EXAM NOTES · MECHANICAL ENGINEERING

Refrigeration & Air-Conditioning

Complete unit-wise notes, worked examples, a question bank and a 35-hour study plan, mapped to **C. P. Arora, *Refrigeration and Air Conditioning*, 3rd edition (Tata McGraw-Hill)**

6	19	25+	300+
Units, as in your syllabus	Diagrams	Worked examples	Exam questions with answers

Use these notes with the book open. Every topic below says which chapter of Arora to read.

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How to use these notes

1. **Follow the 35-hour plan** (next pages). It is weighted by your syllabus contact hours and by how often each topic shows up in exams.
2. For every topic, **read the notes first** (fast, exam-oriented), **then the Arora chapter named in the grey "From the book" box**, and **then solve the questions** for that unit in Part B.
3. Questions marked ★ come up repeatedly in university papers. Do every ★ question at least once.
4. Keep a **refrigerant table (R134a, NH_3, R22), a steam table and a psychrometric chart** with you while practising. Most universities supply these in the exam hall, so learn to read them quickly.
5. The last day belongs to the **Formula sheet** (end of Part A) and the ★ questions.

About the book references. The chapter numbers and titles below follow the published contents of Arora's 3rd edition (24 chapters). Page numbers vary between printings, so this PDF gives the **chapter plus the topic to look for**. Use the book's index for exact pages. If your copy is the 4th edition (2021), the chapter order is almost the same, but check the contents page.

Arora 3rd edition: chapter list

Ch.	Title	Used for
1	Introduction	Unit 1: necessity, applications, methods of refrigeration
2	Refrigeration Machine and Reversed Carnot Cycle	Unit 1: unit of refrigeration, COP, ideal cycles
3	Vapour Compression System	Unit 2: simple, actual cycle, T-s and p-h, sub-cooling and superheating
4	Refrigerants	Unit 3: properties, nomenclature, ODP and GWP
5	Multipressure Systems	Unit 2: multistage, flash gas removal, intercooling, cascade
6	Refrigerant Compressors	Unit 2: volumetric efficiency, actual cycle (supporting)
7	Condensers	Unit 6
8	Expansion Devices	Unit 2 (supporting)
9	Evaporators	Unit 6
10	Complete Vapour Compression System	Units 2 and 6: effect of parameters, component matching
11	Gas Cycle Refrigeration	Unit 1: air refrigeration, Bell-Coleman, aircraft; Unit 3: vortex tube
12	Vapour-Absorption System	Unit 3: NH_3–water, LiBr–water, Electrolux, maximum COP
13	Ejector-Compression System	Unit 3: steam-jet refrigeration
14	Properties of Moist Air	Unit 4: psychrometric properties
15	Psychrometry of Air-Conditioning Processes	Unit 4: processes, summer and winter AC, coils, bypass factor
16	Design Conditions	Unit 5: comfort, effective temperature, comfort chart
17	Solar Radiation	Unit 5: load calculation (solar gain)
18	Heat Transfer through Building Structures	Unit 5: load calculation (U-values, CLTD)
19	Load Calculations and Applied Psychrometrics	Units 4 and 5: RSH, RLH, ERSH, ADP, cmm
20	Design of Air-Conditioning Apparatus	Unit 4: cooling and dehumidifying coils, air washers
21	Transmission and Distribution of Air	(outside your syllabus)

Ch.	Title	Used for
22	Fans	(outside your syllabus)
23	Refrigeration and Air-Conditioning Control	(outside your syllabus)
24	Food Refrigeration and Industrial Air Conditioning	Unit 5: industrial AC requirements

Syllabus → book map

Unit (hours)	Syllabus topic	Read in Arora 3e
1 (8)	Necessity and applications	Ch 1
	Unit of refrigeration, COP, reversed Carnot, types of ideal cycles	Ch 2
	Air refrigeration, Bell-Coleman cycle, open and dense air systems	Ch 11 (opening sections: reversed Brayton / Bell-Coleman cycle)
	Actual air-refrigeration system problems	Ch 11 (solved examples)
	Refrigeration needs of aircraft; actual systems	Ch 11 (aircraft refrigeration: simple, bootstrap, regenerative, reduced-ambient)
2 (10)	Working principle, components, simple VCR cycle, COP	Ch 3
	T-s and p-h charts, sub-cooling, superheating, cycle analysis	Ch 3 (+ refrigerant tables in Appendix)
	Actual cycle; influence of parameters on performance	Ch 3 (actual cycle), Ch 6 (volumetric efficiency), Ch 10
	Necessity of multistaging, multistage systems, cascade	Ch 5
3 (8)	Desirable properties, classification, nomenclature, ozone depletion, global warming	Ch 4
	Vapour absorption; maximum COP; NH_3–water, LiBr–water; three-fluid (Electrolux)	Ch 12
	Steam-jet refrigeration	Ch 13
	Vortex (Hilsch) tube	Ch 11 (last part). If missing in your printing, check the index under “Vortex tube”.
4 (8)	Psychrometric properties	Ch 14
	Processes; summer and winter AC; cooling and dehumidification	Ch 15, Ch 20 (coils)
	Load calculation and applied psychrometry	Ch 19
5 (6)	Human comfort, effective temperature, comfort chart	Ch 16
	Comfort AC; industrial AC requirements	Ch 16, Ch 24
	AC load calculations	Ch 17, 18, 19
6 (2)	Types of evaporators and condensers	Ch 9, Ch 7
	Effect on system performance; design features	Ch 10 (+ end sections of Ch 7 and Ch 9)

The 35-hour study plan

The plan has 35 one-hour blocks. You can run it as **3 days of about 12 hours** or **5 days of 7 hours**. Hours are weighted by contact hours, by how often each topic appears in exams, and by how hard its numericals are. Every block ends with questions, because reading alone does not stick.

Block	Hours	What to do	Questions to solve (Part B)
A. Unit 1: basics and air cycles	**H1–H5**		
	H1	Notes 1.1–1.4 (definitions, TR, COP, Carnot). Read Arora Ch 2.	U1 short questions 1–15
	H2	Notes 1.5–1.6: Bell-Coleman derivation and open vs dense air. Read Arora Ch 11, first part.	U1 theory ★
	H3	Worked examples 1.1–1.2; re-derive COP = 1/(r^(γ−1)/γ − 1) from memory	U1 numericals N1–N5
	H4	Notes 1.7: aircraft systems (simple, bootstrap, regenerative, reduced ambient) and ram effect. Arora Ch 11 aircraft section.	U1 theory on aircraft systems
	H5	Worked example 1.3, aircraft numericals	U1 N6–N10
B. Unit 2: vapour compression	**H6–H13**		
	H6	Notes 2.1–2.3: cycle on T-s and p-h. Practise reading a p-h chart. Arora Ch 3.	U2 short questions
	H7	Worked example 2.1 (simple saturated cycle). Do it once with tables.	U2 N1, N2, N4
	H8	Notes 2.4: sub-cooling and superheating with T-s/p-h sketches. Worked example 2.2.	U2 N3, theory ★
	H9	Notes 2.5–2.6: actual cycle, volumetric efficiency (Arora Ch 6), parameter effects (Ch 10)	U2 N5, N7, theory
	H10	Notes 2.7: why multistage, flash gas removal, intercooling. Arora Ch 5.	U2 theory on multistage
	H11	Worked example 2.3 (two-stage NH_3)	U2 N6, N9
	H12	Notes 2.8: cascade (purpose, refrigerant pairs, COP formula)	U2 N8, cascade theory
	H13	**Unit 1–2 checkpoint**: write all formulas from memory, then fix gaps	Mixed Unit 1–2 ★ questions
C. Unit 3: refrigerants and other systems	**H14–H19**		
	H14	Notes 3.1–3.3: properties, classification, nomenclature. Arora Ch 4.	U3 short questions 1–20
	H15	Notes 3.4: ODP, GWP, Montreal and Kigali, alternatives	U3 theory on environment
	H16	Notes 3.5–3.6: absorption basics, maximum COP derivation, NH_3–water. Arora Ch 12.	U3 N1–N3
	H17	Notes 3.7–3.8: LiBr–water and Electrolux; comparison tables	U3 theory ★
	H18	Notes 3.9: steam jet (Arora Ch 13)	U3 N6, theory

Block	Hours	What to do	Questions to solve (Part B)
	H19	Notes 3.10: vortex tube. Revise Unit 3.	U3 N5, remaining ★
D. Unit 4: psychrometry and AC	**H20–H26**		
	H20	Notes 4.1–4.2: properties and relations. Arora Ch 14.	U4 N1, N2, short questions
	H21	Practise the psychrometric chart: locate states, read h, W, DPT and WBT	U4 N3, N9
	H22	Notes 4.3: all processes, SHF, bypass factor, ADP. Arora Ch 15.	U4 N4, N8
	H23	Mixing, summer and winter AC systems	U4 N5, N7
	H24	Notes 4.5: RSH, RLH, ERSH, ESHF, ADP, cmm. Arora Ch 19.	Worked example 4.4
	H25	Applied psychrometry numericals	U4 N6, N10
	H26	**Unit 3–4 checkpoint**	Mixed ★ questions
E. Unit 5: comfort and loads	**H27–H30**		
	H27	Notes 5.1–5.3: heat balance of the body, effective temperature, comfort chart. Arora Ch 16.	U5 theory ★
	H28	Notes 5.4–5.5: comfort vs industrial AC, inside design conditions. Arora Ch 24.	U5 theory
	H29	Notes 5.6: load components (Ch 17, 18, 19). Worked example 5.1.	U5 N1
	H30	Load numericals	U5 N2–N4
F. Unit 6 and revision	**H31–H35**		
	H31	Notes 6.1–6.3: evaporators and condensers (Arora Ch 9 and Ch 7)	U6 theory ★
	H32	Notes 6.4: effect on performance (Ch 10), worked examples 6.1–6.2	U6 N1–N5
	H33	Objective / MCQ set (Part B, last section)	All MCQs
	H34	Formula sheet, plus every ★ derivation written out once (Carnot, Bell-Coleman, maximum absorption COP, BPF, optimum intermediate pressure)	★ derivations
	H35	Timed mock: 3 long answers and 2 numericals in 90 minutes. Review your mistakes.	Mock paper in Part B

EXAM TIP

Where the marks are. In most university papers, **Unit 2 (VCR) and Unit 4 (psychrometry)** together carry about half the marks, and both are numerical-heavy. Give them your best, freshest hours. Unit 6 has only 2 contact hours, and a short note or a “types of” question is the usual pattern.

PART A: NOTES

Unit 1: Introduction to Refrigeration and Air Refrigeration

From the book (Arora 3e). **Ch 1** Introduction (history, necessity and applications, methods of refrigeration) · **Ch 2** Refrigeration Machine and Reversed Carnot Cycle (unit of refrigeration, COP, heat pump, Carnot limits, reversed Brayton) · **Ch 11** Gas Cycle Refrigeration (Bell-Coleman cycle, open and dense air systems, aircraft refrigeration systems, solved examples). **Skip in Ch 11 if short of time:** detailed regenerative-cycle derivations. Make sure you know the schematic and the reason it is used.

1.1 Refrigeration: definition and necessity

Refrigeration is the process of removing heat from a space or substance and keeping it **below the temperature of its surroundings**. By the second law (Clausius statement), heat does not flow from cold to hot on its own, so **external work (or heat) must be supplied**. A refrigerator is therefore a **heat engine running in reverse**.

Necessity and applications (any 6–8 in an answer):

- **Food preservation**: cold storage, frozen food (−18 °C to −30 °C), dairy, meat, fish, transport refrigeration.
- **Comfort air-conditioning**: homes, offices, theatres, hospitals, vehicles, aircraft.
- **Industrial air-conditioning**: textile, printing, pharmaceutical, precision manufacturing, photographic film, computer and data-centre rooms.
- **Chemical and process industry**: removing heat of reaction, separating gases, liquefying gases (LPG, LNG, O_2, N_2), dewaxing lubricating oil.
- **Medical**: blood banks, vaccine storage, cryosurgery.
- **Ice manufacture**, cooling concrete in dams, ice-skating rinks, shrink-fitting, cooling cutting tools.
- **Cryogenics**: superconductivity, rocket propellants (liquid H_2/O_2), space simulation.

1.2 Methods of producing refrigeration (Arora Ch 1)

1. **Ice / evaporative refrigeration**: melting of ice, evaporation of water (desert coolers).
2. **Vapour compression**: a liquid refrigerant evaporates at low pressure and is compressed mechanically. Most common.
3. **Vapour absorption**: compression is done "thermally" with an absorbent and heat.
4. **Gas (air) cycle**: expansion of a gas in a work-producing device (Bell-Coleman, aircraft).
5. **Steam jet (ejector)**: water flashes at very low pressure created by a steam ejector.
6. **Vortex tube**: compressed air splits into hot and cold streams.
7. **Thermoelectric (Peltier effect)**, **magnetic (adiabatic demagnetisation)** and **dry ice (solid CO_2 sublimation at −78.5 °C)**.

1.3 Unit of refrigeration and COP

Tonne of refrigeration (TR): the refrigerating effect produced by melting **1 ton (2000 lb) of ice at 0 °C in 24 hours**.

$$1\ \text{TR} = \frac{907.2\ \text{kg} \times 335\ \text{kJ/kg}}{24 \times 3600\ \text{s}} \approx 3.5167\ \text{kW} = 211\ \text{kJ/min} \approx 12\,660\ \text{kJ/h}$$

(Older books use 210 kJ/min = 3.5 kW. **State the value you use in your answer**; 3.5 kW is widely accepted.)

Coefficient of performance (COP):

$$\mathrm{COP}_R = \frac{\text{Refrigerating effect}}{\text{Work input}} = \frac{Q_L}{W}, \qquad \mathrm{COP}_{HP} = \frac{Q_H}{W} = \mathrm{COP}_R + 1$$

- COP can be greater than 1 (unlike efficiency). It is a ratio of energies, not an efficiency.
- **Relative COP** = actual COP / theoretical COP.
- **Energy Efficiency Ratio (EER)** = refrigerating effect in Btu/h ÷ power in W = 3.412 × COP.
- **Power per TR** = 3.5167 / COP kW/TR. A useful sanity check: real plants run at 0.6–1.5 kW/TR.

1.4 Reversed Carnot cycle (Arora Ch 2)

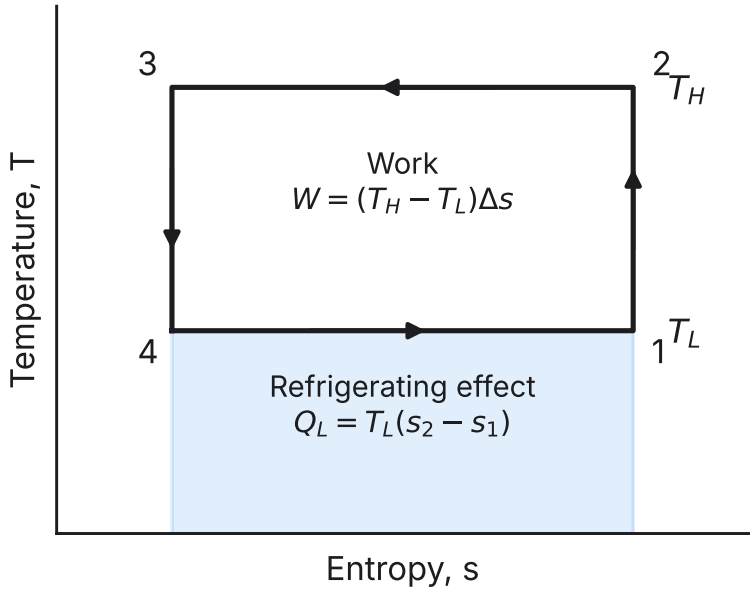

Fig 1.1 Reversed Carnot cycle on T-s. Area under 4-1 is the refrigerating effect; the enclosed rectangle is the work.

Processes (refrigerant inside the vapour dome for a vapour Carnot cycle):

- **1–2**: isentropic compression, $T_L \to T_H$
- **2–3**: isothermal heat rejection at T_H
- **3–4**: isentropic expansion, $T_H \to T_L$
- **4–1**: isothermal heat absorption at T_L (refrigeration)

$$Q_L = T_L(s_1 - s_4), \quad W = (T_H - T_L)(s_1 - s_4)$$

$$\boxed{\mathrm{COP}_{Carnot} = \frac{T_L}{T_H - T_L}}, \qquad \mathrm{COP}_{HP,Carnot} = \frac{T_H}{T_H - T_L}$$

Conclusions (frequently asked):

1. COP increases when T_L rises or T_H falls. **Raising T_L is more effective** than lowering T_H by the same amount, because $\partial\mathrm{COP}/\partial T_L > |\partial\mathrm{COP}/\partial T_H|$.
2. Carnot COP is the **maximum possible** COP between two temperature limits. It depends only on temperatures, not on the working fluid.
3. As $T_L \to 0$ K, COP → 0, so absolute zero cannot be reached.

Why the reversed Carnot cycle is not used in practice:

- *Gas as refrigerant*: isothermal heat transfer needs very slow processes, while isentropic processes need very fast ones. The two cannot happen in one machine, and the T-s rectangle is very thin, so large volumes are needed for little refrigeration.
- *Vapour as refrigerant*: (a) **wet compression** (1–2 in the two-phase region) causes liquid slugging, washes out lubricant and damages valves; (b) the **isentropic expander** (3–4) for a mostly-liquid mixture gives very little work and is costly, so it is replaced by a **throttle valve**. These changes turn the Carnot cycle into the practical vapour compression cycle.

SOLVED

Worked example 1.1 · A Carnot refrigerator works between −10 °C and 35 °C. Find the COP, the power needed per TR, and the COP if the same machine is used as a heat pump.

$T_L = 263.15$ K, $T_H = 308.15$ K → $\text{COP} = 263.15/45 =$ **5.85**. Power per TR = 3.5167 / 5.85 = **0.601 kW**. $\text{COP}_{HP} = 5.85 + 1 =$ **6.85**.

Types of ideal cycles of refrigeration (Arora Ch 2): the **reversed Carnot cycle** (ideal for constant-temperature source and sink), the **reversed Brayton / Bell-Coleman / Joule cycle** (gas cycle: two isentropic and two isobaric processes), the **Lorenz cycle** (ideal for variable-temperature source and sink), the **vapour compression cycle** and the **absorption cycle** (heat-operated, ideal COP derived in Unit 3).

1.5 Air refrigeration: the Bell-Coleman (reversed Brayton / Joule) cycle

Air was the first refrigerant used on ships (1880s, Bell-Coleman machines carrying frozen meat) because it is **safe, non-toxic, non-flammable, free and available everywhere**. Today it is used in **aircraft** and in gas liquefaction.

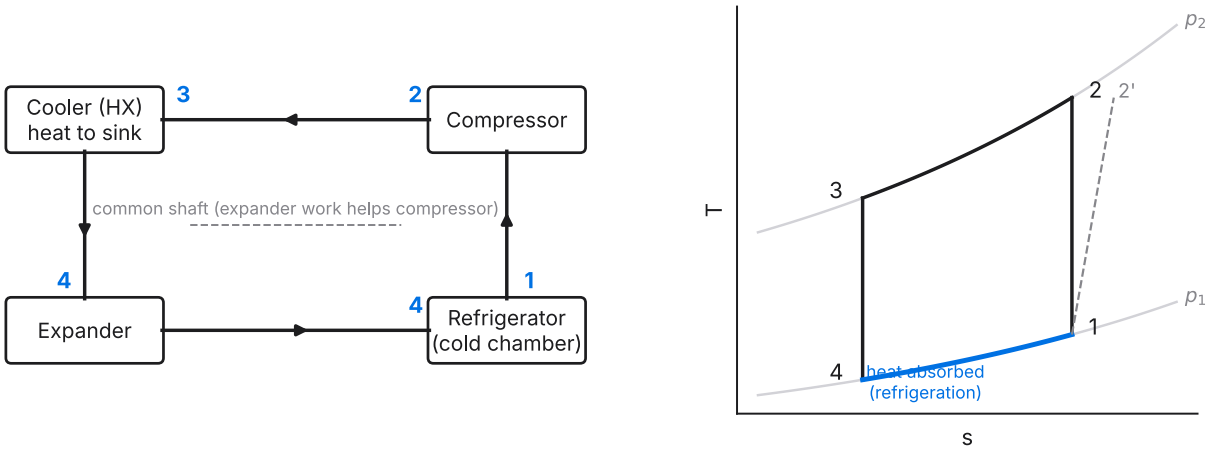

Fig 1.2 Bell-Coleman (closed / dense air) cycle: schematic and T-s diagram. 1–2 isentropic compression, 2–3 constant-pressure cooling, 3–4 isentropic expansion, 4–1 constant-pressure heat absorption (refrigeration).

Processes:

1. **1→2** isentropic compression from p_1 to p_2: $T_2 = T_1 r_p^{(\gamma-1)/\gamma}$.
2. **2→3** constant-pressure cooling in a cooler (heat to cooling water or atmosphere): $Q_{23} = c_p(T_2 - T_3)$.
3. **3→4** isentropic expansion in an expander (engine or turbine): $T_4 = T_3 / r_p^{(\gamma-1)/\gamma}$.
4. **4→1** constant-pressure heat absorption in the refrigerator: $Q_{41} = c_p(T_1 - T_4)$, the **refrigerating effect**.

COP derivation (★ very frequently asked):

$$W_{net} = Q_{23} - Q_{41} = c_p[(T_2 - T_3) - (T_1 - T_4)]$$

$$\text{COP} = \frac{T_1 - T_4}{(T_2 - T_3) - (T_1 - T_4)}$$

For isentropic processes between the same pressures, $\frac{T_2}{T_1} = \frac{T_3}{T_4} = r_p^{(\gamma-1)/\gamma}$, so $T_2 - T_3 = r_p^{(\gamma-1)/\gamma}(T_1 - T_4)$. Therefore:

$$\boxed{\text{COP}_{Bell\text{-}Coleman} = \frac{1}{r_p^{(\gamma-1)/\gamma} - 1} = \frac{T_4}{T_3 - T_4} = \frac{T_1}{T_2 - T_1}}$$

Key points:

- COP depends only on the **pressure ratio**. A lower r_p gives a higher COP but less refrigeration per kg of air, so a larger mass flow is needed.
- The COP is much lower than Carnot COP between the same peak temperatures, because heat is absorbed and rejected at **varying temperature** (no isothermal processes).
- **Polytropic processes** ($pv^n = C$): $T_2 = T_1 r_p^{(n-1)/n}$. Compressor work is $\frac{n}{n-1}R(T_2 - T_1)$ and expander work is $\frac{n}{n-1}R(T_3 - T_4)$ per kg, and RE = $c_p(T_1 - T_4)$ still.
- **With isentropic efficiencies:** $T_2 = T_1 + (T_{2s} - T_1)/\eta_c$ and $T_4 = T_3 - \eta_e(T_3 - T_{4s})$. COP drops sharply, because the work is a *small difference of two large quantities*.

Merits of air refrigeration: the refrigerant is free and safe; the system is light (good for aircraft); leaks are not dangerous; cold air can be supplied directly to the cabin. **Demerits:** low COP (high power per TR); large mass flow, because air carries only sensible heat (low heat capacity per kg); moisture in open systems freezes and chokes valves.

1.6 Open and dense air systems

	Open air system	Dense air (closed) system
Air path	Refrigerated air is passed **directly into the space** and then back to the compressor (or exhausted)	Air is kept in **closed pipes** and cools the space through a heat exchanger (coil)
Pressure at refrigerator	Atmospheric (≈ 1 bar)	Above atmospheric (e.g. 3–4 bar), hence "dense"
Size for a given capacity	Large: low-density air means a large volume flow	**Smaller compressor and expander**, because higher density means less volume per kg (v ∝ 1/p)
Moisture	Moisture from the space enters and **freezes in the expander**; a drier is needed	No moisture problem (dry air sealed in)
COP (same r_p)	Same theoretical COP	Same theoretical COP, but the mean effective pressure is higher, so the machine is smaller and friction losses are relatively lower
Heat exchanger	Not needed at the space	Needed at the space (extra ΔT and cost)

EXAM TIP

Exam phrasing: "Compare open and dense air refrigeration systems." Draw two small schematics and give 5–6 rows of the table above. Key line: *in a dense air system the suction pressure is well above atmospheric, so the volume handled per TR is reduced in proportion to the pressure.* In worked example N5 of Part B, specific volume falls from 0.804 to 0.268 m³/kg at 3 bar.

SOLVED

Worked example 1.2 (Bell-Coleman) · An air refrigerator works between 1 bar and 5 bar. Air leaves the cold chamber and enters the compressor at −5 °C, and is cooled to 30 °C in the cooler before entering the expander. Both processes are isentropic, γ = 1.4, c_p = 1.005 kJ/kg K. For 10 TR, find (a) the COP, (b) the mass flow of air, © the power, and (d) the volume flow at compressor inlet. (e) Repeat (a) to © with isentropic efficiencies of 85% for both compressor and expander.

$r_p^{(\gamma-1)/\gamma} = 5^{0.2857} = 1.5838$

$T_1 = 268$ K → $T_2 = 268 \times 1.5838 = 424.5$ K; $T_3 = 303$ K → $T_4 = 303/1.5838 = 191.3$ K

RE = 1.005 (268 − 191.3) = **77.07 kJ/kg**; W_{net} = 1.005[(424.5 − 303) − (268 − 191.3)] = **45.0 kJ/kg**

(a) COP = 77.07/45.0 = **1.713** (check: 1/(1.5838 − 1) = 1.713 ✓) (b) $\dot{m}$ = 10 × 3.5167 / 77.07 = **0.456 kg/s** © P = 0.456 × 45.0 = **20.5 kW** (d) $v_1 = RT_1/p_1 = 0.287 \times 268/100 = 0.769$ m³/kg → $\dot{V}$ = **0.351 m³/s**

(e) $T_2' = 268 + 156.5/0.85 = 452.1$ K, $T_4' = 303 - 0.85(111.7) = 208.1$ K RE = 1.005(268 − 208.1) = 60.2 kJ/kg; W = 1.005[(452.1 − 268) − (303 − 208.1)] = 89.6 kJ/kg COP = **0.67**; $\dot{m}$ = 0.584 kg/s; P = **52.3 kW**. Losses of 15% per machine cut the COP by more than half.

1.7 Refrigeration needs of aircraft and air-cycle systems

Why aircraft need cooling (★ "refrigeration needs of aircraft"):

1. **Solar radiation** through large transparent canopies and the skin.
2. **Heat from occupants**, avionics and electrical equipment.
3. **Ram effect**: at high speed, air brought to rest relative to the aircraft heats up ($\Delta T = V^2/2c_p$, about 38 K at 1000 km/h) and heats the skin.
4. **Heat from engines** and other machinery.
5. **Cabin pressurisation**: compressing thin outside air to cabin pressure raises its temperature sharply.

Why air cycle and not vapour compression in aircraft:

- **Low weight per TR** of the equipment. Weight is the deciding factor in aircraft.
- Compressed air is **already available** (engine bleed or ram air), and the cold air also **pressurises and ventilates** the cabin directly.
- No refrigerant leak hazard, and simple maintenance.
- The low COP is acceptable, because fuel used to carry extra equipment weight would cost more.

Ram air compression: for flight speed V and ambient T_1, p_1:

$$T_2' = T_1 + \frac{V^2}{2c_p}\ (\text{stagnation}), \qquad \frac{p_2'}{p_1} = \left(\frac{T_2'}{T_1}\right)^{\gamma/(\gamma-1)}, \qquad \eta_{ram} = \frac{p_2 - p_1}{p_2' - p_1}$$

(Note: V in m/s, c_p in J/kg K.) The actual stagnation temperature equals the ideal one, because energy is conserved. Only the pressure recovery is less than ideal.

Dry air rated temperature (DART): the temperature of the air leaving the cooling turbine, assuming no moisture. It is used to compare aircraft cooling systems.

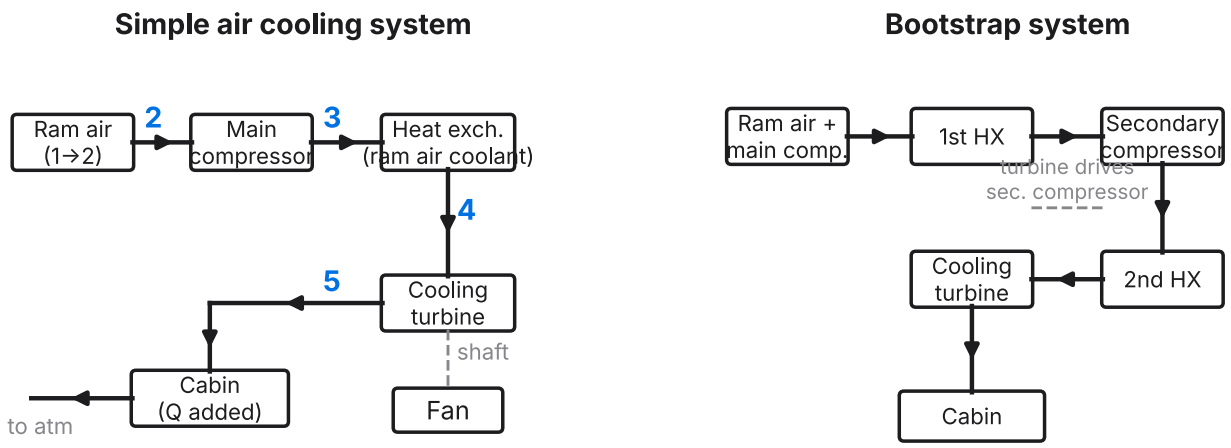

Fig 1.3 Simple and bootstrap air-cycle systems for aircraft (schematic).

Types of aircraft air-refrigeration systems (Arora Ch 11):

System	Arrangement	Use / feature
Simple	Ram air → main compressor → HX (cooled by ram air) → cooling turbine → cabin. The turbine drives a fan that pulls ram air through the HX.	Simple and light. Used in **ground-cooled / low-speed** aircraft. Turbine work drives the fan.
Bootstrap	After the first HX, air is compressed further in a **secondary compressor driven by the cooling turbine**, cooled in a second HX, then expanded.	Higher pressure before the turbine gives a lower turbine-exit temperature. No fan needed. Used in **transport aircraft**. Needs adequate ram air, so not suited to ground cooling.
Regenerative	Part of the cold turbine exhaust is bled back to precool the air before the turbine (regenerative HX).	**High-speed aircraft**, where ram air is too hot to be a good coolant.
Reduced ambient	Two cooling turbines: one expands the ram air used as HX coolant (lowering its temperature), the other supplies the cabin.	**Supersonic** aircraft and rockets, where ram temperatures are very high.
Bootstrap evaporative	Bootstrap plus a water (or alcohol) evaporator between the HXs.	Extra cooling at a lower DART.

Analysis of the simple system (points per Fig 1.3): ambient 1 → ram 2 → compressor 3 → HX 4 → turbine 5 → cabin.

$$\mathrm{RE} = c_p(T_i - T_5), \qquad W = c_p(T_3 - T_2), \qquad \mathrm{COP} = \frac{T_i - T_5}{T_3 - T_2}$$

where T_i is the cabin (inside) temperature. **Turbine work is not deducted**, because it drives the fan or secondary compressor, not the main compressor. Mass of air = $\dfrac{\text{Load (kW)}}{c_p(T_i - T_5)}$.

SOLVED

Worked example 1.3 (simple air cooling system) · An aircraft flies at 1000 km/h where the ambient is 0.35 bar and −15 °C. Ram efficiency is 90%. The main compressor has a pressure ratio of 3 and η_c = 85%. The HX cools the air to 305 K. The cooling turbine (η = 85%) expands it to the cabin pressure of 1.0 bar. The cabin is kept at 25 °C. For 10 TR, find the air mass flow, power and COP.

V = 277.8 m/s → $T_2 = 258 + 277.8^2/(2 \times 1005) =$ **296.4 K** $p_2' = 0.35(296.4/258)^{3.5} = 0.569$ bar → $p_2 = 0.35 + 0.9(0.219) =$ **0.547 bar** $T_3' = 296.4 \times 3^{0.2857} = 405.7$ K → $T_3 = 296.4 + 109.3/0.85 =$ **425.0 K**; $p_3 = 1.641$ bar Turbine: $T_5' = 305/(1.641)^{0.2857} = 264.8$ K → $T_5 = 305 - 0.85(40.2) =$ **270.8 K** (−2.3 °C) RE = 1.005(298 − 270.8) = 27.3 kJ/kg → $\dot{m}$ = 35.17/27.3 = **1.29 kg/s** W = 1.005(425.0 − 296.4) = 129.2 kJ/kg → P = **166 kW**; COP = 27.3/129.2 = **0.21**

The low COP is typical of air-cycle systems: about 4.7 kW per TR, against roughly 1 kW/TR for vapour compression.

1.8 Actual air-refrigeration systems: points to remember

- Compressors and expanders are not isentropic, and there are pressure drops in the HXs. The actual COP is often **0.2–0.6**.
- Moisture: in open systems, moisture freezes at the expander exit, so a water separator is placed after the turbine (high-pressure or low-pressure water separation).
- Mass flow is large because air carries only sensible heat: about 1–2 kg/s for 10 TR.
- Good results need a **low pressure ratio**, efficient turbomachines and **effective HXs**.

EXAM TIP

Unit 1 must-knows: definition of TR (with the 3.5167 kW calculation) · Carnot COP and why it is impractical · Bell-Coleman COP derivation · open vs dense · aircraft needs and the 4–5 system types with sketches · one complete Bell-Coleman numerical and one aircraft numerical.

Unit 2: Vapour Compression Refrigeration

From the book (Arora 3e). **Ch 3** Vapour Compression System (modifications to the reversed Carnot cycle, the vapour compression cycle on T-s and p-h, effect of operating conditions, superheating, sub-cooling, liquid-vapour regenerative HX, actual cycle) · **Ch 5** Multipressure Systems (multistage compression, flash gas removal, intercooling, multi-evaporator systems, cascade) · **Ch 6** Refrigerant Compressors (volumetric efficiency, clearance) · **Ch 10** Complete VC System (performance of the system as a whole). **Practise with:** the refrigerant property tables and p-h charts in the book's appendix.

2.1 Working principle and essential components

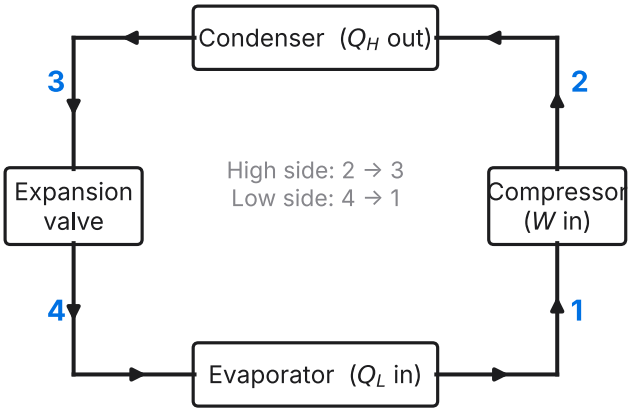

Fig 2.1 Simple vapour compression refrigeration system.

The refrigerant absorbs heat by **evaporating at low pressure and low temperature** in the evaporator, and rejects it by **condensing at high pressure and high temperature** in the condenser. The compressor provides the pressure lift; the expansion device brings the pressure back down.

Component	Function	State in → out
Compressor (reciprocating, rotary, screw, scroll, centrifugal)	Raises the pressure (and so the saturation temperature) of the vapour so that it can reject heat to the ambient	Dry saturated or superheated vapour (low p) → superheated vapour (high p)
Condenser (air or water cooled, evaporative)	Rejects heat: desuperheating, then condensation, then (sometimes) sub-cooling	Superheated vapour → saturated or sub-cooled liquid
Expansion device (capillary, TXV, AXV, float valve)	Reduces pressure (throttling, h = constant) and controls refrigerant flow to the evaporator	High-p liquid → low-p wet mixture (with flash gas)
Evaporator	Absorbs heat from the space or product. This is where the refrigerating effect is produced.	Wet mixture → saturated or superheated vapour

Accessories: receiver, filter-drier, sight glass, solenoid valve, accumulator, oil separator, high/low pressure cut-outs.

2.2 Simple saturated vapour compression cycle

Assumptions: the vapour leaves the evaporator **dry saturated**; the liquid leaves the condenser **saturated**; compression is **isentropic**; there are no pressure drops; expansion is **isenthalpic**.

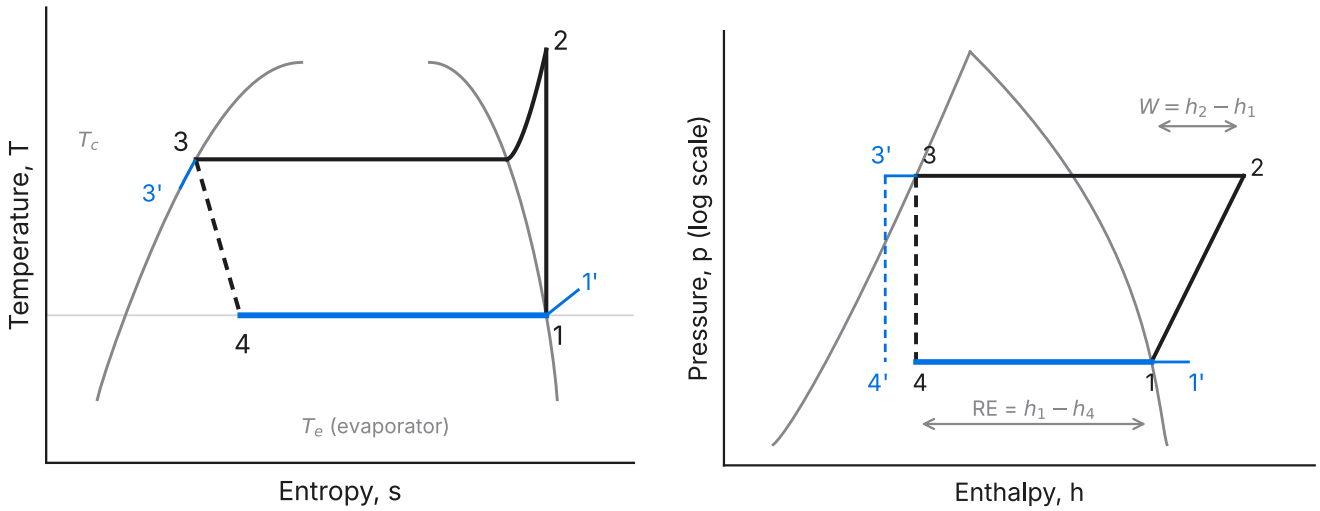

Fig 2.2 Simple VCR cycle (1-2-3-4) on T-s and p-h diagrams. Blue: the effect of sub-cooling (3→3', 4→4') and superheating (1→1').

Process	Name	Energy equation (per kg)
1 → 2	Isentropic compression	$W = h_2 - h_1$, with $s_2 = s_1$
2 → 3	Constant-pressure heat rejection	$q_c = h_2 - h_3$
3 → 4	Throttling	$h_4 = h_3$ (irreversible, $s_4 > s_3$)
4 → 1	Constant-pressure heat absorption	$\text{RE} = h_1 - h_4$

$$\boxed{\text{COP} = \frac{h_1 - h_4}{h_2 - h_1}} \qquad \dot{m} = \frac{211 \times \text{TR}}{h_1 - h_4}\ \text{kg/min} \qquad P = \dot{m}(h_2 - h_1)$$

$$\text{Heat rejection ratio} = \frac{q_c}{\text{RE}} = \frac{h_2 - h_3}{h_1 - h_4} = 1 + \frac{1}{\text{COP}}, \qquad \dot{V}_{suction} = \dot{m}\, v_1$$

Why the p-h chart is preferred: the two constant-pressure processes and the throttling process are **straight lines**, and RE, work and heat rejection are read directly as **horizontal lengths** (enthalpy differences). The T-s chart shows heat as areas, which are hard to measure.

Finding h_2 (state after compression):

- From the **p-h chart**: follow the constant-entropy line from 1 to p_2.
- From **superheat tables**: interpolate at p_c for $s = s_1$.
- If only saturated tables are given, use $s_1 = s_{g2} + c_p \ln(T_2/T_{sat})$ to find T_2, then $h_2 = h_{g2} + c_p(T_2 - T_{sat})$.
- When the vapour at 1 is wet (dryness x_1): $s_1 = s_{f1} + x_1 s_{fg1}$ and $h_1 = h_{f1} + x_1 h_{fg1}$.

SOLVED

Worked example 2.1 · An R134a refrigerator works on a simple saturated cycle between −10 °C (evaporator) and 40 °C (condenser) and produces 5 TR. Data (tables, IIR reference): at −10 °C, p = 2.006 bar, h_g = 392.7 kJ/kg, s_g = 1.733 kJ/kg K, v_g = 0.0996 m³/kg; at 40 °C, p = 10.17 bar, h_f = 256.4 kJ/kg; superheated vapour at 10.17 bar and s = 1.733: h_2 = 426.5 kJ/kg, t_2 = 46.3 °C. Find RE, mass flow, power, COP, Carnot COP, heat rejected and suction volume.

RE = $h_1 - h_4$ = 392.7 − 256.4 = **136.3 kJ/kg** $\dot{m}$ = 5 × 3.5167 / 136.3 = **0.129 kg/s** W = 426.5 − 392.7 = 33.8 kJ/kg → P = 0.129 × 33.8 = **4.36 kW** (0.87 kW/TR) COP = 136.3/33.8 = **4.03**; Carnot COP = 263.15/50 = 5.26; relative COP = 0.77 Q_c = 0.129 (426.5 − 256.4) = **21.95 kW**; $\dot{V}_1$ = 0.129 × 0.0996 = **0.0129 m³/s** Dryness after throttling: x_4 = 0.34. Over a third of the liquid flashes, which is why sub-cooling helps.

2.3 Representation on T-s and p-h charts: what to draw in the exam

- Always draw the **saturation dome**, label p_c **/** T_c and p_e **/** T_e, and number the points the same way on both charts.
- On p-h, the throttling line 3-4 is **vertical**. On T-s, it is a dotted line sloping to the right (irreversible).
- On T-s, the compression line 1-2 is **vertical**, and the discharge vapour is superheated (point 2 above the dome).
- Mark RE, W and q_c as enthalpy lengths on p-h.

2.4 Effect of sub-cooling and superheating (★)

Sub-cooling (3 → 3′): cooling the condensed liquid below its saturation temperature, using a sub-cooler, a larger condenser, or a liquid–suction HX.

- RE increases from $(h_1 - h_4)$ to $(h_1 - h_{4'})$. **Work is unchanged.** So **COP increases** and mass flow per TR decreases.
- Less flash gas at the evaporator inlet.
- It also prevents vapour bubbles forming in the liquid line (which would starve the expansion valve).

Superheating (1 → 1′): the vapour is heated above its saturation temperature before compression.

- **Useful superheat** (heat picked up *inside* the refrigerated space, in the evaporator): RE increases. Work also increases slightly ($h_{2'} - h_{1'} > h_2 - h_1$, because constant-entropy lines diverge). COP may **rise or fall depending on the refrigerant**: it rises for R134a, R12 and R290, and falls for NH_3 and R22.
- **Useless superheat** (heat picked up in the suction line *outside* the space): no gain in RE, more work, **COP decreases**.
- Superheating raises the discharge temperature (oil breakdown, a problem with NH_3) and the specific volume at suction (lower capacity per unit swept volume).
- It is still always kept at about 5 K, to **ensure no liquid enters the compressor** (protection against slugging), and because TXVs need superheat to operate.

Liquid–vapour regenerative heat exchanger: the warm liquid from the condenser sub-cools by heating the cold suction vapour, so $h_3 - h_{3'} = h_{1'} - h_1$. This is useful for R134a and R12, and harmful for NH_3 (high discharge temperature).

SOLVED

Worked example 2.2 · Repeat worked example 2.1 with 5 K sub-cooling and 5 K useful superheat. Data: $h_{1'}$ = 396.9 kJ/kg (s = 1.749, v = 0.1021); $h_{2'}$ = 431.6 kJ/kg (t_2 = 51.0 °C); liquid at 35 °C: $h_{3'}$ = 249.0 kJ/kg.

RE = 396.9 − 249.0 = **147.9 kJ/kg** (+8.5%); W = 431.6 − 396.9 = **34.7 kJ/kg** (+2.7%) COP = **4.26** (against 4.03); $\dot{m}$ = 0.119 kg/s; P = 4.13 kW; discharge temperature 51 °C (against 46 °C).

2.5 Actual vapour compression cycle

The actual cycle differs from the theoretical one in these ways (★ "Differentiate between theoretical and actual VCR cycle"):

1. **Pressure drop in the evaporator and condenser** (friction): the compressor must work over a larger pressure ratio.
2. **Pressure drop and heat gain in the suction line**: superheating, which raises v_1.
3. **Pressure drop across the suction and discharge valves** (wire-drawing).
4. **Non-isentropic compression**: friction and heat transfer through the cylinder walls. Entropy may rise or fall (water-jacketed cylinders). $\eta_{isen} = (h_{2s} - h_1)/(h_2 - h_1)$.
5. **Sub-cooling** of the liquid in the condenser or liquid line.
6. **Heat and pressure losses in the discharge line**.
7. The throttling is not exactly isenthalpic; heat may be gained in the expansion line.

Result: lower COP, a higher discharge temperature, and a lower volumetric efficiency.

Volumetric efficiency of a reciprocating compressor (Arora Ch 6):

$$\eta_v = 1 + C - C\left(\frac{p_d}{p_s}\right)^{1/n}, \qquad C = \frac{V_{clearance}}{V_{swept}}$$

Actual displacement needed: $V_{swept}/\mathrm{s} = \dfrac{\dot{m}\, v_1}{\eta_v} = \dfrac{\pi}{4} D^2 L \cdot \dfrac{N}{60} \cdot k$ (k = number of cylinders, single acting). η_v falls as the pressure ratio rises. When $(p_d/p_s) = \left(\frac{1+C}{C}\right)^n$, $\eta_v = 0$ and the compressor delivers nothing.

2.6 Influence of operating parameters on performance (★)

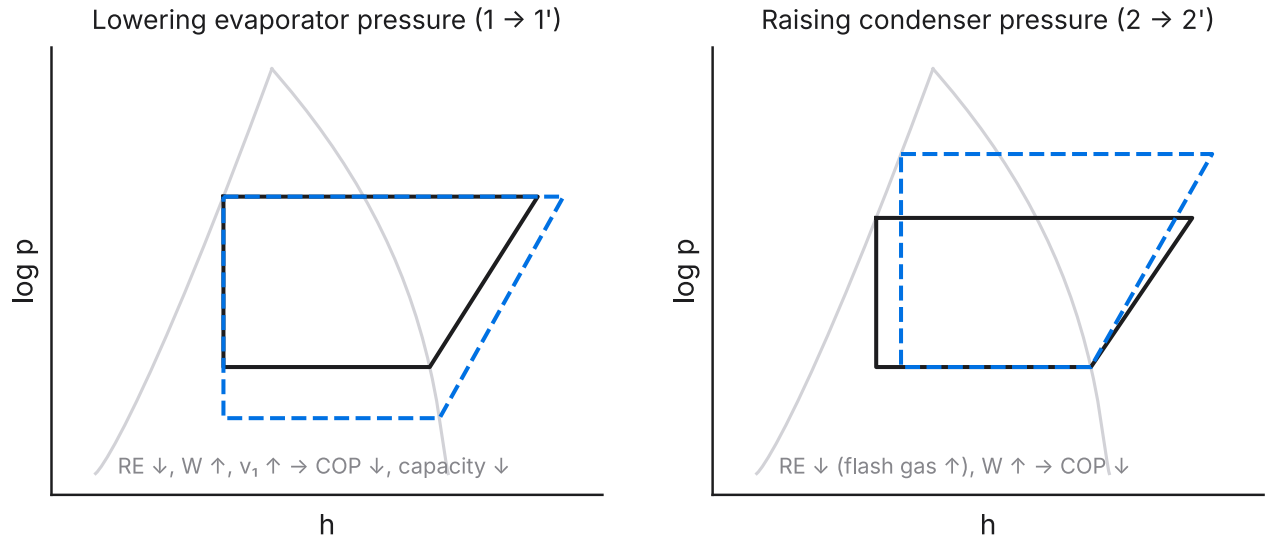

Fig 2.3 Effect of evaporator and condenser pressure on the cycle (schematic).

Change	RE per kg	Work per kg	Suction v_1	COP	Capacity (fixed compressor)
Evaporator temperature ↓	↓ (more flash gas)	↑	↑↑	↓↓	↓↓ (mass flow falls)
Condenser temperature ↑	↓	↑	–	↓	↓
Sub-cooling ↑	↑	same	–	↑	↑
Useful superheat ↑	↑	↑	↑	depends on refrigerant	~
Suction line pressure drop	–	↑	↑	↓	↓

The R134a figures below were computed with the same method as worked example 2.1, per TR:

T_e / T_c (°C)	−5 / 40	−10 / 40	−15 / 40	−20 / 40	−10 / 35	−10 / 45	−10 / 50
COP	4.68	4.03	3.50	3.06	4.65	3.52	3.08
kW/TR	0.75	0.87	1.00	1.15	0.76	1.00	1.14
Suction volume (L/s per TR)	2.09	2.57	3.19	3.98	2.44	2.72	2.89

Rule of thumb: every 1 K drop in evaporator temperature, or rise in condensing temperature, costs about **2–4% of COP**. So **design for the highest evaporator temperature and the lowest condenser temperature** the application allows. This links directly to Unit 6: evaporator and condenser size.

2.7 Multistage compression: necessity and systems (Arora Ch 5)

Why multistage (★): when the evaporator temperature is very low (below about −30 °C) or the pressure ratio is high (> 8–10):

1. **Volumetric efficiency** of a single-stage compressor falls drastically with a high pressure ratio.
2. **Discharge temperature becomes very high** (149 °C for NH_3 at −30/35 °C), which carbonises the oil and harms valves.
3. **Flash gas** after single throttling is large; this vapour does no cooling but must be compressed.
4. **Work increases.** Intercooling between stages brings compression closer to isothermal.

Methods used in multistage systems:

- **Flash gas removal:** throttle in two steps; separate the vapour formed in a **flash chamber** at the intermediate pressure, and send it directly to the HP compressor. Only liquid goes to the evaporator.
- **Intercooling:** desuperheat the LP discharge (a) with a water intercooler (only possible if the intermediate saturation temperature is above the water temperature) or (b) **in a flash intercooler** by bubbling it through liquid refrigerant at p_i.
- **Optimum intermediate pressure** (for minimum work with perfect intercooling, equal pressure ratios):

$$p_i = \sqrt{p_e \, p_c}$$

(For refrigerants this is a good approximation. Arora also gives a corrected relation based on temperatures.)

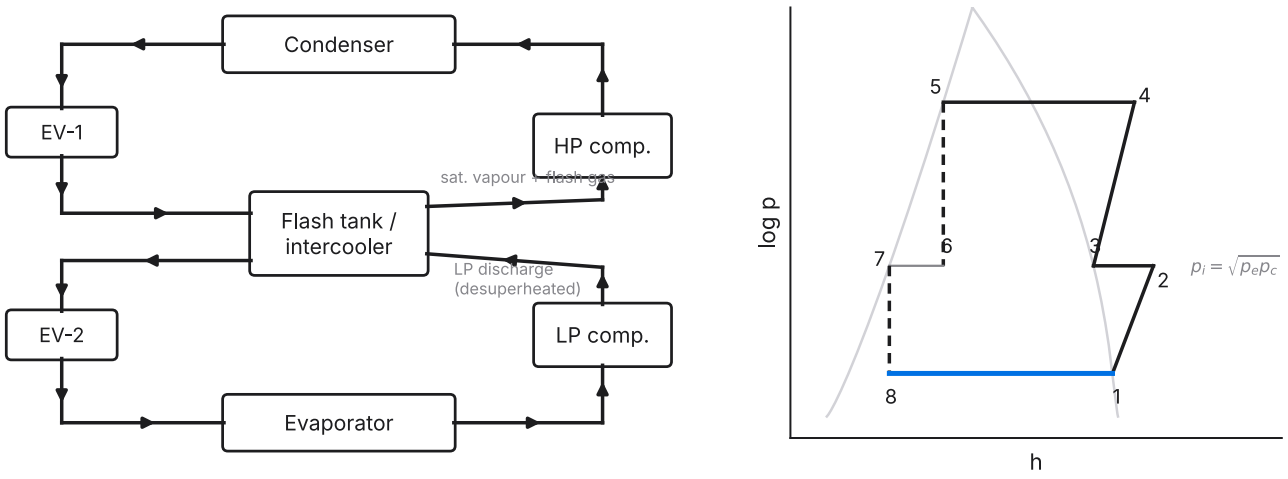

Fig 2.4 Two-stage compression with a flash intercooler (flash gas removal and intercooling) on a p-h diagram.

Analysis (Fig 2.4, flash intercooler):

- LP flow from the evaporator: $\dot{m}_L = \dfrac{Q_L}{h_1 - h_8}$, with $h_8 = h_7 = h_f(p_i)$.
- Energy balance of the flash intercooler: $\dot{m}_L h_2 + \dot{m}_H h_6 = \dot{m}_L h_7 + \dot{m}_H h_3$, so

$$\dot{m}_H = \dot{m}_L \frac{h_2 - h_7}{h_3 - h_6}$$

- Work: $W = \dot{m}_L(h_2 - h_1) + \dot{m}_H(h_4 - h_3)$, and COP = Q_L/W.

SOLVED

Worked example 2.3 (two-stage NH_3, flash intercooler) · 50 TR at −30 °C evaporator and 35 °C condenser. Compare with single-stage. p_e = 1.194 bar, p_c = 13.50 bar → $p_i = \sqrt{1.194 \times 13.5} =$ **4.01 bar** (T_{sat} = −1.8 °C) Enthalpies (kJ/kg): h_1 = 1569.0, h_2 = 1730.3 (49 °C), h_3 = 1605.5, h_4 = 1777.9 (85 °C), $h_5 = h_6$ = 511.6, $h_7 = h_8$ = 337.5 $\dot{m}_L$ = 175.8/(1569.0 − 337.5) = **0.143 kg/s**; $\dot{m}_H$ = 0.143 (1730.3 − 337.5)/(1605.5 − 511.6) = **0.182 kg/s** W = 0.143(161.3) + 0.182(172.4) = 23.0 + 31.3 = **54.4 kW** → COP = **3.23** Single-stage: h_2 = 1942.1 (149 °C!), COP = **2.83**, P = 62.0 kW Two-stage saves 12% power and cuts the discharge temperature from 149 °C to 85 °C.

Other multipressure arrangements (know the sketches): two-stage with water intercooler and liquid sub-cooler; **multiple evaporators at different temperatures** with individual expansion valves and back-pressure valves; individual compressors with multiple expansion valves (compound compression).

2.8 Cascade refrigeration system (★)

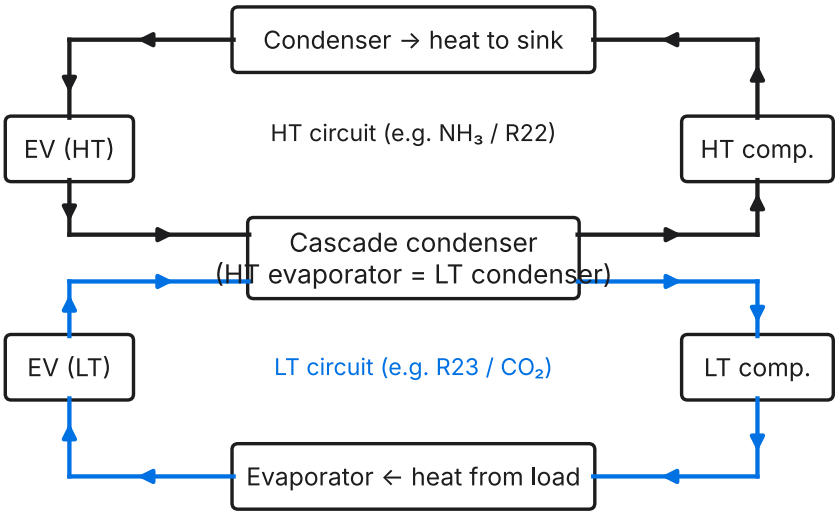

Fig 2.5 Two-stage cascade system. The cascade condenser is the evaporator of the high-temperature (HT) circuit and the condenser of the low-temperature (LT) circuit.

Necessity: for very low temperatures (−70 °C to −100 °C), no single refrigerant is suitable over the whole range:

- A refrigerant suited to low temperature (R23, R170 ethane, R744 CO_2, R1150) has a **very high condensing pressure** at ambient temperature, or is above its **critical temperature**.
- A refrigerant suited to high temperature (NH_3, R22, R134a) has **very low (vacuum) pressures and huge specific volumes** at −80 °C, or **freezes** (NH_3 freezes at −77.7 °C).

So **two (or more) separate circuits with different refrigerants** are used, thermally linked by the cascade condenser. The refrigerants never mix, so oils and refrigerants can be chosen independently.

COP of a two-stage cascade (both stages with COPs c_1 (LT) and c_2 (HT)):

$$\text{Heat rejected by LT} = Q_L\left(1+\tfrac{1}{c_1}\right) = \text{Load on HT}, \qquad \boxed{\text{COP}_{cascade} = \frac{c_1 c_2}{1 + c_1 + c_2}}$$

Example: $c_1 = 3$, $c_2 = 4$ → COP = 12/8 = **1.5**.

Mass flow ratio: $\dfrac{\dot m_{HT}}{\dot m_{LT}} = \dfrac{h_{2,LT} - h_{3,LT}}{h_{1,HT} - h_{4,HT}}$ (from the cascade condenser energy balance).

Applications: liquefying gases, blood plasma and pharmaceutical freeze-drying, environmental test chambers (−70 °C), ultra-low freezers, LNG. **Advantages:** each refrigerant works in its best range, reasonable pressures, and separate oils. **Disadvantages:** an overlap ΔT (about 5 K) is needed in the cascade condenser, which costs COP; and the plant is more complex and costly.

EXAM TIP

Unit 2 must-knows: p-h and T-s sketches with every point labelled · one full table-based numerical (practise until it takes under 15 minutes) · sub-cooling and superheating effects with sketches · actual vs theoretical cycle · volumetric efficiency formula · why multistage, plus the flash intercooler analysis · why cascade, with its COP formula.

Unit 3: Refrigerants and Absorption Refrigeration

From the book (Arora 3e). **Ch 4** Refrigerants (designation, selection, thermodynamic, chemical and physical requirements, secondary refrigerants, ozone depletion and global warming, alternatives) · **Ch 12** Vapour-Absorption System (simple system, maximum COP, common refrigerant-absorbent pairs, NH_3–water system with analyser and rectifier, LiBr–water system, three-fluid / Electrolux system) · **Ch 13** Ejector-Compression System (steam-jet refrigeration, ejector theory) · **Ch 11** (last part) vortex tube.

3.1 Desirable properties of refrigerants (★)

Thermodynamic

1. **Low boiling point** at atmospheric pressure, so the evaporator pressure is slightly *above* atmospheric (no air leaks in).
2. **Moderate condensing pressure**: lighter equipment and less leakage.
3. **High latent heat** of vaporisation: high RE per kg, so less mass flow.
4. **Low specific volume of vapour** at suction: smaller compressor (key for reciprocating compressors).
5. **High critical temperature** (well above condensing temperature) and **low freezing point** (well below evaporating temperature).
6. **Low specific heat of liquid** (less flash gas) and **high specific heat of vapour** (less superheat).
7. **High COP** and **low power per TR**.

Chemical

1. **Non-toxic**, **non-flammable** and **non-explosive** (ASHRAE safety class A1 is best).
2. **Chemically stable** and **inert** with construction materials, oil and moisture.
3. **Zero ODP and low GWP** (environment).

Physical

1. **High thermal conductivity** and **low viscosity**: better heat transfer, less pressure drop.
2. **Good miscibility with oil** (oil return to the compressor) or complete immiscibility (separation).
3. **High dielectric strength**, essential in hermetic compressors.
4. **Easy leak detection** (odour, halide torch, electronic detector).
5. **Low cost** and **ready availability**.

No single refrigerant satisfies all of these, so selection is a compromise. NH_3 has the best thermodynamic properties but is toxic. R12 was ideal except for ozone depletion.

3.2 Classification of refrigerants

Class	Examples	Notes
Primary (take part in the cycle)		
Halocarbons: CFC	R11, R12, R113, R114, R115	Contain Cl; high ODP. **Banned** (Montreal).
Halocarbons: HCFC	R22, R123, R141b	Some Cl; low ODP. **Being phased out**.
Halocarbons: HFC	R32, R125, R134a, R143a, R152a	No Cl, zero ODP, **but high GWP** (Kigali phase-down).

Class	Examples	Notes
HFO	R1234yf, R1234ze	Zero ODP, very low GWP. New generation.
Inorganic	R717 NH_3, R718 water, R744 CO_2, R729 air	Natural refrigerants.
Hydrocarbons	R290 propane, R600a isobutane, R1270 propylene, R170 ethane	Natural, zero ODP, negligible GWP; flammable (A3).
Azeotropic blends (R500 series)	R500, R502, R507A	Boil at a constant temperature like a pure fluid.
Zeotropic blends (R400 series)	R404A, R407C, R410A	Temperature **glide** during phase change; fractionation on leakage.
Secondary (carry the cold, no phase change)	Water, brines (NaCl, $CaCl_2$), glycols (ethylene, propylene)	Used in large plants and ice plants to avoid circulating primary refrigerant throughout.

3.3 Nomenclature (ASHRAE Standard 34) (★)

For a saturated hydrocarbon derivative $\mathrm{C}_m\mathrm{H}_n\mathrm{F}_p\mathrm{Cl}_q$: the refrigerant number is **R (m−1)(n+1)(p)**, and Cl is found from $n + p + q = 2m + 2$.

- Methane series (m = 1) gives two-digit numbers (the leading 0 is dropped): R11, R12, R22.
- Ethane series (m = 2): R1xx, e.g. R134a. The suffix **a, b** marks isomers (a = the most asymmetric arrangement).
- Bromine: add B and the number of Br atoms (R13B1 = CF_3Br).
- **Inorganics: R700 + molecular mass.** NH_3 = R717, H_2O = R718, CO_2 = R744, air = R729.
- **Unsaturated** (olefins): a fourth digit, R1(m−1)(n+1)(p), e.g. R1234yf = $CF_3CF{=}CH_2$.
- **Blends:** R4xx zeotropes, R5xx azeotropes (numbered in order of acceptance).
- **Butane series and others:** R600 (butane), R600a (isobutane).

Worked decoding examples:

Refrigerant	m−1, n+1, p	m, n, p	Cl (q)	Formula
R11	0, 1, 1	1, 0, 1	4−0−1 = 3	CCl_3F
R12	0, 1, 2	1, 0, 2	2	CCl_2F_2
R22	0, 2, 2	1, 1, 2	1	$CHClF_2$
R134a	1, 3, 4	2, 2, 4	0	CH_2FCF_3 ($C_2H_2F_4$)
R123	1, 2, 3	2, 1, 3	2	$CHCl_2CF_3$
R50	0, 5, 0	1, 4, 0	0	CH_4
R290 (propane)	2, 9, 0	3, 8, 0	0	C_3H_8

Reverse: $CHCl_2F$ → m = 1, n = 1, p = 1 → R(0)(2)(1) = **R21**. $C_2Cl_2F_4$ → m = 2, n = 0, p = 4 → **R114**.

3.4 Ozone depletion and global warming (★)

Ozone depletion. The stratospheric ozone layer (15–50 km) absorbs harmful UV-B radiation. CFC molecules are so stable that they survive long enough to reach the stratosphere. There, UV breaks off **chlorine atoms**, which destroy ozone catalytically:

$$\mathrm{Cl} + \mathrm{O_3} \rightarrow \mathrm{ClO} + \mathrm{O_2}, \qquad \mathrm{ClO} + \mathrm{O} \rightarrow \mathrm{Cl} + \mathrm{O_2}$$

One Cl atom can destroy about 100,000 O_3 molecules. The results are the Antarctic ozone hole, and more skin cancer, cataracts and crop damage.

Ozone depletion potential (ODP): depletion caused by 1 kg of the substance relative to 1 kg of **R11 (ODP = 1)**.

Global warming. Refrigerants are strong greenhouse gases that absorb infrared radiation emitted by the earth. **Global warming potential (GWP):** warming caused by 1 kg relative to 1 kg of **CO_2 (GWP = 1)**, usually over a 100-year horizon. **TEWI (total equivalent warming impact)** = direct effect (leakage × GWP) + indirect effect (CO_2 emitted to generate the electricity the plant uses over its life). A low-GWP but inefficient refrigerant can still have a high TEWI.

Refrigerant	Type	ODP	GWP (100 yr, approx.)	Safety
R11	CFC	1.0	4750	A1
R12	CFC	1.0	10 900	A1
R22	HCFC	0.055	1810	A1
R134a	HFC	0	1430	A1
R410A	HFC blend	0	2088	A1
R32	HFC	0	675	A2L
R1234yf	HFO	0	< 1 to 4	A2L
R717 (NH_3)	natural	0	0	B2L
R744 (CO_2)	natural	0	1	A1
R290 / R600a	HC	0	~3	A3

(Values vary slightly between IPCC reports. In exams, quoting the right order of magnitude is enough.)

International protocols: Montreal Protocol (1987) phased out CFCs (in developing countries by 2010) and is phasing out HCFCs (developing countries by 2030). **Kyoto Protocol (1997)** listed HFCs among the greenhouse gases. **Kigali Amendment (2016)** to the Montreal Protocol phases *down* HFCs; India's schedule is a freeze in 2028 and an 85% reduction by 2047.

Alternatives (replacements):

Old	Typical replacement
R12 (domestic refrigerators, car AC)	R134a, R600a (isobutane), R1234yf (car AC)
R22 (room AC, packaged units)	R410A, R32, R290, R407C
R11 (centrifugal chillers)	R123 (interim), R1233zd, R514A
R502 (low-temperature commercial)	R404A, R507A, CO_2 cascade

3.5 Vapour absorption system: principle (★)

The **compressor of a VCR system is replaced by a "thermal compressor"**: an **absorber, a pump and a generator**. Low-pressure refrigerant vapour from the evaporator is **absorbed** by a weak solution in the absorber, which is cooled. The resulting strong solution is **pumped** (a liquid, so little work, since $v\,dp$ is small) to the generator. There, **heat** (steam, hot water, waste heat, solar) drives the refrigerant vapour out at high pressure. The condenser, expansion valve and evaporator work exactly as in VCR.

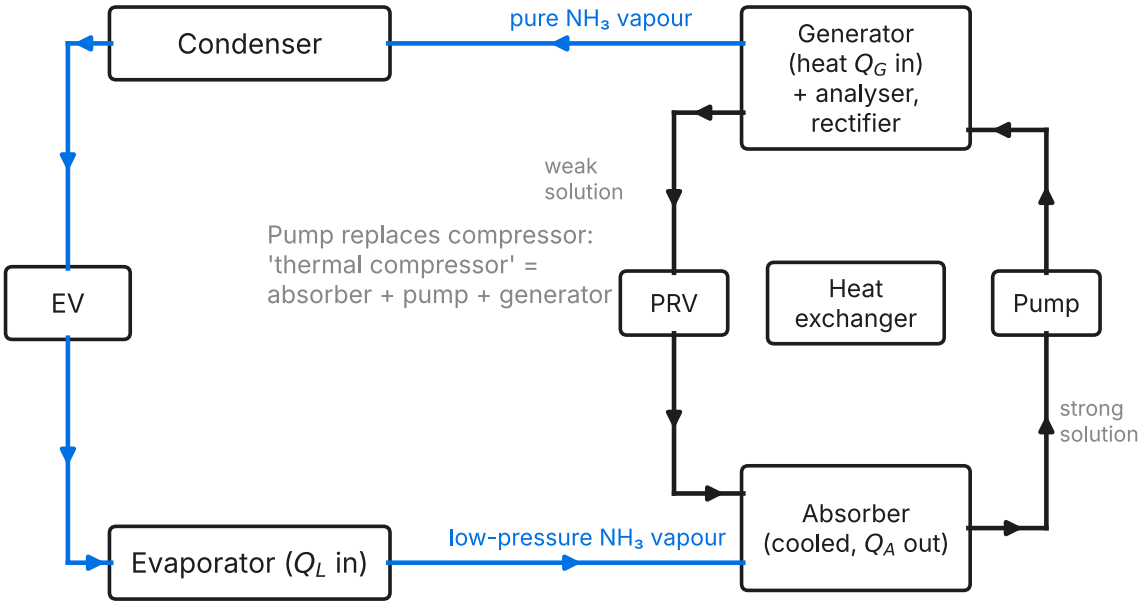

Fig 3.1 Practical NH_3–water vapour absorption system (Arora Ch 12).

Energy balance: $Q_G + Q_E + W_{pump} = Q_C + Q_A$

$$\text{COP}_{VAR} = \frac{Q_E}{Q_G + W_{pump}} \approx \frac{Q_E}{Q_G}$$

Advantages over VCR: uses low-grade heat or waste heat; very few moving parts (only a pump), so quiet with low maintenance and vibration; capacity control is easy; it handles part load well; and leaks are easily detected (NH_3). **Disadvantages:** low COP (0.5–0.8 single effect; about 1.2 double effect); bulky and costly; long start-up; and heat must be rejected from both the condenser and the absorber, so the cooling tower is larger.

3.6 Maximum COP of an absorption system (★ derivation)

Consider an ideal (reversible) system: heat Q_G is received at generator temperature T_G, heat Q_E is absorbed at T_E, and heat $Q_C + Q_A$ is rejected at ambient temperature T_0. Neglect pump work.

First law: $Q_G + Q_E = Q_0$. Second law, reversible: $\frac{Q_G}{T_G} + \frac{Q_E}{T_E} - \frac{Q_0}{T_0} = 0$

Eliminating Q_0: $Q_G\left(\frac{1}{T_0} - \frac{1}{T_G}\right) = Q_E\left(\frac{1}{T_E} - \frac{1}{T_0}\right)$

$$\boxed{\text{COP}_{max} = \frac{Q_E}{Q_G} = \underbrace{\frac{T_G - T_0}{T_G}}_{\text{Carnot engine } \eta} \times \underbrace{\frac{T_E}{T_0 - T_E}}_{\text{Carnot refrigerator COP}}}$$

So the ideal absorption system behaves like a **Carnot engine (between T_G and T_0) driving a Carnot refrigerator (between T_0 and T_E)**. The COP rises with higher T_G, higher T_E and lower T_0.

SOLVED

Worked example 3.1 · Generator at 100 °C, evaporator at −10 °C, ambient 30 °C. η_E = (373.15 − 303.15)/373.15 = 0.188; COP_R = 263.15/40 = 6.58 → **COP_max = 1.23**. Real single-effect NH_3–water plants reach about 0.4–0.6, roughly 40–50% of this ideal.

3.7 NH_3–water system (aqua-ammonia)

NH_3 is the **refrigerant** and water the **absorbent**. Because water is also volatile (the boiling points differ by only about 133 K), the vapour leaving the generator carries water vapour. That water would freeze in the expansion valve and raise the evaporator temperature. So the practical system adds:

1. **Analyser**: a column on top of the generator. Rising vapour meets the incoming cool strong solution, and part of the water condenses.
2. **Rectifier (dephlegmator)**: a water-cooled HX that condenses the remaining water vapour, which drains back to the generator. Nearly pure NH_3 goes to the condenser.
3. **Heat exchanger** between the weak solution (hot, generator to absorber) and the strong solution (cool, absorber to generator). It reduces generator heat and absorber cooling load, and raises the COP.
4. **Liquid sub-cooler** (liquid NH_3 / suction vapour), optional.

Strong solution = rich in NH_3 (leaving the absorber). Weak solution = poor in NH_3 (leaving the generator). Solution properties are read from the **enthalpy–concentration (h–ξ) chart**.

Features: suitable for **sub-zero** temperatures (down to −60 °C); used for cold storage and ice plants; the pressures are high (NH_3 condensing at about 13.5 bar); the system is toxic, so it is kept out of occupied spaces.

3.8 LiBr–water system and the three-fluid (Electrolux) system

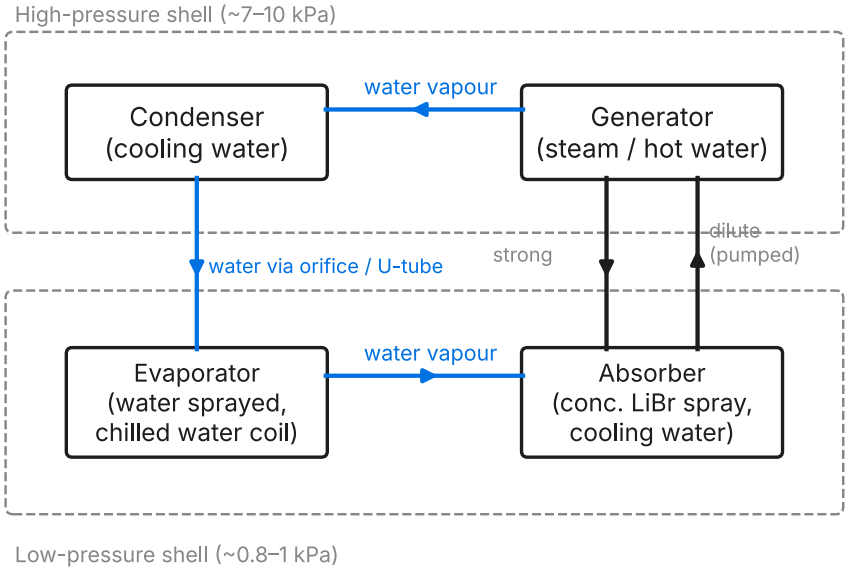

Fig 3.2 Two-shell LiBr–water absorption chiller (schematic).

LiBr–water: water is the refrigerant, LiBr solution the absorbent. LiBr is a salt and non-volatile, so **no analyser or rectifier** is needed.

- Water as refrigerant means the evaporator must work under a **high vacuum** (about 0.87 kPa for 5 °C) and **cannot go below about 0 °C** (in practice 4–10 °C chilled water). It is used for **air-conditioning chillers** (water chilling).

- Construction: one or two shells (HP: generator plus condenser; LP: evaporator plus absorber). Water is sprayed over the chilled-water tubes in the evaporator.
- **Crystallisation**: a concentrated LiBr solution can crystallise and block flow if cooled too much (for example when the cooling water is too cold). Controls prevent this.
- **Corrosive** in the presence of air, so inhibitors are added and the system is purged. Single-effect COP is about 0.7; **double-effect** (two generators, with steam at about 8 bar) gives about 1.0–1.2.

	NH_3–water	**LiBr–water**
Refrigerant / absorbent	NH_3 / water	Water / LiBr
Evaporator temperature	Down to −60 °C (refrigeration)	≥ 4–5 °C (air-conditioning only)
Pressures	High (several bar)	High vacuum
Analyser and rectifier	Required	Not required
COP (single effect)	0.5–0.6	0.7–0.8
Toxicity	Toxic (NH_3)	Non-toxic
Particular problem	Water carry-over	Crystallisation, corrosion, air leaks
Cooling of condenser and absorber	Air or water	Water (cooling tower)

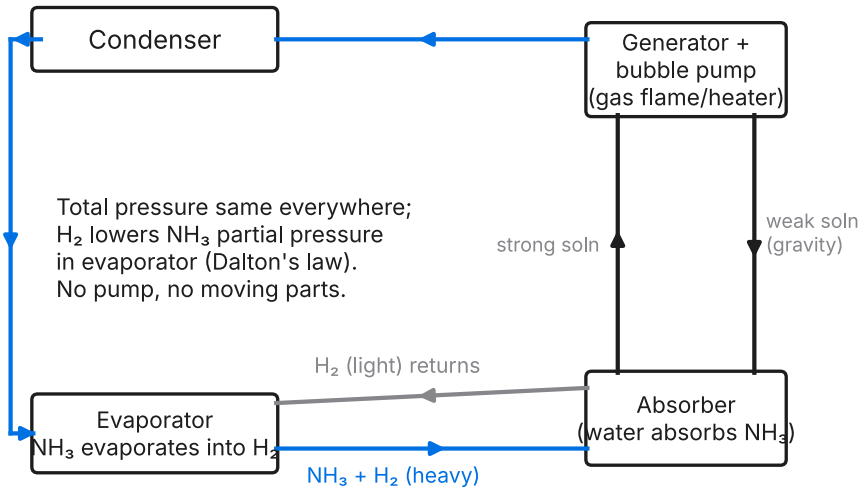

Fig 3.3 Three-fluid (Electrolux / Platen–Munters) absorption refrigerator.

Three-fluid (Electrolux) system: salient features (★)

- Fluids: **NH_3 (refrigerant)**, **water (absorbent)** and **hydrogen (inert gas, pressure equaliser)**.
- **Total pressure is the same throughout** the system, so **no pump and no expansion valve** are needed.
- In the evaporator, NH_3 evaporates into the H_2 atmosphere. By **Dalton's law**, its partial pressure (and so its saturation temperature) is low even though the total pressure is high.
- Circulation by **gravity and a thermosiphon "bubble pump"** in the generator: vapour bubbles lift the solution.
- In the absorber, water absorbs NH_3. The H_2 is light and rises back to the evaporator, while the NH_3 + H_2 mixture (heavier) sinks to the absorber, so the gas circulates naturally.
- Heat source: kerosene flame, LPG or an electric heater. **No moving parts: silent and long-lived.** Used in hotel-room minibars, caravans, and remote areas without electricity. COP is very low (about 0.1–0.2).

3.9 Steam-jet (ejector) refrigeration (Arora Ch 13)

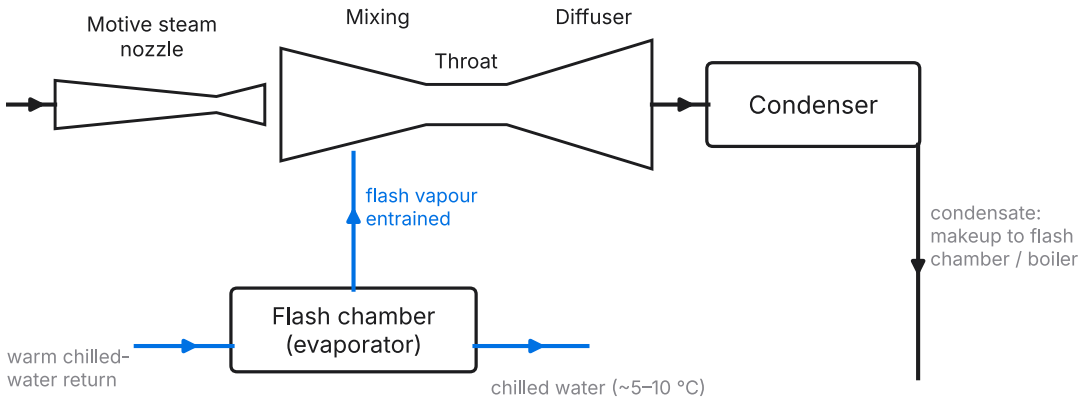

Fig 3.4 Steam-jet refrigeration system.

Principle: water boils at a low temperature if its pressure is low enough (1.23 kPa for 10 °C). A **steam ejector** creates and maintains this vacuum in a **flash chamber**. Part of the water flashes into vapour, taking its latent heat from the rest of the water, which is chilled. Water is the refrigerant, and the ejector replaces the compressor.

Basic components:

1. **Steam nozzle** (convergent–divergent): high-pressure motive steam expands to supersonic velocity and very low pressure.
2. **Mixing (suction) chamber**: the low pressure entrains the flash vapour from the evaporator.
3. **Diffuser**: the mixture's kinetic energy is converted to pressure, compressing it to condenser pressure.
4. **Flash chamber (evaporator)**: warm returning water is sprayed in. About 1% flashes, and the rest is chilled to about 5–10 °C.
5. **Condenser**: condenses the motive steam plus the flash vapour (it needs a lot of cooling water). Non-condensables are removed by secondary ejectors.

Analysis: flash fraction $x = \dfrac{c_w(t_{return} - t_{chilled})}{h_{fg}}$. Refrigeration $Q = \dot{m}_{flash}\, h_{fg}$. The **entrainment ratio** (kg of flash vapour per kg of motive steam) is about 0.3–0.5. COP = Q_E / heat supplied to raise the motive steam.

SOLVED

Worked example 3.2 · Water returns at 16 °C and is chilled to 10 °C in a flash chamber (h_{fg} at 10 °C = 2477 kJ/kg). Flash fraction = 4.187 × 6/2477 = **0.0101** (about 1%). Per TR: chilled water = 3.5167/(4.187 × 6) = 0.140 kg/s; flash vapour = 3.5167/2477 = **1.42 g/s**. The ejector must handle a huge volume (v_g ≈ 106 m³/kg at 10 °C), which is why ejectors and not reciprocating compressors are used.

Advantages: no moving parts, so low maintenance and vibration-free; water is a safe, cheap refrigerant; it uses waste or process steam; it handles very large vapour volumes. **Disadvantages:** low COP (high steam consumption); limited to about 4 °C and above (water freezes); a high vacuum must be kept; a large condenser and a lot of cooling water. **Applications:** chilled water for AC in industries with spare steam (chemical, rubber, food processing); concentrating fruit juice; vacuum cooling of vegetables.

3.10 Vortex tube (Ranque–Hilsch tube) (★)

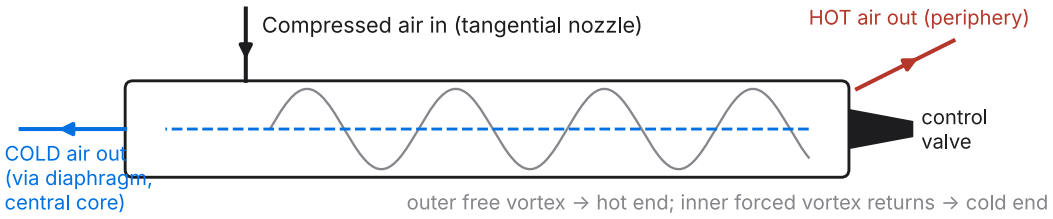

Fig 3.5 Counter-flow vortex tube.

Construction: a tube with a **tangential nozzle** near one end, a **diaphragm with a central orifice** at that end (cold exit), and a **throttle / control valve** at the far end (hot exit).

Working: compressed air (about 5–7 bar) enters tangentially and forms a high-speed **free vortex** spiralling along the wall toward the hot end. Part escapes through the valve as **hot air**. The rest is turned back and returns through the core as an inner **forced vortex** to the cold exit. Energy (angular momentum) passes from the inner core to the outer layer, so the core air leaves **cold** (a 30–50 K drop) and the periphery air leaves **hot**. The **cold fraction** $\mu_c = \dot{m}_c/\dot{m}$ is set by the hot-end valve.

Performance:

$$\mathrm{RE} = \mu_c\, c_p(T_i - T_c), \qquad \mathrm{COP} = \frac{\mu_c c_p(T_i - T_c)}{RT_i \ln(p_i/p_a)}\ \text{(isothermal compression work)}$$

Energy balance: $T_i = \mu_c T_c + (1-\mu_c)T_h$. Maximum refrigeration occurs at a cold fraction of about 0.6–0.7; the largest temperature drop occurs at about 0.2–0.3.

SOLVED

Worked example 3.3 · Air at 300 K, 6 bar enters a vortex tube. The cold fraction is 0.3, the cold air leaves at 263 K, and the ambient is 1 bar. RE = 0.3 × 1.005 × 37 = 11.2 kJ/kg of supply air; isothermal work = 0.287 × 300 × ln 6 = 154.3 kJ/kg → COP = **0.072**. Hot-end temperature = (300 − 0.3 × 263)/0.7 = **315.9 K**.

Advantages: no moving parts and no refrigerant; light, cheap and instant; reliable. **Disadvantages:** very low COP; noisy; needs a supply of compressed air. **Applications:** spot cooling of cutting tools and electronic cabinets, cooling suits for miners and divers, and cooling of hot machinery.

EXAM TIP

Unit 3 must-knows: properties list (classify as thermodynamic, chemical, physical) · decoding R-numbers both ways · ODP, GWP and the protocols · absorption principle with a sketch · **COP_max derivation** · NH_3–water with the role of analyser and rectifier · NH_3–water vs LiBr–water table · Electrolux features (H_2, Dalton's law) · steam-jet sketch and components · vortex tube sketch and working.

Unit 4: Air-Conditioning and Psychrometry

From the book (Arora 3e). **Ch 14** Properties of Moist Air (Gibbs–Dalton law, specific humidity, relative humidity, degree of saturation, dew point, thermodynamic WBT and adiabatic saturation, psychrometric chart) · **Ch 15** Psychrometry of Air-Conditioning Processes (mixing, sensible heating and cooling, humidification, dehumidification, SHF, bypass factor, air washer, summer and winter AC) · **Ch 19** Load Calculations and Applied Psychrometrics (RSHF, GSHF, ESHF, ADP, dehumidified air quantity) · **Ch 20** Design of AC Apparatus (cooling and dehumidifying coils, air washers).

4.1 Moist air and the basic laws

Atmospheric air is a mixture of **dry air** (N_2, O_2, Ar, CO_2) and **water vapour**. Both are treated as ideal gases, and the **Gibbs–Dalton law** applies: $p = p_a + p_v$ (total = partial pressure of dry air + partial pressure of vapour).

Standard atmospheric pressure is p = 101.325 kPa. For dry air R_a = 0.287 kJ/kg K; for vapour R_v = 0.4615 kJ/kg K.

4.2 Psychrometric properties (★ definitions)

Property	Definition / formula
Dry-bulb temperature (DBT), t	Temperature measured by an ordinary thermometer
Wet-bulb temperature (WBT), t'	Temperature shown by a thermometer whose bulb is covered with a wet wick, in moving air (> 2–5 m/s). Equals the thermodynamic WBT (adiabatic saturation temperature) for air–water
Dew-point temperature (DPT), t_d	Temperature at which vapour starts condensing when air is cooled at constant W: $t_d = t_{sat}(p_v)$
Wet-bulb depression	$t - t'$ (zero for saturated air)
Dew-point depression	$t - t_d$
Specific humidity (humidity ratio), W	kg of vapour per kg of dry air: $W = 0.622\,\dfrac{p_v}{p - p_v}$
Relative humidity, ϕ	$\phi = \dfrac{p_v}{p_{vs}}$, the ratio of the vapour's partial pressure to the saturation pressure at the same DBT
Degree of saturation, μ	$\mu = \dfrac{W}{W_s} = \phi\,\dfrac{p - p_{vs}}{p - p_v}$ (slightly less than ϕ)
Enthalpy of moist air, h	$h = 1.005\,t + W(2500 + 1.88\,t)$ kJ/kg dry air
Humid specific heat	$c_{pm} = 1.005 + 1.88\,W \approx 1.0216$ kJ/kg K
Specific volume	$v = \dfrac{R_a T}{p - p_v}$ m³/kg dry air
Vapour density (absolute humidity)	$\rho_v = W/v$

Partial pressure from a wet-bulb reading: Carrier's equation (Arora Ch 14 gives an equivalent form):

$$p_v = p'_{vs} - \frac{(p - p'_{vs})(t - t')}{1544 - 1.44\,t'}$$

where p'_{vs} is the saturation pressure at the WBT t' (all pressures in the same unit, temperatures in °C).

Adiabatic saturation: unsaturated air passes through a long, insulated chamber with water recirculating at t'. It leaves saturated at t'. The energy balance $h_1 + (W_2 - W_1)h'_f = h_2$ shows that **lines of constant WBT are nearly lines of constant enthalpy** on the chart.

Psychrometric chart: DBT on the x-axis and W on the y-axis, with curves of constant φ, inclined lines of constant WBT ≈ constant h, constant specific volume lines, and the saturation curve (φ = 100%), where DBT = WBT = DPT.

SOLVED

Worked example 4.1 · Air at 35 °C DBT and 25 °C WBT, at 1.01325 bar. Find p_v, W, φ, μ, DPT, h and v. From tables: p_{vs}(25 °C) = 3.170 kPa and p_{vs}(35 °C) = 5.629 kPa. p_v = 3.170 − (101.325 − 3.170)(10)/(1544 − 36) = **2.519 kPa** W = 0.622 × 2.519/(101.325 − 2.519) = **0.01586 kg/kg d.a.** φ = 2.519/5.629 = **44.8%**; μ = 0.01586/0.03656 = **0.433** DPT = t_{sat}(2.519 kPa) = **21.2 °C** h = 1.005(35) + 0.01586(2500 + 65.8) = **75.9 kJ/kg d.a.** v = 0.287 × 308.15/(101.325 − 2.519) = **0.895 m³/kg d.a.**

4.3 Psychrometric processes (★)

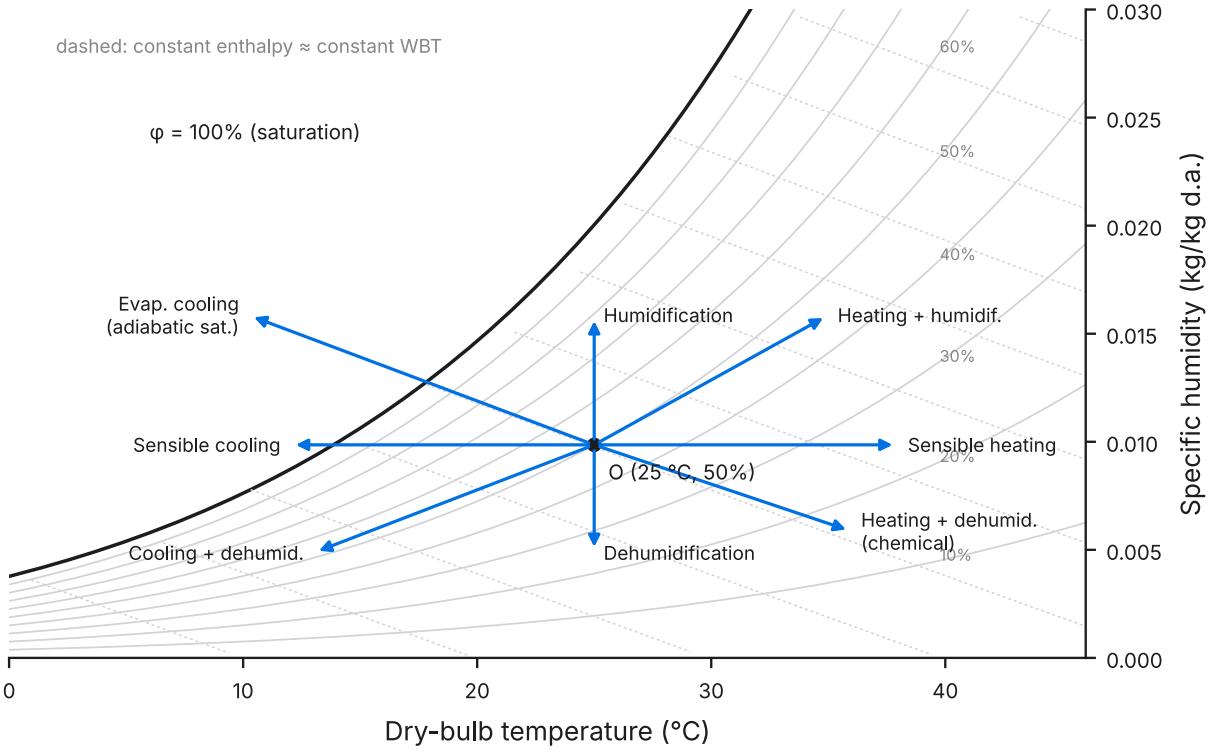

Fig 4.1 Basic psychrometric processes from a common state O (chart computed at 101.325 kPa).

Process	Path on chart	What changes	Heat / mass
Sensible heating	Horizontal →	DBT ↑, W and DPT constant, φ ↓	$Q = \dot{m}_a c_{pm}(t_2 - t_1) = \dot{m}_a(h_2 - h_1)$
Sensible cooling	Horizontal ← (coil surface temperature > DPT)	DBT ↓, W constant, φ ↑	same
Humidification (steam or water at DBT)	Vertical ↑	W ↑	$\dot{m}_w = \dot{m}_a(W_2 - W_1)$
Cooling and dehumidification	Down-left towards the ADP	DBT ↓, W ↓	$Q = \dot{m}_a[(h_1 - h_2) - (W_1 - W_2)h_f]$
Heating and humidification	Up-right	DBT ↑, W ↑	Winter AC

Process	Path on chart	What changes	Heat / mass
Chemical dehumidification (adsorbent: silica gel, alumina)	Down-right, roughly constant h	W ↓, DBT ↑	Latent heat released
Adiabatic (evaporative) cooling	Along constant WBT	DBT ↓, W ↑, $h \approx$ constant	Desert cooler, air washer with recirculated water

Sensible heat factor: $\mathrm{SHF} = \dfrac{\mathrm{SH}}{\mathrm{SH} + \mathrm{LH}}$. SH is the horizontal component on the chart, $\dot{m}_a c_{pm} \Delta t$; LH is the vertical component, $\dot{m}_a h_{fg} \Delta W$.

Handy forms with volume flow (standard air, ρ = 1.2 kg/m³, Q in m³/min, i.e. cmm):

$$\mathrm{SH} = 0.0204 \,\mathrm{cmm}\, \Delta t \ \mathrm{kW}, \qquad \mathrm{LH} = 50 \,\mathrm{cmm}\, \Delta W \ \mathrm{kW}, \qquad \mathrm{TH} = 0.02 \,\mathrm{cmm}\, \Delta h \ \mathrm{kW}$$

Humidifying (saturation) efficiency of an air washer: $\eta_H = \dfrac{t_1 - t_2}{t_1 - t_1'}$ (for evaporative cooling), or the equivalent in terms of W.

Adiabatic mixing of two streams:

$$W_3 = \frac{m_1 W_1 + m_2 W_2}{m_1 + m_2}, \qquad h_3 = \frac{m_1 h_1 + m_2 h_2}{m_1 + m_2}, \qquad t_3 \approx \frac{m_1 t_1 + m_2 t_2}{m_1 + m_2}$$

The mixture state lies on the straight line joining 1 and 2, dividing it in the inverse ratio of the masses. Use **masses**, not volumes, for accuracy.

Cooling coil: bypass factor and ADP

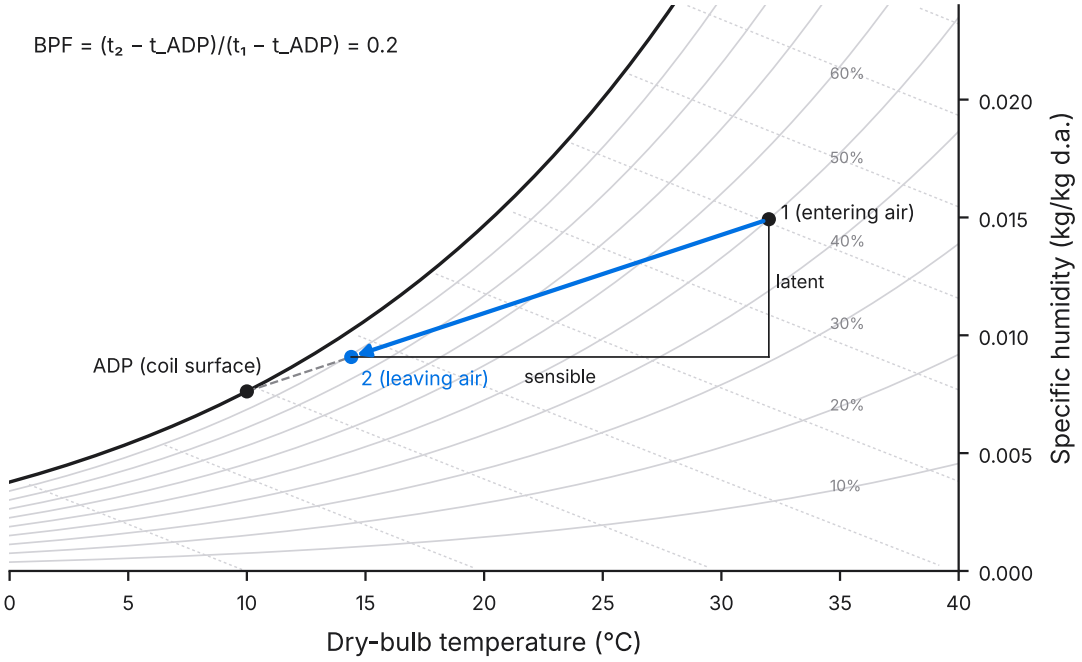

Fig 4.2 Cooling and dehumidifying coil: entering air 1, leaving air 2, apparatus dew point (ADP).

- **ADP (apparatus dew point)**: the effective coil surface temperature, where the line from the entering state through the leaving state meets the saturation curve.
- **Bypass factor (BPF)**: the fraction of air that effectively passes the coil untouched:

$$\mathrm{BPF} = \frac{t_2 - t_{ADP}}{t_1 - t_{ADP}} = \frac{W_2 - W_{ADP}}{W_1 - W_{ADP}} = \frac{h_2 - h_{ADP}}{h_1 - h_{ADP}}, \qquad \text{contact factor} = 1 - \mathrm{BPF}$$

- BPF **falls** with more rows of tubes, closer fin spacing and lower air velocity. Typical values: 0.05–0.3.
- For a heating coil, the same definition applies with the coil surface temperature in place of the ADP.

SOLVED

Worked example 4.2 · Air at 35 °C, 50% RH passes over a cooling coil with ADP 10 °C and BPF 0.15. Find the leaving state, the cooling load per kg of dry air, the SHF and the condensate. W_1 = 0.01777, h_1 = 80.8 kJ/kg; W_{ADP} = 0.00763 t_2 = 10 + 0.15(25) = **13.75 °C**; W_2 = 0.00763 + 0.15(0.01014) = **0.00915**; h_2 = **36.9 kJ/kg**; $\varphi_2 \approx$ 93% Load = 80.8 − 36.9 = **43.8 kJ/kg d.a.** (43.3 after subtracting condensate enthalpy) Sensible = 22.1, latent = 21.8 → SHF = **0.50**; condensate = **0.0086 kg/kg d.a.**

SOLVED

Worked example 4.3 (mixing) · 1 kg/s of outdoor air (35 °C, W = 0.020) mixes with 3 kg/s of return air (25 °C, W = 0.0099). W_3 = (0.020 + 3 × 0.0099)/4 = **0.01243**; h_3 = 59.4 kJ/kg → t_3 = **27.5 °C**

4.4 Summer and winter air-conditioning systems

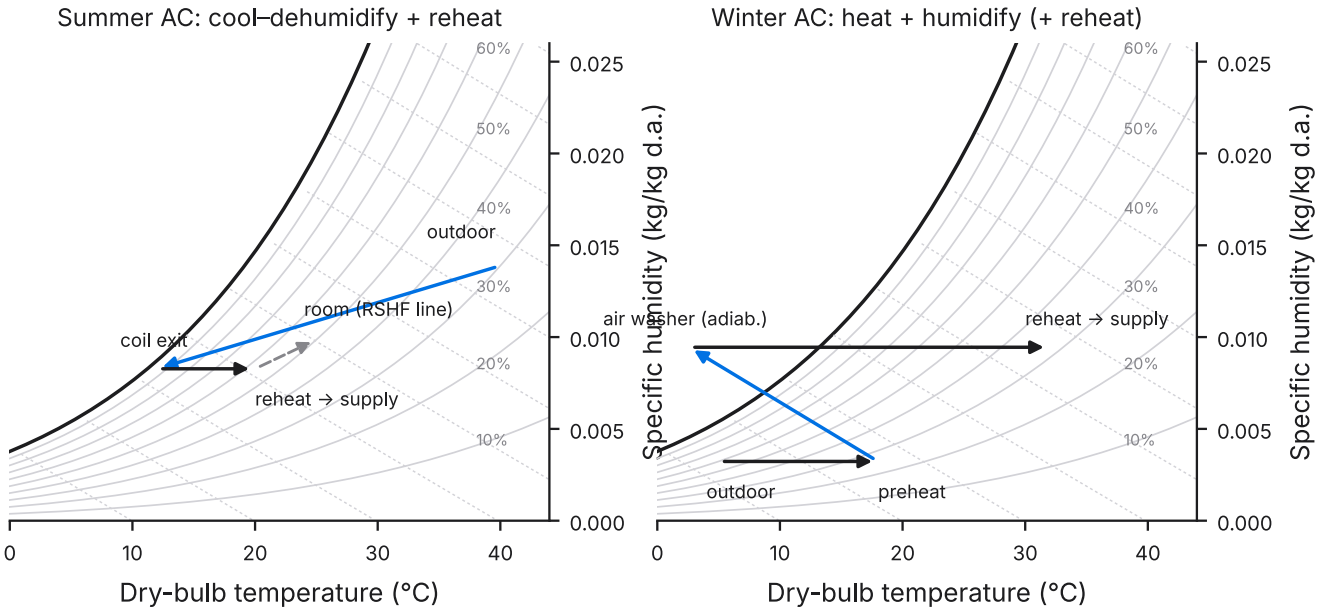

Fig 4.3 Typical summer (left) and winter (right) AC processes on the chart.

Summer AC (hot and humid climate): outdoor air (with or without recirculated air) → **cooling and dehumidifying coil** (cooled below the DPT, so moisture condenses) → (optional) **reheater** to raise the DBT and lower the RH to the supply state → supply fan → room. Reheat is used when the room SHF is low (high latent load) and the coil line cannot reach the required state directly. **For a hot and dry climate:** evaporative cooling (air washer) may be enough.

Components: filter, cooling coil (DX or chilled water), eliminator plates, reheater, humidistat and thermostat, fan, ducts, and the refrigeration plant.

Winter AC: outdoor air (cold, low W) → filter → **preheater** (to avoid freezing in the washer) → **humidifier / air washer** (adiabatic, or with heated spray) → **reheater** to the supply DBT → room. Alternatively: heating coil plus steam injection (a vertical line on the chart).

SOLVED

Worked example 4.4 (winter) · Air at 5 °C, 60% RH is to be brought to 22 °C, 50% RH using a heater and steam humidification. Per kg of dry air: W_1 = 0.00323, W_2 = 0.00823, h_1 = 13.1, h_2 = 43.0 kJ/kg Total heat = **29.9 kJ/kg** (sensible part 17.2); water added = **5.0 g/kg d.a.**

Year-round AC: combines both arrangements, with controls (thermostat and humidistat) switching coils and humidifiers.

4.5 Applied psychrometry: load and supply-air calculations (★ Arora Ch 19)

Sensible heat factors:

- **RSHF** (room) $= \dfrac{\text{RSH}}{\text{RSH} + \text{RLH}}$. The supply air must lie on the RSHF line through the room state.
- **GSHF** (grand) $= \dfrac{\text{TSH}}{\text{GTH}}$, where TSH = RSH + OASH and GTH = RSH + RLH + OASH + OALH.
- **ESHF** (effective) $= \dfrac{\text{ERSH}}{\text{ERSH} + \text{ERLH}}$, with

$$\text{ERSH} = \text{RSH} + \text{BPF} \cdot \text{OASH}, \qquad \text{ERLH} = \text{RLH} + \text{BPF} \cdot \text{OALH}$$

(the bypassed outdoor air acts like an extra room load).

Outdoor air loads: $\text{OASH} = 0.0204\,\text{cmm}_o(t_o - t_i)$, $\text{OALH} = 50\,\text{cmm}_o(W_o - W_i)$.

ADP: the ESHF line drawn through the room state meets the saturation curve at the ADP. Analytically, find t_{ADP} such that

$$\frac{0.0204(t_i - t_{ADP})}{0.0204(t_i - t_{ADP}) + 50(W_i - W_{ADP})} = \text{ESHF}$$

Dehumidified (supply) air quantity:

$$\text{cmm}_d = \frac{\text{ERSH}}{0.0204\,(1 - \text{BPF})(t_i - t_{ADP})}$$

Coil leaving state: $t_{leave} = t_{ADP} + \text{BPF}(t_{mix} - t_{ADP})$. **Refrigeration capacity** = GTH / 3.5167 TR.

SOLVED

Worked example 4.5 (summer AC load, ★ classic) · A room is kept at 25 °C DBT, 50% RH. Outdoor: 43 °C DBT, 27 °C WBT. RSH = 25 kW, RLH = 6 kW. Fresh air = 40 cmm (m³/min). Coil BPF = 0.15. Find ESHF, ADP, supply air quantity, the state of air off the coil, and the plant capacity.

W_i = 0.00988; W_o = 0.01592 (Carrier's equation) OASH = 0.0204 × 40 × 18 = **14.69 kW**; OALH = 50 × 40 × 0.00604 = **12.07 kW** ERSH = 25 + 0.15 × 14.69 = **27.20 kW**; ERLH = 6 + 0.15 × 12.07 = **7.81 kW** ESHF = 27.20/35.01 = **0.777** (RSHF = 0.806) ADP (from chart or by trial) = **11.2 °C** cmm_d = 27.20/[0.0204 × 0.85 × (25 − 11.2)] = **113.5 m³/min** (recirculated air = 73.5 m³/min) Mixed air: t_m = 31.3 °C, W_m = 0.0120 → off-coil: t = 11.2 + 0.15(20.1) = **14.2 °C**, W = 0.00883 Check: 0.0204 × 113.5 × (25 − 14.2) = 25.0 kW = RSH ✓ GTH = 25 + 6 + 14.69 + 12.07 = **57.8 kW = 16.4 TR**

EXAM TIP

Unit 4 must-knows: all definitions and the W, φ, μ, h formulas · Carrier's equation · the process table with sketches · BPF and ADP · mixing · summer and winter system sketches · the full ERSH → ADP → cmm → TR procedure. In the exam, **show the chart sketch even if you compute analytically**.

Unit 5: Human Comfort and Air-Conditioning Loads

From the book (Arora 3e). **Ch 16** Design Conditions (thermal exchange of the body with its environment, factors affecting comfort, effective temperature, comfort chart, inside and outside design conditions, indoor air quality) · **Ch 24** Food Refrigeration and Industrial Air Conditioning (requirements of industrial AC for different industries) · **Ch 17** Solar Radiation and **Ch 18** Heat Transfer through Building Structures (inputs for load calculation) · **Ch 19** Load Calculations (internal and external loads, cooling load estimate).

5.1 Human body heat balance

The body is like a heat engine. It produces heat by **metabolism** (food oxidation) and must reject it to keep its core temperature at about **37 °C**:

$$M - W = Q_{conv} \pm Q_{rad} + Q_{evap} + Q_{resp} + S$$

- M = metabolic rate (about 100 W seated, 120 W light office work, up to 400–500 W heavy work). 1 met = 58.2 W/m² of body surface.
- W = useful mechanical work (usually small).
- Q_{conv}, Q_{rad} = **sensible** heat loss (depends on air temperature, air velocity and mean radiant temperature).
- Q_{evap} = **latent** heat loss through perspiration and skin diffusion (depends on humidity and air velocity).
- Q_{resp} = respiration (sensible and latent).
- S = heat storage. $S > 0$ means the body heats up (discomfort, heat stroke); $S < 0$ means it cools.

Comfort = a heat balance with S = 0 achieved **without stress** on the body's regulating mechanisms (shivering, sweating, blood-flow control).

Body regulatory mechanisms: in hot conditions, vasodilation and sweating; in cold, vasoconstriction and shivering.

5.2 Factors affecting human comfort (★)

Group	Factors
Environmental	(1) **Dry-bulb temperature** (convection); (2) **humidity** (evaporation of sweat; best 30–60% RH); (3) **air velocity** (6–9 m/min, i.e. 0.1–0.15 m/s, is ideal; above 15 m/min feels draughty); (4) **mean radiant temperature** of the surroundings (cold glass, hot roof)
Personal	(5) **Activity level** (metabolic rate); (6) **clothing** (insulation in *clo*, 1 clo = 0.155 m² K/W); also age, sex, state of health, and acclimatisation (people in hot countries prefer warmer conditions)
Air quality	Purity (dust, odours, CO_2, bacteria); ventilation rate

5.3 Effective temperature and the comfort chart (★)

Effective temperature (ET): an index that **combines DBT, humidity and air movement** into one number. It is defined as *the temperature of saturated air (100% RH), moving at very low velocity (about 6–7 m/min), that gives the same sensation of warmth or cold as the actual environment.* Example: air at 30 °C DBT, 40% RH and low velocity may feel the same as still, saturated air at 25 °C, so its ET is 25 °C.

Determined experimentally (ASHRAE/Yaglou), by subjective votes of many people moving between test chambers. It is read from the **ET chart / comfort chart**.

Limitations of ET: it overestimates the effect of humidity at low temperatures and underestimates it at high temperatures; it does not account properly for radiation (corrected by "CET", using the globe temperature); and it ignores clothing and activity. The **New Effective Temperature (ET*)** used in ASHRAE charts is defined at **50% RH** instead of 100%, and accounts for clothing and activity. Fanger's PMV–PPD method is the modern refinement.

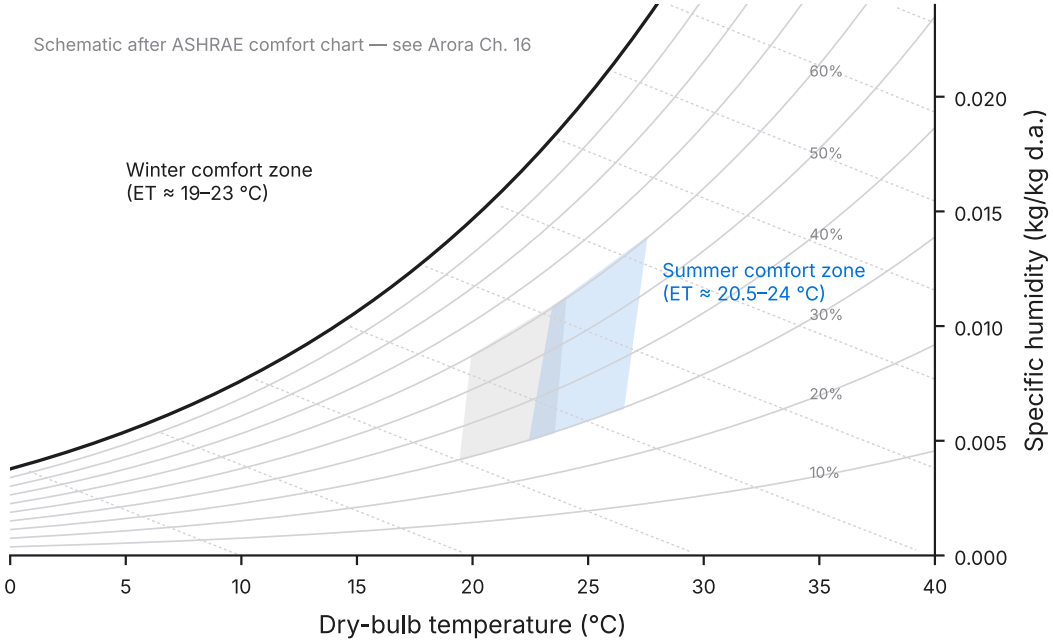

Fig 5.1 Comfort zones on the psychrometric chart (schematic after the ASHRAE comfort chart; see Arora Ch 16 for the actual chart).

Comfort chart (ASHRAE): a psychrometric-type chart with DBT on the x-axis and WBT/humidity lines, with lines of constant ET superimposed and **comfort zones for summer and winter** marked, showing the percentage of people who feel comfortable.

- The comfort zone is bounded by about **30–60% (70%) RH** and ET **≈ 20–24 °C** in summer and **≈ 19–23 °C** in winter.
- The **optimum ET** in summer is about **21.7 °C** (98% of people comfortable) in the classic ASHRAE chart. Indian conditions (acclimatisation) allow a slightly higher figure.
- Summer comfort needs a higher ET than winter because of lighter clothing and acclimatisation to the hot outdoors.
- The chart applies to sedentary or light activity with air velocity of 5–8 m/min. Corrections are needed for heavy activity or higher velocity.

Typical inside design conditions (comfort): **summer 24–26 °C DBT, 50 ± 5% RH**; **winter 20–22 °C DBT, 40–50% RH**. The indoor–outdoor DBT difference in summer should not be too large (≤ 8–10 K) to avoid "thermal shock" when people enter or leave.

Outside design conditions: chosen from local weather data (for example, the DBT and coincident WBT exceeded only 1–2.5% of hours in summer). Typical Indian cities: 40–45 °C DBT, 24–28 °C WBT.

5.4 Comfort air-conditioning

Comfort AC provides conditions for human comfort in homes, offices, hospitals, theatres, auditoriums, restaurants and vehicles. It needs:

1. Control of DBT and RH within the comfort zone.
2. Air distribution with velocities of 6–9 m/min in the occupied zone, and no draughts.

3. **Ventilation** (fresh air) to dilute CO_2, odours and smoke: about **0.28–0.57 cmm per person** (5–10 L/s per person). More in smoking or crowded spaces such as auditoriums and hospitals.
4. Filtering of dust and pollen, and acceptable noise levels.

Types: room (window/split) AC, packaged units, and central systems (all-air, air–water, all-water: fan-coil units).

5.5 Requirements of industrial air-conditioning (★)

Industrial AC controls conditions for a **product or process**, sometimes at conditions outside the human comfort range. Requirements include:

Industry / application	Typical requirement	Reason
Textile (spinning, weaving)	24–27 °C, **RH 50–85%** depending on process (weaving needs a high RH)	Fibre strength and elasticity; avoiding static electricity, thread breakage and fluff
Printing	24–27 °C, **45–50% RH** constant	Paper dimensions stay stable, so colours register correctly; ink drying
Pharmaceuticals	21–24 °C, **low RH (≤ 35%)** for hygroscopic tablets; clean rooms	Prevents moisture absorption; sterility (HEPA filters)
Photographic film	20–24 °C, 40–60% RH, dust-free	Emulsion quality
Precision machining, gauge rooms	**20 ± 0.5 °C**, 40–45% RH	Dimensional accuracy (thermal expansion)
Electronics, semiconductor clean rooms	20–22 °C, 40–50% RH, Class 100 / ISO 5 cleanliness	Contamination and static control
Computer rooms / data centres	20–24 °C, 40–55% RH, high sensible load (SHF ≈ 0.95+)	Equipment reliability, static control
Hospital operation theatres	20–24 °C, 50–60% RH, 100% fresh air, positive pressure	Infection control, anaesthetic gases
Food processing / storage	Product-specific (e.g. cold storage of apples at 0–2 °C, 90% RH)	Shelf life and weight loss
Laboratories	Close control	Repeatability of tests

Comfort vs industrial AC: comfort AC targets people (a broad tolerance band, ventilation for occupants), while industrial AC targets the product or process (often tight tolerances of ±0.5 K and ±2% RH, special filtering, sometimes 100% fresh air, round-the-clock running). Industrial AC must often also provide comfort for the workers.

5.6 Air-conditioning load calculation (★)

The cooling load is the rate at which heat must be removed to keep the space at design conditions. Arora divides it into **sensible** and **latent** loads, and into **external** and **internal** loads.

A. External loads

1. **Solar heat gain through glass**: $Q = A \times \mathrm{SHGF} \times \mathrm{SC} \times \mathrm{CLF}$ (SHGF = solar heat gain factor, SC = shading coefficient, CLF = cooling load factor). Usually the largest single item (Ch 17).
2. **Conduction through glass**: $Q = UA(t_o - t_i)$.
3. **Walls and roof**: $Q = UA \times \mathrm{CLTD}$ (or the *equivalent temperature difference*, which accounts for solar radiation and thermal storage of the wall: the time lag and decrement) (Ch 18).
4. **Partitions, floors, ceilings** next to unconditioned spaces: $UA\Delta t$.

5. **Infiltration** (air leakage through cracks and doors), by the air-change method: cmm = ACH × Volume / 60, giving sensible $0.0204\,\text{cmm}\,\Delta t$ and latent $50\,\text{cmm}\,\Delta W$.
6. **Ventilation (outdoor air)** sensible and latent: added to the coil load (and with the BPF fraction to the effective room load), as in Unit 4.

B. Internal loads

1. **Occupants**: sensible and latent depending on activity (e.g. seated office work about 75 W sensible + 55 W latent; dancing about 90 + 160 W).
2. **Lighting**: wattage × use factor × allowance factor (1.25 for fluorescent tubes, accounting for ballast loss).
3. **Appliances and equipment**: motors (input power, or output/η), computers, cooking (sensible and latent).
4. **Product load** (in industrial or food plants).

C. System heat gains: supply-duct heat gain and leakage, fan motor heat (about 2.5–10%), and a safety factor (about 5–10%).

$$\text{RSH} = \sum \text{sensible room gains} \times (1 + \text{SF}), \quad \text{RLH} = \sum \text{latent room gains} \times (1 + \text{SF})$$

$$\text{GTH} = \text{RSH} + \text{RLH} + \text{OASH} + \text{OALH} \quad \Rightarrow \quad \text{TR} = \text{GTH}/3.5167$$

Heat gain vs cooling load: instantaneous heat gain is not the same as cooling load. Radiant heat is first absorbed by the structure and furniture and released later (the **storage effect**), so the peak cooling load is lower than, and later than, the peak heat gain. Arora handles this with CLF and CLTD.

SOLVED

Worked example 5.1 (office load estimate) · Office 10 m × 8 m × 3 m; inside 25 °C, 50% RH (W_i = 0.00988); outside 43 °C DBT, 27 °C WBT (W_o = 0.01592). Data: 20 occupants (75 W S, 55 W L each); fluorescent lights 15 W/m²; equipment 2 kW; exposed wall net 30 m², U = 1.8 W/m² K, CLTD 12 K; glass 8 m², U = 5.8, solar gain 250 W/m²; roof 80 m², U = 1.2, CLTD 25 K; infiltration 0.5 air changes per hour; ventilation 0.6 cmm per person; safety factor 5%.

Item	Sensible (W)	Latent (W)
Occupants	20 × 75 = 1500	20 × 55 = 1100
Lights: 15 × 80 × 1.25	1500	–
Equipment	2000	–
Wall: 1.8 × 30 × 12	648	–
Glass conduction: 5.8 × 8 × 18	835	–
Glass solar: 8 × 250	2000	–
Roof: 1.2 × 80 × 25	2400	–
Infiltration: 0.5 × 240/60 = 2 cmm	0.0204 × 2 × 18 = 0.734 kW → 734	50 × 2 × 0.00604 = 0.604 kW → 604
Sub-total	11 617	1704
With 5% SF: RSH, RLH	**12 198**	**1789**
Ventilation 12 cmm: OASH, OALH	4406	3622

RSHF = 12 198/13 987 = **0.87**; GTH = 12.2 + 1.79 + 4.41 + 3.62 = **22.0 kW ≈ 6.3 TR**

EXAM TIP

Unit 5 must-knows: body heat balance equation · comfort factors (environmental and personal) · ET definition, how it is found, and its limits · comfort chart sketch with zones · inside design conditions · comfort vs industrial AC, plus the industry table (textile, printing, pharma, precision) · list of load components with formulas · one load table like worked example 5.1.

Unit 6: Evaporators and Condensers

From the book (Arora 3e). **Ch 7** Condensers (heat rejection ratio, types: air-cooled, water-cooled, evaporative; heat transfer in condensers; fouling) · **Ch 9** Evaporators (types: flooded, DX, natural and forced convection; frosting and defrosting; heat transfer) · **Ch 10** Complete Vapour Compression System (how condenser and evaporator size affect system balance and performance).

6.1 Condensers

Function: to reject the heat absorbed in the evaporator **plus** the compressor work to the sink (air or water), in three zones: **desuperheating → condensing → sub-cooling**.

$$Q_c = \dot{m}(h_2 - h_3) = Q_e + W, \qquad \text{Heat rejection ratio (HRR)} = \frac{Q_c}{Q_e} = 1 + \frac{1}{\text{COP}}$$

HRR is about 1.15–1.3 for AC applications, and higher for low-temperature (low-COP) and hermetic systems (motor heat goes into the refrigerant).

Classification and comparison (★):

Type	Construction	Merits	Demerits / use
Air-cooled: natural convection	Wire-and-tube or plate on the back of domestic refrigerators	No fan, silent, cheap	Low U (≈ 5–10 W/m² K); small capacity only
Air-cooled: forced convection	Finned copper/aluminium tube coils with a fan	No water needed, easy installation, low maintenance	Higher condensing temperature (ambient + 10–15 K), so lower COP; noisy; used up to ~20 TR, window/split AC
Water-cooled: double tube (tube-in-tube)	Water in the inner tube, refrigerant in the annulus, counterflow	Simple, good counterflow ΔT	Hard to clean; small capacity
Water-cooled: shell and coil	Water coil inside a welded shell	Compact, cheap	Cleaned chemically only; medium capacity
Water-cooled: shell and tube	Refrigerant condenses on the shell side over horizontal water tubes (multi-pass)	**Large capacity**, high U (700–1200 W/m² K), mechanically cleanable, the shell acts as a receiver	Needs a cooling tower and water treatment; used in large plants and chillers
Evaporative	Refrigerant coil sprayed with recirculated water while a fan blows air through. Heat leaves mainly by evaporating the water.	Condensing temperature close to the **ambient WBT**; uses only about 5% of the water of a once-through condenser; combines condenser and cooling tower	Scaling and corrosion; needs a water supply; used in large NH_3 plants and dry climates

Heat transfer in condensers:

$$Q = UA\,\Delta T_{lm}, \qquad \frac{1}{U_o} = \frac{1}{h_o} + R_{f,o} + \frac{xA_o}{kA_m} + \left(R_{f,i} + \frac{1}{h_i}\right)\frac{A_o}{A_i}$$

- **Film condensation** coefficient (Nusselt) for horizontal tubes: $\bar{h} = 0.725\left[\dfrac{k^3\rho^2 g h_{fg}}{\mu D \Delta T}\right]^{1/4}$. More tubes in a vertical row lower the coefficient (inundation, $N^{-1/4}$).
- **Fouling (scale) factors** reduce U, so the condensing temperature rises. Cleaning is therefore important.

- **Non-condensable gases** (air) collect in the condenser, raise the total pressure and blanket the tubes. They are removed by **purging**.
- **Wilson plot**: an experimental method to separate the water-side and refrigerant-side coefficients.

6.2 Evaporators

Function: to absorb heat from the space or product; this is where refrigeration happens. The refrigerant enters as a low-quality wet mixture and leaves dry or slightly superheated.

Classification:

1. **By the method of feeding refrigerant:**
   - **Dry expansion (DX):** the expansion valve meters just enough refrigerant to evaporate completely, with superheat at the exit. Little refrigerant charge and good oil return, but part of the surface is used for superheating, so the surface is used less efficiently.
   - **Flooded:** a float valve keeps the evaporator full of liquid; vapour is separated in a surge drum and liquid recirculates. **High heat-transfer coefficient** and fully wetted surface, but a large refrigerant charge and oil-return problems. Used in large NH_3 plants and chillers.
   - **Liquid overfeed** (pump-recirculated): the pump supplies 3–4 times the evaporation rate; best heat transfer.
2. **By construction: bare tube** (pipe coil); **finned tube** (air coolers); **plate type** (domestic freezers, roll-bond); **shell and tube** (flooded or DX, for liquid chilling); **shell and coil**; **double tube**; **Baudelot** (water flows as a film over vertical plates; milk and water cooling near 0 °C); **brazed plate HX**.
3. **By the fluid cooled:** air coolers (natural or forced convection: fan-coil units, unit coolers) and **liquid chillers** (water, brine).
4. **By the mode of air flow:** natural convection (cold rooms, domestic) and forced convection.

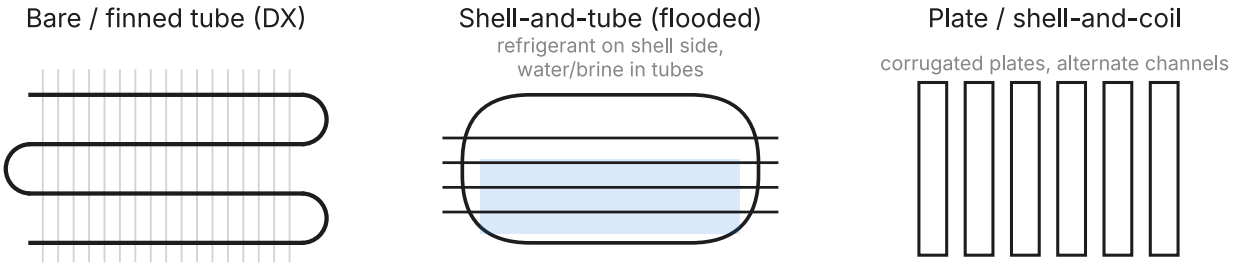

Fig 6.1 Common evaporator constructions (schematic).

Frosting and defrosting (air coolers below 0 °C): frost insulates the coil and blocks air flow, so both capacity and evaporator temperature fall. **Defrosting methods:** manual (switch off), off-cycle, **electric heaters**, **hot-gas defrost** (diverting compressor discharge to the evaporator), water spray, reverse cycle.

6.3 Design features of evaporators and condensers (★)

Design feature	Effect
Heat-transfer area (size)	Larger area → smaller ΔT between refrigerant and fluid → higher evaporating and lower condensing temperature → **higher COP and capacity**, but more first cost
Fins (spacing, height, material)	Increase the air-side area (air side has a low h). Close fins raise area but cause frost blockage and a higher air pressure drop. Typical spacing: 2–3 mm for AC coils, 6–12 mm for freezers.
Number of rows (depth)	More rows → lower bypass factor, more dehumidification, but a higher air-side pressure drop
Face velocity	2–3 m/s for cooling coils (higher velocity carries condensate over, and needs eliminators)

Design feature	Effect
Refrigerant circuiting / velocity	High velocity gives a good h and oil return, but a large pressure drop (which lowers the effective evaporator temperature). Multiple parallel circuits with a distributor are used.
Tube material and enhancement	Copper (halocarbons), steel (NH_3); internally grooved or enhanced tubes increase the refrigerant-side h
Flooded vs DX	Flooded gives a higher U, but a larger charge and oil handling
Water velocity and passes (shell and tube)	1–2.5 m/s: higher h and less fouling, but a higher pump power and erosion
Fouling allowance / cleanability	Removable heads, straight tubes for mechanical cleaning; a design fouling factor of about 0.000088–0.000176 m² K/W
Air-cooled condenser placement	Avoid recirculation of hot air and direct sun. Every 1 K rise in air-on temperature raises the condensing temperature by about 1 K.

6.4 Effect of evaporator and condenser on system performance (Arora Ch 10)

The system operates where the **compressor capacity curve meets the evaporator (and condenser) capacity curves**. This is the balance point.

- **Undersized evaporator, frosted coil, low air flow, or oil fouling** → a larger ΔT is needed → the **evaporator temperature falls** → specific volume rises, RE falls and work rises → **capacity and COP fall**, and air dehumidification increases. *From the R134a figures in Unit 2: dropping T_e from −10 to −15 °C cuts COP from 4.03 to 3.50 (−13%) and raises kW/TR from 0.87 to 1.00.*
- **Undersized or fouled condenser, high ambient, air in the system, or low water flow** → the **condensing temperature rises** → more flash gas, more work, higher discharge temperature → **COP falls** (4.03 → 3.08 for 40 → 50 °C), and the compressor motor may overload.
- **Oversized evaporator/condenser** → better COP, but higher cost and refrigerant charge. With a DX evaporator, an oversized coil can cause poor oil return at low velocity.
- **Superheat setting of the TXV** too high → part of the evaporator is starved, so capacity falls; too low → liquid floods back.
- **Sub-cooling in the condenser** (or a separate sub-cooler) → RE rises, so capacity rises at the same work.

SOLVED

Worked example 6.1 (water-cooled condenser) · A 10 TR plant has COP 3.5. Refrigerant condenses at 40 °C, and cooling water enters at 30 °C and leaves at 35 °C. U = 800 W/m² K. Find the heat rejected, the water flow and the condenser area. Q_c = 35.17(1 + 1/3.5) = **45.2 kW**; $\dot{m}_w$ = 45.2/(4.187 × 5) = **2.16 kg/s** LMTD = (10 − 5)/ln(10/5) = **7.21 K**; A = 45 210/(800 × 7.21) = **7.84 m²**

SOLVED

Worked example 6.2 (water chiller evaporator) · 10 TR; chilled water 12 °C → 7 °C; refrigerant evaporates at 2 °C; U = 600 W/m² K. LMTD = (10 − 5)/ln 2 = 7.21 K; A = 35 167/(600 × 7.21) = **8.13 m²**. If fouling cuts U by 20%, the same area needs LMTD ≈ 9.0 K, so the evaporating temperature drops by about 2 K and COP falls by roughly 4–7%.

EXAM TIP

Unit 6 must-knows: HRR · condenser types with merits and demerits (especially the evaporative condenser) · evaporator types, DX vs flooded · frosting and defrost methods · the design-features table · how a fouled condenser or undersized evaporator lowers COP (link to Unit 2 parameter effects).

Formula sheet: last-day revision

Unit 1: basics and air cycle

- 1 TR = 3.5167 kW = 211 kJ/min (≈ 3.5 kW, 210 kJ/min)
- $\text{COP}_R = Q_L/W$; $\text{COP}_{HP} = \text{COP}_R + 1$
- Carnot: $\text{COP} = \dfrac{T_L}{T_H - T_L}$
- Bell-Coleman: $\text{COP} = \dfrac{1}{r_p^{(\gamma-1)/\gamma} - 1} = \dfrac{T_1}{T_2 - T_1}$
- $T_2 = T_1 r_p^{(\gamma-1)/\gamma}$; $T_4 = T_3/r_p^{(\gamma-1)/\gamma}$
- Polytropic work: $\frac{n}{n-1}R(T_2 - T_1)$
- $\eta_c = \dfrac{T_{2s} - T_1}{T_2 - T_1}$, $\eta_t = \dfrac{T_3 - T_4}{T_3 - T_{4s}}$
- Ram: $T_2 = T_1 + \dfrac{V^2}{2c_p}$; $\dfrac{p_2'}{p_1} = (T_2/T_1)^{\gamma/(\gamma-1)}$; $\eta_{ram} = \dfrac{p_2 - p_1}{p_2' - p_1}$
- Aircraft: $\text{COP} = \dfrac{T_{cabin} - T_5}{T_3 - T_2}$; $\dot{m} = \dfrac{Q}{c_p(T_{cabin} - T_5)}$

Unit 2: vapour compression

- RE = $h_1 - h_4$; W = $h_2 - h_1$; $h_4 = h_3$
- COP = $\dfrac{h_1 - h_4}{h_2 - h_1}$
- $\dot{m} = \dfrac{3.5167\,\text{TR}}{h_1 - h_4}$ kg/s; $\dot{V} = \dot{m} v_1$
- HRR = $1 + 1/\text{COP}$
- Wet state: $h = h_f + x h_{fg}$, $s = s_f + x s_{fg}$
- Superheat: $s_1 = s_g + c_p \ln(T/T_s)$, $h = h_g + c_p(T - T_s)$
- $\eta_v = 1 + C - C(p_d/p_s)^{1/n}$
- Swept volume: $\frac{\pi}{4} D^2 L \frac{N}{60} k = \dot{m} v_1/\eta_v$
- $p_i = \sqrt{p_e p_c}$
- Flash intercooler: $\dot{m}_H = \dot{m}_L \dfrac{h_2 - h_7}{h_3 - h_6}$
- Cascade: $\text{COP} = \dfrac{c_1 c_2}{1 + c_1 + c_2}$

Unit 3: refrigerants and other systems

- R-number: (m−1)(n+1)(p); Cl = 2m + 2 − n − p
- Inorganic: R700 + molecular mass
- Absorption: $\text{COP}_{max} = \dfrac{T_E}{T_0 - T_E} \cdot \dfrac{T_G - T_0}{T_G}$
- $Q_G + Q_E + W_P = Q_C + Q_A$
- Steam jet: flash fraction = $c_w \Delta t / h_{fg}$
- Vortex tube: RE = $\mu_c c_p (T_i - T_c)$; $W = R T_i \ln(p_i/p_a)$
- $T_i = \mu_c T_c + (1 - \mu_c) T_h$

Unit 4: psychrometry

- $W = 0.622 \dfrac{p_v}{p - p_v}$; $\phi = p_v/p_{vs}$
- $\mu = W/W_s = \phi \dfrac{p - p_{vs}}{p - p_v}$
- $h = 1.005t + W(2500 + 1.88t)$
- $v = R_a T/(p - p_v)$
- Carrier: $p_v = p'_{vs} - \dfrac{(p - p'_{vs})(t - t')}{1544 - 1.44t'}$
- BPF = $\dfrac{t_2 - t_{ADP}}{t_1 - t_{ADP}}$; CF = 1 − BPF
- SH = 0.0204 cmm Δt; LH = 50 cmm ΔW; TH = 0.02 cmm Δh (kW)
- SHF = SH/(SH + LH)
- Mixing: $W_3 = \sum mW / \sum m$, $h_3 = \sum mh / \sum m$
- Evap. cooler: $\eta = \dfrac{t_1 - t_2}{t_1 - t'}$

Units 4 and 5: load and applied psychrometry

- ERSH = RSH + BPF·OASH; ERLH = RLH + BPF·OALH
- ESHF = ERSH/(ERSH + ERLH)
- $\text{cmm}_d = \dfrac{\text{ERSH}}{0.0204(1 - \text{BPF})(t_i - t_{ADP})}$
- GTH = RSH + RLH + OASH + OALH; TR = GTH/3.5167
- Glass solar: A·SHGF·SC·CLF; wall: U·A·CLTD
- Infiltration cmm = ACH × Volume/60
- Lights: W × 1.25 (fluorescent)
- Body: $M - W = Q_{conv} + Q_{rad} + Q_{evap} + Q_{resp} + S$

Unit 6: evaporators and condensers

- $Q_c = Q_e + W = Q_e(1 + 1/\text{COP})$
- $Q = UA\,\text{LMTD}$; LMTD = $\dfrac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)}$
- $\dot{m}_w = Q_c/(c_w \Delta t_w)$
- $1/U = 1/h_o + R_f + x/k + 1/h_i$ (thin tube)
- Nusselt (horizontal tube): $h = 0.725[k^3 \rho^2 g h_{fg}/(\mu D \Delta T)]^{1/4}$

PART B: QUESTION BANK

Previous-year-pattern question bank

What this bank is. These questions are compiled in the style and wording of questions that keep recurring in university end-semester RAC papers (AKTU, RTU, MDU/KUK, RGPV, GTU, VTU, Anna and others that use Arora as the textbook), plus GATE/ESE objective patterns. They are **not** copied from one specific paper, so no years are attached. ★ **= asked again and again.** Every numerical has a final answer (computed for these notes) so you can check your work. Where property values are given in the question, use them, so that your answer matches regardless of which tables you use.

Unit 1: Introduction and air refrigeration

A. Short-answer questions (2–3 marks)

1. ★ Define refrigeration. How does it differ from cooling?
2. ★ Define one tonne of refrigeration and show that 1 TR ≈ 3.5 kW.
3. ★ Define COP. Why is it called "coefficient" and not "efficiency"?
4. What is the relation between COP of a heat pump and COP of a refrigerator working between the same limits?
5. Define relative COP and EER.
6. State the Clausius statement of the second law and its relevance to refrigeration.
7. ★ Why is the reversed Carnot cycle not used in practice for (a) gases, (b) vapours?
8. Which is more effective for raising Carnot COP: increasing T_L or decreasing T_H by the same amount? Prove it.
9. List any six applications of refrigeration.
10. Name the methods of producing refrigeration.
11. Why is air used as a refrigerant in aircraft?
12. ★ What is meant by a "dense air" refrigeration system?
13. Define ram efficiency.
14. What is DART (dry air rated temperature)?
15. Why is the COP of an air refrigeration system low?
16. What is the effect of pressure ratio on the COP of a Bell-Coleman cycle?
17. Name the four processes of the Bell-Coleman cycle.
18. What is the function of the cooling turbine in an aircraft system?
19. Distinguish between a refrigerator and a heat pump.
20. Why is a water separator used in aircraft air-cycle systems?

B. Long-answer questions (theory and derivations)

1. ★ With a neat T-s diagram, derive the COP of the reversed Carnot cycle. Explain its practical limitations.
2. ★ Explain the Bell-Coleman (reversed Brayton) cycle with schematic, p-v and T-s diagrams, and derive its COP in terms of the pressure ratio.
3. ★ Compare open and dense air refrigeration systems with neat sketches.
4. ★ Explain the need for aircraft refrigeration. Why is air-cycle refrigeration preferred in aircraft?
5. ★ With neat sketches and T-s diagrams, explain (a) the simple air cooling system and (b) the bootstrap air cooling system for aircraft.
6. Explain the regenerative and reduced-ambient air cooling systems. Where is each used?

7. Compare the simple, bootstrap, regenerative and reduced-ambient systems in a table.
8. Discuss the merits and demerits of air refrigeration over vapour compression refrigeration.
9. Derive an expression for the stagnation temperature and pressure due to ram effect.
10. Explain the effect of compressor and expander efficiencies on the COP of an air refrigeration cycle.
11. Write a short note on the history and applications of refrigeration.
12. "The COP of an air refrigerator depends only on pressure ratio." Justify the statement and state its implication for design.

C. Numericals (with answers)

1. ★ A Carnot refrigerator extracts 150 kJ/min from a space at −20 °C and rejects heat to the atmosphere at 30 °C. Find the COP and the power required. **[COP = 5.06; P = 0.494 kW]**
2. A refrigerator has a COP one-third that of a Carnot refrigerator working between 250 K and 300 K. Find the power needed per TR. **[COP_Carnot = 5, actual = 1.67; 2.11 kW]**
3. ★ A Bell-Coleman refrigerator works between 1 bar and 5.5 bar. Air enters the compressor at −10 °C and leaves the cooler at 25 °C. Compression and expansion are isentropic (γ = 1.4). For 15 TR, find the COP, the mass of air, the power and the volume of air handled by the compressor. **[T_2 = 428.3 K, T_4 = 183.2 K, COP = 1.59, $\dot{m}$ = 0.656 kg/s, P = 33.1 kW, $\dot{V}$ = 0.496 m³/s]**
4. Repeat Q3 taking compression and expansion as polytropic with n = 1.25 (work by $\frac{n}{n-1}R\Delta T$). **[T_2 = 370.1 K, T_4 = 212.0 K, W_c = 153.4, W_e = 123.6, RE = 51.4 kJ/kg, COP = 1.72, $\dot{m}$ = 1.03 kg/s, P = 30.6 kW]**
5. ★ A dense air machine works between 3 bar and 12 bar. Air enters the compressor at 7 °C and the expander at 27 °C. Find the COP and RE per kg, and compare the specific volume at compressor inlet with an open system at 1 bar. **[COP = 2.06, T_4 = 201.9 K, RE = 78.5 kJ/kg, v = 0.268 vs 0.804 m³/kg]**
6. ★ In an aircraft system, air at 3.5 bar is cooled in an HX to 330 K and expanded in a cooling turbine (η = 80%) to 1 bar. The cabin is at 25 °C. For 10 TR, find the turbine exit temperature and the mass of air. **[T_5' = 230.7 K, T_5 = 250.6 K (−22.6 °C), RE = 47.8 kJ/kg, $\dot{m}$ = 0.735 kg/s]**
7. An aircraft flies at 900 km/h where the ambient is −20 °C and 0.5 bar. With 80% ram efficiency, find the ram air temperature and pressure. **[T = 284.2 K, p_2' = 0.75 bar, p_2 = 0.70 bar]**
8. ★ (Simple system, full) Speed 1000 km/h; ambient 0.35 bar, −15 °C; ram efficiency 90%; compressor pressure ratio 3, η = 85%; HX exit 305 K; turbine η = 85% to 1 bar; cabin 25 °C; 10 TR. Find the mass flow, power and COP. **[$\dot{m}$ = 1.29 kg/s, P = 166 kW, COP = 0.21]**
9. A Bell-Coleman machine works between 1 bar and 5 bar, from −5 °C (compressor inlet) to 30 °C (expander inlet), with η_c = η_e = 0.85. For 10 TR, find the COP and power. **[COP = 0.67, P = 52.3 kW]**
10. A Carnot heat pump supplies 30 kW to a room at 22 °C with outdoor air at 2 °C. Find the power input. **[COP_HP = 14.75; W = 2.03 kW]**

Unit 2: Vapour compression refrigeration

A. Short-answer questions

1. ★ Draw the simple VCR cycle on p-h and T-s diagrams.
2. ★ Why is the throttle valve used instead of an expansion engine in VCR?
3. ★ Why is the p-h chart preferred for VCR cycle analysis?
4. What is flash gas? How does it affect performance?
5. ★ What is the effect of sub-cooling on COP? Show it on a p-h diagram.
6. Distinguish between useful and useless superheat.
7. Why is some superheat always provided at the compressor suction?
8. Define volumetric efficiency of a compressor. What factors affect it?
9. What is wet compression? Why is dry compression preferred?
10. What is a liquid–suction heat exchanger?

11. Define heat rejection ratio.
12. ★ What is the effect of lowering the evaporator pressure on COP and capacity?
13. ★ What are the limitations of single-stage compression at low evaporator temperatures?
14. Write the expression for the optimum intermediate pressure in two-stage compression.
15. What is a flash intercooler?
16. ★ What is a cascade system? Where is it used?
17. Name suitable refrigerant pairs for a cascade system.
18. Why can't NH_3 be used in the low-temperature stage of a −90 °C cascade?
19. What is meant by "compound compression"?
20. List the essential components of a VCR plant and their functions.

B. Long-answer questions

1. ★ Explain the working of a simple VCR system with a schematic, and represent it on T-s and p-h charts. Derive the COP.
2. ★ Explain the effect of (a) sub-cooling and (b) superheating on the performance of a VCR system with the help of T-s and p-h charts.
3. ★ Differentiate between the theoretical and actual VCR cycles with a p-h diagram.
4. ★ Discuss the effect of evaporator pressure and condenser pressure on the performance of a VCR system.
5. Derive an expression for the volumetric efficiency of a reciprocating compressor considering clearance.
6. ★ What is the necessity of multistage compression? Explain a two-stage compression system with flash intercooler, and draw it on a p-h diagram.
7. Explain flash gas removal and intercooling, and their benefits.
8. ★ Explain the working of a cascade refrigeration system with a neat sketch. Derive its overall COP in terms of the individual COPs.
9. Explain a multi-evaporator system with individual expansion valves and back-pressure valves.
10. Compare vapour compression and air refrigeration systems.
11. Show that the optimum intermediate pressure for minimum work in two-stage compression with perfect intercooling is $\sqrt{p_1 p_2}$.
12. Explain the role of the liquid–vapour regenerative heat exchanger. When does it improve COP?

C. Numericals (with answers)

1. ★ An R134a plant of 10 TR works on a simple saturated cycle between −10 °C and 40 °C. Data: h_1 = 392.7, h_2 = 426.5, h_3 = 256.4 kJ/kg, v_1 = 0.0996 m³/kg. Find the mass flow, power, COP, heat rejected and suction volume. **[ṁ = 0.258 kg/s, P = 8.73 kW, COP = 4.03, Q_c = 43.9 kW, V̇ = 0.0257 m³/s]**
2. ★ An ammonia plant of 20 TR works between −15 °C and 30 °C on a simple saturated cycle. Data: h_1 = 1590, h_2 = 1821 (99 °C), h_3 = 487.2 kJ/kg, v_1 = 0.509 m³/kg. Find the COP, mass flow, power and Carnot COP. **[COP = 4.77, ṁ = 0.0638 kg/s, P = 14.75 kW, COP_Carnot = 5.74]**
3. ★ An R134a system works between 0 °C and 45 °C with 10 K superheat and 5 K sub-cooling (5 TR). Data: h_1 = 407.5, h_2 = 437.7, h_3 = 256.4 kJ/kg. Find the COP and power. **[RE = 151.1 kJ/kg, COP = 5.0, ṁ = 0.116 kg/s, P = 3.52 kW]**
4. An R22 plant of 15 TR runs between −5 °C and 35 °C on a saturated cycle. Data: h_1 = 403.2, h_2 = 432.2, h_3 = 243.1 kJ/kg. Find the COP and power. **[COP = 5.51, ṁ = 0.330 kg/s, P = 9.58 kW]**
5. ★ For Q1, a 4-cylinder single-acting compressor runs at 1440 rpm with bore = stroke and η_v = 0.8. Find the bore. **[Swept volume = 32.1 L/s, D = L = 75 mm]**
6. ★ A two-stage NH_3 system with a flash intercooler produces 10 TR at −40 °C and condenses at 30 °C. The intermediate pressure is the geometric mean. Data (kJ/kg): h_1 = 1553.8, h_2 = 1737.7, h_3 = 1595.8, h_4 = 1795.1, h_5 = 487.2, h_7 = 299.1. Find p_i, the mass flows, power and COP. Compare with single stage (h_2 =

1989.6). **[p_i = 2.89 bar, ṁ_L = 0.028 kg/s, ṁ_H = 0.0364 kg/s, W = 12.4 kW, COP = 2.84; single stage COP = 2.45, T_2 = 167 °C]**

7. A compressor has 4% clearance and works between pressures in the ratio 6, with n = 1.15. Find the volumetric efficiency and the pressure ratio at which delivery stops. **[η_v = 0.85, r = 42.4]**
8. ★ The LT and HT stages of a cascade have COPs of 2.5 and 3.5. Find the overall COP. **[1.25]**
9. Evaporator and condenser pressures are 1.2 bar and 13.5 bar. Find the optimum intermediate pressure. **[4.02 bar]**
10. An R134a heat pump works between 0 °C and 45 °C (saturated). h_1 = 398.6, h_2 = 427.2, h_3 = 263.9 kJ/kg. Find COP_R and COP_HP. **[4.70, 5.70]**
11. In Q1, what happens to COP if the evaporator temperature falls to −20 °C (all else the same)? **[COP falls to 3.06; kW/TR rises from 0.87 to 1.15]**
12. Repeat worked example 2.1 with 5 K sub-cooling and 5 K superheat (h values in the notes). **[COP = 4.26]**

Unit 3: Refrigerants and absorption refrigeration

A. Short-answer questions

1. ★ List the desirable thermodynamic properties of a refrigerant.
2. ★ What are secondary refrigerants? Give examples.
3. ★ Identify: R12, R22, R134a, R717, R744, R290, R600a.
4. ★ Write the chemical formula of R11 and R123, and the R-number of $CHCl_2F$ and $C_2Cl_2F_4$.
5. ★ Define ODP and GWP.
6. What is TEWI?
7. Distinguish between azeotropic and zeotropic refrigerant blends. What is temperature glide?
8. Why are CFCs harmful to the ozone layer?
9. State the purpose of the Montreal Protocol and the Kigali Amendment.
10. Name two alternatives each for R12 and R22.
11. Why is NH_3 preferred in large cold storage plants?
12. Why should a refrigerant have a high dielectric strength?
13. ★ Why is an analyser and rectifier needed in an NH_3–water system but not in a LiBr–water system?
14. What is the role of hydrogen in an Electrolux refrigerator?
15. ★ Why can a LiBr–water system not produce temperatures below 0 °C?
16. What is crystallisation in LiBr systems?
17. Name the refrigerant–absorbent pairs commonly used in absorption systems.
18. What is a "thermal compressor"?
19. What is the entrainment ratio of a steam ejector?
20. ★ What is a vortex tube? Who invented it?
21. What is meant by cold fraction in a vortex tube?
22. Why does a steam-jet system need a very high vacuum?

B. Long-answer questions

1. ★ Discuss the desirable properties of an ideal refrigerant under thermodynamic, chemical and physical headings.
2. ★ Explain the classification of refrigerants with examples, and the ASHRAE nomenclature system.
3. ★ Explain ozone depletion and global warming due to refrigerants, and discuss the alternative refrigerants.
4. ★ Explain the working of a practical NH_3–water vapour absorption system with a neat sketch. State the functions of the analyser, rectifier and heat exchanger.

5. ★ Derive the expression for the maximum COP of an absorption refrigeration system.
6. ★ Explain the working of a LiBr–water absorption system with a neat sketch. What are its limitations?
7. ★ Explain the three-fluid (Electrolux) absorption refrigerator. What are its salient features?
8. Compare vapour absorption and vapour compression systems.
9. Compare NH_3–water and LiBr–water systems.
10. ★ Explain the working principle of a steam-jet refrigeration system with a neat sketch. State its basic components, advantages and limitations.
11. ★ Explain the principle and operation of a vortex tube (Hilsch tube). State its advantages and applications.
12. Discuss the properties and applications of NH_3, R134a, CO_2 and hydrocarbons as refrigerants.

C. Numericals (with answers)

1. ★ Find the maximum COP of an absorption system with heating at 100 °C, refrigeration at −10 °C and ambient at 30 °C. **[1.23]**
2. Repeat for a generator at 110 °C, evaporator at −15 °C and ambient at 35 °C. **[1.01]**
3. An absorption plant of 10 TR has an actual COP of 0.6. Find the heat supplied to the generator. If the ideal COP (130 °C, −10 °C, 30 °C) is used, what is the minimum heat? **[58.6 kW; COP_max = 1.63 → 21.5 kW]**
4. ★ Find the formula and R-number: (a) $CHClF_2$, (b) CCl_2F_2, © C_2H_6, (d) CH_4, (e) $C_2HCl_2F_3$. **[R22, R12, R170, R50, R123]**
5. A vortex tube is supplied with air at 5 bar and 295 K. The cold fraction is 0.25 and the cold air leaves at 255 K. Find the RE per kg of supply air and the COP (isothermal compression). **[RE = 10.05 kJ/kg, W = 136.3 kJ/kg, COP = 0.074]**
6. ★ A steam-jet system chills water from 15 °C to 10 °C for a 50 TR load. Find the chilled water flow and the rate of flashing (h_{fg} = 2477 kJ/kg). **[8.40 kg/s; 0.071 kg/s]**
7. In an absorption system, Q_E = 35 kW, Q_G = 50 kW, pump work = 0.5 kW and Q_C = 38 kW. Find Q_A and the COP. **[Q_A = 47.5 kW; COP = 0.69]**

Unit 4: Air-conditioning and psychrometry

A. Short-answer questions

1. ★ Define DBT, WBT and DPT. When are all three equal?
2. ★ Define specific humidity, relative humidity and degree of saturation.
3. ★ What is the apparatus dew point (ADP)?
4. ★ Define bypass factor. On what does it depend?
5. Define sensible heat factor. What does a low RSHF indicate?
6. Why do constant WBT lines nearly coincide with constant enthalpy lines?
7. What is adiabatic saturation?
8. Distinguish between relative humidity and degree of saturation.
9. What is the dew point? What happens if a cooling coil's surface temperature is below the dew point?
10. ★ Show sensible heating, cooling and dehumidification, and evaporative cooling on a psychrometric chart.
11. What is chemical dehumidification?
12. Define ERSH and ESHF.
13. What is the need for reheat in summer AC?
14. Why is a preheater used in winter AC?
15. What is the humidifying efficiency of an air washer?
16. State the Gibbs–Dalton law.

17. What is the use of Carrier's equation?
18. Define GSHF.

B. Long-answer questions

1. ★ Define and explain the psychrometric properties of air. Derive $W = 0.622\,p_v/(p - p_v)$.
2. ★ Explain the psychrometric chart and show all basic processes on it.
3. ★ Explain the cooling and dehumidification process. Define bypass factor and ADP, and show them on a chart.
4. ★ Explain the summer air-conditioning system for hot and humid weather with a sketch and its psychrometric process.
5. ★ Explain the winter air-conditioning system with a sketch and its psychrometric representation.
6. Explain the adiabatic mixing of two air streams and derive expressions for the mixture properties.
7. ★ Explain RSHF, GSHF and ESHF. Show how the ADP and supply air quantity are found.
8. Derive the expression for the thermodynamic wet-bulb temperature (adiabatic saturation).
9. Explain the year-round air-conditioning system.
10. Explain the factors affecting the bypass factor of a coil.

C. Numericals (with answers)

1. ★ Air at 30 °C DBT and 22 °C WBT, 1.01325 bar. Find p_v, W, φ, DPT, h, v and μ. **[p_v = 2.12 kPa, W = 0.0133, φ = 50%, DPT = 18.5 °C, h = 64.2 kJ/kg, v = 0.877 m³/kg, μ = 0.489]**
2. ★ Air at 30 °C and 60% RH. Find W, DPT, h, v and WBT. **[W = 0.0161, DPT = 21.4 °C, h = 71.2 kJ/kg, v = 0.881 m³/kg, WBT ≈ 23.8 °C]**
3. Air at 10 °C and 80% RH is heated to 30 °C. Find the heat added per kg and the final RH. **[20.3 kJ/kg; 23%]**
4. ★ 100 m³/min of air at 30 °C and 60% RH passes over a coil with ADP 12 °C and BPF 0.2. Find the leaving state, the capacity in TR and the condensate. **[t_2 = 15.6 °C, W_2 = 0.0102, RH ≈ 92%; $\dot{m}$ = 1.89 kg/s; Q = 56.2 kW = 16.0 TR; condensate ≈ 39.9 kg/h]**
5. ★ 50 m³/min of air at 35 °C, 50% RH is mixed with 150 m³/min at 25 °C, 50% RH. Find the mixture condition. **[W = 0.0118, h = 57.7 kJ/kg, t = 27.4 °C, RH ≈ 52%]**
6. ★ A room is kept at 26 °C and 50% RH. RSH = 30 kW, RLH = 8 kW. Outdoor: 40 °C DBT, 26 °C WBT. Fresh air = 50 cmm, BPF = 0.1. Find ERSH, ERLH, ESHF, ADP, dehumidified air quantity and capacity. **[OASH = 14.3, OALH = 12.3 kW; ERSH = 31.4, ERLH = 9.2 kW; ESHF = 0.77; ADP ≈ 12.2 °C; 124 m³/min; GTH = 64.6 kW = 18.4 TR]**
7. ★ Air at 10 °C and 40% RH is to be conditioned to 22 °C and 50% RH. Find the heat and moisture added per kg of dry air. **[25.3 kJ/kg; 5.2 g/kg]**
8. ★ Air at 40 °C and 20% RH passes through an evaporative cooler of 80% efficiency. Find the exit DBT. **[WBT ≈ 22.0 °C; exit ≈ 25.6 °C, W ≈ 0.0150]**
9. Air at 35 °C DBT and 25 °C WBT: find the properties (worked example 4.1). **[W = 0.01586, φ = 44.8%, DPT = 21.2 °C, h = 75.9 kJ/kg]**
10. The summer AC problem of worked example 4.5: re-solve without looking. **[ADP 11.2 °C, 113.5 cmm, 16.4 TR]**

Unit 5: Human comfort and loads

A. Short-answer questions

1. ★ Define effective temperature.
2. ★ List the factors affecting human comfort.
3. What is the comfort chart?

4. What are the typical inside design conditions for summer comfort AC?
5. What is metabolism? State typical metabolic rates.
6. Define clo and met.
7. What is mean radiant temperature?
8. Why is the summer comfort ET higher than the winter one?
9. What is meant by thermal shock?
10. Differentiate between comfort and industrial air-conditioning.
11. What are RSH and RLH?
12. Name the sources of the sensible and latent heat load in a building.
13. What is infiltration? How is it estimated?
14. What is CLTD?
15. Why is a factor of 1.25 applied to fluorescent lighting load?
16. What is the difference between heat gain and cooling load?

B. Long-answer questions

1. ★ Explain the heat balance of the human body and the body's regulatory mechanisms.
2. ★ Explain effective temperature and the factors governing it. Draw and explain the ASHRAE comfort chart.
3. ★ Discuss the factors affecting human comfort.
4. ★ Discuss the requirements of industrial air-conditioning for (a) textile, (b) printing, © pharmaceutical and (d) precision manufacturing industries.
5. ★ Explain the various components of the cooling load in an air-conditioning load calculation.
6. Explain how the solar heat gain through glass and walls is estimated.
7. Explain the concept of outside and inside design conditions.
8. Discuss indoor air quality and ventilation requirements.
9. Explain the limitations of effective temperature as a comfort index.

C. Numericals (with answers)

1. ★ Estimate the cooling load of the office in worked example 5.1, but with 30 occupants, glass area 12 m^2 (wall unchanged) and ventilation 0.6 cmm/person. **[RSH = 14.47 kW, RLH = 2.37 kW (incl. 5% SF), RSHF = 0.86; OASH = 6.61 kW, OALH = 5.44 kW; GTH = 28.9 kW = 8.2 TR]**
2. ★ An auditorium seats 50 people, with 0.28 cmm of fresh air per person. Outdoor: 38 °C, 50% RH; inside: 25 °C, 50% RH. Find the ventilation sensible and latent loads. **[OASH = 3.71 kW; OALH = 7.81 kW]**
3. 0.5 air changes per hour infiltrate a 240 m^3 room (43 °C outside, 25 °C inside, $\Delta W = 0.00604$). Find the infiltration loads. **[0.73 kW sensible; 0.60 kW latent]**
4. A wall of 30 m^2 ($U = 1.8$ W/m^2 K) has a CLTD of 12 K, and a 1.2 kW fluorescent installation is in use. Find the sensible load. **[648 + 1500 = 2148 W]**

Unit 6: Evaporators and condensers

A. Short-answer questions

1. ★ Define heat rejection ratio.
2. ★ Classify condensers.
3. ★ What is an evaporative condenser? Where is it used?
4. Why does air in a condenser reduce performance? How is it removed?
5. ★ Distinguish between flooded and dry-expansion evaporators.
6. What is a Baudelot cooler?

7. ★ Why is defrosting necessary? Name the methods.
8. What is the effect of fouling on condenser performance?
9. What is a Wilson plot?
10. Why are fins provided on the air side of evaporators and condensers?
11. What is liquid overfeed?
12. What is the effect of an undersized evaporator on system performance?

B. Long-answer questions

1. ★ Classify condensers and explain the shell-and-tube and evaporative condensers with neat sketches.
2. ★ Compare air-cooled, water-cooled and evaporative condensers.
3. ★ Classify evaporators and explain flooded and DX evaporators with sketches.
4. ★ Discuss the effect of evaporator and condenser design features on the performance of refrigeration and AC systems.
5. Explain frosting and the various defrosting methods.
6. Explain the factors affecting heat transfer in condensers.
7. Explain the balance of a compressor–condenser–evaporator system.

C. Numericals (with answers)

1. ★ A 15 TR plant has COP 3.2. The refrigerant condenses at 42 °C; water enters at 30 °C and leaves at 36 °C. With U = 900 W/m² K, find the heat rejected, water flow and area. **[Q_c = 69.2 kW, ṁ_w = 2.76 kg/s, LMTD = 8.66 K, A = 8.9 m²]**
2. An air-cooled condenser rejects 20 kW with a 10 K air temperature rise. Find the air mass and volume flow (ρ = 1.16 kg/m³). **[1.99 kg/s; 1.72 m³/s]**
3. ★ A 20 TR water chiller evaporator cools water from 12 °C to 4 °C at an evaporating temperature of 0 °C. U = 500 W/m² K. Find the LMTD, area and water flow. **[LMTD = 7.28 K, A = 19.3 m², ṁ_w = 2.10 kg/s]**
4. Find the HRR for COPs of 3 and 5. **[1.33, 1.20]**
5. ★ Using the R134a table in Unit 2, estimate the percentage drop in COP when the evaporator temperature falls from −10 °C to −15 °C due to frosting. **[4.03 → 3.50, −13%]**

Objective questions (GATE/ESE pattern)

1. 1 TR is approximately: (a) 2.5 kW (b) 3.5 kW © 4.5 kW (d) 1 kW **— b**
2. The COP of a Carnot refrigerator between 250 K and 300 K is: (a) 6 (b) 5 © 1.2 (d) 0.2 **— b**
3. The COP of a heat pump is always: (a) less than 1 (b) COP_R − 1 © COP_R + 1 (d) equal to COP_R **— c**
4. In the Bell-Coleman cycle, heat is rejected at: (a) constant temperature (b) constant pressure © constant volume (d) constant entropy **— b**
5. The COP of an air refrigeration cycle increases when the pressure ratio: (a) increases (b) decreases © stays the same (d) none **— b**
6. The bootstrap system has: (a) one HX (b) two HXs and a secondary compressor © an evaporator only (d) no turbine **— b**
7. In VCR, throttling occurs at constant: (a) entropy (b) temperature © enthalpy (d) pressure **— c**
8. Sub-cooling the liquid refrigerant: (a) raises RE and COP (b) raises work © lowers COP (d) has no effect **— a**
9. Lowering the evaporator temperature at constant condenser temperature: (a) increases COP (b) decreases COP © no change (d) increases RE **— b**
10. The optimum intermediate pressure in two-stage compression is: (a) $(p_1 + p_2)/2$ (b) $\sqrt{p_1 p_2}$ © $p_1 p_2$ (d) p_2/p_1 **— b**

11. The overall COP of a cascade with stage COPs c_1, c_2 is: (a) $c_1 c_2$ (b) $c_1 c_2/(c_1 + c_2)$ © $c_1 c_2/(1 + c_1 + c_2)$ (d) $c_1 + c_2$ — **c**
12. Volumetric efficiency of a reciprocating compressor decreases with: (a) a decrease in clearance (b) an increase in pressure ratio © a decrease in pressure ratio (d) none — **b**
13. The R-number of NH_3 is: (a) R717 (b) R718 © R744 (d) R729 — **a**
14. R134a is: (a) CFC (b) HCFC © HFC (d) HC — **c**
15. ODP is measured relative to: (a) CO_2 (b) R12 © R11 (d) R22 — **c**
16. GWP is measured relative to: (a) CO_2 (b) R11 © CH_4 (d) H_2O — **a**
17. In a LiBr–water system, the refrigerant is: (a) LiBr (b) water © NH_3 (d) H_2 — **b**
18. In the Electrolux refrigerator, hydrogen is used to: (a) absorb NH_3 (b) lower the partial pressure of NH_3 in the evaporator © act as refrigerant (d) cool the absorber — **b**
19. The maximum COP of an absorption system depends on: (a) T_G only (b) T_G, T_0 and T_E © T_E only (d) refrigerant properties — **b**
20. An analyser and rectifier are used in: (a) LiBr–water (b) NH_3–water © steam jet (d) vortex tube — **b**
21. In steam-jet refrigeration, the refrigerant is: (a) steam (b) water © NH_3 (d) air — **b**
22. In a vortex tube, cold air leaves from: (a) the periphery near the valve (b) the central orifice near the nozzle © the middle of the tube (d) both ends — **b**
23. Constant WBT lines on the psychrometric chart nearly coincide with constant: (a) DBT (b) RH © enthalpy (d) specific volume — **c**
24. During sensible heating, the following stays constant: (a) RH (b) DBT © specific humidity (d) WBT — **c**
25. When air is saturated: (a) DBT > WBT (b) DBT = WBT = DPT © WBT > DPT (d) DPT > DBT — **b**
26. During adiabatic saturation, the following stays nearly constant: (a) DBT (b) W © WBT (d) DPT — **c**
27. A bypass factor of 0.2 means the contact factor is: (a) 0.2 (b) 0.8 © 1.2 (d) 5 — **b**
28. The bypass factor decreases with: (a) fewer rows (b) higher air velocity © more rows of tubes (d) wider fin spacing — **c**
29. Effective temperature combines: (a) DBT, RH, air velocity (b) DBT, MRT © WBT only (d) DPT, DBT — **a**
30. In summer comfort AC, the recommended inside condition is about: (a) 18 °C, 80% RH (b) 25 °C, 50% RH © 30 °C, 30% RH (d) 21 °C, 90% RH — **b**
31. Printing presses need constant humidity mainly for: (a) worker comfort (b) paper dimensional stability © ink colour (d) fire safety — **b**
32. The heat rejection ratio for COP = 4 is: (a) 1.25 (b) 0.75 © 4 (d) 5 — **a**
33. An evaporative condenser brings the condensing temperature close to the: (a) ambient DBT (b) ambient WBT © DPT (d) water inlet temperature — **b**
34. A flooded evaporator has, compared with DX: (a) a lower U (b) a higher U and a larger charge © a smaller charge (d) no surge drum — **b**
35. Non-condensable gases in a condenser: (a) raise the condenser pressure (b) lower it © raise COP (d) have no effect — **a**

Mock paper (3 hours · attempt any five · 14 marks each)

1. (a) With a T-s diagram, derive the COP of the Bell-Coleman cycle. (7) (b) Numerical U1-C3. (7)
2. (a) Explain the bootstrap air cooling system for aircraft. (7) (b) Compare open and dense air systems. (7)
3. (a) Explain the effect of sub-cooling and superheating on a p-h chart. (6) (b) Numerical U2-C1. (8)
4. (a) Why multistage compression? Explain a two-stage system with a flash intercooler. (7) (b) Explain a cascade system and derive its COP. (7)
5. (a) Explain the NH_3–water absorption system and derive its maximum COP. (8) (b) Explain the Electrolux refrigerator. (6)

6. (a) Explain the desirable properties and nomenclature of refrigerants, and the environmental issues. (7) (b) Explain the vortex tube **or** steam-jet refrigeration. (7)
7. (a) Define the psychrometric properties and show the basic processes on a chart. (6) (b) Numerical U4-C6. (8)
8. (a) Explain effective temperature and the comfort chart. (7) (b) Write short notes on the requirements of industrial AC and on evaporative condensers. (7)

EXAM TIP

On exam day: draw a sketch with every theory answer (schematic + T-s/p-h, or a psychrometric chart sketch). Underline the final numerical answers with units. State your assumptions (1 TR = 3.5 kW, c_p = 1.005, γ = 1.4) at the start of each numerical.