

## AC System Equipment Specification, Installation and Operational Options for Improved Indoor Humidity Control

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### ABSTRACT

High-efficiency home designs have significantly reduced sensible cooling loads, and some building codes and IAQ standards have begun requiring continuous outdoor ventilation air. These trends have led to an increased prevalence of high indoor humidity conditions (Rudd and Henderson 2007). This paper presents a summary of low-cost or no-cost equipment selection and operation options for conventional residential air-conditioning equipment that can result in lower indoor humidity levels. These options should be evaluated and employed to the extent possible prior to considering the added first cost and operating costs of separate dehumidification equipment.

### INTRODUCTION

There has been increased concern about controlling high humidity levels in homes as energy efficiently as possible. Newer energy codes and above-code programs encourage energy-efficient envelope methods that reduce sensible cooling loads from roofs, windows, floors, walls and doors. Multifamily housing is particularly problematic as many units have only one or two surfaces exposed to outdoors. Less sensible cooling load from outdoors means that the latent (dehumidification) portion of the cooling load, from infiltration, mechanical ventilation, and internal moisture sources, is a larger portion of the total cooling load. Especially during weather with high humidity but mild temperatures, interior humidity levels may become higher than desired in these homes.

There are many products on the market and many strategies that can successfully reduce the incidence of the problem. Some may have a minor impact on energy use while others can use significantly more energy in order to further dry the conditioned space. This paper summarizes selection, installation and operational issues for conventional direct-expansion air conditioners that can be employed prior to seeking the added first cost and operating costs of separate dehumidification equipment.

### SELECT AC SYSTEMS WITH LOWER SHR

There exists a wide range of conventional direct expansion AC equipment in the marketplace, and different equipment combinations (coil, air handler, condensing unit) result in systems with different dehumidification performance. Sensible Heat Ratio (SHR) is the ratio of sensible cooling capacity (temperature reduction) of the system to the total cooling capacity of the system (temperature reduction plus dehumidification). Lower SHR indicates better dehumidification performance (i.e., a larger portion of the system's total cooling capacity is devoted to dehumidification).

Figure 1 shows the steady-state SHR for various air conditioners (air-cooled direct expansion [DX] systems with single compressor speed) based on both laboratory measurements and manufacturer's catalog data. At standard rating conditions of 80°F (26.7°C) dry-bulb temperature and 67°F (19.4°C) wet-bulb temperature air entering the indoor unit (AHRI 2006), the equipment SHR values range from 0.67 to 0.8. Thus, the dehumidification fraction (one minus SHR) varies from 0.2 (20%) to 0.33 (33%) for the different systems. The data for these systems also indicate that, for the most part, the change in SHR with variation in inlet air wet-bulb temperature is fairly consistent for each unit. For improved dehumidification performance, equipment combinations with lower SHR should be specified.

For 2-stage cooling equipment, the SHR at both high speed (e.g., AHRI Rating Conditions of 80°F drybulb [26.7°C] / 67°F [19.4°C] wetbulb indoors and 95°F drybulb outdoors) and at low speed (e.g., 80°F drybulb [26.7°C] / 67°F [19.4°C] wetbulb and 82°F drybulb outdoors) should be reviewed. In some cases due to mismatches between the ratio of cooling capacity at high and low compressor speed versus the supply air fan speed ratio at high and low speed, the system can have a relatively low SHR at high speed (e.g., 0.73 at 350 cfm/ton [47 L/s per kW]) but a high SHR at low speed (e.g., 0.8 at 450 cfm/ton [60 L/s per kW]). In this example, the fan speed is not reduced in the proper proportion to match the reduction in compressor capacity when

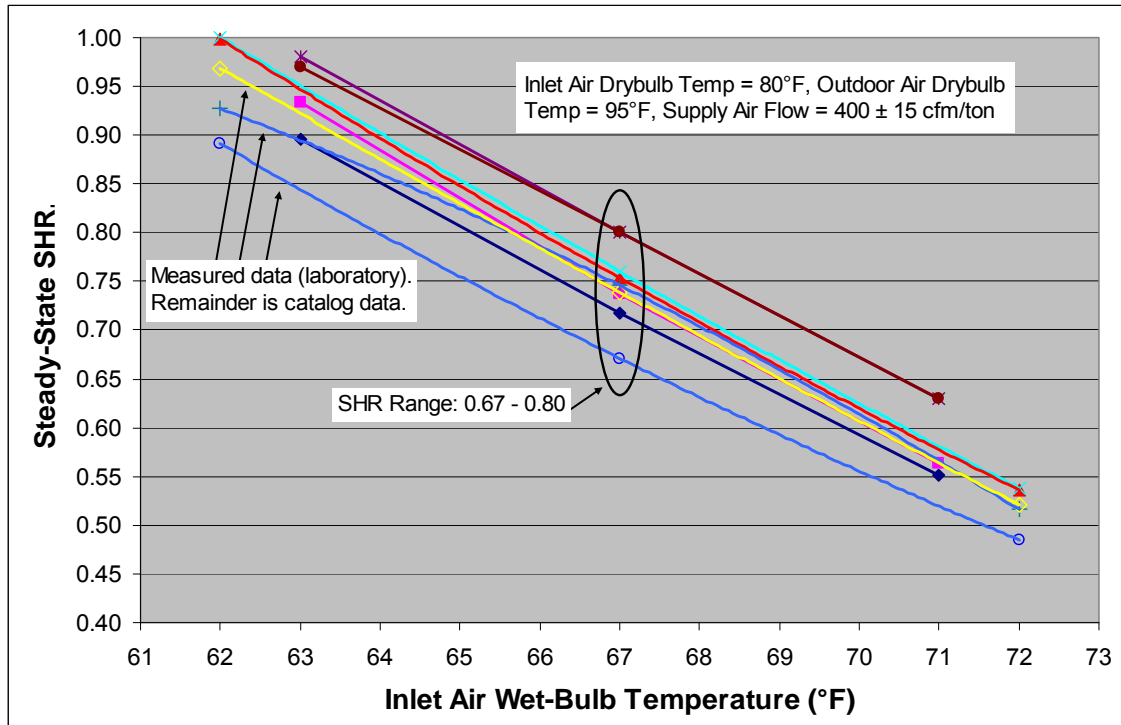


Figure 1. Variations in Steady-State Sensible Heat Ratio

changing from high speed to low speed. As a result, the system operates with a high SHR (less dehumidification) at the low operating speed, which is the speed where the system operates most often. For systems installed in humid-climate residences, the SHR at both low speed and high speed operation (e.g., at AHRI rating conditions) should typically be on the lower end of the available range (e.g., <0.73).

Proper equipment selection requires evaluating the full range of sensible and latent loads expected throughout the cooling season (i.e., not just at a single design condition). These cooling loads then need to be compared with the performance of available AC equipment. Since influences from ductwork air leakage, moisture capacitance due to interior building materials and furnishings, and part-load equipment performance should also be considered, annual computer simulations are usually necessary for selecting the best equipment for each application.

#### REDUCE SUPPLY AIR FLOW RATE

Reducing the supply air flow rate through a cooling system (e.g., reducing the fan motor speed) will increase its dehumidification performance. This will typically result in a lower energy efficiency ratio (EER) (Parker et al. 1997), but the efficiency decrease may be modest in some cases depending on the supply air fan and compressor characteristics.

Operating at too low of a supply air flow rate could cause coil icing and/or sweating ductwork (i.e., moisture condensing on the outside of the ductwork due to cold surface temperatures). Placing ductwork and the air handler in conditioned space or increasing duct insulation levels can help alleviate the sweating ductwork issue, but the potential for coil icing remains so a lower limit on supply air flow rate is required.

The supply air flow rate can be set near 350 cfm/ton [47 L/s per kW] if the air handling unit (AHU) and/or supply air ductwork is located outside of the conditioned space. If the AHU and supply air ductwork are both located within the conditioned space (i.e., within the thermal and air boundaries for the home), then there may be the opportunity to reduce air flows even lower (e.g., 300-320 cfm/ton [40-43 L/s per kW]). However, only operate the supply air flow as low as is needed for adequate dehumidification since over-dehumidification of the conditioned space results in increased energy use.

The following information is taken from Table 1:

Point 1: 1225 cfm (580 L/s), 35.6 kBtuh (10.4 kW) =  
 413 cfm/ton (55.6 L/s per kW)  
 Sensible capacity = 35.6 x **0.78** =  
 27.77 kBtuh (8.1 kW)

**Table 1.** Air conditioner equipment performance over a range of operating conditions (Lennox 2007)

Entering Wet Bulb Temperature	Total Air Volume		Outdoor Air Temperature											
			85°F (29°C)						95°F (35°C)					
			Total Cooling Capacity		Comp Motor kW Input	Sensible To Total Ratio (S/T) Dry Bulb			Total Cooling Capacity		Comp Motor kW Input	Sensible To Total Ratio (S/T) Dry Bulb		
						75°F 24°C	80°F 27°C	85°F 29°C				75°F 24°C	80°F 27°C	85°F 29°C
cfm	L/s	kBtuh	kW				kBtuh	kW						
63°F (17°C)	1090	515	34.6	10.1	2.29	.76	.90	1.00	32.8	9.6	2.61	.77	.93	1.00
	1225	580	35.4	10.4	2.29	.78	.94	1.00	33.6	9.8	2.62	.80	.97	1.00
	1380	650	36.2	10.6	2.30	.81	.98	1.00	34.4	10.1	2.62	.84	.99	1.00
67°F (19°C)	1090	515	36.6	10.7	2.31	.60	.73	.87	34.8	10.2	2.62	.60	.75	.89
	1225	580	37.6	11.0	2.31	.61	.76	.90	35.6	10.4	2.63	.62	.78	.93
	1380	650	38.5	11.3	2.31	.63	.79	.95	36.4	10.7	2.63	.64	.81	.97
71°F (22°C)	1090	515	38.5	11.3	2.32	.45	.58	.71	36.8	10.8	2.62	.45	.59	.72
	1225	580	39.5	11.6	2.32	.46	.60	.74	37.6	11.0	2.64	.46	.61	.76
	1380	650	40.5	11.9	2.33	.46	.62	.77	38.5	11.3	2.64	.47	.63	.79

Point 2: 1090 cfm (515 L/s), 34.8 kBtuh (10.2 kW) = 376 cfm/ton (50.5 L/s per kW).  
Sensible capacity = 34.8 x **0.75** = 26.1 kBtuh (7.7 kW)

Reducing the air flow rate across the cooling coil lowers the sensible heat ratio (SHR, or S/T Ratio in Table 1) of the cooling equipment, and the lower SHR means the unit will remove more moisture from the air when it operates. In addition, the lower sensible capacity means the system will run a little longer to achieve the same dry-bulb temperature set point resulting in additional dehumidification.

Table 2 presents computer simulation results showing the impacts of reduced supply air flow rate on indoor humidity levels (annual hours above 60%

RH) and energy use (Henderson et al. 2007). Some of the key characteristics for the modeled houses are summarized in Table 3. The computer simulation program accounts for the moisture capacitance of interior building materials and furnishings, ductwork air leakage, part-load air conditioner performance, and AUTO fan control was modeled for these cases (i.e., supply air fan operates only when the compressor operates). For these simulations, it was assumed that the ductwork size and duct air leakage decreased with the lower supply air flow rate. However, the unit's nominal cooling capacity and the fan power rate (Watts per cfm of air flow) remained the same.

In reality, the normalized fan power (Watts per cfm) increases slightly when the supply airflow rate

**Table 2.** Impact of Lower Supply Air Flow on High Indoor Humidity and AC Energy Use

Miami	HERS Reference House			High Efficiency House			
	Hours above 60% RH (hrs)	AC Runtime (hrs)	Relative Energy Use (%)	Hours above 60% RH (hrs)	AC Runtime (hrs)	Relative Energy Use (%)	
Variable Infiltration (S-G)	400 cfm/ton	724	2075.9	100%	1,641	1911.0	100%
	300 cfm/ton	533	2015.2	100%	1,306	1876.1	101%
Constant Infiltration	400 cfm/ton	1,583	2166.2	100%	3,909	2170.4	100%
	300 cfm/ton	1,116	2104.6	100%	2,861	2128.7	101%

Notes: See Table 3 for house characteristics. Supply air flow and fan power decrease proportionally at 300 cfm/ton. Unit size remains the same. Duct surface area changes proportionally with air flow. Duct air leakage held as a constant percentage of supply air flow in each case (no leakage for high efficiency house). Fan power maintained at 0.35 W/cfm (0.74 W per L/s). S-G = Sherman-Grimsrud method. For Constant Infiltration/Ventilation case, outdoor air flow continuous at 98 cfm (46 L/s). "Relative Energy Use (%)" is relative to the 400 cfm/ton case for each infiltration scenario (e.g., 101% means a 1% increase compared to the 400 cfm/ton case for the same infiltration scenario).

**Table 3.** Summary Characteristics of Modeled Houses

	HERS Reference House	High Efficiency House
Insulation Levels	R11.2 (wall) R24.9 (ceiling)	R19.4 (wall) R38.5 (ceiling)
Windows	Miami: U-value = 1.2 SHGC = 0.40 Houston: U-value = 0.75 SHGC = 0.40	Miami: U-value = 0.75 SHGC = 0.20 Houston: U-value = 0.65 SHGC = 0.20
Infiltration (Sherman-Grimsrud method) ELA per envelope area	3.9 in <sup>2</sup> per 100 ft <sup>2</sup> SLA = 0.00048	2.5 in <sup>2</sup> per 100 ft <sup>2</sup> 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	

Note: A complete list of assumptions used for each house is given in the full report (Henderson et al. 2007). Both houses were single-story, slab-on-grade construction with ventilated attic.

is decreased from 400 to 300 cfm per ton (54 to 40 L/s per kW). The Table 2 results include the assumption that supply fan power decreased in proportion with the air flow at 300 cfm/ton (i.e., the normalized fan power remained constant at 0.35 W/cfm [0.74 W per L/s]). If the fan power is assumed to be 0.4 W/cfm (0.85 W per L/s) at 300 cfm/ton (40 L/s per kW), this is more representative of “riding the fan curve” with a forward-curve centrifugal fan. With this assumption, the hours above 60% RH do not change significantly from those shown in Table 2; however, the relative energy use for the 300 cfm/ton (40 L/s per kW) scenario increases to 102-103% (i.e., 2 to 3% increase over the corresponding 400 cfm/ton case).

#### DISABLE SUPPLY AIR FAN OVERRUN

Most air handlers have the ability to operate the supply air fan for a brief period after the compressor shuts off (i.e., Supply Air Fan Overrun). This control method provides some additional sensible cooling, which can increase the Seasonal Energy Efficiency Ratio (SEER) rating somewhat (typical increase  $\leq$  0.5 SEER point). However under normal operating conditions, the additional “cooling” comes primarily from evaporating moisture from the wet cooling coil at the end of the compressor operation cycle and sending that moisture back into the conditioned space. This would be acceptable for systems installed in dry climates -- sending the extra moisture back into the conditioned space would be beneficial. But in

hot/humid climates, this supply air fan strategy significantly increases indoor humidity levels which may lead to occupant discomfort and indoor air quality issues.

Air handlers are normally shipped with supply air fan overrun “enabled” (i.e., same configuration used for the SEER rating tests). However, the installation guide typically provides instructions on how to disable this feature. It is recommended that air conditioner (AC) contractors disable the “Supply Air Fan Overrun” feature on new system installations located in hot/humid climates. And if homeowners with existing AC systems are experiencing high indoor humidity levels, disabling supply air fan overrun may be an easy solution to the problem.

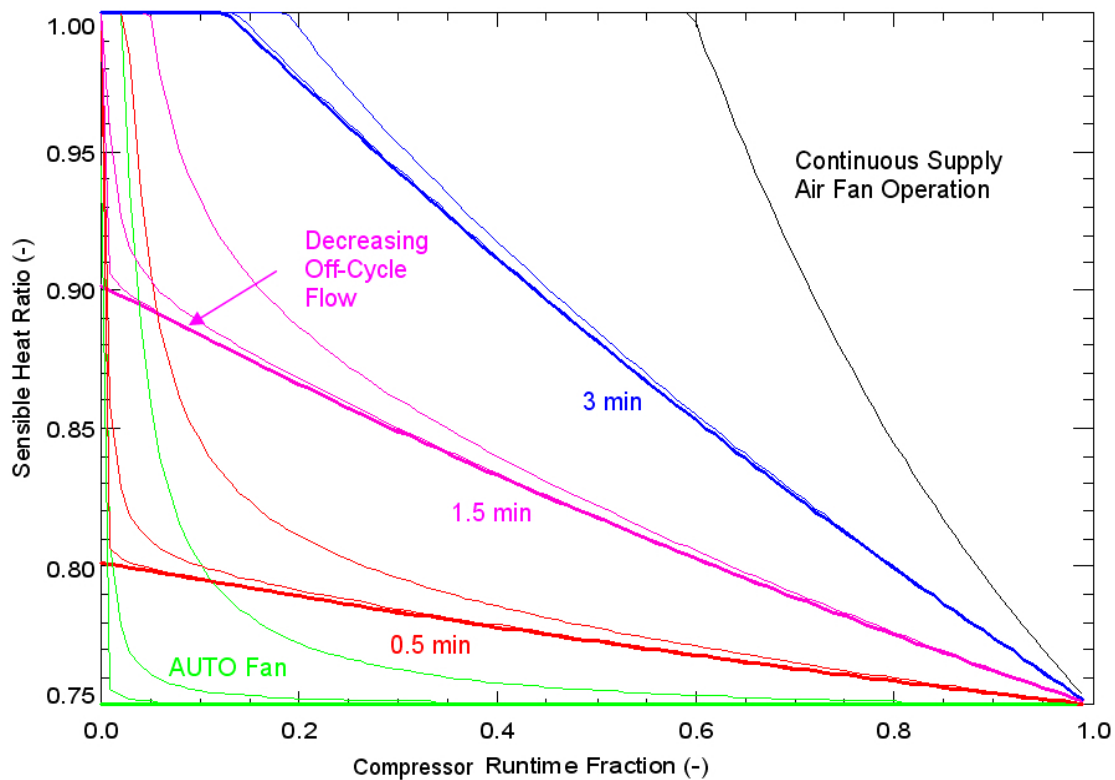
Table 4 summarizes computer simulation results showing the impacts of supply air fan overrun on indoor humidity levels (annual hours above 60% RH) and energy use (Henderson et al. 2007). With a typical 90-second fan overrun, the indoor humidity levels increase significantly. For this particular simulation, the hours above 60% RH rose by 80% (1,583 hrs  $\Rightarrow$  2,854 hrs). The difference in energy use was minor.

Figure 2 shows the impact of a supply air fan overrun strategy that keeps the fan on at the same airflow rate for a fixed length of time after the

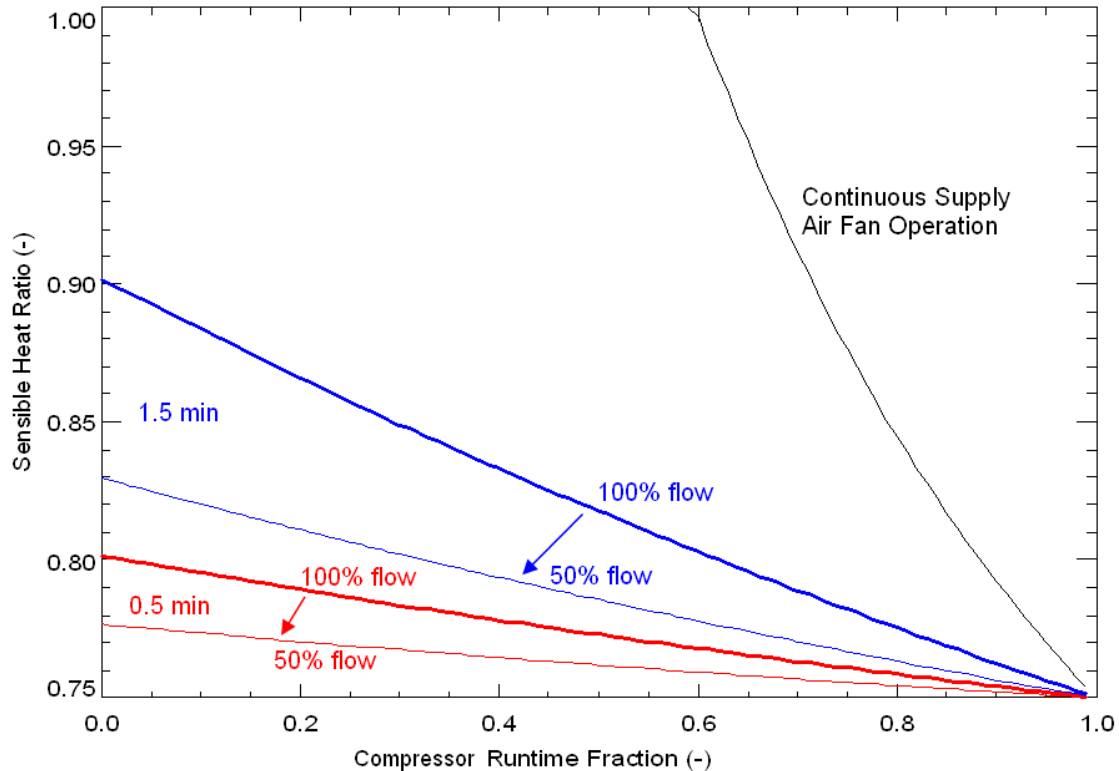
**Table 4.** Impact of Fan Delays on Dehumidification Performance

Miami		Hours above 60% RH (hrs)	AC Runtime (hrs)	AC Electric Use (kWh)	Supply Fan Electric Use (kWh)	Total HVAC Electric Use (kWh)	Relative Energy Use (%)
Base Case	AUTO Fan	1,583	2,166	5,201	859	6,411	100%
Fan Overrun (1s)	AUTO Fan	1,587	2,165	5,199	860	6,409	100%
Fan Overrun (30 s)	AUTO Fan	1,894	2,143	5,159	890	6,398	100%
Fan Overrun (90 s)	AUTO Fan	2,854	2,096	5,075	948	6,373	99%

Notes: HERS Reference House (see Table 3 for characteristics), constant infiltration/ventilation at 98 cfm (46 L/s), 400 cfm/ton (54 L/s per kW) supply air flow rate. "Relative Energy Use (%)" is relative to the Base Case (e.g., 99% means a 1% reduction from the base case Total HVAC Electric Use).



**Figure 2.** Latent Capacity Degradation with Supply Air Fan Overrun



**Figure 3.** Latent Capacity Degradation with Supply Air Fan Overrun at Reduced Air Flow

compressor on cycle (Shirey et al. 2006). The plotted lines are results from a mathematical model verified with laboratory test data. The overrun delays shown on the plot are for 0.5, 1.5, and 3 minutes of supply air fan operation after the compressor turns off. In each case the thicker line is for a simple model that adds a fan delay but assumes no moisture evaporation from the wet cooling coil for the remainder of the off-cycle when the fan is off. The thinner lines associated with each time delay use a two off-cycle interval model which assumes that the 2<sup>nd</sup> interval in the compressor off-cycle when the fan is off has a very small airflow ( $10^2$ ,  $10^3$ , and  $10^4$  times less are shown on the plot). As expected the two models converge at very small off-cycle flow rates.

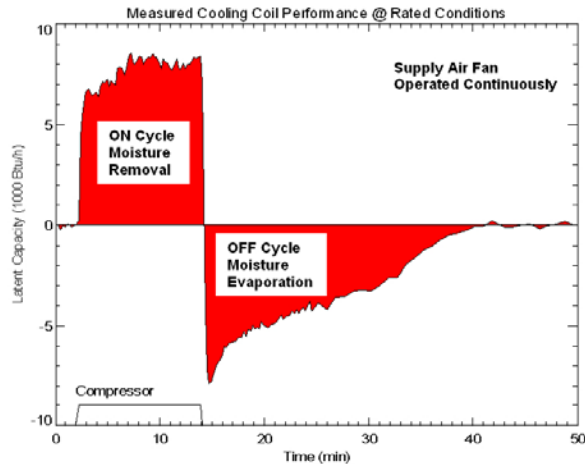
Some manufacturers implement a supply air fan delay that maintains 50% of full flow for a brief period after the compressor shuts off. Figure 3 shows the impact of reducing airflow during the fan delay period (Shirey et al. 2006). The reduced air flow during the fan overrun period does, in part, mitigate the latent capacity degradation. However, any supply air fan overrun negatively impacts the dehumidification performance of the system and is therefore not recommended for homes located in hot/humid climates.

#### OPERATE SUPPLY AIR FAN IN AUTO MODE

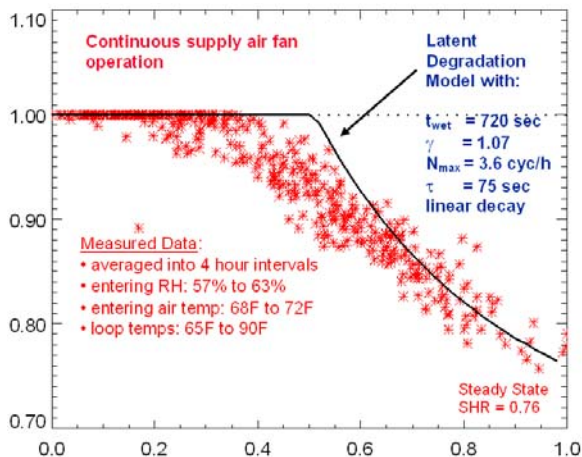
Most thermostats allow the supply air fan of the AC system to operate in AUTO mode (i.e., supply air fan cycles on and off in tandem with the compressor) or the ON mode (i.e., supply air fan operates continuously while the compressor cycles on and off to meet the cooling set point). Operating the supply air fan continuously when the compressor is off (i.e., ON mode) causes moisture from the wet cooling coil to be evaporated into the airstream and sent back into the conditioned zone, thereby increasing indoor humidity levels. While the moisture evaporation process is the same, the dehumidification performance degradation with fan ON mode is even more pronounced than that described previously for supply air fan overrun; that is, fan ON mode is like having a very long supply air fan overrun period.

Figure 4 shows the dehumidification performance for a typical cooling system with continuous supply air fan operation (Henderson 1990). Moisture removal starts shortly after compressor operation begins and eventually reaches steady state. After the compressor turns off but the supply air fan continues to operate, moisture from the wet cooling coil evaporates into the supply air stream. Figure 5 shows the negative impact on

dehumidification performance from a monitored field test site (Henderson 1998). For this site with a single-speed air conditioner, the system provided virtually no net dehumidification for compressor runtime fractions less than 0.4, and dehumidification performance was severely degraded for higher compressor runtime fractions.



**Figure 4.** On-cycle Condensation and Off-Cycle Evaporation of Moisture from a Cooling Coil



**Figure 5.** Comparison of Measured SHR with First-Generation Latent Degradation Model

Figure 6 and Table 5 compare indoor humidity levels achieved with AUTO fan mode and fan ON (constant) mode. Figure 6 shows measured daily indoor humidity levels versus outdoor air humidity levels during summer weather at a Florida field test site (Shirey et al. 2006). For this residence, constant supply air fan operation resulted in significantly higher indoor humidity levels (approximately 20 gr/lb higher) compared to AUTO fan mode, in spite of the fact that the home had a two-stage unit operating with low fan speed during the compressor

off cycle. With a thermostat set point of 77°F, indoor relative humidity levels were approximately 50% RH with AUTO fan mode but rose to 60-65% RH with constant fan operation.

Table 5 summarizes computer simulation results showing the impacts of supply air fan operating mode on indoor humidity levels and energy use for a typical new house (HERS Reference) and a high efficiency house (Henderson et al. 2007). For both houses, continuous supply air fan mode increases indoor humidity levels tremendously (2X to 4X increase in hours > 60% RH). Continuous fan operation also results in a large increase in energy use due to increased supply air fan electrical energy use and fan motor heat, and increased duct air leakage (HERS Reference House only)

Homeowners may feel compelled to operate the AC system's supply air fan in the ON (constant) mode for a variety of reasons. These reasons might include the perceived need for continuous air flow for occupant comfort or improved air cleaning from a high-efficiency filtration system installed in the air distribution system. But for residential applications in hot/humid climates, it is recommended that the system be operated in the AUTO mode (with no supply air fan overrun) to avoid the negative impacts on system dehumidification performance. If the homeowner feels that additional air circulation or air filtration is required beyond that obtained with AUTO fan operation, then local devices should be considered (e.g., ceiling fans, room air cleaners, etc.).

### PROPERLY SIZE THE AC SYSTEM

Energy codes frequently contain provisions to limit AC equipment oversizing (e.g.,  $\leq 115\%$  of the size determined by ACCA Manual J [ACCA 2006]), and these provisions should be retained. However, the input assumptions for the sizing calculations can greatly impact the results. Therefore, methods for additional checking of equipment sizing assumptions and verification of the calculations results should be implemented to the extent possible.

In a field study of more than 300 Florida homes conducted in the early 1990s, AC systems sized 20% larger than the Manual J value consumed 3.7% more cooling energy (James et al. 1997). In this same study, systems sized 50% larger than the Manual J value consumed 9.3% more cooling energy. A recent computer simulation study (Henderson et al. 2007) indicates a slightly smaller impact on energy use, due mainly to recent data which implies that cycling degradation for cooling equipment (i.e., inefficiencies due to on/off cycling) seems to be improving over

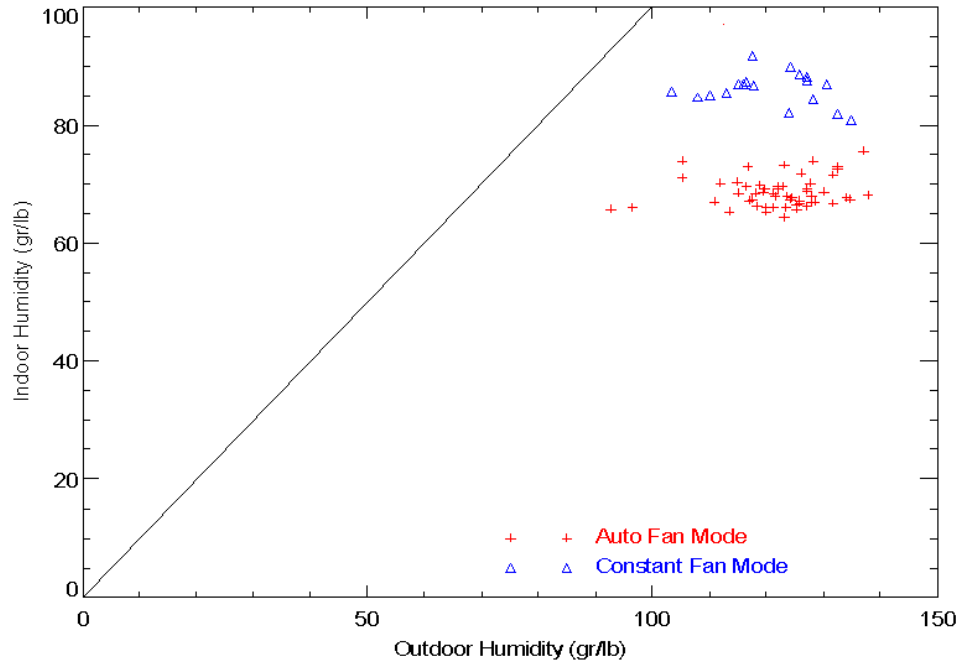


Figure 6. Daily Humidity Ratios with Constant and AUTO Fan Modes at a Florida Residence

Table 5. Impact of Supply Air Fan Operating Mode on Indoor Humidity Levels and Energy Use

HERS Reference House		Hours above 60% RH (hrs)	AC Runtime (hrs)	AC Electric Use (kWh)	Supply Fan Electric Use (kWh)	Total HVAC Electric Use (kWh)	Relative Energy Use (%)
Miami	AUTO Fan	1,583	2,166	5,201	859	6,411	100%
	CONST Fan	7,283	2,089	5,165	3,434	8,598	134%
Houston	AUTO Fan	1,557	1,411	3,670	678	4,697	100%
	CONST Fan	5,411	1,351	3,630	3,679	7,309	156%

High Efficiency House		Hours above 60% RH (hrs)	AC Runtime (hrs)	AC Electric Use (kWh)	Supply Fan Electric Use (kWh)	Total HVAC Electric Use (kWh)	Relative Energy Use (%)
Miami	AUTO Fan	3,909	2,170	3,187	520	4,057	100%
	CONST Fan	7,971	2,102	3,180	2,085	5,265	130%
Houston	AUTO Fan	2,623	1,434	2,485	449	3,284	100%
	CONST Fan	5,656	1,393	2,492	2,453	4,944	151%

Notes: See Table 3 for summary of house characteristics. Constant infiltration/ventilation at 98 cfm (46 L/s), AUTO Fan case includes additional 40 Watts of power for mechanical exhaust fan that runs continuously year-round to exhaust 58 cfm (27 L/s). "Relative Energy Use (%)" is relative to the AUTO Fan case for each house and city (e.g., 134% means a 34% increase over the Total HVAC Electric Use for the AUTO Fan base case).

past levels (see Figure 7). Regardless, proper system sizing remains important in terms of system first costs, peak electric demand and indoor humidity levels.

**REFRIGERANT EXPANSION DEVICE**

Air-conditioning systems include a refrigerant metering device which separates the high pressure

and low pressure sides of the system. There are two common types of metering devices: fixed orifice (e.g., capillary tube or short tube restrictor) and thermostatic expansion valves (TXVs). Fixed orifice expansion devices are inexpensive and reliable, but tend to yield lower system efficiencies when cooling loads vary. TXVs are more expensive and have some moving parts which might impact reliability, but use



of these metering devices typically produces higher efficiency units due to its ability to meter the refrigerant flow rate over a wide range of cooling loads. Good TXV operation must include proper location and attachment of the sensing bulb, so installation of TXVs (at the factory or in the field) needs to be completed with care.

Hard-shutoff TXVs do a better job at maintaining high/low refrigerant pressures during compressor off period, which leads to lower start-up losses when the compressor restarts. The compressor must be able to handle the higher starting torque requirements.

**INSTALL AIRTIGHT DUCTWORK**

The previous sections describe equipment selection and operation issues. A key installation issue is air leakage from the forced air distribution system. Duct air leakage can have significant negative impacts on indoor humidity levels, energy consumption, and indoor air quality (Modera 1989, Cummings et al. 1990, Cummings et al. 1991).

If the ductwork and/or air handler are located outside of the conditioned space (e.g., attic or garage), then ductwork air leakage can cause entrainment of humid outdoor air which can significantly increase the latent cooling load to be met by the air conditioner. Locating the ductwork and air handler within the conditioned space helps alleviate this issue. Regardless of location, it is important that ductwork be sealed airtight, including the connection of the ductwork to the air handler where air pressure differences are greatest.

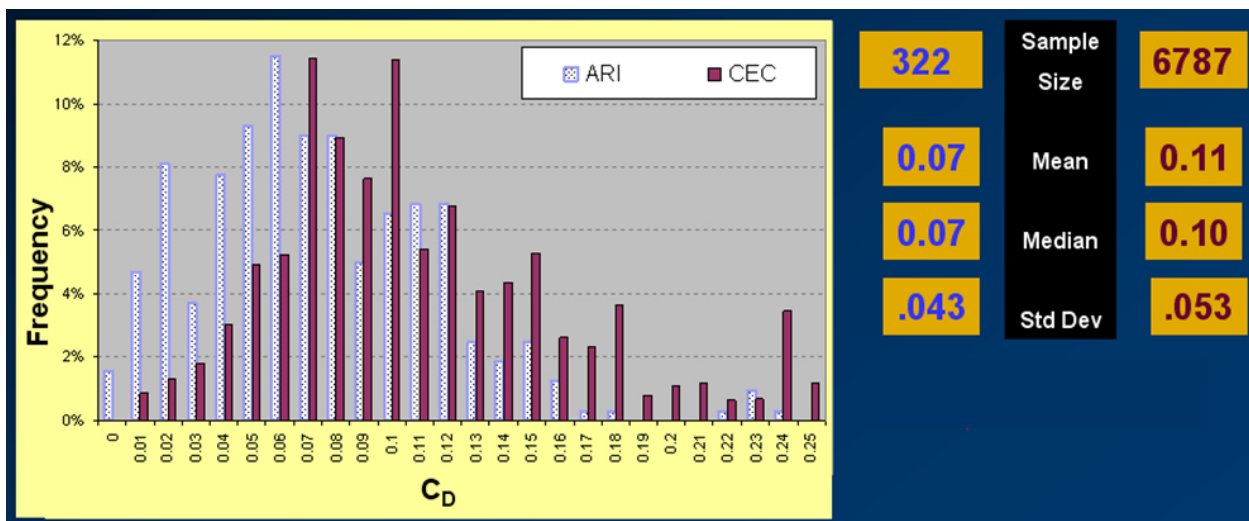
The Florida Energy Code currently encourages installing airtight ductwork by providing an energy credit for tight tested ductwork (via Compliance Method A). Other code agencies should also investigate methods for promoting improved ductwork installation practices and performance testing.

**SUMMARY AND CONCLUSIONS**

The need for residential air-conditioning systems with improved dehumidification performance is on the rise. While there are many products and strategies available, there are a number of low-cost or no-cost options regarding the selection, installation and operation of conventional AC equipment which can result in lower indoor humidity levels. These include:

- Selecting AC systems with a lower sensible heat ratio,
- Reducing the supply air flow rate,
- Disabling supply air fan overrun,
- Operating the supply air fan in AUTO mode (without fan overrun) instead of fan ON (constant) mode,
- Properly sizing the AC system, and
- Specifying an AC system with a thermostatic expansion valve (TXV)
- Install airtight ductwork

The costs and benefits of these options should be assessed prior to considering the added first cost and operating costs of separate dehumidification equipment.



**Figure 7.** Cooling Cycle Degradation Coefficient for Residential Air Conditions (Dougherty 2002)

**ACKNOWLEDGEMENT**

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**REFERENCES**

- ACCA. 2006. Manual J Residential Load Calculation (8<sup>th</sup> Edition), Arlington, VA: Air Conditioning Contractors of America.
- AHRI. 2006. Performance of Unitary Air-Conditioning and Air-Source Heat Pump Equipment, Standard 210/240, Arlington, VA: Air-Conditioning, Heating, and Refrigeration Institute.
- Cummings, J.B., Tooley, J.J., Moyer, N.A. and Dunsmore, R. 1990. Impacts of Duct Leakage on Infiltration Rates, Space Conditioning Energy Use, and Peak Electrical Demand in Florida Homes, Proceedings of the ACEEE 1990 Summer Study, Pacific Grove, CA. Washington, DC: American Council for an Energy-Efficient Economy.
- Cummings, J.B., Tooley, J.J., and Moyer, N. 1991. Investigation of Air Distribution System Leakage and Its Impact in Central Florida Homes, FSEC-CR-397-91. Cocoa,FL: Florida Solar Energy Center.
- Dougherty, B.P., J.J. Filliben and A.I. Aviles. 2002. Central Air Conditioner Test Procedure Workshop: A Technical Discussion on New Defaults for  $C_D^c$  (power point presentation to U.S. DOE). Gaithersburg, MD: National Institute of Science and Technology.
- Henderson, H.I. 1990. An Experimental Investigation of the Effects of Wet and Dry Coil Conditions on Cyclic Performance in the SEER Procedure, Proceedings of USNC/IIR Refrigeration Conference at Purdue University, July 1990, West Lafayette, IN.
- Henderson, H.I. 1998. The Impact of Part-Load Air-Conditioner Operation on Dehumidification Performance: Validating a Latent Capacity Degradation Model, Proceedings of ASHRAE IAQ & Energy '98, pp. 115-122.
- Henderson, H.I., D. Shirey and R. Raustad. 2007. Closing the Gap: Getting Full Performance from Residential Central Air Conditioners – Develop New Climate-Sensitive Air Conditioner: Simulation Results and Cost Benefit Analysis, FSEC-CR-CR-1716-07. Cocoa, FL: Florida Solar Energy Center.
- James, P., J.E. Cummings, J. Sonne, R. Vieira, J. Klongerbo. 1997. The Effect of Residential Equipment Capacity on Energy Use, Demand, and Run-Time, FSEC-PF-328-97. Cocoa, FL: Florida Solar Energy Center.
- Lennox. 2007. Engineering Data, Expanded Rating Tables, XP15 Heat Pump Outdoor Units. Bulletin No. 210459R, Model XP15-036 with AHU CBX32MV-036. April.
- Modera, M. P. 1989. Residential Duct System Leakage: Magnitude, Impacts and Potential for Reduction. *ASHRAE Transactions* 95(2): 561–569.
- Parker, D.S., J.R. Sherwin, R.A. Raustad, D.B. Shirey. 1997. Impact of Evaporator Coil Airflow in Residential Air-Conditioning Systems. *ASHRAE Transactions* 103(2): 395-405.
- Rudd, A. and H.I. Henderson. 2007. Monitored Indoor Moisture and Temperature Conditions in Humid-Climate U.S. Residences, DA-07-046, *ASHRAE Transactions* 113(1): 435-449.
- Shirey, D.B., H.I. Henderson and R. Raustad. 2006. Understanding the Dehumidification Performance of Air-Conditioning Equipment at Part-Load Conditions, Final Report, FSEC-CR-1537-05. Cocoa, FL: Florida Solar Energy Center.