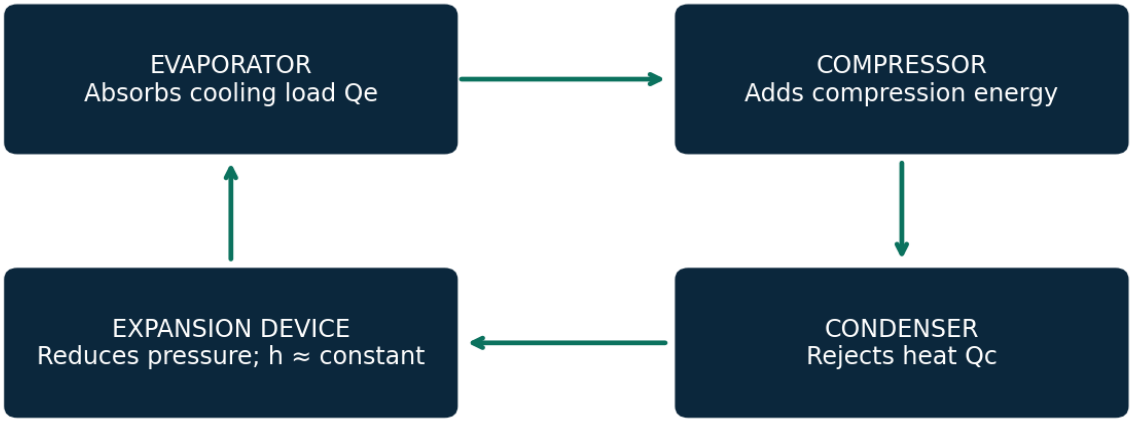
Refrigeration Component Sizing

Step-by-step calculation manual | Version 1 | 6 October 2026



Commercial refrigeration • Air conditioning • Cold rooms

1. Purpose and calculation map

This manual accompanies Refrigeration Component Studio, version 1 (6 October 2026). It covers subcritical direct-expansion systems with air-cooled condensers for commercial refrigeration, air conditioning and cold rooms. Inputs use SI units; selected results also show Btu/h, refrigeration tons (TR) and CFM.

The software calculates loads, design operating temperatures, component duties, airflow, face area and screening heat-transfer requirements. It does not select a compressor model from a built-in manufacturer map or produce final tube, fin and refrigerant-circuit geometry. Refrigerant selection labels the design requirement; it does not generate refrigerant properties.

Procedure: establish the cooling requirement → set evaporating and condensing temperatures → calculate evaporator airflow and area → obtain compressor performance → calculate condenser heat rejection → verify the matched system. Sources [1–3] support the load/selection workflow and heat-rejection basis.

The diagram shows energy flow. Qe is heat absorbed in the evaporator. Qc includes this heat and the energy transferred to the refrigerant during compression. Electrical compressor input is an approximation of that added heat when casing losses are neglected.

2. Collect application inputs

Cold rooms: enter room/product target temperature, dimensions, insulation U-value, external design temperature, door/infiltration load, internal heat, product mass and specific heat, product entering/final temperature, pull-down time, defrost heat and available refrigeration hours. Room mode uses one U-value and ambient for all six surfaces.

Commercial refrigeration: use a cabinet/load calculation that includes insulation, door openings, lighting, fan heat, product loading and defrost. Enter the known cooling duty if these items were calculated separately. Do not substitute compressor nominal horsepower for cooling capacity.

Air conditioning: obtain total and sensible design loads from a building load calculation, or calculate from entering/leaving moist-air states at a known dry-air mass flow. The application does not perform a whole-building solar, ventilation or occupancy calculation. Enter total load and sensible heat ratio (SHR).

For every application specify refrigerant, actual condenser entering-air temperature, electrical supply, return-gas/superheat conditions and liquid subcooling when requesting manufacturer selection. The software prompts for the thermal inputs; final equipment selection also requires these supplier-specific data.

Application examples are starting assumptions, not recommended operating limits. Choose density, TD, COP and face velocity for the real design. No default COP is refrigerant-specific.

3. Cooling-load calculations

Transmission: Qtrans = U × A × (Tout – Tin) / 1000, in kW. U is W/(m²·K), A is m². For different surfaces sum Ui × Ai × ΔTi. Room mode uses A = 2(LH + WH) + 2LW and clips negative envelope heat gain to zero.

Infiltration: Qinfil = m_da × (houtside – hinside), in kW when dry-air mass flow is kg/s and moist-air enthalpy is kJ/kg dry air. The software accepts an already-calculated average infiltration load; it does not infer door leakage from room size.

Product sensible energy: Eproduct = m × cp × (Ti – Tf) / 3600, in kWh. Product load during pull-down = Eproduct / tproduct, in kW. If freezing occurs, add sensible cooling above/below freezing and latent heat. Software freezing-energy input is the additional daily energy not already counted in the entered sensible term.

Daily energy: Edaily = (Qtrans + Qinfil + Qinternal) × 24 + Eproduct + Efreezing + Edefrost. All motor, fan and defrost values must represent heat entering the refrigerated space.

Operating-hour duty: Qaverage = Edaily / trefrigeration. Product pull-down duty: Qpeak = Qtrans + Qinfil + Qinternal + (Eproduct + Efreezing) / tproduct. Software base duty = max(Qaverage, Qpeak). Final duty Qe = Qbase × (1 + allowance/100).

This simplified daily model assumes continuous average infiltration/internal loads, one daily product batch and defrost outside available refrigeration hours. Review coincident peak events separately. If using known-load mode, load is already the design duty; the hours field does not rescale it. Source [1].

4. Worked cold-room load

Inputs: L = 6 m, W = 4 m, H = 3 m; U = 0.25 W/(m²·K); room 2°C; ambient 38°C; infiltration 0.50 kW; internal 0.40 kW; 500 kg product/day; cp = 3.5 kJ/(kg·K); product 20°C to 2°C; 10-hour product pull-down; 2 kWh/day defrost heat; refrigeration 20 h/day; allowance 10%. No product freezing.

Step 1 — Area: A = 2(6×3 + 4×3) + 2(6×4) = 108 m².

Step 2 — Transmission: Qtrans = 0.25 × 108 × (38–2) / 1000 = 0.972 kW.

Step 3 — Product energy: Eproduct = 500 × 3.5 × (20–2) / 3600 = 8.750 kWh.

Step 4 — Daily energy: Edaily = (0.972+0.500+0.400)×24 + 8.750 + 2 = 55.678 kWh/day.

Step 5 — Duties: Qaverage = 55.678/20 = 2.784 kW. Qpeak = 0.972+0.500+0.400+8.750/10 = 2.747 kW.

Step 6 — Design duty: Qe = max(2.784,2.747) × 1.10 = 3.062 kW. This is the component target before manufacturer-specific gross/net corrections. Confirm fan heat is counted exactly once.

5. Operating point and evaporator sizing

TD■ = evaporator entering-air temperature – refrigerant evaporating saturation temperature. Te = Tair,in – TD■. For air at 2°C and TD■ = 6 K, Te = –4°C. TD■ differs from the air temperature drop across the coil.

TD■ = refrigerant condensing temperature – condenser entering-air temperature. Tc = Tambient + TD■. At 38°C ambient and TD■ = 12 K, Tc = 50°C. Compressor selection must also account for suction/discharge piping pressure losses.

Evaporator sensible duty: Qs = Qe × SHR. Air volume flow: Vdot = Qs / (ρ × cp,air × ΔTair). Use kW, kg/m³ and kJ/(kg·K) to obtain m³/s. This sensible relation does not independently predict humidity removal.

Example: Qe = 10 kW, SHR = 0.90, ρ = 1.20 kg/m³, cp,air = 1.006 kJ/(kg·K), ΔTair = 3 K. Vdot = 9/(1.2×1.006×3) = 2.485 m³/s = 5,266 CFM. At 2 m/s face velocity, Afront = Vdot/vface = 1.243 m².

For a wet coil, verify Qt = m_da × (hin – hout) and moisture removal = m_da × (Win – Wout), where W is humidity ratio. SHR = Qs/Qt. Those checks require psychrometric states; entering SHR is an assumption, not a prediction.

Coil development: choose face height/width → tube diameter/wall and pitches → tube pattern → rows and fin spacing → circuits/distributor → iterate heat transfer and air/refrigerant pressure drop. Match the fan to coil plus complete air-path resistance. Check drainage, moisture carryover, frost and defrost. Final geometry requires a validated coil model or supplier rating.

For blends, use the manufacturer’s dew/bubble/mean-temperature rating convention. High glide cannot be modeled accurately as one constant refrigerant temperature.

6. Compressor duty and refrigerant mass flow

Estimated input: Pcomp = Qe / COP. At 10 kW cooling and an assumed COP of 3, Pcomp = 3.333 kW. COP is dimensionless and must correspond to the operating condition. A nominal motor power or horsepower is not a rated cooling capacity.

Use manufacturer selection data at the exact refrigerant, suction/discharge saturation conditions, return-gas temperature/superheat, subcooling/liquid temperature, voltage, frequency and speed. Required rated cooling capacity is the design evaporator duty on a consistent rating boundary. Verify capacity and power using the actual manufacturer map. Sources [1,3].

Refrigerant-side calculation: $m_r = Qe / (h_{evap,out} - h_{evap,in})$. Across an approximately adiabatic expansion valve, $h_{evap,in} \approx h_{liquid,before\ valve}$. With Qe in kW and h in kJ/kg, m_r is kg/s.

Illustrative enthalpy arithmetic only: $h_{evap,out}$ = 400 kJ/kg; h_{liquid} = 250 kJ/kg; Qe = 10 kW. m_r = 10/(400–250) = 0.06667 kg/s = 240 kg/h. These numbers are not property data for any specified refrigerant.

For preliminary swept displacement, $V_{swept} = m_r \times v_{suction} / \eta v$, in m^3/s, using specific volume at compressor suction and volumetric efficiency. Neither $v_{suction}$ nor ηv is calculated in this version. Compressor map data are preferable to an assumed ηv.

Check normal operation, hot ambient, warm-product pull-down, low load and defrost recovery against the compressor envelope. Review discharge temperature, current, minimum motor cooling, superheat, oil return and capacity-control requirements.

7. Condenser heat rejection and surface

Approximate condenser duty: $Q_c \approx Q_e + P_{comp}$. At Q_e = 10 kW and COP = 3, Q_c = 13.333 kW. Use supplier heat-rejection data where available; not all electrical input reaches the condenser if heat is lost from the casing or oil cooling. Source [2].

With consistent refrigerant state data: $Q_c = m_r \times (h_{cond,in} - h_{cond,out})$. Software uses condenser inlet discharge enthalpy and liquid-before-valve enthalpy, neglecting liquid-line heat transfer. Using m_r = 0.06667 kg/s, hdischarge = 445 and hliquid = 250 kJ/kg gives Q_c = 13.000 kW.

Condenser airflow: $\dot{V}_c = Q_c/(\rho \times c_{p,air} \times \Delta T_{air,c})$. At Q_c = 13.333 kW, density 1.2, c_p 1.006 and air rise 7 K: $\dot{V}_c$ = 1.578 m³/s = 3,343 CFM. Select actual fan airflow against installed resistance.

Catalog index: $Q_c/TD_{\blacksquare}$ = 13.333/12 = 1.111 kW/K. This entering-air-TD index is not the actual coil UA. Apply supplier refrigerant and rating corrections.

For preliminary isothermal heat transfer: $\Delta T_{lm} = (\Delta T1 - \Delta T2)/\ln(\Delta T1/\Delta T2)$. If the differences are equal, ΔT_{lm} equals that common difference. $Q = U \times A \times F \times \Delta T_{lm}$. Software assumes F = 1 and uses total external surface as the U-value area basis.

Condenser example: $\Delta T1$ = 50–38 = 12 K; $\Delta T2$ = 50–45 = 5 K. ΔT_{lm} = 7/ln(12/5) = 7.996 K; UA = 13.333/7.996 = 1.668 kW/K. At an explicitly assumed U = 35 W/(m²·K), A = 13,333/(35×7.996) = 47.64 m² external surface.

Evaporator dry equivalent: differences 6 and 3 K give ΔT_{lm} = 4.328 K, UA = 10/4.328 = 2.310 kW/K. At assumed U = 30 W/(m²·K), A = 77.02 m². A wet/frosted coil requires coupled heat and mass transfer; this result cannot establish latent capacity.

A final condenser model separates desuperheating, condensation and subcooling, accounts for glide and pressure drop, and checks refrigerant distribution. U is geometry/flow dependent; do not treat assumed U-values as universal constants.

8. Application examples and software workflow

Cold-room example uses the known-load default: 10 kW, air 2°C, TD■ 6 K, ambient 38°C, TD■ 12 K, SHR 0.90, air drop 3 K, COP 3. Results: Te –4°C; Tc 50°C; compressor input 3.333 kW; condenser duty 13.333 kW; evaporator flow 2.485 m³/s.

Commercial freezer example: 3 kW, air –18°C, TD■ 7 K, ambient 35°C, TD■ 12 K, SHR 0.90, air drop 3 K, COP 1.5. Results: Te –25°C; Tc 47°C; input 2.000 kW; condenser duty 5.000 kW; evaporator flow 0.746 m³/s. Verify frost/defrost and the low-temperature compressor envelope.

Air-conditioning example: 10 kW total, air 27°C, TD■ 20 K, ambient 35°C, TD■ 12 K, SHR 0.75, air drop 12 K, COP 3.2. Results: Te 7°C; Tc 47°C; input 3.125 kW; condenser duty 13.125 kW; evaporator flow 0.518 m³/s (about 1,097 CFM). Leaving dry-bulb is 15°C. Humidity/latent performance still needs verification.

Step 1 — Choose an application; the example inputs change. Step 2 — Choose known-load or room-load mode and enter the required load inputs. Step 3 — Choose refrigerant and set temperatures/TD. Step 4 — Enter SHR, air drop, density, specific heat and face velocity. Step 5 — Choose assumed COP or manufacturer electrical input at the design point. Step 6 — Optionally enter verified enthalpies and U-values. Step 7 — Click Calculate component duties. Step 8 — Review all warnings and print the results.

Results refresh when Calculate is clicked; changing a field alone does not recalculate. Changing application or loading an example resets its principal example settings. Inputs/results are session-only and are not a saved project database.

9. Validation, troubleshooting and sources

Input checks: positive duty, COP, airflow parameters and TD; SHR between 0 and 1; refrigeration hours no more than 24; leaving evaporator air warmer than evaporating refrigerant; condenser air rise smaller than condenser TD. At equal approach the idealized LMTD model is unsuitable.

Energy balance: compare refrigerant-side evaporator/condenser duty and compressor input using consistent state boundaries. Large differences can indicate mixed units, incorrect enthalpies, inconsistent operating conditions, line heat gains or casing heat losses.

Prototype validation: measure airflows and entering/leaving dry/wet-bulb, suction/discharge pressures, refrigerant temperatures, superheat, subcooling, compressor electrical input and pull-down time. Check hot ambient, steady state, defrost recovery and low load.

Conversions: 1 kW = 3,412.142 Btu/h; 1 TR = 3.516853 kW; 1 $m^3/s \approx 2{,}118.88$ CFM; a 1 K temperature difference equals a 1°C difference and 1.8°F difference. Absolute temperatures require their proper conversion.

[1] Copeland, Considerations for Sizing and Selecting Condensing Units for WICFs: https://e360blog.copeland.com/considerations-for-sizing-and-selecting-condensing-units-for-wicfs/ — load categories, design evaporating/condensing conditions and selection process.

[2] Heatcraft / Chandler, Air Cooled Condensers Technical Guide, CH-ACCTG, November 2020, condenser selection pp. 6–8: https://www.heatcraftrpd.com/dA/2dd9a0bd1b/CH-TB-C-AIRCOOLED-11-264.pdf — total heat rejection and manufacturer compression-heat data.

[3] Copeland Online Product Information: https://webapps.copeland.com/online-product-information/ — manufacturer capacity, power and application data. Use model-specific documentation for actual selection.

The energy, mass-flow, airflow and LMTD equations are standard conservation/heat-transfer relations derived here under the stated assumptions; the cited selection guides do not validate the illustrative COP, SHR, density or U inputs. Example enthalpies are fabricated arithmetic inputs and must be replaced by verified properties for real equipment. References checked 6 October 2026.