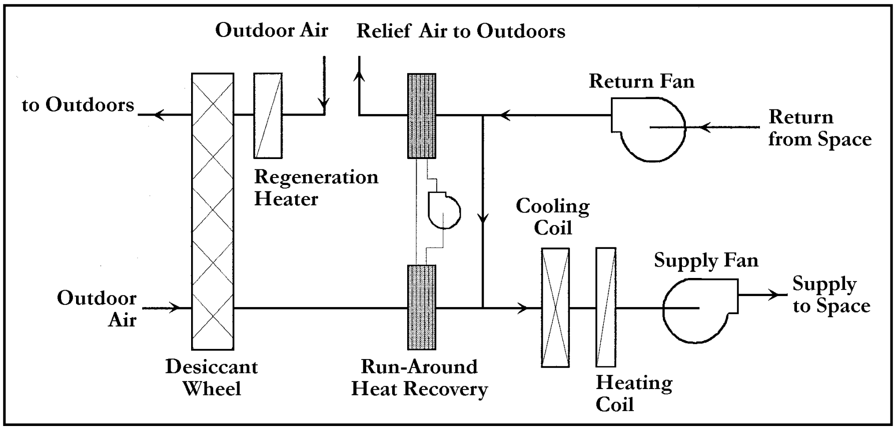

Figure 3: Desiccant system alternative.

This application would result in nearly 1800 occupied hours exceeding 60% rh. The curves show the cost of providing varying levels of humidity control with the other three systems. (No curve is plotted for the baseline system. Active humidity control with the baseline system actually increases the relative humidity in the space because the space temperature drops without adequate moisture removal capacity.) The curves should be viewed from right to left, with the initial case representing no active humidity control for the particular system alternative.

As the curve progresses to the left, successively greater levels of humidity control were modeled. Each increase in humidity control reduces the number of hours exceeding 60% rh and increases the energy cost.

The enhanced reheat and enthalpy wheel systems with DX cooling capacity sized at 200% of the baseline capacity are costly to operate and would have to be sized even larger to eliminate all hours above 60% rh. For this application, the desiccant system is able to eliminate more hours of high humidity with a significantly lower energy cost. This is accomplished, as well, with less DX tonnage than originally specified in the baseline system, offering the potential of a lower first cost than competing technologies. Note that these comparisons assume that all reheat energy is provided by condenser waste heat or some other free source. If reheat energy costs were included, the differences would be much greater.

Moisture Load Modeling Limitations

DOE-2.1E is one of the most sophisticated building energy simulation programs used by design engineers and re-

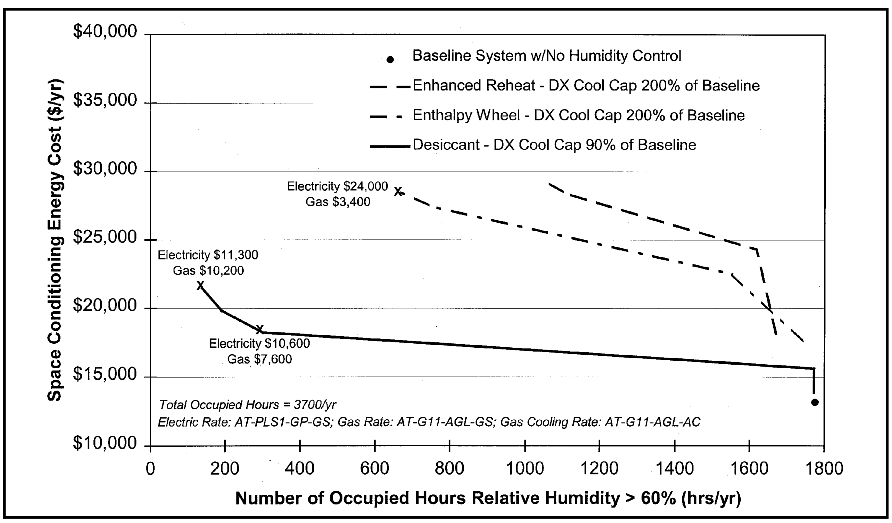

Figure 4: Space conditioning energy cost vs. humidity control for a movie theater in Atlanta.

searchers around the world. However, it has several key modeling limitations related to predicting indoor humidity levels:

• All moisture balances are instantaneous—the moisture capacitance of interior building materials and furnishings is neglected.

• Moisture from infiltration is neglected during hours when the fan system is off.

• The impact of continuous supply fan operation on latent capacity at part-load conditions is not modeled.

While most simulation programs model thermal capacitance effects that are due to the mass of the building, they neglect to consider that most interior building materials and furnishings are able to sorb and desorb significant amounts of moisture. The amount of moisture capacitance depends on the amount and type of materials within the building. Moisture capacitance tends to yield higher average humidity levels in buildings while limiting indoor humidity fluctuations.

Moisture infiltration during hours when the fan system is off can significantly increase indoor humidity when outdoor humidity levels are high. At the same time, interior building materials and furnishings sorb and store some of this moisture. When the HVAC system turns back on, the stored moisture is released and results in an increased latent load on the air-conditioning system.

In most commercial buildings, supply air fans run continuously during occupied hours to ensure proper air circulation and ventilation. This mode of fan operation, however, decreases the dehumidification performance of single-speed DX air conditioners at part-load conditions.[16] (Note: this type of problem would also be seen for constant volume chilled water systems with two-way control valves.

Furthermore, two-stage DX units would have a similar problem but to a lesser extent.) Fan operation after the compressor shuts off evaporates moisture from the wet cooling coil and drain pan back into the supply airstream. The amount of moisture evaporated depends on the physical characteristics of the evaporator coil and drain pan (fin spacing, pan slope, etc.), the thermostat cycling rate and the time constant of air conditioner latent performance at compressor start-up.[17] In addition, the constant air circulation increases the evaporator temperature when the compressor cycles off, thereby delaying dehumidification during the next compressor operating period until the evaporator temperature falls below the dew point temperature of the air.

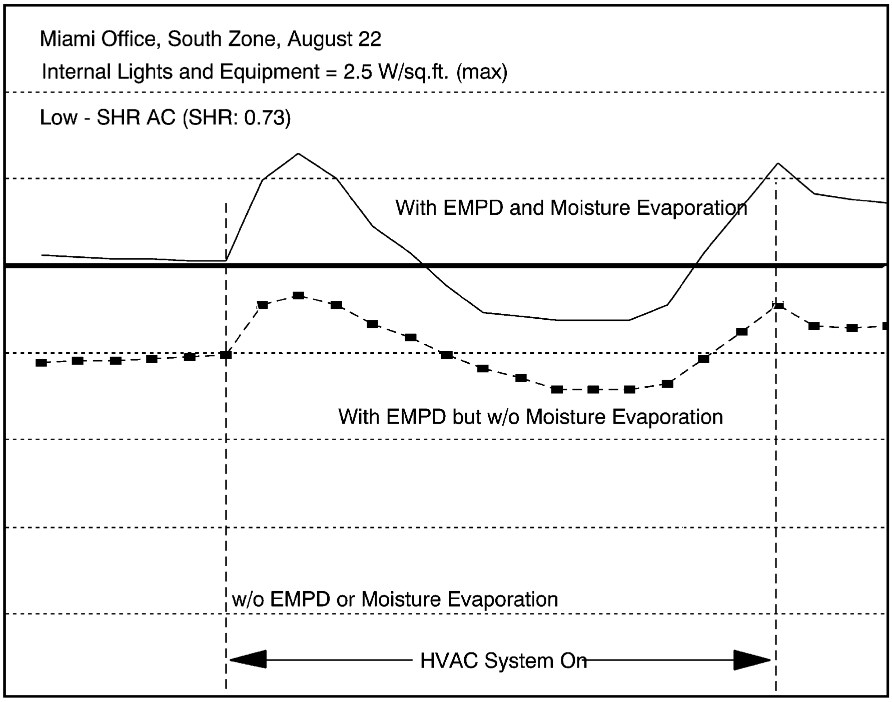

Figure 5: Impacts of moisture transport/storage in building materials and evaporation from the cooling coil on indoor humidity.

It is uncertain what impact these modeling limitations have on the indoor humidity levels predicted by DOE-2. To address this issue, work is underway to compare the DOE-2 output with results generated by FSEC 2.3, another building energy simulation program that is capable of modeling the moisture effects discussed above.[18] FSEC 2.3 is a general-purpose software package especially designed to simulate complex building science problems. This public domain software models both thermal and moisture transport and storage in typical building materials and furnishings. It also models moisture infiltration loads when the fan systems are off and moisture evaporation from the cooling coil during part-load conditions.

While the comparisons between DOE-2 and FSEC 2.3 are still in progress, a recent study by Shirey et al. compared the indoor humidity predicted by FSEC 2.3 with and without the moisture-modeling algorithms enabled. In both cases, the moisture infiltration loads were still considered when the fan systems were off.[19] *Figure 5* presents the indoor humidity profile for a small Miami office on a typical summer day with and without the use of moisture models. The office is a 4000 ft^2 (372 m^2) section of a larger single-story building. The office is partitioned into two zones, each served by a conventional direct-expansion (DX) air conditioner (SHR=0.73 at ARI rating conditions). The constant-volume air handlers were operated continuously from 6 a.m. to 9 p.m. on weekdays, and outdoor ventilation air was provided at 20 cfm (10 L/s) per person according to Standard 62-1989.

The solid line in *Figure 5* represents the indoor humidity with the moisture models enabled (with EMPD and moisture evaporation). EMPD, effective moisture penetration depth, is the moisture transport/storage model that was used.[20] The solid squares represent indoor humidity with the EMPD model enabled but the model for moisture evaporation from the cooling coil disabled. While the profiles are similar, moisture evaporation from the cooling coil increases the overall humidity level by about 5% rh for this particular day.

The effects of moisture evaporation on indoor humidity vary from hour to hour based on the run-time fraction of the compressor. The rise in rh from 6 a.m. to 8 a.m. is caused by the introduction of outdoor ventilation air that is extremely humid, but its temperature is nearly the same as indoors. Since interior lighting, equipment and people loads are also very low during these hours, the thermostatically-controlled AC compressor only operates for a small fraction of the hour and provides very little dehumidification. The solid triangles represent indoor humidity when both the EMPD and moisture evaporation models are disabled.

Note that humidity changes at a much faster rate than the other two profiles since the effect of moisture sorption/desorption from building materials is neglected. The low indoor humidity achieved while the HVAC system operates is due to two factors. Since moisture desorption from the building materials and furnishings is neglected, the latent load on the air conditioner is simply the moisture due to infiltration, outdoor ventilation air and internal generation by the occupants. In addition, the latent capacity of the air conditioner is overestimated at part-load conditions (e.g., 6 a.m. to 9 a.m.), since moisture evaporation from the cooling coil is not considered.

Conclusions

With Standard 62-1989 ventilation levels becoming applicable in more and more state and local building codes, HVAC equipment manufacturers, specifiers and end users are faced with the challenge of providing ventilation air treatment while maintaining control of indoor relative humidity. A variety of outside air preconditioning equipment options are being promoted by HVAC equipment manufacturers to address this challenge. These options include various reheat techniques, enthalpy wheels and desiccant dehumidification. Depending on the application, the initial and operating costs of these options can vary dramatically. Careful analysis is required to insure a cost-effective solution for the client.

Building energy simulation programs are currently used by HVAC design engineers to predict performance and estimate the operating costs of the various systems under consideration. Building owners evaluate the estimated costs and benefits prior to final selection. As public concern over IAQ increases, designers and building owners will demand that similar tools be developed to estimate IAQ and allow cost/benefit analyses to be performed. Since high indoor humidity can adversely affect IAQ, building simulation programs will be increasingly required to predict indoor humidity levels, including the effects of moisture capacitance and coil evaporation, for various building and HVAC system combinations.

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