The Case of the Wet Filters

Investigating the link between wet filters and blow-through air-handling units in health-care and laboratory facilities

By

Gerald Williams, PE, LEED AP

With ever-increasing pressure to design for lower space temperatures and relative humidities, engineers are providing colder, drier supply-air streams requiring deeper cooling coils and lower chilled-water temperatures for health-care and laboratory applications. A solution often employed is a blow-through air-handling unit (AHU) with the cooling coil and high-efficiency final filters on the discharge side of the fan. With this arrangement, fan heat appears on the entering side of the cooling coil, resulting in a potentially lower supply-air temperature and a reduced air-delivery requirement.

The use of blow-through-type AHUs in hospitals and laboratories is not new. What is new are the low dry-bulb and dew-point temperatures they often are designed to achieve. The resultant operational issues require a better understanding of the physical and psychrometric processes at work, which will be explained in this article.

Historical Perspective

About 30 years ago, I was just out of college, working for a consulting engineering firm in the Midwest. One day, one of the principals asked me to accompany him on a trip to investigate a problem at a local hospital.

When the principal and I arrived at the hospital, the chief engineer took us to a large AHU serving a major portion of the patient-care areas, providing 100-percent outside air for makeup. The blow-through-type AHU had a large airfoil fan discharging into a target plate designed to spread air over the face of an eight-row, 12-fin-per-inch chilled-water coil located just downstream. Several feet beyond the chilled-water coil, which was discharging approximately 52°F air, was a large bank of 95-percent-efficient final filters. Everything on the discharge side of the coil was wet. This included all of the filters in the approximately 8-ft-tall-by-16-ft-wide array, which were soaked along the sides, at the top, and at the bottom. The chief engineer wanted to eliminate the problem because the relatively expensive filters had to be changed every few weeks.

My boss and the chief engineer agreed the problem likely was carryover off of the cooling coil resulting from high-velocity air being discharged from the fan and unevenly distributed across the face of the coil. (For more on carryover, see the sidebar on the next page.) My boss and I presented a proposal to study and correct the problem, but the chief engineer apparently found another way to address the issue, as we never heard from him again.

Although I did not voice my concerns at the time, something about what I saw that day always bothered me. Conventional wisdom certainly pointed to carryover off of the coil as the source of the problem, but as hard as I tried, I could not see a single droplet of water coming off the face of the coil. There was no bridging between the fins, no overflowing primary or intermediate drain pans--literally, nothing unusual. In short, I could see no evidence of carryover by studying the exit face of the coil. I was convinced that another process--one I did not yet understand--was at work.

In the years that followed, I saw that phenomenon to varying degrees of severity several other times, always involving AHUs with blow-through cooling coils and downstream high-efficiency filters. In 2005, the following investigation provided an opportunity to truly understand the processes involved.

Case Study

The problem. Early in 2005, two identical blow-through air-handling systems (Figure 1) were installed on a medical campus as part of a multiphase project. The characteristics of the fan and cooling coils in each AHU were as follows:

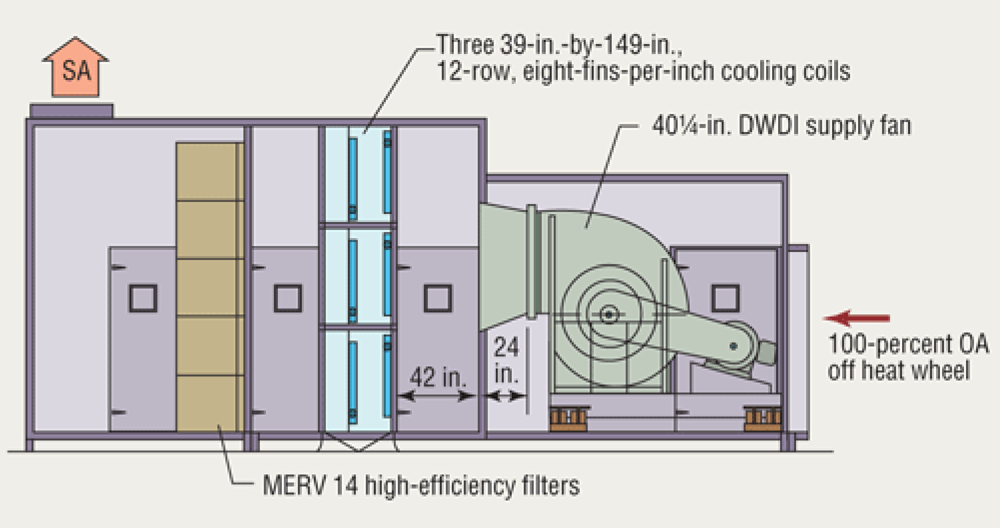


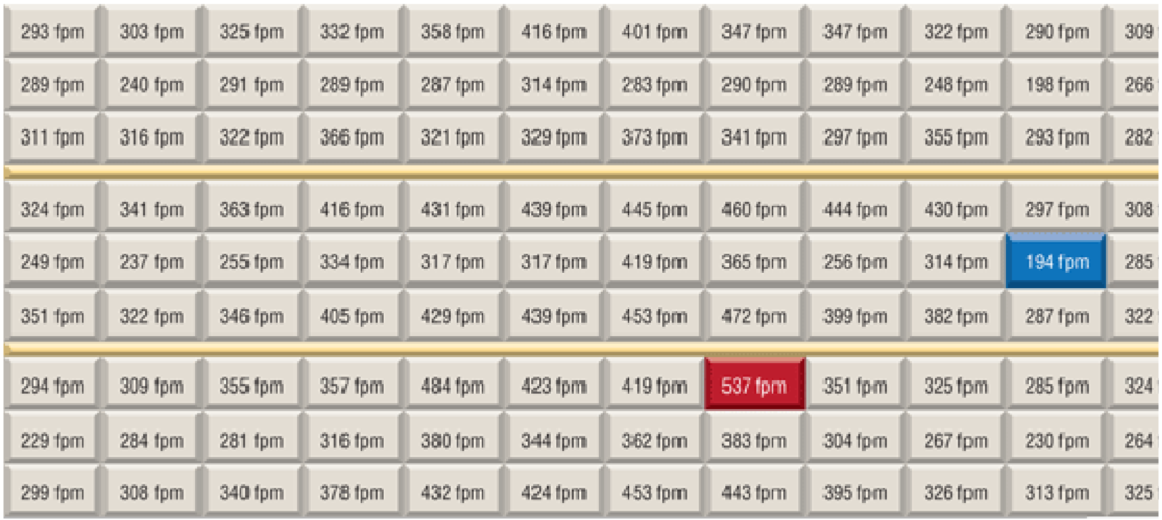
FIGURE 1. One of two identical blow-through AHUs installed on medical campus.

- Fan: 54,000 cfm, 40.25-in. double-width-double-inlet (DWDI) airfoil fan; 10.95-in. total pressure; 112.0 bhp; 125-hp motor with variable-speed drive.
- Cooling coils: Three 39-in.-by-149-in., 12-row chilled-water coils with eight mildly enhanced fins per inch; 648-gpm chilled water at 42°F; 88.1°F-dry-bulb (db), 70.6°F-wet-bulb (wb) entering air; 49°F-db design leaving air; 121.06-sq-ft total coil face area; 446-fpm average coil face velocity, with primary and intermediate drain pans.

The operational problem was the wetting of the cooling-coil discharge chamber of each AHU and the soaking of all of the MERV (minimum-efficiency reporting value) 14 filters installed downstream. The physical-plant director was anxious to solve the problem because of both the cost of frequent filter changes and the possibility of mold forming in the system.

A quick review of Figure 1--specifically, the absence of a target plate on the discharge of the fan--could lead one to conclude that coil carryover was the problem. Yet the average coil face velocity was quite low: less than 450 fpm. Additionally, the fan supplied a variable-air-volume system; even during summer conditions, the fan never was seen to operate above 45 Hz (75 percent of full speed).

With one of the units operating at 45 Hz, a field test was conducted to determine actual fan/coil-system performance. For each of the three coils, face velocity was measured at 36 locations, with each location measuring approximately 1.12 sq ft. Table 1 shows raw face-velocity data. Figure 2 shows curves of constant face velocity as a percentage above and below the average face velocity.



293 fpm	303 fpm	325 fpm	332 fpm	358 fpm	416 fpm	401 fpm	347 fpm	347 fpm	322 fpm	290 fpm	309
289 fpm	240 fpm	291 fpm	289 fpm	287 fpm	314 fpm	283 fpm	290 fpm	289 fpm	248 fpm	198 fpm	266
311 fpm	316 fpm	322 fpm	366 fpm	321 fpm	329 fpm	373 fpm	341 fpm	297 fpm	355 fpm	293 fpm	282
324 fpm	341 fpm	363 fpm	416 fpm	431 fpm	439 fpm	445 fpm	460 fpm	444 fpm	430 fpm	297 fpm	308
249 fpm	237 fpm	255 fpm	334 fpm	317 fpm	317 fpm	419 fpm	365 fpm	256 fpm	314 fpm	194 fpm	285
351 fpm	322 fpm	346 fpm	405 fpm	429 fpm	439 fpm	453 fpm	472 fpm	399 fpm	382 fpm	287 fpm	322
294 fpm	309 fpm	355 fpm	357 fpm	484 fpm	423 fpm	419 fpm	537 fpm	351 fpm	325 fpm	285 fpm	324
229 fpm	284 fpm	281 fpm	316 fpm	380 fpm	344 fpm	362 fpm	383 fpm	304 fpm	267 fpm	230 fpm	264
299 fpm	308 fpm	340 fpm	378 fpm	432 fpm	424 fpm	453 fpm	443 fpm	395 fpm	326 fpm	313 fpm	325

Notes:

1. Cooling-coil bank consists of three stacked coils, each 39 in. high by 149 in. long, with a total face area of 121.06 sq ft.
2. Velocity measurements made at the center of 108 grids, each 13 in. high by 12.5 in. wide.
3. Average face velocity 339 fpm with drive at 45 Hz.
4. Highest face velocity 537 fpm, 58 percent above average face velocity (shown in red).
5. Lowest face velocity 194 fpm, 43 percent below average face velocity (shown in blue).

Williams FIGURE Table 1 (27 or 41 picas wide)

TABLE 1. Raw face-velocity data.

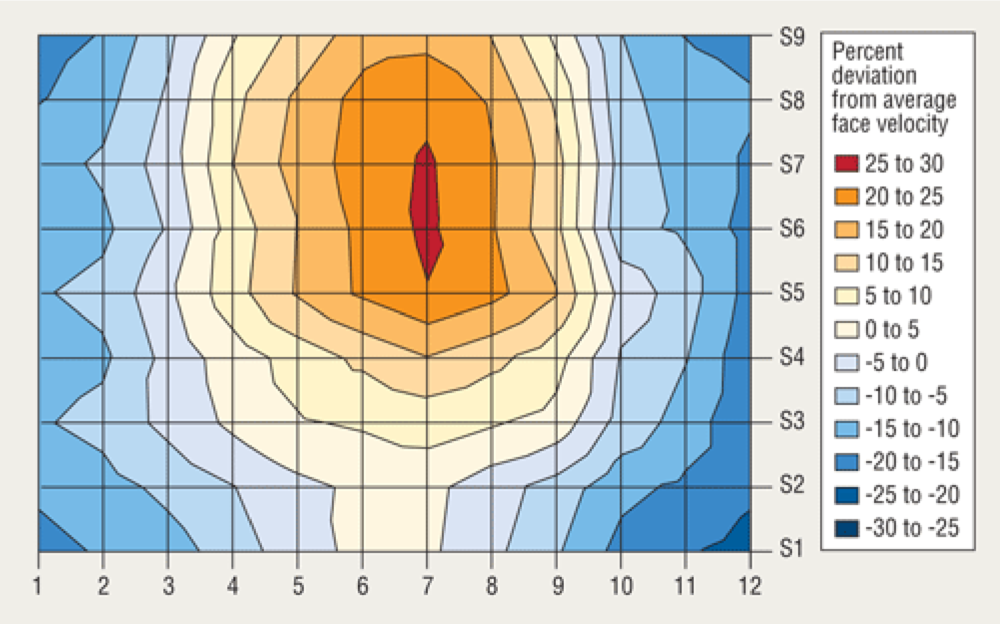
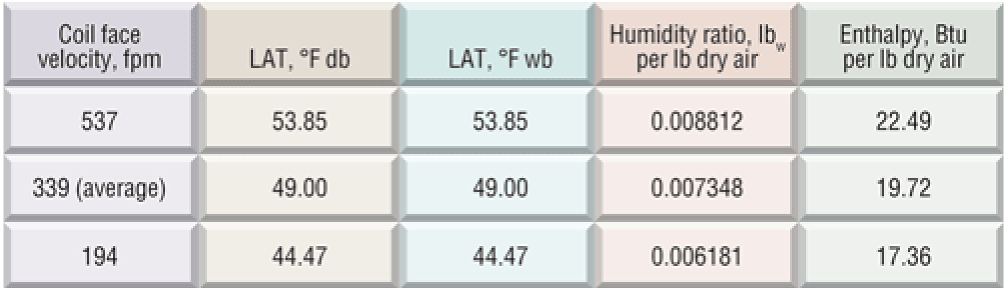


FIGURE 2. Averaged face-velocity profiles.

Figure 2 clearly indicates the signature left by the high-velocity fan discharge toward the top center of the coils. Interestingly, at 45-Hz operation, the four highest face velocities were 537 fpm, 484 fpm, 472 fpm, and 460 fpm, while the average face velocity was 339 fpm; at 60-Hz operation (which seemed unlikely from the loads calculated), the four highest face velocities would have been 716 fpm, 645 fpm, 629 fpm, and 613 fpm, while the average face velocity would have been 452 fpm. According to data from a manufacturer of a physically similar coil with equivalent fin character and spacing, carryover should begin to occur at approximately 615-fpm face velocity.

At the normal operating condition of approximately 45 Hz, then, carryover should not have occurred. Even at 60 Hz, carryover would have occurred at only three or four points on the coil (about 4 percent of the total face area). How, then, could the entire filter bank have become soaked?

The solution. A diagnostic computer simulation was performed. First, operation with a 339-fpm average face velocity was simulated, revealing that 250 gpm of chilled water was required to maintain a 49°F supply-air temperature. Then, operation at this chilled-water flow rate, but with the face velocity altered to the highest reading of 537 fpm and the lowest reading of 194 fpm, was simulated. The results are given in Table 2.



Coil face velocity, fpm	LAT, °F db	LAT, °F wb	Humidity ratio, lb_w per lb dry air	Enthalpy, Btu per lb dry air
537	53.85	53.85	0.008812	22.49
339 (average)	49.00	49.00	0.007348	19.72
194	44.47	44.47	0.006181	17.36

TABLE 2. Results of diagnostic computer simulation.

Although the psychrometrics at work were more complicated than the simulations indicated, the deep cooling coil clearly was producing different leaving-air temperatures based on the varying velocities on the coil face. Furthermore, all of the leaving conditions were at saturation. A psychrometric-chart saturation curve enhanced to make the process easier to visualize is shown in Figure 3. The mix-air humidity ratios of all adjacent air streams are in the fog region (the area to the left of the saturation curve). Thus, the mix condition of any air stream, regardless of face velocity, would have yielded a state point consisting of saturated air and suspended water droplets, or fog. An approximate calculation indicates that for this particular test point and entering-air condition, the coil produced about 12 lb per hour of water droplets as fog. That was more than enough to cause the soaking of the filters because the water droplets could not have re-evaporated without heat having been added to the air stream.

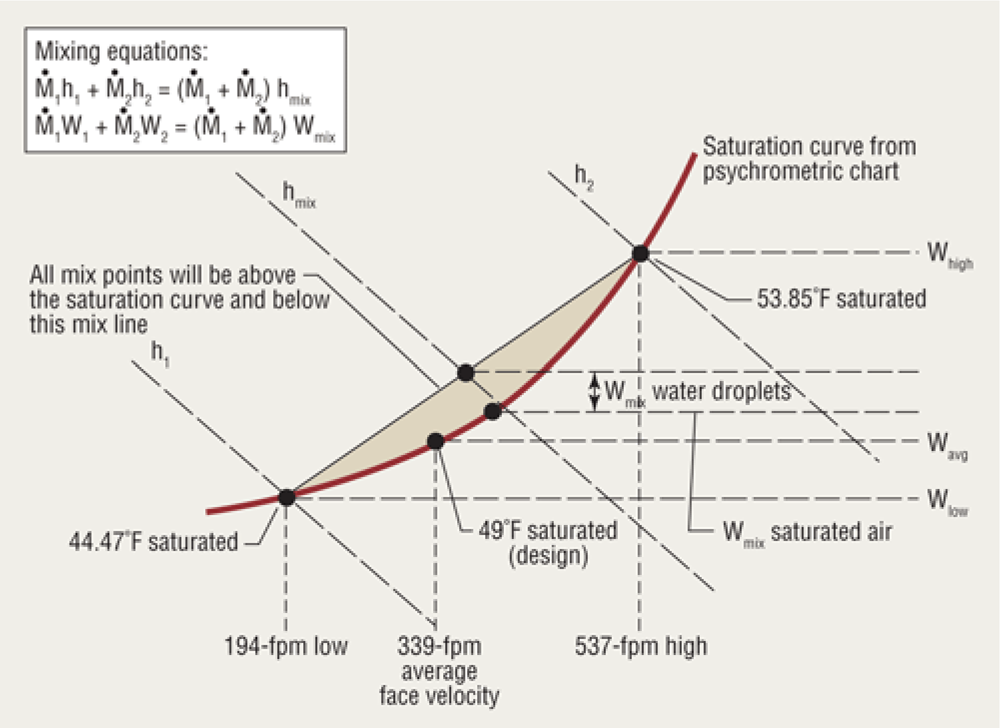


FIGURE 3. Enhanced psychrometric-chart saturation curve.

The solution was to provide reheat to the leaving-air stream. To that end, the 12-row cooling coil was replaced with a 10-row cooling coil, which load calculations indicated could easily handle the actual load, and a one-row reheat coil (Figure 4). Return chilled water from the 10-row coil was routed to the reheat coil to increase the temperature of the leaving-air stream by 1.7°F. The psychrometrics of this modification are shown in Figure 5. Because the unit may need to operate at 60 Hz in the future, a target plate was installed at the fan discharge to even the velocity distribution across the coil and prevent actual carryover. The addition of the target plate reduced the face-velocity variation to a range of +34 percent to –32 percent.

The retrofitted system was operated during the summers of 2006 and 2007 with no wetting of the final filters.

Obviously, the problem was created not by the blow-through unit, but by the extremely uneven distribution of air across the face of the cooling coil. Even with a target plate installed, airflow variations of ±25 percent or more are not uncommon. A prudent designer anticipating operation at low discharge temperatures should: (1) make every effort to ensure face velocity is as uniform as possible and (2) simulate operation at flow rates at least ±25 percent of design to determine if the formation of water droplets (fog) downstream may be an issue, in which case reheat must be applied to minimally warm the exit air, moving the mix points away from the fog region above the saturation curve.

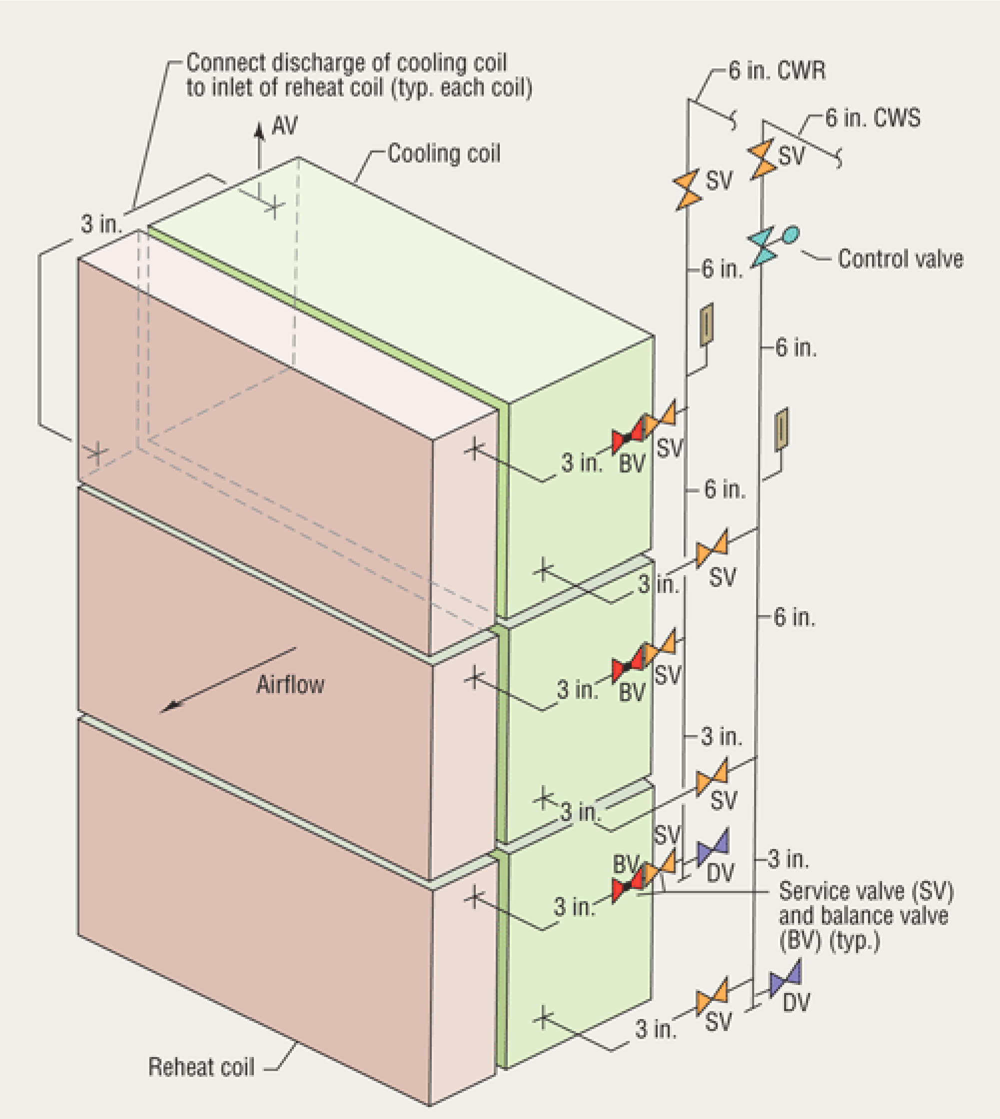


FIGURE 4. Retrofit of blow-through air-handling unit.

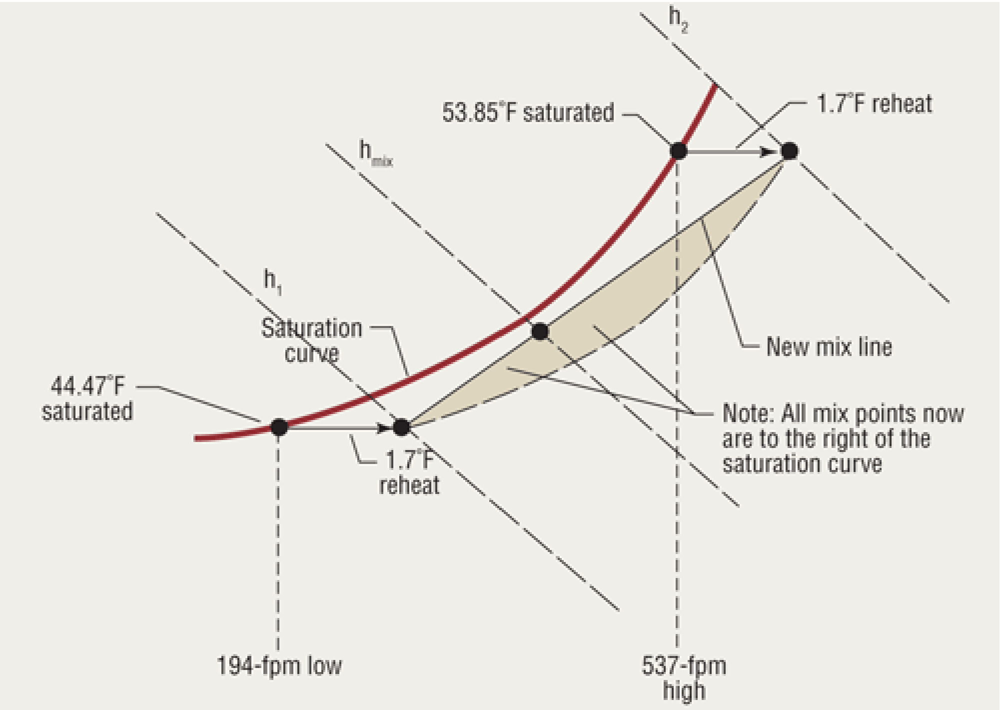


FIGURE 5. Psychrometrics of blow-through-air-handling-unit retrofit.

Sidebar: Condensate Carryover

Condensate carryover is a mechanical process by which liquid water that condenses as part of the cooling and dehumidifying process is blown off the exit face of a cooling coil by the kinetic energy of an air stream. The velocity at which carryover begins to occur depends on:

- Coil-fin spacing.
- Fin-surface character (waviness).
- Coatings that may have been applied to the fin surface to delay the onset of carryover.
- Fin-surface cleanliness.
- Tube spacing.
- Coil-face height.
- The amount of moisture on the coil.

This limiting face velocity generally is not related to the number of rows of depth of a coil, except to the extent additional rows modify fin-surface cleanliness and face height. Consult the manufacturer to determine the carryover characteristics of a particular coil.

The president of McClure Engineering Associates and a member of HPAC Engineering's Editorial Advisory Board, Gerald J. Williams, PE, LEED AP, has more than 30 years of experience in the design, analysis, and commissioning of building HVAC systems. He can be contacted at jwilliams@mcclureeng.com.