FLORIDA SOLAR ENERGY CENTER®

Dehumidification at Part Load

Authors

Shirey III, Don
Henderson Jr., Hugh

Original Publication

Don Shirey III, Hugh Henderson Jr., “Dehumidification at Part Load ”, ASHRAE Journal, April 2004. This article may not be copied and/or distributed electronically or in paper form without permission of ASHRAE.

Publication Number

FSEC-GP-151-06

Copyright

Disclaimer

small building applications

Dehumi

By **Don B. Shirey III,** Member ASHRAE, and **Hugh I. Henderson Jr., P.E.,** Member ASHRAE

Engineers and equipment manufacturers need a better understanding of dehumidification performance at part-load conditions to evaluate the impacts of their design choices on indoor humidity levels, occupant comfort, and indoor air quality. Data from previous field test studies[1,2] show that the moisture removal capacity of a cooling coil degrades at part-load conditions — especially when the supply air fan operates continuously.

Figure 1 illustrates this concept with transient data from a laboratory test. Degradation occurs because a portion of the moisture that condenses on the coil surfaces during the cooling on cycle (blue data) evaporates back into the airstream when the coil is off (green data). The data in the plot shows that the transient off-cycle performance of the coil is essentially adiabatic with sensible cooling (red data) provided in conjunction with evaporation of moisture (green data) back into the airstream. The off-cycle sensible cooling diminishes with time as the amount of available moisture on the coil surfaces decreases. As a result, a cooling coil that cycles on and off in response to a control or thermostat signal will provide a smaller fraction of its total cooling capacity as moisture removal when the system spends relatively more time with the coil off. Conversely, the full latent removal capability of the system is only realized when the coil operates continuously.

The net impact of this latent degradation phenomenon is that dehumidification performance depends on the runtime fraction of the cooling coil (load divided by steady-state capacity). *Fig-*

About the Authors

Don B. Shirey III is principal research engineer at the Florida Solar Energy Center in Cocoa, Fla. **Hugh I. Henderson Jr., P.E.,** is a principal with CDH Energy in Cazenovia, N.Y.

dification at Part Load

ure 2 shows the field-measured impact of part-load operation on the sensible heat ratio (SHR) of a residential water-to-air heat pump with continuous supply air fan operation.[2] When the cooling system operates at steady-state conditions (i.e., at a runtime fraction of one), the effective SHR of the system is 0.76. However, as the compressor runs less often, the effective SHR of the cooling coil increases, meaning that less moisture removal is provided. In this case, the cooling system provided no latent removal for compressor runtime fractions less than 40%. Most cooling coils spend a large number of hours at part-load conditions. As a result, there is considerable degradation in the moisture removal capacity for a system across the cooling season. This part-load degradation causes space humidity levels to drift upwards, especially on days when cooling loads are modest.

A Model to Predict Latent Degradation

Henderson and Rengarajan[3] developed a mathematical model to predict the degradation of latent (dehumidification) capacity of single-stage cooling equipment at part-load conditions. This model, shown as a line on *Figure 2*, demonstrates agreement with these measured data. The model parameters t_{wet} and γ were derived from on-site measurements for this system to be 720 seconds (12 minutes) and 1.07 respectively.[2]

Figure 3 shows the meaning of the model parameters t_{wet} and γ. An amount of moisture (M_o) must build up on the coil before condensate falls from the coil. After this time (t_o), all the latent capacity provided by the coil is "useful" moisture removal since this condensate leaves the system through the drain. When the coil cycles off and the supply air fan continues to operate, the initial mass of moisture buildup on the coil (M_o) evaporates back into the airstream. If the cooling coil cycles back on before all the moisture has evaporated, then the time until the first condensate removal is reduced for this next cooling cycle since the coil starts out partially wetted.

The parameter t_{wet} is the ratio of the coil's moisture holding capacity (M_o) and steady-state latent capacity (Q_L). t_{wet} is the nominal time for moisture to fall from the coil (starting from a dry coil and ignoring transient effects at startup). The other parameter γ is defined as the ratio of the initial evaporation rate (Q_e) and the steady-state latent capacity (Q_L). The latent degradation model requires two additional parameters that also are associated with engineering models for part-load efficiency.[3] These additional parameters include τ, the time constant associated with latent capacity at startup (for the system in *Figure 2* it was assumed to be 75 seconds). The other parameter N_{max} is the maximum cycling rate of the thermostat as defined in the NEMA thermostat test standard.[4]

A project was initiated in 2001 to collect additional laboratory and field measurements of part-load cooling coil performance. These data are used to verify the existing latent degradation mathematical model and to refine or extend the model to predict latent degradation for a wider range of cooling systems (e.g., multistage cooling equipment and constant air volume chilled water systems).

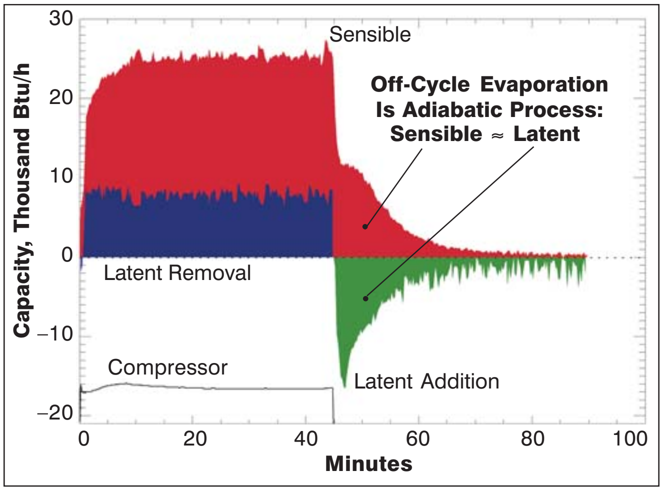

Figure 1: Transient sensible & latent capacity of cooling coil over an operating cycle (supply air fan operates continuously).

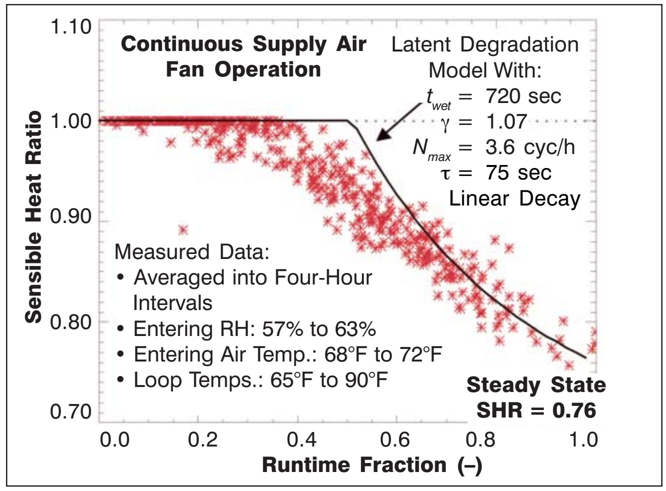

Figure 2: Field data showing the net impact of part-load operation on sensible heat ratio.[2]

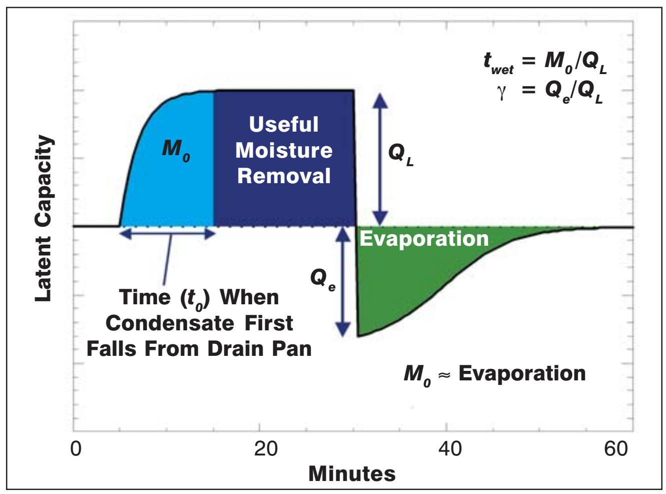

Figure 3: Concepts of moisture buildup and evaporation.

Initial Test Results

For this project, a psychrometric testing facility was set up for evaluating air conditioner cooling coils with cooling capacities up to 3 tons (10.5 kW). The facility includes indoor and outdoor test chambers capable of maintaining constant temperature and humidity conditions as specified in ANSI/ASHRAE Standard 37, *Methods of Testing for Rating Unitary Air-Conditioning and Heat Pump Equipment*.[5] The facility is able to hold the desired conditions, even for transient testing, where the cooling equipment is cycled on and off.

Laboratory testing has been completed for several coils. Each coil is tested at various entering air temperature/humidity conditions, airflow rates, and coil refrigerant temperatures. Overall, the test results from the laboratory tend to confirm the following trends:

• The off-cycle evaporation process becomes adiabatic after refrigerant migration inside the coil and system has subsided one or two minutes after the compressor stops operating. The type of refrigerant expansion device has some impact on the length of this transition.

• The mass of moisture retained on the coil surface is mostly a function of coil surface geometry with some secondary dependence on dew-point temperature and velocity of entering air.

• The calculated values of the model parameter t_{wet} are generally in line with the measured condensate delay time (t_o). The delay time is a strong function of the entering air conditions. As entering conditions are more humid, moisture builds up faster on the coil, so the time to first condensate removal is shorter. For example, the delay time for Coil 2 varies from 40 to 10 minutes as the entering dew point goes from 50°F to 70°F (10°C to 21°C).

• The moisture evaporation rate during the off cycle is a function of the wet-bulb depression (i.e., the difference between the wet-bulb and dry-bulb temperatures) of the entering air, as would be expected for an evaporative cooler. The retained moisture on the cooling coil evaporates more quickly with drier inlet air conditions.

• The off-cycle evaporation trend implies that the wetted surface area of the coil decreases in proportion to the remaining moisture mass. So the wet coil acts as an evaporative cooler with progressively less surface area as moisture is evaporated from the coil.

Table 1 compares the latent performance model parameters determined for the five coils that have been tested in the laboratory to date. The tested coils ranged in size from 1.5 to 3 tons (5.3–10.5 kW), with the total fin surface area varying by nearly a factor of two from the largest to the smallest coil. The moisture-holding capacity per total finned surface area has been very similar for most of the lab-tested coils at 8 to 9 lbs per 1,000 ft^2 (39 to 44 g/m^2) of fin area. Less accurate field measurements of moisture-holding capacity also have resulted in a similar range of values: typically 6 to 10 lb per 1,000 ft^2 (30 to 50 g/m^2). The one exception observed thus far is lab Coil 4. This vertical slab coil with wavy fins retained 50% to 60% more moisture per unit surface area than the other coils. Through continued lab testing, we intend to quantify how fin spacing,

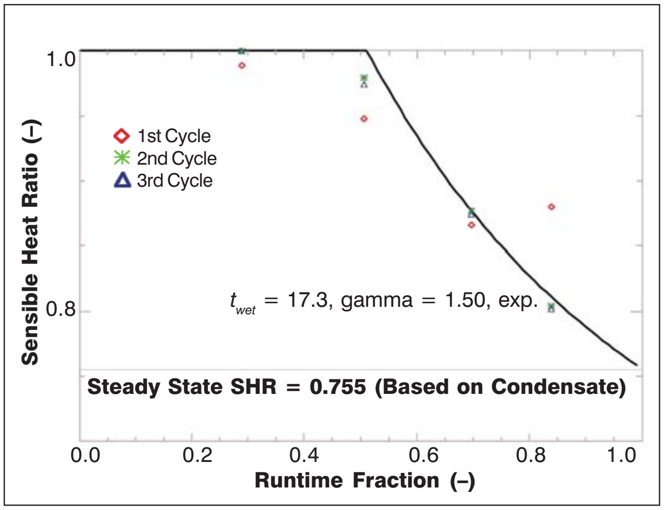

Figure 4: Comparing measured latent degradation to LHR model at nominal conditions (Coil 2).

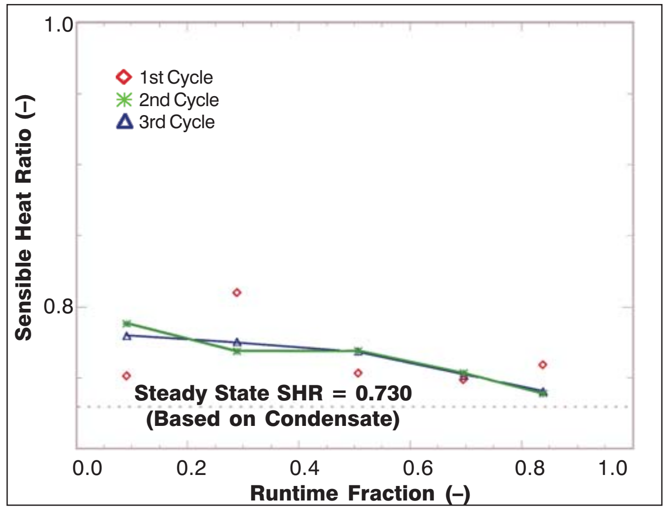

Figure 5: Measured latent degradation with cycling fan at nominal conditions.

fin type, and coil orientation affect the moisture-holding capacity of the coil.

The time for condensate to first fall from the coil (t_o) varied from 12 minutes to 33 minutes for the lab test coils at nominal conditions. Similar variations were observed for the model parameter t_{wet}. The results from field testing have generally confirmed these parameter values for other cooling coils.

As part of the test program, we have completed a series of quasi-steady cyclic tests in the laboratory with differing lengths of compressor on and off times to simulate real-world cycling performance. The lengths of the on and off times were selected to correspond with the NEMA thermostat curve with a maximum cycle rate of three cycles/hour.[4] The quasi-steady lab testing showed the same degradation trends observed in the field (e.g., in *Figure 2*) and confirmed that the model by Henderson and Rengarajan[3] could reasonably predict part-load dehumidification performance. The data points shown in *Figure 4* for Coil 2 correspond to the first, second, and third operating cycles at the same inlet air conditions and discrete runtime fractions. Typically, the results for the second and third cycles for a given runtime fraction are in close agreement, implying that at least two cycles are necessary to achieve quasi-steady conditions. The triangles from the third cycle match the latent degradation model, which is shown as a line on the plot. The parameters for the model include a t_{wet} of 17.3 minutes and gamma of 1.5 for Coil 2. Similar tests have been completed at other operating conditions that also show agreement between the model and the measured data.

	Cooling Capacity	Fin Surface Area	Moisture-Holding Capacity, M_o		Cond. Delay Time, t_o	t_{wet}
	tons (kW)	ft² (m²)	lb (kg)	lb/1,000 ft² (g/m²)	Min	Min
Coil 1 (Slanted Slab, Three Rows, 13 fpi, Plain Fins, Orifice)	3.0 (10.5)	243.8 (22.7)	2.1 (0.95)	8.6 (42.1)	13.5	16.5
Coil 2 – Normal Air Flow Rate (A-coil, Three Rows, 15.5 fpi, Lanced Sine-Wave Fins, TXV)	2.4 (8.4)	237.8 (22.1)	2.0 (0.91)	8.4 (41.1)	16.3	17.0
Coil 3 – Coil 2 with Low Airflow (A-Coil, Three rows, 15.5 fpi, Lanced Sine-Wave Fins, TXV)	1.5 (5.3)	237.8 (22.1)	2.0 (0.91)	8.4 (41.1)	32.5	29.0
Coil 4 (Vert. Slab, Two rows, 14 fpi, Wavy Fins, Orifice)	1.8 (6.3)	138.3 (12.8)	1.9 (0.86)	13.7 (67.0)	23.5	18.5
Coil 5 (Slanted Slab, Four Rows, 12 fpi, Wavy Fins, Orifice)	2.3 (8.1)	162.7 (15.1)	1.4 (0.64)	8.6 (42.1)	11.5	9.0

Notes: 1. Cooling capacity includes sensible and latent cooling at nominal conditions with airflow rate of 400 cfm/ton (54 L/s per kW). Nominal conditions correspond to ASHRAE Test A test point.
2. Fin surface area is gross fin area (coil face area × coil depth × fin spacing × 2).
3. Condensate delay time and t_{wet} are at nominal conditions.

Table 1: Comparing measured performance parameters for lab-tested cooling coils.

Laboratory testing also has been used to evaluate cooling coils with the AUTO fan mode, where the supply air fan cycles on and off with the compressor. *Figure 5* shows that degradation in latent capacity is even apparent with this fan operating mode, although to much lesser extent than with continuous supply air fan operation. Most of the coils tested in the lab so far have shown some amount of degradation in the AUTO fan mode. Data collected from field test sites also have confirmed that degradation occurs with both continuous (ON) and cycling (AUTO) fan operation.

Practical Implications on Equipment Design and Controls

This research project also measured the performance of cooling coils at six different residential and commercial field test sites. The data from these test sites generally have confirmed the findings of laboratory testing. The field measurements also have demonstrated that control and equipment configuration issues have a significant impact on latent degradation and the ability to control space humidity levels.

For instance, field measurements for a 10-ton (35 kW) packaged rooftop unit in a retail application showed that having two stages of cooling capacity significantly reduced the impact of latent degradation. The SHR of the cooling coil was maintained fairly well at part-load conditions since the system spent many hours with the first stage operating continuously. Therefore, humidity control was reasonably maintained in this commercial application even with continuous supply air fan operation. In contrast, a single-stage rooftop would have resulted in extremely poor space humidity control in this application since the compressor would have operated for several thousand hours at less than a 50% runtime fraction.

The field results show that the expected negative impact of constant fan operation in commercial applications can be greatly mitigated by specifying multistage equipment. However, engineers and designers must be sure to specify rooftop equipment with coil circuiting that provides good dehumidification at part load (i.e., using face-split evaporator coils instead of row-split coils).

Similarly, a two-stage residential cooling system with variable-speed air handler was monitored in a Florida home and was shown to experience little part-load latent degradation. Since the fan speed was modulated properly with the compressor cooling capacity, the SHR of the system was held low enough to provide adequate dehumidification when operating in first-stage cooling. The system operated continuously for hundreds of hours at low capacity, so good humidity control was maintained continuously. The high-efficiency, two-speed residential systems that now are available provide longer compressor on-times, which can result in better indoor humidity control if supply air fan speed is properly modulated with cooling capacity.

The results from this research project also confirm and quantify the impact equipment oversizing has on humidity control. Whether operating with continuous fan operation, as is common in commercial applications, or in the AUTO fan mode, as most residential systems do, dehumidification performance degrades at part-load conditions. Oversizing AC equipment increases the time spent at part load and results in higher space humidity levels. Carefully sizing equipment to match the cooling load requirements results in better humidity control and higher system efficiency since part-load losses are minimized.

Modulated chilled water coils in large commercial systems also experience latent degradation at part-load conditions. Field testing of a constant air volume chilled water coil in a Florida commercial building confirmed the expected drop in latent capacity as the water flow rate through the chilled water coil modulates to match the load requirements. In applications where improved humidity control is important, designers should consider controlling cooling capacity by bypassing air around the coil. The air bypass method clearly provides better humidity control at part load compared to systems that modulate capacity by varying the chilled water flow rate. Variable air volume systems also provide good dehumidification when controlled to an appropriate discharge air temperature.

Better Simulation Tools

The key to developing better building designs is to provide engineers and other building design professionals with better tools to understand the implications of their design decisions. Dynamic (e.g., hourly) building simulation tools provide the best means for engineers to evaluate how well their designs perform at part-load conditions.

Unfortunately, most mainstream hourly building simulation tools currently do not predict space humidity levels properly at part load. Since these mainstream simulation models do not consider part-load degradation, they tend to overestimate the moisture removal capacity of cooling equipment and predict that space humidity levels are maintained at lower levels than are observed in practice. Kosar et al.[6] showed that ignoring latent degradation causes hourly building simulation models to underpredict space humidity levels by 5% to 10% RH under typical conditions in a small office application.

The latent degradation model from Henderson and Rengarajan has already been incorporated into some whole building hourly simulation models[7,8,9] and will be available soon in other models.[10] This research project has confirmed the validity of the latent degradation model for single-capacity systems and is working to refine the model to consider more applications. We are also working to understand the moisture-related characteristics of cooling coils so we can develop guidelines for selecting model parameters for an array of cooling coil and equipment configurations.

Summary

The latent capacity of a cooling coil degrades at part-load conditions. This degradation is most significant when the supply air fan operates continuously with a single-stage cooling coil. However, some degradation also occurs with modulated and staged cooling systems as well. Continuous supply air fan operation is used in nearly all commercial buildings to provide the outdoor air ventilation requirements prescribed by ANSI/ASHRAE Standard 62, *Ventilation for Acceptable Indoor Air Quality*[11] and to provide air circulation for occupant comfort. The impact of latent degradation must be considered in these circumstances.

A recent utility research project indicated that 2% of Florida homeowners operated the supply air fan continuously.[12] This fan operation mode may become more prevalent in residential applications if its use is recommended for central air filtration systems (e.g., UVC lamps or high-efficiency air filters), ventilation requirements,[13] or occupant comfort. However, most residential air conditioners cycle the supply air fan on and off with the cooling coil in response to the thermostat signal (AUTO fan control). Most homeowners in humid regions inherently know this operating mode is preferable since it provides reasonable moisture removal. However, laboratory data have confirmed that even AUTO fan control can result in significant degradation in dehumidification performance at part load.

This study is working to quantify the equipment characteristics, control modes, and operating conditions where latent degradation is a concern. Algorithms to predict latent degradation are being incorporated into hourly whole building energy analysis tools to allow building designers and equipment manufacturers to quantify the impact as well.

Acknowledgments

This article was prepared with the support of the U.S. Department of Energy, under Award No. DE-FC26-01NT41253. However, any opinions, findings, conclusions, or recommendations expressed herein are those of the authors and do no necessarily reflect the views of the DOE.

References

1. Khattar, M., N. Ramanan, and M. Swami. 1985. "Fan cycling effects on air conditioner moisture removal performance in warm, humid climates." Presented at the International Symposium on Moisture and Humidity. April. Washington, D.C.

2. Henderson, H. 1998. "The impact of part-load air-conditioner operation on dehumidification performance: Validating a latent capacity degradation model." Presented at ASHRAE's IAQ & Energy '98 conference.

3. Henderson, H., and K. Rengarajan. 1996. "A model to predict the latent capacity of air conditioners and heat pumps at part-load conditions with constant fan operation." *ASHRAE Transactions* 102(1): 266–274.

4. NEMA. 1990. *Residential Controls—Electric Wall-Mounted Room Thermostats*, NEMA Standards Publication No. DC 3. National Electrical Manufacturers Association, Washington, D.C.

5. ANSI/ASHRAE Standard 37-1988, *Methods of Testing for Rating Unitary Air-Conditioning and Heat Pump Equipment.*

6. Kosar, D.R., M.J. Witte, D.B. Shirey, and R.L. Hedrick. 1998. "Dehumidification issues of Standard 62-1989." *ASHRAE Journal* 40(3).

7. EPRI. 2000. Supermarket Simulation Tool Version 3: Installation and User Guide, Electric Power Research Institute, Customer Solutions Group, Palo Alto, Calif. CM-111112-R4.

8. Shirey, D.B. and K. Rengarajan. "Impacts of ASHRAE Standard 62-1989 on small Florida offices." *ASHRAE Transactions* 102(1):153–165.

9. EPA. 2001. *Indoor Humidity Assessment Tool — Reference Manual*. Available at www.epa.gov/iaq/schooldesign/saves.html.

10. EnergyPlus. 2003. *Engineering Reference Manual for Energy Plus.* Available at www.energyplus.gov.

11. ANSI/ASHRAE Standard 62-2001, *Ventilation for Acceptable Indoor Air Quality.*

12. Personal correspondence with D. Parker regarding a large scale utility study: Parker, D.S., "Research highlights from a large scale residential monitoring study in a hot climate." *Proceedings of International Symposium on Highly Efficient Use of Energy and Reduction of its Environmental Impact*, pp. 108–116, Japan Society for the Promotion of Science Research for the Future Program, JPS-RFTF97P01002, Osaka, Japan, January 2002. (Also published as FSEC-PF369-02, Florida Solar Energy Center, Cocoa, Fla.)

13. ASHRAE Standard 62.2-2003, *Ventilation for Acceptable Indoor Air Quality in Low-Rise Residential Buildings.*