Delhi Technological University · B.Tech Mechanical Engineering

Refrigeration & Air-Conditioning

Every previous-year question from the shared PYQ folder, solved in full and laid out to match the marks on each paper.

PAPER	COURSE	SESSION	MARKS
End-Term Examination (VII Sem)	ME407	Nov 2024	50 · 3 h
Mid-Semester Examination (VII Sem)	ME407	Sep 2019	30 · 1.5 h
Mid-Semester Examination (V Sem)	ME359	Sep 2019	25 · 1.5 h

How these answers are built. DTU does not publish model answers, so each answer follows how marks are usually split in DTU RAC papers: definition or principle → neat labelled sketch (schematic + T–s / p–h / psychrometric chart) → working or derivation → substituted calculation with units → boxed result. The space given to each part follows the marks printed on the paper.

Property data. Where a paper gives property values, those values are used exactly. Otherwise refrigerant and moist-air properties are taken from standard equations of state (IIR reference state for halocarbons: h = 200 kJ/kg, s = 1.0 kJ/kg·K for saturated liquid at 0 °C, the same as the tables in C.P. Arora). If your table uses a different reference, the absolute enthalpies shift, but every enthalpy *difference*, and so every COP, power and mass-flow result, stays the same.

Sources: *Question papers for end term theory examinations Nov–Dec 2024* (pp. 434–435) and the mid-semester photos (Sep 2019 compilation, pp. 266–269) from the shared Google Drive folder. All 1,807 pages in the folder (May 2023, May 2024 and Nov 2024 compilations, PYQs.pdf and the photos) were OCR-searched. These three papers are the only RAC papers in it. Original scans are in the Appendix.

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What's inside

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Q1 Apjohn / psychrometric properties · Q2 humidification · Q3 spray washer, mixing · Q4 ET, ducts, expansion devices · Q5 refrigerants · Q6 VAS, bootstrap · Q7 R22 plant, cascade

Paper 2 · ME407 Mid-Sem, Sep 2019 — Reversed Carnot, R12 cycle with LSHX, refrigerants, aircraft, multistage

Paper 3 · ME359 Mid-Sem, Sep 2019 — Condenser pressure, NH_3 ice plant, wet/dry compression, refrigerants, two-stage

Appendix A · One-page formula sheet — for last-minute revision

Appendix B · Original question papers — scans / photos as found in the folder

Some topics come up in more than one paper: **multistage / flash intercooling** (P2 Q5, P3 Q5–Q6), **refrigerant properties & nomenclature** (P1 Q5, P2 Q3, P3 Q4), **aircraft / bootstrap systems** (P1 Q6b, P2 Q4). Learn these first.

PAPER 1 · VII SEMESTER B.TECH · END-TERM EXAMINATION

ME407 · End-Term · Nov 2024

Refrigeration and Air-Conditioning · Time 3 h · Max. marks 50 · Attempt any FIVE · All questions carry equal marks · Refrigeration tables and charts allowed

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Q1 (a) **An air sample has DBT and WBT as 25 °C and 20 °C. Calculate without using psychrometric chart (i) relative humidity (ii) DPT (iii) partial pressure of water vapour (iv) enthalpy (v) DPT. Use Apjohn equation $p_v = p'_v - \dfrac{1.8\,p\,(t - t')}{2700}$.** [5] CO4

GIVEN

$t = 25\,^\circ\text{C}$ (DBT), $t' = 20\,^\circ\text{C}$ (WBT), barometric pressure $p = 1.01325$ bar $= 101.325$ kPa (assumed standard). From steam tables: $p'_v = p_s(20\,^\circ\text{C}) = 2.339$ kPa, $p_s(25\,^\circ\text{C}) = 3.169$ kPa.

(III) PARTIAL PRESSURE OF VAPOUR (APJOHN), WITH T IN °C AND P IN KPA

$$p_v = 2.339 - \frac{1.8 \times 101.325 \times (25 - 20)}{2700} = 2.339 - 0.338 = \mathbf{2.002\ kPa}$$

(I) RELATIVE HUMIDITY

$$\phi = \frac{p_v}{p_s(25^\circ\text{C})} = \frac{2.002}{3.169} = 0.632 \;\Rightarrow\; \phi \approx \mathbf{63.2\,\%}$$

(II)/(V) DEW-POINT TEMPERATURE

DPT is the saturation temperature at p_v. From steam tables, $p_s = 2.002$ kPa lies between 17 °C (1.938 kPa) and 18 °C (2.064 kPa):

$$t_{dp} = 17 + \frac{2.002 - 1.938}{2.064 - 1.938} = \mathbf{17.5\,^\circ\text{C}}$$

SPECIFIC HUMIDITY (NEEDED FOR ENTHALPY)

$$\omega = 0.622\frac{p_v}{p - p_v} = 0.622 \times \frac{2.002}{101.325 - 2.002} = 0.01253\ \text{kg/kg d.a.}$$

(IV) ENTHALPY

$$h = 1.005\,t + \omega\,(2500 + 1.88\,t) = 1.005(25) + 0.01253\,(2500 + 47) = 25.13 + 31.92 = \mathbf{57.05\ kJ/kg\ d.a.}$$

Item (v) repeats DPT (17.5 °C, as above). If the examiner meant **degree of saturation**: $\omega_s = 0.622 \times 3.169/(101.325 - 3.169) = 0.02009$, so $\mu = \omega/\omega_s = 0.01253/0.02009 = 0.624$.

> **Answer:** $p_v = 2.0$ kPa · RH = 63.2 % · DPT = 17.5 °C · $\omega = 0.0125$ kg/kg d.a. · $h = 57.05$ kJ/kg d.a. · ($\mu = 0.624$)

Scoring tip — Write the Apjohn equation, state the units (p in kPa, t in °C), and give each of the five answers on its own line with units. Marks are lost for a missing DPT interpolation step.

Q1 (b) **Define specific humidity and prove that it is given by $0.622\,\dfrac{p_v}{p - p_v}$, where p is barometric pressure.** [5] CO4

Definition. Specific humidity (humidity ratio) ω is the mass of water vapour present per kg of *dry air* in a moist-air mixture: $\omega = m_v/m_a$ (kg vapour / kg dry air).

Proof. Moist air is treated as a mixture of two ideal gases, dry air and water vapour, that fill the same volume V at the same temperature T. By Dalton's law the barometric pressure is $p = p_a + p_v$.

$$p_a V = m_a R_a T, \qquad p_v V = m_v R_v T$$

$$\omega = \frac{m_v}{m_a} = \frac{p_v V/(R_v T)}{p_a V/(R_a T)} = \frac{R_a}{R_v}\cdot\frac{p_v}{p_a}$$

With $R_a = \bar{R}/M_a = 8314/28.97 = 287$ J/kg·K and $R_v = 8314/18.02 = 461.5$ J/kg·K:

$$\frac{R_a}{R_v} = \frac{287}{461.5} = \frac{M_v}{M_a} = \frac{18.02}{28.97} = 0.622$$

$$\boxed{\omega = 0.622\,\frac{p_v}{p_a} = 0.622\,\frac{p_v}{p - p_v}}$$

Because p is nearly constant, ω depends only on p_v. So ω (and the dew point) stays constant in any process where no moisture is added or removed, e.g. sensible heating or cooling. Maximum value: $\omega_s = 0.622\,p_s/(p - p_s)$ for saturated air.

Scoring tip — State both assumptions (ideal-gas mixture, Dalton's law) and show the 0.622 = M_v/M_a step explicitly. That step is what the 5 marks are for.

Q2 (a) Explain the process of cooling and humidification (adiabatic humidification only). Specify the season for which this process is suitable. [5] CO4

Process. Unsaturated air is passed through a spray of *recirculated* water (air washer / evaporative cooler) or over wetted pads. No heat is supplied or removed from outside, so the process is adiabatic. The recirculated water settles at the thermodynamic wet-bulb temperature t' of the incoming air.

- Water evaporates into the air. The latent heat for this evaporation comes from the **sensible heat of the air**, so the DBT falls.
- The vapour added carries that heat back into the air as latent heat, so the **enthalpy stays (almost) constant**. The process follows the constant-WBT (≈ constant-enthalpy) line on the chart.
- Results: DBT ↓, specific humidity ω ↑, RH ↑, WBT constant, DPT ↑, h ≈ constant. The lowest temperature reachable is the WBT (exit air saturated, point 2′).

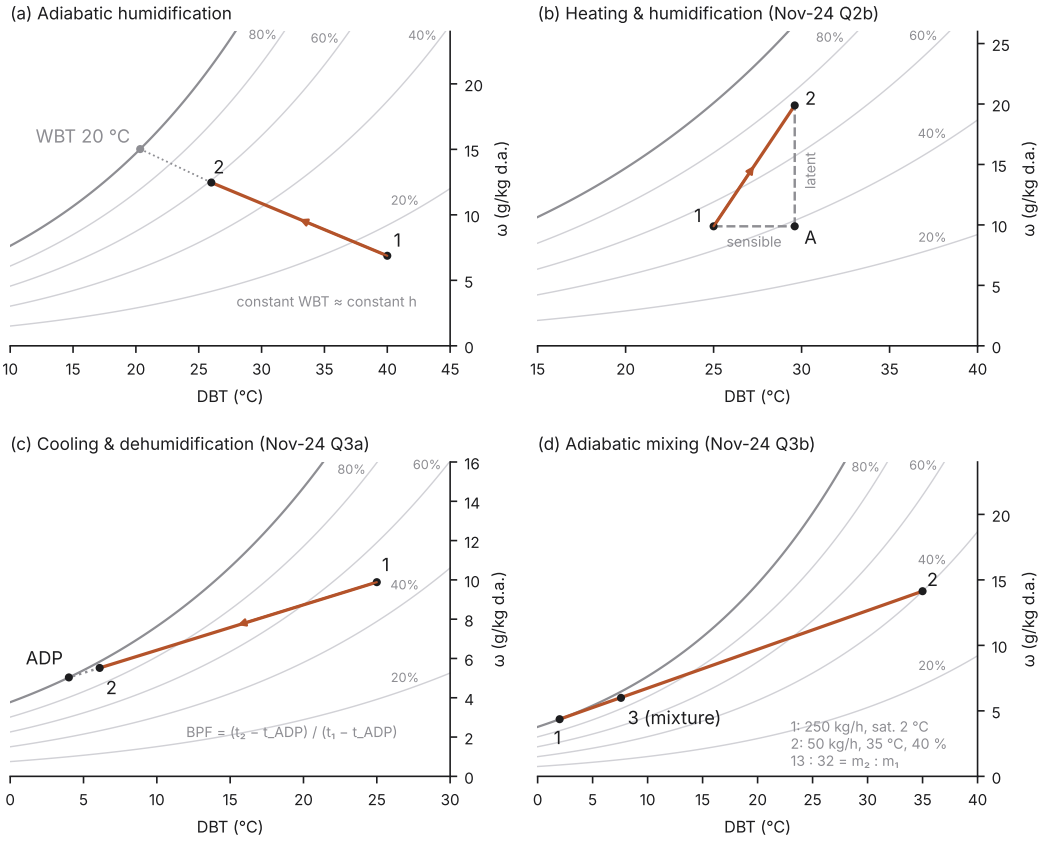

Fig. 1 — Psychrometric processes. Panel (a): adiabatic humidification 1→2 along the constant-WBT line. Panels (b)–(d) are the processes in Q2(b), Q3(a), Q3(b).

Energy balance: $h_1 + (\omega_2 - \omega_1)\,h_f(t') = h_2$. Since $(\omega_2 - \omega_1)h_f$ is very small, $h_1 \approx h_2$.

Humidifying (saturation) efficiency: $\eta_H = \dfrac{t_1 - t_2}{t_1 - t'} = \dfrac{\omega_2 - \omega_1}{\omega_s' - \omega_1}$.

Suitable season: hot and dry summer (e.g. April–June in Delhi or Rajasthan, when RH is low and the wet-bulb depression is large). It cools and humidifies at the same time, which is the basis of desert coolers and air washers. It is *not* suitable for the humid monsoon season, because the air is already near saturation: very little cooling is possible and the air becomes uncomfortably humid.

Q2 (b) **25 kg/min of air at a DBT of 25 °C and 50 % RH is heated and humidified such that the WBT of air at exit is 26 °C and the increase in water vapour is 0.01 kg/kg of d.a. Calculate (i) sensible and latent heat added to air (ii) SHF.** [5] CO4

STATE 1 (INLET): 25 °C, 50 % RH, P = 101.325 KPA

$$p_{v1} = 0.5 \times 3.169 = 1.585\ \text{kPa}; \quad \omega_1 = 0.622\frac{1.585}{101.325 - 1.585} = 0.00988; \quad h_1 = 25.13 + 0.00988(2547) = 5$$

STATE 2 (EXIT): WBT = 26 °C, $\Omega_2 = \Omega_1 + 0.01 = 0.01988$

Enthalpy is (practically) constant along a WBT line, so h_2 equals the enthalpy of saturated air at 26 °C:

$$\omega_s(26) = 0.622\frac{3.363}{101.325 - 3.363} = 0.02136; \quad h_2 = 1.005(26) + 0.02136(2500 + 48.9) = 80.57\ \text{kJ/kg}$$

$$t_2:\ 80.57 = 1.005\,t_2 + 0.01988\,(2500 + 1.88\,t_2) \Rightarrow t_2 = 29.6\,^\circ\text{C}\ (\text{RH}_2 \approx 76\,\%)$$

SPLIT THE PROCESS THROUGH POINT A ($T_A = T_2$, $\Omega_A = \Omega_1$), FIG. 1(B)

$$h_A = 1.005(29.6) + 0.00988\,(2500 + 1.88 \times 29.6) = 55.01\ \text{kJ/kg}$$

Quantity	Expression	kJ/kg d.a.	kJ/min	kW
Sensible heat	$\dot{m}(h_A - h_1)$	4.71	117.9	1.96
Latent heat	$\dot{m}(h_2 - h_A)$	25.56	638.9	10.65
Total heat	$\dot{m}(h_2 - h_1)$	30.27	756.8	12.61

$$\text{SHF} = \frac{\text{SH}}{\text{SH} + \text{LH}} = \frac{117.9}{756.8} = \mathbf{0.156}$$

> **Answer:** Sensible heat ≈ 118 kJ/min (1.96 kW); latent heat ≈ 639 kJ/min (10.65 kW); total ≈ 757 kJ/min (12.6 kW); **SHF ≈ 0.16**. Exit state ≈ 29.6 °C DBT, 76 % RH.

Quick check: latent ≈ $\dot{m}\,\Delta\omega\,h_{fg} = 25 \times 0.01 \times 2500 = 625$ kJ/min. Chart readings give the same answers within ±2 %.

Q3 (a) **The air at 25 °C and 50 % RH is passed through a spray washer at 4 °C. Specify the process which will take place during this interaction of air and spray washer. Assume bypass factor as 0.1 for spray washer, calculate (i) heat added or removed from the air (ii) final condition of air.** [5] CO4

Process. DPT of the inlet air = $t_{sat}(1.585\ \text{kPa}) = 13.9\,^\circ\text{C}$. The spray water at 4 °C is **below the dew point**, so the air is **cooled and dehumidified**: DBT, ω, h and DPT all fall. The water temperature acts as the apparatus dew point (ADP = 4 °C), Fig. 1(c).

$$\text{BPF} = \frac{t_2 - t_{ADP}}{t_1 - t_{ADP}} \Rightarrow t_2 = 4 + 0.1(25 - 4) = \mathbf{6.1\,^\circ\text{C}}$$

$$\omega_{ADP} = 0.622\frac{0.8135}{101.325 - 0.8135} = 0.005035; \quad \omega_2 = \omega_{ADP} + 0.1(\omega_1 - \omega_{ADP}) = 0.005519\ \text{kg/kg}$$

$$h_2 = 1.005(6.1) + 0.005519\,(2500 + 11.5) = 19.99\ \text{kJ/kg}; \qquad h_1 = 50.30\ \text{kJ/kg}$$

$$q = h_1 - h_2 = 50.30 - 19.99 = \mathbf{30.3\ kJ/kg\ d.a.\ (removed)}$$

Moisture condensed = $\omega_1 - \omega_2 = 0.00988 - 0.00552 = 0.00436$ kg/kg d.a. Final RH: $p_{v2} = 0.891$ kPa, $p_s(6.1) = 0.944$ kPa, so $\phi_2 \approx 94.5\,\%$.

> **Answer:** Cooling and dehumidification. Heat removed ≈ **30.3 kJ/kg of dry air**. Final state: **6.1 °C DBT, ω = 5.5 g/kg d.a., RH ≈ 95 %, h ≈ 20 kJ/kg**.

Q3 (b) **250 kg/h of air saturated at 2 °C is adiabatically mixed with 50 kg/h of air at 35 °C and 40 % RH. Determine the final state of air.** [5] CO4

Stream	ṁ (kg/h)	p_v (kPa)	ω (kg/kg)	h (kJ/kg)
1: saturated, 2 °C	250	0.706	0.004364	12.94
2: 35 °C, 40 % RH	50	0.4 × 5.629 = 2.252	0.014136	71.45

Mass and energy balances for adiabatic mixing (taking the given flows as dry-air flows):

$$\omega_3 = \frac{\dot m_1\omega_1 + \dot m_2\omega_2}{\dot m_1 + \dot m_2} = \frac{250(0.004364) + 50(0.014136)}{300} = 0.005993\ \text{kg/kg}$$

$$h_3 = \frac{250(12.94) + 50(71.45)}{300} = 22.69\ \text{kJ/kg}$$

$$22.69 = 1.005\,t_3 + 0.005993\,(2500 + 1.88\,t_3) \Rightarrow t_3 = 7.58\,^\circ\text{C}$$

$p_{v3} = \omega_3 p/(0.622 + \omega_3) = 0.967$ kPa; $p_s(7.58) = 1.044$ kPa, so $\phi_3 = 92.7\,\%$. Check: the mass-weighted DBT $(250 \times 2 + 50 \times 35)/300 = 7.5\,^\circ$C agrees. The mixture point 3 lies on line 1–2, dividing it in the ratio $\overline{13} : \overline{32} = \dot m_2 : \dot m_1 = 1 : 5$ (Fig. 1d).

> **Final state:** DBT ≈ **7.6 °C**, ω ≈ **6.0 g/kg d.a.**, RH ≈ **93 %**, h ≈ **22.7 kJ/kg d.a.** (DPT ≈ 6.5 °C)

Q4 (a) **What is meant by effective temperature? What are the factors on which effective temperature depends?** [5]

Effective temperature (ET) is an index that combines the effects of dry-bulb temperature, humidity and air motion on human comfort into one number. It is defined as *the temperature of saturated (100 % RH), still air (≈ 0.1 m/s) that would produce the same feeling of warmth or coolness as the actual environment*. Example: air at 30 °C DBT, 30 % RH moving at 0.5 m/s may feel the same as still saturated air at 24 °C, so its ET is 24 °C. It was determined experimentally by ASHRAE (Houghten & Yaglou) with groups of people and is plotted on the **comfort chart** (DBT vs WBT with ET lines and comfort zones).

Factors on which ET depends:

- **Climatic/environmental:** dry-bulb temperature, relative humidity (or WBT), air velocity, mean radiant temperature of surroundings.
- **Activity:** metabolic rate (sedentary, light, heavy work).
- **Clothing:** its insulation value (clo).
- **Age and sex** of occupants (women and older people prefer ≈ 0.5 °C higher ET).
- **Duration of stay** and acclimatisation / climate of the region (people in hot climates prefer higher ET).
- **Density of occupants** and the season.

ASHRAE comfort chart: summer optimum ≈ 21.7 °C ET, winter optimum ≈ 19 °C ET, with 60–70 % of people comfortable over a band of about ±2 °C around the optimum. For Indian conditions, comfort design is about 24–26 °C DBT, 50–60 % RH, 6–9 m/min air velocity.

Q4 (b) (i) What are the various duct design methods? (ii) What are the various types of expansion devices? [2.5 + 2.5]

(i) Duct design methods

1. **Velocity reduction method:** a high velocity is chosen at the fan outlet and reduced arbitrarily at each branch (e.g. 8 → 6 → 4 m/s). Sizes come from $A = Q/V$. Simple, but needs an experienced designer, and branches need dampers to balance.
2. **Equal friction method:** the same pressure loss per metre of length (e.g. 0.8–1 Pa/m) is kept for all sections, read from the friction chart. This is the most widely used method for low-velocity supply, return and exhaust ducts. It gives automatic reduction of velocity along the flow, but short runs still need dampers.
3. **Static regain method:** each section is sized so that the rise in static pressure from the velocity drop after each branch just offsets the friction loss of the next section. Static pressure is then the same at every outlet. Best for high-velocity systems and long ducts with many outlets. Gives a balanced system but larger ducts and more calculation.

Others: constant-velocity method (industrial exhaust and conveying), T-method and computer optimisation methods.

(ii) Types of expansion devices

Device	Working / control	Typical use
Capillary tube	Long (1–6 m), small-bore (0.5–2 mm) tube. Fixed restriction; pressure drop by friction and acceleration. No moving parts; pressures equalise at shut-down.	Domestic refrigerators, window/split ACs (< ≈ 10 kW)
Hand-operated expansion valve	Needle valve opened manually; needs an attendant.	Large ammonia plants (bypass), old installations
Automatic (constant-pressure) expansion valve	Diaphragm and spring hold the evaporator pressure constant.	Constant-load systems, water coolers
Thermostatic expansion valve (TXV)	Bulb on the suction line senses superheat. Keeps constant superheat (≈ 5 K) by matching flow to load.	Most commercial / AC systems
Float valves (low-side / high-side)	A float keeps the liquid level constant in the evaporator (LSF) or condenser receiver (HSF).	Flooded evaporators, chillers
Electronic expansion valve (EEV)	Stepper motor or pulse-width valve driven by a microcontroller using p and T sensors.	Inverter ACs, VRF, chillers

Q5 (a) What are the desirable characteristics of a refrigerant? [5]

THERMODYNAMIC

- Low normal boiling point; evaporator pressure slightly *above* atmospheric (so no air or moisture leaks in).
- Moderate condensing pressure (light, cheap equipment).
- High latent heat of vaporisation (large refrigerating effect, low mass flow).

CHEMICAL, PHYSICAL, SAFETY, ENVIRONMENTAL, ECONOMIC

- Non-toxic, non-flammable, non-explosive.
- Chemically stable and inert; non-corrosive to metals; compatible with oil and seals.
- Suitable oil miscibility; does not react with moisture.
- High thermal conductivity, low viscosity (better heat transfer, lower pressure drop).

- Low specific volume of suction vapour (small compressor).
- High critical temperature (well above condensing temperature); low freezing point.
- Low specific heat of liquid and high specific heat of vapour (less flash gas, less superheat), giving high COP and low power per TR.
- Low compression ratio and low discharge temperature.
- High dielectric strength (hermetic compressors); leaks easy to detect.
- **Zero ODP and low GWP** (Montreal & Kigali protocols).
- Cheap and easily available.

Q5 (b) **Decode the chemical name of the following refrigerants (i) R22 (ii) R717. (iii) Can CO_2 be used as a refrigerant?** [5]

Rule for halocarbons $\mathrm{C}_m\mathrm{H}_n\mathrm{F}_p\mathrm{Cl}_q$: the number is **R (m−1)(n+1)(p)**, and $q = 2m + 2 - n - p$ (saturated compounds).

(i) R22 → R 0 2 2: m − 1 = 0 → m = 1; n + 1 = 2 → n = 1; p = 2; q = 2(1) + 2 − 1 − 2 = 1.
Formula **$CHClF_2$, chlorodifluoromethane** (an HCFC; ODP 0.055, GWP ≈ 1810; being phased out).

(ii) R717 → 700 series = inorganic refrigerants; number = 700 + molecular mass. Molecular mass 17 → **NH_3, ammonia** (ODP 0, GWP 0; toxic, mildly flammable; excellent thermodynamic properties; ice plants, cold storages).

(iii) Yes. CO_2 is refrigerant **R744** (700 + 44). Advantages: natural, ODP 0, GWP 1, non-toxic, non-flammable, cheap, very high volumetric refrigerating capacity (small compressor and pipes). Limitations: low critical temperature (31.1 °C) and very high pressures (≈ 35 bar evaporating, 90–120 bar gas-cooler), so the cycle is *transcritical* in warm climates with lower COP; it needs heavy, special components. Uses: low stage of cascade systems, supermarket refrigeration, heat-pump water heaters, automobile AC, dry-ice manufacture.

Q6 (a) **Explain the working of a vapour absorption system with the help of a neat sketch.** [5]

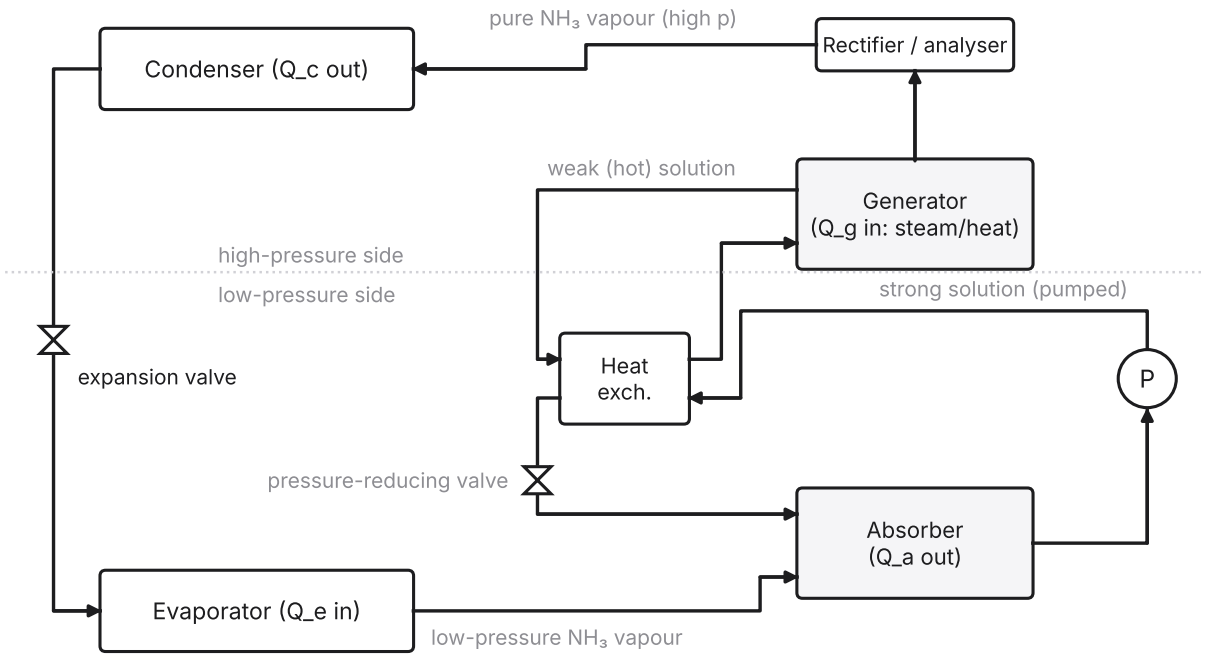

Fig. 2 — Practical aqua-ammonia (NH_3–H_2O) vapour absorption system. NH_3 is the refrigerant, water the absorbent.

Principle. The compressor of a VCR system is replaced by a **"thermal compressor"**: absorber + pump + generator. Ammonia vapour is absorbed into water at low pressure, the liquid solution is pumped to high pressure (very little work, because a liquid is pumped), and the ammonia is then driven off by heat. Energy input is mainly **low-grade heat** (steam, exhaust or solar heat).

1. **Evaporator:** low-pressure liquid NH_3 evaporates, taking heat Q_e from the cold space.
2. **Absorber:** the vapour is absorbed by weak aqua-ammonia. Absorption releases heat Q_a, which is removed by cooling water (absorption capacity falls as temperature rises). The result is a **strong (NH_3-rich) solution**.
3. **Pump:** raises the strong solution to condenser pressure.
4. **Heat exchanger:** the cold strong solution is pre-heated by the hot weak solution returning from the generator. This reduces generator heat and absorber cooling.
5. **Generator:** heat Q_g boils NH_3 out of the solution. The **analyser and rectifier** remove water vapour so that nearly pure NH_3 goes to the condenser (water would otherwise freeze or choke the expansion valve).
6. **Condenser → expansion valve → evaporator**, as in VCR. The hot weak solution returns through the heat exchanger and a pressure-reducing valve to the absorber.

COP = $Q_e/(Q_g + W_p) \approx Q_e/Q_g$ (practical 0.4–0.7). Maximum (ideal) COP with generator at T_g, surroundings (condenser and absorber) at T_0 and evaporator at T_e:

$$\mathrm{COP}_{max} = \frac{T_e}{T_0 - T_e} \times \frac{T_g - T_0}{T_g}$$

Merits: uses waste heat, few moving parts (quiet, low maintenance), capacity can be controlled easily, no compressor wear, no ozone-depleting refrigerant. **Demerits:** low COP, bulky, needs a rectifier (NH_3–H_2O).

Q6 (b) **Explain the working of a Boot Strap system with the help of a neat sketch. Also draw the T-s diagram.** [5]

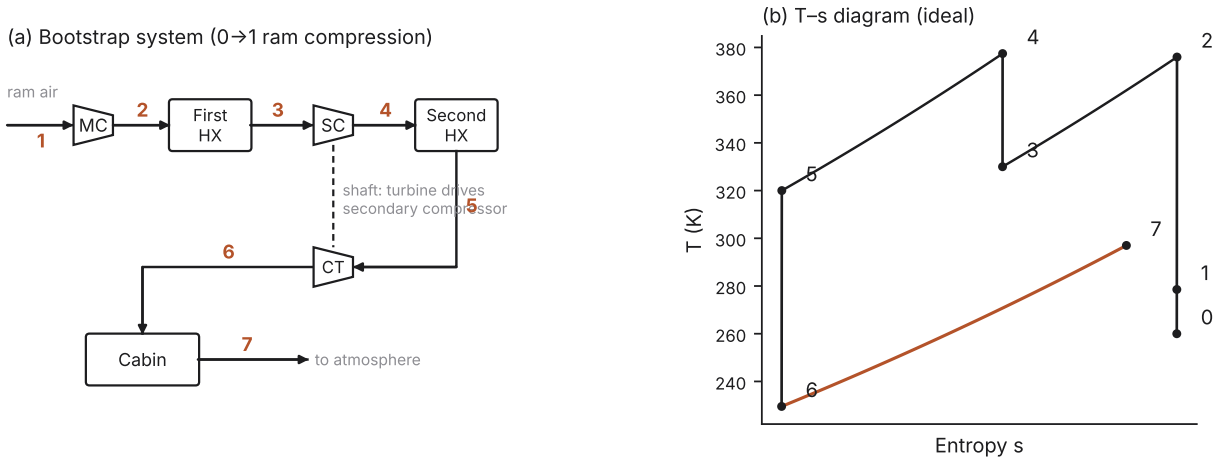

Fig. 3 — Bootstrap air-cycle refrigeration system for aircraft and its T–s diagram (ideal processes).

Working. The bootstrap system adds a **secondary compressor** driven by the cooling turbine, so the turbine work is used to raise the pressure further before final cooling. This gives a larger temperature drop in the turbine.

1. **0→1 Ram compression:** atmospheric air is rammed into the intake by the aircraft's speed (p and T rise).
2. **1→2 Main compressor** (bleed from the engine compressor) raises the pressure further.
3. **2→3 First heat exchanger:** compressed air is cooled at constant pressure by ram air.
4. **3→4 Secondary compressor** (driven by the turbine) compresses the air further.
5. **4→5 Second heat exchanger:** cooled again by ram air.
6. **5→6 Cooling turbine:** the air expands to cabin pressure, its temperature drops well below cabin temperature, and the work produced drives the secondary compressor.
7. **6→7 Cabin:** cold air absorbs the cabin heat load at constant pressure, then is exhausted.

$$\text{Refrigerating effect } = \dot{m}c_p(T_7 - T_6), \quad W_{\text{turbine}} = W_{\text{sec. comp.}} \Rightarrow \text{COP} = \frac{T_7 - T_6}{T_2 - T_1}$$

Use: transport aircraft, where ram-air temperature is too high for a simple system to give enough cooling. Its drawback is that it needs high aircraft speed for ram cooling. For that reason fans are added on the ground (the "bootstrap evaporative" variant also adds water evaporation).

Q7 (a) **A 5 TR air-conditioning plant (refrigerant R22) working on vapour compression refrigeration system operates between the temperature conditions of 6 °C and 40 °C. The refrigerant leaves the evaporator in superheated state at 10 °C and enters the compressor. The refrigerant leaves the condenser in saturated state. Calculate (i) mass flow rate of refrigerant in kg/s (ii) compressor power in kW and (iii) COP of the system.** [5]

ASSUMPTIONS

Isentropic compression; no pressure drops; superheat (4 K) takes place in the evaporator, so it counts as useful refrigeration; 1 TR = 3.5167 kW (211 kJ/min).

PROPERTIES OF R22 (FROM TABLES / CHART)

State	Condition	p (bar)	t (°C)	h (kJ/kg)	s (kJ/kg·K)
1	Superheated vapour at evaporator pressure, compressor inlet	6.03	10	410.25	1.7531
2	Superheated vapour, $s_2 = s_1$ at condenser pressure	15.34	58.9	433.84	1.7531
3	Saturated liquid at 40 °C	15.34	40	249.65	—
4	After throttling, $h_4 = h_3$	6.03	6	249.65	—

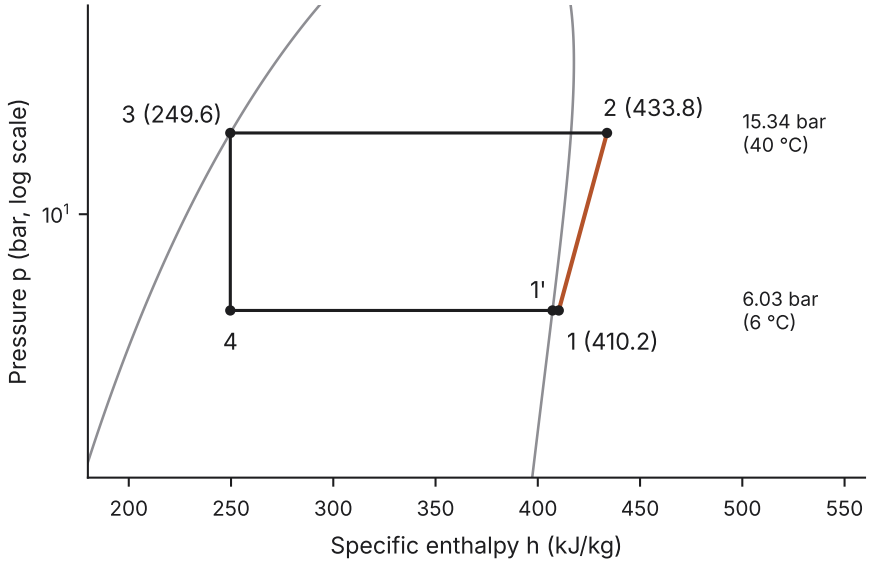

Fig. 4 — R22 cycle on the p–h chart (state 1′ = saturated vapour at 6 °C).

$$\text{Refrigerating effect } q_e = h_1 - h_4 = 410.25 - 249.65 = 160.60 \text{ kJ/kg}$$

$$\text{(i)} \quad \dot{m} = \frac{5 \times 3.5167}{160.60} = \mathbf{0.1095 \text{ kg/s}} \ (6.57 \text{ kg/min})$$

$$w_c = h_2 - h_1 = 433.84 - 410.25 = 23.59 \text{ kJ/kg}$$

$$\text{(ii)} \quad P = \dot{m}\, w_c = 0.1095 \times 23.59 = \mathbf{2.58 \text{ kW}}$$

$$\text{(iii)} \quad \text{COP} = \frac{q_e}{w_c} = \frac{160.60}{23.59} = \mathbf{6.81} \qquad \left(\text{COP}_{Carnot} = \frac{279.15}{313.15 - 279.15} = 8.21\right)$$

Answer: ṁ = 0.110 kg/s · compressor power = 2.58 kW · COP = 6.81. (Condenser heat = ṁ(h₂ − h₃) = 20.2 kW; discharge temperature ≈ 59 °C; suction volume flow ≈ 0.26 m³/min.)

Scoring tip — If your table's R22 enthalpies use a different datum (e.g. h = 0 at −40 °C), each h shifts by the same constant. Differences such as 160.6 and 23.6 kJ/kg, and so every final answer, stay the same. Always draw the p–h sketch and mark states 1–4.

Q7 (b) **Draw the schematic and P-h diagrams of a Cascade refrigeration system and briefly explain its working.** [5]

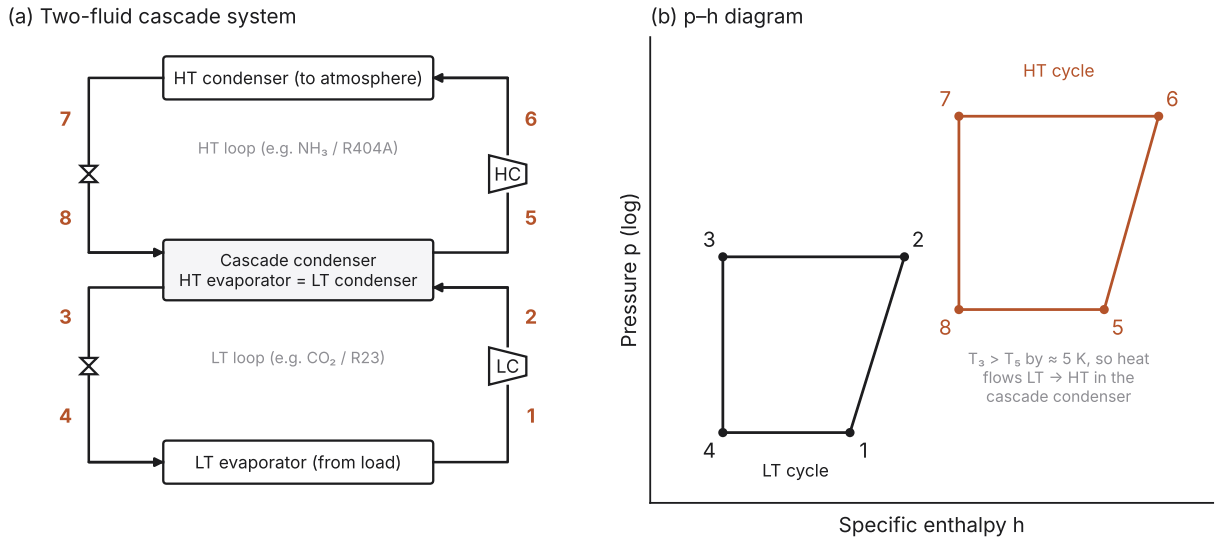

Fig. 5 — Two-stage cascade system: two independent VCR circuits coupled by a cascade condenser.

Need. For very low temperatures (−40 to −100 °C) one refrigerant cannot work over the whole range. A low-boiling refrigerant (CO_2, R23, ethane) would have enormous condensing pressure or be above its critical point at ambient temperature. A high-boiling refrigerant (NH_3, R404A) would have evaporator pressures far below atmospheric and huge specific volumes.

Working. Two (or more) separate VCR circuits, each with **its own refrigerant**, are connected in series thermally:

- **Low-temperature (LT) cycle 1-2-3-4:** its evaporator takes the refrigeration load at the lowest temperature. Its condenser rejects heat into the **cascade condenser**.
- **Cascade condenser:** a heat exchanger that is the condenser of the LT cycle and also the evaporator of the high-temperature (HT) cycle. The LT condensing temperature is kept about 5 K above the HT evaporating temperature so that heat flows.
- **High-temperature (HT) cycle 5-6-7-8:** it absorbs the heat $Q_e + W_{LT}$ and rejects it to the atmosphere in its condenser.

$$\dot{m}_{HT}(h_5 - h_8) = \dot{m}_{LT}(h_2 - h_3), \qquad \text{COP} = \frac{\dot{m}_{LT}(h_1 - h_4)}{\dot{m}_{LT}(h_2 - h_1) + \dot{m}_{HT}(h_6 - h_5)}$$

Advantages: a suitable refrigerant in each temperature range, moderate pressures and pressure ratios in each stage, no oil or refrigerant mixing between circuits, higher volumetric efficiency.
Disadvantages: extra heat exchanger and temperature overlap (irreversibility); more complex.
Applications: liquefaction of gases, blood and vaccine storage at −80 °C, environmental test chambers, freeze drying.

ME407 · Mid-Semester · Sep 2019

Refrigeration and Air-Conditioning · Time 1.5 h · Max. marks 30 · Attempt all questions · Assume suitable missing data, if any

Q1 **With the help of suitable T-s diagram, describe the processes and practical limitations of reversed Carnot cycle if refrigerant is: a) Gas, b) Vapour.** [3 + 3]

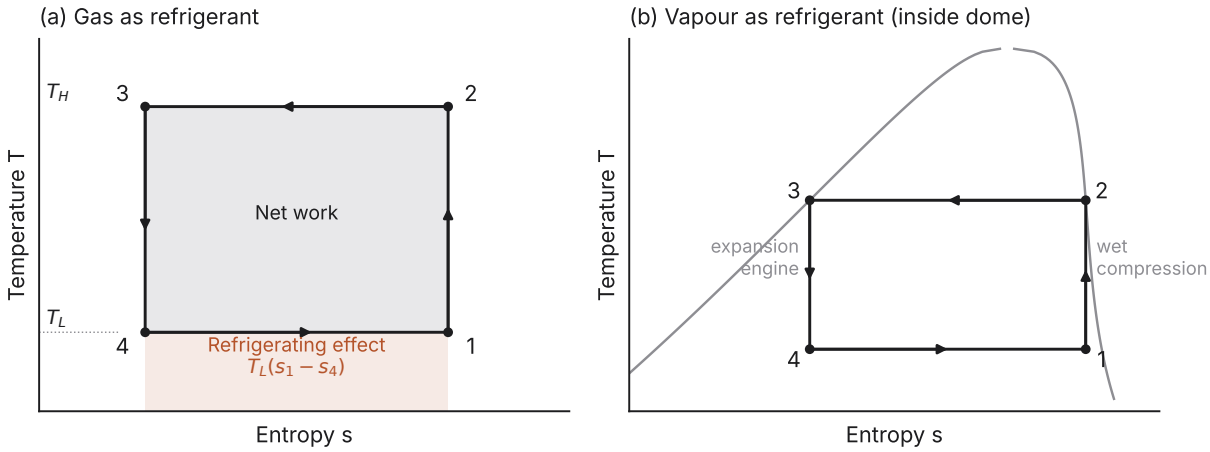

Fig. 6 — Reversed Carnot cycle: (a) with a gas, (b) with a vapour inside the saturation dome.

Processes (same for both): 1→2 isentropic compression from T_L to T_H; 2→3 isothermal heat rejection at T_H; 3→4 isentropic expansion from T_H to T_L; 4→1 isothermal heat absorption at T_L (refrigerating effect).

$$\mathrm{COP}_{Carnot} = \frac{Q_L}{W_{net}} = \frac{T_L(s_1 - s_4)}{(T_H - T_L)(s_1 - s_4)} = \frac{T_L}{T_H - T_L}$$

It is the **maximum COP** possible between two temperatures and depends only on them.

(a) Gas as refrigerant: practical limitations

- Isothermal heat transfer with a gas (2→3, 4→1) means heat must be added or removed *while* the gas is expanding or being compressed. That needs an extremely slow process (quasi-static) and very large heat-transfer area. But isentropic processes 1→2 and 3→4 must be very fast to be adiabatic. Both cannot be achieved in one machine.
- On the p–v diagram the isothermals and isentropics of a gas are nearly parallel, so the cycle area (net work and refrigerating effect) is very small for a large swept volume. The machine would be bulky, and mechanical friction would be comparable with the net work.
- So the gas cycle is modified into the **Bell–Coleman (reversed Brayton)** cycle with constant-pressure heat exchange.

(b) Vapour as refrigerant: practical limitations

- Isothermal heat transfer is practical here, because evaporation and condensation at constant pressure are also at constant temperature. This is a big advantage over the gas cycle.
- **Wet compression 1→2**: liquid droplets in the compressor wash away the lubricant from cylinder walls (wear), can cause liquid hammering and valve damage, and evaporate unevenly. The compressor is not isentropic in practice. So **dry (saturated or superheated) compression** is used instead.
- **Isentropic expansion 3→4** needs an expansion engine handling a wet mixture. Its work output is very small compared with the cost and complexity of the engine, so it is replaced by a simple **throttle (expansion) valve** ($h_3 = h_4$).

- Heat transfer through a zero temperature difference needs infinite area. Real evaporators and condensers need a finite ΔT, which lowers COP.
- These modifications produce the practical **vapour compression refrigeration cycle**.

Q2 **A simple saturation cycle (refrigerant is R12) operates at temperature of 35 °C and −15 °C for the condenser and evaporator. Determine the COP and HP/TR of the system. If a liquid-vapor heat exchanger is installed in the system, with the temperature of vapor leaving the heat exchanger at 15 °C, what will be the change in COP and HP/TR?** [2 + 4]

T (°C)	h_f	h_g	s_g	20 K superheat: h, s	40 K superheat: h, s
35 (p_c)	69.5	201.5	0.6839	216.4, 0.731	231.0, 0.7741
−15 (p_e)	22.3	181.0	0.7052	193.2, 0.751	205.7, 0.7942

Data given in the paper (h in kJ/kg, s in kJ/kg·K).

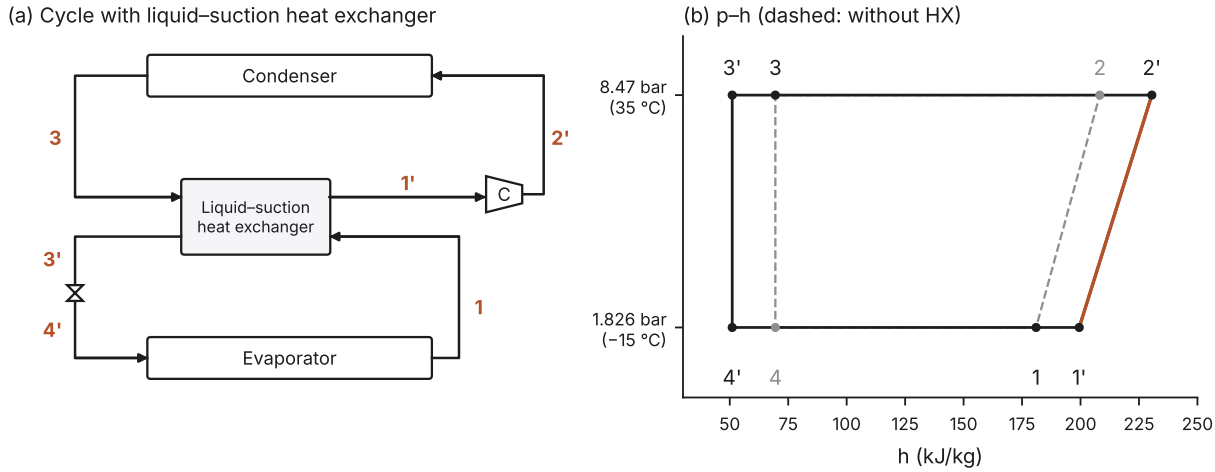

Fig. 7 — R12 cycle without (dashed) and with the liquid–suction heat exchanger (solid), using the given property values.

Part 1: simple saturation cycle

$h_1 = h_g(-15) = 181.0$, $s_1 = 0.7052$. Compression to 35 °C pressure: since $0.6839 < s_1 < 0.731$, interpolate between saturated vapour and 20 K superheat:

$$h_2 = 201.5 + \frac{0.7052 - 0.6839}{0.731 - 0.6839}(216.4 - 201.5) = 201.5 + 0.4522(14.9) = 208.24\ \text{kJ/kg}$$

$$h_3 = h_4 = h_f(35) = 69.5; \quad q_e = 181.0 - 69.5 = 111.5; \quad w = 208.24 - 181.0 = 27.24\ \text{kJ/kg}$$

$$\text{COP} = \frac{111.5}{27.24} = \mathbf{4.09}$$

$$\text{HP/TR} = \frac{3.5167}{\text{COP} \times 0.7457} = \frac{3.5167}{4.09 \times 0.7457} = \mathbf{1.152} \quad (\dot{m} = 0.03154\ \text{kg/s per TR},\ 0.859\ \text{kW/TR})$$

Part 2: with liquid–vapour (suction) heat exchanger

The vapour is superheated in the HX from −15 °C to 15 °C (30 K superheat at p_e). Interpolating midway between the 20 K and 40 K data:

$$h_{1'} = \tfrac{1}{2}(193.2 + 205.7) = 199.45\ \text{kJ/kg}, \qquad s_{1'} = \tfrac{1}{2}(0.751 + 0.7942) = 0.7726\ \text{kJ/kg·K}$$

Energy balance on the HX (heat lost by liquid = heat gained by vapour):

$$h_3 - h_{3'} = h_{1'} - h_1 \Rightarrow h_{3'} = 69.5 - (199.45 - 181.0) = 51.05\ \text{kJ/kg} = h_{4'}$$

Compression from $s_{1'} = 0.7726$ to p_c: between 20 K (0.731) and 40 K (0.7741) superheat:

$$h_{2'} = 216.4 + \frac{0.7726 - 0.731}{0.7741 - 0.731}(231.0 - 216.4) = 216.4 + 0.9652(14.6) = 230.49\ \text{kJ/kg}$$

$$q_e = h_1 - h_{4'} = 181.0 - 51.05 = 129.95; \quad w = h_{2'} - h_{1'} = 230.49 - 199.45 = 31.04\ \text{kJ/kg}$$

$$\text{COP} = \frac{129.95}{31.04} = \mathbf{4.19}; \qquad \text{HP/TR} = \frac{3.5167}{4.19 \times 0.7457} = \mathbf{1.127}$$

Cycle	q_e (kJ/kg)	w (kJ/kg)	COP	$\dot{m}$ per TR (kg/s)	HP/TR
Simple saturation	111.5	27.24	4.09	0.0315	1.152
With LSHX	129.95	31.04	4.19	0.0271	1.127
Change	+16.5 %	+14.0 %	**+2.3 %**	−14.2 %	**−2.2 %**

> **Answer:** Simple cycle COP = 4.09, HP/TR = 1.15. With the LSHX, COP = 4.19 (**increases by ≈ 2.3 %**) and HP/TR = 1.13 (**decreases by ≈ 2.2 %**). For R12 the LSHX is beneficial: the gain in refrigerating effect from subcooling outweighs the extra compressor work caused by superheated suction. The HX also makes sure no liquid reaches the compressor.

Q3 (a) Write the chemical formula of the following refrigerants: R11, R13B1, R134, R744. [3]

Rule: R(m−1)(n+1)(p) for $\text{C}_m\text{H}_n\text{F}_p\text{Cl}_q$, $q = 2m + 2 - n - p$. A suffix "B" followed by a number means that many Cl atoms are replaced by Br. 700-series: inorganic, 700 + molecular mass.

Refrigerant	Decoding	Formula and name
R11 (R011)	m = 1, n = 0, p = 1, q = 4 − 0 − 1 = 3	**CCl_3F**, trichlorofluoromethane (CFC)
R13B1 (R013B1)	m = 1, n = 0, p = 3, q = 1 → the one Cl replaced by Br	**$CBrF_3$**, bromotrifluoromethane (halon 1301)
R134	m = 2, n = 2, p = 4, q = 6 − 2 − 4 = 0	**$C_2H_2F_4$ = CHF_2–CHF_2**, 1,1,2,2-tetrafluoroethane (symmetric isomer; R134a is CH_2F–CF_3)
R744	700 + 44 (molecular mass)	**CO_2**, carbon dioxide

Q3 (b) Briefly describe the meaning of ODP and GWP of CFC refrigerants. [3]

ODP (Ozone Depletion Potential): the ratio of the stratospheric ozone destroyed by a unit mass of a substance to that destroyed by the same mass of **R11 (ODP = 1)**. CFCs are very stable, so they reach the stratosphere. There UV radiation frees chlorine atoms, which destroy ozone catalytically (Cl + O_3 → ClO + O_2; ClO + O → Cl + O_2). One Cl atom can destroy about 10^5 O_3 molecules. CFCs have high ODP (R11 = 1, R12 = 1). HCFCs are lower (R22 ≈ 0.055). HFCs, NH_3 and CO_2 are 0.

GWP (Global Warming Potential): the heat trapped in the atmosphere by a unit mass of a gas over a time horizon (usually 100 years), relative to the same mass of **CO_2 (GWP = 1)**. CFCs absorb infrared radiation strongly and stay in the atmosphere for a long time, so their GWP is very high (R11 ≈ 4,750; R12 ≈ 10,900; for comparison R134a ≈ 1,430).

For these reasons CFCs were phased out under the **Montreal Protocol (1987)**, and high-GWP HFCs are being phased down under the **Kigali Amendment (2016)**. TEWI (Total Equivalent Warming Impact) adds the indirect CO_2 from the energy a system uses.

Q4 With the help of suitable schematic and T-s diagram, describe the Simple and Bootstrap system of aircraft refrigeration. [6]

Why air cycle in aircraft: air is light, free and non-toxic, and needs no heavy refrigerant circuit. Compressed air is readily available from the engine compressor and can also pressurise the cabin. The weight per TR is low, even though COP is low.

(i) Simple air-cooling system

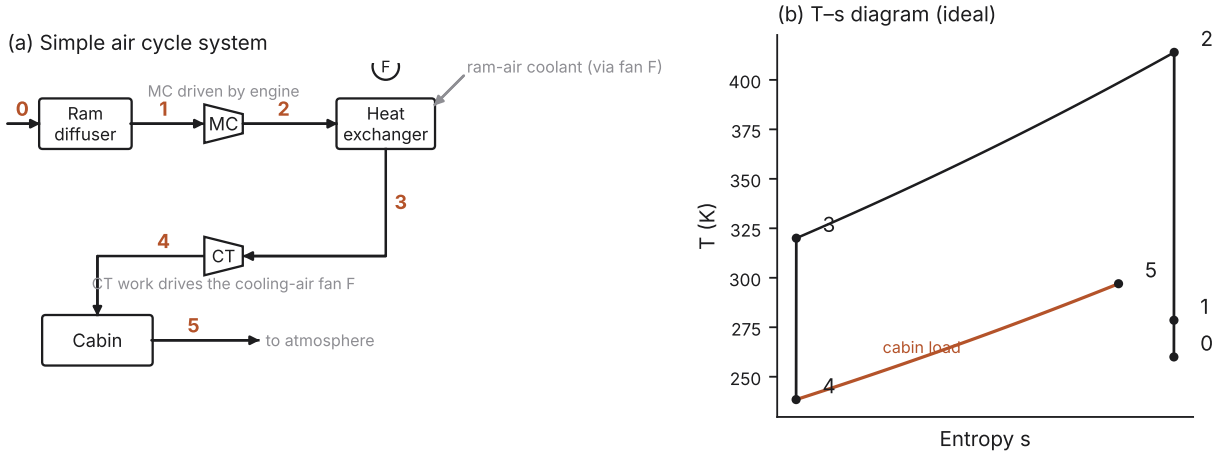

Fig. 8 — Simple air-cycle system and T–s diagram (2′, 4′ would denote actual non-isentropic states).

- 0→1: ram compression of ambient air in the diffuser by the forward speed of the aircraft.
- 1→2: further compression in the main compressor (engine-driven or engine bleed).
- 2→3: cooling at constant pressure in a heat exchanger by ram air. A fan, driven by the cooling turbine, draws the ram air through the HX.
- 3→4: expansion in the cooling turbine to cabin pressure. T_4 is well below cabin temperature.
- 4→5: the cold air absorbs the cabin heat load and leaves at T_5 (cabin temperature).

$$\text{RE} = \dot{m}c_p(T_5 - T_4), \quad W = \dot{m}c_p(T_2 - T_1), \quad \text{COP} = \frac{T_5 - T_4}{T_2 - T_1}$$

Suitable for: fighter or high-speed aircraft and ground cooling (it needs a good ram-air supply). Simple and light.

(ii) Bootstrap system

Here the cooling turbine drives a **secondary compressor** instead of a fan. After the first heat exchanger (2→3) the air is compressed again (3→4), cooled in a second heat exchanger (4→5) and expanded in the turbine (5→6) to cabin pressure. Because the turbine inlet pressure is higher, the temperature drop is larger. Schematic and T–s diagram: Fig. 3 (Paper 1, Q6b).

$$\text{COP} = \frac{T_7 - T_6}{T_2 - T_1} \quad (\text{only the main compressor work is external})$$

Suitable for: transport aircraft. Its drawback is that it needs high flight speed for enough ram cooling, so it is unsuitable for ground cooling without auxiliary fans.

Feature	Simple	Bootstrap
Turbine work used for	Driving the cooling fan	Driving the secondary compressor
Heat exchangers	One	Two (primary and secondary)
Ground cooling	Possible (fan)	Poor (needs ram air)
Typical use	Fighter / high-speed aircraft	Transport / cargo aircraft

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Q5 (a) **What do you mean by multistage compression? Describe the necessity of multistaging in VCR cycle.** [1 + 2]

Multistage compression means raising the refrigerant vapour from evaporator to condenser pressure in two or more compressors (or stages) in series, with intercooling and/or flash-gas removal between stages.

Necessity: when the evaporator temperature is low or the condenser temperature high, the pressure ratio p_c/p_e becomes large (more than about 8–10). In a single stage this causes:

- **Low volumetric efficiency** from re-expansion of clearance vapour, $\eta_v = 1 + C - C(p_c/p_e)^{1/n}$. Capacity falls steeply, and at very high ratios it falls to zero.
- **Very high discharge temperature**: oil carbonises and breaks down, valves are damaged, and the refrigerant (especially NH_3) decomposes.
- Large **flash gas** after throttling, so the refrigerating effect is low.
- **High power** per TR. Compression is far from isothermal.

Multistaging with intercooling brings compression closer to isothermal (less work). It removes flash gas (more refrigerating effect, lower LP compressor flow), raises volumetric efficiency, lowers discharge temperature, and allows different evaporator temperatures. Optimum intermediate pressure: $p_i = \sqrt{p_e\, p_c}$. Rule of thumb: NH_3 single stage down to about −25 to −30 °C; below that, two stages.

Q5 (b) **With the help of suitable schematic and P-h diagram, describe the flash intercooling.** [3]

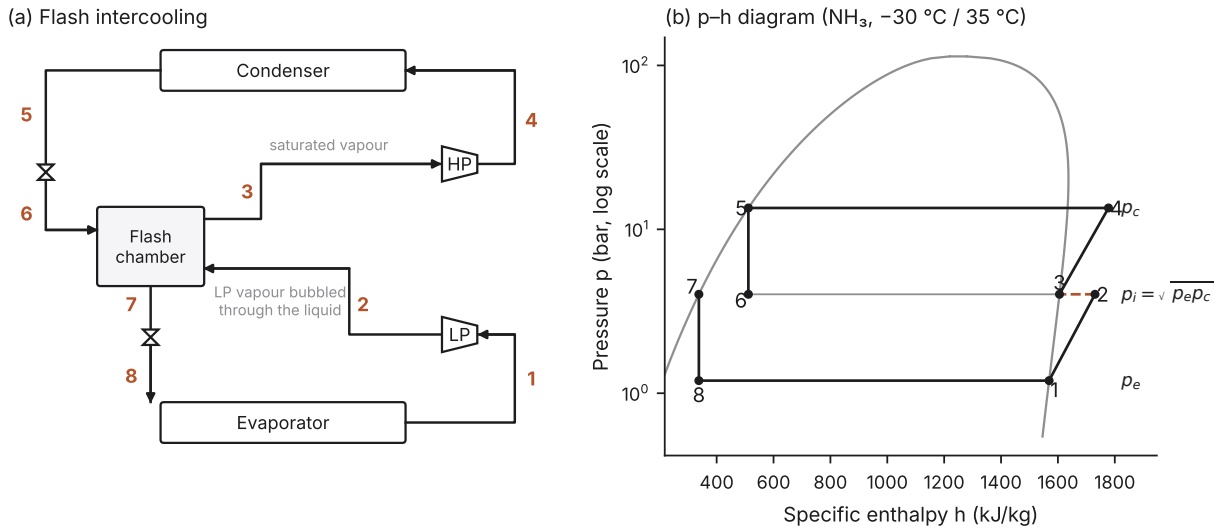

Fig. 9 — Two-stage compression with flash intercooler (flash chamber at intermediate pressure p_i).

Working. Liquid from the condenser (5) is throttled to the intermediate pressure (6) into the flash chamber. The flash vapour separates. The superheated discharge of the LP compressor (2) is **bubbled through the liquid** in the flash chamber. There it is desuperheated to saturated vapour (3) by evaporating some of the liquid. This is the **flash intercooling**. Saturated vapour (3), made up of flash vapour, desuperheated LP vapour and the vapour formed by intercooling, goes to the HP compressor (3→4). Saturated liquid (7) is throttled to the evaporator (8). So it gives both **intercooling** (lower HP work and discharge temperature) and **flash-gas removal** (larger refrigerating effect $h_1 - h_8$).

$$\dot{m}_1 = \frac{Q_e}{h_1 - h_8}; \qquad \text{flash-chamber balance: } \dot{m}_1 h_2 + \dot{m}_2 h_5 = \dot{m}_1 h_7 + \dot{m}_2 h_3 \Rightarrow \dot{m}_2 = \dot{m}_1 \frac{h_2 - h_7}{h_3 - h_5}$$

$$\text{COP} = \frac{\dot{m}_1 (h_1 - h_8)}{\dot{m}_1 (h_2 - h_1) + \dot{m}_2 (h_4 - h_3)}$$

ME359 · Mid-Semester · Sep 2019

Refrigeration and Air Conditioning · Time 1.5 h · Max. marks 25 · Answer any five questions · Refrigeration tables and charts allowed · Assume suitable missing data

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Q1 **Explain the reason for variation in condenser pressure. Discuss its effect on the performance of the vapour compression refrigeration system with the help of P-h diagram.** [4]

Reasons for variation in condenser pressure (condenser pressure = saturation pressure at the condensing temperature):

- Seasonal or daily change in **ambient air / cooling-water temperature** (summer gives higher p_c).
- **Reduced cooling-water flow** or condenser fan failure; blocked or dirty air-cooled fins.
- **Fouling / scaling** of condenser tubes (lower U).
- **Non-condensable gases** (air) in the condenser, which add their partial pressure.
- **Overcharge** of refrigerant: liquid backs up and floods condenser area.
- Increased heat load (higher mass flow), or an undersized condenser.

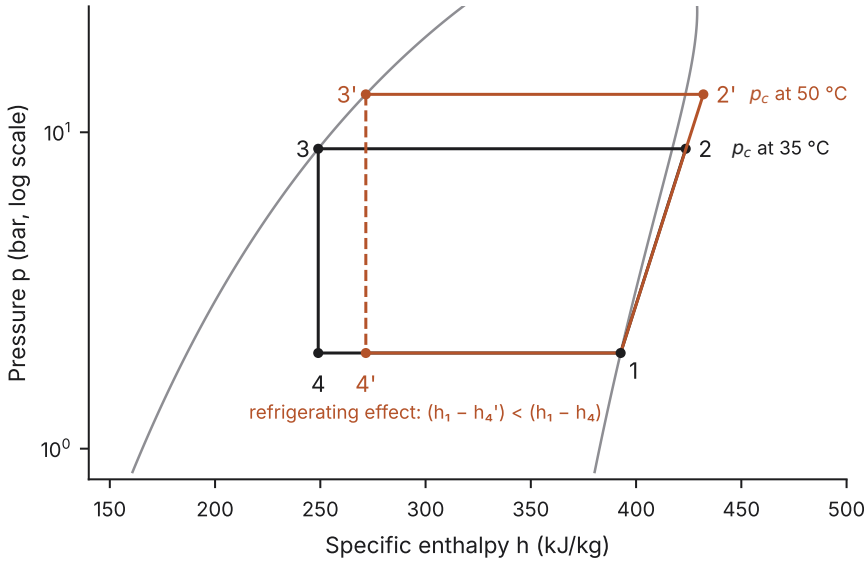

Fig. 10 — Rise in condenser pressure (35 °C → 50 °C) at fixed evaporator pressure: cycle 1-2-3-4 becomes 1-2′-3′-4′.

Effects of an increase in p_c (evaporator pressure constant):

- Compressor work rises: $h_{2'} - h_1 > h_2 - h_1$.
- Refrigerating effect falls, because the throttling line moves right: $h_1 - h_{4'} < h_1 - h_4$.
- So **COP falls** and **power per TR rises**. Mass flow per TR rises.
- Higher pressure ratio, so volumetric efficiency and capacity fall. Discharge temperature rises (oil breakdown).

$$\text{COP} = \frac{h_1 - h_4}{h_2 - h_1} \downarrow \quad \text{as } p_c \uparrow$$

A **decrease** in p_c has the opposite effect (higher COP), but p_c cannot be lowered too much, because the expansion valve needs a minimum pressure difference to feed the evaporator. (Example for R134a at −10 °C evaporation: raising condensing from 35 to 50 °C lowers the ideal COP from about 4.6 to about 3.1.)

Q2 **In a 5 TR vapour compression system used as an ice plant working between evaporator and condenser temperatures of −10 °C and 35 °C respectively, the dryness fraction of the refrigerant at discharge from compressor is 0.97. The refrigerant leaves the condenser in the saturated liquid state. Calculate specific refrigeration effect, specific compressor work, mass flow rate of refrigerant (kg/s) and COP of the system.** [6]

T_{sat} (°C)	h_f (kJ/kg)	h_g (kJ/kg)	s_f (kJ/kg·K)	s_g (kJ/kg·K)
−10	154.056	1450.22	0.8297	5.755
35	366.072	1488.57	1.566	5.2086

Data given in the paper (these are NH_3 properties). The compression is wet: it ends at $x_2 = 0.97$ at condenser pressure and is assumed isentropic.

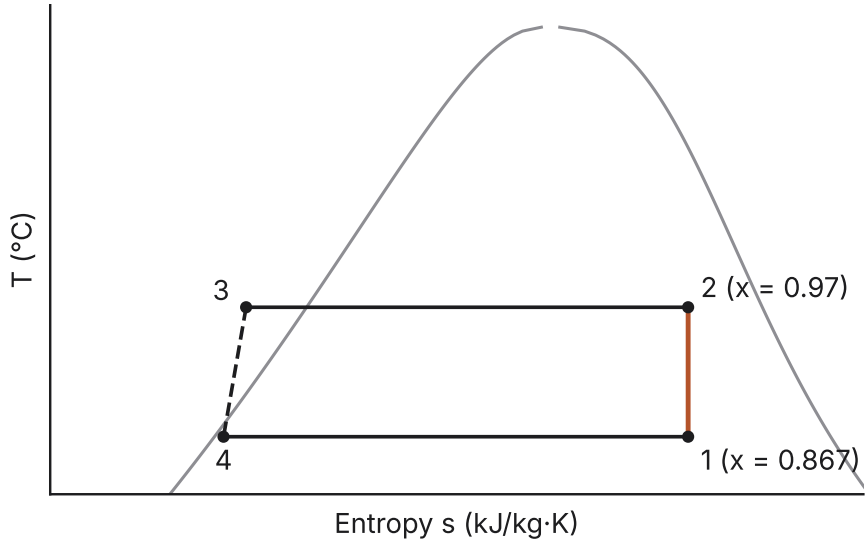

Fig. 11 — Wet compression cycle on T–s: 1→2 isentropic, $x_1 = 0.867 \rightarrow x_2 = 0.97$.

STATE 2 (COMPRESSOR DISCHARGE, 35 °C, X = 0.97)

$$h_2 = 366.072 + 0.97(1488.57 - 366.072) = 1454.90 \text{ kJ/kg}$$

$$s_2 = 1.566 + 0.97(5.2086 - 1.566) = 5.0993 \text{ kJ/kg·K}$$

STATE 1 (COMPRESSOR SUCTION, −10 °C), $S_1 = S_2$

$$x_1 = \frac{5.0993 - 0.8297}{5.755 - 0.8297} = 0.8669; \qquad h_1 = 154.056 + 0.8669(1450.22 - 154.056) = 1277.67 \text{ kJ/kg}$$

STATES 3, 4: SATURATED LIQUID AT 35 °C, THROTTLED: $H_4 = H_3 = 366.07$ KJ/KG

$$\text{Specific refrigerating effect } = h_1 - h_4 = 1277.67 - 366.07 = \mathbf{911.6 \text{ kJ/kg}}$$

$$\text{Specific compressor work } = h_2 - h_1 = 1454.90 - 1277.67 = \mathbf{177.2 \text{ kJ/kg}}$$

$$\dot{m} = \frac{5 \times 3.5167}{911.6} = \mathbf{0.0193 \text{ kg/s}} \ (1.157 \text{ kg/min})$$

$$\text{COP} = \frac{911.6}{177.2} = \mathbf{5.14} \qquad (\text{Power} = 0.0193 \times 177.2 = 3.42 \text{ kW})$$

> **Answer:** RE = 911.6 kJ/kg · W = 177.2 kJ/kg · ṁ = 0.0193 kg/s · COP = 5.14 (compressor power 3.42 kW; COP_{Carnot} = 263.15/45 = 5.85).

If 1 TR is taken as 210 kJ/min (3.5 kW): ṁ = 0.0192 kg/s. Every other result is unchanged.

Q3 **What is the difference between dry and wet compression?** [3]

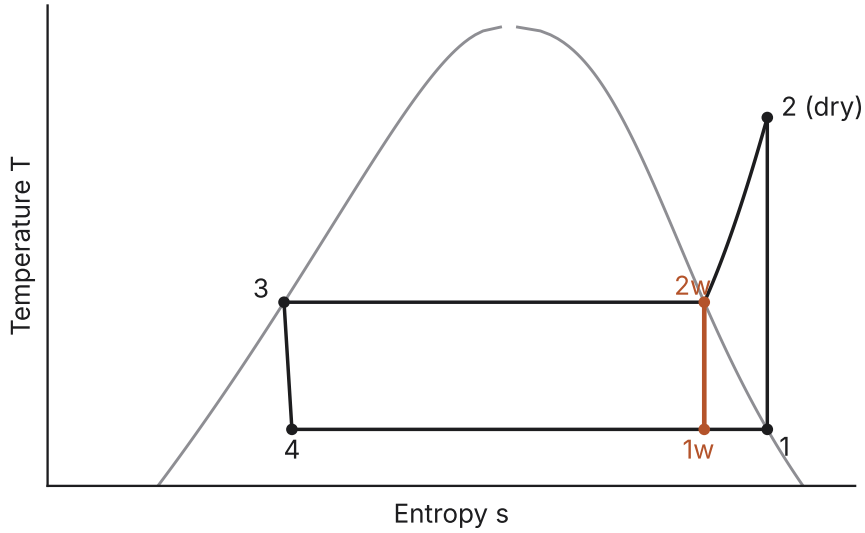

Fig. 12 — Dry compression 1→2 (saturated vapour in, superheated out) versus wet compression 1w→2w (wet vapour in, dry saturated out).

Aspect	Dry compression	Wet compression
Suction state	Dry saturated (or superheated) vapour	Wet vapour ($x < 1$)
Discharge state	Superheated vapour	Dry saturated (or still wet) vapour
Refrigerating effect	Larger (full evaporation, $h_1 = h_g$)	Smaller (refrigerant leaves evaporator partly liquid)
COP (theory)	Slightly lower than Carnot because of the superheat horn	Closer to Carnot in theory
Practical COP	Higher: good volumetric efficiency	Lower: liquid evaporating in the cylinder reduces η_v
Compressor safety	Safe; no liquid	Liquid droplets wash off lubricant, cause wear, liquid hammering and valve damage
Discharge temperature	Higher	Lower (useful for NH_3, but risky)
Practice	Always preferred (often with slight superheat for safety)	Avoided in reciprocating compressors

Q4 **Write six desirable properties of refrigerants. Name the conventional and alternative refrigerant for a refrigerator.** [3]

1. Low boiling point, with evaporator pressure slightly above atmospheric (no air or moisture ingress).
2. Moderate condensing pressure (light and cheap equipment) and high critical temperature.
3. High latent heat of vaporisation and low specific volume of vapour (small mass flow, small compressor).
4. Non-toxic, non-flammable, non-corrosive and chemically stable; compatible with oil.
5. High thermal conductivity, low viscosity, and leaks easy to detect.
6. Zero ODP and low GWP; low cost and easily available.

Domestic refrigerator: conventional refrigerant **R12 (CCl_2F_2, CFC)**, now banned (high ODP). Alternatives: **R134a (CH_2FCF_3, HFC, ODP 0)** and, in most new Indian refrigerators, **R600a**

(isobutane, ODP 0, GWP ≈ 3). Newer options include R1234yf (HFO).

Q5 **When is it recommended to go for multistage compression? Explain with reason.** [3]

Multistage compression is recommended when the **pressure ratio p_c/p_e is high** (more than about 8–10). This happens with very low evaporator temperatures (e.g. NH_3 below about −25 to −30 °C), high condensing temperatures, or when several evaporators work at different temperatures. Reasons:

- **Volumetric efficiency**: $\eta_v = 1 + C - C(p_c/p_e)^{1/n}$ falls sharply with pressure ratio. Splitting the ratio, e.g. $\sqrt{p_c/p_e}$ per stage, keeps it high.
- **Discharge temperature**: single-stage discharge becomes very high (oil carbonisation, NH_3 dissociation, valve failure). Intercooling between stages limits it.
- **Power saving**: intercooling makes compression closer to isothermal, so less work is needed.
- **Flash-gas removal** at intermediate pressure increases the refrigerating effect and reduces the vapour the LP compressor must handle.
- Mechanical balance and smaller cylinder sizes; it also allows multiple-temperature loads.

Q6 **Draw a neat sketch of a two stage compression refrigeration system with flash gas removal. Also draw P-h diagram and specify relevant equations for calculation of COP.** [4]

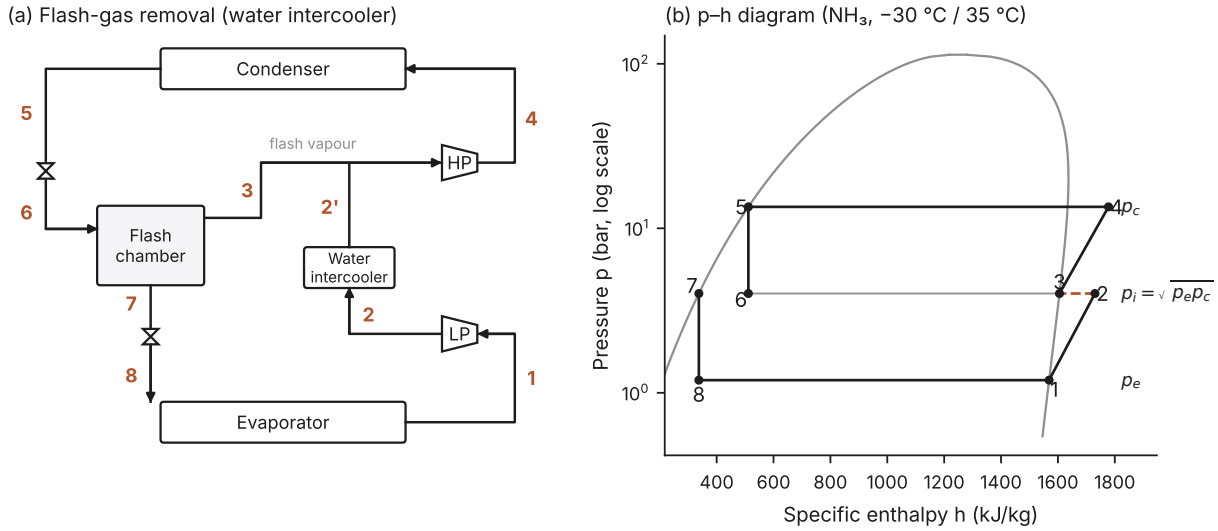

Fig. 13 — Two-stage compression with flash-gas removal (flash chamber) and water intercooler between stages.

Working. Saturated liquid from the condenser (5) is throttled to intermediate pressure p_i (6). In the flash chamber the **flash vapour (3) is separated and sent straight to the HP compressor**, so it does not pass through the evaporator. Only saturated liquid (7) is throttled to the evaporator (8), so the refrigerating effect rises from $h_1 - h_5$ to $h_1 - h_7$. The LP compressor (1→2) discharges through a water intercooler (2→2′). This vapour mixes with the flash vapour before the HP compressor (→4).

EQUATIONS (REFRIGERATION CAPACITY Q_E, KW)

$$\dot{m}_{LP} = \frac{Q_e}{h_1 - h_8} \quad (h_8 = h_7)$$

$$x_6 = \frac{h_6 - h_7}{h_3 - h_7}\ (h_6 = h_5), \qquad \dot{m}_{HP} = \frac{\dot{m}_{LP}}{1 - x_6}, \qquad \dot{m}_{flash} = \dot{m}_{HP}\, x_6$$

$$\text{HP suction (mixing): } h_m = \frac{\dot{m}_{LP} h_{2'} + \dot{m}_{flash} h_3}{\dot{m}_{HP}}\ (= h_3 \text{ if cooled to saturation})$$

$$W_{LP} = \dot{m}_{LP}(h_2 - h_1), \qquad W_{HP} = \dot{m}_{HP}(h_4 - h_m)$$

$$\boxed{\text{COP} = \frac{\dot{m}_{LP}(h_1 - h_8)}{\dot{m}_{LP}(h_2 - h_1) + \dot{m}_{HP}(h_4 - h_m)}} \qquad p_i = \sqrt{p_e\, p_c}\ \text{(optimum)}$$

For comparison, Fig. 9 (Paper 2, Q5b) shows the flash-intercooler variant, in which the LP vapour is bubbled through the flash-chamber liquid instead of passing through a water intercooler.

APPENDIX A

Formula sheet

Everything used above, on one page.

BASICS

1 TR = 3.5167 kW = 211 kJ/min (≈ 210 kJ/min, 3.5 kW) · 1 HP = 0.7457 kW

$\mathrm{COP}_R = Q_L/W$, $\mathrm{COP}_{HP} = \mathrm{COP}_R + 1$, $\mathrm{COP}_{Carnot} = T_L/(T_H - T_L)$

$\mathrm{HP/TR} = 3.5167/(0.7457\,\mathrm{COP}) = 4.716/\mathrm{COP}$

SIMPLE VCR (1 SUCTION, 2 DISCHARGE, 3 LIQUID, 4 AFTER VALVE)

RE = $h_1 - h_4$, W = $h_2 - h_1$, $h_3 = h_4$, $\dot{m}$ = Q/RE, $Q_C = \dot{m}(h_2 - h_3)$

Wet mixture: $h = h_f + x\,h_{fg}$, $s = s_f + x\,s_{fg}$

Superheat: $h = h_g + c_{p,v}(T - T_{sat})$; subcool: $h = h_f - c_{p,l}\Delta T$

$\eta_v = 1 + C - C\,(p_c/p_e)^{1/n}$ · optimum $p_i = \sqrt{p_e p_c}$

LSHX: $h_3 - h_{3'} = h_{1'} - h_1$

AIR REFRIGERATION

$T_2/T_1 = (p_2/p_1)^{(\gamma-1)/\gamma}$ · Bell–Coleman $\mathrm{COP} = 1/(r_p^{(\gamma-1)/\gamma} - 1)$

Simple: COP = $(T_5 - T_4)/(T_2 - T_1)$ · Bootstrap: $(T_7 - T_6)/(T_2 - T_1)$

ABSORPTION

$$\mathrm{COP}_{max} = \frac{T_e(T_g - T_0)}{T_g(T_0 - T_e)}$$

PSYCHROMETRY (P = 101.325 KPA)

$\omega = 0.622\,p_v/(p - p_v)$ · $\phi = p_v/p_s$ · $\mu = \omega/\omega_s$

$h = 1.005t + \omega(2500 + 1.88t)$ kJ/kg d.a.

Apjohn / Carrier: $p_v = p'_v - \dfrac{1.8\,p\,(t - t')}{2700}$

DPT = $t_{sat}(p_v)$ · humid specific heat $c_{pa} = 1.005 + 1.88\omega \approx 1.022$

Sensible heat $= \dot{m}(h_A - h_1) \approx \dot{m}\,c_{pa}\,\Delta t$ · Latent $= \dot{m}\,\Delta\omega\,h_{fg}$ (2500)

$\mathrm{SHF} = \mathrm{SH}/(\mathrm{SH} + \mathrm{LH})$

BPF $= \dfrac{t_2 - t_{ADP}}{t_1 - t_{ADP}}$, contact factor = 1 − BPF

Mixing: $\omega_3 = \dfrac{\sum \dot{m}\omega}{\sum \dot{m}}$, $h_3 = \dfrac{\sum \dot{m}h}{\sum \dot{m}}$

Humidifying efficiency $= (t_1 - t_2)/(t_1 - t')$

Room loads: $\mathrm{SH} = 0.0204\,Q\,\Delta t$ kW, $\mathrm{LH} = 50\,Q\,\Delta\omega$ kW (Q in m³/min, ρ ≈ 1.2 kg/m³)

REFRIGERANT NUMBERING

$\mathrm{C}_m\mathrm{H}_n\mathrm{F}_p\mathrm{Cl}_q$ → R(m−1)(n+1)(p), $q = 2m + 2 - n - p$; "B" = Br replacing Cl; 500-series azeotropes; 400 zeotropes; 600 organics (R600a isobutane); 700 + M inorganic (R717 NH_3, R718 H_2O, R744 CO_2)

APPENDIX B

Original question papers

As found in the shared folder, kept for verification of the question text.

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(VII SEMESTER)

B.Tech.

END TERM EXAMINATION *Nov-2024*

COURSE CODE ME407 (Refrigeration and Air-Conditioning)

Time: 3:00 Hours *Max. Marks: 50*

Note : All questions are compulsory. Attempt any FIVE question. All questions carry equal marks. Assume suitable missing data, if any. Use of refrigeration tables and chart is allowed.

Q.1(a) An air sample has DBT and WBT as 25°C and 20°C. Calculate without using psychrometric chart (i) Relative humidity (ii) DPT (iii) Partial pressure of water vapour (iv) Enthalpy (v) DPT. Use Apjohn equation $P_v = P'_v - \frac{1.8\,p\,(t-t')}{2700}$. [5][CO4]

(b) Define specific humidity and prove that it given by $0.622\frac{P_v}{P-P_v}$ where P is barometric pressure. [5][CO4]

2(a) Explain the process of cooling and humidification (adiabatic humidification only). Specify the season for which this process is suitable. [5][CO4]

(b) The 25 kg/ min of air at a DBT of 25°C and 50% RH is heated and humidified such that the WBT of air at exit is 26°C and the increase in water vapour is 0.01 kg / kg of d.a. Calculate (i) sensible and latent heat added to air (ii) SHF. [5][CO4]

3(a) The air at 25°C and 50% RH is passed through a spray washer at 4°C. Specify the process which will take place during this interaction of air and spray washer. Assume bypass factor as 0.1 for spray washer, calculate (i) heat added or removed from the air (ii) final condition of air. [5][CO4]

(b) 250 kg/h of air saturated at 2°C is adiabatically mixed with 50 kg/h of air at 35°C and 40% RH. Determine the final state of air. [5][CO4]

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Q.4(a) What is meant by effective temperature? What are the factors o which effective temperature depends? [5][CO

(b) (i) What are the various duct design methods? [2.5][C(

(b)(ii) What are various types of expansion devices? [2.5][CC

Q.5 (a) What are the desirable characteristics of a refrigerant? [5][CC

(b) Decode the chemical name of the following refrigerant (i) R22 (ii 717. (iii) Can CO_2 be used as a refrigerant? [5][CC

6(a) Explain the working of a vapour absorption system with the help c neat sketch. [5][CO

(b) Explain the working of a Boot Strap system with the help of neat sket Also draw the T-s diagram. [5][CO

7(a) A 5 TR Air-Conditioning plant (refrigerant R22) working on vap compression refrigeration system operates between the temperat conditions of 6°C and 40° C. The refrigerant leaves the evaporator superheated state at 10° C and enters the compressor. The refrigerant lea the condenser in saturated state. Calculate (i) mass flow rate of refrigerant in kg/s (ii) Compressor power in kW and (iii) COP of system. [5][CC

(b) Draw the schematic and P-h diagrams of a Cascade refrigeration syst and briefly explain its working. [5][CC

ME407 End-Term Nov 2024, page 2 (compilation p. 435). The right margin is cut off in the source scan.

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7TH SEMESTER

B.Tech.

MID SEMESTER EXAMINATION

SEPT-2019

ME407; REFRIGERATION AND AIR-CONDITIONING

Time: 1:30 Hours

Max. Marks: 30

Note : Attempt all question. Assume suitable missing data, if any.

Q.1 With the help of suitable T-s diagram, describe the processes and practical limitations of reversed Carnot cycle if refrigerant is: a) Gas, b) Vapor (3+3)

Q.2 A simple saturation cycle (refrigerant is R12) operates at temperature of 35°C and -15°C for the condenser and evaporator. Determine the COP and HP/TR of the system.
If a liquid-vapor heat exchanger is installed in the system, with the temperature of vapor leaving the heat exchanger at 15°C. What will be change in COP and HP/TR

The property of R12 are given as:

T(°C)	h_f	h_g	s_g	For 20K Superheat		For 40K Superheat	
				h	s	h	s
35	69.5	201.5	0.6839	216.4	0.731	231.0	0.7741
-15	22.3	181	0.7052	193.2	0.751	205.7	0.7942

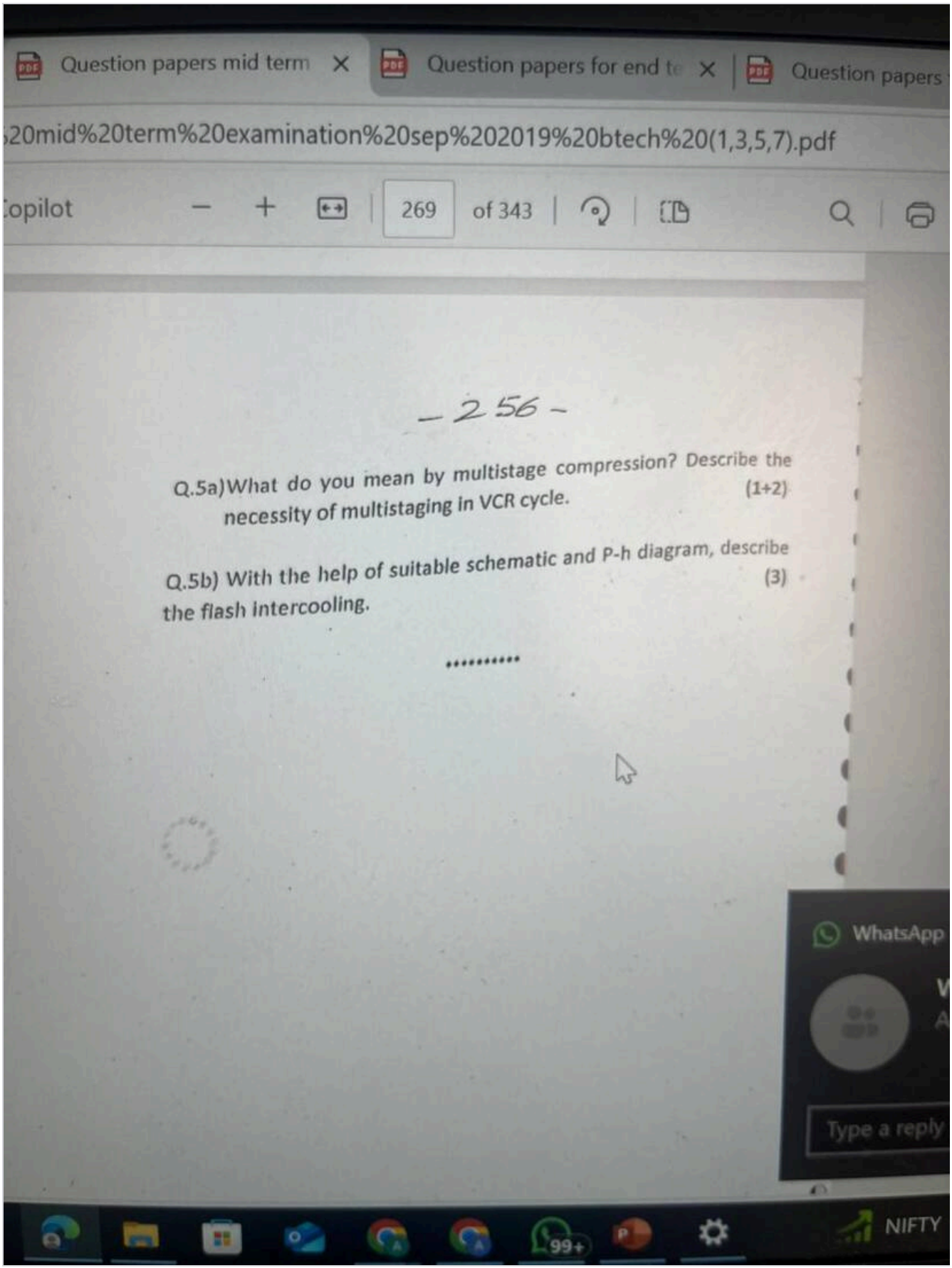
Unit of enthalpy and entropy are kJ/kg and kJ/kg-K respectively (2+4)

Q.3 a) Write the chemical formula of the following refrigerants: R11, R13B1, R134, R744 (3)

Q.3 b) Briefly describe the meaning of ODP and GWP of CFC refrigerants (3)

Q.4 With the help of suitable schematic and T-s diagram, describe the Simple and Bootstrap system of aircraft refrigeration. (6)

ME407 Mid-Sem Sep 2019, page 1 (photo of compilation p. 268)



ME407 Mid-Sem Sep 2019, page 2 (photo of compilation p. 269)

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5th SEMESTER

B.Tech.

MID SEMESTER EXAMINATION SEP.2019

ME359 : REFRIGERATION AND AIR CONDITIONING

Time: 1:30 Hours *Max. Marks: 25*

Note : Answer any five questions. Use of refrigeration tables and charts is allowed. All questions carry equal marks. Assume suitable missing data, If any.

1. Explain the reason for variation in condenser pressure. Discuss its effect on the performance of the vapour compression refrigeration system with the help of P-h diagram. 4

2. In a 5 TR vapour compression system used as an ice plant working between evaporator and condenser temperatures of -10° C and 35° C respectively. The dryness fraction of the refrigerant at discharge from compressor is 0.97. The refrigerant leaves the condenser in the saturated liquid state. Calculate specific refrigeration effect, specific compressor work, mass flow rate of refrigerant (kg/s) and COP of the system. The following properties of the refrigerant may be used. 6

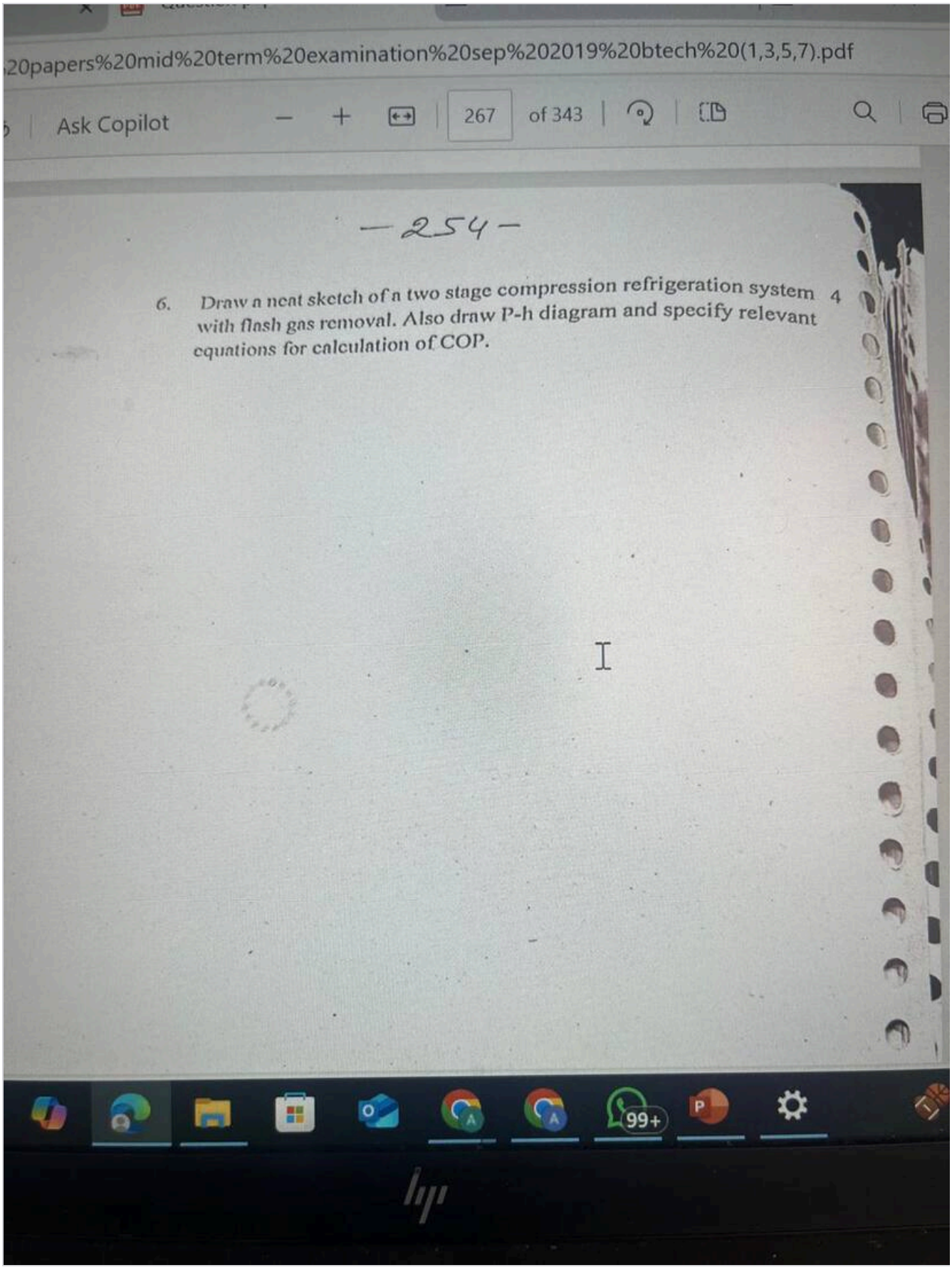
Tsat (C)	h_f (kJ/kg)	h_g (kJ/kg)	S_f (kJ/kgK)	S_g (kJ/kgK)
-10	154.056	1450.22	0.8297	5.755
35	366.072	1488.57	1.566	5.2086

3. What is the difference between dry and wet compression? 3

4 Write six desirable properties of refrigerants. Name the conventional and alternative refrigerant for a refrigerator. 3

5. When is it recommended to go for multistage compression? Explain with reason. 3

ME359 Mid-Sem Sep 2019, page 1 (photo of compilation p. 266)



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