**Understanding the Dehumidification Performance of Air-Conditioning Equipment at Part-Load Conditions**

**Abstract**

Cooling coils provide both sensible cooling and moisture removal. Data from field test studies have demonstrated that the moisture removal capacity of a cooling coil degrades at part load conditions. Degradation occurs because moisture that condenses on the coil surfaces during the on cycle evaporates back into air stream when the coil is off. This paper presents initial laboratory and field results that characterize the part load dehumidification performance of various cooling systems. The lab and field measurements compare well to theoretical algorithms that have been developed to predict this part load phenomenon (Henderson and Rengarajan 1996). The lab data have also confirmed many of the underlying assumptions of the theoretical model. The paper also discusses the types of applications and control modes where part load latent degradation is the greatest concern.

**Introduction and Background**

Cooling coils provide both sensible cooling and moisture removal. Moisture removal or dehumidification occurs when coil surfaces are colder than the dew point of the entering air. The mix of latent and sensible capacity provided by a cooling coil at steady state conditions is well understood. The ratio of sensible to total capacity – or the sensible heat ratio (SHR) – depends on the physical characteristics of the coil as well as the refrigerant temperature inside the tube and the temperature and humidity of the entering air stream.

Data from field test studies (Khattar et al. 1985; Henderson 1998) have demonstrated that the moisture removal capacity of a cooling coil degrades at part load conditions – especially when the supply air fan operates continuously. Degradation occurs because moisture that condenses on the coil surfaces during the cooling on cycle evaporates back into air stream when the coil is off. The data in Figure 1 show that the off-cycle performance of the coil is essentially adiabatic with sensible cooling provided by the evaporation of moisture back into the air stream. The off-cycle sensible cooling diminishes with time as the available quantity of 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 have less net moisture removal as the system spends relatively more time with the coil deactivated.

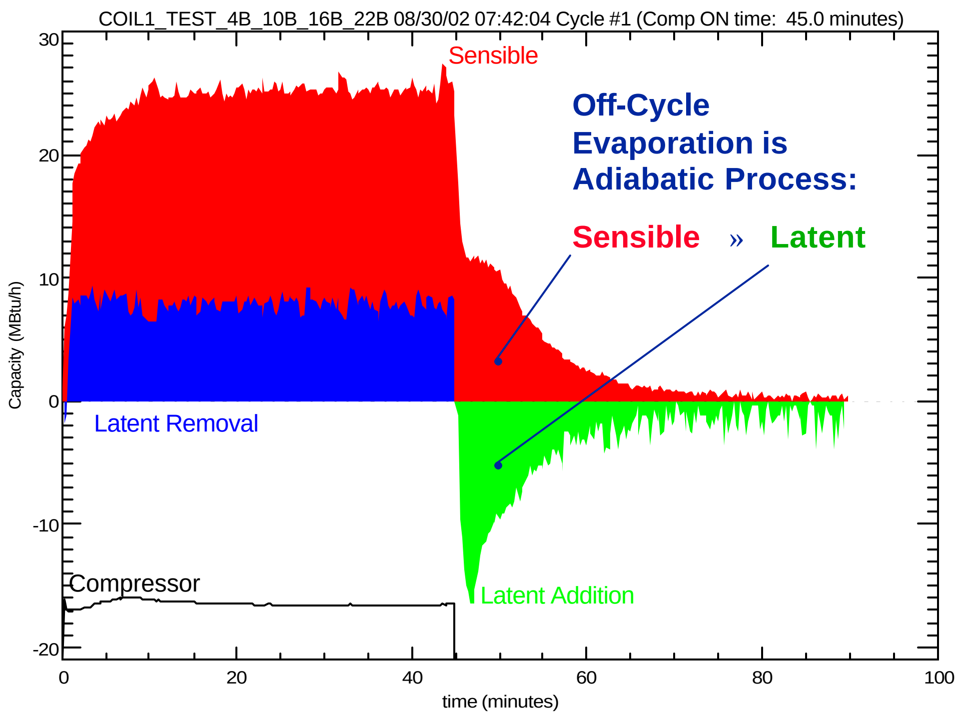


**Figure 1. Transient Sensible and Latent Capacity of Cooling Coil Over an Operating Cycle**

This paper introduces the concept of latent degradation and presents the measured results from recent laboratory and field studies that further quantify the impact this phenomenon.

**Latent Degradation Concepts**

Because of the off-cycle evaporation phenomenon described above, moisture is only removed from the space when it leaves through the condensate drain. Therefore moisture removal only begins when enough moisture has built up on fins and coil surfaces so that condensate drains from the system. Figure 2 graphically shows these concepts. An amount of moisture ($M_o$) must first build up on the coil before condensate falls from the coil. After this time ($t_o$), all the latent capacity provided the coil is "useful" moisture removal since this condensate leaves the system. When the coil cycles off, the initial mass of moisture buildup on the coil ($M_o$) evaporates back into the air stream. If the cooling coil cycles back on before the all the moisture has evaporated, then the time until the first condensate removal is reduced since the coil is already partially wetted.

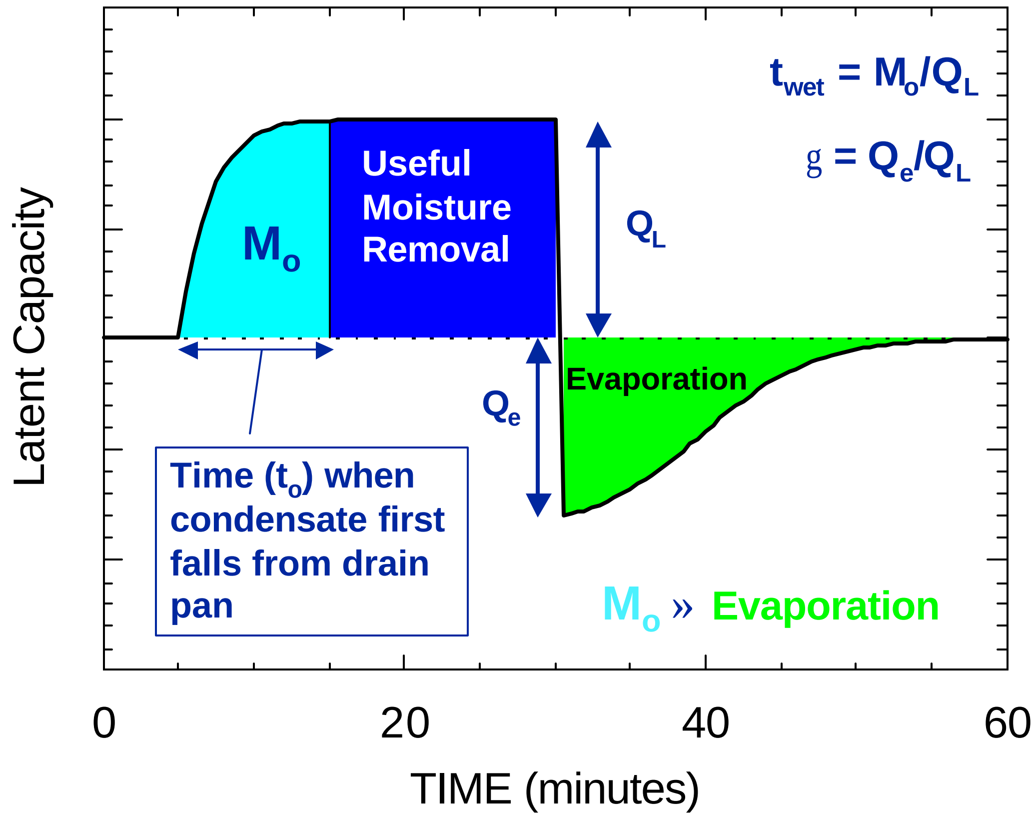


**Figure 2. Concepts of Moisture Buildup and Evaporation on the Coil**

Several useful dimensionless or normalized parameters can be derived from hypothetical example above. The ratio of the coil's moisture holding capacity ($M_o$) and steady-state latent capacity ($Q_L$) we have defined as $t_{wet}$: the nominal time for moisture to fall from the coil (ignoring transient effects at startup). The ratio of the initial evaporation rate ($Q_e$) and the steady-state latent capacity ($Q_L$) also provide a normalized measure of transient, part load performance.

Figure 3 shows the measured impact of part-load operation on the sensible heat ratio (SHR) of a conventional water-to-air heat pump with continuous supply air fan operation. When the system operates at steady-state conditions (i.e., a runtime fraction of one), the effective SHR of the system is 0.76. However, as the unit runs less often, the effective SHR of the cooling coil increases, meaning that less moisture removal is provided. This cooling system only provides sensible capacity once the runtime fraction drops below about 0.4. The most cooling coils spend a large portion of the annual operating hours at this part load condition. As a result, there is considerable degradation in the moisture removal capacity of this system across the cooling season.

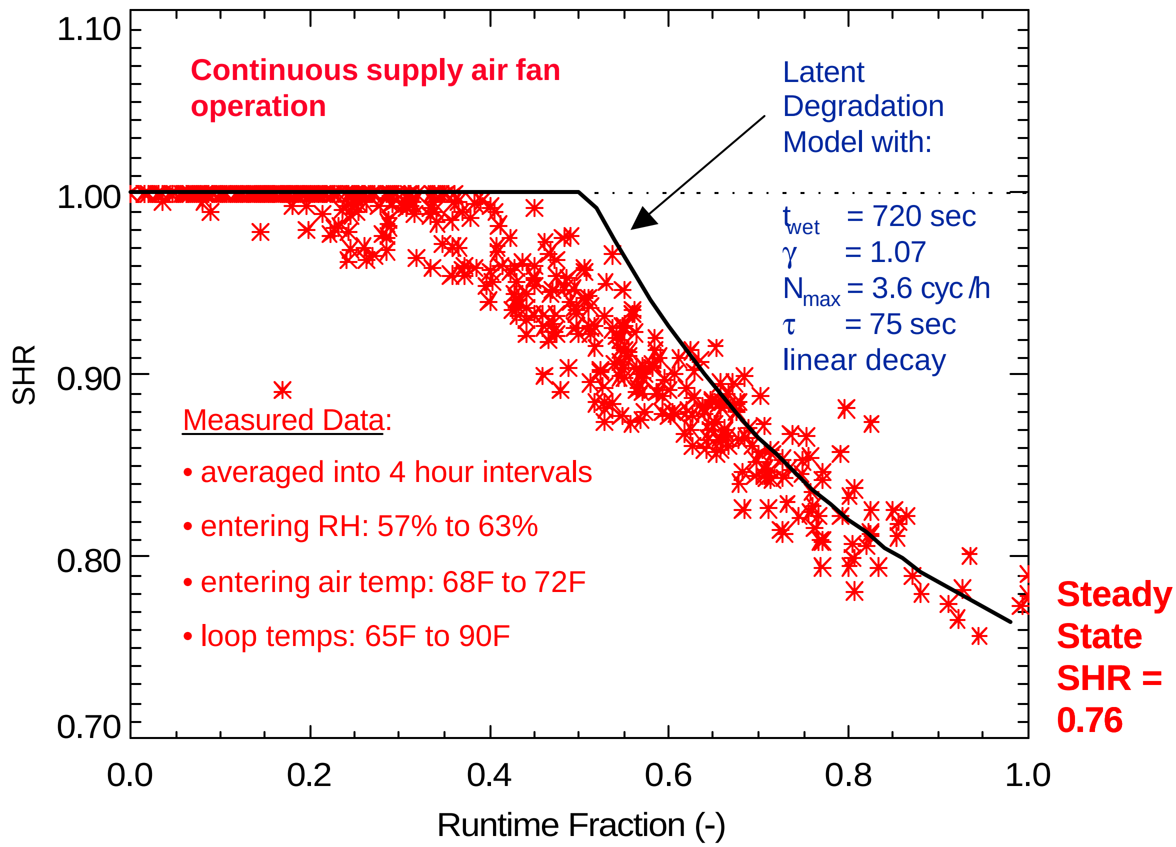


**Figure 3. Field Data Showing the Net Impact of Part Load Operation on Sensible Heat Ratio (Henderson 1998)**

Henderson and Rengarajan (1996) developed a mathematical model to predict the degradation of latent (dehumidification) capacity at part-load conditions. This model, shown as a line on Figure 3 demonstrates good agreement with this experimental data set. The model parameters $t_{wet}$ and $\gamma$ were derived from on-site measurements for this system to be 720 seconds and 1.07 respectively (Henderson 1998). The parameter $\tau$ is the time constant associated with latent capacity at startup (in this case 75 seconds). $N_{max}$ is the maximum cycling rate of the thermostat as defined in the NEMA thermostat test standard (1990). The maximum cycling rate of 3.6 cycles per hour is more rapid than the normally assumed cycling rate of 3 (which corresponds to 10 minutes on and 10 minutes off at 50% load).

### Laboratory Test Results

A psychrometric testing facility at the Florida Solar Energy Center was setup for testing air conditioner systems with cooling capacities up to 3 tons (11 kW). The test facility consists of indoor and outdoor chambers capable of maintaining constant temperature and humidity conditions for each section as specified in ASHRAE Standard 37. The instrumentation used in the lab is schematically shown in Figure 4. The instrumentation and room controls were further modified to allow for transient testing of air conditioner performance. For instance, a high precision chilled mirror dew point hygrometer with sampling tree was used instead of the wet bulb socks because of its faster response time. Similarly, the conditioning equipment and controls in the indoor and outdoor chamber were modified in order to hold constant space conditions as the cooling system cycled on and off.

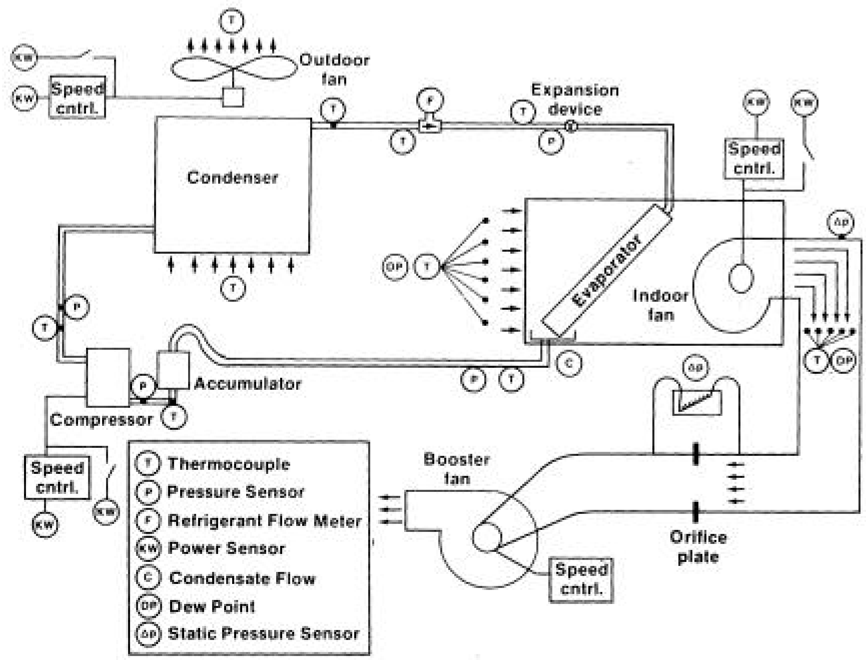


**Figure 4. Schematic of Psychrometric Chambers/Coil Testing Apparatus at FSEC**

Data from a typical laboratory run at the nominal operating point of 80°F dry bulb, 67°F wet bulb are shown in Figure 5. The plot shows the second in a series of on/off cycles. The sensible and latent cooling capacity show a slight delay on startup, bur after 40 minutes the system has reached steady state. At this point the latent capacity calculated from psychrometric properties of moist air and the measured airflow (7.4 MBtu/h [2.2 kW]) is approximately equal to the latent capacity determined from the condensate flow rate (6.7 MBtu/h [2.0 kW]). There was a delay of 16.3 minutes for the first condensate pulse to fall from the drain pan in this system.

At the beginning of the off cycle, the sensible capacity quickly drops as the system makes the transition to an evaporative cooler. Starting at the transition point identified in Figure 5 (11.9 MBtu/h [3.5 kW]), the sensible cooling is approximately equal to the latent energy associated with moisture addition. Integrating the latent (or sensible) capacity over the off cycle provides an indication of the amount of moisture that is retained on the coil surfaces. In this case, starting the integration after a one-minute delay and continuing to the designated integration point, a moisture mass of 2 lbs [0.9 kg] is retained on the coil. Dividing the moisture mass by the steady state latent capacity indicates that $t_{wet}$ is about 17 minutes in this case, which is very close to measured condensate delay time of 16.3 minutes.

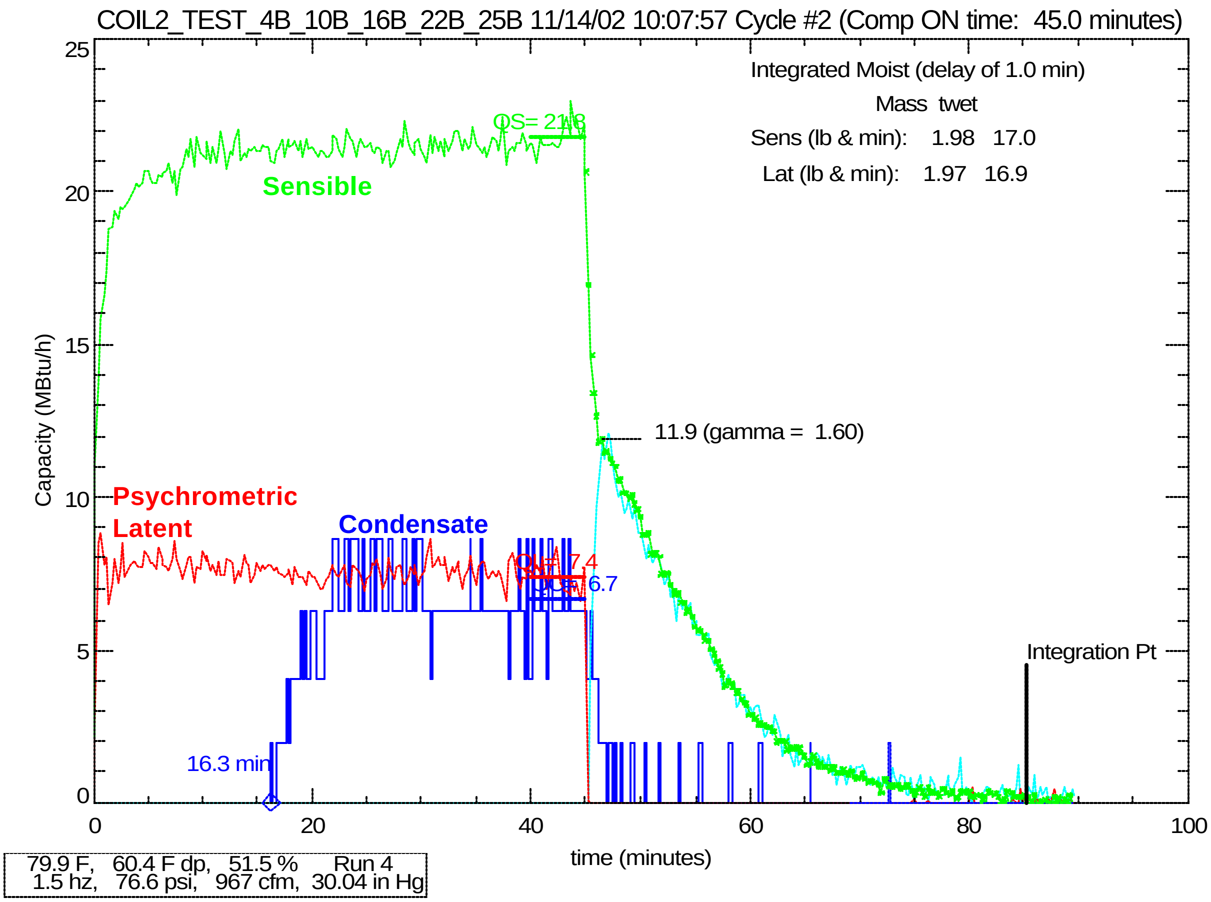


**Figure 5. Laboratory Test Data for a Typical Test Run**

This type of test has been completed for several coils. Each coil is tested at various entering air conditions, flow rates, and coil refrigerant temperatures. In addition, several quasi-steady cyclic tests were completed with differing lengths of on and off times. The lengths of the on and off times were selected to correspond the NEMA thermostat curve with a maximum cycle rate of 3 cycles/hour (NEMA 1990).

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 after 1 or 2 minutes. The type of expansion device has some impact on the duration of the transition.

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

- The calculated values of the parameter $t_{wet}$ are generally in line with the measured condensate delay time. The delay time is strong function of the entering conditions. As entering conditions are more humid, moisture buildings up faster on the coil so the time to the first condensate pulse is shorter. For the coil shown in Figure 5, the delay time varies from 40 to 10 minutes as the entering dew point goes from 50°F to 70°F [10°C to 21°C].

- The evaporation rate during the off-cycle is function if the wet bulb depression, as would be expected for an evaporative cooler. The retained moisture evaporates more quickly at dryer 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.

The table below compares the overall latent performance parameters determined for the coils that have been tested to date. The tested coils have ranged in size from 1.5 to 3 tons [6.3 to 1.5 kW]. The total surface area of the fins varies by nearly a factor of 2 from the largest to the smallest coil. Surprisingly, the amount of moisture retained on the coil surface showed much less variation, ranging from 1.9 to 2.5 lbs [0.86 to 1.14 kg]. The retained moisture per unit surface area was 30% greater for Coil 4, the vertical slab coil with plain fins. The time for condensate to first fall from the coil varied from 13 minutes up to 33 minutes, with similar variations for the parameter $t_{wet}$. Similar results and trends have been realized from the field measurements.

**Table 1. Comparing Measured Performance Parameters for Tested Cooling Coils**

| | **Cooling Capacity (ton / kW)** | **Fin Surface Area ($ft^2$ / $m^2$)** | **Retained Moisture Mass (lb / kg)** | **Condensate Delay Time (min)** | **$t_{wet}$ (min)** |
|---|---|---|---|---|---|
| Coil 1<br>(Slanted slab, 3 row, 17 fpi, plain fins) | 3.0 / 10.5 | 253.3 / 23.5 | 2.1 / 0.95 | 13.5 | 16.5 |
| Coil 2 – Normal Flow<br>(A-coil, 3 rows, 15 fpi, lanced fins) | 2.5 / 8.8 | 237.8 / 22.1 | 2.5 / 1.14 | 16.3 | 19.0 |
| Coil 2 – Low Flow<br>(A-coil, 3 rows, 15 fpi, lanced fins) | 1.5 / 5.3 | 237.8 / 22.1 | 2.5 / 1.14 | 33.3 | 35.4 |
| Coil 4<br>(vert. slab, 2 rows, 14 fpi, plain fins) | 1.8 / 6.3 | 138.3 / 12.8 | 1.9 / 0.86 | 23.5 | 18.5 |

Notes: 1- Cooling capacity includes sensible and latent cooling at nominal conditions. Nominal conditions correspond to ASHRAE Test A or ISO T1 test points.
2- Surface area is gross fin area (coil face area x coil depth x fin spacing x 2).
3- Condensate delay time and $t_{wet}$ are at nominal conditions

The quasi-steady testing also confirmed that the model by Henderson and Rengarajan (1996) could reasonably predict partload performance over a range of conditions. The data points shown on the plot correspond to the first, second, and third operating cycles at the same conditions. Typically the second and third cycles correspond, implying that quasi-steady conditions have been achieved by the second cycle. The triangles from the third cycle correspond well to 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.

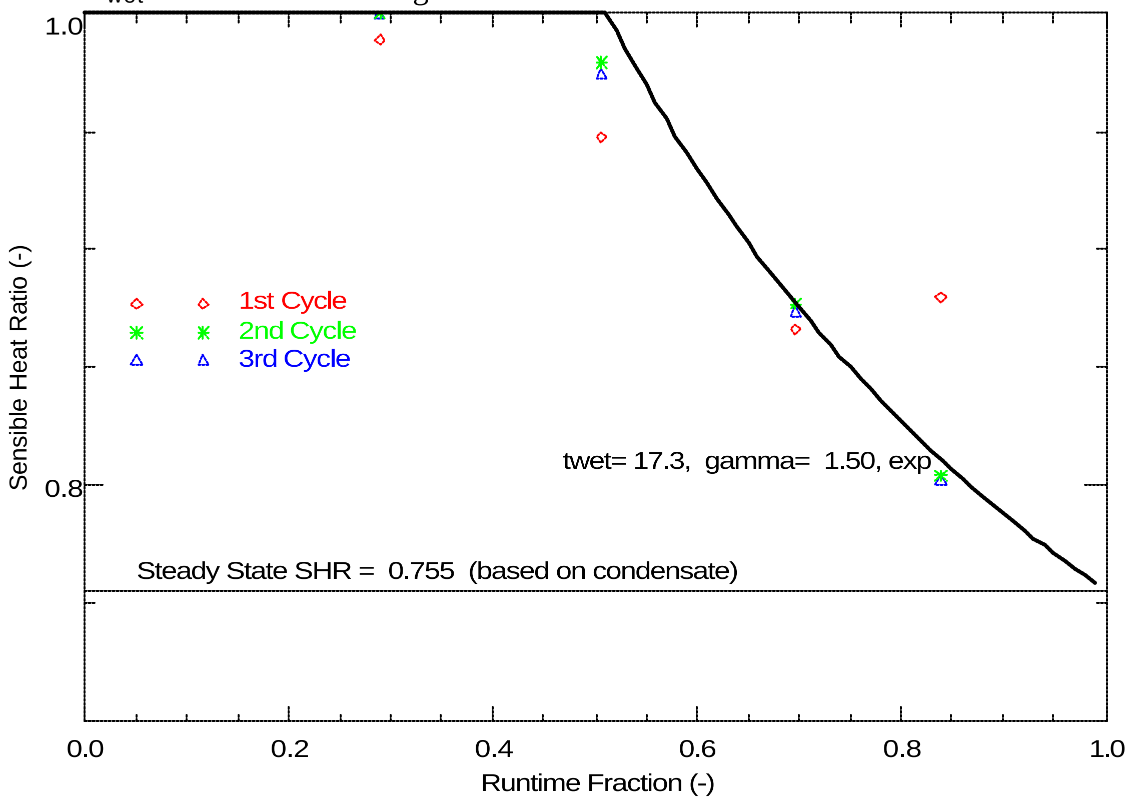


**Figure 6. Comparing Measured Latent Degradation to LHR Model at Nominal Conditions (Coil 2)**

The laboratory testing also evaluated cooling coils in the AUTO fan mode: with the supply fan cycling on and off with the compressor. Figure 7 shows that some degradation in latent capacity is apparent in this operating mode, even though it would seem that very little off-cycle evaporation should occur with the fan off. So far, most of the coils tested in the lab have shown some amount of degradation in the AUTO fan mode. Figure 8 shows similar part load degradation trends from a residential field test site. In this case significant degradation was also apparent in the AUTO fan mode. The limited amount of data in the constant fan mode also agreed well with the latent degradation model.

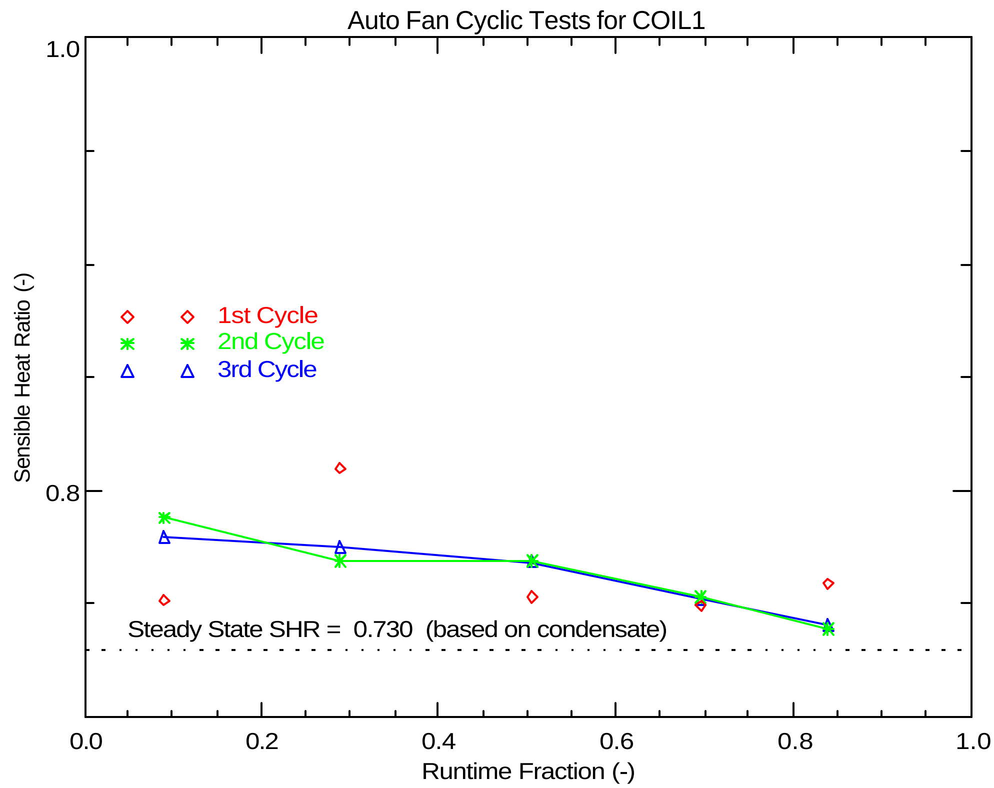


**Figure 7. Measured Latent Degradation with Cycling Fan at Nominal Conditions**

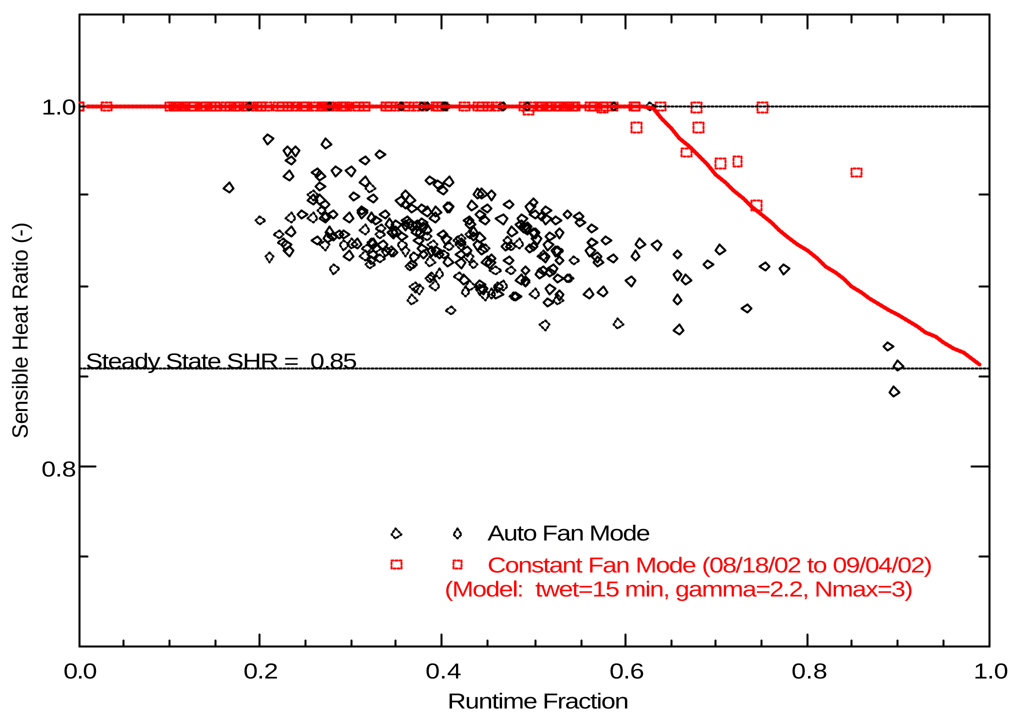


**Figure 8. Measured Latent Degradation from a Residential Field Test Site (AUTO and CONST Fan Modes)**

## Implications of Latent Degradation

The data from field test sites also confirmed the general findings of the laboratory testing. The field measurements have also demonstrated the impact that control issues such as multiple capacity stages have on latent degradation. For instance measurements on a 10 ton [35 kW] packaged rooftop in a retail application showed that having two stages of capacity significantly reduced the impact of latent degradation since the system spends many hours with first stage operating. Therefore, latent degradation was modest in spite of constant fan operation. Similarly, a two-speed residential cooling system installed in a Florida home was shown experience very little degradation. Field testing of chilled water coils in a Florida commercial building have shown latent degradation impacts are significant as the chilled water coil modulates to match the load requirements.

The latent degradation model from Henderson and Regarajan (1996) has been incorporated into several whole building hourly simulation models to predict the degradation of latent capacity at part load conditions (Shirey and Rengarajan 1996; EPRI 2000; EnergyPlus 2003). These building models do not consider this part load degradation effect and therefore tend to over estimate the moisture removal capacity of cooling equipment and predict that space humidity levels are maintained at lower levels then is observed in practice. Kosar et al. (1998) showed that ignoring latent degradation causes hourly building simulation models to under predict space humidity levels by 5-10% RH under typical conditions.

## Summary

Latent degradation is most significant when the fan operates continuously on a single-stage cooling coil. However, some degradation also occurs on all modulated and stage cooling systems as well. Continuous supply air fan operation is required in nearly all commercial buildings in order to provide the outdoor air ventilation requirements prescribed by ASHRAE Standard 62 (ASHRAE 2001) and equivalent international ventilation standards. Constant fan operation is also becoming more popular for residential applications for a variety of reasons, including the use of various air cleaning technologies such as high-efficiency filters and $UV_c$ lamps. The impact of latent degradation must be considered in these circumstances.

In the US, most residential air conditioners cycle the supply air fan on and off with the cooling coil cycle in response to the thermostat signal (AUTO fan control). Homeowners in humid regions inherently know that this operating mode is preferable since it provides reasonable moisture removal. However, the laboratory data have confirmed that even AUTO fan control can result in significant degradation in dehumidification performance at part load. This study is attempting 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 in order to allow other engineers to quantify the impact as well.

## Acknowledgements

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