



Matching the Sensible Heat Ratio of Air Conditioning Equipment with the Building Load SHR

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Table of Contents

TABLE OF CONTENTS.....	I
LIST OF TABLES	II
LIST OF FIGURES	III
1. INTRODUCTION	1-1
1.1. THE MYTH OF THE DECLINING LATENT CAPACITY OF UNITARY AIR CONDITIONERS	1-1
1.2. BUILDING HUMIDITY CONTROL PROBLEMS ARE <i>NOT</i> A MYTH	1-2
1.3. THE RATIO OF SENSIBLE AND LATENT COOLING LOADS IN BUILDINGS HAS CHANGED.....	1-3
1.4. OPTIONS FOR INCREASING THE LATENT CAPACITY	1-5
2. THE EVOLVING SENSIBLE HEAT RATIO OF COMMERCIAL BUILDINGS.....	2-1
2.1. SENSIBLE AND LATENT COOLING LOADS	2-2
2.2. CHANGING BUILDING ENERGY EFFICIENCY AND VENTILATION STANDARDS – ASHRAE STANDARD 90.1 AND ASHRAE STANDARD 62	2-3
2.2.1. <i>Building Envelope</i>	2-4
2.2.2. <i>Equipment Loads</i>	2-5
2.2.3. <i>Ventilation and Infiltration</i>	2-6
2.3. <i>BUILDING COOLING LOAD CALCULATIONS FOR A RANGE OF CLIMATE ZONES</i>	2-8
2.3.1. <i>The Evolution of Envelope and Lighting Specifications Over Time</i>	2-9
2.3.2. <i>Analysis Results</i>	2-9
3. THE SYSTEM SENSIBLE HEAT RATIO OF COMBINATIONS OF HVAC EQUIPMENT 3-1	
3.1. SYSTEM SHR – GENERAL CASE	3-1
3.2. SYSTEM SHR – UNITARY AIR CONDITIONER COMBINED WITH AN AIR TO AIR ENTHALPY RECOVERY EXCHANGER	3-2
3.2.1. <i>Example Calculation of Combined SHR for Unitary and ERV</i>	3-3
3.2.2. <i>Illustrative Example – the Example Office Building of Section 2.3</i>	3-4
3.2.3. <i>Sensitivity Analysis</i>	3-6
3.2.3.1. Infiltration.....	3-6
3.2.3.2. ERV Latent Effectiveness < ERV Sensible Effectiveness	3-8
3.2.4. <i>Calculation of Unitary + ERV SHRs for the Examples in Section 3.2.2</i>	3-8
4. SUMMARY OF KEY FINDINGS	4-1
5. CONCLUSIONS.....	5-1
6. REFERENCES	6-1

List of Tables

Table 2-1: Changing Minimum Outdoor Make-Up Air Ventilation Rates Per ASHRAE Standard 62	2-7
Table 2-2: Required CFM/ft ² , Based on Estimated Maximum Occupancy and Required CFM/Person	2-7
Table 2-3: Weighted Average Ventilation per Floorspace Required for Office Building	2-9
Table 2-4: Input Values of the Envelope	2-11
Table 2-5: Design Conditions	2-11
Table 3-1: Combined SHR Calculation	3-4
Table 3-2: Comparison of the Building Cooling Load SHR with the Combined Unitary + ERV SHR in Miami Climatic Conditions	3-5
Table 3-3: Comparison of the Building Cooling Load SHR with the Combined Unitary + ERV SHR in Boston Climatic Conditions	3-5
Table 3-4: Sensitivity Analysis – Impact of Infiltration (at 0.5 Air Changes/Hour)	3-7
Table 3-5: Sensitivity Analysis: $\epsilon_{\text{latent}} = \epsilon_{\text{sens}} - 0.05$ (Unitary SHR = 0.75)	3-8
Table 3-6: Ventilation Portion of Design Load, Corresponding ERV Net Recovery Capacity at $\epsilon=0.7$	3-8
Table 3-7: Design Load and Unitary Capacity Net of the ERV NRC	3-9
Table 3-8: Combined SHR (Unitary SHR = 0.75).....	3-9

List of Figures

Figure 1-1: Airstream Temperature Passing Through Evaporator Coil.....	1-2
Figure 1-2: Indoor RH vs. Building Cooling Load SHR with Conventional Unitary Air Conditioning Equipment.....	1-5
Figure 2-1: ASHRAE 90.1 Maximum Wall Assembly “U” Values, Over Time	2-4
Figure 2-2: ASHRAE 90.1 Maximum Roof "U" Values Over Time	2-5
Figure 2-3: ASHRAE 90.1 Maximum Solar Heat Gain Coefficient Over Time.....	2-5
Figure 2-4: ASHRAE 90.1 Maximum Lighting Power Density (for Office Space) and Estimated Office Equipment Power Density	2-6
Figure 2-5: Model Office Building Configuration.....	2-8
Figure 2-6: Area Plot of Basic Constituents of the Sensible and Latent Cooling Loads – Miami / Latent Design Day / 1st Floor North Zone / 1981-1989	2-10
Figure 2-7: Area Plot of Basic Constituents of the Sensible and Latent Cooling Loads – Miami / Latent Design Day / 1st Floor North Zone / 1989-1999	2-10
Figure 2-8: Building Load SHR from 1975 – Present, at Sensible Design Condition (High Outdoor Temperature)	2-12
Figure 2-9: Building Load SHR from 1975 – Present, at Latent Design Condition (High Outdoor Humidity).....	2-12
Figure 2-10: Building Load SHR from 1975 – Present, at Shoulder or Part Load Condition (High Outdoor Humidity, Moderate Temperature)	2-13
Figure 3-1: Unitary Air Conditioner with a Factory Integrated AAHX	3-2
Figure 3-2: Comparison of the Building Cooling Load SHR with the Combined Unitary + ERV SHR in Miami and Boston Climatic Conditions	3-6
Figure 3-3: Impact of Infiltration on the Match of the Building Cooling Load SHR with the Combined Unitary + ERV SHR (at the Latent Design Condition).....	3-7

Executive Summary

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 both poor interior humidity control and liquid water sources such as plumbing leaks and rain water leakage. Moisture and mold problems can cause serious health problems among the 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 is often attributed to the inability of conventional unitary air conditioners to handle the moisture loads. A common conception holds that the increased efficiency of modern air conditioning equipment has resulted in reduced latent (moisture removal) capacity.

The Myth of the Declining Latent Cooling Capacity of Unitary Air Conditioners

When air conditioning equipment cools air, it reduces the temperature of the air (sensible cooling) and it reduces the moisture content of the air (latent cooling) by causing a portion of the water vapor in the air to condense into liquid water. 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 the 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 1980’s 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 equipment, no such direct relationship exists. ARI [Amrane, 2003] has recently conducted an extensive review of the relationship between EER and SHR, both for current equipment and earlier era 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 the SHR.

The Ratio of Sensible and Latent Cooling Loads in Buildings Has Changed

While the SHR of unitary air conditioning equipment has not changed, building loads have changed substantially. The energy efficiency improvement measures — better wall and roof insulation, reduced window U-values, increased solar shading, more energy-efficient lighting — that have been driven by ASHRAE Standard 90.1 have almost exclusively reduced sensible 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, outdoor humidity levels and the consequent latent loading is quite high, while both 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 (RH) 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, with 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 re-evaporate, causing further deterioration of the net latent capacity and driving up the indoor RH.

The Impact of Changing Building Ventilation and Efficiency Standards

Over 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 US 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 90.1 were created to set minimum standards for building ventilation and for energy related aspects of building design, respectively. They were initially introduced in the early 1970's. Since then, major revisions have been published at regular intervals. Over the past 30 years, the outdoor ventilation airflow rates required by ASHRAE Std. 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. ASHRAE 90.1 has mandated reductions in various contributions to the sensible cooling load of a building, but has left latent loads largely unaffected. When these changes are accounted for in the cooling load model of a typical office building (utilizing 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 E-1 to E-3, which plots the building cooling load SHR trend for 7 cities at three cooling load design conditions.

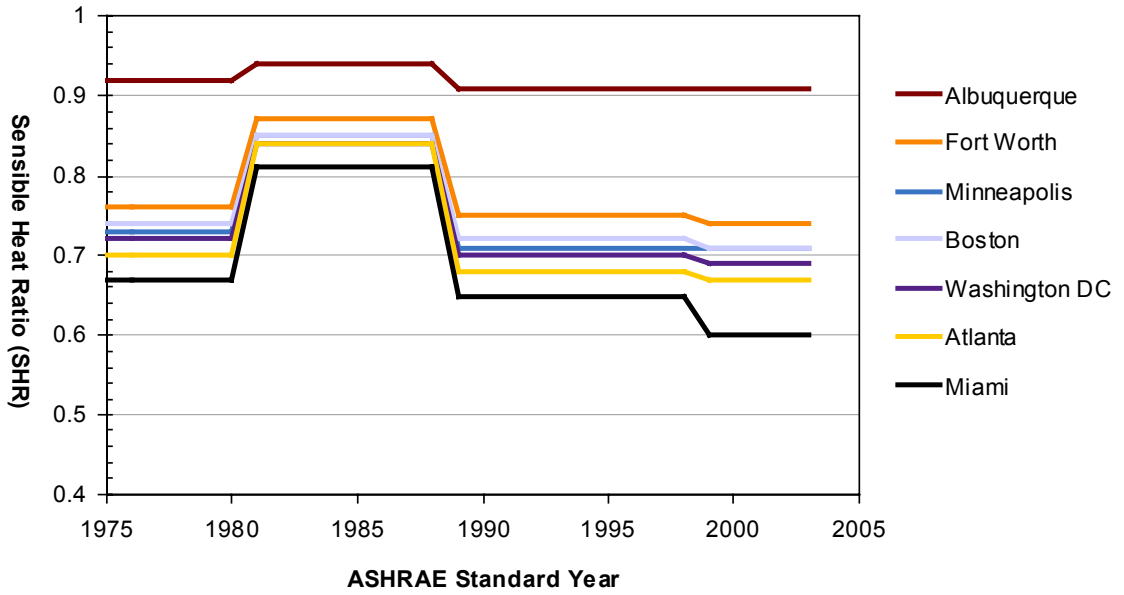


Figure E-1: Building Load SHR from 1975 – Present, at Sensible Design Condition (High Outdoor Temperature)

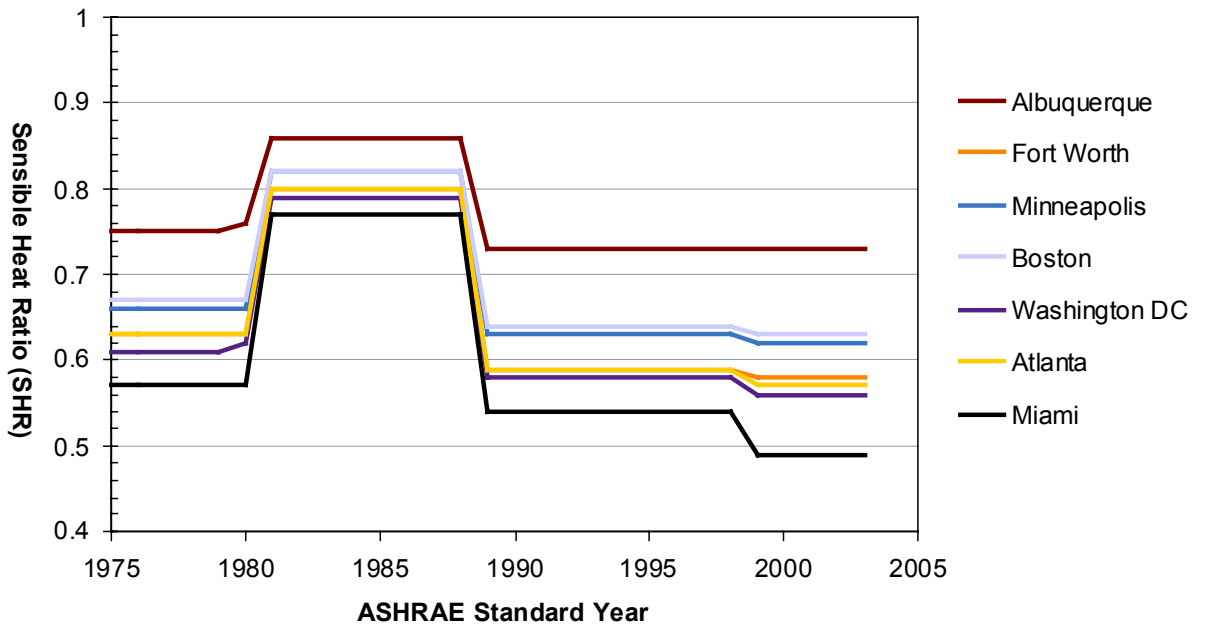


Figure E-2: Building Load SHR from 1975 – Present, at Latent Design Condition (High Outdoor Humidity)

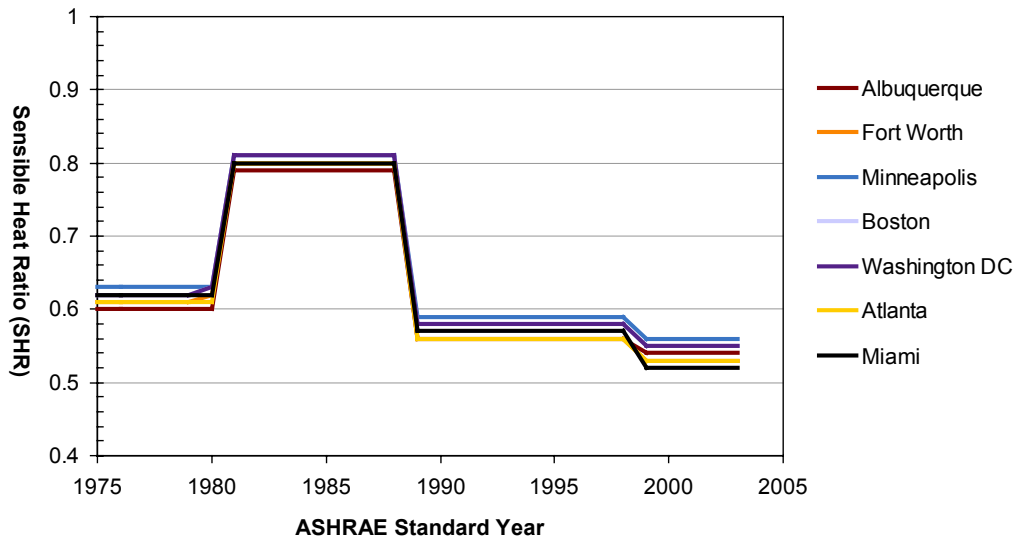


Figure E-3: Building Load SHR from 1975 – Present, at Shoulder or Part Load Condition (High Outdoor Humidity, Moderate Temperature)

The Sensible Heat Ratio of HVAC Equipment –Options for Increasing the Latent Capacity

There are a variety of options for adjusting the latent capacity of unitary air conditioning equipment. These include:

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

The Combined SHR of a Unitary Air Conditioner with Air to Air Enthalpy Recovery Ventilation (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 conditioning system with another system or component that operates with a low SHR. Examples include mechanical or desiccant dehumidifiers and enthalpy recovery ventilation systems. Unlike dehumidification systems that use new energy to reduce humidity, enthalpy recovery uses recovered

energy from the building exhaust air as the energy source thereby increasing efficiency and reducing peak electric demand. The SHR of enthalpy recovery also adjusts automatically to changing conditions; as the outdoor humidity increases, so does the latent capacity of enthalpy recovery. This characteristic allows for the introduction of additional outside air without negatively impacting the SHR balance. Figure E-4 compares the SHR of the building load (from the examples in Figures E-1 through E-3) 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 effectivenesses of enthalpy recovery are both modeled at 70%). The figure shows a close match of SHR across a range of humid weather cooling conditions.

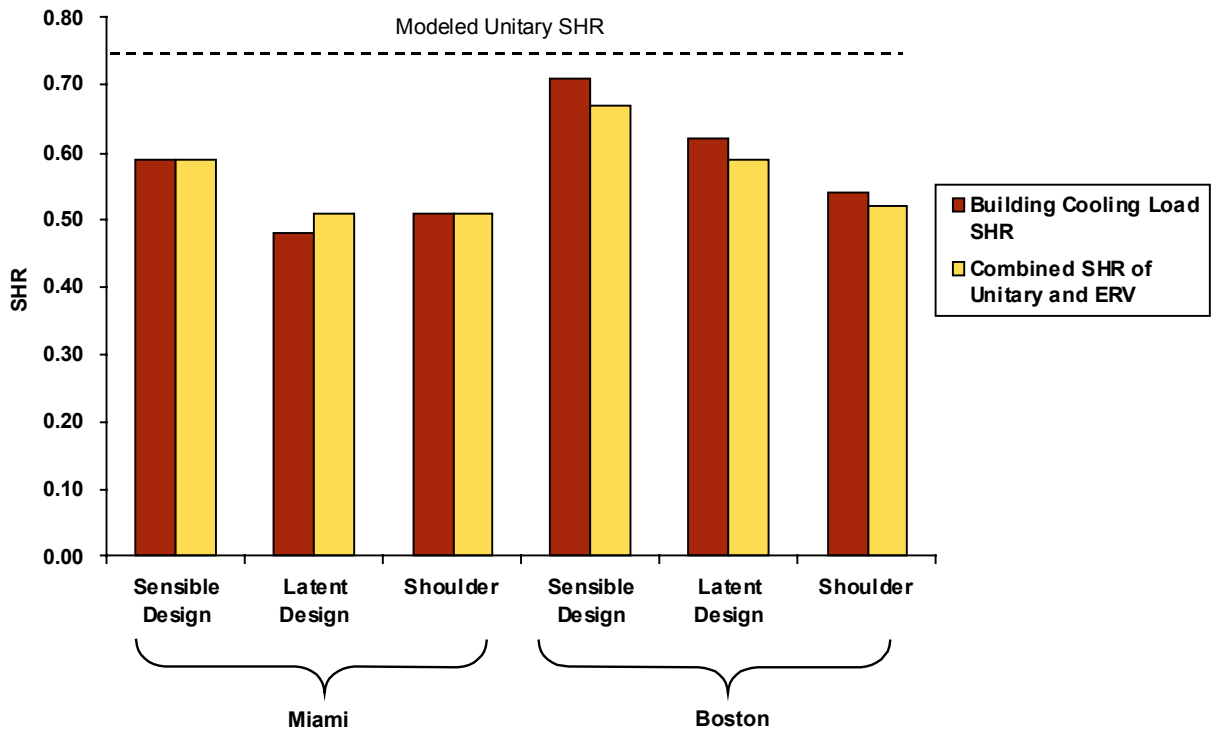


Figure E-4: Comparison of the Building SHR with the Combined Unitary +ERV SHR



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 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.

1. Introduction

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 both poor interior humidity control and liquid water sources such as plumbing leaks and rain water leakage. Moisture and mold problems can cause serious health problems among the 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 is often attributed to the inability of conventional unitary air conditioners to handle the moisture loads. A common conception holds that the increased efficiency of modern air conditioning equipment has resulted in reduced latent (moisture removal) capacity.

1.1. The Myth of the Declining Latent Capacity of Unitary Air Conditioners

The split between the sensible and latent cooling capacity of a unitary air conditioner is commonly reported as the sensible heat ratio (SHR). The SHR is the ratio of the sensible cooling capacity to the total cooling capacity. The remaining fraction, $1 - \text{SHR}$, is the latent fraction of the total cooling capacity.

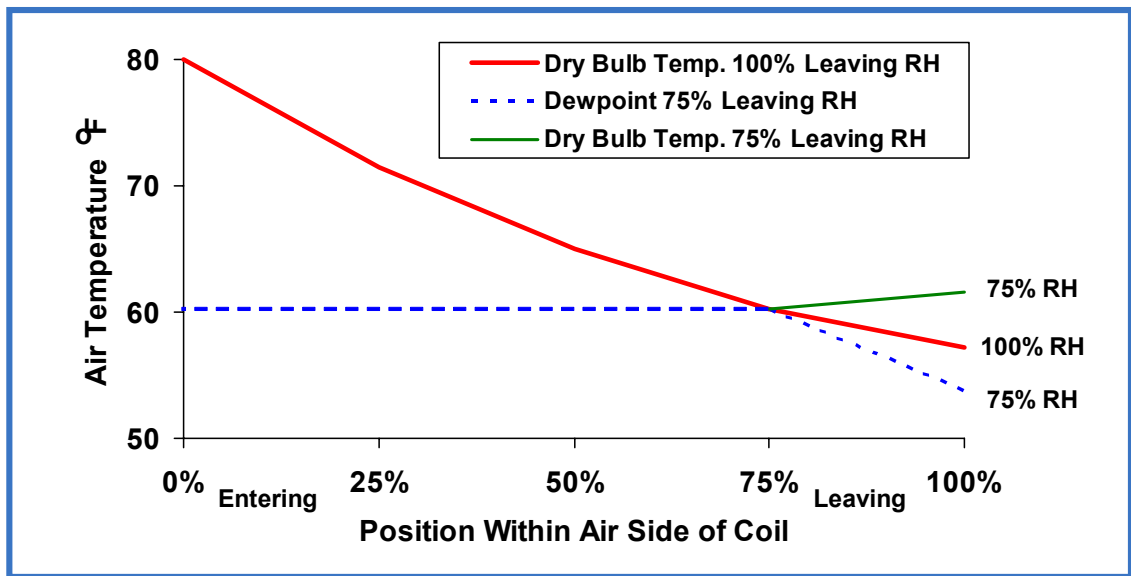
It has become a matter of “common knowledge” that the increases in the energy efficiency ratio (EER) of unitary air conditioners that have occurred since the early 1980’s have been accompanied by a decrease in the latent cooling capacity as a fraction of the total cooling capacity. The “standard” explanation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

In the real world of commercially available equipment, no such trend exists. ARI [Amrane, 2003] has recently conducted an extensive review of the relationship between EER and SHR, both for current equipment and earlier era equipment going back to the 1970s. The consistent finding is that SHRs of individual unitary air conditioner models have 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 the year of manufacture and the SHR.

In a basic sense, this should not be a particularly surprising result. At standard indoor rating conditions of 80°F DB, 67°F WB temperatures, and at a nominal air flow rate of 400 CFM/ton, latent heat removal at steady-state test conditions will fall within a fairly narrow range. Figure 1-1 illustrates the fundamental reason. If the air were cooled uniformly as it passed through the coil, it would need to be cooled down to 60.3 °F to reach the dewpoint and begin moisture condensation. If the air left the coil saturated

with water vapor (100% RH) the leaving dry bulb temperature and dewpoint would be 57.2 °F and the resulting SHR would be 0.81.

In real coils the air is not cooled uniformly as it passes through the coil. The fins are colder than the air and there is a temperature gradient between the air and the fins. As a result, moisture condensation on the finned evaporator surfaces begins before the average, or bulk, air temperature reaches the dewpoint. If the fin temperature is less than the dewpoint, moisture will condense on the fin and water vapor will diffuse through the laminar boundary layer (in parallel with sensible heat conduction through the same boundary layer). It is possible, and commonplace, for the moisture transfer to the fins to exceed the sensible heat transfer and for the bulk air to exit the coil with a dewpoint below the exit dry bulb temperature. For example, as shown in Figure 1-1, if the air exits the coil at an average relative humidity of 75%, the dry bulb temperature will be 61.6°F and the dewpoint will be 53.8°F. The corresponding SHR for 100% exit RH and 75% exit RH are 0.81 and 0.65, respectively (the same range noted by Amrane). Because sensible heat transfer and latent heat transfer are both limited by (heat) conduction and (water vapor) diffusion, respectively, through the same laminar boundary layer covering the fin surface, the divergence between sensible heat conduction and moisture diffusion falls within a relatively narrow range.



***Figure 1-1: Airstream Temperature Passing Through Evaporator Coil**

1.2. Building Humidity Control Problems are *Not* a Myth

During the last decade at least, indoor environment problems including so-called “sick building syndrome”, mold growth, and poor humidity control have become commonplace. 10% of the articles in the HVAC trade press over the past 3 years have dealt with this issue. The growing number of lawsuits arising from these problems has led builders and HVAC contractors to identify this as the largest risk element in their

business. A seminar (attended by the author) on how to address moisture problems at the January, 2003 ASHRAE Winter meeting in Chicago drew close to 500 attendees. Moisture related building problems arise from both poor interior humidity control and liquid water sources such as plumbing leaks and rain water leakage. With persistently high indoor humidity levels, damp areas do not dry out, whatever the original source of the moisture.

A variety of studies have examined the impact of ASHRAE Standard 62-1989 (the restoration of ventilation make-up air requirements to pre-1981 levels) on building humidity.

The Florida Solar Energy Center studied the impact of ASHRAE 62-1989 ventilation levels for a variety of commercial and institution building types in high humidity climates (Rengarajan, 1996; Chasar, 1998; Shirey, 1996; Davanagene, 1997). An essential conclusion of these studies is that conventional unitary air conditioning, by itself, is unable to maintain a satisfactory indoor humidity level.

An ASHRAE Research project (Brandenmuehl, 2000) analyzed humidity control performance in a variety of commercial building types, in a variety of climates. The project was premised on the observations that the increased outdoor ventilation air requirements of ASHRAE 62-1989 increase latent loads at the same time that sensible loads are decreasing due to more energy efficient building design. The study applies the concept of the sensible heat ratio of the total building cooling load. The study concludes that in many climate areas, the SHR of the total building cooling load often falls well below the SHR of conventional unitary air conditioning equipment.

Two unpublished presentations at the ASHRAE 2003 annual meeting (Henderson, 2003; Shirey, 2003) presented the results of modeled and tested dehumidification performance of unitary air conditioners operating in a cycling mode. In the commonplace operating mode of constantly running fan, with compressor on-off cycling for temperature control, condensed water left on the evaporator fin surfaces at the end of a compressor on cycle re-evaporates during the compressor off cycle. As a result, dehumidification performance is degraded to below steady state levels.

In the worst case, moisture and mold problems can cause serious health problems among the occupants and reduce productivity. In extreme cases, buildings are rendered uninhabitable, with the only remedies being to strip the building down to the frame work and start again or outright demolition of the building. The financial consequences of these situations can be disastrous for the uninsured and are reflected in rapidly escalating liability insurance rates.

1.3. The Ratio of Sensible and Latent Cooling Loads in Buildings Has Changed

If the SHR of unitary air conditioning equipment hasn't changed while building humidity problems have become more prevalent, what has changed? The answer is that

buildings have changed. The energy efficiency improvement measures — better wall and roof insulation, reduced window U-values, increased solar shading, more energy-efficient lighting — that have been driven by ASHRAE Standard 90.1 have almost exclusively reduced sensible cooling loads. Latent cooling loads, which for most commercial buildings are primarily due to ventilation, infiltration, and the occupants, have not changed substantially.

The effect of this for most buildings constructed or thoroughly remodeled since 1990 is to raise the latent cooling load relative to the sensible cooling load at all load conditions. At typical cooling design conditions, outdoor humidity levels and the consequent latent loading is quite high, while both 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 before the latent load is met, the result is that steady state indoor relative humidity increases from the desired range around 50% to a range around 70%.

When the sensible heat ratio of the air conditioning system does not match the SHR of the building, the desired building interior comfort condition cannot be met. Figure 1-2 illustrates the adjustment that occurs. The interior relative humidity increases, with the increased moisture content of the air lowering the operating SHR of the air conditioner until the latent load and moisture removal capacity of the air conditioner match. At light sensible loading – so called “shoulder” (or part load) conditions where the outdoor ambient temperature approximates 70°F – with high outdoor humidity, unitary air conditioners cycle on-off frequently with short on cycles. During the off cycle, moisture on the coil can re-evaporate, causing further deterioration of the net latent capacity, driving the RH up even higher [Henderson, 2003, and Shirey, 2003]. Note that indoor relative humidity levels need to be maintained consistently below 65% to avoid mold growth and optimum comfort is generally experienced between 40% and 50% RH.

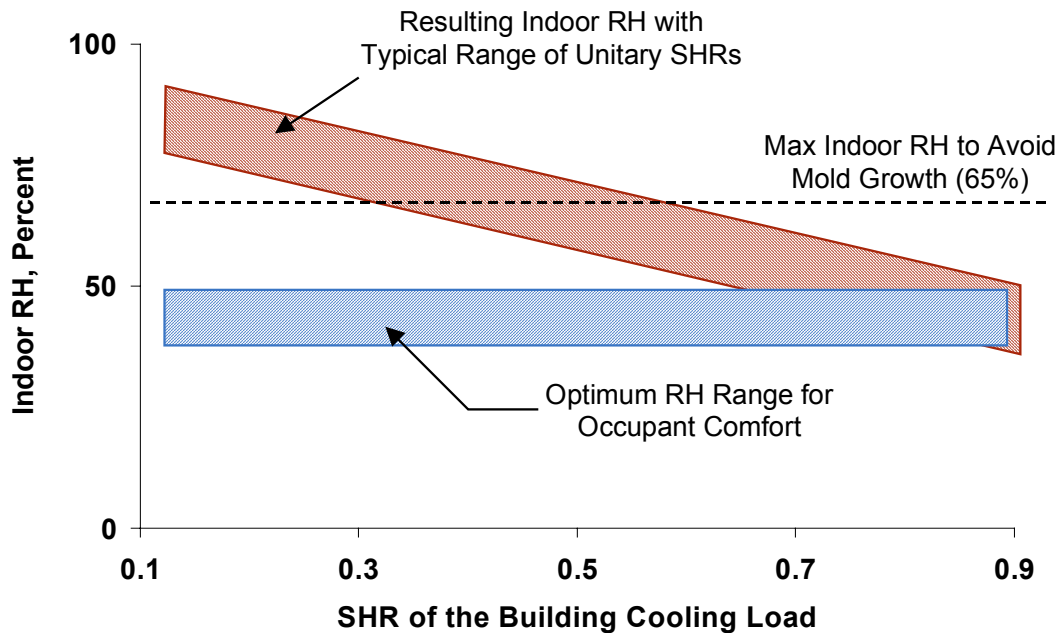


Figure 1-2: Indoor RH vs. Building Cooling Load SHR with Conventional Unitary Air Conditioning Equipment

At typical part load (65°F to 75°F ambient) conditions, the effect of reduced internal loads and reduced solar load is that the sensible cooling load is quite small. During a significant portion of the time within this temperature range, the ambient relative humidity is quite high (above 90%, 100% during periods of rainfall). The resulting latent load can be close to the latent loads at design cooling conditions, well above the sensible load. Air conditioning equipment with an SHR that is inherently between 0.65 and 0.80 at standard rating conditions simply cannot maintain a comfortable indoor relative humidity.

Section 2 of this report examines the evolution of commercial building SHR's over time.

1.4. Options for Increasing the Latent Capacity

There are a variety of options for adjusting the latent capacity of unitary air conditioning equipment. These include:

- Overcooling part of the air stream to increase the latent fraction, e.g., coil bypass or reduced air flow through the coil
- Overcooling, as above, and reheating
- Overcooling and reheating using heat transfer between the air entering and leaving the evaporator to precool and reheat, reducing energy penalties. Examples include:

- Heat exchangers in series with the evaporator coil, such as heat pipe heat exchangers
- Reheating by subcooling the liquid refrigerant, which increases the evaporator cooling capacity, so the air is overcooled in the evaporator
- Active desiccants
- Enthalpy recovery to precondition incoming outdoor make-up air

Section 3 provides a method to estimate the combined, system SHR when components are combined, specifically, enthalpy recovery added to or integrated with a unitary air conditioner.

2. The Evolving Sensible Heat Ratio of Commercial Buildings

The design of commercial buildings has evolved over the past decades, as different materials and ways of using them have become available and as the requirements for functionality and safety have evolved. In this study, the scope is limited to the small and medium commercial buildings most likely to use unitary air conditioning equipment, and the evaluation of the energy related design aspects of these buildings over the past 30 years.

By the 1970's, most, if not all, of modern commercial construction practices were well established. The structural uses of steel and concrete were well known, a wide variety of façade styles ranging from masonry to concrete to metal to glass were readily available. In all but the coldest climates in the U.S., comfort cooling air conditioning had become a given for commercial buildings.

The 50's and 60's were a period of low and declining energy prices, particularly the price of electricity. Commercial building design and construction practices reflected this economic reality. The combination of the 1973 and 1980 disruptions of Middle Eastern oil supplies and the start of the long term decline of U.S. petroleum production reversed this trend, and more significantly changed the political perception so that a wide range of policy initiatives including automobile fuel economy standards, consumer appliance efficiency standards and building efficiency standards were instituted and subsequently tightened to varying degrees.

For commercial buildings, two design and performance standards are particularly germane to the subject of how sensible and latent cooling loads have changed over the past 30 years:

- ASHRAE Standard 90.1, "Energy Standard for Buildings Except Low-Rise Residential Buildings"
- ASHRAE Standard 62, "Ventilation for Acceptable Indoor Air Quality"

The former explicitly sets minimum standards in several areas affecting building energy performance. The latter specifies minimum forced mechanical ventilation outdoor make-up air flow rates, a major source of latent cooling load and a lesser source of sensible cooling load, depending on how high the outdoor temperature and humidity are. Both standards were introduced in the 1970s. Subsequent revisions of 90.1 have steadily increased the requirements for envelope effectiveness and equipment (particularly lighting and space conditioning) efficiency. Almost all of the loads affected by 90.1 requirements are sensible loads. Minimum ventilation rates prescribed by ASHRAE Standard 62 were decreased abruptly in 1981, in response to the perceived urgency to address energy issues, then were restored to previous levels in 1989 in response to the sick building syndrome.

2.1. Sensible and Latent Cooling Loads

The cooling load of a building is the rate that heat must be removed from the building at a particular outdoor condition to maintain a desired indoor comfort condition. The total cooling load consists of sensible and latent portions. The sensible load is that portion associated with maintaining the temperature of the interior space. The latent load is the portion of the load associated with removing excess moisture from the space. It is the latent heat of vaporization of the humidity (water vapor) that must be condensed out of the air in the conditioned space to maintain the desired humidity level.

The magnitude of the sensible and latent cooling loads depends on the desired interior comfort condition – the dry bulb temperature and relative humidity – that is to be maintained. Reducing either the desired temperature or RH increases the cooling load. ASHRAE Standard 55 is a distillation of research on requirements for thermal comfort. It defines a range of temperature and relative humidity that is generally acceptable to occupants of the conditioned space. A common air conditioning season comfort setpoint is 75°F and 50% RH, which is in the lower portion of the ASHRAE 55 comfort range.

There are a variety of individual contributions to the sensible cooling load of a typical commercial building:

- Outdoor ventilation make-up air (temperature in excess of the 75°F \pm level generally maintained indoors for comfort) carries sensible heat into the conditioned space
- Infiltration of outdoor air through leaks in the building envelope also carries sensible heat into the conditioned space
- Thermal conduction through the building envelope – walls, roofs, windows, bottom floors
- Solar load through windows
- Internally generated heat loads
 - Lights
 - Ventilation blower motors
 - Other motor loads – e.g., elevators
 - Office equipment – computers, photocopiers, printers, telephone, etc.
 - Other miscellaneous “plug” loads
 - Human occupants

The latent load consists of all of the sources that add water vapor to the building conditioned space. Individual contributions to the latent cooling load of a typical commercial building include:

- Outdoor ventilation make-up air (humidity in excess of the 50% level generally maintained indoors for comfort and IAQ purposes) carries excess humidity into the conditioned space
- Infiltration of outdoor air through leaks in the building envelope also carries excess humidity into the conditioned space
- Moisture permeation through the building envelope (generally insignificant)
- Human occupants – via respiration and perspiration
- Indoor plants (generally insignificant)
- Commercial cooking equipment, if applicable

Most of these contributors to the sensible and latent load can be controlled by design and a corresponding investment in better building materials and components – e.g., insulation, windows, – and better construction practices – e.g. leak-tight envelope assembly.

2.2. Changing Building Energy Efficiency and Ventilation Standards – ASHRAE Standard 90.1 and ASHRAE Standard 62

ASHRAE standards 62 and 90.1 were created to set minimum standards for building ventilation and for energy related aspects of building design, respectively. They were initially introduced in the early 1970's. Since then, major revisions have been published at regular intervals (listed here in reverse chronological order).

- ASHRAE Standard 62-2001 *Ventilation for Acceptable Indoor Air Quality*
 - 62-1999 *Ventilation for Acceptable Indoor Air Quality*
 - 62-1989 *Ventilation for Acceptable Indoor Air Quality*
 - 62-1981 *Ventilation for Acceptable Indoor Air Quality*
 - 62-1973 *Standards for Natural and Mechanical Ventilation* (original version of this standard)
- ASHRAE Standard 90.1-2001 *Energy Standard for Buildings Except Low-Rise Residential Buildings*
 - 90.1-1999 *Energy Efficient Design of New Buildings Except Low-Rise Residential Buildings*
 - 90.1-1989 *Energy Efficient Design of New Buildings Except Low-Rise Residential Buildings*
 - 90A-1980 *Energy Conservation in New Building Design*
 - 90-1975 *Energy Conservation in New Building Design* (original version of this standard)

The net effect, over time, of revisions to these standards is discussed below.

2.2.1. Building Envelope

ASHRAE 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 90.1 defined climate zones. Instead, the maximum 90.1 U-Value can be tracked for particular cities, as the successive versions of 90.1 apply to the specific climate characteristics (e.g. design heating degree-days and cooling degree-days) of each city. Figure 2-1 plots the ASHRAE 90.1 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 3 to 4, which reduces the wall conduction contribution to the sensible load by a like amount.

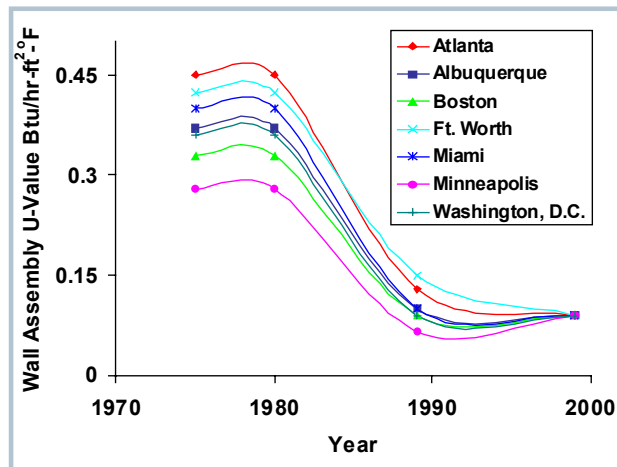


Figure 2-1: ASHRAE 90.1 Maximum Wall Assembly “U” Values, Over Time

Figure 2-2 plots the ASHRAE 90.1 maximum roof U-Value over the period during which ASHRAE 90 has been in effect. Modest (~30%) reductions in the maximum U-Value have occurred during this time period.

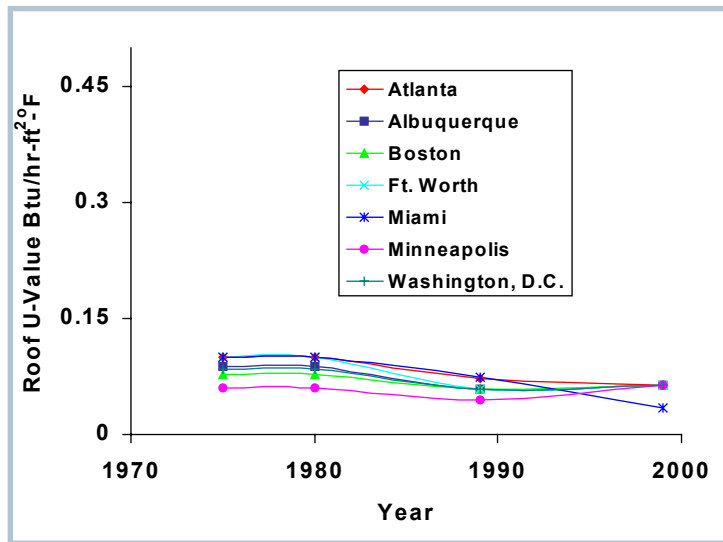


Figure 2-2: ASHRAE 90.1 Maximum Roof "U" Values Over Time

Figure 2-3 plots the maximum solar heat gain coefficient for window glazings over the life of ASHRAE 90. It controls the fraction of incident solar radiation that enters the conditioned space.

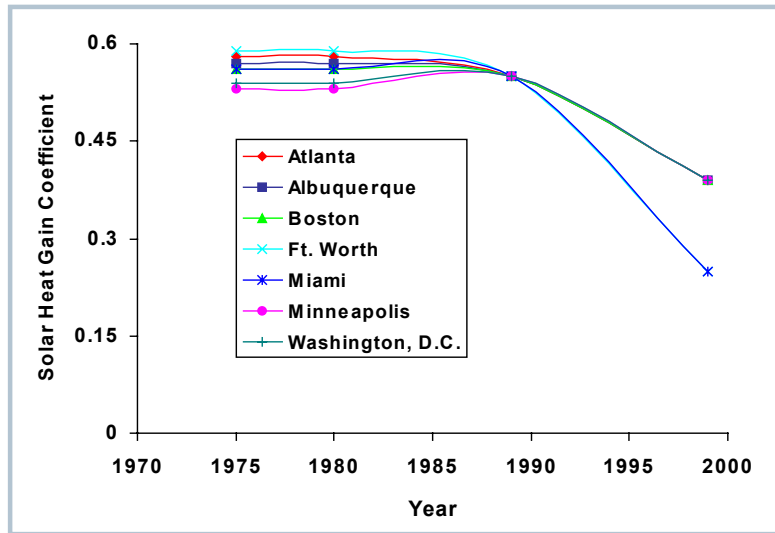


Figure 2-3: ASHRAE 90.1 Maximum Solar Heat Gain Coefficient Over Time

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.

2.2.2. Equipment Loads

Internal heat gains from energy dissipated by equipment within the building is a significant fraction of the building cooling loads (and contributes to heating the building during the cold periods of the year). ASHRAE 90.1 specifically controls two of the

contributors to internal heat gain – lighting and motors. Minimum motor efficiencies are currently set at the EAct 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 2-4. It has been reduced by 40% since the first version of ASHRAE 90. ASHRAE 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. Estimated office equipment power densities are also plotted in Figure 2-4. As is the case with envelope requirements, the ASHRAE 90 requirements to reduce internal loads addresses sensible load only.

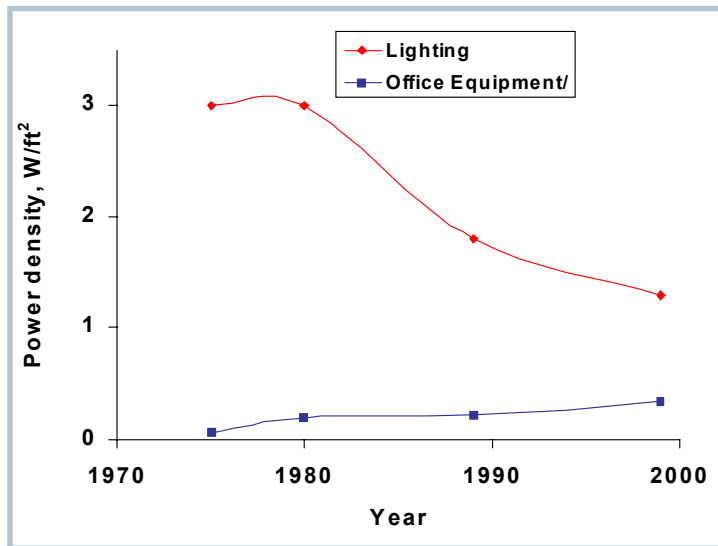


Figure 2-4: ASHRAE 90.1 Maximum Lighting Power Density (for Office Space) and Estimated Office Equipment Power Density

2.2.3. Ventilation and Infiltration

Outdoor air enters the conditioned space of a building (displacing conditioned indoor air in the process) via two basic processes – mechanical ventilation and infiltration. Mechanical ventilation is the deliberate forced draft supply of outdoor air to the conditioned space by motor driven blowers. ASHRAE Standard 62 specifies minimum outdoor air ventilation rates. Air also enters via infiltration – the uncontrolled flow of air into the conditioned space through leaks in the building envelope. Infiltration air flows are driven by natural convection (difference in the buoyancy of indoor and outdoor air, sometimes referred to as stack effect, a more significant factor in tall buildings), wind, and pressure differentials caused by the building HVAC system.

Table 2-1 provides a sampling of the minimum mechanical ventilation air flow requirements in ASHRAE 62, for a representative selection of types of commercial space. The minimums in Table 2-1 are in terms of CFM/person. As a practical matter, ventilation is supplied to an interior space, so CFM/person needs to be related to the size of the interior space. The maximum design occupancy density is used to make this connection. Applying the minimum CFM/person and the maximum occupancy density

together yields the minimum required CFM per square foot, provided in Table 2-2 for the same selection of occupancy types as Table 2-1. Based on either of these tables, it is

Table 2-1: Changing Minimum Outdoor Make-Up Air Ventilation Rates Per ASHRAE Standard 62

Version of ASHRAE 62	Minimum Required Outdoor Air Per Person in CFM (non-smoking)				
	Office Space	Meeting Space	Retail Space	Classrooms	Dining Rooms
62 - 2001	20	20	10*	15	20
62 - 1999	20	20	10*	15	20
62 - 1989	20	20	10*	15	20
62 - 1981	5	7	5	5	7
62 - 1973	15; (15-25)**	25; (33-40)**	7; (10-15)**	10; (10-15)**	10; (15-20)**

*At estimated maximum occupancy at specified CFM/ft²

** (Recommended)

Table 2-2: Required CFM/ft², Based on Estimated Maximum Occupancy and Required CFM/Person

Version of ASHRAE 62	Required Outdoor Ventilation Air CFM/ft ²				
	Offices	Conference Room	Retail/(Street Level)	Classroom	Dining Room
62 - 2001	0.14	1.0	0.3	0.75	1.4
62 - 1999	0.14	1.0	0.3	0.75	1.4
62 - 1989	0.14	1.0	0.3	0.75	1.4
62 - 1981	0.035	0.3	0.15	0.25	0.49
62 - 1973	0.15	1.5	0.21	0.50	0.70

evident that maximum ventilation rates were decreased dramatically in 1981, in response to the perceived energy crisis. Then minimums were increased to somewhat higher than original levels in 1989, this time in response to widespread reports of “sick-building syndrome”.

Infiltration levels depend on the integrity of the building envelope. The leak-tightness of existing buildings varies widely, as discussed, for example by (Persily, 1998).

ASHRAE Std 90.1 requires sealing the building envelope to minimize leakage. Over time the trend has been to require tighter envelopes.

The impact of outside air (due to either leakage or mechanical ventilation) on latent and sensible loads depends on the outdoor temperature and humidity. In most climate regions in the U.S., outdoor air tends to bring a larger amount of latent load than sensible load, so the 1989 increase in mechanical ventilation requirements tended to increase latent loads at the same time that 90.1 requirements were significantly reducing sensible loads.

2.3. Building Cooling Load Calculations for a Range of Climate Zones

The individual changes in U-Values, lighting power density, and ventilation affect individual contributors to the total building sensible and latent cooling loads. To evaluate the total impact, a hypothetical medium sized (~50,000 sq. ft.) office building was designed, as shown in Figure 2-5.

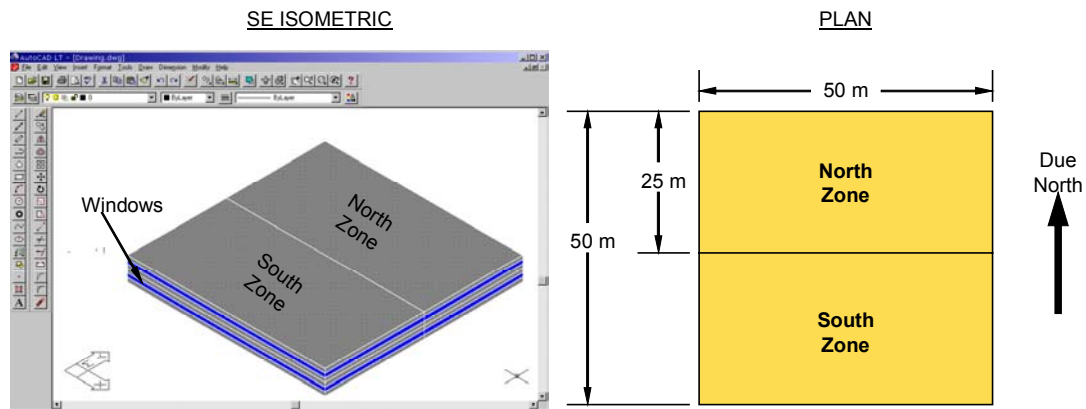


Figure 2-5: Model Office Building Configuration

As indicated in Figure 2-5, the model office building has two stories with a square footprint, 50 m (164 ft) on a side. The building has 4 space conditioning zones, the north and south halves of each floor. The breakdown of floorspace of the building was assumed to be 80% offices, 10% conference rooms, and 10% reception/lobbies. Table 2-3 summarizes the calculation of the average ventilation per square foot required by each revision of ASHRAE 62.

The DOE EnergyPlus building energy simulation software was used to calculate cooling loads for the model office building. As discussed below, the levels of U-values, internal load, and ventilation rates specified by ASHRAE Standards 62 and 90.1 as they evolved over time were systematically applied to the EnergyPlus model of the office building to evaluate the total impact of these changes.

Table 2-3: Weighted Average Ventilation per Floorspace Required for Office Building

Version of ASHRAE 62	Required Outdoor Ventilation Air, CFM/ft ²			
	Offices	Conference Room	Reception/Lobby	Weighted Average*
62 - 2001	0.14	1.0	0.9	0.292
62 - 1999	0.14	1.0	0.9	0.292
62 - 1989	0.14	1.0	0.9	0.292
62 - 1981	0.035	0.30	0.15	0.073
62 - 1973	0.15	1.5	0.30	0.30

*Based on 80% offices, 10% conference rooms, 10% reception/Lobbies

2.3.1. The Evolution of Envelope and Lighting Specifications Over Time

Table 2-4 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 2-4.

2.3.2. Analysis Results

Typical results of an EnergyPlus design simulation are shown in Figures 2-6 and 2-7. The ASHRAE 62 ventilation rate is applied 24 hr/day, the result of interest is generally the design condition during occupied daytime hours. Both plots are for the first floor north zone of the hypothetical office building in Miami, Figure 2-6 is for the standards in place from 1981-1989, Figure 2-7 for 1989-1999. Note the significant drop in sensible load and the large increase in latent load, from pre-1989 to post 1989.

Figures 2-8 through 2-10 summarize the calculated SHR for the office building described in Section 2.3, when designed for and located in each of seven representative cities. 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 outdoor conditions are summarized in Table 2-5 for each of the seven climate cities. Each of the three plots shows the SHR that would be obtained for the building as a whole (the average of 4 zones), with the building designed to meet the ASHRAE 90.1 energy performance and the ASHRAE 62 ventilation requirements that were in effect that year.

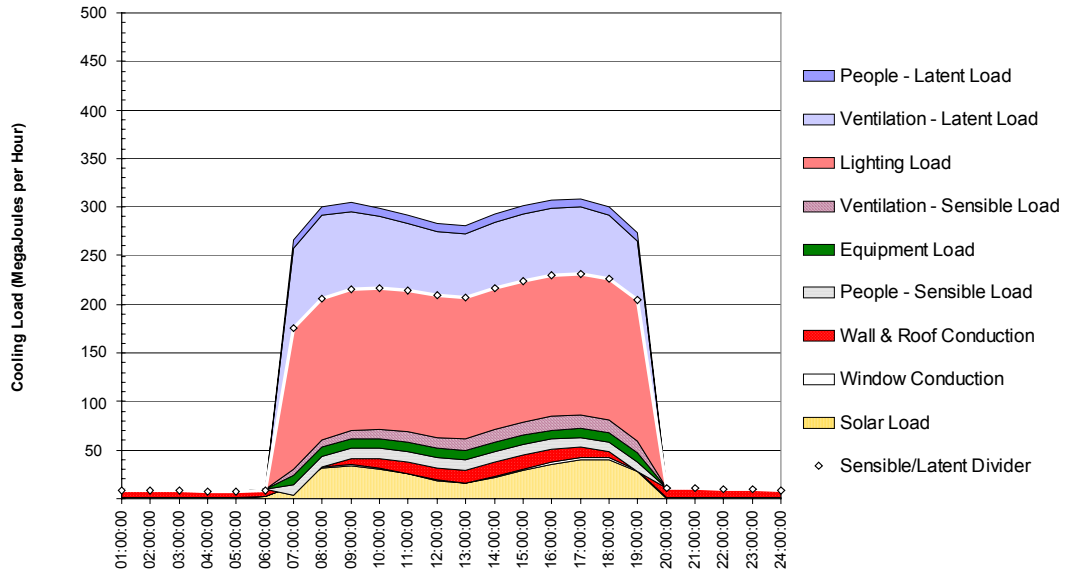


Figure 2-6: Area Plot of Basic Constituents of the Sensible and Latent Cooling Loads – Miami / Latent Design Day / 1st Floor North Zone / 1981-1989

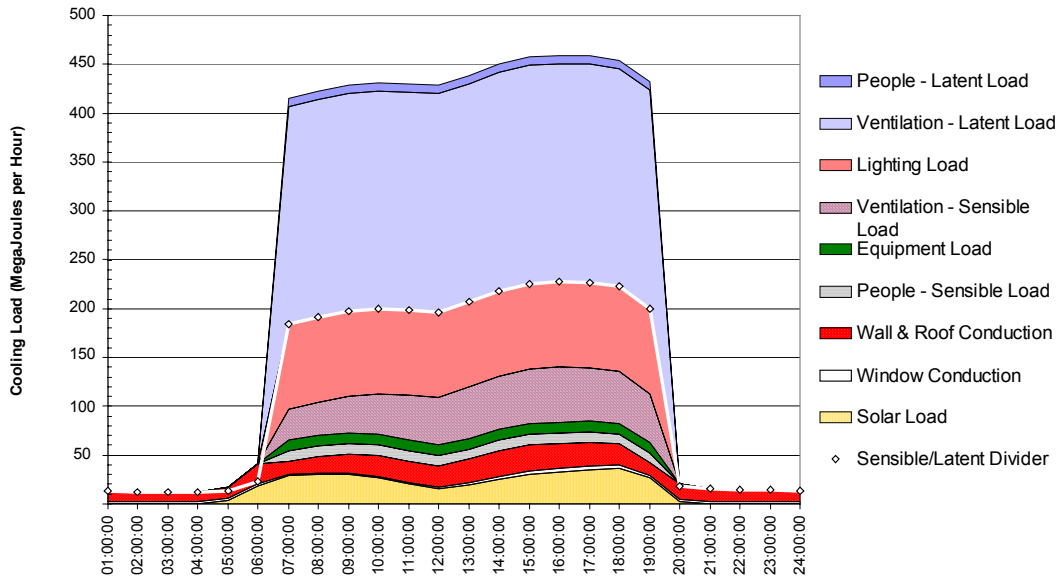


Figure 2-7: Area Plot of Basic Constituents of the Sensible and Latent Cooling Loads – Miami / Latent Design Day / 1st Floor North Zone / 1989-1999

Table 2-4: Input Values of the Envelope

Specification	Years	Value for Climate Location						
		Atlanta	Albuquerque	Boston	Ft. Worth	Miami	Minn., MN	Wash., D.C.
Wall Ass'y U-Value Btu/hrft ² F	1975-1989	0.45	0.37	0.33	0.425	0.4	0.28	0.36
	1989-1999	0.13	0.1	0.089	0.15	1.0	0.065	0.089
	1999-	0.089	0.089	0.089	0.089	0.089	0.089	0.089
Roof U-Value Btu/hrft ² F	1975-1989	0.1	0.088	0.078	0.1	0.1	0.06	0.084
	1989-1999	0.072	0.059	0.058	0.058	0.074	0.045	0.058
	1999-	0.063	0.063	0.063	0.063	0.034	0.063	0.063
Solar Heat Gain Coefficient	1975-1989	0.58	0.57	0.56	0.59	0.56	0.53	0.54
	1989-1999	0.55	0.55	0.55	0.55	0.55	0.55	0.55
	1999-	0.39	0.39	0.39	0.25	0.25	0.39	0.39

Table 2-5: Design Conditions

Design Day	Design Temps	Temperature for Climate Locations, °F						
		Atlanta	Albuquerque	Boston	Ft. Worth	Miami	Minn., MN	Wash., D.C.
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	60	75	79	80	76	79
HDD 65		2,991	4,425	5,641	2,304	200	7,981	4,707
CDD50		5,038	3,908	2,897	6,557	9,474	2,680	3,709

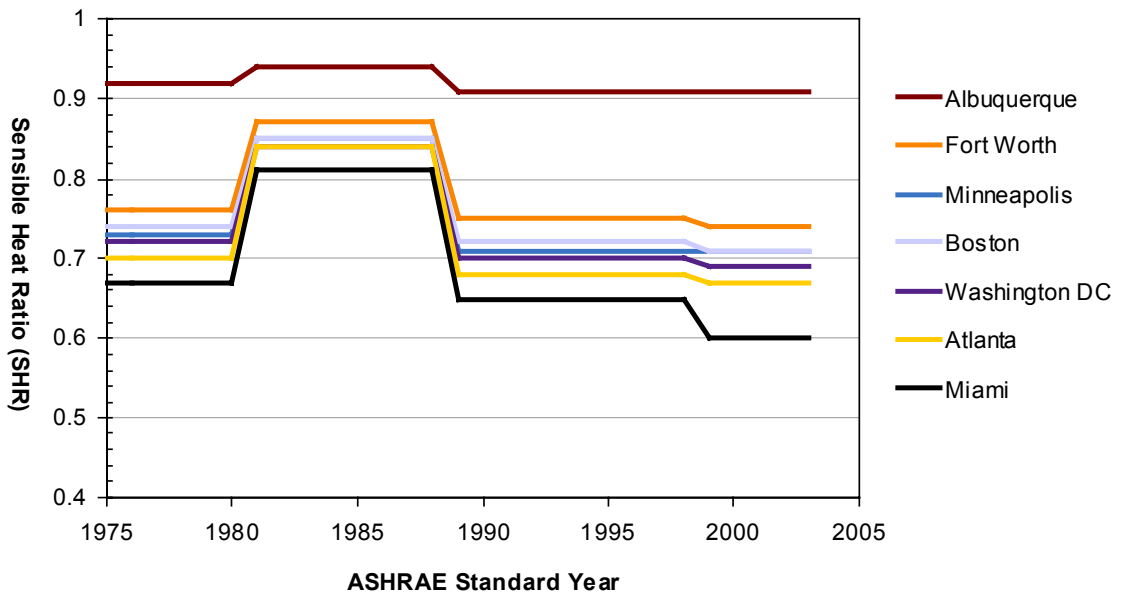


Figure 2-8: Building Load SHR from 1975 – Present, at Sensible Design Condition (High Outdoor Temperature)

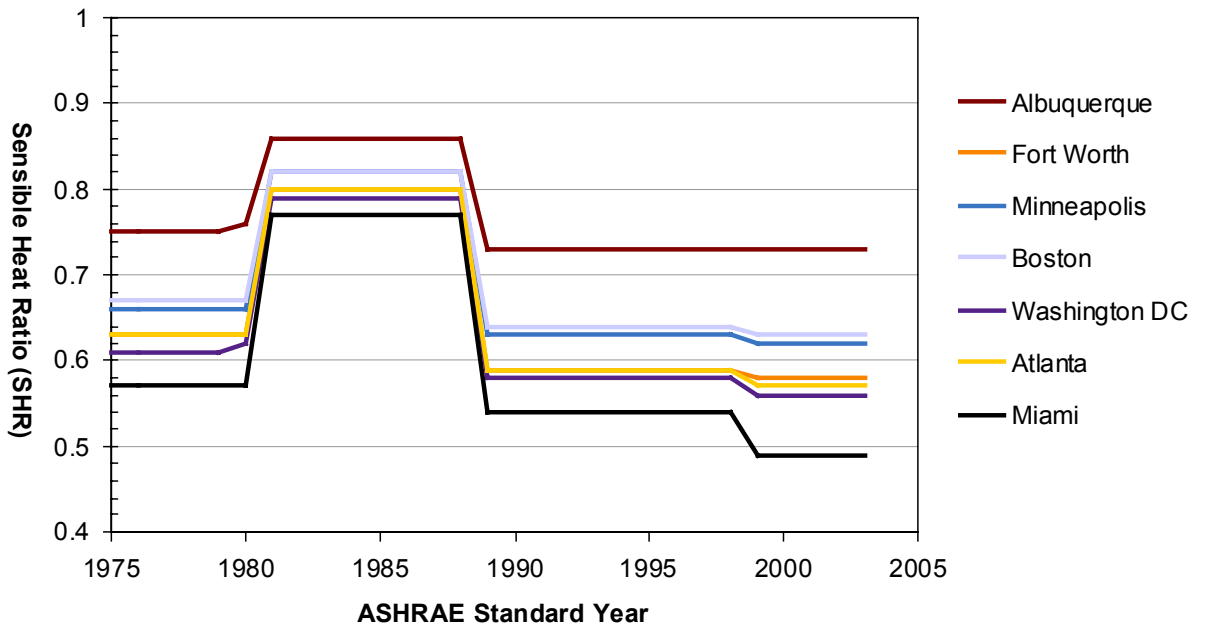


Figure 2-9: Building Load SHR from 1975 – Present, at Latent Design Condition (High Outdoor Humidity)

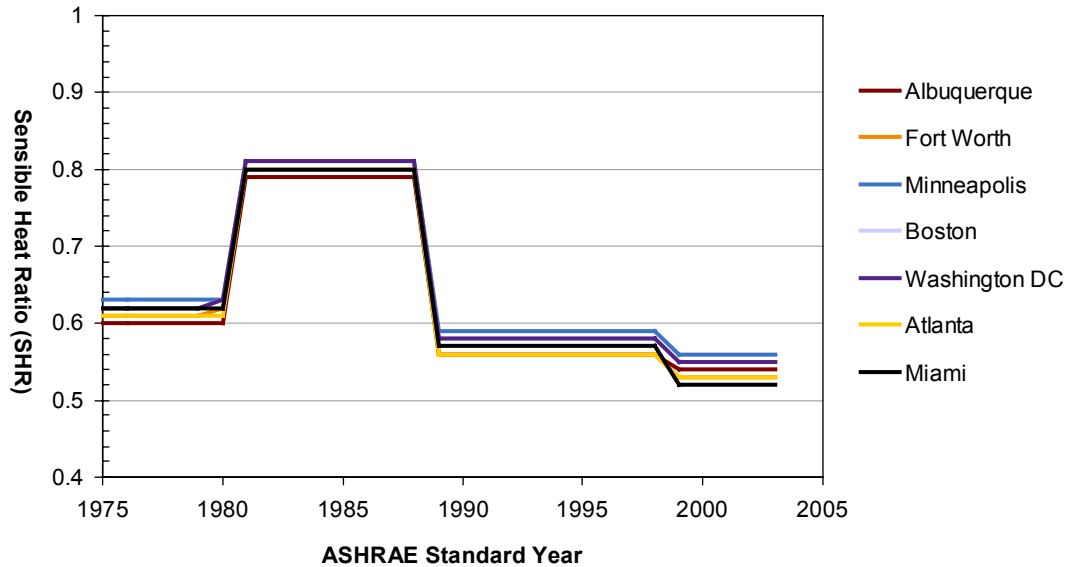


Figure 2-10: Building Load SHR from 1975 – Present, at Shoulder or Part Load Condition (High Outdoor Humidity, Moderate Temperature)

Some observations from/about Figures 2-8 through 2-10:

- Except in dry climates, (e.g. Albuquerque) the combined effect of the 1989 revisions to ASHRAE Standards 62 and 90.1 were to dramatically reduce the sensible heat ratio (i.e., increase the latent ratio) of the building cooling load, to levels below the normal range of unitary air conditioner SHR levels.
- When the outdoor ambient temperature is in the vicinity of 70°F, with high outdoor dewpoint, the SHR and the total cooling load fall to particularly low values.

A set of detailed results for all 7 climate areas is provided in the separate Appendix Volume. For each climate area (city), the following is provided:

- A table summarizing the design conditions
- Plots of SHR vs. time for each zone, at each of the three design conditions
- A bar chart giving the design day loads, for each zone at each of the three design conditions and for each time frame.

3. The System Sensible Heat Ratio of Combinations of HVAC Equipment

It is evident from the preceding section that conventional unitary air conditioning equipment by itself is not capable of maintaining indoor relative humidity close to 50% across the range of conditions normally encountered. A mismatch in the SHR of the building load and the SHR of the cooling equipment can be addressed by:

- Using the building and equipment as designed and allowing the relative humidity to adjust while the interior temperature set point is met. When the SHR of the building load is much less than the SHR of the cooling equipment, higher indoor humidity will result. Often this will be unsatisfactory because high indoor humidity provides poor comfort and encourages mold growth.
- Modifying the design of the HVAC system to better match its SHR to the building load SHR across the anticipated range of operating conditions, enabling the indoor RH to be maintained within the desirable range close to 50%.

Two basic approaches to lowering the cooling system SHR to match the building SHR are:

- Internally modify the unitary air conditioner to reduce the SHR and provide for control of the SHR to meet the instantaneous sensible and latent loads
- Add additional components or equipment to the basic unitary air conditioner to reduce and control the SHR

This section addresses the latter of these two basic approaches providing a method to estimate the combined SHR of a combination of pieces of HVAC equipment and components. Specific focus is then given to the combination of a unitary air conditioning system with enthalpy recovery ventilation, with a method to estimate the combined SHR and examples that show the effectiveness of this combination in following the SHR of the building cooling load as it varies with outdoor conditions. Unlike dehumidification systems that use new energy to reduce humidity, enthalpy recovery uses recovered energy from the building exhaust air as the energy source thereby increasing efficiency and reducing peak electric demand. The SHR of enthalpy recovery also adjusts automatically to changing conditions; as the outdoor humidity increases, so does the latent capacity of enthalpy recovery.

3.1. System SHR – General Case

When a building or a distinct interior zone of a building is cooled by more than one piece of HVAC equipment, the combined SHR is the sum of the sensible cooling capacity from each separately rated HVAC system and component divided by the sum of the total cooling capacities of these systems and components:

$$\text{Combined SHR} = \frac{\text{SensCap}_1 + \text{SensCap}_2 + \dots + \text{SensCap}_n}{\text{Cap}_1 + \text{Cap}_2 + \dots + \text{Cap}_n} \quad (1)$$

Or

$$\text{Combined SHR} = \frac{\text{SHR}_1\text{Cap}_1 + \text{SHR}_2\text{Cap}_2 + \dots + \text{SHR}_n\text{Cap}_n}{\text{Cap}_1 + \text{Cap}_2 + \dots + \text{Cap}_n} \quad (2)$$

The individual capacities and SHRs can often be determined from manufacturer’s equipment performance ratings for the specified conditions.

3.2. System SHR – Unitary Air Conditioner Combined with an Air to Air Enthalpy Recovery Exchanger

A system/component combination of interest is an energy recovery heat exchanger integrated with a unitary air conditioner, as shown in Figure 3-1. This combination may be a fully factory integrated unit, a unitary air conditioner with an add-on accessory ERV, or a separate unitary air conditioner and ERV make-up air unit installed in the same building. The capacity and SHR of Unitary air conditioners are rated at standard conditions. Many manufacturers also provide performance ratings for a range of outdoor air and return air conditions. ARI-1060, the current rating standard for ERV provides for determination of the sensible heat, latent heat, and total energy transfer effectivenesses, but not for capacities at a given set of conditions.

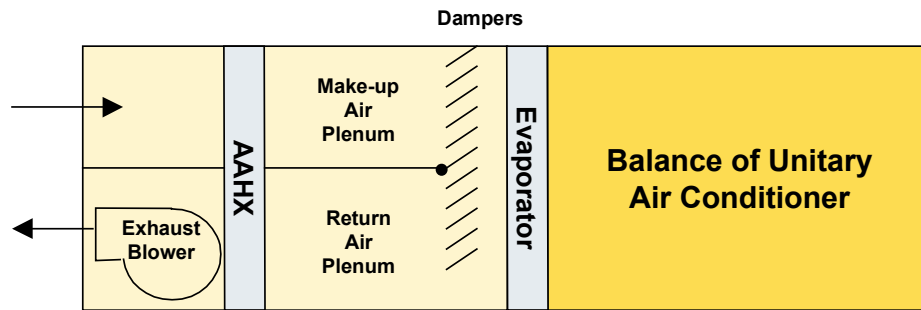


Figure 3-1: Unitary Air Conditioner with a Factory Integrated AAHX

The recovered cooling capacity of an ERV, referred to as the net recovery capacity or NRC is

$$\text{NRC} = \dot{m} \epsilon_{\text{total}} \Delta h_{\text{total}}, \text{ where}$$

\dot{m} = mass flow rate of ventilation air

ϵ_{total} = total enthalpy transfer effectiveness

Δh_{total} = enthalpy difference between outdoor air and return air

To be precise, ϵ_{total} should be the value at the particular condition of interest. The value determined at ARI-1060 standard conditions (a high humidity load condition) is a very

good approximation for the comparatively humid conditions that are the primary concern in the context of this study, and these can be used with little impact on the accuracy of calculations.

The sensible capacity of the ERV is

$NRC_{sens} = \dot{m} \epsilon_{sens} \Delta h_{sens}$, where where Δh_{sens} = sensible enthalpy difference between outdoor air and return air, i.e., $c_p \Delta T$

ϵ_{sens} = sensible enthalpy transfer effectiveness

The SHR of the ERV is

$$NRC_{sens} / NRC = \dot{m} \epsilon_{sens} \Delta h_{sens} / \dot{m} \epsilon_{total} \Delta h_{total} \quad (3)$$

$$= \frac{\epsilon_{sens} \Delta h_{sens}}{\epsilon_{total} \Delta h_{total}} \quad (3a)$$

The first part of (3a), $\epsilon_{sens}/\epsilon_{total}$, is a performance characteristic of the ERV. The second part of (3a), $\Delta h_{sens}/\Delta h_{total}$ is the sensible heat ratio of the enthalpy difference between outdoor and indoor conditions and will be referred to as SHR_{vent}

$$SHR_{vent} = \Delta h_{sens} / \Delta h_{total} \quad (4)$$

Thus, the SHR of an ERV is

$$\epsilon_{sens} / \epsilon_{total} SHR_{vent} \quad (4a)$$

Equation (4a) is a simple mathematical statement of an important attribute of enthalpy recovery heat exchangers—the SHR of the ERV tracks the SHR of the ventilation load, since $\epsilon_{sens}/\epsilon_{total}$ is constant (or nearly so) for most enthalpy recovery heat exchangers on the market.

The combined unitary + ERV SHR is

$$\text{Combined SHR} = \frac{\frac{\epsilon_{sens}}{\epsilon_{total}} SHR_{vent} NRC + SHR_{unitary} \text{ Unitary Capacity}}{NRC + \text{Unitary Capacity}} \quad (5)$$

3.2.1. Example Calculation of Combined SHR for Unitary and ERV

Several sample calculations of the combined SHR are summarized in Table 3-1 for the case where the indoor conditions are set at 75/63 and the unitary air conditioner operates at 400 CFM/ton of nominal capacity with an SHR of 0.75. Estimates are

provided for three levels of ventilation make up-air flow rates and at two outdoor temperatures.

Table 3-1: Combined SHR Calculation

Outdoor DB/WB Temperature	Ventilation Air flow ÷ Total	Unitary		ERV $\epsilon=0.7^*$		Combined SHR
		Capacity Tons	SHR	NRC Tons	SHR	
95/78	.2	10	.75	2.69	.375	.67
	.5	10	.75	6.73	.375	.60
	1.0	10	.75	13.47	.375	.53
75/74	.2	10	.75	1.93	0	.63
	.5	10	.75	4.82	0	.51
	1.0	10	.75	9.64	0	.38

*Assumed $\epsilon_{tot} = \epsilon_{sens} = \epsilon_{lat} = 0.7$

3.2.2. Illustrative Example – the Example Office Building of Section 2.3

In Section 2, the essence of the sensible heat ratio issue was shown to be two-fold:

- Overall, the trend to more energy efficient commercial buildings has reduced sensible loads, while latent loads have remained constant, reducing the SHR of the building cooling load at any given condition.
- It has always been the case that at moderate temperature, high humidity outdoor conditions, the SHR of the building load will be lower than it would be at design ambient conditions, often well below the SHR of conventional unitary air conditioning equipment.

To illustrate the effectiveness of the combination of enthalpy recovery ventilation and conventional unitary air conditioning equipment in matching the cooling system SHR to the building SHR, the building SHR’s calculated in Section 2.3 at sensible design conditions, latent design conditions, and humid shoulder day conditions can be compared to the combined SHR of unitary and ERV sized to meet the design load. Tables 3-2 and 3-3 make this comparison for the model building in Section 2.3

- For each of the three conditions addressed in Section 2 – sensible design, latent design, and humid shoulder – in Miami and Boston
- For building envelope and equipment performance meeting ASHRAE 90.1 –1999
- For outdoor ventilation air flow rates meeting ASHRAE 62-1999
- With a unitary air conditioner whose SHR is 0.75 when the indoor RH is 50%.
- The underlying analysis is summarized in Section 3.2.4

Table 3-2, which summarizes the results for the office building in Miami, shows that the low SHR of the ventilation load accounts for most of the total humidity load. This observation applies to each of the three design conditions that were examined, even the “sensible” (high ambient dry bulb temperature) design conditions. Using an ERV to

precondition the outdoor ventilation air adds significant latent capacity to the combined unitary + ERV cooling system, so the combined SHR comes very close to matching the SHR of the building load.

Table 3-2: Comparison of the Building Cooling Load SHR with the Combined Unitary + ERV SHR in Miami Climatic Conditions

Design Day Type	Ambient Temperature		Building Cooling Load SHR			Unitary + ERV SHR
	DB, °F	WB, °F	Envelope +Internal	Ventilation	Total	
Sensible	91	77	0.95	0.31	0.59	.588
Latent	87	80	0.95	0.19	0.48	.506
Shoulder	75	72	0.94	0	0.51	.506

Table 3-3, for Boston shows similar matching of cooling system SHR to building load SHR when an ERV is used in combination with unitary air conditioning. While Boston is less humid than Miami, outdoor ventilation make-up air still accounts for the majority of the humidity load at each of the three design conditions.

Table 3-3: Comparison of the Building Cooling Load SHR with the Combined Unitary + ERV SHR in Boston Climatic Conditions

Design Day Type	Ambient Temperature		Building Cooling Load SHR			Unitary + ERV SHR
	DB, °F	WB, °F	Envelope +Internal	Ventilation	Total	
Sensible	91	73	0.96	0.42	0.71	.668
Latent	87	75	0.95	0.28	0.62	.588
Shoulder	75	72	0.95	0	0.54	.518

The results in Tables 3-2 and 3-3 are summarized in bar chart format in Figure 3-2.

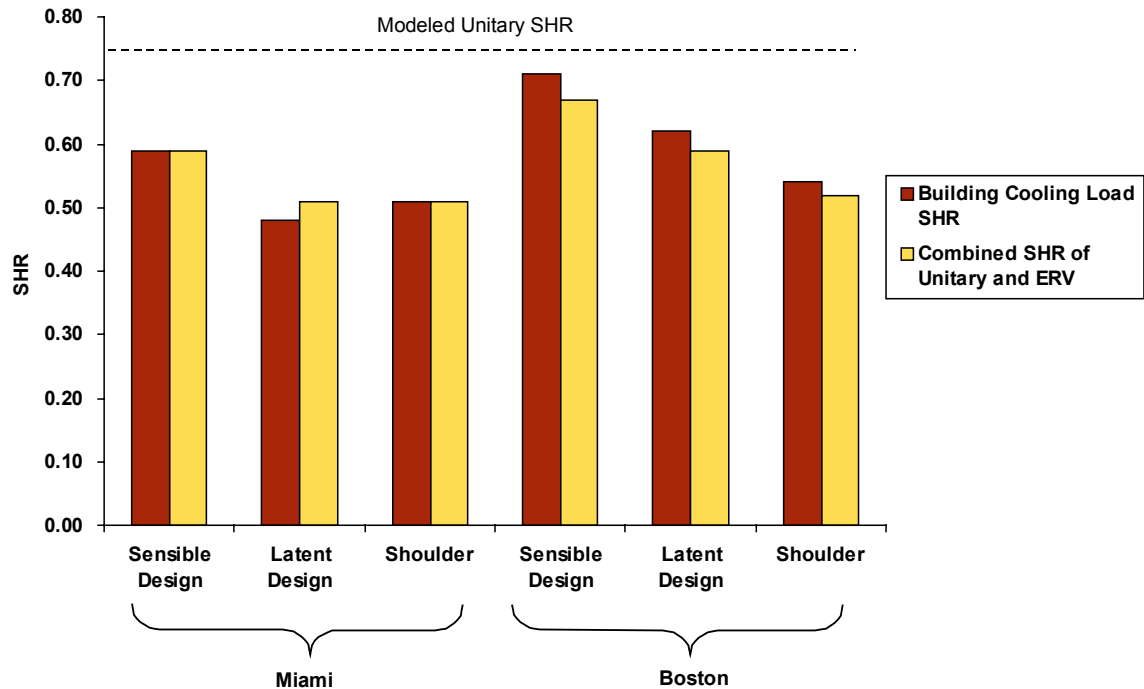


Figure 3-2: Comparison of the Building Cooling Load SHR with the Combined Unitary + ERV SHR in Miami and Boston Climatic Conditions

3.2.3. Sensitivity Analysis

3.2.3.1. Infiltration

The preceding examples (Tables 3-2 and 3-3) assume a leak-tight (no infiltration) envelope with all outdoor make-up air supplied by mechanical ventilation and an ERV. The majority of commercial buildings do exhibit some envelope leakage (Persily, 1998). Outdoor air that infiltrates through leaks in the building envelope adds to the latent and sensible load in the same fashion as ventilation make-up air, but is not accessible for energy recovery. ASHRAE 90.1 specifies minimum levels of window assembly tightness and the trend is toward requiring tighter envelopes overall.

To illustrate the impact of a finite level of infiltration, the latent design day cases for Miami and Boston have been recalculated with an infiltration rate of 0.5 air changes per hour (ACH). This level of infiltration brings unconditioned outside air into the space at about 30% of the flow rate of the mechanical ventilation outdoor air supply meeting ASHRAE 62. Table 3-4 summarizes the results and provides a comparison with the results for zero infiltration in Tables 3-2 and 3-3. In the zero infiltration case, the building cooling load SHR and the combined unitary/ERV SHR are comparable, for a unitary air conditioner having an SHR at the high end of the range (0.75) of typical unitary SHRs.

Table 3-4: Sensitivity Analysis – Impact of Infiltration (at 0.5 Air Changes/Hour)

Location & Design Condition	Infiltration: Zero		Infiltration: 0.5 ACH	
	Building Load SHR	Unitary + ERV SHR	Building Load SHR	Unitary + ERV SHR
Miami-Latent Design Day	0.48	0.51	0.44	0.54
Boston-Latent Design Day	0.62	0.59	0.58	0.61

The impact of unconditioned outside air infiltrating into the space is to increase the design cooling load and to reduce the SHR of the building cooling load. The unitary portion of the total cooling capacity increases (the NRC of the ERV is fixed), so the combined SHR increases. As a result, there is a mismatch of the building load SHR and the cooling system SHR. The results in Table 3-4 are summarized in bar chart format in Figure 3-3. Key points:

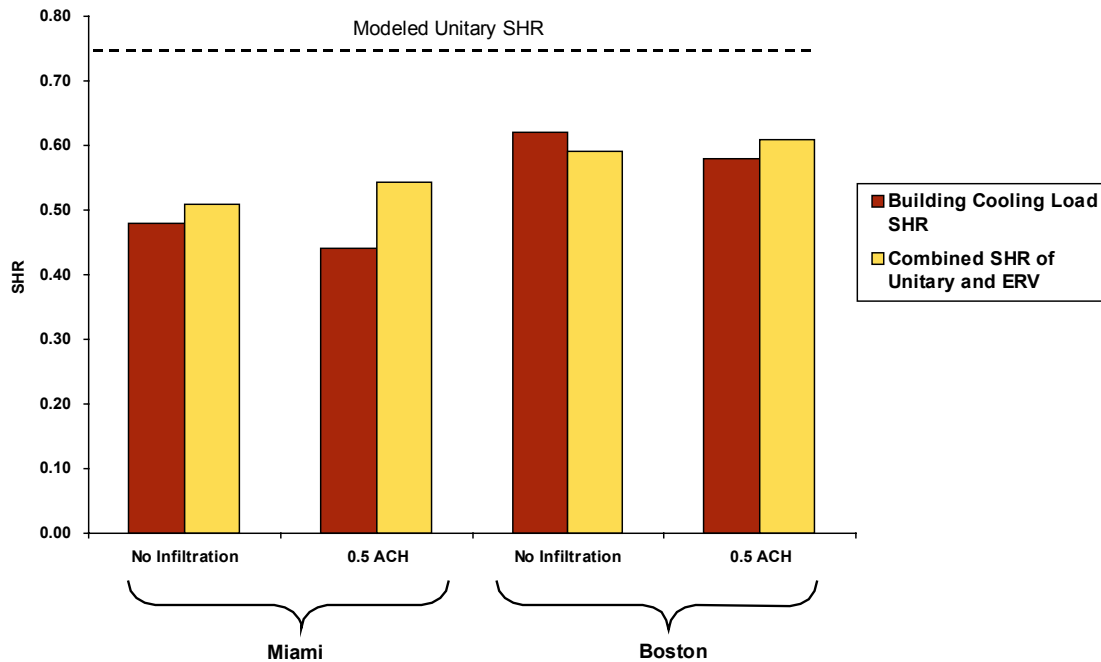


Figure 3-3: Impact of Infiltration on the Match of the Building Cooling Load SHR with the Combined Unitary + ERV SHR (at the Latent Design Condition)

- Even with infiltration added to the mechanical ventilation, the effect of the ERV is to reduce the cooling system SHR significantly from the unitary SHR.
- In a building that has this level of infiltration, to meet the imposed humidity load it would be necessary to specify unitary equipment that has a lower SHR at rating conditions. This may need to be supplemented by additional moisture removal

capacity that could be provided by heat pipes, evaporator bypass, lower CFM/ton, liquid subcooling/air reheat, active desiccants, etc.

- This example also illustrates the importance of minimizing infiltration in a humid climate.

3.2.3.2. ERV Latent Effectiveness < ERV Sensible Effectiveness

In the interest of simplicity, the examples in Tables 3-2 and 3-3 were calculated on the basis of an ERV with equal sensible and latent enthalpy transfer effectiveness. Enthalpy exchangers on the market typically have a latent effectiveness several percentage points less than the sensible effectiveness at a given operating condition. To examine the effect of this difference, the latent design day cases for Miami and Boston were calculated with $\epsilon_{total} = 0.7$, as in the base case, but with $\epsilon_{lat} = 5$ percentage points less than ϵ_{sens} . (i.e., $\epsilon_{lat} = \epsilon_{sens} - 0.05$). The results summarized in Table 3-5 show that the impact on the combined SHR of the unitary + ERV is negligible.

Table 3-5: Sensitivity Analysis: $\epsilon_{latent} = \epsilon_{sens} - 0.05$ (Unitary SHR = 0.75)

Location & Design Condition	Unitary Capacity Tons	ERV Capacity and SHR			Combined SHR of Unitary + ERV	
		NRC, tons	SHR _{vent}	ERV SHR	$\epsilon_{lat} = \epsilon_{sens} - 0.05$	$\epsilon_{lat} = \epsilon_{sens}$
Miami-Latent	80.5	62.4	0.191	0.199	0.509	0.506
Boston-Latent	75.8	41.1	0.289	0.302	0.592	0.588

3.2.4. Calculation of Unitary + ERV SHR for the Examples in Section 3.2.2

The following tables provide additional details of the calculations used to create Tables 3-2 and 3-3.

Table 3-6 summarizes the calculation of the ventilation make-up air cooling load and the net recovery capacity of the ERV. Δh_{total} is the enthalpy difference between the ambient air at the indicated dry/wet bulb temperature and the indoor enthalpy at 75°F dry bulb, 63°F wet bulb.

Table 3-6: Ventilation Portion of Design Load, Corresponding ERV Net Recovery Capacity at $\epsilon=0.7$

Location/Condition	Ambient Temperature		Δh_{total} Btu/lb	Ventilation Load, Tons*	ERV($\epsilon=0.7$) NRC, Tons
	DB, °F	WB, °F			
Miami – Sensible	91	77	11.94	70.3	49.2
Miami – Latent	87	80	15.14	89.2	62.4
Boston – Sensible	91	73	8.11	47.8	33.4
Boston – Latent	87	75	9.97	58.7	41.1
Shoulder	75	72	7.22	42.5	29.8

*Ventilation load = $15,710 \times .075 \times 60 \times \Delta h_{total}$

Table 3-7 summarizes the calculation of the net capacity of the unitary air conditioner needed to meet the total cooling load at the indicated ambient condition. The “design load” was determined by the EnergyPlus run for the indicated design condition. The

“ERV NRC” was determined in Table 3-6 and the Net Unitary Capacity is obtained by subtracting the ERV NRC from the design load.

Table 3-7: Design Load and Unitary Capacity Net of the ERV NRC

Location/Condition	Design Load, Tons	ERV NRC, Tons	Net Unitary Capacity
Miami – Sensible	129.65	49.2	80.45
Miami – Latent	142.9	62.4	80.5
Miami – Shoulder	91.8	29.8	62.0
Boston – Sensible	112.95	33.4	79.55
Boston – Latent	116.9	41.1	75.8
Boston – Shoulder	93.2	29.8	63.4

Table 3-8 summarizes the computation of the combined unitary + ERV SHR, for a unitary air conditioner having an SHR of 0.75.

Table 3-8: Combined SHR (Unitary SHR = 0.75)

Location/Condition	Unitary Capacity, Tons	ERV		Unitary +ERV Combined SHR
		NRC, Tons	SHR	
Miami Sensible	80.5	49.2	0.322	0.588
Miami Latent	80.5	62.4	0.191	0.506
Miami Shoulder	62.0	29.8	0	0.506
Boston Sensible	79.5	33.4	0.474	0.668
Boston Latent	75.8	41.1	0.289	0.588
Boston Shoulder	63.4	29.8	0	0.518

4. Summary of Key Findings

As evidenced by the trade journal literature and the technical programs at HVAC technical conferences, there is significantly increased awareness of inadequate humidity and moisture control in buildings, and the adverse consequences this has on occupant comfort and mold growth. This situation has been attributed incorrectly to increasing SHRs (reduced moisture removal capacity) of typical air conditioning equipment as efficiency levels have increased. ARI data shows that this is simply not the case. Over several decades, and over a wide range of EER and SEER levels, the range of commercially manufactured unitary air conditioner SHRs has remained 0.65 to 0.80.

- The design of buildings has changed in response to energy codes. Sensible loads are most readily reduced. Latent loads usually originate from ventilation and occupant respiration/perspiration, so they are tied to the needs and activities of the occupants, and have not been subject to any reduction.
- Analysis of a hypothetical office building designed to meet ASHRAE 90.1 energy standards and ASHRAE 62 ventilation standards shows a significant trend of falling sensible heat ratio of the total building load as these standards evolved from the early 1970's to the present time.
- Conventional unitary air conditioning is often unable to handle the latent loads while maintaining a desirable indoor relative humidity level close to 50% RH. To maintain 50% RH, conventional equipment needs to be modified or supplemented by equipment that can remove a high proportion of latent load.
- A particularly effective combination of equipment for handling added latent loads is an enthalpy recovery heat exchanger and a unitary air conditioner. A high proportion of the latent load originates with outside air, the enthalpy exchanger (having essentially constant sensible and latent transfer effectiveness) in effect intercepts a significant portion of the (often high latent) cooling load of the outside air.
- A basic methodology is presented in Section 3.1 for calculating the combined, or system, SHR of two or more air conditioning systems or components working together to condition building space or zone.
- In reasonably tight buildings (low infiltration), a unitary air conditioner with an SHR in the 0.70 or 0.75 range combined with an ERV having a latent transfer effectiveness on the order of 70% will inherently follow the variations in outdoor humidity levels and maintain the indoor RH close to 50% for the vast majority of operating hours



5. Conclusions

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 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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