# DEVELOPMENT OF A HUMID CLIMATE DEFINITION

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**ABSTRACT**

The role of humidity in indoor air quality has become of increasing concern in recent years. High indoor humidities can result in microbial growth on building surfaces, resulting in poor indoor air quality, as well as damage to the building and its contents. In addition to the IAQ impacts, high indoor humidity can cause occupant discomfort.

The public review draft of ASHRAE Standard 62-1989R included requirements for installation of dehumidification controls in buildings with mechanical cooling located in humid climates. The draft standard included a definition of humid climate: *where, during the warmest six consecutive months of a typical year, the wetbulb temperature is 19°C (67°F) or higher for 3500 hours or more, or 23°C (73°F) or higher for 1750 hours or more.* This definition is that used in the 1993 ASHRAE Handbook of Fundamentals to define the humid climate region. The only areas in the continental United States which meet these criteria are close to the Gulf coast, all of Florida, and along the Atlantic coast as far north as southern North Carolina.

climate region need to be carefully designed with regard to humidity control, it is also clear that buildings in other areas have an equal need for humidity control.

The work described in this paper examines a number of potential indicators of "humid climate" and correlates them with the prevalence of indoor humidity problems in three building types. The FSEC 2.3 energy simulation computer program (Kerestecioglu et al. 1989) was used to simulate the three building types, using weather from 10 cities in the southeastern U.S. The FSEC software was selected because it is capable of accurately modeling moisture transfer within the building space and the dehumidification performance of cooling coils at part-load conditions, and predicting resulting humidity levels. The buildings modeled were a retail store (similar to a K-Mart or Wal-Mart), a large office building, and a fast food restaurant. Existing building models were employed for this study with ventilation rates in accordance with ASHRAE Standard 62-1989. The HVAC systems used were typical for these building types, without any special humidity control measures. The selected indicators of humidity problems are the number of hours per year with space humidity above 60% RH and the number of *occupied* hours with space humidity above 60% RH.

TMY2 weather data (NREL 1995) for 10 cities was used for the annual building energy simulations. TMY2 data was also used to calculate a number of potential humid climate parameters for the same 10 cities. These included:

- the number of hours and the wetbulb-degree hours above 3 different wetbulb temperatures,
- the number of hours and grain-hours above 4 different humidity ratios, and
- the sensible, latent and total Ventilation Load Index (VLI).

The VLI is the load (latent, sensible or total) generated by bringing one cfm of outdoor air to space neutral conditions over the course of one year (Harriman, et al. 1997).

indoor humidity problems was analyzed and compared. Implications of using the selected parameters to define a humid climate will be discussed.

## INTRODUCTION

Buildings in humid climates can encounter indoor air quality problems related to high indoor humidity. High indoor humidities can result in microbial growth on building surfaces, resulting in poor indoor air quality, as well as damage to the building and its contents. In addition to the IAQ impacts, high indoor humidity can cause occupant discomfort.

Buildings designed in humid climates should include specific measures to avoid high indoor humidities. Frequently, the HVAC systems in these buildings will include equipment and controls to provide additional dehumidification capabilities. All building designs for humid climates should consider the indoor humidity levels which will result from various design choices.

One question which naturally follows from the above, however, is "What is a humid climate?" While it is obvious that Miami or Houston is in a humid climate, it becomes much less obvious for more northern locations. Various definitions of humid climate are available.

ASHRAE provided definitions of humid climate and fringe climate in the 1993 Handbook of Fundamentals (ASHRAE). "A humid climate can be defined as one in which one or both of the following conditions occur: 1. A 67°F or higher wet-bulb temperature for 3500 h or more during the warmest 6 consecutive months of the year. 2. A 73°F or higher wet-bulb temperature for 1750 h or more during the warmest 6 consecutive months of the year." Fringe climate is defined similarly, except the hours are reduced to 3000 and 1500 for the two conditions.

The latest Handbook of Fundamentals (ASHRAE 1997) no longer uses these definitions, but defines a warm, humid cooling climate which is the same as the 1993 fringe climate definition.

The warm, humid cooling climate defined by ASHRAE covers all of Florida and Louisiana, and the areas near the Gulf coast of Texas, Mississippi and Alabama, as well as the Atlantic coast as far north as southeast North Carolina. Figure 1 shows the areas of the continental U. S. covered by the ASHRAE definitions.

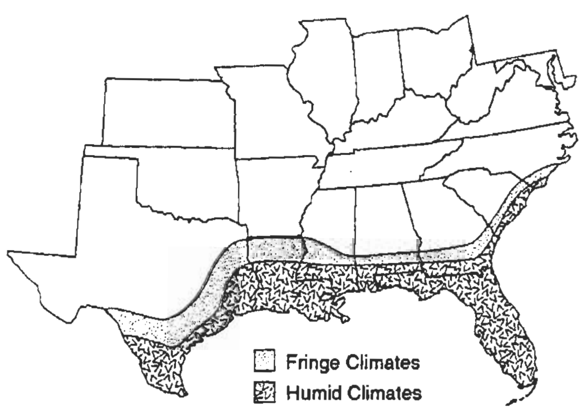


**Figure 1. Areas of ASHRAE humid climate and fringe climate.**

The design firm of CH2M HILL published a manual on designing for hot, humid climates (Odom and DuBose 1996) which provides an alternate humid climate definition:, "A humid climate is defined as one where the average monthly latent load of outside air meets or exceeds the average monthly sensible load for any month during the cooling season." A map is provided, see Figure 2, which shows this definition to cover all of North and South Carolina, Georgia, Alabama, Mississippi, Louisiana, and Florida, the southeastern third of Arkansas and the southeastern half of Texas.

The definition of humid climate is significant in that it alerts designers that they should carefully consider the dehumidification performance of their system designs for buildings constructed in this humid environment. Potentially, this consideration could be required by code or standard. For example, the public review draft of ASHRAE Standard 62-1989R included requirements for installation of dehumidification controls in buildings with mechanical cooling located in humid climates.

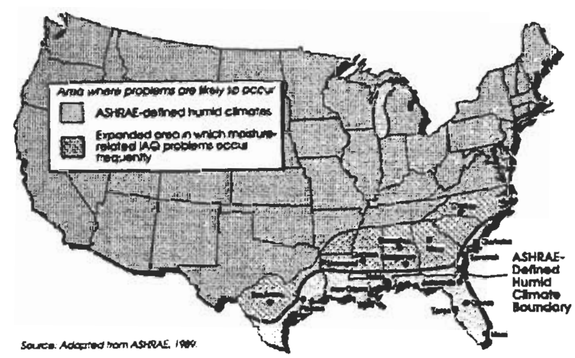


**Figure 2. Alternate humid climate area covered by the definition by Odom and DuBose of CH2M HILL.**

## METHODOLOGY

The root question we are addressing is, "Under what climatic conditions do HVAC systems require consideration of specialized humidity control designs?" To determine this, we analyzed several common building types operating in a number of locations in the southeastern U.S., to determine the degree of indoor humidity problems they experience when standard HVAC systems are used. To extend these results to additional locations, we then examined a number of weather parameters and how well each correlated to the degree of humidity problems experienced.

Three building types were investigated: a retail store (similar to a K-Mart or Wal-Mart), a large office building (5 stories, 100,000 ft² [9,290 m²]), and a fast food restaurant. The performance of these buildings was modeled using the FSEC 2.3 building simulation program. The building models were previously developed for other projects. The only modifications were to use "standard" HVAC equipment (described below) and ventilation rates which comply with ASHRAE Standard 62-89 (ASHRAE 1989).

Operation of each building was simulated in 10 cities: Miami; Shreveport, LA; Montgomery, AL; Columbia, SC; Richmond, VA; Waco, TX; Little Rock, AR; Nashville, TN; Tulsa, OK; and St. Louis. TMY2 weather data was used for each city (NREL 1995).

For each of two zones in each building, the relative humidity was recorded for each hour of the year. The number of hours with indoor humidity above 60% was summed and used as the indicator of humidity problems. 60% was selected as the threshold for humidity problems based on recommendations in ASHRAE Standard 62-89.

A computer program developed by the Gas Research Institute, Binmaker (GRI 1997), was then used to develop a number of weather parameters for comparison to the simulation results. Binmaker uses TMY2 weather data files, and computes bin data according to the user's specifications. The resulting bin data was then exported to a spreadsheet program for additional calculations. 31 parameters were computed for each city. These were:

1) number of hours with wet bulb temperature above 64°F (18°C), 67°F (19°C) and 73°F (23°C);
2) wetbulb degree-hrs above 64°F (18°C), 67°F (19°C) and 73°F (23°C);
3) number of hours with humidity ratios above 55 grains/pound of dry air (7.9 g/kg), 60 gr/lb (8.6 g/kg), 65 gr/lb (9.3 g/kg) and 78 gr/lb (11.1 g/kg);
4) grain-hours above 55 gr/lb (7.9 g/kg), 60 gr/lb (8.6 g/kg), 65 gr/lb (9.3 g/kg) and 78 gr/lb (11.1 g/kg);
5) latent ventilation load index;
6) sensible ventilation load index; and
7) total ventilation load index.

The 14 parameters listed in items 1 through 4 were calculated in two ways, for both total hours per year and for the hottest 6 full months of the year. The ventilation load index (Harriman, et al. 1997) is the cooling load (latent, sensible or total) generated by bringing one cfm of outdoor air to space neutral conditions over the course of one year. The ventilation load index (VLI) has units of ton-hrs/cfm (kWh/L/sec). VLIs were calculated using nominal space conditions of 75°F (24°C), 50% RH.

Based on graphical analysis (see Figure 3), it appears that the relationship between outdoor humidity and number of hours with indoor humidity above 60% is linear. To compare the ability of each of the 31 weather parameters to predict humidity problems, correlation coefficients were calculated for each. Correlation coefficients were calculated for each weather parameter against hours with humidity above 60% for each of 2 zones in each building.

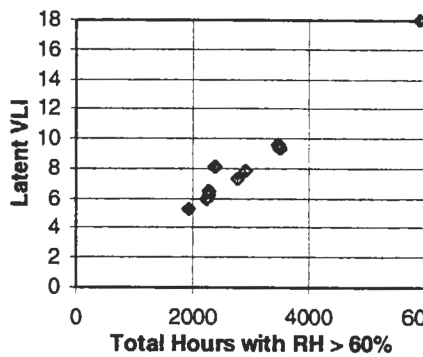


**Figure 3. Example of the linear relationship between outdoor humidity and indoor hours above 60% RH.**

**BUILDING SIMULATION SOFTWARE**

Since the emphasis of this study was on predicting indoor humidity levels and correlating occurrences of high indoor humidity to ambient weather conditions, the computer simulation software FSEC 2.3 was used to simulate the three building types and the performance of their respective HVAC equipment (Kerestecioglu et al. 1989). FSEC 2.3 is a general-purpose software package especially designed to simulate complex building science problems. This public domain software was selected due to it's unique ability to model both thermal and moisture transport and storage in typical building materials and furnishings, a key requirement when simulating building/air conditioner interactions in humid climates. It also models moisture infiltration loads when the fan systems are off and the effects of continuous fan operation on the latent capacity of air conditioners. These capabilities allow for the prediction and evaluation of indoor humidity levels for each simulation time step (hourly for this study).

While most simulation programs model thermal capacitance effects 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. For this study, moisture storage in building materials and furnishings was modeled with the effective moisture penetration depth (EMPD) method described by Kerestecioglu et al. (1990). The EMPD approach is based on the theory that sorption-desorption of moisture takes place primarily at the surface of an absorbing material. This approach is computationally more efficient than detailed numerical methods and is more rational than the lumped-air methods used by some simulation models.

FSEC 2.3 software also accounts for moisture infiltration when the fan systems are off. This moisture load 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 (Khattar et al. 1987). Supply fan operation after the compressor shuts off evaporates moisture from the wet cooling coil and drain pan back into the supply air stream. 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. A model for predicting the latent capacity of air conditioners at part-load conditions with constant fan operation was utilized for the simulations of rooftop packaged equipment for the retail store and fast-food restaurant (Henderson and Rengarajan 1996).

## MODELED BUILDING CHARACTERISTICS

Three building types were modeled for this study: a large office building, a retail store, and a fast-food restaurant. The building models were taken from a previous study of the indoor humidity, energy and economic impacts of ASHRAE Standard 62-1989 on seven building types in Florida's hot and humid climate (Chasar et al. 1997, Rengarajan et al. 1997, Rengarajan et al. 1998). A description of the building, HVAC system, internal loads and schedules for each of the three building types used for the present study are summarized in Table 1.

The selected HVAC systems were typical for these building types without any special humidity control measures. Economizer control was not used on any of the buildings. For the fast-food restaurant and retail store, rooftop packaged DX equipment with electric strip heat was modeled under conventional thermostat control. The supply air flow for the retail store DX equipment was set at 400 cfm/ton (53.7 L/kW·s) according to conventional practice. The supply flow rate for the restaurant was reduced to 300 cfm/ton (40.3 L/kW·s) because of the high outdoor air requirements and the resulting need for improved dehumidification for this application.

A chilled water, variable-air-volume (VAV) system was modeled for the large office building. Two separate VAV air handlers were used to condition each floor of the office building, one for the core and one for the perimeter zones. The air handlers were operated to provide 55°F (12.8°C) supply air, although supply air temperature was reset based on outdoor air temperature (Figure 4). The design supply air temperature for the perimeter zone air handlers was 55°F (12.8°C), but was 60°F (15.6°C) for the core zone air handlers. This was necessary so that adequate cooling would be available when the supply air temperature was reset (Figure 4), because the cooling load in the core was primarily due to internal loads, and therefore independent of outdoor temperature. Cooling capacity control for the office perimeter zones was provided by parallel fan-powered mixing boxes.

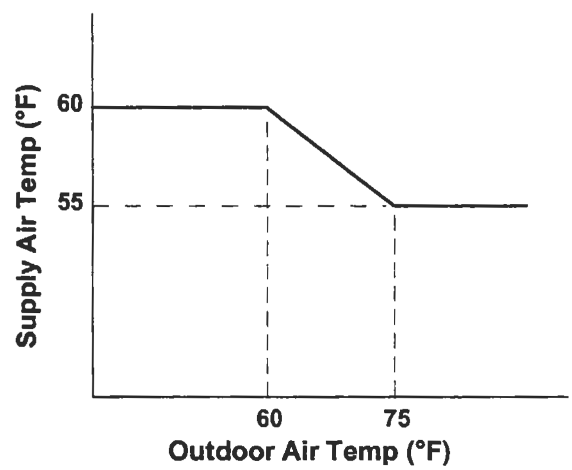


**Figure 4. Supply air temperature reset control used by the office building air handlers.**

**Table 1 Description of Building Characteristics.**

| Prototype Characteristics | Large Office | Retail Store | | Fast Food Restaurant | |
|---|---|---|---|---|---|
| | | Showroom | Stockroom | Dining Rm. | Kitchen |
| **Physical Characteristics:** | | | | | |
| Floor Area, ft² (m²) | 100,000 (9,290) | 72,650 (6,750) | 14,220 (1,321) | 1,600 (149) | 1,600 (149) |
| Number of Stories | 5 | 1 | | 1 | |
| Height per Story , ft (m) | 14 (4.3) | 17 (5.2) | | 10 (3.0) | |
| Wall U-value, Btu/ft²·h·°F (W/m²·°C) | 0.08 (0.45) | 0.13 (0.74) | | 0.08 (0.45) | |
| Roof U-value, Btu/ft²·h·°F (W/m²·°C) | 0.07 (0.40) | 0.07 (0.40) | | 0.07 (0.40) | |
| **Glazing Characteristics:** | | | | | |
| Percent Glass in Exterior Walls (%, avg.) | 25 (equally distributed) | 2.7 (south entrance only) | | 21 | |
| Glass Type | Double pane | Single Pane | | Single Pane | |
| U-value, Btu/ft²·h·°F (W/m²·°C) | 0.60 (3.41) | 1.06 (6.02) | | 1.06 (6.02) | |
| **Operating Conditions:** | | | | | |
| Cooling Set Point/Setup, °F (°C) | 75/85 (24/29) | 74/80 (23/27) | | 75/off (24) | 80/off (27) |
| Heating Set Point/Setback, °F (°C) | 70/55 (21/13) | 70/60 (21/16) | | 70/off (21) | 70/off (21) |
| **HVAC Equipment:** | | | | | |
| Air Handling System Type | Variable Air Volume | Constant Volume | | Constant Volume | |
| Cooling Plant Type | Air-Cooled Recip. Chillers | Rooftop Packaged DX | | Rooftop Packaged DX | |
| Heating Plant Type | Electric Resistance | Electric Resistance | | Electric Resistance | |
| Outside Air | 20 cfm (10 L/s) per person | 0.3 cfm/ft² (1.5 L/s/m²) | 0.15 cfm/ft² (0.75 L/s/m²) | 20 cfm/person* (9.4 L/s/per) | 900 cfm** (425 L/s) |
| **Internal Loads (peak):** | | | | | |
| Occupants, ft²/person (m²/person) | 150 (13.9) | 120 (11.1) | 710 (66.0) | 14.5 (1.4) | 133 (12.4) |
| Lighting, Watts/ft² (Watts/m²) | 1.25 (13.5) | 1.94 (20.9) | 1.28 (13.8) | 1.3 (14.0) | 1.4 (15.1) |
| Equipment/Misc, Watts/ft² (Watts/m²) | 1.25 (13.5) | 0.5 (5.4) | 0.2 (2.2) | 0.0 (0.0) | 5.6 (60.3) |
| Schedule | 7 a.m. - 5 p.m. weekdays, with some evening work, closed weekends | Daytime and evening operations 7 days per week, reduced hours on Sunday | | 5 a.m. to 11 p.m. weekdays, extended to 12 midnight on weekends | |

Notes: * - Outside air based on average occupancy for the space over the duration of system operation (per ASHRAE Std. 62-89, Section 6.1.3.4).
** - Makeup air for hood exhaust required kitchen ventilation which exceeded ASHRAE 62-89 minimum of 15 cfm (8 L/s) per person.

The outdoor ventilation rates for the modeled buildings were established in accordance with ASHRAE Standard 62-1989 (ASHRAE 1989). For the large office, the outdoor air quantity delivered was based on the design occupancy and the prescribed minimum ventilation rate of 20 cfm (9.4 L/s) per person. The ventilation air was continuously provided to each air handler during the hours of HVAC system operation regardless of the cooling or heating load to be met. In a similar fashion, outdoor ventilation for the retail store showroom and stockroom was continuously provided at 0.3 cfm/ft$^2$ (1.5 L/s·m$^2$) and 0.15 cfm/ft$^2$ (0.75 L/s·m$^2$), respectively, during system operation.

Determination of outdoor ventilation for the fast-food restaurant was slightly more complicated. ASHRAE Standard 62-89 specifies a minimum of 20 cfm (9.4 L/s) per person of outdoor ventilation air for fast food dining areas. Due to the variable nature of dining room occupancy, Section 6.1.3.4 of Standard 62-89 was applied to set the outdoor air quantity based on the average occupancy during system operation rather than the design occupancy. Since the average occupancy was relatively small compared to the peak (20% weekdays, 40% weekends), the average occupancy was set at one-half of the maximum as stipulated in Section 6.1.3.4.

ASHRAE Standard 62-89 sets a minimum outdoor air requirement of 15 cfm/person (8 L/s/person) for commercial kitchens. The standard also notes that makeup air for hood exhaust may require more ventilating air. In fact, Standard 62-89 requires that the combination of outdoor air and transfer air of acceptable quality from adjacent spaces must provide an exhaust rate of not less than 1.5 cfm/ft$^2$ (7.5 L/s·m$^2$). For the restaurant kitchen modeled for this study, we assumed that 30 percent of the exhaust requirement was provided by transfer air from the dining room. One half of the remaining exhaust requirement was provided by make-up air introduced locally at the hood (i.e., it did not thermally impact the conditioned space). The other half of the remaining exhaust requirement, plus some additional outdoor air for positive space pressurization, was furnished as outdoor air through the rooftop packaged air conditioner.

## SIMULATION RESULTS

The focus of the simulations was on determining humidity levels in the space. For each of the buildings, humidity level data were collected for two zones. The retail store and fast food restaurant only had two zones each. For the office building, the ground floor core and west perimeter zone of a typical middle floor were monitored. For each hour of the simulation (8,760 hours), the space humidity was recorded.

Table 2 shows the number of hours per year that the relative humidity in each space exceeded 60%. As would be expected, the number of hours generally is higher for cities which are located more to the southeast. The perimeter zone of the large office was an exception to this trend.

While the number of hours above 60% RH is quite large in many cases, it is also of interest to know how much above 60% the humidity was.

Table 3 shows the average percemt RH amount by which space humidity exceeded 60% for all hours with humidity above 60% RH. There are no obvious trends in this data. The amount above 60% RH is fairly constant across all cities for a given zone. The increase ranges from about 2% in the office building to almost 9% in the restaurant dining zone.

**Table 2. The number of hours per year when space humidity was above 60% RH.**

| | Fast-food Restaurant | | Large Office | | Retail Store | |
|---|---|---|---|---|---|---|
| | Dining | Kitchen | Core | Perimeter | Showroom | Stockroom |
| Miami | 4,782 | 2,674 | 3,722 | 119 | 5,891 | 6,050 |
| Montgomery | 3,630 | 2,003 | 2,383 | 611 | 3,520 | 3,141 |
| Shreveport | 3,622 | 1,958 | 2,495 | 705 | 3,451 | 3,507 |
| Columbia | 3,194 | 1,731 | 2,041 | 477 | 2,906 | 2,964 |
| Little Rock | 3,009 | 1,589 | 2,033 | 348 | 2,762 | 2,780 |
| Nashville | 2,799 | 1,565 | 1,795 | 286 | 2,283 | 2,512 |
| Richmond | 2,646 | 1,578 | 1,659 | 264 | 2,262 | 2,087 |
| Tulsa | 2,512 | 1,109 | 1,680 | 455 | 2,281 | 2,191 |
| St. Louis | 2,427 | 1,366 | 1,730 | 376 | 1,944 | 2,011 |
| Waco | 2,379 | 861 | 1,422 | 355 | 2,399 | 2,059 |

Table 3. The average percent RH above 60% for those hours over 60% RH.

| | Fast-food Restaurant | | Large Office | | Retail Store | |
|---|---|---|---|---|---|---|
| | Dining | Kitchen | Core | Perimeter | Showroom | Stockroom |
| Miami | 5.8% | 3.4% | 1.9% | 1.5% | 5.0% | 4.6% |
| Montgomery | 7.2% | 3.9% | 2.0% | 2.7% | 5.4% | 4.6% |
| Shreveport | 7.7% | 4.4% | 2.0% | 2.5% | 5.3% | 5.4% |
| Columbia | 7.9% | 4.4% | 2.0% | 2.5% | 4.9% | 4.9% |
| Little Rock | 7.5% | 4.3% | 2.1% | 2.7% | 4.4% | 4.8% |
| Nashville | 8.5% | 5.0% | 2.3% | 2.1% | 5.0% | 5.4% |
| Richmond | 8.9% | 5.2% | 2.3% | 2.1% | 5.6% | 5.2% |
| Tulsa | 6.4% | 3.5% | 1.9% | 3.3% | 4.2% | 4.2% |
| St. Louis | 8.4% | 5.0% | 2.7% | 2.7% | 5.3% | 5.8% |
| Waco | 6.0% | 3.9% | 1.8% | 2.3% | 4.1% | 4.7% |

Table 4. The number of occupied hours per year when space humidity was above 60% RH.

| | Fast-food Restaurant | | Large Office | | Retail Store | |
|---|---|---|---|---|---|---|
| | Dining | Kitchen | Core | Perimeter | Showroom | Stockroom |
| Miami | 3248 | 1575 | 1614 | 86 | 3330 | 3499 |
| Montgomery | 2564 | 1291 | 1160 | 281 | 1966 | 1707 |
| Shreveport | 2541 | 1255 | 1131 | 260 | 1886 | 1926 |
| Columbia | 2172 | 1112 | 848 | 190 | 1553 | 1582 |
| Little Rock | 2094 | 1022 | 884 | 144 | 1536 | 1499 |
| Nashville | 1948 | 1003 | 748 | 123 | 1285 | 1365 |
| Richmond | 1848 | 1027 | 677 | 129 | 1223 | 1095 |
| Tulsa | 1758 | 716 | 780 | 224 | 1353 | 1265 |
| St. Louis | 1695 | 944 | 696 | 181 | 1093 | 1075 |
| Waco | 1660 | 548 | 770 | 184 | 1530 | 1234 |

Table 5. The percentage of occupied hours per year when space humidity was above 60% RH.

| | Fast-food Restaurant | | Large Office | | Retail Store | |
|---|---|---|---|---|---|---|
| | Dining | Kitchen | Core | Perimeter | Showroom | Stockroom |
| Miami | 49% | 24% | 44% | 2% | 68% | 71% |
| Montgomery | 38% | 19% | 32% | 8% | 40% | 35% |
| Shreveport | 38% | 19% | 31% | 7% | 38% | 39% |
| Columbia | 33% | 17% | 23% | 5% | 32% | 32% |
| Little Rock | 31% | 15% | 24% | 4% | 31% | 31% |
| Nashville | 29% | 15% | 20% | 3% | 26% | 28% |
| Richmond | 28% | 15% | 19% | 4% | 25% | 22% |
| Tulsa | 26% | 11% | 21% | 6% | 28% | 26% |
| St. Louis | 25% | 14% | 19% | 5% | 22% | 22% |
| Waco | 25% | 8% | 21% | 5% | 31% | 25% |

Table 4 is similar to Table 2, except the hours shown are only from occupied periods of the year. Table 5 shows the same data as Table 4, except that the hours per year are converted into percent of occupied hours when the humidity is above 60%.

The number of hours per year with high indoor humidity is striking. For the zones selected, except the office building perimeter zone, humidity is high for 2,674 to 6,050 hours per year in Miami, and for 861 to 3,630 hours per year in the other cites. This is 30% to 69% of all hours in Miami, and 10% to 41% in the other cities.

During occupied periods, the story is similar. High humidity occurs 548 to 2,564 hours per year, excluding Miami and the office perimeter zone. With total occupied hours per year of 6,674 for the restaurant, 3,654 for the office, and 4,902 for the retail store, high humidity occurred during as much as 40% of the occupied hours, excluding Miami.

The hours above 60% RH reflect the inability of the conventional HVAC systems to provide adequate dehumidification. Outdoor ventilation requirements for the fast-food restaurant and retail store are quite high, and conventional rooftop packaged equipment does not provide sufficient dehumidification to offset the resulting large latent load. The ventilation requirements for the large office are lower, and the high humidity hours are significantly less than for the other two building types. Based on a previous simulation study of the Miami large office (Rengarajan et al. 1997), a large fraction of the high RH hours are caused by the supply temperature reset strategy. This strategy, encouraged for energy-savings when combined with chilled water temperature reset, tends to promote high indoor humidity in humid climates. Simply deleting the supply temperature reset strategy would significantly lower indoor humidity levels. The impacts on energy use may be small since lower supply fan energy use would offset some of the increase in chiller energy use.

The high percentage of time when humidity is high indicates that there are likely to be discomfort problems in these buildings, and microbial growth problems are possible. It is equally clear that the systems used in these buildings should be carefully designed to provide adequate dehumidification.

## WEATHER PARAMETER CORRELATIONS

As was mentioned earlier, 31 weather parameters were calculated for each of the 10 cities. For each set of humidity results, i.e. for each building zone with recorded humidity data, correlation coefficients, *r*, between the hours over 60% and each of the weather parameters were computed.

The correlation coefficient is the ratio of the standard deviation of the x data to the standard deviation of the y data, multiplied by the slope of a linear regression. The *r* value varies from 1 to -1, with a value close to 1 indicating that when one value is high, it is likely that the other will also be high. When *r* is close to -1, it is likely that when one value is high, the other will be low. A value close to zero indicates a lack of linear association between the two variables, i.e., the value of one variable does not allow prediction of the value of the other.

In general, correlations were fairly good. For five of the six building zones, *r* values ranged from 0.806 to 0.925. Two sets of humidity data were collected: for all hours and for occupied hours only. The *r* values for these two sets were very similar, with the range for the occupied hours being somewhat smaller: 0.819 to 0.921.

There were two exceptions to the generally good correlations. First, the *r* values for the west perimeter zone of the office building ranged from -0.318 to 0.073. These indicate that the indoor humidity is not significantly affected by the weather. This was due to the idiosyncracies of the HVAC system controls, with all four perimeter zones on a floor served by a single airhandler. Second, the Sensible VLI had low correlations for all zones, ranging from 0.196 to 0.779, excluding the office perimeter zone. This is perhaps not surprising because the sensible VLI was developed to be independent of latent loads.

To try and identify the best weather parameter to use in developing a humid climate definition, the correlation coefficients for five building zones (excluding the office perimeter zone) were averaged for total and for occupied hours. Table 6 lists the weather parameters which had the 6 highest average correlation coefficients and the associated average *r* values.

Table 7 lists the values of the first six weather parameters listed in Table 6 for each of the 10 cities where buildings were simulated. For these cities, excluding Miami, the values vary by about a factor of 2. Figure 5 shows a map of the southeastern U. S. with bands of Latent VLI values plotted. It is interesting to compare this map to the maps in Figures 1 and 2. This type of map can be developed for other parameters, but it is expected that they would have a similar appearance.

**Table 6. The weather parameters which had the highest average correlation coefficients**

| | Average Correlation Coefficient | |
|---|---|---|
| **Weather Parameter** | **Total** | **Occupied** |
| GrHrs>78 - 6 Month | 0.925 | 0.921 |
| WB°-Hrs>73 - 6 Month | 0.924 | 0.916 |
| WB°-Hrs>73 - Annual | 0.924 | 0.916 |
| GrHrs>78 - Annual | 0.920 | 0.912 |
| Latent VLI | 0.917 | 0.910 |
| GrHrs>65 - 6 Month | 0.914 | 0.912 |

Notes: 1) "GrHrs>78" means the grain-hours above 78 gr/lb, for those hours with humidity ratio greater than 78, etc.
2) "6 Month" indicates that the value is for the 6 warmest months of the year.
3) "Annual" indicates that the value is for the entire year.

**Table 7. Values of selected weather parameters for 10 southeastern U. S. cities.**

| | Thousands of Grain-Hours above | | | Wetbulb Degree-Hours above | | Latent VLI |
|---|---|---|---|---|---|---|
| | 78 gr/lb | | 65 gr/lb | 78°F Wetbulb | | |
| | 6 hottest mo. | Whole year | 6 hottest mo. | 6 hottest mo. | Whole year | Whole year |
| **Miami** | 174 | 222 | 231 | 8,096 | 8,210 | 18.0 |
| **Shreveport** | 105 | 114 | 153 | 3,570 | 3,574 | 9.7 |
| **Montgomery** | 108 | 112 | 156 | 3,434 | 3,434 | 9.4 |
| **Waco** | 86 | 93 | 132 | 2,044 | 2,048 | 8.2 |
| **Columbia** | 85 | 89 | 129 | 1,900 | 1,900 | 7.8 |
| **Little Rock** | 80 | 82 | 123 | 2,468 | 2,468 | 7.3 |
| **Tulsa** | 75 | 76 | 117 | 2,164 | 2,164 | 6.5 |
| **Nashville** | 70 | 72 | 109 | 1,602 | 1,602 | 6.2 |
| **Richmond** | 66 | 66 | 102 | 1,082 | 1,082 | 5.9 |
| **St. Louis** | 58 | 59 | 94 | 1,512 | 1,512 | 5.3 |

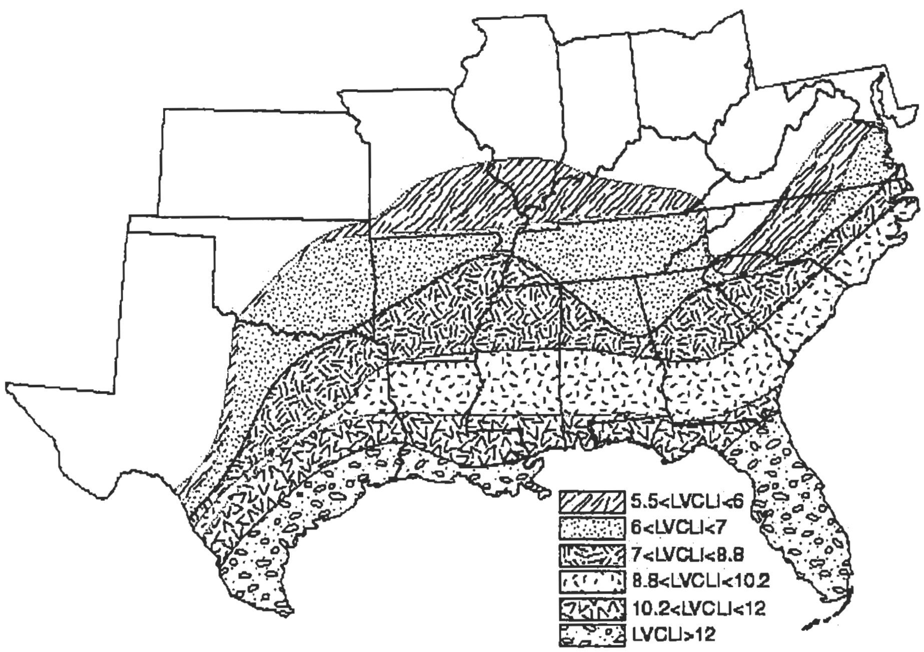


**Figure 5. Zones of constant latent ventilation load index.**

## CONCLUSIONS AND RECOMMENDATIONS

We have shown that buildings in the southeastern U.S., designed without specialized humidity control capabilities, can result in high indoor humidity levels. This is true for several common types of buildings, which have very different characteristics.

The level of humidity problems, specifically the number of hours per year with indoor humidity above 60% RH, is related to the outdoor humidity levels. Several specific weather parameters seem to be particularly useful in predicting indoor humidity problems. These include:

- Grain-hours above 78 gr/lb for the 6 hottest months of the year.
- Wetbulb degree-hours above 73°F wetbulb for the 6 hottest months of the year.
- Annual wetbulb degree-hours above 73°F wetbulb.
- Annual grain-hours above 78 gr/lb.
- Latent ventilation load index on a 75°F, 50% RH basis.
- Grain-hours above 65 gr/lb for the 6 hottest months of the year.

The indoor humidity levels encountered in the simulations, shown in Tables 2 and 4, make it difficult to draw a boundary which says that one city is not in a humid climate while the others are. Based on the simulation results, it appears that careful attention to dehumidification capabilities should be paid by building designers in all of these cities, especially buildings with high ventilation rates, such as restaurants and retail stores. By the criteria set earlier, "Under what climatic conditions do HVAC systems require consideration of specialized humidity control designs?" all of these cities would be included in the humid climate.

Based on this, we conclude that an appropriate definition of humid climate might be: "A humid climate is one which, during the hottest 6 months of the year, the total grain-hours above 78 gr/lb exceeds 55,000." Analysis of additional cities, however, should be done to identify where the humid climate ends. This would then refine the number of grain-hours above 78 gr/lb that would be used in the humid climate definition.

The definition of humid climate must be matched to its application. The definition above is intended for use in alerting the building and system designer that dehumidification should be addressed in the building's design. For other uses, such as mandating specified in a code or standard, other definitions may be more appropriate.

An additional conclusion is that use of "grain-hours above 78 gr/lb during the hottest 6 months of the year" is an improved basis for any future definition of humid climate. Use of grain-hours, instead of simply hours above a specified humidity level, provides extra weight to those hours with especially high humidity. This change appears to provide a better correlation to occurrences of high indoor humidity.

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