Can Conventional Cooling Equipment Meet Dehumidification Needs for Houses in Humid Climates?

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ABSTRACT

Traditionally, conventional cooling systems have been considered adequate for meeting dehumidification loads even in the most humid climates. However, as high-efficiency home designs have reduced sensible loads and codes have begun requiring continuous outdoor ventilation air, high indoor humidity conditions are becoming more prevalent (Rudd and Henderson 2007). This paper presents detailed simulation results that confirm that high-efficiency envelope designs with continuous ventilation are more likely to require explicit dehumidification to maintain acceptable indoor humidity levels (Henderson, Shirey and Raustad 2007). The TRNSYS-based simulation model considers several aspects of building and equipment performance that are critical to quantifying these impacts, including detailed coil models, off-cycle moisture evaporation, infiltration and ventilation impacts, and duct air leakage. The simulation model was used to evaluate the efficacy of using conventional dehumidifiers as well as enhanced cooling technologies that provide more moisture removal in conjunction with sensible cooling (i.e., a lower sensible heat ratio). The enhanced cooling systems use condenser reheat as well as control enhancements that reduce supply air flow rates and/or conditionally overcool the space. The simulation results show that many of the high humidity periods occur when sensible cooling loads are modest or non-existent. Therefore, systems must provide explicit dehumidification with little-to-no sensible cooling or allow for some degree of overcooling in order to maintain space conditions below 60% RH.

Introduction

Traditionally, conventional unitary air-conditioning systems have been considered adequate for meeting dehumidification loads even in the most humid climates. However, changes in residential ventilation standards and reduced sensible cooling loads with high-efficiency residential home designs may be straining the ability of conventional cooling equipment to adequately control indoor humidity levels.

Field testing of typical Florida homes often show that adequate humidity control – i.e., space humidity levels below 60% RH – are usually maintained in homes that use conventional cooling equipment. For instance field testing by Henderson, Rengarajan and Raustad (1991) in the early 1990s showed that summer time humidity levels rarely exceeded 60% RH in a study of more than 24 Florida homes. Similarly, more recent field testing by Shirey, Henderson and Raustad (2006) showed similar results for conventional air conditioners – as long as the fan cycled on and off with the air conditioner.

However, space humidity data recently collected on several newer, high-efficiency homes showed much different behavior: with humidity levels often exceeding 60 or even 70% RH when only conventional cooling system were used (Rudd and Henderson 2007). These homes included a mix of Energy Star homes as well as even more energy-efficient Building America home designs with ducts in the conditioned space, very high insulation levels, high quality windows, and tight building envelopes. Many of the 43 homes in this sample also included ventilation systems to supply fresh air in accordance with ASHRAE Standard 62.2. The homes included a wide range of HVAC equipment, including: 1) conventional cooling systems, 2) cooling systems with enhanced dehumidification systems, and 3) dehumidifiers. Overall, the test results implied that high-efficiency homes and homes with mechanical ventilation were more likely to have higher space humidity levels, though the relative contribution of each factor was not readily apparent from the field data.

Approach

The analysis presented in this paper was developed using a detailed TRNSYS model of a house to predict space humidity levels and energy impacts. The model was developed to assess whether conventional systems have adequate dehumidification capacity or if additional dehumidification is necessary to properly control space humidity levels. The TRNSYS model is used to compare the performance of various standard and enhanced cooling and dehumidification systems, as well as to understand the impacts of different equipment control approaches. The analysis also seeks to understand the impact that mechanical ventilation and high-efficiency residential construction have on dehumidification requirements.

A model of the house was developed in TRNSYS 16 using the Type 56 multi-zone building model (University of Wisconsin et al. 2004). The model framework was originally developed to compare the performance of gas-fired and conventional cooling and dehumidification systems in commercial building applications (Henderson and Sand 2003). The model treats the slab-on-grade house as a single zone and includes a second zone for the attic. The model includes the following features that are key to providing a realistic prediction of space humidity levels and HVAC energy use:

- an accurate building model that simulates the thermal performance of the building envelope and internal heat gains as well as the impact of infiltration and duct air leakage in the attic zone,
- realistic equipment models for the air conditioner and other HVAC components that accurately reflect the performance variations with operating conditions,
- realistic representations of part-load equipment operation in terms of dehumidification performance and efficiency,
- realistic representations of various fan, ventilation, and dehumidification control strategies, including controllable overcooling, fan delays, and distribution of ventilation air with the main air handler fan.

Simulation Assumptions

Special attention was paid to the issue of equipment sizing to insure that the systems were "right-sized" consistent with industry understanding of best practice. Equipment was separately sized for each climate. Simulations were completed for several humid climate locations (Miami, Houston, Jacksonville, Wilmington) as well as some intermediate climates (Atlanta, Fort Worth, Washington). Two 2,000 sq ft houses were simulated in each climate,

1. A HERS (Home Energy Rating System) reference house with the appropriate building construction details and internal gains, as defined by the 2006 RESNET Standard (consistent with 2004 IECC "Supplement" minimum efficiency standards), and
2. A high-efficiency house that would qualify for a Federal Tax Credit under the Energy Policy Act of 2005. These houses have combined annual cooling and heating loads that are 33% to 53% lower than the HERS reference house.

A complete list of assumptions used for each house is given in the full report (Henderson, Shirey and Raustad 2007). Some of the key differences between the houses are summarized in Table 1 below. Both houses were modeled as single-story, slab-on-grade with a ventilated attic. The window fractions defined for the HERS reference were used in both cases, and it was assumed that the houses were not shaded by adjacent houses. The insulations levels, window characteristics, infiltration rates, and duct leakage rate represent the primary differences between the two houses. The resulting internal moisture gain is 24% lower than the net moisture gain of 13.6 lb/day (6.2 kg/day) recommended for this average occupancy level under ASHRAE Standard 160P. The internal moisture gains turn out to be one of the key assumptions for accurately predicting space humidity levels[1].

Table 1. Summary Characteristics for each House

	HERS Reference House	High Efficiency House
Insulation Levels (Region-1, Miami)	R11.2 (wall) R24.9 (ceiling)	R19.4 (wall) R38.5 (ceiling)
Windows (Region-1, Miami)	U-value = 1.2 SHGC = 0.40	U-value = 0.75 SHGC = 0.20
Infiltration (Sherman-Grimsrud method) ELA per envelope area	3.9 in^2 per 100 ft^2 SLA = 0.00048	2.5 in^2 per 100 ft^2 SLA = 0.000307
Duct Leakage	80% distribution efficiency	No leakage (ducts in space)
Set Points	75°F cooling 70°F heating	
Internal moisture gains (from people only)	10.3 lb/day	
Occupancy	3 max, 2.4 avg	

The conventional conditioner is assumed to be a nominal 13 SEER unit. The actual as-installed fan power is assumed to be 0.35 Watt per cfm. Therefore, the "as-installed" SEER with realistic fan power becomes 11.7 Btu/Wh. The AC unit was sized separately for each house and climate to the nearest 0.1 ton[2]. The high-efficiency house generally required a 30% to 40% smaller unit.

Input parameters for modeling latent capacity degradation were selected to represent the part-load dehumidification performance characteristics described by Shirey, Henderson, and Raustad (2006). Latent capacity degradation occurs primarily when the supply air fan operates

[1] A similar study completed by Walker and Sherman (2006) used the ASHRAE 160P assumptions about internal moisture load.
[2] The full report evaluated the impact of equipment sizing on energy use and space humidity levels, and found a relatively modest impact due to 30% oversizing (within ±3% for energy use; only small changes in hours above 60% RH)

continuously while the compressor cycles on and off to meet the sensible cooling load (i.e., to satisfy the temperature thermostat). However, they also noted that some minor amount of latent capacity degradation also occurs in the AUTO Fan mode (supply air fan cycles on and off with the compressor). Degradation parameters were selected to represent both AUTO and CONSTANT Fan as well as hybrid scenarios where supply air fan operation continues for a brief period after compressor operation (fan delay) or the fan is periodically activated to provide outdoor ventilation air and/or zone air mixing.

Outdoor air infiltration and ventilation were modeled using two different scenarios: variable infiltration determined by the Sherman-Grimsrud Method with no additional mechanical ventilation, and constant ventilation mechanically provided in a manner consistent with ASHRAE 62.2. Assuming 4 bedrooms in this 2,000 sq ft home, the later scenario required a ventilation rate of 57.5 cfm. The standard implicitly assumes another 40 cfm (2% of floor area) is provided as background infiltration (in reality, this will vary depending on wind speed, indoor/outdoor temperature difference, supply air fan operation and duct leakage, etc.). For this study, we assumed that a total 97.5 cfm is provided continuously for the mechanical ventilation case (with fan power of 350 kWh per year). This approach provided an "outer bound" estimate of the impact of continuous ventilation required by ASHRAE 62.2.

Results

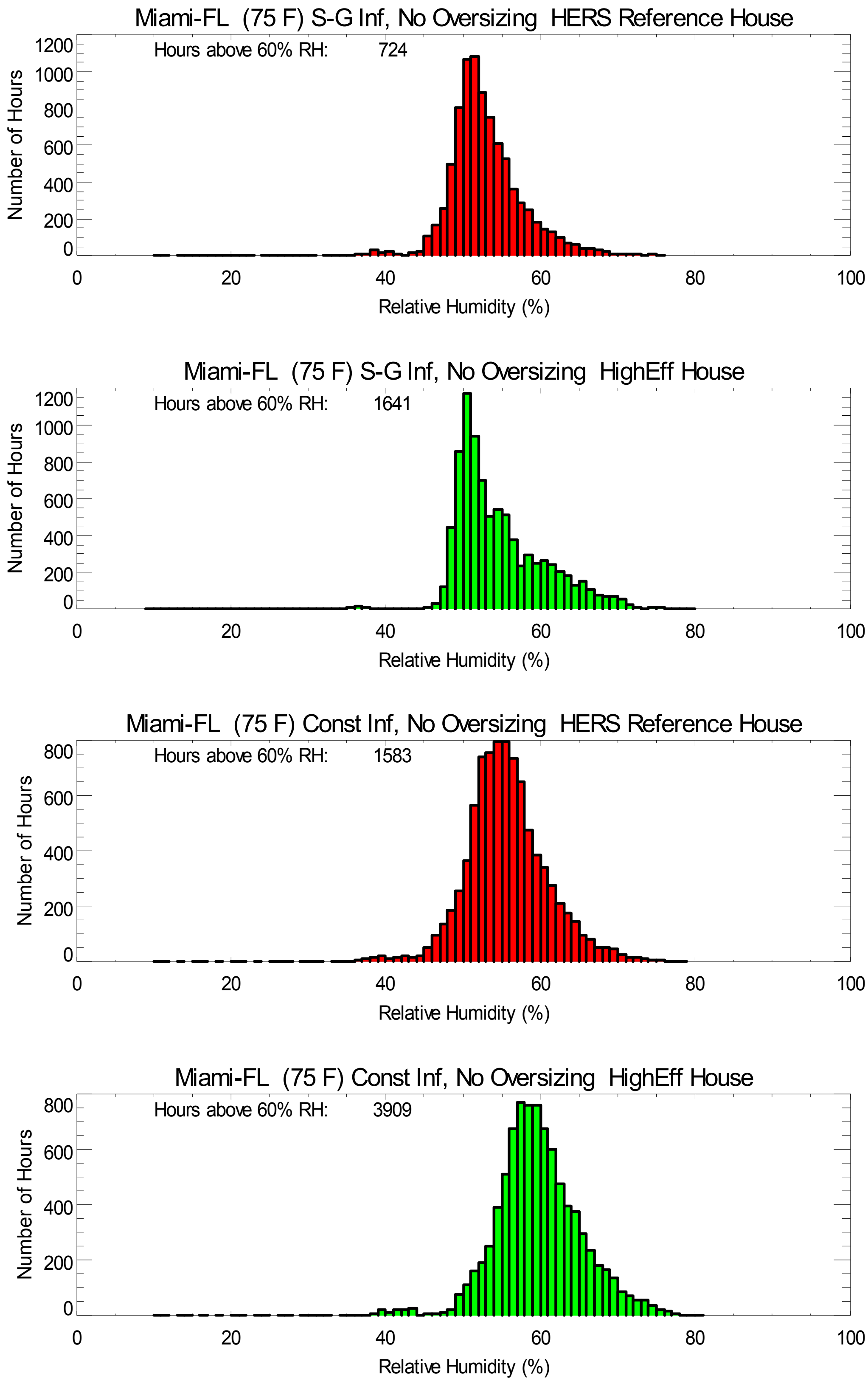
Conventional Cooling Systems

The hourly simulation model was used to evaluate the wide range of options.

Figure 1 shows the resulting distribution of space humidity levels for the case of the HERS Reference and High Efficiency Houses in Miami with variable infiltration (S-G Inf) and constant ventilation (Const Inf) using conventional cooling equipment. All data in the histograms are for the case with a cooling set point of 75°F and no equipment over-sizing. A key metric is the number of hours per year that the indoor air exceeds 60% RH. For the case with variable infiltration (S-G Inf) the hours over 60% are 724 hours for HERS Reference hours and 1641 hours for the High Efficiency House. When the houses are continuously ventilated at 97.5 cfm, the amount of time over 60% RH increase to 1583 hours and 3909 hours, respectively.

Table 2 summarizes the space humidity results for all climates evaluated as well as with different cooling set point temperatures. The results clearly show the following trends:

Figure 1. Comparison of Relative Humidity Distributions for Miami with a 75°F Set Point



- In the humid climates (Miami, Houston, Jacksonville) the hours over 60% RH are consistently greater for the High Efficiency house which has lower sensible cooling loads. In more moderate climates, the results are mixed and the impacts are smaller.
- In the humid climates (Miami, Houston, Jacksonville) the hours over 60% RH are consistently greater when constant ventilation is provided as compared to natural (variable) infiltration.
- When the cooling set point is increased from 75°F to 78°F, the hours over 60% RH consistently increases. This somewhat counter-intuitive result happens because there are fewer hours when the air conditioner operates at the higher set point – so less moisture is removed by the air conditioner due to lower compressor runtimes.

Table 2. Hours Above 60% RH with Conventional AC System

			Hours above 60 % RH	
			HERS Ref	High Eff
Variable Infiltration (Sherman-Grimsrud)	75 F	Miami-FL	724	1,641
		Jacksonville-FL	622	976
		Atlanta-GA	193	73
		Sterling-VA	46	-
		Houston-TX	1017	1,400
		Fort_Worth-TX	131	29
		Wilmington-NC	588	253
	78 F	Miami-FL	1667	2,699
		Jacksonville-FL	1153	1,768
		Atlanta-GA	385	118
		Sterling-VA	119	5
		Houston-TX	1535	2,040
		Fort_Worth-TX	305	51
		Wilmington-NC	974	468
Constant Ventilation (0.3656 ACH or 97.5 cfm)	75 F	Miami-FL	1,583	3,909
		Jacksonville-FL	1,391	2,833
		Atlanta-GA	384	355
		Sterling-VA	268	342
		Houston-TX	1,557	2,623
		Fort_Worth-TX	216	191
		Wilmington-NC	1,384	1,750
	78 F	Miami-FL	2,473	4,592
		Jacksonville-FL	1,954	3,297
		Atlanta-GA	563	449
		Sterling-VA	318	299
		Houston-TX	1,991	2,955
		Fort_Worth-TX	385	263
		Wilmington-NC	1,772	1,841

The shade plots in

Figure 2 show when the high humidity periods occur for Miami (for two specific cases). The hours for each day are shown as a vertical stripe on the plots. The different shades indicate how far the indoor humidity was above 60% RH for each hour. For light gray periods, the humidity was below 60% RH. The next three darker shades of gray progressively indicate that the indoor humidity was between 60-65% RH, 65-70% RH, or greater than 70% RH.

The high-humidity periods tend to occur in the early morning hours and in the swing seasons (i.e., spring and fall) when sensible cooling loads are low. The high-humidity problems are exacerbated by reducing the sensible cooling loads (e.g., high-efficiency building envelope improvements) and providing continuous outdoor ventilation air at times when natural infiltration is normally modest. Each shade plot also includes the number of hours above 60% RH overall, and the number of hours above 60%RH for periods when the air conditioner provides cooling. A large portion of the high humidity hours occur at times when cooling operation is not normally required. This implies that enhanced cooling systems that provide a modest amount of additional dehumidification when indoor humidity is high may not able to fully meet these loads, since no sensible cooling is required (i.e., these systems will not operate without a thermostat call). Addressing these high-humidity periods will require explicit dehumidification (without sensible cooling) or some means to overcool the space when humidity levels are high.

Dehumidifiers and Enhanced Cooling Systems

Several enhancements are possible to increase the dehumidification performance of a conventional air conditioner. These low-cost control improvements typically reduce the sensible heat ratio (SHR) of the cooling coil, causing the system to provide more moisture removal and less sensible cooling capacity.

One enhancement is lowering the cooling set point (or overcooling) in response to high humidity levels. One manufacturer allows up to a 3ºF decrease in space temperature as the space humidity increases above a dehumidification set point. Table 3 shows the results for this control scenario in Miami. As described previously, many of the high humidity hours occur when cooling is just barely required. As a result, pushing the cooling set point down at these times forces more cooling and dehumidification and lowers the total hours above 60% RH from 1,583 to 1,070 in Miami (similar results were observed for Houston). The energy penalty is about 6% for this scenario. Overcooling the space, however, may also result in occupant comfort issues.

Another common dehumidification enhancement listed in Table 3 is to lower the supply air fan speed when indoor humidity exceeds a set point. This more passive dehumidification approach lowers the SHR of the cooling coil but does not directly prolong cooling coil operation. Lowering the fan speed to 80% when the RH is above 55% provides a more modest dehumidification improvement: decreasing the time over 60% RH from 1,583 to 1,251 hours in Miami (similar results were observed for Houston). The more modest impact of this control approach underscores the fact that most high-humidity periods occur when cooling loads are low and run-time for single-stage AC units is short.

Figure 2. Shade Plots showing Humid Periods for Houses in Miami

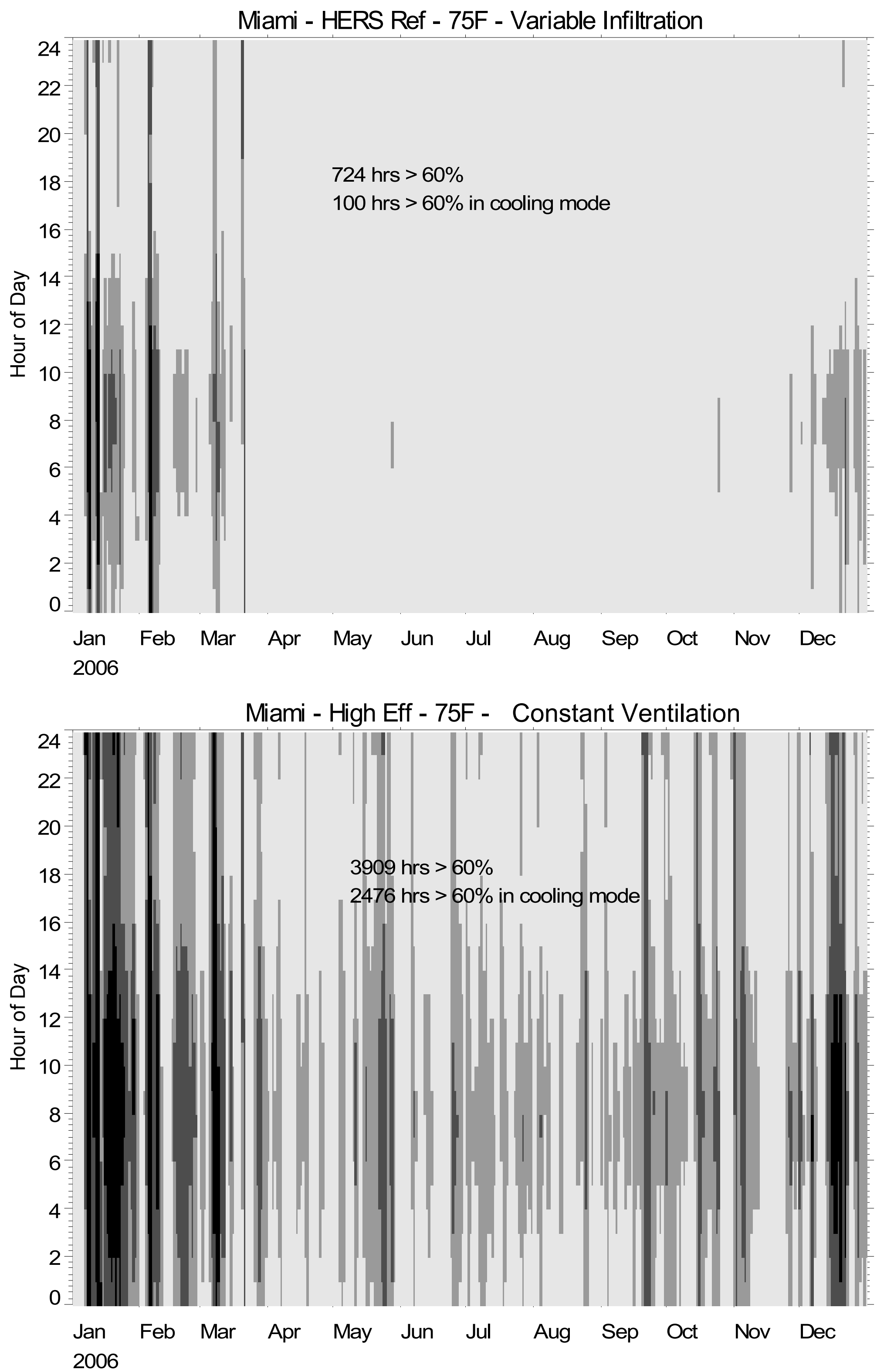


Table 3. Impact of Enhanced AC Controls (HERS Reference House, Constant Ventilation)

Miami	Hours above 60% RH (hrs)	AC Runtime (hrs)	AC Electric Use (kWh)	Supply Fan Electric Use (kWh)	Total HVAC Electric Use[1] (kWh)	Relative Energy Use (%)
Base Case	1,583	2,166	5,201	859	6,411	100%
Over Cooling by 3F	1,070	2,314	5,533	918	6,801	106%
80% Supply Airflow	1,251	2,223	5,281	821	6,451	101%

Notes: Overcooling proportional to how much RH exceeds 55% set point
Lower supply air flow activated when humidity exceeds 55% set point
HERS Reference House with 75°F cooling set point
1 – Includes additional 350 kWh for mechanical exhaust fan

Dehumidifiers (DHs) can be used to directly meet high humidity loads all of the time. However, dehumidifiers add sensible heat to the space which increases cooling operation. Other hybrid dehumidification/cooling options are also available that are potentially more energy efficient and/or add less heat to the space than a standard dehumidifier. The dehumidification options evaluated in this paper include:

- Conventional Dehumidifier. This standalone 75 pint/day unit as a nominal efficiency of 2.6 pints/kWh. The DH operates with independent controls to a dehumidification setpoint of 55% RH.
- Santa Fe high efficiency dehumidifier from Therma-Stor. This 105 pint/day unit has a nominal efficiency of 5.4 pints/kWh, which is twice the dehumidification efficiency of a standard DH unit. This higher efficiency unit adds less sensible heat to the space (so less AC operation is required). Like the standard DH, this standalone unit was also assumed to control to a set point of 55% RH.
- Mini MAU. A very small makeup air unit (MAU) could be used to provide continuous treatment of ventilation air. The MAU is assumed to have its own fan and pretreated air is provided into the supply duct after (or in parallel with) the cooling coil. Only a 0.2 or 0.3 ton unit would be required to dehumidify 57.5 cfm of outdoor air. The condenser coil is located in the cold supply air. The unit was assumed to have 1 stage of cooling capacity and require separate fan power equivalent to 0.7 W/cfm (40 Watts).
- Residential Munters HCU. Munters makes a commercial desiccant unit that regenerates the desiccant wheel with condenser heat. We have assumed that a unit 20% the size of the HCU-1004 could be applied in a house as a ducted dehumidifier (separate ductwork from that used by the conventional AC unit). The 200 cfm HCU unit would be mounted outdoors. It would treat recirculated air from the house and provide no outdoor ventilation air.
- Enthalpy Wheel/ERV. A small energy recovery ventilator (ERV) is assumed to treat 57.5 cfm of incoming outdoor air while exhausting the same amount of air. The ERV runs continuously and is decoupled from the AC supply air fan (which is in the AUTO mode). The assumed effectiveness is 75% and the fan power for both the exhaust and return fans is assumed to be 0.5 W/cfm (on each side).

- Conventional AC with Heat Pipe HX. A heat pipe heat exchanger can be used to enhance the dehumidification performance of a conventional DX cooling coil by precooling and then reheating the supply air flow. The heat pipe is assumed to be a 2-row, 11 fpi plate fin coil with a face area of 3 ft^2 on each side of the cooling coil (heat exchanger effectiveness ~ 0.32). The fan power is increased to 0.4 W/cfm (from 0.35 W/cfm for the conventional AC) to account for the extra air-side pressure drop.
- Condenser Reheat. A condenser reheat coil is installed downstream of the cooling coil to provide free air reheat. The cooling coil operates to hold either the cooling or dehumidification set point. The reheat coil modulates its heat output to maintain the space temperature at a set point temperature 0.5ºF below the cooling set point (if cooling coil operation driven by dehumidification overcools the space). The maximum heating capacity of the condenser reheat coil is assumed to be 75% of the nominal cooling coil capacity, or roughly half of the total condenser heat rejection. As a result the compressor discharge pressure is assumed to be still driven by outdoor conditions since at least half of the condenser heat must be rejected to ambient.

Table 4 and

Table 5 show the simulation results for these options in the Miami HERS Reference and High-Efficiency houses, respectively. The conventional standalone dehumidifier eliminates all hours above 60% RH, but meeting the additional dehumidification load increases overall HVAC system energy use significantly compared to the conventional AC system which does not meet this load. Even a 50% smaller dehumidifier (37.5 pint/day) was able to meet these loads in all the locations (see the full report). Somewhat surprisingly, the sensible heat added to the space by the conventional dehumidifier did not have a large impact on the air conditioner runtime. For instance, while the dehumidifier ran for more than 1000 hours in the Miami HERS reference house, the increase in air conditioner runtime was only 128 hours (6%). The dehumidification hours often occur when sensible cooling is not required, so the added heat is not a large cooling penalty.

The high-efficiency Santa Fe DH unit provides good dehumidification with about half the energy penalty of the standard dehumidifier in all cases. The small scale MAU also provides good dehumidification with a slightly lower energy penalty than the standard dehumidifier (in reality the condenser sees air from the cooling coil outlet instead of the space which was assumed for these simulations, so the actual efficiency for this case would probably be even higher than was predicted here). The Munters HCU provides good dehumidification with a very low energy penalty. One major issue with this unit, however, is the need to be installed outside in order to reject heat (and moisture) to ambient.

Table 4. Impact of Other Dehumidification Equipment Options: HERS Reference House, Constant Ventilation

Miami	Hours above 60% RH (hrs)	AC Runtime (hrs)	Dehumid Runtime (hrs)	AC Electric Use (kWh)	Supply Fan Electric Use (kWh)	Mech. Exh. Fan Use (kWh)	DH Unit Electric Use (kWh)	DH FAN Electric Use (kWh)	Total Electric Use (kWh)	Relative Energy Use (%)
Conventional AC	1,583	2,166	-	5,201	859	350	-	-	6,411	100%
Standalone Dehum (75 pint)	-	2,294	1,083	5,480	908	350	1,189	-	7,927	124%
Santa Fe High Efficiency DH	-	2,248	859	5,374	890	350	628	-	7,243	113%
Mini MAU (0.2 tons, 288 cfm/ton)	77	2,269	2,410	5,418	898	-	449	353	7,471	117%
Mini MAU (0.3 tons, 192 cfm/ton)	34	2,271	1,905	5,421	898	-	502	353	7,527	117%
Munters HCU	-	2,046	897	4,898	813	350	777	69	6,976	109%
Conv AC (AUTO) w/ ERV (CONST)	1,201	1,563	7,135	3,830	621	-	205	252	5,161	80%
AC with HP HX	675	2,478	-	5,644	1,122	350	-	-	7,116	111%
Condenser Reheat System	-	2,635	-	6,212	1,042	350	-	-	7,603	119%

Table 5. Impact of Other Dehumidification Equipment Options: High-Efficiency House, Constant Ventilation

Miami	Hours above 60% RH (hrs)	AC Runtime (hrs)	Dehumid Runtime (hrs)	AC Electric Use (kWh)	Supply Fan Electric Use (kWh)	Mech. Exh. Fan Use (kWh)	DH Unit Electric Use (kWh)	DH FAN Electric Use (kWh)	Total Electric Use (kWh)	Relative Energy Use (%)
Conventional AC	3,909	2,170	-	3,187	520	350	-	-	4,057	100%
Standalone Dehum (75 pint)	-	2,592	1,486	3,738	620	350	1,640	-	6,348	156%
Santa Fe High Efficiency DH	-	2,476	1,206	3,581	592	350	888	-	5,411	133%
Mini MAU (0.2 tons, 288 cfm/ton)	6	2,551	3,508	3,680	610	-	673	353	5,667	140%
Mini MAU (0.3 tons, 192 cfm/ton)	5	2,559	2,604	3,692	612	-	708	353	5,718	141%
Munters HCU	-	1,859	1,425	2,713	446	350	1,240	110	4,968	122%
Conv AC (AUTO) w/ ERV (CONST)	1,726	1,445	6,984	2,152	347	-	201	252	3,203	79%
AC with HP HX	1,583	2,596	-	3,571	710	350	-	-	4,631	114%
Condenser Reheat System	-	3,344	-	4,664	799	350	-	-	5,813	143%

As expected, the ERV is more energy efficient than any of the other options simulated; though, the ERV option still results in a large number of hours with elevated space humidity levels. The ERV has a modest humidity impact since it can not provide dehumidification when the indoor-to-outdoor humidity differential is small. The simple condenser reheat system provides good humidity control with an energy penalty that is slightly lower than the standalone dehumidifier.

More recent analyses (Henderson and Rice 2008) have looked at other types of HVAC systems such as the Lennox Humiditrol unit. This analysis used a slightly different set of building assumptions than described above and looked at different cities; however, the relative results are still informative. The Humiditrol system uses a subcool/reheat coil to lower the SHR of the AC cooling coil. In the dehumidification (DH) operating mode, the unit lowers the supply air flow rate to 60% of the nominal cooling flow and activates the subcool/reheat coil. When the outdoor temperature is below 75ºF, the unit switches to enhanced DH mode and the outdoor fan cycles or modulates to provide an even lower SHR. Generally, the SHR cannot go lower than zero (i.e., no simultaneous heating and dehumidification). The DH modes are activated when the space drops below the cooling set point. The dehumidification set point was set at 55% RH. The amount of overcooling below the cooling set point is user selectable (we used settings of 2ºF and 4ºF). In summary, this unit employs a combination of subcool/reheat, reduced airflow, and overcooling to provide dehumidification on demand.

The XP-13 series Humiditrol unit provided good humidity control with about half the energy use premium of a standard standalone dehumidifier, as long as sufficient overcooling was allowed. With an overcooling setting of 4°F, there were almost no hours over 60% RH. Even with a 4°F overcooling setting, very little overcooling occurred during the normal cooling season: the overcooling simply continued system operation at times when cooling was not normally required (because the cooling set point was already satisfied). The degree of humidity control for the Humiditrol unit – i.e., its ability to hold space humidity levels below 60% RH – was much better than most of the enhanced cooling options and nearly matched the levels achieved by the dehumidifier options. Its energy performance was on par with the high-efficiency Santa Fe dehumidifier.

Conclusions

The simulation results for the standard HERS Reference house with natural infiltration (wind and temperature difference driven) were generally consistent with field experience: a conventional AC provides adequate dehumidification to hold the space below 60% RH for all but a few hundred hours of the year in most U.S. climates. The annual time above 60% RH ranged from 588 to 1017 hours in the four most humid climates and 46 to 193 hours in the three less humid climates.

Indoor humidity levels were typically higher for the High-Efficiency house compared to the standard house, in spite of lower infiltration rates. The reduction in sensible cooling loads due to better walls and windows reduces air conditioner runtimes and results in less dehumidification provided by the air conditioner coil.

Adding continuous ventilation, similar to what is required by ASHRAE Standard 62.2-2004, significantly increases the number of hours over 60% RH in all scenarios. The main reason is that more ventilation is provided at moderate outdoor conditions, when the driving potential for natural infiltration is small. While the impact of continuous ventilation on the total cooling load is small at these outdoor conditions, the moisture introduced at these times can have a big impact on space humidity levels since air conditioner runtimes are low.

Because high humidity conditions occur at times when cooling loads are small, lowering the cooling set point temperature can have a big impact on space humidity levels. Relaxing the cooling set point to 78°F substantially increased the hours above 60% RH.

Due to the nature of when the high-humidity periods occur, cooling systems that provide enhanced dehumidification in conjunction with thermostat-based cooling can only provide limited improvement in humidity control since there is no call for cooling during many of these high-humidity periods. Only systems that provide explicit dehumidification – such as a standalone dehumidifier -- can completely control humidity during these times. The conventional wisdom assumes that the heat added to the space by a dehumidifier provides significant additional cooling load that increases AC operation. However, the results from this analysis shows that the energy penalty due to this heat addition is small since the dehumidifier often operates when the space is just below the cooling set point. The energy impact from operating a dehumidifier in tandem with a conventional air conditioner is primarily from the additional air dehumidification that is provided.

The results of this analysis show that humidity control is primarily a problem when sensible (temperature) cooling loads are very modest or non-existent. Dehumidification needs in the summer are a small portion of the annual dehumidification load. There is some evidence that new, energy-efficient homes will increase the need for swing season dehumidification. Therefore, technologies that can efficiently provide dehumidification with little or no sensible cooling are likely to be the most successful.

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