Improving Humidity Control With

ENERGY RECOVERY VENTILATION

By **John Dieckmann,** Member ASHRAE

Energy recovery ventilation (ERV) generally is thought of as an energy-saving measure, using the exhaust air from a building to precondition (precool and dehumidify during the cooling season and preheat and humidify during the heating season) incoming ventilation air, with only the small energy penalty of the blower power to overcome the pressure drop in the ERV. Indeed, this technology is a proven means of reducing cooling and heating loads and energy use, allowing downsizing of cooling and heating equipment and reducing the magnitude of the peak electric load on design condition cooling days. An equally compelling reason for using ERV is to allow unitary air-conditioning equipment to maintain the indoor relative humidity within the desirable 40% to 50% range under virtually all conditions, including high humidity outdoor conditions.

ERV is implemented with an air-to-air enthalpy exchanger, as a component of an engineered air handler, integrated with a unitary air conditioner, or as a component of a makeup air unit. As illustrated in *Figure 1*, an air-to-air enthalpy exchanger transfers sensible and latent heat (i.e., heat and water vapor, or total enthalpy) between conditioned air being exhausted from the building (at T_3 and H_3) and outside air being supplied to the building (at T_1 and H_1). During the cooling season, the warm, humid outside air is precooled by transferring heat and moisture to the cooler, drier air being exhausted from the building, so that T_2 and H_2 are lower than T_1 and H_1. The opposite is true in heating mode (i.e., building exhaust air preheats the cooler incoming outdoor air via the enthalpy exchanger), but the primary focus of this article is the cooling season.

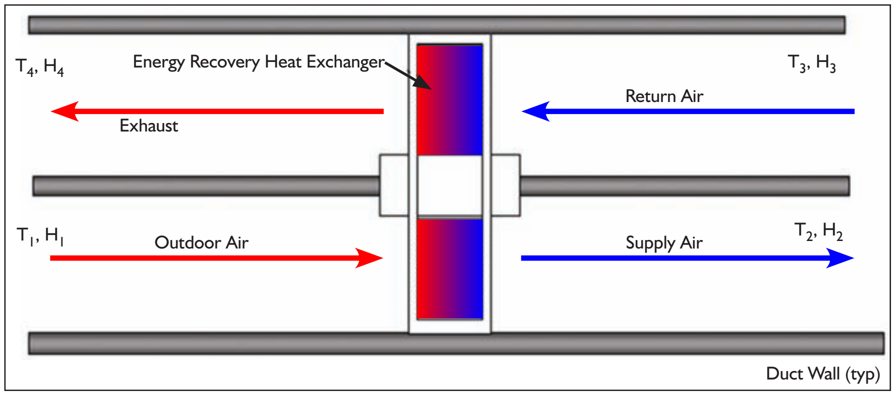

Figure 1: Generic configuration of an enthalpy exchanger.

Rotary wheel type enthalpy exchangers have a heat and mass transfer core material that is coated with a desiccant. The core alternately passes through the outdoor supply air and exhaust airstreams as the wheel rotates, absorbing heat and moisture from the warmer, more humid airstream, and desorbing heat and moisture to the cooler, drier airstream. Heat and moisture are stored in the core material as the core passes from one airstream to the other.

In many types of commercial buildings—e.g., offices, retail, hotels, hospitals—the main design humidity load is the moisture brought into the occupied space of the building with ventilation air at hot, humid ambient conditions. The reason why ERV has a positive impact on building humidity control is that an ERV will remove a large portion of the moisture from the ventilation air before it enters the building by transferring it to the less humid exhaust air from the building. To appreciate why this matters, a review of how sensible and latent cooling loads in commercial buildings have evolved provides useful perspective.

In recent years, an increasing number of buildings and their occupants have experienced serious moisture problems, including so-called sick building syndrome, mold growth, and occupant discomfort due to high humidity. These problems arise from poor interior humidity control and liquid water sources such as plumbing leaks and rain water leakage (Serious cases of liquid water cannot be remediated by dehumidification alone. The water leaks need to be addressed). Moisture and mold problems can cause serious health problems among occupants and reduce productivity. In extreme cases, buildings have been rendered uninhabitable. The financial consequences of these situations can be substantial and are reflected in rapidly escalating liability insurance rates. These impacts have been widely reported and documented by research and in the press.

Poor humidity control often is attributed to the inability of conventional unitary air conditioners to handle the moisture loads. A common conception is that the increased efficiency of modern air-conditioning equipment has resulted in reduced latent (moisture removal) capacity.

Myth of Declining Latent Cooling Capacity of Unitary ACs

When air-conditioning equipment cools air, it reduces the temperature of the air (sensible cooling) and it reduces the moisture content of the air (dehumidifies the air) by causing a portion of the water vapor in the air to condense into liquid water (latent cooling, so-called because the latent heat of moisture condensation from the air is the portion of the cooling load associated with dehumidification). The dehumidification effectiveness of air-conditioning equipment is commonly characterized by the sensible heat ratio (SHR), which is the ratio of the sensible cooling capacity to the total (sensible + latent) cooling capacity. Reducing SHR increases the portion of the total cooling capacity that is providing dehumidification.

It has become a matter of common wisdom that increases in the energy-efficiency ratio (EER) of unitary air conditioners that have occurred since the early 1980s have been accompanied by a decrease in the latent cooling capacity as a fraction of the total cooling capacity (i.e., increased SHR). The wisdom explains that among the ways that efficiency is increased, increasing the evaporator surface area results in a higher evaporator temperature and, therefore, less moisture removal capacity.

In the real world of commercially available air-conditioning equipment, no such direct relationship exists. ARI[1] conducted an extensive review of the relationship between EER and SHR, for current equipment and equipment going back to the 1970s. The consistent finding is that the SHR of individual unitary air conditioner models has varied between 0.65 and 0.80 at all EER levels, from 1970 to the present, with no statistically significant correlation between either the EER and the SHR or between the year of manufacture and SHR.

Ratio of Sensible and Latent Cooling Loads Has Changed

Although the SHR of unitary air-conditioning equipment has not changed, building loads have changed substantially. The energy-efficiency improvement measures such as better wall and roof insulation, reduced window U-values, increased solar shading, more energy-efficient lighting, which have been driven by Standard 90.1 have almost exclusively reduced sensible

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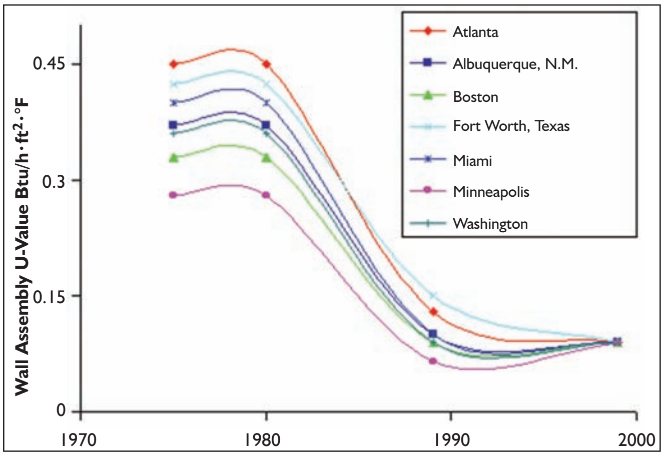

Figure 2: 90.1 maximum wall assembly U-values over time.

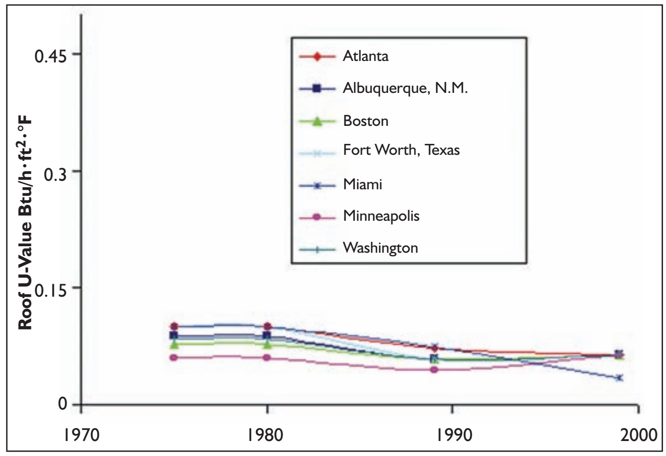

Figure 3: 90.1 maximum roof U-values over time.

cooling loads. Latent cooling loads, which for most commercial buildings are primarily due to ventilation, infiltration, and occupants, have not changed substantially.

The effect for most buildings constructed or thoroughly remodeled since 1990 is to raise the latent cooling load relative to the sensible cooling load at all conditions. At typical cooling design conditions (in most parts of the U.S., at high dry-bulb and high wet-bulb conditions), outdoor humidity levels and the consequent latent loading is quite high, while envelope and internal sensible loads have been reduced, compared to buildings constructed prior to 1990. Conventional unitary air conditioning tends to satisfy the sensible load well before the latent load is met, with the result that the steady-state indoor relative humidity increases from the desired range around 50% to around 70% (uncomfortable and supportive of mold growth). At moderate outdoor temperatures, 65°F to 70°F (18°C to 21°C), with

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high outdoor humidity, the low sensible load results in unitary air conditioners cycling on-off frequently with short on cycles. During the off cycle, moisture on the coil can reevaporate, causing further deterioration of the net latent capacity[2,3] and driving up the indoor RH.

Changing Building Ventilation and Efficiency Standards

During the past 30 years, two trends have affected commercial building cooling loads and the ratio of latent to sensible load. The first of these is a drive to improve the energy efficiency of the U.S. economy, reflected in tightened efficiency standards for automobiles, appliances and buildings. The second trend has been the recognition of the need for minimum levels of outdoor ventilation air to maintain reasonably healthy conditions inside buildings. ASHRAE Standards 62 and 90 (they became 62.1 and 90.1 years later) were initially introduced in the early 1970s, and created to set minimum standards for building ventilation and for energy-related aspects of building design, respectively. Since then, major revisions have been published at regular intervals. During the past 30 years, the outdoor ventilation airflow rates required by Standard 62 were first reduced then restored to earlier levels. In most commercial and institutional buildings outdoor ventilation air is the main source of humidity load, so the net effect has been to leave latent loads unchanged. Standard 90.1 has mandated reductions in various contributions to the sensible cooling load of a building, but has left latent loads largely unaffected.

Building Envelope

Standard 90.1 specifies minimum envelope thermal performance by specifying maximum U-values for wall and roof assemblies. Variations in climate are accounted for, so that U-values are cost-effective relative to the ambient conditions in the cooling and heating seasons of a given locale. Over time, the scheme for defining climate zones has changed, so it is not possible to track the changing requirements in U-value over time through particular Standard 90.1 defined climate zones. Instead, the maximum Standard 90.1 U-value can be tracked for particular cities, as the successive versions of Standard 90.1 apply to the specific climate characteristics (e.g., design heating degree days and cooling degree days) of each city.

Figure 2 plots Standard 90.1's maximum wall assembly U-value for seven cities that represent the range of climate conditions in the continental U.S., from cold to moderate to hot-humid to hot-dry. The required maximum U-values have decreased by a factor of three to four, which reduces the wall conduction contribution to the sensible load by a like amount.

Figure 3 plots the Standard 90.1 maximum roof U-value over the period during which Standard 90 has been in effect. Modest (~30%) reductions in the maximum U-value have occurred during this time period.

Figure 4 plots the maximum solar heat gain coefficient for window glazings over the life of Standard 90. It controls the fraction of incident solar radiation that enters the conditioned space.

The key point is that each of these measures—wall assembly and roof U-values, solar heat gain coefficients—addresses contributors to the sensible cooling load only.

Equipment Loads

Internal heat gains from energy dissipated by equipment within the building are a significant fraction of the building cooling loads (and contribute to heating the building during the cold periods of the year). Standard 90.1 specifically controls two of the contributors to internal heat gain: lighting and motors. Minimum motor efficiencies are set at the EPAct minimums for general purpose motors, a modest increase in minimum efficiency compared to earlier revisions of 90.1. Maximum lighting power density is plotted in *Figure 5*. It has been reduced by 40% since the first version of Standard 90. Standard 90.1 does not specify efficiencies or power levels for office equipment, but power densities have been increasing as the usage of computers, printers, etc. has increased, offsetting

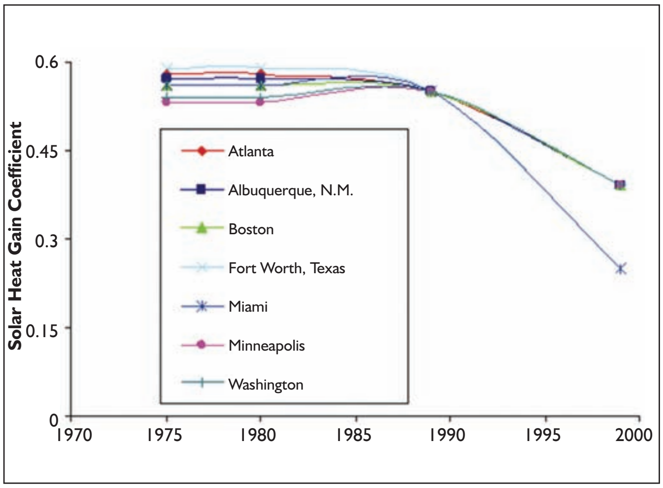

Figure 4: 90.1 maximum solar heat gain coefficient over time.

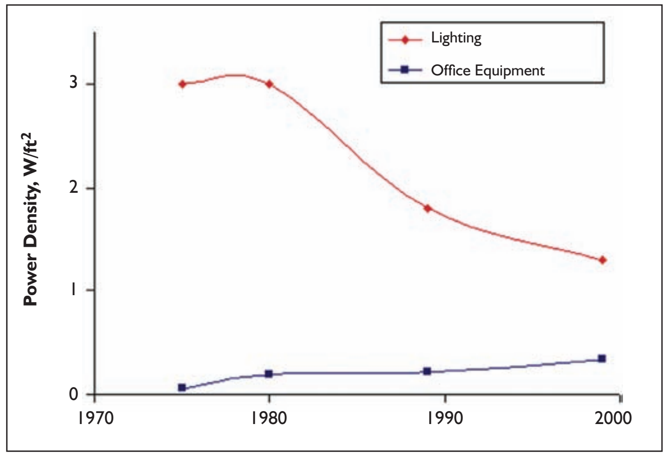

Figure 5: 90.1 maximum lighting power density (for office space) and estimated office equipment power density.

some of the reduction of allowed lighting power use. Estimated office equipment power densities are also plotted in *Figure 5*. As is the case with envelope requirements, the Standard 90 requirements to reduce internal loads addresses sensible load only.

Version of Standard 62	Required Outdoor Ventilation Air, cfm/ft²			
	Offices	Conference Room	Reception/Lobby	Weighted Average*
62-2001	0.140	1.00	0.90	0.292
62-1999	0.140	1.00	0.90	0.292
62-1989	0.140	1.00	0.90	0.292
62-1981	0.035	0.30	0.15	0.073
62-1973	0.150	1.50	0.30	0.300

* Based on 80% offices, 10% conference rooms and 10% reception/lobbies.

Table 1: Weighted average ventilation per floor space required for office building.

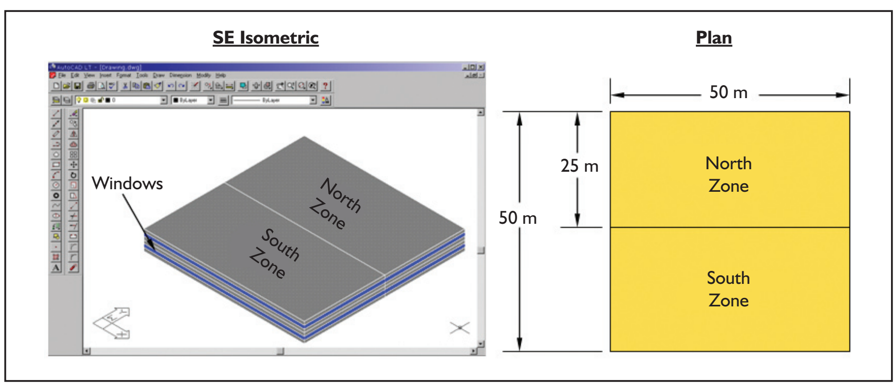

Figure 6: Model office building configuration.

Analysis of a Typical Medium-Size Office Building

A hypothetical medium-size (~50,000 ft² [~4645 m²]) office building was designed, as shown in *Figure 6*, to evaluate the total impact of these changes in U-values, lighting power density, and ventilation on the total building sensible and latent cooling loads.

As indicated in *Figure 6*, the model office building has two stories with a square footprint, 50 m (164 ft) on a side. The building has four space conditioning zones, the north and south halves of each floor. The breakdown of floor space of the building was assumed to be 80% offices, 10% conference rooms, and 10% reception/lobbies. *Table 1* summarizes the calculation of the average ventilation per square foot required by each revision of Standard 62. Note that ventilation airflow rates were reduced in the 1980s in response to the energy crisis of that era, but were then restored to previous levels (approximately) when cases of sick building syndrome and ongoing IAQ research recognized the need for increased amounts of ventilation airflow.

Evolution of Envelope and Lighting Specifications Over Time

Table 2 summarizes the input values of the envelope specifications for the load model calculations in each of the seven cities that were chosen to represent different climate zones within the U.S. Lighting and office equipment power densities are taken from *Figure 5*.

When these changes are accounted for in the cooling load model of a typical office building (using the DOE EnergyPlus building energy simulation software), the increase in humidity loading (reduction in the SHR) since the 1980s is dramatic, as illustrated in *Figures 7* to *9*, which plots this building cooling load SHR trend for seven cities, at three cooling load design conditions. These three figures give the SHR at three design points—a high outdoor temperature/high coincident wet-bulb temperature day, a high outdoor humidity (high latent load) day, and a high humidity shoulder day. These three design

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		Value for Climate Location						
Specification	Years	Atlanta	Albuquerque, N.M.	Boston	Fort Worth, Texas	Miami	Minneapolis	Washington
Wall Assembly U-Value Btu/h·ft²·°F	1975–1989	0.450	0.370	0.330	0.425	0.400	0.280	0.360
	1989–1999	0.130	0.100	0.089	0.150	1.000	0.065	0.089
	1999–Present	0.089	0.089	0.089	0.089	0.089	0.089	0.089
Roof U-Value Btu/h·ft²·°F	1975–1989	0.100	0.088	0.078	0.100	0.100	0.060	0.084
	1989–1999	0.072	0.059	0.058	0.058	0.074	0.045	0.058
	1999–Present	0.063	0.063	0.063	0.063	0.034	0.063	0.063
Solar Heat Gain Coefficient	1975–1989	0.580	0.570	0.560	0.590	0.560	0.530	0.540
	1989–1999	0.550	0.550	0.550	0.550	0.550	0.550	0.550
	1999–Present	0.390	0.390	0.390	0.250	0.250	0.390	0.390

Table 2: Input values of the envelope.

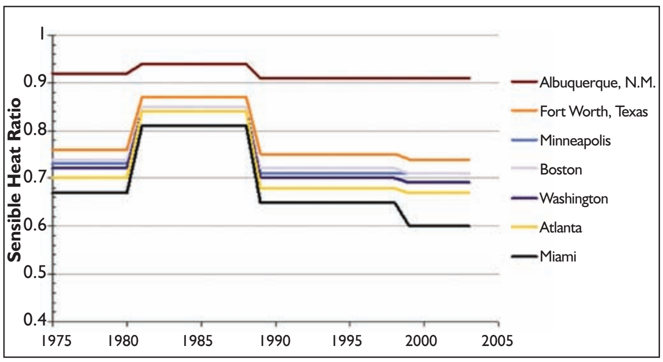

Figure 7: Building load SHR from 1975 to present, at sensible design condition (high outdoor temperature).

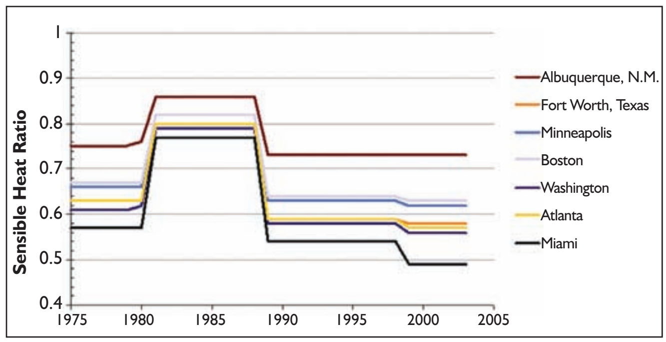

Figure 8: Building load SHR from 1975 to present, at latent design condition (high outdoor humidity).

outdoor conditions are summarized in *Table 3* for each of the seven climate cities.

SHR of HVAC: Options for Increasing Latent Capacity

A variety of options exist for adjusting the latent capacity of the unitary air conditioning equipment that is used to condition the interior space of a significant portion of commercial building space. These include:

- Overcooling part of the airstream to increase the latent fraction, e.g., coil bypass or reduced airflow through the coil;
- Overcooling, as above, and reheating (undoing excess sensible cooling);
- Active desiccants; and
- Enthalpy recovery.

Combined SHR of a Unitary AC with Air-to-Air ERV

The current level of building cooling load SHRs fall outside the range of SHRs that basic unitary air conditioners can provide. One approach to reducing the cooling equipment SHR is to combine a unitary air-conditioning system with another system or component that operates with a low SHR. Examples include dedicated mechanical dehumidifiers or desiccant dehumidifiers and enthalpy recovery ventilation systems. Unlike dehumidifi-

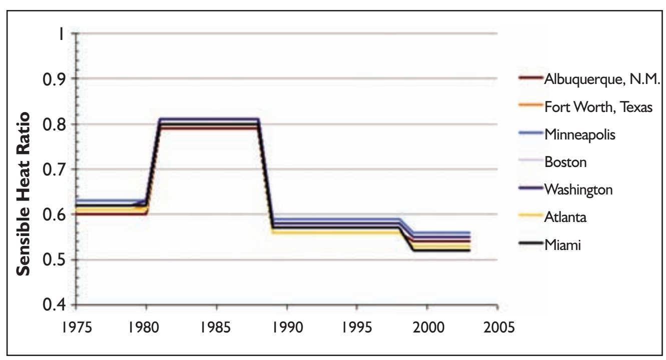

Figure 9: Building load SHR from 1975 to present, at shoulder or part-load condition (high outdoor humidity, moderate temperature).

cation systems that use new energy to reduce humidity, enthalpy recovery uses recovered energy from the building exhaust air as the energy source thereby increasing system efficiency and reducing peak electric demand (ARI Guideline V[4] provides an accepted method for calculating the combined efficiency of a unitary air conditioner combined with an air-to-air enthalpy recovery exchanger.)

The combined capacity and SHR of an ERV integrated with a unitary air conditioner can be calculated by adding the individual contributions to the latent and sensible cooling from

Design Day	Design Temperatures	Temperature for Climate Locations, °F Atlanta	Albuquerque, N.M.	Boston	Fort Worth, Texas	Miami	Minneapolis	Washington
Cooling Design Day	Dry Bulb	93	96	91	100	91	91	95
	Dew Point	68	26	65	64	72	65	68
	Wet Bulb	75	60	73	75	77	73	76
Latent Design Day	Dry Bulb	88	83	87	92	87	88	89
	Dew Point	73	60	70	74	78	71	76
	Wet Bulb	77	68	75	79	80	76	79
HDD 65		2,991	4,425	5,641	2,304	200	7,981	4,707
CDD 50		5,038	3,908	2,897	6,557	9,474	2,680	3,709

Table 3: Design conditions.

each component and dividing the combined sensible capacity by the combined total capacity. The latent and sensible cooling provided by the enthalpy exchanger varies with outdoor conditions; as the outdoor humidity increases, so does the latent capacity of enthalpy recovery. In this way the SHR of enthalpy recovery adjusts automatically to changing outdoor conditions. This characteristic allows for the introduction of additional outside air without negatively impacting the SHR balance.

Figure 10 compares the SHR of the building load (from the examples in *Figures 7* through *9*) with the combined SHR of a unitary air conditioner with an enthalpy recovery exchanger. (The modeled SHR of the unitary air conditioner alone, as indicated by the dotted line, is typical of commercially available products, while the latent and sensible effectiveness of the ERV are both modeled at 70%). The figures show a close match of SHR across a range of humid weather cooling conditions, leading to stable humidity levels in the indoor space.

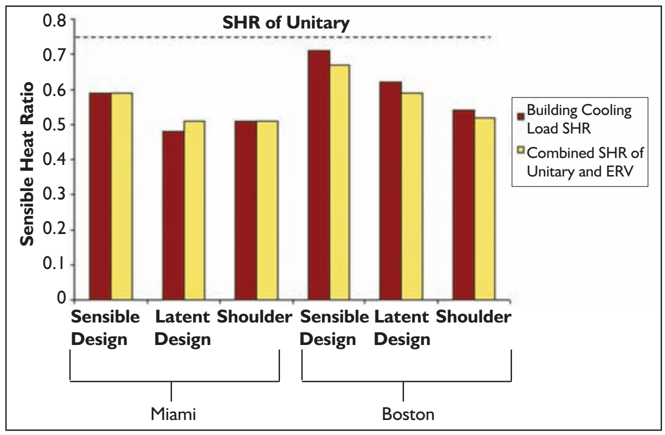

Figure 10: Comparison of the building SHR with the combined unitary + ERV SHR.

Conclusion

The evolution of ASHRAE standards and building technology over the past 30 years has resulted in a mismatch between the SHR of the typical unitary air conditioner and that of the typical commercial building load. The resulting loss of indoor humidity control can cause structural, comfort, and health problems. Enthalpy recovery ventilation systems can address these issues by allowing standard unitary packaged cooling equipment to closely match building SHR, while conserving energy and reducing peak demand.

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