

Air Handling Unit

Selection & Troubleshooting Handbook

Practical Submittal Review, Controls and Performance Checks
for HVAC & MEP Engineers

SPECIFY • REVIEW • COMPARE • VERIFY • DIAGNOSE

SI units with labelled IP conversions

Hot-and-humid climate baseline, with guidance for other climates

Worked examples • Troubleshooting cases • Submittal checklists • RFIs

First edition — 2026

Contents

How to Use This Handbook.....	V
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PART A FOUNDATIONS AND REQUIREMENTS

1	AHU Types, Applications and System Arrangements.....	1
1.1	Scope and definitions.....	1
1.2	Classifying AHUs.....	2
1.3	Functional sections.....	3
1.4	Blow-through and draw-through arrangements.....	4
1.5	Fan and damper arrangements for building pressure.....	8
1.6	System arrangements.....	9
1.7	Application guide.....	10
1.8	Hot-humid design drivers.....	11
1.9	Selecting an arrangement.....	11
2	Converting Project Requirements into an AHU Specification.....	14
2.1	The hierarchy of requirements.....	14
2.2	Inputs the specification needs.....	15
2.3	Performance versus prescriptive specification.....	16
2.4	The AHU data sheet.....	17
2.5	Stating airflow on a clear basis.....	19
2.6	Stating pressure on a clear basis.....	20
2.7	Ratings, certification and tests.....	21
2.8	Tolerances and acceptance criteria.....	22
2.9	Specifying for comparable tenders.....	24
3	Design Conditions, Ventilation, Diversity and Psychrometric Processes.....	27
3.1	Outdoor design conditions.....	27
3.2	Indoor design conditions.....	28
3.3	Ventilation airflow.....	29
3.4	Diversity — where it applies.....	30
3.5	Psychrometric properties.....	31
3.6	AHU processes.....	33
4	Airflow, Sensible and Latent Loads, and Entering and Leaving Air Conditions. .	38
4.1	From room load to coil load.....	38
4.2	Supply airflow.....	39
4.3	Supply humidity.....	40
4.4	Coil entering and leaving states.....	40
4.5	Leaving conditions for DOAS units.....	42
4.6	Part-load behaviour.....	43

PART B COMPONENTS AND CONTROLS

5	Cooling-Coil Selection.....	47
5.1	What a coil selection must deliver.....	47

CHAPTER 1

AHU Types, Applications and System Arrangements

Before any coil, fan or filter is selected, the arrangement of the unit and the system it serves has already decided most of its part-load behaviour. This chapter sets out the choices and their consequences.

🕒 LEARNING OBJECTIVES

- Define what is, and is not, an air handling unit for specification and rating purposes.
- Classify AHUs by construction, location, airflow path, service and cooling source.
- Identify each functional section of an AHU and the review questions it raises.
- Quantify the consequences of blow-through versus draw-through fan placement.
- Choose fan and damper arrangements that control building pressure.
- Match AHU and system arrangements to common building types, with hot-humid climates as the baseline.

1.1 Scope and definitions

An **air handling unit (AHU)** is a factory-made, encased assembly of one or more fans and the components needed to perform some combination of circulating, filtering, heating, cooling, dehumidifying, humidifying, mixing and recovering energy from air, delivered to and from the building through ductwork. The definition matters because it decides which rating standards, certification programmes and test methods apply, and therefore what a supplier can be required to prove.

Two families of rating standards dominate international projects:

- **AHRI 430 (I-P) / AHRI 431 (SI), 2020 editions** rate the *supply fan* of a central-station AHU — fan speed, fan electrical power and shaft power — with the fan tested inside its cabinet. Their scope covers encased assemblies without an integral source of mechanical cooling and capable of more than 250 Pa [1.0 in. w.g.] external static pressure. Their exclusions include DX dedicated outdoor air system units rated under AHRI 920. Casing mechanical performance is rated separately under AHRI 1350 (I-P) / 1351 (SI).
- **EN 1886** (mechanical performance: casing strength, leakage, filter bypass, thermal transmittance and thermal bridging) and **EN 13053:2019** (rating and performance of whole units and their components) form the European framework, and underpin Eurovent certification. EN 13053:2019 applies to units with a design airflow above 250 m³/h. EN 1886:2025, published in July 2025, supersedes EN 1886:2007; projects and certification schemes may still reference the 2007 edition during transition, so confirm which edition the specification invokes.

Neither framework rates the complete thermal performance of a custom AHU as a single certified number. Coil capacity, heat recovery effectiveness, acoustic data and filter performance each come from

$$W = [(2501 - 2.326 t^*) W_s^* - 1.006 (t - t^*)] / (2501 + 1.86 t - 4.186 t^*) \quad (3.9)$$

where

t^* — thermodynamic wet-bulb temperature, °C

W_s^* — saturation humidity ratio at t^* , from Eqs. (3.4) and (3.5)

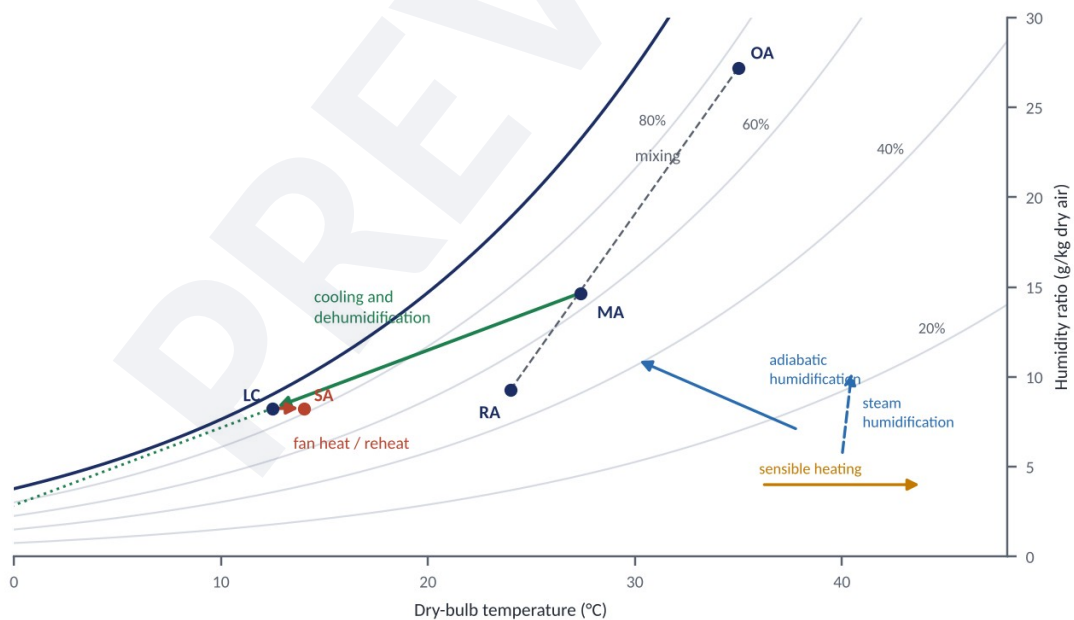
Equation (3.9) is solved for t^* by iteration. A psychrometer reading approximates t^* closely only when adequately ventilated and shielded from radiation; for field work, a calibrated dew-point or capacitive RH sensor is usually more reliable (Ch. 14).

! ENGINEERING CAUTION | Charts versus equations

Psychrometric charts are excellent for seeing a process and poor for auditing one. Reading a chart to 0.5 kJ/kg on a coil with a 30 kJ/kg enthalpy change is a 1.7% error before any other uncertainty. For submittal review and acceptance calculations, compute states from the equations or validated software, and use the chart to check that the answer is sensible.

3.6 AHU processes

Every AHU process moves the air state in a characteristic direction. Figure 3.1 shows them on a chart; Table 3.3 summarises what each changes and which equation governs it.



Point positions illustrative (Gulf dehumidification design point, 30% outdoor air, $p = 101.325$ kPa).

Figure 3.1 AHU processes on the psychrometric chart

⚠ ENGINEERING CAUTION | District-cooling requirements are contractual

District-cooling operators set supply temperature, minimum return temperature or ΔT , maximum flow and pressure-drop limits at the energy transfer station, and sometimes specific requirements for terminal coils and control valves. These are conditions of connection. Obtain the operator's current requirements before coils are selected, include them on the data sheet, and require the coil selection to be run at the operator's supply temperature and ΔT [MANDATORY where they apply].

Σ WORKED EXAMPLE | WE-5.1 — Face area, water flow and tube velocity [ILLUSTRATIVE]

Given. The coil of WE-4.1: $4.684 \text{ m}^3/\text{s}$ of standard air ($\dot{m} = 5.621 \text{ kg/s}$), entering 26.3°C with $W = 0.01288 \text{ kg/kg}$, total load 145.3 kW . Specification limit 2.5 m/s face velocity at standard air. District-cooling supply 4.5°C , return 13.5°C (ΔT 9.0 K , illustrative). Candidate coil: finned height 1.22 m , finned length 1.60 m , 24 circuits, tube ID 14.4 mm .

Find. (a) Face velocity on standard and actual air. (b) Water flow, compared with a 5.5 K ΔT plant. (c) Tube velocity and Reynolds number at design and at 30% flow.

(a) Face velocity (Eq. 5.1): $A_{\text{face}} = 1.22 \times 1.60 = 1.952 \text{ m}^2$. Standard air: $4.684 / 1.952 = \mathbf{2.40 \text{ m/s}}$ [472 fpm] ✓. Actual entering density (Eq. 2.1) $= 1.170 \text{ kg/m}^3 \rightarrow \dot{V} = 5.621 / 1.170 = 4.804 \text{ m}^3/\text{s} \rightarrow \mathbf{2.46 \text{ m/s}}$. Both within the limit, with little margin on actual air.

(b) Water flow (Eq. 5.3): $\dot{m}_w = 145.3 / (4.19 \times 9.0) = \mathbf{3.85 \text{ kg/s} \approx 3.85 \text{ L/s}}$. At ΔT 5.5 K it would be $145.3 / (4.19 \times 5.5) = 6.31 \text{ L/s}$ — 64% more water and, at equal pipe sizes, about 2.7 times the pipe friction (flow ratio squared).

(c) Tube velocity (Eq. 5.2): tube area $= \pi/4 \times 0.0144^2 = 1.629 \times 10^{-4} \text{ m}^2$. $v_w = 0.00385 / (24 \times 1.629 \times 10^{-4}) = \mathbf{0.99 \text{ m/s}}$; $Re = 0.99 \times 0.0144 / 1.38 \times 10^{-6} \approx \mathbf{10\,300}$ (turbulent). At 30% flow: $v_w \approx 0.30 \text{ m/s}$, $Re \approx \mathbf{3\,100}$ — transitional.

Check. Re scales linearly with flow: $10\,286 \times 0.30 = 3\,086$ ✓. Water flow cross-check: $3.853 \times 4.19 \times 9.0 = 145.3 \text{ kW}$ ✓.

Interpretation. The face area meets the limit on both air bases. The high- ΔT selection uses much less water, but its circuiting leaves the coil in transitional flow at low part load. Ask the supplier for the coil's performance at 30–50% flow, or for fewer, longer circuits that keep Re higher, accepting a higher water ΔP . This is a selection to discuss, not to reject.

5.5 Auditing a coil selection

A coil selection sheet makes a set of claims that must be mutually consistent. Before accepting one, check: (1) the air-side energy balance, $\dot{m}(h_{\text{in}} - h_{\text{out}})$, against the stated total; (2) the water-side balance, $\dot{m}_w c_{p,w} \Delta t_w$, against the same total; (3) the sensible capacity against $\dot{m} c_p \Delta t$; (4) the leaving state against the required leaving dew point; (5) the leaving state's plausibility — RH below 100%, an ADP above entering water temperature, a bypass factor consistent with the number of rows. Energy balances should agree within

✕ COMMON MISTAKE | Common coil-selection mistakes

- Specifying capacity in kW without leaving dew point; accepting a coil that meets kW but not the latent duty (WE-5.2).
- Checking face velocity on standard air only, for a coil handling hot outdoor air.
- Selecting at the wrong water ΔT , or ignoring district-cooling requirements.
- Parallel-flow piping of a coil selected for counterflow.
- Deep, dense coils in dusty sites with no access for cleaning; ignoring the ventilation standard's cleanability limit.
- Accepting dry-coil air ΔP for fan selection.
- Coated coils selected on uncoated performance data.
- No eliminator or stacking on a tall, high-velocity dehumidifying coil.

✓ REVIEW CHECKLIST | Chapter 5 — Coil submittal review

- ☐ Entering conditions match the governing design point; airflow basis stated.
- ☐ Leaving DB and dew point (or W) meet the required leaving state.
- ☐ Air-side, water-side and sensible balances agree within about 2–3%.
- ☐ Face velocity within limit on the stated basis; carryover limit confirmed for the fin and coating.
- ☐ Water ΔT , supply temperature and flow match plant or district-cooling requirements.
- ☐ Tube velocity in the recommended range; part-load flow regime acceptable.
- ☐ Wet-coil air ΔP stated and used for fan selection; dry-coil ΔP checked against the ventilation standard.
- ☐ Water ΔP compatible with valve sizing and authority (or PICV basis).
- ☐ Rows, fin density, materials, coating system and corrosion test basis as specified; coated performance shown.
- ☐ Counterflow connections, vents and drains confirmed on drawings.
- ☐ Stacking, intermediate drain pans and eliminator provided where needed.
- ☐ Withdrawal space and access for cleaning confirmed.

References

Chapter 5 references

Ref.	Document	Edition note
5.1	AHRI 410-2023, Forced-Circulation Air-Cooling and Air-Heating Coils. AHRI.	Current edition at publication (Oct 2026)
5.2	2024 ASHRAE Handbook — HVAC Systems and Equipment (coils). ASHRAE.	Current edition at publication (Oct 2026)

7.3 Fan curves and the operating range

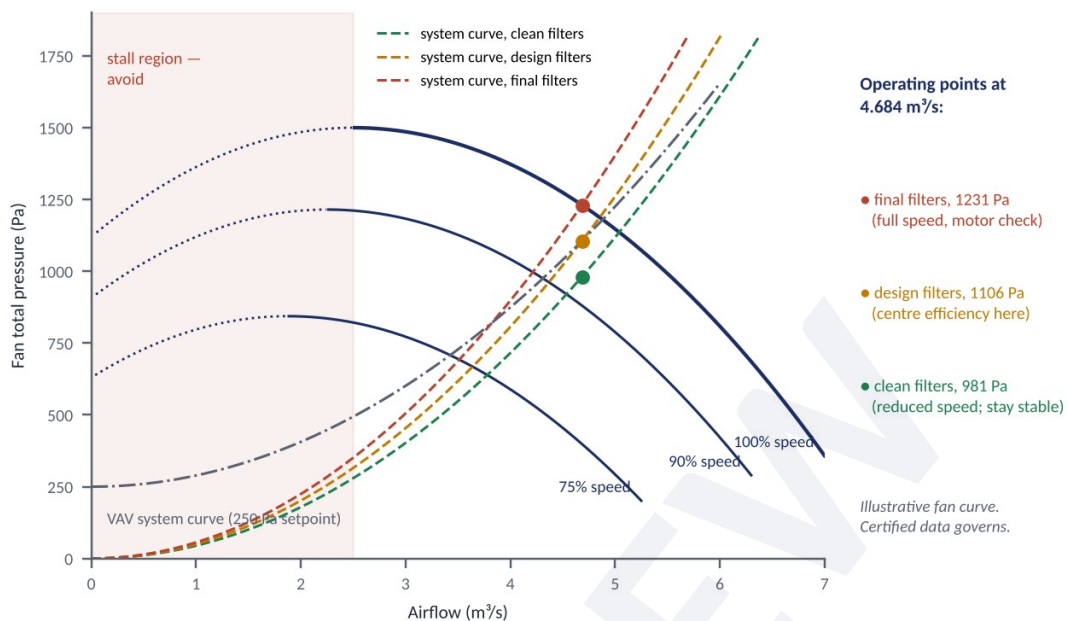


Figure 7.1 Fan curve, system curves and operating range

- **Stay right of the stall region** at every operating point, including minimum VAV airflow at maximum static-pressure setpoint. Operation near or left of the peak of the pressure curve causes pulsation, noise and vibration (TC-6).
- **Centre the efficiency** around the design state, not the final or clean state.
- **Check the motor at the worst point** — usually final filters at full speed — and, for forward-curved fans, at the highest airflow the system can produce, because their power rises with airflow.
- **Size for real, not nominal, wheel diameters** where arrays use several small fans: aggregate data must come from tests or validated software for the array as built.

7.4 Fan laws and their limits

For a given fan operating at geometrically similar points (the same position relative to its curve):

$$\dot{V} \propto N \quad p \propto \rho N^2 \quad P \propto \rho N^3 \quad (7.3)$$

where

- $\dot{V}$ — airflow, m³/s
- N — rotational speed, rev/s or r/min
- p — fan pressure (static or total, consistently), Pa
- P — air or shaft power, W
- ρ — air density, kg/m³

The laws apply when the system curve passes through the origin, so that pressure varies with the square of airflow. They do **not** apply directly when part of the system pressure is fixed — a VAV duct static-pressure

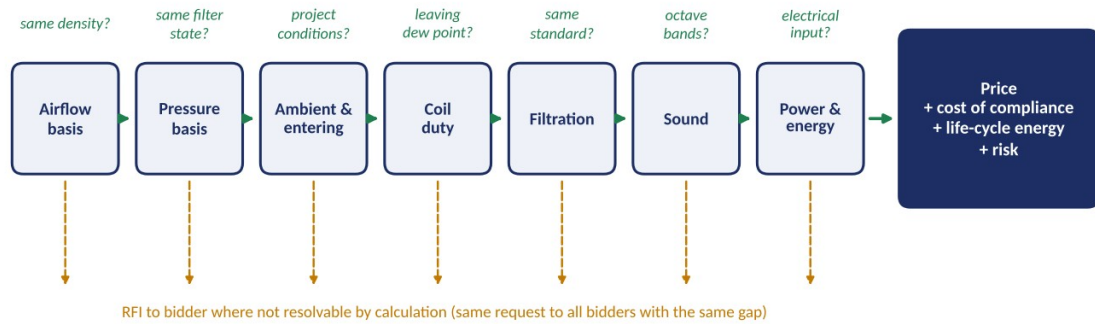


Figure 13.1 Offer normalisation workflow

Σ WORKED EXAMPLE | WE-13.1 — Normalising three offers for the same AHU [ILLUSTRATIVE]

Given. Three synthetic offers (X, Y, Z) for the AHU of WE-4.1 and WE-7.1. Specification basis: 4.684 m³/s standard air; ESP 450 Pa including a 50 Pa external attenuator; fan to deliver design airflow at final filters; coil 145 kW with leaving dew point ≤ 11.2 °C at project entering conditions; face velocity ≤ 2.5 m/s; ISO 16890 filters (ISO ePM10 50% + ISO ePM1 60%); octave-band sound power; electrical input at design filters to be stated. Energy evaluation: 6000 h/year at 0.30 currency units/kWh over 10 years at 6% discount rate (present-value factor 7.36). All prices are in the same currency units.

Table 13.1 Offers as received

Attribute	Offer X	Offer Y	Offer Z
Price as offered	100 000	88 000	96 000
Airflow as offered	4.684 m ³ /s std ✓	16 000 m ³ /h actual at fan inlet = 4.444 m ³ /s actual ≈ 4.53 m ³ /s std (−3.3%) ✗	4.684 m ³ /s std ✓
ESP basis	450 Pa incl. attenuator ✓	400 Pa, attenuator excluded ✗	450 Pa incl. attenuator ✓
Fan selected at	Design filters; final shown ✓	Clean filters ✗	Final filters ✓
Coil rating basis	Project conditions; DP 11.1 °C ✓	Standard rating conditions, not project ✗	Project conditions; DP 11.2 °C ✓
Coil face velocity	2.40 m/s ✓	2.45 m/s ✓	2.90 m/s, no eliminator ✗
Filters	ISO ePM10 50% / ePM1 60% ✓	G4 / F7 (EN 779) ✗	ISO ePM10 50% / ePM1 50% ✗ (lower final class)
Sound data	Octave bands, all paths ✓	82 dB(A) only ✗	Octave bands, all paths ✓

 TROUBLESHOOTING CASE | TC-2 — Poor cooling despite "design" chilled-water flow [HYPOTHETICAL]

Symptom. Supply air at 15.5 °C against 12.5 °C setpoint on hot afternoons; cooling valve 100% open. The plant operator reports design flow to the AHU.

Measurements. CHW entering 7.2 °C (design 4.5 °C); leaving 10.4 °C (ΔT 3.2 K against 9 K design). Clamp-on flow 3.9 L/s. Coil surface: lower rows cold, upper headers noticeably warmer.

Hypotheses. (a) High CHW supply temperature from the energy-transfer station; (b) air trapped in the coil; (c) parallel-flow piping.

Verification. District-cooling ETS secondary supply setpoint found at 7.0 °C after a controller replacement. Coil vent at the top header releases air for several seconds. Piping inspected: counterflow, correct.

Root cause. Raised secondary supply temperature (loss of capacity and dehumidification) compounded by air in the upper circuits, which also explains the very low water ΔT .

Corrective options. Restore ETS setpoint per the operator's connection requirements; vent the coil and add an automatic air vent at the high point; check system air separation. Re-verify capacity at measured conditions (Ch. 14).

Lesson. "Design flow" is not "design capacity". Low water ΔT with a fully open valve points to the water side — supply temperature, air in coils, or bypassing.

 TROUBLESHOOTING CASE | TC-3 — Space temperature met but RH 65–70% [HYPOTHETICAL]

Symptom. Office meets 24 °C but occupants complain of clamminess; space RH sensors read 65–70% on mild, cloudy days, 50–55% on hot days.

Measurements. Constant-volume AHU. On mild days, leaving-coil temperature trends at 17–18 °C with the valve modulating at 35–45%; outdoor air at design fraction.

Hypotheses. (a) Part-load dew-point loss in CAV; (b) excess outdoor air; (c) infiltration.

Verification. OA airflow measured at design. Building pressure slightly positive. Trends match WE-4.3(a): sensible load halves, latent load stays, coil warms, space RH rises to about 67–70%.

Root cause. Capacity control by valve modulation on space temperature in a constant-volume system.

Corrective options. Short term: leaving-coil temperature control at about 12.5 °C with reheat enabled in dehumidification mode, using recovered heat if available. Medium term: convert to single-zone VAV with minimum airflow in dehumidification mode, or add heat-pipe reheat (WE-6.1).

Lesson. In humid climates, a unit can meet temperature and fail humidity. Test humidity control at part load (Ch. 14).

END OF PREVIEW

The full handbook continues

You have seen 9 of 182 pages. The complete edition covers the whole AHU workflow: specify, review, compare, verify and diagnose.

16

chapters

32

worked examples

11

troubleshooting cases

16

RFI templates

24

technical figures

115

cited references

CHAPTERS

- | | |
|---|---|
| 1 AHU Types and System Arrangements | 9 Casing: Leakage, Insulation and Hygiene |
| 2 Requirements to AHU Specification | 10 Condensate Drainage |
| 3 Design Conditions and Psychrometrics | 11 Sound, Vibration and Attenuation |
| 4 Airflow, Loads and Coil Conditions | 12 Controls, Interlocks and Humidity Control |
| 5 Cooling-Coil Selection | 13 Submittal Review and Offer Comparison |
| 6 Heating, Reheat and Heat Recovery | 14 Installation, Commissioning and Handover |
| 7 Fan Selection | 15 Troubleshooting |
| 8 Filtration | 16 Maintenance, Degradation and Retrofit |

APPENDICES

- | | |
|---|--|
| A Checklists and data-sheet template | E Conversion reference |
| B RFI library | F Psychrometric equations and property data |
| C Glossary | G Bibliography |
| D Symbols and subscripts | H Terminology and assumptions register |

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SI units with IP conversions · hot-humid (GCC) baseline with guidance for other climates