

NEW PEAK MOISTURE DESIGN DATA IN THE 1997 ASHRAE HANDBOOK OF FUNDAMENTALS

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ABSTRACT

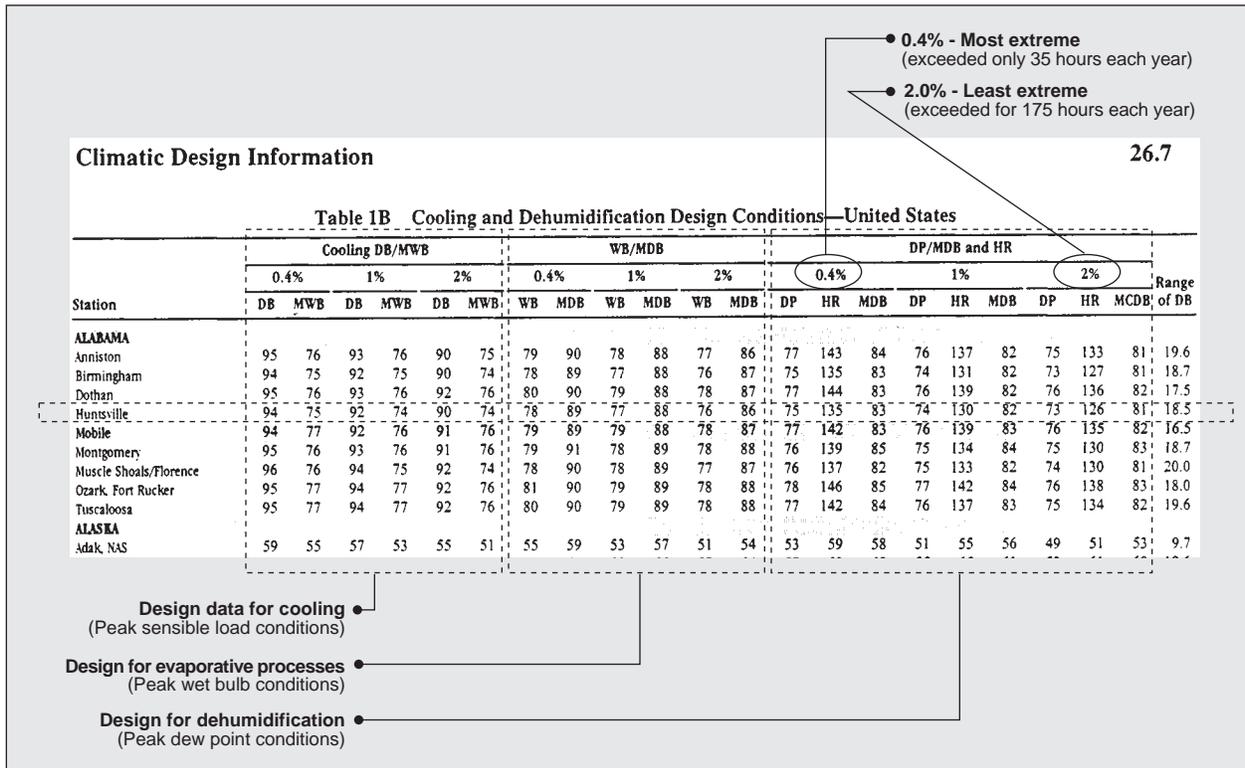
Chapter 26 of the 1997 edition of the Handbook of Fundamentals published by ASHRAE (American Society of Heating, Refrigerating and Air Conditioning Engineers) contains climatic design data that has been completely revised, recalculated and expanded.

Designers of air conditioning systems for hot and humid climates will be pleased to note that, for the first time, the chapter contains correct values for peak moisture conditions. This is in sharp contrast to older editions, which contained only the average moisture during periods of peak dry bulb temperatures. The new data show that using earlier, temperature-based data for humidity design underestimates the true peak moisture loads by 30 to 50%, depending on the humidity control level in the space. This paper explains the new data elements and suggests some of its potential implications for engineers designing air conditioning systems for hot and humid climates.

BACKGROUND

Nearly 100 years ago, Willis Carrier was assigned the job of improving humidity control at the Sackett-Williams Lithography and Printing Company in Brooklyn, New York. He chose to accomplish that task by chilling the incoming fresh air below its dew point to remove its moisture load. Some sources cite that project as the beginning of the modern era of air conditioning.¹ But in the years since Carrier dehumidified that printing plant, the air conditioning industry has focused on temperature control, often losing sight of the moisture component of air conditioning.

Evidence of that single-minded focus on temperature is shown by the climatic design data published in the Handbook of Fundamentals up through its 1993 edition. Chapter 24 of that volume contains data which allow an engineer to size air conditioning equipment to remove peak sensible temperature loads—but it does not allow accurate calculation of peak moisture loads.²



Most the air conditioning industry has not noticed this shortcoming, but the problem could not be ignored by the manufacturers of precision dehumidification equipment. The success of their systems is based on accurately quantifying peak moisture loads. So dehumidification manufacturers investigated the behavior of weather data, and published their own peak moisture load information³ The industry as a whole, however, does not have full confidence in such manufacturer-financed research, since larger loads seemed to mostly benefit the manufacturer. Also, the sensible peak load data from ASHRAE always appeared to include peak moisture loads. The Handbook of Fundamentals shows the average wet bulb temperatures during periods of peak dry bulb temperatures. Despite warnings in that chapter, most engineers assumed that the peak sensible load point was also the peak moisture load.²

To resolve the controversy about peak moisture loads, ASHRAE Technical Committee 3.5 (Desiccant and Sorption Technologies) asked the society to perform research into the true peak moisture conditions in the United States and Canada. That research, completed in 1995,⁴ showed that indeed, the sensible heat load data creates a false sense of security with respect to dehumidification design. Peak moisture loads are 20 to 50% greater than what would be expected from looking at periods of peak dry bulb temperature. Peak moisture levels actually occur at moderate, rather than at extreme dry bulb temperatures—after rainstorms, and during early morning hours, when condensed moisture is evaporating into air at ground level.

However, the results of the research were not apparent to design engineers, who almost universally use the Handbook of Fundamentals—not research reports—to design systems. So TC 3.5 joined with TC 4.2 (Weather Data) to perform more extensive research into weather at all the locations—international as well as domestic—that are shown in Climatic Design Chapter of the Handbook. That research project⁵ was expanded beyond moisture to include other important missing data elements. It also investigated hundreds of additional international locations. The results are now contained in Chapter 26 of the 1997 ASHRAE Handbook of Fundamentals.

1% SEASONAL PEAK VALUES VS. 0.4% ANNUAL PEAK VALUES

Before reviewing the data in the new Chapter, it will be useful to explain a major change in calculation methodology. The 1993 and earlier editions displayed seasonal extremes, but the 1997 edition shows the annual extremes.

In older editions, extreme cooling values were calculated according to the percent of the summer-season

hours that a given temperature was likely to be exceeded. The summer season was assumed to be June, July, August and September in the Northern Hemisphere, and December through March in the Southern Hemisphere. So the “1% value for temperature” represented the dry bulb temperature that is not likely to be exceeded for more than 1% of the 2928 hours of the summer season. In other words, expect to have only 29 hours above that temperature during the summer. But this calculation methodology was not consistent throughout all stations listed in the 1993 handbook, and it has other shortcomings.

In Canada, the 1% value was calculated against the hours in July alone, rather than the full summer season. Also, it is difficult to define the “summer” as any universal sequence of months at locations with marine climates near the equator. Monsoons and ocean evaporation may dominate local weather patterns, creating extreme conditions during different periods compared to the better-defined seasons of continental climates further from the equator.

To ensure consistency worldwide, and to improve accuracy in tropical climates, the new ASHRAE extremes were calculated on an annual, rather than a seasonal basis for all locations. The hourly percentages were adjusted, so that in absolute terms the values for the new extremes are not much different from the extreme temperatures in older editions. For example, the 1% seasonal values for dry bulb temperatures corresponded well with 0.4% annual values at most locations. So the new column of 0.4% values takes the place of the old 1% values. The new 1% annual value corresponds to the old 2.5% seasonal value, and the new 2% annual value takes the place of the old 5% seasonal value. Similar adjustments were made to heating design values, so that the new 99.6% annual dry bulb temperatures corresponds to the old 99% seasonal values.

These changes in calculation methodology must be kept in mind when comparing the extreme values of the 1993 handbook to values in the 1997 edition. Differences between these editions are likely to reflect this calculation change, rather than any significant climate change between 1993 and 1997.

NEW DATA ELEMENTS FOR COOLING AND DEHUMIDIFICATION DESIGN

Figure 1 shows the values and the layout for the new cooling and dehumidification design data in Chapter 26 of the 1997 Fundamentals. There are three major groups of data, including peak values for dry bulb, wet bulb and dew point. For each of these variables, values are displayed for the 0.4%, 1% and 2% annual extremes, along with mean coincident values for other variables.

Mean coincident” values are the average of that

coincident variable during the period when the primary variable is extreme. For example, in Huntsville, Alabama, the 0.4% dry bulb temperature is 94°F. That means that in Huntsville, it not likely that the temperature will exceed 94°F for more than 35 hours each year ($8760 \times 0.004 = 35$). To calculate the mean coincident wet bulb temperature, the researchers identified those 35 hourly observations, and calculated the average value of the wet bulb temperatures that occurred during those specific hours. So in Huntsville, the average wet bulb temperature is likely to be 75°F whenever the dry bulb temperature is 94°F and above.

PEAK COOLING LOAD - DB/MWB

The display of peak cooling loads is not significantly different from that in previous editions, except for the fact that values are calculated as annual rather than seasonal extremes, as explained above.

PEAK EVAPORATION LOAD - WB/MDB

Peak load conditions for evaporative processes are shown in the next group of columns, and again, the values are annual rather than seasonal. Another change from earlier editions is that the researchers have calculated the mean dry bulb temperature coincident to the peak wet bulb temperatures. In Huntsville, for example, the average dry bulb temperature is likely to be 89°F when the wet bulb temperature is at or above 78°F. This additional information is useful when sizing cooling towers or other evaporative cooling equipment. The difference between the dry bulb and wet bulb temperatures—the “wet bulb depression”—is one of the principal driving forces for adiabatic drying of products and adiabatic cooling of water and air.⁶

PEAK DEHUMIDIFICATION LOAD - DP/MDB AND HR

These columns are entirely new. The data they contain did not exist in previous editions of the handbook. In each group of three columns, the first column contains the peak dew point (DP), the second contains the humidity ratio (HR) and the third contains the mean

coincident dry bulb temperature (MDB). Returning to the example of Huntsville, the 0.4% peak value for dew point is 75°F. That dew point represents a humidity ratio (moisture content) of 135 grains of water per pound of air. And when the dew point is that high or higher, the average dry bulb temperature (MDB) is likely to be 83°F.

Load calculations use humidity ratio and not dew point, but the dew point is the value recorded by the weather stations. So the humidity ratio is included in the new chapter to save the engineer the trouble of converting the dew point to grains per pound or grams per kilogram. At constant pressure, that conversion is a simple look-up from a table. But most of the locations are not precisely at sea level, and an accurate conversion must consider the atmospheric pressure at the measurement station. As a convenience the engineer, the completed conversion to humidity ratio is shown for each dew point in the tables.

RANGE OF DB

The final column contains the average daily range of dry bulb temperature during the hottest month of the year. The value is used in cooling load calculations. These values have been recalculated using the most current long-term weather records, but they not different in either use or derivation from the values displayed in earlier editions of the handbook.

COMPARING PEAK VALUES FOR HEAT AND MOISTURE

From the perspective of the designer of air conditioning systems for hot and humid climates, the most significant news about the new data is the difference between the old, “assumed” moisture peak at the peak sensible design point, and the new “true” moisture peak.

Consider the example of Huntsville, Alabama. As seen in figures 1 and 2, the peak sensible heat load occurs at 94°F dry bulb, and 75°F wet bulb. That represents a humidity ratio of 96 grains per lb. Before the 1997 edition, the engineer might well have assumed

	0.4% Dry Bulb Design				0.4% Dew Point Design				Loads are larger at the peak dew point			
	DB	MCWB	HR	Enthalpy	DP	MCDB	HR	Enthalpy	Moisture	Enthalpy	Total Load	Moisture Load
	(°F)	(°F)	(gr/lb)	(Btu/lb)	(°F)	(°F)	(gr/lb)	(Btu/lb)	Δ (gr/lb)	Δ (Btu/lb)	vs. 75° 50%	vs. 65 gr/lb
Boston, MA	91	73	93	36.5	72	80	119	38.0	26	1.5	117%	193%
Chicago, IL	91	74	98	37.3	74	84	130	40.5	32	3.2	133%	197%
Huntsville, AL	94	75	96	38.2	75	83	135	42.5	39	4.3	141%	226%
Miami, FL	91	77	117	40.5	78	83	144	42.8	27	2.3	118%	152%
St. Louis, MO	95	76	104	39.3	76	85	138	41.8	34	2.5	122%	187%

Figure 2. Comparing moisture levels and loads at 0.4% dry bulb vs. 0.4% dew point

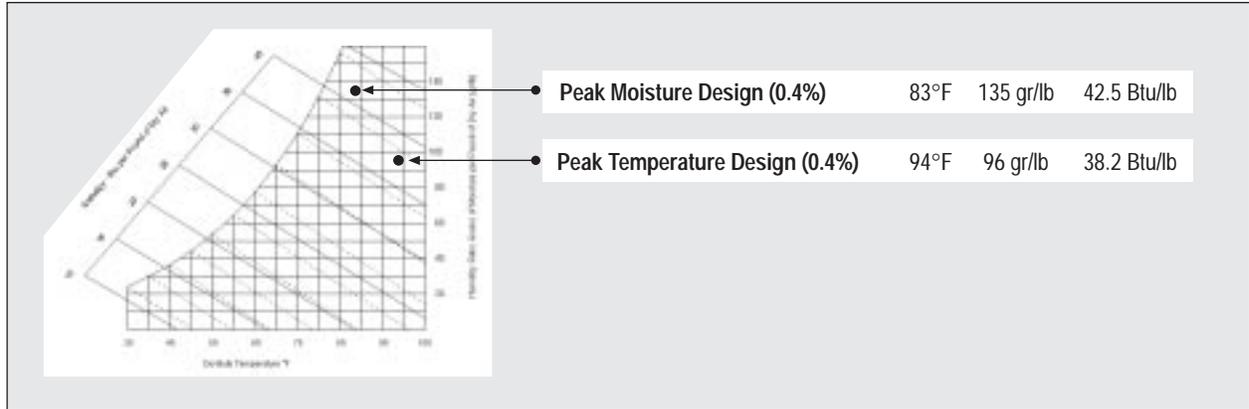


Figure 3. Psychrometric plot of 0.4% cooling and dehumidification design points - Huntsville, AL

that 96 gr/lb is the peak moisture level. But it is now apparent that the peak moisture is 135 gr/lb, and that such conditions usually occur at temperatures of 83°F.

Figure 3 shows these points on a psychrometric chart, where other differences become apparent. For example, the total heat—the enthalpy—of the peak temperature is 38.2 Btu/lb, compared to 42.5 Btu/lb for the peak moisture condition. In other words, the maximum total heat load occurs during the moisture peak, not the temperature peak—and the difference is rather large.

Figure 4 shows how these points compare when calculating the loads from ventilation air in a building maintained at 75°F and 50%rh. Assuming the building is a small quick-service restaurant, and needs 2,000 scfm of ventilation air, the load at peak temperatures is $2,000 \times 4.5 \times (38.2 - 28.1) = 90,900 \text{ Btu/h} \div 12,000 = 7.6$ tons. But at the peak moisture condition, the total load is $2,000 \times 4.5 \times (42.5 - 28.1) = 129,600 \text{ Btu/h} \div 12,000 = 10.8$ tons. In other words, for ventilation air in Huntsville, the total cooling load at the moisture peak is 42% greater than at the temperature peak.

If the air conditioning unit had been sized for the temperature peak, one might expect that it would have

difficulty maintaining control during periods of peak moisture, when the load is 41% greater. And in fact, such difficulties are reported by owners of quick-service restaurants in humid climates.

Considering moisture alone, the load difference between these two extremes is even greater. At the temperature peak, the moisture load is $2,000 \times 4.5 \times (96 - 65) = 279,000 \text{ gr/hr} \div 7,000 = 39.9 \text{ lbs/hr}$. In sharp contrast, the load at the peak moisture condition is $2,000 \times 4.5 \times (135 - 65) = 630,000 \text{ gr/hr} \div 7,000 = 90 \text{ lbs/hr}$.

In other words, if an engineer assumed that the peak moisture load occurred during periods of peak temperature, he or she would be surprised to learn that the true peak moisture load is actually 226% of that value. Systems designed under such misimpressions would be undersized by 63%, and would probably have difficulty controlling humidity.

POSSIBLE IMPLICATIONS OF THE NEW PEAK MOISTURE LOAD DATA

These moisture loads are not different than in the past—they are merely more apparent since they are displayed in the ASHRAE Handbook of Fundamentals.

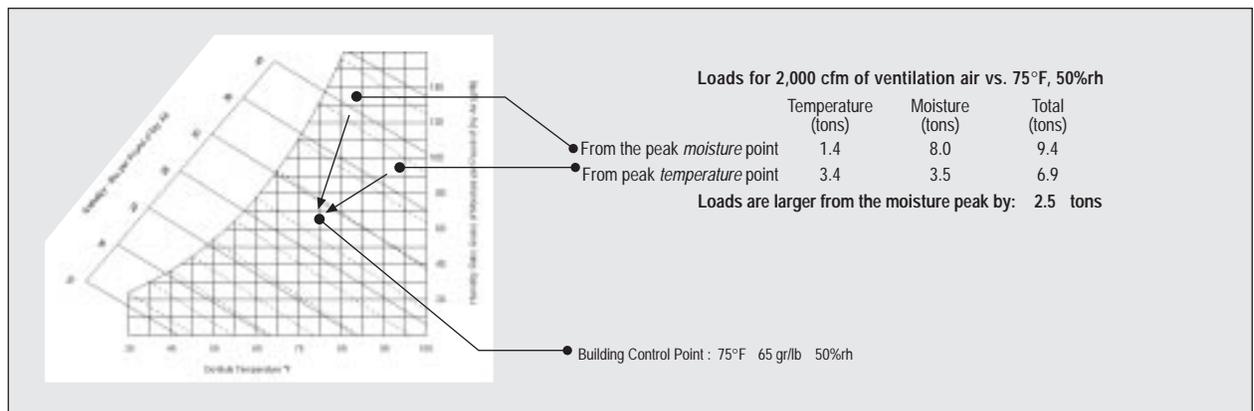


Figure 4. Comparing ventilation loads for cooling and dehumidification design points - Huntsville, AL

Engineers and equipment manufacturers might logically ask: why do these loads matter, if existing equipment and systems perform to the owners expectations? The answer probably depends on each project. If past practices produced equipment and systems which satisfy building owners and occupants, then one need not be concerned about the fact that the true moisture loads are much larger than previously shown in the ASHRAE handbook.

However, where systems or equipment have not performed to expectations, the design and manufacturing communities might wish to consider if these previously-hidden moisture loads are responsible for problems not explained by other causes.

It is probably best to leave such deliberations in the hands of those who have the facts about specific cases, but some questions might be in order as engineers and owners consider the implications of these moisture loads.

CONSIDERATIONS FOR MANUFACTURERS AND GOVERNMENT REGULATORS

Since the oil embargoes of the 1970's, the industry have been encouraged (and in some cases, required) to produce cooling units with high efficiency in removing sensible heat. The industry has achieved this goal admirably, with units now available with seasonal energy efficiency ratios (SEER's) above 10. On the other hand, these units have become noted for their poor humidity-control performance.⁷ It may be that they are incapable of removing high moisture loads, or it may be that they need component or control modifications to meet the load at peak moisture conditions.

Manufacturers can consider the evidence available from their service departments in light of the fact that high moisture loads occur at "off-peak" temperatures. Such evidence may help determine if modifications are needed for particular equipment, or if different equipment is needed for applications involving large amounts of fresh air, where moisture loads are especially high.

Government regulators may wish to consider the definition of "high efficiency" in light of the fact that high moisture loads occur at moderate temperature. If equipment succeeds in controlling temperature at peak sensible load conditions, but fails to control temperature and moisture during periods of high moisture, perhaps the definition of "high efficiency" needs to include some measure of moisture removal performance in addition to sensible heat removal.

CONSIDERATIONS FOR SYSTEM DESIGNERS

When humidity control is not a consideration, there may be no need to act on the fact that moisture loads are higher than formerly expected. On the other hand,

if system have difficulty controlling comfort or difficulty maintaining temperature control at "off-peak" temperatures, it may be that high moisture loads are consuming the equipment capacity that was in place to remove sensible heat loads. In those cases, it might be useful to take the suggestion made in the new Chapter 26—calculate performance of the system at peak moisture conditions after it has been sized for the peak temperature conditions.⁸ If the selected equipment does not meet the loads at the peak moisture conditions, the engineer may consider different equipment, or save money by obtaining agreement with the owner that off-peak hours need not be as comfortable as on-peak hours.

Another alternative in the face of high moisture loads is to remove the largest load before it enters the building. In nearly all systems, the largest moisture load will be contained in the ventilation air brought into the building to assure dilution of internally-generated contaminants. The engineer may wish to consider treating that ventilation air with a separate system dedicated to handling the ventilation load alone. This allows the rest of the cooling systems to remove primarily sensible heat, so they are not overwhelmed by the moisture loads from ventilation air. Such strategies have been used with great success by owners of large retail buildings, where ventilation loads are very high⁹

System designers may as wish to consider the value of "economizers" in light of the new moisture load data. An economizer is a system of dampers and fans which bring in large amounts of outside air when that outside air is cooler than air inside the building. The idea is to save energy that would be used to cool warmer recirculating air, replacing part or all of the recirculating air with cooler outside air. But the new data suggest that bringing cool air may simply load the building with moisture, which increases the total load on the system, even though the temperature component of the load is reduced. The only way to be certain would be to analyze the loads over the whole year, and to examine the moisture loads separately from the heat loads. Computer programs to identify these load independently are now available to supplement the peak design data in the ASHRAE handbook¹⁰

CONSIDERATIONS FOR BUILDING OWNERS AND OCCUPANTS

Owners and building occupants may wish to consider their comfort experiences in light of the this moisture design data. If comfort has been generally maintained in the past, then no changes are necessary. But if summer comfort has been elusive with respect to either temperature or humidity, or if the building has persistent musty odors, it may be useful to examine the equipment and it's operating sequences with respect to moisture removal. Either in-house or outside consult-

ing engineers could be assigned to survey the installed equipment, define its operating sequences, and determine if it is capable of removing the high moisture loads that come from code-required ventilation air.

In many cases, the equipment or air flows of existing systems can be adjusted, reducing temperature removal, while increasing moisture removal without additional equipment. In fact, recent field experience suggests that if humidity is controlled at 45 to 50%, temperatures can be allowed to rise to 78°F or higher while maintaining comfort. Where adjustments alone do not solve a problem—and assuming the problem is worth spending money to solve—then owners might consider removing the ventilation moisture load with a new system dedicated to that purpose. That strategy can add moisture removal capacity without disrupting existing systems.

Certainly the lowest-cost way to improve system performance with respect to moisture removal is to maintain it well. Specifically, if chilled water temperatures have risen because of under-investment in maintenance, or if clogged filters have reduced air flows, then the simple, low-cost solution is to change filters and refurbish the chiller.

For new buildings in the planning stage, owners who have experienced problems in the past with either temperature or humidity control might do well to discuss the new humidity design data with the engineer who will be responsible for the new systems. If humidity is of secondary concern, the owner could simply request that the engineer check system performance at the peak moisture condition to ensure that humidity is not likely to rise above expected levels. Where humidity is of primary importance, the owner and engineer should discuss the issue of design data in more depth, coming to an agreement on which data will be used to calculate the loads.

CONCLUSION

Ultimately, the owner is the real authority deciding which design data to use to build or operate an air conditioning system. Neither building codes nor ASHRAE nor any outside consultant can make the final decision about which peak load data to use for which purpose. Each owner must weigh the financial costs, comfort trade-offs and any liability factors to arrive at the best compromise between flawless year-round control and economic reality. But now, at least, Chapter 26 of the 1997 ASHRAE Handbook of Fundamentals allows owners and engineers to make better decisions in light of our much-improved understanding of peak moisture loads.

ACKNOWLEDGEMENTS

The weather data in the 1997 Handbook of Fundamentals were calculated by a research team headed by Professor Donald Colliver, Ph.D, P.E. and Professor Richard Gates, Ph.D, P.E. of the Agricultural Engineering Faculty of the University of Kentucky. Without their experience and dedication, the massive job of calculating revised design conditions for 1459 locations could never have been accomplished in time for publication in the 1997 Handbook of Fundamentals. The engineering community should also be grateful to the members of ASHRAE Technical Committee 4.2, and their research subcommittee chaired by Robert Morris, Chief of the Industrial Meteorology Branch of Environment Canada. Without their support over eight years, and their insightful supervision of RP-754 and RP-890, accurate update of the weather data would not have been possible. Finally, thanks to the members of ASHRAE TC 3.5, without whom no weather data update project would have been initiated.

NOTES

1. "Everyone talks about the weather; Willis Carrier did something about it". Air Conditioning, Heating & Refrigeration News, July 31st, 1989, pp. 3-4 Business News Publishing, Troy, Michigan. USA
2. "Note that a dew point calculated from design dry-bulb and mean coincident wet bulb temperatures is generally significantly lower than the dew point that corresponds to the same nominal percentile" is the brief, (and not entirely clear) caution provided in paragraph 4 of page 24.3 of the 1993 ASHRAE Handbook of Fundamentals.
3. Appendix A, The Dehumidification Handbook, 2nd Edition. 1990. L.G. Harriman III, Editor. Munters Cargocaire, Amesbury, MA USA
4. "1, 2.5 and 5% occurrences of extreme dew point temperature with mean coincident dry bulb temperatures." Research Project 754RP, 1995. American Society of Heating Refrigerating and Air Conditioning Engineers, Atlanta, GA USA
5. "Updating the tables of weather design information in the Handbook of Fundamentals." Research Project 890-RP. 1997. American Society of Heating Refrigerating and Air Conditioning Engineers, Atlanta, GA USA
6. An adiabatic process is one which proceeds without the addition of energy from outside the system. Examples include evaporation of water in a rainstorm and crop drying in the field. Some air conditioning processes

are also called adiabatic, because the drying or humidifying proceeds without addition of energy other than fan and pump energy which moves air and water through the system.

7. These difficulties have been widely-reported in many applications. One example of the problems created by such units is that of the hotel and motel industry, which estimates that it loses over \$86 million every year to mold and mildew damage. That problem is apparently caused in part by “high-efficiency” units in hotel rooms, which do not operate long enough to dehumidify the air. They remove sensible heat very efficiently, so they satisfy the room thermostat and shut off quickly—before any dehumidification is accomplished. (Mold and Mildew in Hotels and Motels. [1991] A report of the Executive Engineers Group of the American Hotel and Motel Association, Washington, DC USA)

8. Paragraph 3. page 26.3 of the 1997 ASHRAE Handbook of Fundamentals: (Referring to the design dew point values) “These values are especially useful for applications involving humidity control, such as desiccant cooling and dehumidification and fresh air ventilation systems. The values are also used as a check point when analysing the behavior of cooling systems at part load conditions, particularly when such systems are used for humidity control as a secondary function.”

9. “Gas-fired Desiccant System for Retail Super Center” Spears and Judge. ASHRAE Journal October 1997 pp: 65-69 “...During a two-month test period, the (desiccant) store temperature set point was raised from 74° to 78°F while the relative humidity was maintained at 45% with no comfort complaints. During that period, the (desiccant) store utility cost was \$10,448 and the (conventional cooling store, with set point of 74°F and no humidity control set point) energy cost was \$12,063.”

10. One such program is called “BinMaker(TM)”, published by the Gas Research Institute, Chicago, IL. It contains a complete year of 8760 hourly weather observations for 239 U.S. locations, and was used to calculate the annual VLI's (Ventilation Load Indexes) described in: “Dehumidification and Cooling Loads From Ventilation Air”, Harriman, Plager and Kosar, ASHRAE Journal, November 1997, pp:37-45

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