Refrigeration component sizing manual

CALCULATION METHODS AND WORKED EXAMPLES

Software and manual revision 2.6.1
6 October 2026

Commercial refrigeration, air conditioning and cold rooms

This revised manual combines the existing load, cycle, compressor, TXV and selection chapters with expanded instructions for coil geometry, fan selection, refrigerant inventory and pressure-drop calculations.

For the evaporator and condenser, it explains inputs, formulas, unit conversions, numerical examples and how calculated losses feed the fan and coil summaries.

Engineering scope

Manufacturer methods remain available. The advanced refrigerant option solves local pressure and properties through prescribed enthalpy regions; it does not solve a coupled air/refrigerant heat-transfer distribution or update the full refrigeration cycle with those losses. Air-surface coefficient data must match the actual fin pattern and velocity basis.

Worked assumed values teach arithmetic. The R134a pressure-profile example is actual output from the shipped numerical engine. Neither illustrative values nor numerical convergence replace validation against supplier or laboratory data.

Revision	Changes
2.6.1	Expanded manual; worked calculations and workflow examples.
2.6.0	Calculated air/refrigerant losses; advanced local-state iteration, Reynolds-dependent coefficients and mesh check.
2.5 series	Coil geometry, face area, separate summaries, coil volume and ten fan-curve points.

Contents and calculation workflow

Pages	Calculation group
3-6	Compressor class, evaporating/condensing temperature and COP guidance
7-11	Cold-room loads, product loads and compressor power
12-16	Thermal sizing, refrigerant properties, mass flow and tube screening
17-20	Manufacturer-data coil/fan workflow, dry UA capacity and pressure allowances
21-27	TXV selection, cold-room example, source register and condenser reference temperatures
28-32	Geometry, surface UA, face area, coil volume and fan operating point
33-34	Calculated air pressure drop and Reynolds-dependent coefficients
35-38	Darcy, homogeneous, Lockhart-Martinelli and void/acceleration calculations
39-40	Advanced local-state solver and complete R134a worked example
41-42	Mesh checks, integration, project log and updated references

The core sizing chapters on pages 3-27 are retained. Their manufacturer-input pressure-loss descriptions are one selectable method; pages 33-41 explain the new calculated alternatives. The reference zero-loss cycle remains separate from calculated coil losses.

Suggested sequence

1. Establish design load and operating temperatures; solve the refrigerant reference cycle.
2. Enter both coil geometries and choose thermal and pressure-loss methods.
3. Load or verify mass flow, properties, pressure/enthalpy boundaries and validated air coefficients.
4. Calculate coil losses, enter fan points 1-10 and solve each fan operating point.
5. Review velocity, thermal duty, saturation-loss allowance, mesh sensitivity and supplier limits; save a comparison run.

Compressor application classes

Secop identifies typical evaporating ranges by back-pressure application. These categories overlap; they do not define a universal operating envelope. Exact limits depend on compressor model, refrigerant, condensing condition, return gas and motor cooling [S4].

Class	Typical evaporating range	Common application
LBP	−45 to −10 °C	Freezers and refrigeration with freezer sections
MBP	−20 to 0 °C	Chilled cabinets and medium-temperature cooling
HBP	−5 to +15 °C	Higher evaporating-temperature cooling applications

Example: a −18°C freezer with 7 K entering-air TD needs approximately −25°C evaporation, within the typical LBP guide. A 2°C cold room with 6 K TD gives −4°C, within the overlapping MBP and HBP guides. Compressor selection must still follow its map and envelope.

What temperature is being controlled

Cooled-space target or evaporator entering air is separate from outdoor condenser ambient. Evaporator TD uses entering-coil air, which can differ from the room setpoint because of stratification or outdoor-air mixing. Condenser TD uses the actual air entering its coil, including recirculation and installation effects.

Refrigerant blend conventions

Use manufacturer-defined dew, bubble or mean-temperature conventions consistently. A high-glide refrigerant needs a segmented coil treatment. The cycle workspace supplies CoolProp HEOS refrigerant properties. Property-model availability does not authorize the charge or application. The cycle is single-stage and subcritical; CO_2 at normal warm condenser conditions needs a separate transcritical gas-cooler model.

Evaporator temperature guidance

ASHRAE refrigerated-storage guidance identifies approximately 6 K TD for humidity-sensitive unit-cooler selection [S1]. The software applies that value to the exposed-product guide. Packaged and frozen-storage ranges below are engineering screening assumptions, not universal ASHRAE requirements.

Application	TD starting point	Evaporation basis
Exposed chilled products	6 K source-backed starting point	Entering air minus 6 K
Packaged chilled products	6–10 K screening; default 8 K	Entering air minus selected TD
Frozen storage	5–10 K screening; default 7 K	Entering air minus selected TD
Comfort DX air conditioning	5–10°C evaporation screening; default 7°C	Mixed entering air minus target evaporation

Te = Tair,in − TD_e. Lower TD generally requires more coil surface and helps reduce product drying; higher TD can increase moisture removal. Room humidity cannot be predicted from TD alone. Check product requirements, coil rating, fan distribution and moisture balance.

Entering air °C	Illustrative TD K	Evaporating °C
-25	7	-32
-18	7	-25
-5	6	-11
2	6	-4
5	6	-1
10	6	4
27	20	7

The final row is an AC example; the preceding rows are cold-space examples. Verify frost/defrost whenever evaporating below freezing. Te must remain below leaving-air temperature in the simplified dry model.

Condensing temperature guidance

ASHRAE remote air-cooled refrigeration guidance uses generally accepted condenser TD values of 8 K for medium-temperature systems and 5.5 K for low-temperature systems [S1]. These values apply to that remote-refrigeration context, not every packaged unit or AC condenser.

Design entering air °C	Remote medium °C	Remote low °C	AC screening °C
25	33	30.5	37
30	38	35.5	42
35	43	40.5	47
40	48	45.5	52
45	53	50.5	57

$T_c = T_{condenser,in} + TD_c$. The AC example uses 12 K; a 10–15 K AC band is an engineering screening assumption. Supplier ratings and the compressor operating envelope override this guide. Small packaged refrigeration equipment can require a different TD.

Use the site design weather condition and actual condenser intake. Outdoor design dry-bulb is not yearly average weather. ASHRAE load guidance identifies the 0.4% outdoor design basis for refrigeration [S2]. Condenser selection is based on total heat rejection and actual rating conditions [S7].

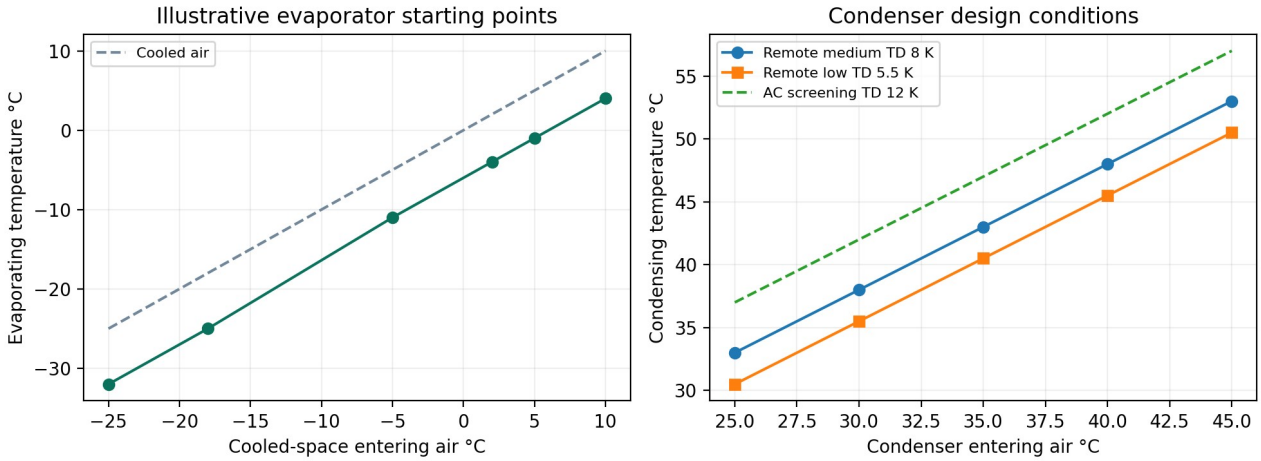


Figure 2 Starting temperature relationships. Frozen-room TD and AC condenser TD curves are engineering assumptions; remote condenser curves use the cited ASHRAE values.

COP guidance and efficiency boundaries

There is no universal recommended COP for an application. COP depends on evaporating/condensing temperatures, refrigerant, model, speed, superheat, subcooling and the power boundary. Manufacturer performance should replace assumed COP [S5].

Application	Screening compressor COP	Basis
Low-temperature refrigeration	0.6–1.8	Engineering range near 40–50°C condensation
Medium-temperature refrigeration	1.2–3.0	Engineering range near 40–50°C condensation
Comfort air conditioning	2.5–4.0	Engineering range near typical cooling design point

These ranges are not ASHRAE prescriptions, legal minima or guarantees. A small hermetic unit can fall below them; a large system or lower temperature lift can exceed them. As one published example, Secop SC10G R134a at 220 V / 50 Hz gives COP 0.86 at −23.3/54.4°C and 1.59 at −6.7/54.4°C; return-gas and liquid temperatures differ between the stated ratings [S5].

COPcompressor = Qe / Pcompressor. Whole-system COP includes fans, pumps, controls and time-averaged defrost, with consistent net cooling. Do not insert a whole-system COP and then count compressor input as though auxiliary power all reaches the refrigerant.

COPCarnot = (Te + 273.15) / (Tc − Te). At Te = −4°C and Tc = 50°C, Carnot COP = 4.984. The optional software estimate is 0.35 × 4.984 = 1.744. The 0.35 factor is an explicit engineering assumption, not a fitted refrigerant model. The Carnot result is an upper bound, not achievable design performance.

Cooling EER in Btu/Wh divided by 3.412 gives cooling COP with the same boundary. Seasonal SEER/IEER must not be substituted for a single design-point COP. Rated compressor input is the preferred software input when available.

Detailed cold room load method

ASHRAE separates transmission, product, internal, infiltration and equipment heat [S2]. The improved model follows these categories, using user-defined schedules and physical inputs. It improves the earlier uniform-envelope estimate but does not implement a full doorway-flow or transient product model.

Envelope and thermal bridges

For each surface i: Qi = Ui × Ai × max(0, Tadjacent,i + solar addition,i − Troom) / 1000, in kW. Four wall areas are LH, WH, LH, WH. Roof and floor areas are LW. Door area is deducted from wall 1 and calculated separately.

Qenvelope = sum(Qi) × (1 + bridge allowance/100) + Udoor × Adoor × max(0,Tdoor − Troom)/1000. Use assembly U-values and actual adjacent temperature. A single bridge percentage is a screening allowance. The floor calculation is a simplified steady boundary; ground-coupled slab analysis may be needed.

U = 1 / (1/hi + sum(thickness/k) + 1/ho). Use aged insulation conductivity, actual panels and thermal bridges. Solar offsets must be justified for exposure and averaging period; building AC requires its own load procedure.

Daily energy and coincident peak

Daily envelope energy = Qenvelope × 24. Scheduled source energy = source kW × operating hours. Eday sums envelope, infiltration, product, packaging, internal heat, defrost and other entered energy.

Qaverage = Eday / refrigeration hours. Qpeak = envelope + peak infiltration + product/packaging energy divided by pull-down hours + scheduled sources assumed coincident + respiration. Base duty = max(Qaverage,Qpeak). Selected duty = base × (1 + design allowance/100). Defrost recovery is included in average energy; review recovery-time peaks separately.

Moist air infiltration calculation

Enter incoming dry-bulb/RH, room dry-bulb/RH, site pressure, total daily air exchanges and peak hourly exchanges. Air exchanges should come from measured traffic or a separate analysis. The program does not infer them from door area or apply the ASHRAE door-opening correlation.

Saturation vapor pressure is evaluated from the ASHRAE Hyland–Wexler equations using absolute temperature. Use ice saturation at and below 0.01°C and liquid-water saturation above it. The calculator restricts these inputs to −60 to +60°C. Source [S3].

pv = RH/100 × pws. W = 0.621945 × pv/(p − pv), kg water/kg dry air. h = 1.006T + W(2501 + 1.86T), kJ/kg dry air. v = 0.287042(T + 273.15)(1 + 1.607858W)/p, m^3/kg dry air, with p in kPa.

Daily incoming dry-air mass = room volume × daily air exchanges / incoming specific volume. Eair = mass × max(0, hin − hroom) / 3600, in kWh/day. Peak load uses room volume × peak hourly air exchanges / incoming v × enthalpy difference / 3600, in kW.

Below-freezing rooms add a screening ice term: mass × max(0,Win − Wroom) × [333.5 + 2.1 × max(0,−Troom)] / 3600. This treats removed moisture as ice cooled to room temperature. Actual frost temperature and defrost heat recovery can differ; it is not a moisture-transport simulation.

The model clips negative infiltration heat to zero. It assumes all net moisture removal loads the refrigeration system; no desiccant or airlock conditioning is modeled. Do not count the same infiltration as a separate known kW load in detailed mode.

Product and internal heat calculations

Product sensible and freezing energy

Without freezing: Eproduct = mass × cp × (Tinitial − Tfinal) / 3600. With freezing: sum sensible energy above the freezing point, effective latent heat for a batch crossing it, and sensible energy below it. Use different specific heats above and below freezing.

For initial temperature above freezing and final temperature below: Eproduct = m × [cp,above(Tinitial − Tfreeze) + Leffective + cp,below(Tfreeze − Tfinal)] / 3600. If initially frozen, only below-freezing sensible cooling applies. A batch initially at its freezing point is treated as initially unfrozen if the freezing option is enabled.

Leffective must reflect product composition and frozen fraction. A fixed latent term is a lumped energy model, not a progressive freezing curve. Required product duty is energy/pull-down hours; verify air exposure, stacking, packaging and core temperature independently.

Packaging and respiration

Epackaging = mpack × cppack × (Tpack,in − Tpack,out) / 3600. Respiration input is total W for the stored inventory, included for 24 hours. Obtain commodity-specific product properties and respiration at the storage temperature; no generic food database is built in.

Schedules and defrost

Epeople = count × total W/person × hours / 1000. Lighting, evaporator fans and other equipment each use W × hours /1000. Use heat entering the room, not merely equipment nameplate power for a motor located outside it.

Edefrost = heater W × total daily defrost hours × heat fraction entering the room /1000. The fraction depends on drainage and method. Do not enter the same defrost energy elsewhere. Refrigeration hours should exclude defrost and downtime; verify daily availability and peak recovery.

Compressor power methods

Compressor duty can mean required cooling capacity or required drive power. Cooling duty Qe is established from the heat load. Electrical input P is calculated or obtained separately. Four input methods are provided.

Method	Equation	Best use
Assumed COP	P = Qe / COP	Early screening with documented assumption
Rated performance	P from supplier at actual operating point	Preferred equipment selection basis
Actual enthalpy rise	Pfluid = mass flow × (h2 − h1)	Verified actual refrigerant states
Isentropic work	Pfluid ≈ mass flow × (h2s − h1) / efficiency	Cycle model with verified compressor efficiency

Mass flow = Qe/(h_evaporator,out − h_liquid). The expansion device is assumed adiabatic with equal upstream/downstream enthalpy. h1 is at compressor suction, not necessarily at evaporator exit. h2 is the actual discharge state. h2s is found at discharge pressure with entropy equal to the suction state.

For actual fluid-state efficiency: eta_is,fluid = (h2s − h1)/(h2 − h1). Some manufacturer isentropic efficiencies use shaft or electrical input instead; preserve that boundary rather than applying an additional efficiency twice. Source [S6].

Steady compressor balance, neglecting kinetic/potential energy: work input = mass flow × (h2 − h1) + heat lost from the chosen compressor boundary. The software neglects casing heat loss. External mechanical and motor losses may require conversion; the software combines applicable losses in eta_conversion, so P_electric = Pfluid/eta_conversion.

For a hermetic compressor, discharge enthalpy may already contain internal motor losses. Use eta_conversion = 1 for that complete electrical/fluid boundary under negligible casing loss, or use manufacturer electrical input. Dividing by motor efficiency again can overstate power.

Worked compressor calculation without COP

Illustrative arithmetic inputs only: Qe = 10 kW, evaporator outlet enthalpy = 400 kJ/kg, liquid enthalpy = 250 kJ/kg, compressor suction enthalpy h1 = 405 kJ/kg. These are not a validated state set for any named refrigerant.

Actual discharge state

1 Mass flow = 10/(400 − 250) = 0.066667 kg/s.
2 With actual discharge h2 = 450 kJ/kg, Pfluid = 0.066667 × (450 − 405) = 3.000 kW.
3 If an external-drive combined conversion efficiency is 0.90, estimated electrical input = 3/0.90 = 3.333 kW.
4 If h2 includes internal hermetic motor heat, use the appropriate complete boundary rather than adding that 0.90 correction again.

Isentropic discharge state

1 Obtain h2s = 435 kJ/kg at discharge pressure and suction entropy.
2 Ideal fluid work = 0.066667 × (435 − 405) = 2.000 kW.
3 With eta_is,fluid = 0.70, actual fluid work = 2/0.70 = 2.857 kW.
4 With external conversion efficiency 0.90, electrical input = 2.857/0.90 = 3.175 kW.
5 Predicted actual discharge enthalpy = 405 + (435 − 405)/0.70 = 447.857 kJ/kg.

The software lets you enter actual h2 for condenser heat rejection separately. Keep it consistent with the isentropic prediction. Line heat gain h1 − h_evaporator,out also reaches the condenser; it is not useful room cooling. Enthalpy-based heat rejection therefore has a different boundary from Qe + electrical input when these effects or external losses are present.

Other checks

Measured real electrical input can be obtained from a power analyzer. For a balanced three-phase load, P ≈ sqrt(3) × Vline × Iline × power factor /1000. Single phase: P ≈ V × I × power factor /1000. These are checks, not implemented input modes; inverter harmonics may require direct measurement. Current alone does not establish compressor power.

Evaporator and condenser sizing

Evaporator sensible duty Qs = Qe × SHR. Volume airflow = Qs/(rho × cp,air × air temperature drop). Face area = volume airflow / selected face velocity. For wet coils verify total capacity from dry-air mass flow × enthalpy change and moisture balance; entered SHR does not predict latent capacity.

Condenser duty can use supplier total heat rejection or mass flow × (h_condenser,in − h_liquid,out). The simple Qc ≈ Qe + P input approximation neglects casing losses and line heat gains. Airflow = Qc/(rho × cp,air × condenser air rise). Each coil uses its own entered density and air heat capacity at actual temperature, humidity and altitude. Condenser air rise must be smaller than its entering-air TD.

For a dry isothermal screening model, LMTD = (difference1 − difference2)/ln(difference1/difference2). Evaporator sensible UA = Qs/LMTD and sensible external area = Qs×1000/(U×LMTD); condenser isothermal UA = Qc/LMTD, with F = 1. U must use the same area basis. These calculations do not replace wet/frosted evaporator or segmented condenser modeling.

Example: 10 kW evaporator, SHR 0.90, rho 1.2 kg/m³, cp 1.006 kJ/(kg K), air drop 3 K gives 2.485 m³/s. At 2 m/s face velocity, face area = 1.243 m². With COP 3, compressor input = 3.333 kW and simple heat rejection = 13.333 kW.

At condenser Te-independent Tc = 50°C and entering air 38°C with 7 K air rise, the condenser LMTD = (12−5)/ln(12/5) = 7.996 K. UA = 13.333/7.996 = 1.668 kW/K. A catalog heat-rejection/TD index = 13.333/12 = 1.111 kW/K; this index is not UA.

Final coil design requires face dimensions, tube OD/wall/pitches, fin spacing, rows and refrigerant circuits. Validate heat transfer, refrigerant distribution, air/refrigerant pressure drop, fan curve, condensate carryover, frost, sound and serviceability. Velocity and mass-flux circuit screening is supplied in the cycle workspace; final tube count and circuit optimization require a validated coil model. Air-side manufacturer curve inputs and dry UA checks are covered below.

Refrigerant properties and current cycle points

The standalone engine embeds CoolProp WebAssembly 7.2.1dev; Python properties use CoolProp 7.2.0 HEOS. The catalogue contains 116 constructible model entries and 31 unavailable blend entries, including alternate representations. These are not 116 unique approved refrigerants. Transport properties required for pressure models may be unavailable even when thermodynamic states can be solved [S8, U5].

UI point	State and calculation
1	Compressor suction / superheated vapor: T1 = dew(Pe) + SH
2s	Ideal compressor discharge: s2s = s1 at Pc; reference point
2	Actual discharge: h2 = h1 + (h2s - h1)/eta_is
3	Condenser saturated vapor at Pc
4	Condenser saturated liquid at Pc; bubble reference
5	Subcooled liquid / valve inlet: T5 = bubble(Pc) - SC
6	Valve outlet: h6 = h5 at Pe; isenthalpic expansion
7	Evaporator saturated vapor at Pe; dew reference

Select evaporating dew temperature and condensing bubble or dew convention. Liquid subcooling always uses bubble temperature at the condensing pressure. The P-H chart uses absolute pressure on a logarithmic axis. Each point shows temperature, pressure, enthalpy, entropy, density and quality. Saturation points 3/4/7 are not additional compressors or valves.

The cycle model brackets quality or temperature to invert enthalpy/entropy at fixed pressure, using up to 70 iterations and a 10^-6 kJ/kg or kJ/(kg K) residual. State convergence is not simultaneous equipment-performance matching. Point 2s is an ideal reference and does not lie on the actual four-process loop. Advanced coil-loss states on pages 39-41 remain a separate calculation.

Worked refrigerant cycle with seven point numbering

R134a example: evaporating dew -4 C, condensing bubble 40 C, 5 K outlet superheat, 3 K subcooling and fluid isentropic efficiency 0.70. Values below are calculated by the current property engine.

Point	T C	P kPa abs	h kJ/kg	s kJ/(kg K)
1	1.000	252.676	400.634	1.74555
2	61.819	1016.593	443.153	1.78433
3	40.000	1016.593	419.429	1.71106
4	40.000	1016.593	256.409	1.19048
5	37.000	1016.593	251.942	1.17614
6	-4.000	252.676	251.942	1.19325
7	-4.000	252.676	396.252	1.72941
2s	49.873	1016.593	430.398	1.74555

```
Refrigerating_effect = h1 - h6 = h1 - h5
Mass_flow = Qe/(h1 - h6)
Fluid_work = mass_flow (h2 - h1)
Condenser_rejection = mass_flow (h2 - h5)
Fluid_COP = (h1 - h6)/(h2 - h1)
```

1. Cooling effect = 148.692173 kJ/kg. For 10 kW duty, mass flow = 0.067253 kg/s = 242.111 kg/h.

2. Refrigerant compression work = 2.859548 kW; fluid COP = 3.497056.

3. Condenser heat rejection = 12.859548 kW. Energy closure Qc = Qe + fluid work holds at this model boundary.

4. For fixed mass flow, SH and SC change cooling effect and therefore Qe. For fixed design duty, SH and SC change inferred mass flow and compressor/condenser duty.

Use the fluid-to-electrical conversion boundary when predicting electrical power. The ideal compressor line and connecting chart segments do not establish an exact process path. Points 3/4/7 mark saturation references; point 5 includes subcooling and point 1 includes selected superheat.

Mass flow options and capacity boundaries

Mass flow follows load and refrigerating effect; it is not a fixed application recommendation. In one steady loop, total mass flow is the same through the components. Parallel circuits divide that flow according to distribution; the pressure screening assumes equal circuits.

Dropdown method	Calculation or input
Off	No independent mass input; infer Qe/qe when a thermodynamic power method requires it.
Cooling effect	mass = Qe/(h_evap_out - h_evap_in); current reference cycle uses h1 - h6.
Manual	Enter kg/s from measured/supplier data and review implied capacity.
Displacement	mass = swept m3/h x volumetric efficiency / suction specific volume / 3600.

1. Displacement example: 15 m3/h x 0.75 / 0.05 m3/kg / 3600 = 0.0625 kg/s. Efficiency is entered supplier data, not predicted.

2. For reference-cycle fixed mass 0.050 kg/s and the R134a cooling effect from page 14, calculate Qe = 0.050 x cooling effect. Compressor fluid work = 0.050 x (h2 - h1).

3. Compare this calculated capacity with the independent design load. A capacity shortfall remains a shortfall even if the mass flow is manually specified.

The cycle capacity-basis selector determines whether duty is fixed or inferred from entered mass flow. The main sizing mass-flow dropdown determines how a separate mass input is calculated. These controls have distinct roles; record the selected basis in comparisons.

Illustrative application	Fluid and Te/Tc C	kg/s at 10 kW
Chilled room	R134a -4/40	0.06725
Frozen storage	R404A -25/40	0.09847
Commercial chilled	R513A.mix -5/40	0.07919
Comfort DX AC	R410A 7/47	0.06428

Application values use 5 K SH, 3 K SC and 70% fluid efficiency. They are calculated examples, not ASHRAE prescribed flows. Replace them with actual model conditions, evaporator boundaries and manufacturer data.

Refrigerant tube and circuit screening

```
Volume_flow = mass_flow / density
Tube_flow_area = mass_flow / (density design_velocity)
Required_ID = sqrt(4 mass_flow / (pi density velocity))
Mass_flux_per_circuit = mass_flow / (circuits pi ID^2/4)
```

Use suction point 1 density for suction screening, discharge point 2 density for discharge screening, and subcooled-liquid point 5 density for liquid-line screening. A saturated point or outlet density is not an average density for an entire coil.

1. Obtain a current refrigerant state and mass flow. Select application/manufacturer velocity guidance and pressure-loss allowances.

2. Calculate line flow area and required ID. Verify actual wall thickness, pressure rating, material and available size with the supplier.

3. At minimum capacity, recompute vapor-line velocity and oil-return suitability. Riser geometry, traps and compressor requirements must be checked.

4. For coil circuit screening choose a target mass flux from a validated model or supplier, then compare entered circuits with the resulting flux. A generic exploratory flux is not a universal design recommendation.

5. Use total straight tube count only once. Equal-circuit pressure methods require an integer number of tubes per circuit; final circuit layouts also need bends, passes, distributors and actual flow distribution.

ASHRAE Refrigeration 2026 Ch 1 gas-line screening guides include suction 3.5-20 m/s, suction risers 6-20 m/s, discharge 10-18 m/s, and receiver-to-evaporator liquid below 1.5 m/s. Copeland oil-return guidance includes minimum horizontal/vertical suction velocities; use the applicable compressor documentation rather than applying all guides universally [S9, S10].

Velocity screening is not a final pipe or coil selection. Pressure drop, oil return, heat gain, flash margin and rated equipment boundaries must agree. Pages 35-41 explain the new smooth-tube coil pressure calculations; they do not infer microfin enhancement or supplier distributor behavior.

Advanced coil and fan selection inputs

Calculation scope dropdown: Duty and compressor sizing only hides coil airflow and UA result details and the advanced coil/fan workspace. Include evaporator and condenser sizing enables these calculations. Heat-rejection and evaporator duty remain available in both modes because the cycle still needs these energy quantities.

For each coil, use manufacturer thermal and air-pressure data for its geometry and actual design state. Required design inputs include refrigerant, face dimensions, tube diameter and wall, rows, fin type/spacing, circuits/distributor, entering air dry-bulb and humidity, refrigerant temperatures, superheat/subcooling, mass flow and wet/frost/dirty condition. The selected coil data must embody these parameters; the software does not infer them from a generic coil family.

Input group	Required selection data
Fan	CFM versus Pa or inches water; static/total basis; curve speed/density; fan count; allowable speed and stall region
Air resistance	Clean dry coil pressure at reference CFM; wet/frost and fouling factors; other variable losses; fixed pressure; local coil flow exponent
Thermal	Validated overall UA at reference CFM and local flow exponent, or manufacturer total-capacity versus CFM curve at the exact design state
Refrigerant	Supplier coil/header/distributor pressure loss; application-specific allowable saturation loss
Checks	Face area and velocity; capacity at operating CFM; motor efficiency/input; fan heat, sound, carryover, frost and serviceability

The demonstration fan curve is synthetic and must not be used to select real equipment. Manufacturer capacity curves are piecewise linearly interpolated only inside the supplied airflow range. Fan curves also are not extrapolated. Input pressure values must all use the selected static or total pressure basis; an equipment external-static curve may already include internal coil/filter losses, so do not add those losses twice [S12, S13].

Fan operating point equations and pressure budget

Correct a single-fan curve under similarity assumptions: CFMnew = CFMrated × speed ratio; pressure_new = pressure_rated × density ratio × speed ratio². Identical parallel fans multiply volume at equal pressure. Parallel-fan interactions and installed system effects require supplier confirmation. Fan affinity laws do not validate large speed changes or excluded stall regions.

At airflow V, coil loss = clean dry loss_ref × wet/frost factor × fouling factor × (V/Vref)^n. Other variable loss = other_ref × (V/Vref)². System pressure = coil + other + fixed pressure. The exponent n and factors must be fitted or obtained from the coil manufacturer. A quadratic resistance curve is a local approximation, especially when frost evolves.

The operating point solves fan pressure(V) = system pressure(V) within each measured fan-curve interval by bisection. The curve chart plots both in Pa versus actual CFM. All intersections are identified. A local stability screen requires fan slope lower than system slope; multiple intersections or excluded manufacturer stall ranges require engineering review [S12].

Maximum in-service coil loss at the required CFM = fan pressure(required)/(1 + pressure reserve) − other loss(required) − fixed pressure. The corresponding clean dry budget divides this by wet/frost and fouling multipliers. A negative budget means the fan cannot support even non-coil losses; displayed allowance is zero. This is a fan-dependent recommendation, not a universal ASHRAE coil maximum.

Required CFM for sensible evaporator screening = Qe × SHR/(rho × cp × air temperature drop) × 2118.88. Condenser CFM = Qc/(rho × cp × air rise) × 2118.88. At the solved operating point, estimated electrical fan power = volume_m³/s × pressure_Pa / efficiency /1000 kW. Use a verified fan power curve for final motor selection; assumed efficiency is not a measured motor result.

Include evaporator fan heat in room load once. Changing the fan selection can change the load; update the fan heat schedule and calculate again. The software does not automatically converge that heat-load/fan selection loop. Use dirty/wet data and frost/defrost conditions appropriate to the application [S13].

Coil capacity at the selected airflow

Dry constant-temperature screening uses UA(V) = UAref × (V/Vthermal,ref)^m, with entered local exponent m. Air heat-capacity rate C = rho × volume × cp in kW/K. Effectiveness epsilon = 1 − exp(−UA/C). Capacity = epsilon × C × absolute(entering air − refrigerant saturation). Leaving air follows the sensible energy balance. This is the Cr = 0 dry-coil limit, not a complete segmented coil model.

For evaporators this dry check compares with required sensible duty Qe × SHR; it does not establish total or latent capacity. For condensers the isothermal screening neglects desuperheating, subcooling and blend glide, so verify those regions with manufacturer data. Wet/frost coils and high-glide blends require a more detailed selection.

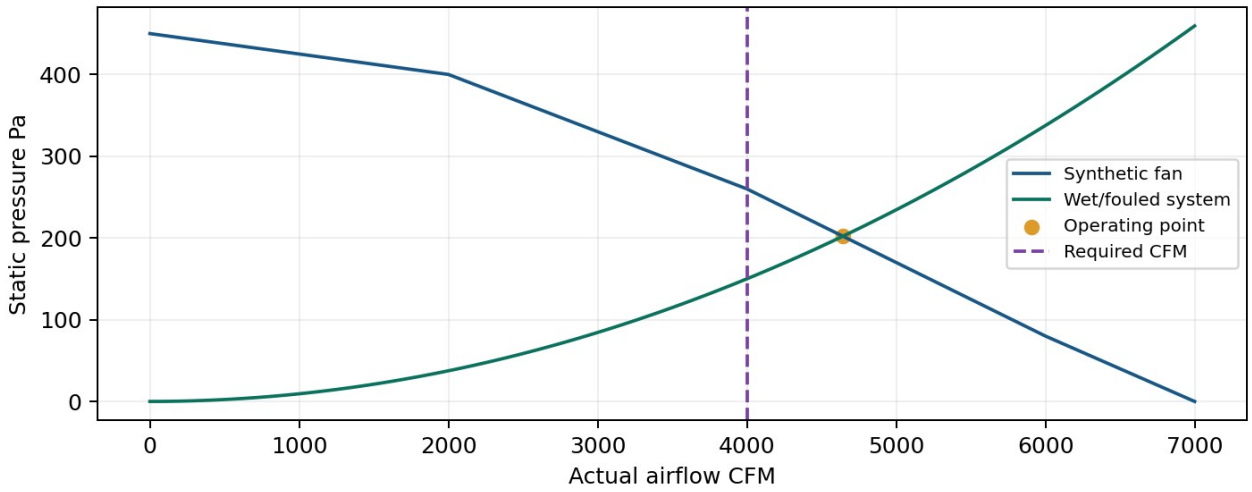
Manufacturer curve mode instead interpolates total evaporator cooling capacity or total condenser heat rejection at the fan operating airflow. It compares against total required duty. Curve conditions must match entering dry/wet bulb, refrigerant, Te/Tc convention, superheat, subcooling, geometry and circuitry. Leaving-air temperature is not inferred from a total-capacity curve without a sensible/latent split.

Worked synthetic fan example

One fan: (0 CFM,450 Pa), (2000,400), (4000,260), (6000,80), (7000,0). Reference resistance at 4000 CFM: dry coil 100 Pa, other 30 Pa, fixed 0; wet factor 1, fouling factor 1.2, exponent 2; density 1.2 kg/m³, speed ratio 1; entered pressure reserve 10%.

At required 4000 CFM, system pressure = 100×1.2+30 = 150 Pa. Available fan pressure = 260 Pa. Maximum in-service coil budget = 260/1.10 −30 = 206.36 Pa. Clean dry budget = 206.36/1.2 = 171.97 Pa. Reserve is a user assumption.

Intersection gives 4643.16 CFM at 202.12 Pa. At 55% air-to-electrical efficiency, input is 0.805 kW. Face area 1.5 m² gives 1.461 m/s. With UAref 2 kW/K at 4000 CFM, flow exponent 0.6, entering air 2°C and saturation −4°C, dry capacity is 8.929 kW; this is below a 9 kW sensible target despite adequate fan airflow.



Pressure loss at current cycle connections

Fan losses are air-side Pa. Coil/line refrigerant losses affect refrigerant pressure, phase and compressor requirements; do not add refrigerant kPa to fan static pressure.

UI boundary	Guide or allowance	Selection issue
1 suction	1 K equivalent, dew reference	Pressure loss versus oil return
2 discharge	1 K equivalent, dew reference	Actual hot-gas temperature differs from saturation
5 liquid line	0.5 K screen, bubble reference	Check downstream flash margin
6 to 1 evaporator	Entered supplier/application coil allowance	Include specified distributor/header boundary
2 to 5 condenser	Entered supplier/application coil allowance	Circuit and phase regions
5 to 6 valve	Intentional expansion pressure difference	Valve capacity and stable control

```
Equivalent_loss_K = Tsat(Pin) - Tsat(Pin - DP)
DP_limit = Pin - Psat[Tsat(Pin) - allowed_deltaT]
```

Use the same dew/bubble convention at both pressures and absolute pressure units. Refrigerant loss must be smaller than inlet pressure. ASHRAE line guides expressed as saturation-temperature loss do not automatically apply to two-phase coils; the coil allowance remains application/supplier input [S9].

1. R134a at -4 C dew: inlet 252.676 kPa abs. A 10 kPa suction loss produces approximately 1.073 K equivalent loss; 1 K corresponds to approximately 9.333 kPa at this inlet condition.

2. For liquid, compare downstream bubble temperature with entered liquid temperature to screen flashing. Subcooling must also cover heat gain and elevation outside the modeled boundary.

3. Choose manufacturer, basic calculated or advanced calculated coil pressure loss. Review the resulting saturation-loss allowance separately from the air-side fan loss.

These checks do not modify the reference zero-loss P-H cycle or recalculate compressor work using actual suction/discharge losses. Pages 39-41 update pressure along a coil with prescribed enthalpy boundaries, but do not couple the full cycle and compressor map. Zero entered losses remain zero assumptions, not completed equipment validation.

Thermostatic expansion valve sizing inputs

The TXV worksheet uses the operating-condition groups shown in the supplied Coolselector screenshot. It is an independent dry DX sizing worksheet, not a reproduction of the Danfoss valve/orifice database. The screenshot identifies R134a, capacity, evaporating/condensing temperature, useful/additional superheat, subcooling/additional subcooling, load percentage and distributor pressure drop. It also shows optional discharge temperature and connection/family criteria.

Select kW, refrigeration tons or Btu/h capacity. Select °C with K differences or °F with °F differences. The unit dropdown converts existing entries. Absolute temperatures convert as (°F −32)×5/9; temperature differences convert as °F difference×5/9, without subtracting 32. Additional discharge temperature is recorded only and does not set useful TXV cooling duty.

Input	Meaning
Useful superheat	Temperature rise from evaporation dew to evaporator outlet; contributes to useful refrigerating effect
Additional superheat	Suction-line temperature rise outside the evaporator; excluded from useful cooling effect
Subcooling and additional subcooling	Entered total liquid cooling from condensing bubble temperature at Pc; do not duplicate measured total SC
Load percent	Operating duty as a fraction of entered design capacity
Distributor and evaporator loss	Downstream pressure allowance from valve exit to evaporator outlet
Liquid friction and lift	Upstream losses reduce valve inlet pressure; lift is net upward elevation
Manufacturer check	Corrected per-valve capacity, MOPD, equalization, family and connection criteria

Danfoss identifies refrigerant, operating range, connection and superheat-control characteristics as family-specific selection requirements [S14]. Supply manufacturer capacity corrected to actual liquid temperature, evaporation, net differential, refrigerant and superheat. A generic orifice equation for incompressible liquid is not used because flashing and valve-specific behavior can govern expansion-valve capacity.

TXV pressure budget and worked example

Operating duty = entered design cooling capacity × load percentage/100. Useful evaporator outlet h uses Pe and Te,dew + useful SH. Liquid h uses Pc and total bubble-referenced subcooling. Required mass = operating duty/(h_evap,out − h_liquid). Additional suction superheat changes suction h but not useful evaporator duty.

Static liquid head = rho_liquid × g × net upward lift, divided by 1000 to give kPa. Valve inlet pressure = Pc − liquid friction − static head. Valve outlet requirement = Pe + distributor loss + evaporator loss. Net available TXV differential = inlet pressure − outlet requirement. Check positive differential at full load and lowest condensing condition.

Liquid flash margin = bubble temperature at valve inlet pressure minus entered liquid temperature. Zero or negative margin warns of possible flashing. Liquid friction is treated without heat gain; static head assumes liquid density. After flashing this static-head approximation is not valid. Include actual fittings, accessories, elevation, heat gain and manufacturer valve inlet requirements.

Screenshot conditions: R134a; 2.843 TR; evaporation 14°F (−10°C); condensation 86°F (30°C); useful superheat 14.4°F difference (8 K); subcooling 3.6°F difference (2 K); additional SH/SC zero; load 100%; distributor pressure drop 0 psi. For this worked example liquid friction, evaporator loss and elevation are also assumed zero, because those values are not supplied in the screenshot.

This property model gives useful effect 160.640 kJ/kg, duty 9.998 kW and flow 0.06224 kg/s or 8.233 lb/min. Net differential = 770.196 − 200.603 = 569.593 kPa (82.612 psi). The screenshot shows 8.267 lb/min; our result differs slightly and should not be claimed as an exact Coolselector match. Property and rating conventions and rounded inputs can differ.

Parallel valves divide duty and mass equally only when the associated evaporators actually share the load evenly. Enter corrected manufacturer capacity per valve; required duty/capacity reports loading and flags insufficient capacity. Enter MOPD to check differential against the specified maximum. Capacity at one point does not prove adequate minimum-load stability or acceptable oversizing.

With downstream distributor/evaporator loss, internal equalization triggers a manufacturer review warning. Confirm external equalization requirements, bulb location, static/operating superheat, MOP charge/range, working pressure, connections and material. No particular Danfoss valve family or orifice is automatically approved by this worksheet [S14].

Worked detailed cold room calculation

Example inputs: 6 × 4 × 3 m; room 2°C; 0.25 W/(m² K) for every surface; four adjacent wall temperatures and roof 35°C; roof solar addition 5 K; floor boundary 20°C; door 2 m² at U 0.35 and adjacent 30°C; bridge allowance 10%.

Infiltration: 30°C / 60% RH into 2°C / 85% RH, pressure 101.325 kPa; 10 room volumes/day; peak 2 room volumes/hour. Product: 500 kg at cp 3.5, 20→2°C, no freezing, pull-down 10 h. Packaging: 20 kg at cp 1.5, 20→2°C.

Internal heat: 1 person at 300 W for 2 h; lights 100 W for 8 h; evaporator fan 150 W for 20 h; other equipment/respiration zero. Defrost: 2000 W for 1 h, fraction 0.5. Refrigeration 20 h/day; allowance 10%.

Calculated item	Result
Envelope transmission	0.916 kW
Daily infiltration	13.599 kWh/day
Peak infiltration	2.720 kW
Product energy	8.750 kWh/day
Packaging energy	0.150 kWh/day
Scheduled internal heat	4.400 kWh/day
Defrost into room	1.000 kWh/day
Daily total	49.873 kWh/day
Operating-hour duty	2.494 kW
Coincident peak duty	5.075 kW

Base duty = max(average, peak) = 5.075 kW. With 10% allowance, selected cooling duty = 5.583 kW. Different traffic, product schedules or surface conditions can change this result materially.

Validation and selection review

Check input units, realistic schedules, fan heat counted once, product properties, humidity and pressure. Compare load components to measured energy and previous similar installations. Peak concurrency is conservative and should be revised from actual schedules if excessive.

Temperature checks reject invalid approaches. Class guidance warns outside typical application bands. COP at or above the ideal Carnot bound is flagged. Enthalpy heat rejection is compared with cooling plus electrical input, but the difference must be interpreted using actual line and motor boundaries.

For final compressor selection evaluate normal running, hot ambient, warm pull-down, minimum load and defrost recovery. Check the exact operating envelope, discharge temperature, current, return gas, oil return, voltage/frequency and capacity controls. Software class ranges do not approve a model.

Prototype validation should measure airflows, entering/leaving dry and wet bulb, refrigerant pressures and temperatures, superheat, subcooling, electrical power and product core pull-down. Numerical software tests verify implementation of equations; they do not validate supplier performance or the selected assumptions.

Launching the two versions

Standalone: extract all files and open index.html in your laptop browser. Calculations work offline. Python Web: install Python 3.10 or later, extract all files, run start_windows.bat or python app.py, keep the window open, and open http://127.0.0.1:8000 in Edge or Chrome on that laptop. The in-app browser may not reach a laptop-local server. The first Python launch installs pinned CoolProp 7.2.0 in a local virtual environment and requires internet access. Thereafter calculations work locally. To install manually, run python -m pip install -r requirements.txt before python app.py --open.

The Python launcher now checks for the Python launcher or python command, reports errors and opens the browser after the server starts. For a port conflict, run python app.py --port 8001. The local server is not an internet production deployment.

Technical sources and recommendation provenance

S1 ASHRAE Handbook Refrigeration 2018 Chapter 15 Retail Food Store Refrigeration and Equipment
https://handbook.ashrae.org/handbooks/R18/SI/r18_ch15/r18_ch15_si.aspx
S2 ASHRAE Handbook Refrigeration 2018 Chapter 24 Refrigerated Facility Loads
https://handbook.ashrae.org/handbooks/R18/SI/r18_ch24/r18_ch24_si.aspx
S3 ASHRAE Handbook Fundamentals 2025 Chapter 1 Psychrometrics
https://handbook.ashrae.org/Handbooks/F25/SI/F25_Ch01/F25_Ch01_si.aspx
S4 Secop Product Portfolio March 2026 application class definitions
https://www.secop.com/fileadmin/user_upload/pdf-files/secop_product_portfolio_03-2026_desh000b1002.pdf
S5 Secop SC10G Technical Data September 2026 model specific ratings
https://www.secop.com/fileadmin/user_upload/SEPS/datasheets/en/sc10g_104g8000_r134a_220v_208v_50hz_60hz_09-2026_ds.pdf
S6 ASHRAE Handbook HVAC Systems and Equipment 2020 Chapter 38 Compressors
https://handbook.ashrae.org/Handbooks/S20/SI/s20_ch38/s20_ch38_si.aspx
S7 ASHRAE Handbook HVAC Systems and Equipment 2020 Chapter 39 Condensers
https://handbook.ashrae.org/handbooks/S20/SI/s20_ch39/s20_ch39_si.aspx
S8 CoolProp thermodynamic property library and High Level API
https://coolprop.org/coolprop/HighLevelAPI.html
S9 ASHRAE Handbook Refrigeration 2026 Chapter 1 Halocarbon and Hydrocarbon Refrigeration Systems
https://handbook.ashrae.org/Handbooks/R26/SI/r26_ch01/r26_ch01_si.aspx
S10 Copeland Suction line pipe sizing to ensure refrigerant velocity
https://copeland.custhelp.com/app/answers/detail/a_id/3968
S11 NIST EVAP COND Version 5 detailed finned tube coil simulation
https://www.nist.gov/services-resources/software/evap-cond-version-50
S12 AMCA Straightening Out Fan Curves October 2020
https://www.amca.org/educate/articles-and-technical-papers/amca-inmotion-articles/straightening-out-fan-curves.html
S13 Trane Coil Selection for Cleanability February 2021
https://www.trane.com/content/dam/Trane/Commercial/global/learning-center/engineers-newsletters/APP-PRC006A-EN_02252021.pdf
S14 Danfoss Thermostatic Expansion Valves and Coolselector selection resources
https://www.danfoss.com/en-us/products/dcs/valves/thermostatic-expansion-valves/thermostatic-expansion-valves/

Sources reviewed 6 October 2026. Publicly accessible chapter editions are identified explicitly; this manual does not claim all guidance comes from the latest ASHRAE Refrigeration edition. The calculator is not certified or endorsed by ASHRAE or Secop.

Engineering assumptions include packaged/frozen evaporator TD bands, AC evaporating and condensing bands, broad COP screening ranges, the 35% Carnot factor, thermal-bridge percentage, all example load/property values and schedule assumptions. Source recommendations should be used only in the stated application context.

Condenser configuration reference temperatures

Release 2.0.1 adds the following reference tables. Calculator formulas and preset values are unchanged. Use this section alongside Design guidance and source S7. Ambient means dry-bulb air actually entering the condenser, which can exceed local weather temperature because of recirculation.

Configuration	Meaning	Preset TD
Remote air cooled medium temperature	Separate air-cooled condenser serving chilled-product refrigeration	8 K
Remote air cooled low temperature	Separate air-cooled condenser serving frozen-product refrigeration	5.5 K
Packaged AC screening assumption	Air-conditioning condenser using a preliminary engineering assumption	12 K
Custom TD	User-entered approach based on supplier data or a validated design	Entered value

Medium and low temperature identify the evaporator application, not the outdoor temperature. Condenser TD is a temperature difference: Tcondensing = Tair entering + TDcond. A difference of 1 K equals a difference of 1°C or 1.8°F.

Condenser preset	TD K	25°C air	30°C air	35°C air	40°C air	45°C air
Remote medium	8	33	38	43	48	53
Remote low	5.5	30.5	35.5	40.5	45.5	50.5
Packaged AC	12	37	42	47	52	57

All results in the last five columns are condensing temperatures in °C. With Custom TD, add the entered TD to each entering-air temperature. Example: 35°C entering air + 10 K TD = 45°C condensing temperature.

ASHRAE condenser guidance and selection

ASHRAE Handbook HVAC Systems and Equipment 2020 Chapter 39 Condensers [S7] describes the following typical air-cooled condenser TD ranges at the indicated evaporating conditions. These are typical selection ranges, not universal mandatory limits.

Application	Reference evaporation	Typical condenser TD
Low-temperature refrigeration	−40 to −30°C	5–8 K
Medium-temperature refrigeration	Approximately −7°C	8–11 K
Air conditioning	Approximately +7°C	14–17 K

The current remote presets of 8 K and 5.5 K fall within these respective ranges. The packaged AC preset of 12 K is a separate engineering screening assumption below the ASHRAE typical 14–17 K range; it must not be presented as an ASHRAE recommendation. At 35°C entering air, the ASHRAE AC range gives 49–52°C condensation. For an ASHRAE-range comparison, enter 14–17 K using Custom TD and verify manufacturer performance.

Step by step selection

1 Identify the application and refrigerant and determine the required total heat rejection, including compressor heat within the chosen boundary.

2 Select the site design entering-air dry-bulb temperature. Source S7 suggests the 1% design-day condition; assess recirculation, installation restrictions and process reliability requirements.

3 Choose an initial TD from the appropriate application range or manufacturer data.

4 Calculate condensing temperature by adding TD to entering-air temperature. Example: 40°C air + 8 K TD = 48°C condensation.

5 Check compressor capacity, power and operating envelope at the resulting condensing and evaporating conditions.

6 Select the condenser using manufacturer total-heat-rejection ratings, actual airflow and installation conditions. Iterate TD, condenser size, fan operating point and compressor selection until duties match.

For refrigerant blends, document the saturation-temperature convention used by the supplier and the software; dew, bubble and midpoint temperatures are not interchangeable. These tables do not replace manufacturer rating conventions. Lower TD generally requires more condenser conductance or airflow for the same heat rejection.

Source S7: https://handbook.ashrae.org/handbooks/S20/SI/s20_ch39/s20_ch39_si.aspx — reviewed 6 October 2026. Existing source S1 provides the retail-food-store context for the remote presets. This revision distinguishes software assumptions from source guidance.

Coil geometry and surface calculations

Apply this sequence independently to evaporator and condenser. The shipped model uses rectangular continuous plate fins; a developed-area factor adjusts fin area but does not predict wavy/louvered enhancement.

Input	Example
Tube OD / wall / ID	9.525 / 0.350 / 8.825 mm
Rows / tubes per row / circuits	3 / 12 / 3
Straight tube length / transverse pitch / row pitch	1.000 m / 25.4 mm / 22.0 mm
Pattern / fin density / fin thickness	Staggered / 10 fins per inch / 0.120 mm

1. Convert dimensions to meters. ID = OD - 2 wall = 8.825 mm. N = rows x tubes per row = 36.

2. Total straight length = N x length = 36 m. Fin pitch = 0.0254/FPI = 0.00254 m; full fin count = floor(1/0.00254) = 393.

3. Staggered face-height estimate H = (tubes per row + 0.5) x transverse pitch = 0.3175 m. Depth = rows x longitudinal pitch = 0.066 m.

4. Face area = H x length = 0.3175 m2. Internal tube area Ai = pi ID total length.

```
Af = 2 Nf [H Depth - N pi OD^2/4] x area_factor
Ab = pi OD N (L - Nf fin_thickness)
Ao = Af + Ab
```

Results: Ai = 0.9981 m2; fin area Af = 14.4544 m2; exposed tube area Ab = 1.0264 m2; external area Ao = 15.4808 m2.

The height convention is a schematic estimate, not a finished coil casing dimension. Frontal clear area is not the minimum free area through a staggered bank. Bends, headers and distributors are excluded from these straight-tube surface calculations.

Surface resistance and dry coil capacity

Surface mode requires validated air and refrigerant film coefficients and fin efficiency at the rating condition. The following coefficient values are illustrative, not refrigerant/fin recommendations.

Parameter	Example
ha / hr	60 / 2000 W/(m2 K)
Fin efficiency / tube conductivity	0.85 / 385 W/(m K)
Air/ref fouling and contact resistance	Zero for this arithmetic example

```
Rair = 1/[ha (Ab + eta_fin Af)]
Rwall = ln(OD/ID)/(2 pi k_tube total_length)
Rref = 1/(hr Ai)
Rtotal = Rair + Rwall + Rref + Rf_air/Ao + Rf_ref/Ai + Rcontact
UA_kW_per_K = 1/Rtotal/1000
```

1. Use the areas from page 28. Rair = 0.001252 K/W; Rwall = 0.00000088 K/W; Rref = 0.000501 K/W.

2. With the zero fouling/contact assumption, UA = 0.5702 kW/K. Overall external surface efficiency = (Ab + eta_fin Af)/Ao = 0.8599.

3. At operating flow, UA_op = UA_rated (CFM_op/CFM_rated)^n. n is a validated local fit, not a universal exponent.

4. Dry air capacity rate C_air = rho Q cp in kW/K. Effectiveness = 1 - exp(-UA_op/C_air). Dry capacity = effectiveness C_air |T_air - T_refrigerant|.

Example: Q = 1 m3/s, rho = 1.2 kg/m3, cp = 1.006 kJ/(kg K), UA = 2 kW/K and temperature difference = 10 K. C = 1.2072 kW/K; effectiveness = 0.8092; dry sensible capacity = 9.769 kW (rounding shown).

Wet latent performance, frosting and detailed condenser heat regions are not predicted by this constant-temperature dry UA relation. For those cases, use an actual-condition manufacturer total-capacity curve or a validated thermal model.

Face area sensible cooling and latent cooling

```
Q_air_m3_per_s = CFM x 0.00047194745
A_face = Q_air / target_face_velocity
SHR = sensible_cooling / total_cooling
Q_sensible = Q_total SHR
Q_latent = Q_total (1 - SHR)
```

1. Select required design airflow, current fan operating airflow, or entered airflow. Use actual CFM at the selected air condition.

2. For 4000 CFM and target 2.0 m/s: volume flow = 1.887790 m3/s; recommended gross face area = 0.943895 m2 = 10.160 ft2.

3. Apply the recommendation to the fan check only when face area is an independent input. In surface-geometry mode, revise geometry dimensions instead.

4. For total duty 10 kW and entered design SHR 0.90: sensible duty = 9.0 kW; latent duty = 1.0 kW.

5. If a manufacturer total capacity at actual conditions is 11 kW and its validated SHR is 0.80: available sensible = 8.8 kW and available latent = 2.2 kW.

Carryover guidance

The traditional wet-coil screening guide is 500 ft/min = 2.54 m/s. The software starting target 2.0 m/s is 394 ft/min and is an engineering choice. The selected supplier limit governs. Fin density/design, wettability, condensate loading, air distribution and drainage affect carryover. This applies to wet DX and chilled-water coils; a lower velocity is not a guarantee of zero carryover. Frosted freezer coils need their supplier criteria [U1].

Gross face velocity differs from minimum-passage velocity used in pressure-loss correlations. Do not substitute one for the other. Load SHR is an input allocation, not a predicted coil SHR. Changing face area does not create a new supplier capacity/loss curve.

Internal coil volume and refrigerant amount

```
Tube_volume_m3 = pi ID_m^2 total_tube_length_m / 4
Coil_volume_L = 1000 Tube_volume_m3 + entered_extra_volume_L
Liquid_filled_mass_kg = rho_liquid coil_volume_L / 1000
```

Quantity	Evaporator	Condenser
ID / rows / tubes per row / length	8.825 mm / 3 / 12 / 1 m	8.825 mm / 4 / 12 / 1 m
Straight tube count / total length	36 / 36 m	48 / 48 m
Straight tube volume	2.20202 L	2.93603 L
Entered bend/header/distributor volume	0.20 L	0.30 L
Total entered coil volume	2.40202 L	3.23603 L

1. Read the selected ID method for each coil. Count physical straight tubes once; circuits do not multiply total tube count.

2. Calculate cylindrical tube volume and add measured extra internal volumes inside that coil boundary once.

3. For an illustrative liquid density 1000 kg/m3, masses are 2.40202 kg and 3.23603 kg; combined coil volume and liquid-filled mass are 5.63805 L and 5.63805 kg.

4. Automatic density uses condenser saturated-liquid point 4 at its displayed bubble temperature for both coil bounds. Alternatively enter validated liquid density and its reference temperature.

A point in a thermodynamic diagram has no physical volume. This calculation counts only the two entered coils. It excludes system lines, receiver, accumulator and compressor. The 100% liquid-filled mass is a theoretical inventory bound at the stated density, not an allowable maximum system charge or a filling instruction. Actual operating coils contain vapor and liquid; permissible maximum charge requires refrigerant, installation and equipment safety limits.

Fan curve input and operating point

For each coil enter points 1-10 as actual CFM and pressure in the selected units. Use at least two complete points in increasing CFM order. Unused rows remain entirely blank. Enter supplier curve speed, density, static/total basis and valid operating limits.

```
Corrected_Q = rated_Q speed_ratio parallel_fan_count
Corrected_P = rated_P (rho_design/rho_rated) speed_ratio^2
System_P(Q) = coil_loss(Q) + other_Pref (Q/Qref)^2 + fixed_P
Fan_P(Qop) = System_P(Qop)
```

Example curve segment: 2000 CFM / 400 Pa and 4000 CFM / 260 Pa. For 4000-CFM reference losses of clean coil 100 Pa, dirty factor 1.2, other loss 30 Pa and fixed loss zero, the system loss is 150 (Q/4000)^2 Pa.

1. Within that segment, linear fan pressure = 540 - 0.07 Q.

2. Solve 540 - 0.07 Q = 150 (Q/4000)^2. Its positive root is approximately 4724.7 CFM, which lies outside the 2000-4000 segment. It is not a valid intersection on that segment.

3. Use the next supplied fan segment, for example 4000 CFM / 260 Pa to 6000 CFM / 80 Pa. Here P = 620 - 0.09 Q. Solve this segment with the same system curve.

4. The positive root is approximately 4643.2 CFM (verify with the exact quadratic). It lies in the 4000-6000 segment and is therefore usable for this illustrative curve.

At the selected intersection, calculate face velocity, coil/system pressure and fan power: electrical kW = Q_m3/s system_P_Pa / efficiency / 1000. Check thermal capacity and excluded stall regions separately.

Pressure reserve

Maximum in-service coil budget at required airflow = fan pressure/(1 + entered reserve) - other variable loss - fixed loss. For 260 Pa fan pressure, reserve 10% and other loss 30 Pa: budget = 206.36 Pa. Divide by wet/frost and dirty factors for a clean-dry budget. Reserve is a user design assumption, not a universal standard.

Calculated air pressure drop with constant coefficients

Select Geometry plus validated loss coefficients. Required inputs are actual reference CFM, design density, physical rows, minimum free area, per-row K, entry/exit K, a matching source and its CFM validity range.

```
v_min = Q_actual / A_min
K_total = rows K_row + K_entry_exit
DeltaP_clean = K_total rho_air v_min^2 / 2
```

Input	Illustrative value
CFM / air density	4000 / 1.2 kg/m3
Minimum free area / rows	0.50 m2 / 3
Per-row K / entry-exit K	2.0 / 1.0
Wet/frost multiplier / fouling multiplier	1.0 / 1.2

1. Convert airflow: 4000 x 0.00047194745 = 1.887790 m3/s.

2. Minimum velocity = 1.887790/0.50 = 3.775580 m/s. Total coefficient = 3 x 2 + 1 = 7.

3. Clean dry pressure drop = 7 x 1.2 x 3.775580 squared / 2 = 59.8710 Pa.

4. Fan solver applies wet/frost and dirty factors once: operating-condition loss at reference = 71.8452 Pa. At 2000 CFM it is one quarter of that value for constant coefficients.

5. Calculate the fan operating point; the calculated coil curve is combined with the other system losses.

These K values are fabricated arithmetic inputs. They are not recommendations for any fin type. The minimum area and K must use the same correlation velocity basis. A tested point can calibrate K_total = 2 DeltaP_test/(rho_test v_min,test squared), but this local calibration cannot be transferred to an unrelated geometry. The basic model uses a quadratic flow law; its coefficients must remain valid over the flow range [U2].

Reynolds dependent air pressure calculation

```
Re = rho_air v_min D_h / mu_air
Krow = linear_interpolation[validated (Re, Krow) data]
DeltaP_clean = (rows Krow + Kentry_exit) rho_air v_min^2/2
```

Illustrative two-point coefficient data: Re 400 / Krow 3.0 and Re 800 / Krow 2.0. Use Q = 4000 CFM, minimum area 0.50 m2, rho = 1.2 kg/m3, correlation hydraulic diameter 2.0 mm, viscosity 0.0000181 Pa s, 3 rows and entry/exit K = 1.0.

1. v_min = 3.775580 m/s. Re = 1.2 x 3.775580 x 0.002 / 0.0000181 = 500.629.

2. Interpolate Krow = 3 + (Re - 400)/(800 - 400) x (2 - 3) = 2.74843.

3. Clean loss = (3 Krow + 1) x 1.2 x v_min squared / 2 = 79.0749 Pa.

4. At every fan/system evaluation flow, recompute velocity, Re and Krow. This makes the coil curve non-quadratic. Existing wet/dirty multipliers still apply once.

5. Enter coverage for the entire nonzero fan evaluation range. Outside-curve Reynolds numbers are rejected; zero airflow has zero loss.

These coefficient points are illustrative, not a validated fin correlation. D_h must match the source definition. Do not infer a new louver/wave correlation solely from tube pattern or fins per inch. Changing the air density or viscosity changes Re and the coefficient interpolation. The supplied density is constant over the coil; an air thermal/density profile is not solved by this pressure method.

Single phase refrigerant pressure drop

The basic smooth-tube option can represent an all-liquid region with both qualities zero, or an all-vapor region with both qualities one. Supply property/reference conditions explicitly.

```
G = total_mass_flow / circuits / (pi ID^2/4)
Re = G ID / mu
f_Darcy = 64/Re for laminar flow
DeltaP_f = f_Darcy (L_circuit/ID) G^2/(2 rho)
```

Input	Liquid-only example
ID / equal circuits / circuit length	0.010 m / 2 / 10 m
Total mass flow	0.015707963 kg/s
Density / viscosity	1000 kg/m3 / 0.001 Pa s
Minor K / elevation / extra length	Zero / zero / zero

1. Circuit flow = 0.015707963/2 = 0.007853982 kg/s. Tube flow area = pi x 0.010 squared / 4 = 0.0000785398 m2.

2. Mass flux G = 100 kg/(m2 s). Re = 100 x 0.010 / 0.001 = 1000; Darcy factor = 64/1000 = 0.064.

3. Friction pressure drop = 0.064 x (10/0.010) x 100 squared /(2 x 1000) = 320 Pa = 0.320 kPa.

4. With no phase/density change, acceleration is zero. With no minor losses or elevation, net loss is 0.320 kPa.

Turbulent single-phase friction is solved by Colebrook: 1/sqrt(f) = -2 log10[roughness/(3.7 ID) + 2.51/(Re sqrt(f))]. The basic screen flags transitional segments. Fanning friction factor is one quarter of Darcy friction factor; do not interchange them. Equal circuits need an integer tube count per circuit. Add actual bend centerline length once, and use appropriate K for localized losses [U2, U3].

Homogeneous two phase screening example

```
1/rho_h = (1-x)/rho_l + x/rho_v
mu_h = (1-x) mu_l + x mu_v [stated model assumption]
DeltaP_total = friction + minor + acceleration + elevation
```

For a constant-quality arithmetic check use x = 0.5 at both ends, G = 100 kg/(m2 s), ID = 0.010 m, circuit length 10 m, liquid/vapor densities 1000/10 kg/m3 and viscosities 0.001/0.00001 Pa s. Enter minor K = 2 and outlet elevation 1 m above inlet.

1. Density = 1/(0.5/1000 + 0.5/10) = 19.801980 kg/m3. Viscosity = 0.000505 Pa s.

2. Re = G ID/mu = 1980.198; f_Darcy = 64/Re = 0.03232.

3. Friction = f (10/0.010) G squared /(2 rho_h) = 8160.800 Pa.

4. Minor loss = 2 G squared /(2 rho_h) = 505.000 Pa. Elevation = rho_h g (1 m) = 194.191 Pa.

5. Acceleration is zero because endpoint quality and densities match.

6. Total = 8859.991 Pa = 8.859991 kPa.

The software basic method integrates 40 midpoint segments when quality changes linearly. It holds phase densities/viscosities at entered reference values. The linear mixture-viscosity relation is a screening assumption, not a universal validated two-phase property relation. Slip and flow distribution are not resolved. The advanced method below updates local saturation/phase properties with pressure.

Lockhart Martinelli friction example

The advanced smooth-tube formulation uses individual phase flow rates and the ACHP documented implementation. It is selectable independently of the void-fraction closure. Zero roughness is required for this implementation.

```
Re_l = G (1-x) ID / mu_l; Re_v = G x ID / mu_v
a_l = 2 fF_l G^2 (1-x)^2/(rho_l ID)
a_v = 2 fF_v G^2 x^2/(rho_v ID)
Two_phase_gradient = a_l + C sqrt(a_l a_v) + a_v
```

At x = 0.5, G = 100, ID = 0.010 m and the same liquid/vapor properties as page 36:

1. Re_l = 500; Re_v = 50000. The phase friction factors here are Fanning factors.

2. Liquid is below Re 1000: fF_l = 16/Re_l = 0.032000. Vapor is above Re 2000: fF_v = 0.046/Re_v^0.2 = 0.005284.

3. Phase-alone gradients are a_l = 16.000 Pa/m and a_v = 264.201 Pa/m. Liquid Re below 1500 and vapor Re above 1500 gives C = 12.

4. Two-phase gradient = 16.000 + 12 sqrt(16.000 x 264.201) + 264.201 = 1060.405 Pa/m. A one-meter constant-property segment has this friction loss.

Phase Reynolds classification at 1500	C
Both above threshold	20
Liquid below; vapor above	12
Liquid above; vapor below	10
Both below threshold	5

For Re 1000-2000, the implementation blends the laminar and turbulent Fanning expressions. The phase endpoint limits are single-phase flow. The correlation is a selectable engineering model, not a claim of validity for all fluids, microfins, diameters or operating regimes [U3, U4].

Void fraction acceleration and elevation

```
alpha_h = 1/[1 + (1-x)/x (rho_v/rho_l)]
alpha_Zivi = 1/[1 + (1-x)/x (rho_v/rho_l)^(2/3)]
rho_bulk = alpha rho_v + (1-alpha) rho_l
S = x^2/(alpha rho_v) + (1-x)^2/[(1-alpha)rho_l]
DeltaP_acc = G^2 (S_out - S_in)
```

1. At x = 0.5 and phase densities 1000/10 kg/m3: homogeneous void fraction = 0.990099; Zivi void fraction = 0.955643.

2. For homogeneous closure, S = (1-x)/rho_l + x/rho_v = 0.0505 m3/kg. For the Zivi closure, S = 0.031796 m3/kg.

3. For a separate homogeneous acceleration example at fixed phase densities, x rises from 0.2 to 0.8. S_in = 0.0208; S_out = 0.0802 m3/kg. With G = 100, acceleration loss = 100 squared x (0.0802 - 0.0208) = 594 Pa.

4. Condensation reverses that endpoint change and produces a negative acceleration contribution (pressure recovery). Do not multiply this endpoint result by tube length.

5. Elevation uses bulk density: DeltaP_z = rho_bulk g Delta_z. Upward outlet elevation is positive; downward is negative. Local minor loss is K_segment G squared/(2 rho_bulk).

These are separate demonstrations of closures and contributions, not values to add to the unrelated constant-quality friction example. At x = 0 or 1, the momentum expression uses liquid or vapor endpoint limits to avoid division by zero. Zivi is an optional slip closure; applying it with homogeneous Darcy friction is explicitly a hybrid screening assumption [U3].

Advanced local pressure iteration procedure

Choose Segmented local-property pressure iteration. Load current cycle boundaries, then verify inlet pressure, enthalpies, mass flow, complete circuit length, zero roughness for Lockhart-Martinelli, minor K, elevation and segment count.

1. Split each equal-flow circuit into N segments. Prescribe h(s) = h_in + (h_out - h_in) s/L. This assumes uniform heat per length.

2. At local pressure, retrieve saturation endpoints and transport properties. For h between liquid/vapor endpoints, quality is (h - h_l)/(h_v - h_l). Otherwise solve a bounded temperature bracket at local pressure for the single-phase enthalpy.

3. Initialize a segment outlet pressure. Evaluate midpoint friction, minor loss and elevation, plus endpoint momentum/acceleration.

4. Update p_out = p_in - [DeltaP_f + DeltaP_K + DeltaP_z + DeltaP_acc]. Recompute properties at the revised pressure.

5. Use 50% relaxation until local pressure residual is below 0.1 Pa, or report failure after 35 iterations. Reject pressures below 1 kPa abs or unsupported property brackets.

6. Advance to the next segment. Sum contributions and record local distance, phase, temperature, pressure, enthalpy, quality and density.

7. Repeat at 2N segments. Compare total pressure drops; display sensitivity and a pressure-distance chart. Inputs changed during iteration invalidate the result.

Thermal boundary assumption

The inlet and outlet enthalpies specify heat transfer; their distribution is prescribed, not solved from fin coefficients or wet-air states. This allows the solver to traverse desuperheating, two-phase and subcooling regions, but the predicted region lengths depend on that assumption. Circuit distribution and full-cycle pressure/capacity feedback are not solved.

N is 10-80; the comparison uses up to 160 segments. Local iteration convergence and a fine mesh establish numerical behavior, not laboratory accuracy. Unsupported refrigerant transport properties are reported.

Worked advanced R134a condenser calculation

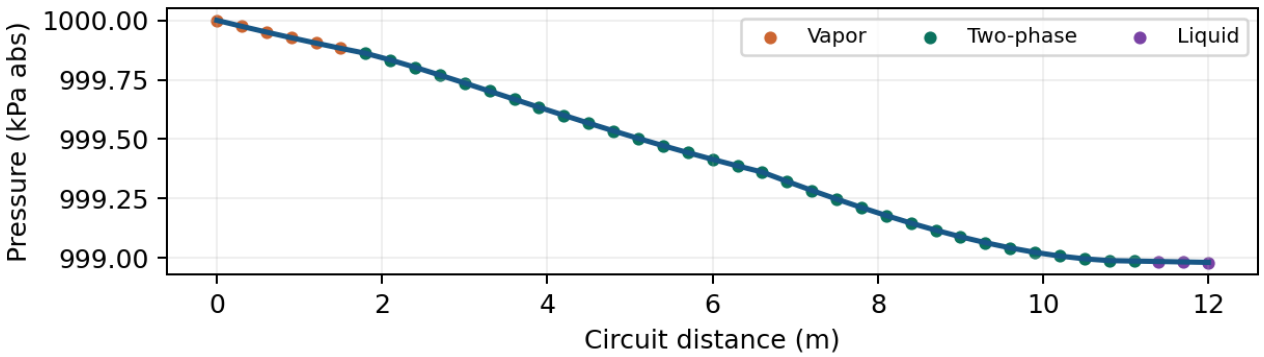
Actual numerical output from the shipped engine. Inlet pressure 1000 kPa abs; h_in 450 kJ/kg; h_out 240 kJ/kg; total flow 0.010 kg/s; three equal circuits; ID 8.825 mm; each circuit length 12 m. Smooth-tube Lockhart-Martinelli friction and homogeneous void closure; K = 0, elevation = 0, roughness = 0.

1. Tube area = 0.00006117 m2. Circuit mass flow = 0.010/3 = 0.00333333 kg/s. Mass flux = 54.49535 kg/(m2 s).

2. At 40 segments, each segment is 0.300 m and the prescribed enthalpy decreases by 5.25 kJ/kg. Retrieve local phase properties and iterate each outlet pressure.

3. Read the resulting state table. Qualities appear only in the two-phase region; liquid/vapor single-phase states have no vapor mass quality.

Distance m	Phase	T C	P kPa abs	h kJ/kg	Quality
0.0	Vapor	68.082	1000.000	450.000	-
3.0	Two-phase	39.378	999.736	397.500	0.8677
6.0	Two-phase	39.366	999.415	345.000	0.5470
9.0	Two-phase	39.354	999.090	292.500	0.2264
12.0	Liquid	28.809	998.982	240.000	-



Fine-mesh total loss = 1.018408 kPa; outlet pressure = 998.981592 kPa abs. Friction = 1.087306 kPa; acceleration = -0.068897 kPa; minor/elevation are zero. The negative acceleration term represents condensation pressure recovery.

Mesh checks and fan integration example

Check	Numerical example
20-segment total loss	1.018852 kPa
40-segment total loss	1.018408 kPa
Absolute difference	0.000444 kPa
Relative sensitivity	0.0436%
Local pressure iterations at 40 segments	370

```
Mesh_sensitivity_percent =
100 |DP_2N - DP_N| / max(|DP_2N|, 1 Pa)
```

The 2% sensitivity flag is an engineering screening choice, not an ASHRAE, AHRI or AMCA accuracy requirement. Refine and investigate the phase boundary/heat profile when sensitivity is large. Even a small value only shows mesh sensitivity for the selected model.

1. Calculated air loss writes the clean-dry reference pressure field. In Reynolds mode, the fan solver evaluates the coefficient function directly over its airflow range.

2. Calculated refrigerant loss writes the coil refrigerant-pressure-drop field in kPa. It remains independent of the fan pressure, which is an air-side quantity.

3. Calculate each coil fan point. The software reports operating CFM, velocity, coil/system loss, fan power, pressure budget and model capacity in the separate coil summaries.

4. Check equivalent saturation-temperature decrease against the entered application/supplier allowance. Inlet and outlet saturation states use the selected coil loss; the reference P-H cycle does not change.

5. Compare the numerical output against supplier/lab results at the same geometry, refrigerant, boundary conditions, flow and wet/frost condition.

Negative net refrigerant pressure recovery cannot be transferred into the nonnegative coil-loss field; a signed-pressure model is needed. Microfins, distributor/header flow, oil, imbalance, dryout and critical flow require additional validated treatments.

Saving comparisons definitions and source updates

1. Use Calculation log / project files to save a run after solving the current duties and each selected fan curve.

2. Download project JSON to preserve inputs and saved runs. Upload restores the current inputs and log; numerical models should be rerun and warnings reviewed.

3. Compare runs using actual values, difference or percent change. Difference = new - baseline. Percent = 100 (new - baseline)/baseline; a zero baseline has no defined percent.

4. Download comparison CSV when needed. Yellow cells identify changes from the preceding displayed run. Fan operating figures are current only when their saved input snapshot matches the present inputs.

Symbol	Meaning / units
Q air / G refrigerant	Actual volume flow m3/s / tube mass flux kg/(m2 s)
Aface / Amin / Dh	Gross coil face m2 / minimum passage m2 / correlation hydraulic diameter m
K / fD / fF	Loss coefficient / Darcy factor / Fanning factor; fD = 4 fF
x / alpha / h	Vapor mass quality / void fraction / specific enthalpy J/kg internally
rho / mu / p	Density kg/m3 / dynamic viscosity Pa s / pressure Pa absolute internally

Updated sources below supplement the original source register on page 25. These identify formulas/model scope; they do not validate the illustrative inputs.

[U1] Trane Performance Air Handler catalog, Coil Optimization: traditional 500 fpm carryover guide and supplier-specific performance. https://www.trane.com/content/dam/Trane/Commercial/lar/es/product-systems/Sistemas-de-agua-helada/Performance-climate-changer/CSAA/Catalogo/CLCH-PRC022-EN_092013_Performance%20Air%20Handler%20Catalog.pdf

[U2] NASA TM 20230011420: hydraulic channel loss structure. https://ntrs.nasa.gov/api/citations/20230011420/downloads/TM-20230011420.pdf

[U3] ACHP project Fluid Correlations: smooth-tube friction, void fraction and acceleration. https://achp.readthedocs.io/en/latest/ACHPComponents/FluidCorrelations.html

[U4] Chisholm 1967, theoretical basis for Lockhart-Martinelli. https://doi.org/10.1016/0017-9310(67)90047-6

[U5] CoolProp state/property API: https://coolprop.org/coolprop/LowLevelAPI.html . [U6] NIST EVAP-COND scope: https://www.nist.gov/publications/evap-cond-simulation-models-finned-tube-heat-exchangers-circuitry-optimization-version . NIST software is a reference, not this calculation engine.