ENGINEERING WHITE PAPER

TRICOIL® Variable Air Volume Application

Reset-schedule dehumidification control for VAV reheat systems, using a single recuperative run-around loop

Room latent cooling capacity held constant, 100%–50% turndown

No separate reheat energy source required

September 2026

1. Purpose

This paper presents the TRICOIL® recuperative run-around loop applied to a Variable Air Volume (VAV) reheat system, and works a complete numeric design example showing how the method solves VAV's characteristic part-load dehumidification weakness. It updates and verifies a 2003 technical bulletin originally issued by Sensible Equipment Company; the underlying method and example are unchanged, but the supporting calculations have been independently re-verified (Section 9) and one arithmetic transcription error in the original has been corrected.

2. The VAV Part-Load Dehumidification Problem

VAV systems provide an economical means of controlling multiple zone temperatures from a single air handling unit: supply air volume to each zone is throttled up or down in response to the zone's dry-bulb temperature, rising to meet greater cooling demand and falling as the zone cools.

That throttling creates a well-known side effect. Supply air delivered to a room does two jobs at once — it removes sensible heat, and it removes moisture (latent cooling). Reducing supply air volume achieves the intended reduction in sensible cooling, but it also reduces latent cooling, since less air means less dehumidification capacity delivered to the space. In most applications a modest loss of part-load dehumidification capacity is unimportant. Where it is not — classrooms, auditoriums, and other occupancies with a high latent load relative to sensible load, or where humidity control is critical — the drop in room dehumidification at reduced airflow can push room relative humidity above target at exactly the part-load conditions the system spends most of its time in.

3. The TRICOIL® Method Applied to VAV

TRICOIL® addresses this by resetting both the supply air dry-bulb temperature and the supply air dew point in response to changes in supply air volume, rather than holding either fixed:

- **Supply air dry-bulb temperature is reset upward as supply air volume drops.** This keeps the delivered air volume — and with it, room air movement — higher than a fixed-temperature reset would allow at the same sensible load, sustaining comfort-level circulation at part load, and keeps supply air warm enough to avoid cold air dropping out of the diffuser or causing diffuser sweating.
- **Supply air dew point is reset downward as supply air volume drops.** Drier supply air removes more moisture per unit of air delivered — this is the core of the method, decoupling dehumidification capacity from supply air volume so the room's required latent cooling can be held constant even as sensible-driven airflow is throttled down.

The result is a system that tracks the sensible load with air volume in the usual VAV fashion, while holding latent (dehumidification) capacity essentially flat across the full turndown range — solving the part-load humidity problem inherent to conventional VAV reheat.

4. Worked Example

Given: A room has a peak room sensible heat gain (RSHG) of 260.4 mbh and a room latent heat gain (RLHG) of 69.7 mbh. At the minimum part-load condition under consideration, room sensible load falls to 97.7 mbh while the room latent cooling requirement remains at 69.7 mbh (latent load does not scale down with sensible load — it is driven by occupancy and infiltration, not by the sensible cooling need). The room is to be held at 75°F and 50% RH (humidity ratio = 0.0093 lb water/lb dry air) across the full range from maximum to minimum sensible load.

Solution: using a 20°F supply-air-to-room-air temperature differential at peak load and a 15°F differential at part load:

Peak Load Supply Air Volume

$$\mathrm{CFM\ SA} = \mathrm{RSHG}\ /\ [1.085 \times (\mathrm{T\ ROOM} - \mathrm{T\ SUPPLY\ AIR})] = 260{,}400\ /\ [1.085 \times (75-55)] = 12{,}000\ \mathrm{cfm}$$

Part Load Supply Air Volume

$$\mathrm{CFM\ SA} = \mathrm{RSHG}\ /\ [1.085 \times (\mathrm{T\ ROOM} - \mathrm{T\ SUPPLY\ AIR})] = 97{,}700\ /\ [1.085 \times (75-60)] = 6{,}000\ \mathrm{cfm}$$

(The part-load supply air temperature is 60°F, not 55°F — see Section 9 for a note on this correction relative to the original 2003 bulletin.)

Peak Load Supply-Air-to-Room-Air Humidity Ratio Differential

$$\Delta \mathrm{W1\ (ROOM - SUPPLY\ AIR)} = \mathrm{RLHG}\ /\ [4840 \times \mathrm{CFM\ SA}] = 69{,}700\ /\ [4840 \times 12{,}000] = 0.0012\ \mathrm{lb\ water/lb\ dry\ air}$$

Subtracting ΔW1 from the room humidity ratio gives a peak supply air humidity ratio of 0.0081 lb/lb, corresponding to a supply air dew point of 51.4°F.

Part load: part-load supply air volume is 50% of peak (6,000 cfm vs. 12,000 cfm). To deliver the same 69.7 mbh of room latent cooling from half the air volume, the supply air needs twice the latent cooling effect per unit volume — so the humidity ratio differential must double: ΔW2 = 2 × 0.0012 = 0.0024 lb/lb. That puts the part-load supply air humidity ratio at 0.0069 lb/lb, corresponding to a 47.2°F dew point.

Note that although airflow at this part-load point is 50% of peak, the sensible load it corresponds to is only 37.5% of peak (97.7 ÷ 260.4 mbh) — because the supply-air-to-room temperature differential itself widened from 20°F at peak to 15°F at part load; see the reset schedule below.

5. Reset Schedule

% Air Volume	Supply Air Temp (°F)	Supply Air Dew Point (°F)	Room Sensible Cooling	Room Latent Cooling
100	55	51.4	100%	100%
70	60	49.7	52.5%	100%
50	60	47.2	37.5%	100%
Below 50	60	47.2	Varies with airflow	100%

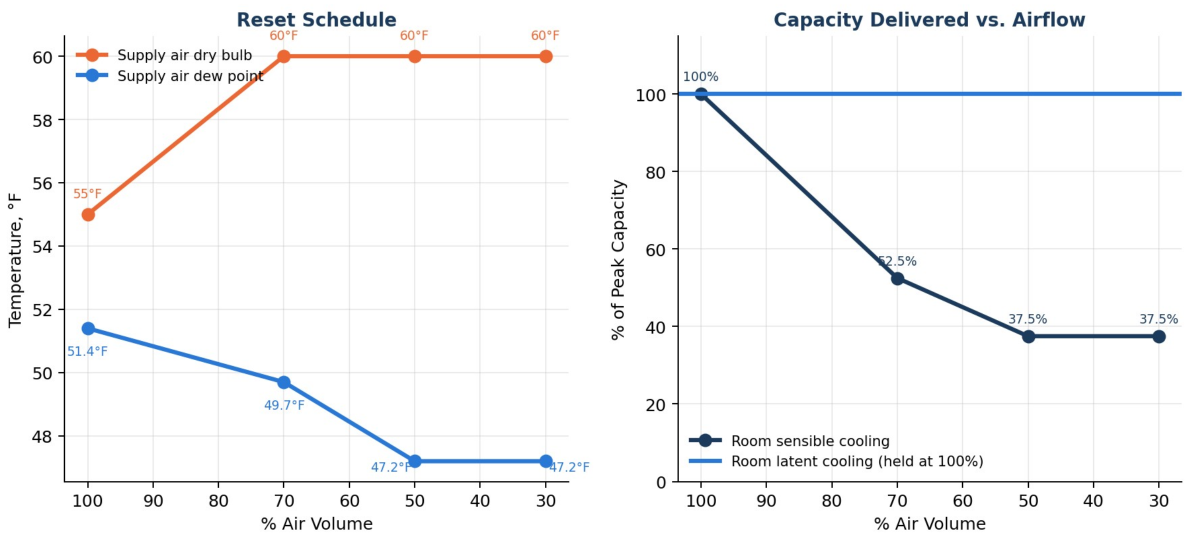

Figure 1 — Reset schedule: supply air dry bulb resets up and dew point resets down as airflow falls, holding room latent cooling constant while sensible cooling tracks airflow.

- The leaving chilled-water-coil air temperature is used in place of supply air dew point for the control system, since dry-bulb temperature sensors are more economical and more reliable in service than dew point sensors.
- The reset uses 70% air volume as the point where the dry-bulb temperature reset completes (transitioning from the 55°F peak-load supply temperature to the 60°F value held from 70% down through minimum turndown). This breakpoint is judgment-based rather than derived from a hard constraint — 70% has worked well across a range of applications in practice.
- Supply air dew point continues to fall as airflow decreases below 70%, all the way to the minimum turndown point, so that dehumidification capacity per unit airflow keeps rising to offset the shrinking air volume.
- Supply air dry-bulb temperature is held flat at 60°F below the 70% breakpoint; further increases in delivered air volume to meet reduced sensible load come from the widening room-to-supply-air ΔT design basis at part load, not from further dry-bulb reset.

6. Piping Configuration

Air passes through three coils in series, sharing a single closed loop with a variable-speed circulator pump: precool coil → driver (dehumidification) coil → reheat coil → supply fan → supply air.

TRICOIL® Piping Diagram — Variable Air Volume Application

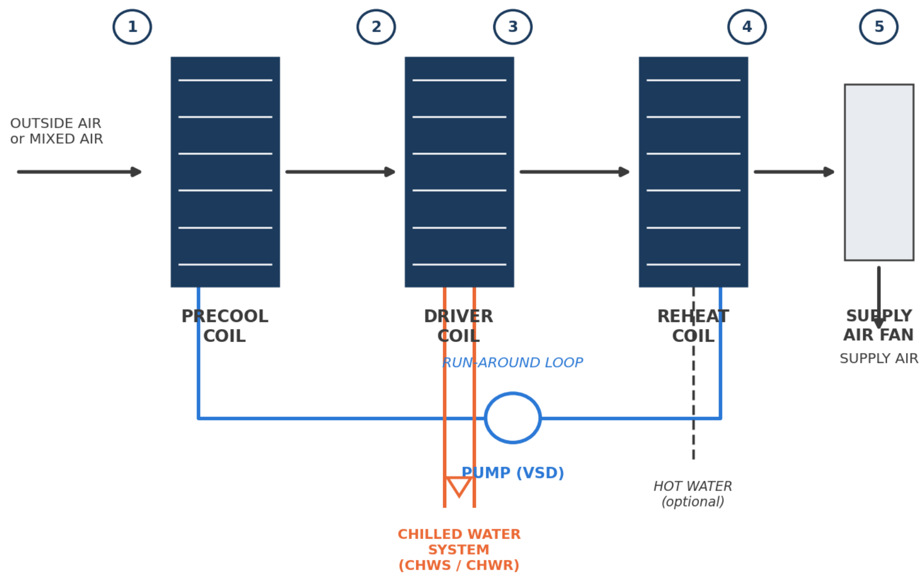

Figure 2 — TRICOIL® piping diagram: the precool and reheat coils share one closed run-around loop; the driver (dehumidification) coil carries its own dedicated chilled-water loop.

The precool coil and reheat coil are connected on the same closed run-around loop — heat picked up cooling the entering outdoor/mixed air at the precool coil is carried by that loop to the reheat coil, where it reheats the cold, near-saturated air leaving the dehumidification coil back up toward supply temperature. The precool coil is sized to supply 100% of the reheat duty across the full reset range, so no external reheat energy source is required. A hot water valve is provided for heating duty only — used if design conditions require heating, or if minimum outside air intake produces a low mixed-air temperature during winter operation.

Control points, referenced to the numbered state points in Section 7's performance table:

- **Point (5), supply air temperature —** the circulator pump speed modulates to hold the point (5) temperature to the reset schedule (Section 5). Because the reheat coil's dry-bulb temperature rise always equals the precool coil's dry-bulb temperature drop (the two coils share one loop and one heat quantity), this single pump-speed control handles both sides of the loop simultaneously.
- **Point (3), leaving chilled-water-coil air temperature —** the chilled water valve modulates to hold the point (3) temperature to the reset schedule, using dry-bulb temperature as the economical surrogate for supply air dew point discussed in Section 5.

7. Performance Summary — Peak Load vs. Part Load

Air-side conditions at each of the five state points shown in Section 6 (1 = entering outdoor/mixed air, 2 = leaving precool coil, 3 = leaving dehumidification coil, 4 = leaving reheat coil, 5 = supply air after fan):

Peak Load

Point	1	2	3	4	5
Dry bulb, °F	80	78	51.4	53.4	55
Wet bulb, °F	67	66.4	51.4	52.3	52.9
Humidity ratio, lb/lb	0.0113	0.0113	0.0081	0.0081	0.0081

Part Load

Point	1	2	3	4	5
Dry bulb, °F	80	68.8	47.2	58.4	60
Wet bulb, °F	67	63.3	47.2	53.4	52.8
Humidity ratio, lb/lb	0.0113	0.0113	0.0069	0.0069	0.0069

Consistency Checks

- **Reheat rise equals precool drop, at both loads.** Peak: reheat coil rises 53.4 − 51.4 = 2.0°F, precool coil drops 80 − 78 = 2.0°F. Part load: reheat coil rises 58.4 − 47.2 = 11.2°F, precool coil drops 80 − 68.8 = 11.2°F. This confirms the single run-around loop correctly balances reheat duty against precool duty at both operating points.
- **Latent cooling capacity is constant across the airflow range.** At 100% airflow: 12,000 cfm × 4840 × 0.0012 = 69,700 Btu/hr (69.7 mbh). At 50% airflow (37.5% of peak sensible load): 6,000 cfm × 4840 × 0.0024 = 69,700 Btu/hr — identical. The reset schedule holds the room's full latent load across the entire turndown range, which is the entire point of the method.

8. Coil Selection — Classroom Study

The coil selections below were run through Coil Company's EZ-Coil selection software (v5.5.0.0) for an actual classroom application of this TRICOIL® VAV method ("Classroom Study" project, September 2026). These are independent, vendor-generated selections against real design conditions — distinct from the illustrative round-number example worked in Section 4 — but they use the same 12,000 cfm peak / 6,000 cfm part-load airflow split and the same three-coil TRICOIL configuration (precool → dehumidification "driver" coil → reheat).

Coil	CFM	Rows /FPI	Circuiti ng	Ent. Air DB/WB	Lvg. Air DB/WB	Total/Sens. (MBH)	Ent./Lvg. Fluid (°F)	GPM
Dehumid. (Driver) — Peak	12,000	6/10	9F/22P	76.4/65.3	51.7/51.1 (rq'd 53.5/53.4)	501.4 / 325.1	43.0 / 55.0	83.36
Dehumid. (Driver) — Part Load	6,000	6/10	9F/22P	72.4/66.2	49.6/49.6	292.9 / 150.7	43.0 / 57.9	38.50
Precool — Part Load	6,000	2/10	2F/16P	83.9/69.8	72.4/66.2 (rq'd 77.0/70.0)	76.3 / 76.3	57.1 / 75.8	8.00
Reheat — Part Load	6,000	2/10	2F/16P	48.4 DB	60.1 DB (rq'd 60.0)	76.2 / 76.2	75.8 / 57.1	8.00

Model numbers (all Coil Company EZ-Coil, 5/8" tube, 1.5"×1.299" sinewave fin, 16 Ga galvanized casing): dehumidification (peak and part load — same physical coil) WC58E-F10-07202400-SCA-R; precool WC58E-B10-07202400-SCA-R; reheat HC58E-B10-07202400-SCA-R.

The dehumidification coil is one physical coil, sized for peak load (12,000 cfm) and operated at reduced airflow and water flow at part load — so it carries the same model, rows/FPI, and circuiting (9 Feeds/22 Pass) at both operating points, as shown above. Its part-load figures are a rating of that coil at the reduced condition rather than an independent selection, so there is no separate "required" leaving condition for that row — 49.6/49.6°F is simply what the coil delivers at 6,000 cfm, 38.50 GPM, and 72.4°F/66.2°F entering air.

Why the dehumidification coil needed a data point at both airflows, while precool and reheat were selected only at part load: the dehumidification coil's duty differs materially between peak (501 MBH) and part load (293 MBH), so both operating points needed independent verification. The precool and reheat coils see their largest duty and largest ΔT at the part-load condition instead — consistent with Section 7's illustrative example, where peak-load reheat/precool duty was only 2.0°F versus 11.2°F at part load — so the 6,000 cfm part-load selection is the governing case for sizing those two coils.

Air-side chain, precool → dehumidification: the precool coil's actual leaving air (72.4°F DB / 66.2°F WB) matches the dehumidification coil's entering air (72.4°F DB / 66.2°F WB) exactly — confirming the two coils are correctly chained in series at part load.

Air-side chain, dehumidification → reheat: the dehumidification coil's part-load leaving air (49.6°F DB) runs about 1.2°F above the reheat coil's modeled entering air (48.4°F DB). A follow-up reheat coil selection against a 49.6°F entering condition would close this chain exactly; noting it here rather than adjusting the reheat numbers without new vendor data.

Run-around loop closure, confirmed against the actual selections: the precool coil rejects 76.3 MBH to the loop; the reheat coil absorbs 76.2 MBH from it — a 0.13% difference. Loop water chains exactly in both directions (precool leaving fluid 75.8°F = reheat entering fluid 75.8°F; reheat leaving fluid 57.1°F =

precool entering fluid 57.1°F), at an identical 8.00 GPM — confirming the two coils are correctly modeled on one common closed loop.

Precool drop vs. reheat rise, this project's actual conditions: precool drops 83.9°F → 72.4°F (11.5°F) while reheat rises 48.4°F → 60.1°F (11.7°F) — matching within 0.2°F, and consistent with the same design principle demonstrated in Section 7's illustrative example. Full independent energy-balance checks (water- and air-side) are documented in the companion coil-selection verification memo.

9. Summary

Conventional VAV reheat throttles supply air volume to track sensible load and, in doing so, throttles dehumidification capacity along with it — a real liability in latent-load-sensitive spaces like classrooms and auditoriums. The TRICOIL® recuperative run-around loop breaks that coupling: by resetting supply air dew point downward and supply air dry-bulb temperature upward as airflow drops, it holds room latent cooling capacity flat across the full turndown range while still tracking sensible load with air volume in the usual VAV manner — all from a single closed loop and one variable-speed circulator pump, with no separate reheat energy source required. The worked example in Section 4 shows this holding exactly: 69.7 mbh of latent capacity delivered at both 100% and 50% airflow, verified independently in Section 10 below, and confirmed against real, vendor-selected coils for an actual classroom application in Section 8.

10. Note on This Revision

This paper is a modernized presentation of a technical bulletin originally issued by Sensible Equipment Company, Inc. in 2003. The method, the given design conditions, and the final results are unchanged from the original.

One correction was made: the original bulletin's "Part Load Supply Air Volume" equation displayed a leftover (75–55) temperature substitution copied from the peak-load line above it, rather than the (75−60) that its own stated 15°F part-load differential requires. The final answer (6,000 cfm) was already correct in the original — only the printed intermediate arithmetic has been corrected here.

Every other value in the worked example, reset schedule, and performance table was independently re-verified using precise ASHRAE psychrometric correlations and matches the original to within normal chart-reading precision (typically ±0.2–0.3°F on dew points read from a hand-drawn psychrometric chart).