

Heating Ventilation Air Conditioning (315373)

Model Question Paper — Set B · Model Answers

Q.1

a) Wet bulb depression and degree of saturation.

2 M

Wet bulb depression (WBD): The difference between dry bulb temperature and wet bulb temperature, $WBD = DBT - WBT$. It is zero for saturated air; larger WBD means drier air.

Degree of saturation (μ): The ratio of the specific humidity of moist air to the specific humidity of saturated air at the same temperature, $\mu = \omega/\omega_s$. It is 0 for dry air and 1 for saturated air.

(Each definition 1 M)

b) Sensible Heat Factor (SHF).

2 M

Sensible heat factor (SHF) is the ratio of sensible heat to the total heat (sensible + latent) removed or added in a process / room: $SHF = SH/(SH + LH)$.

Example: if a room has 12 kW sensible and 3 kW latent load, $SHF = 12/15 = 0.8$. On the psychrometric chart a line of given SHF through the room state is the *room SHF line*; supply air must lie on it. High SHF (≈ 0.9) $\rightarrow$ mostly temperature change; low SHF $\rightarrow$ large moisture load.

(Definition with formula 1 M, explanation 1 M)

c) Advantages of hermetically sealed compressor.

2 M

- No shaft seal, so no leakage of refrigerant.
- Compact, light and occupies less space.
- Quiet, vibration-free running (internal spring mounting).
- No belt or coupling — no alignment problem; low maintenance.
- Lubrication is sealed for life; low initial cost in small sizes.
- Motor is cooled by suction vapour.

(Any four, $\frac{1}{2}$ M each)

d) Capillary tube — advantages and limitations.

2 M

- **Advantages:** simple, cheapest, no moving parts; pressures equalise in off-cycle so a low starting-torque motor is enough.
- **Limitations:** cannot adjust to load variations; easily choked by moisture/dirt — so used only for small units with nearly constant load.

(Two advantages 1 M, two limitations 1 M)

e) Basement ventilation.

2 M

Basement ventilation: Basements (car parks, stores) have no natural openings, so **mechanical ventilation** is provided to remove CO, exhaust fumes, smoke, heat and moisture and to supply fresh air (typically 6–12 air changes per hour, higher in fire mode). Fresh air is supplied through

shafts by supply fans; exhaust fans with ducts or **jet (induction) fans** push the polluted air to exhaust shafts. CO sensors control fan speed.

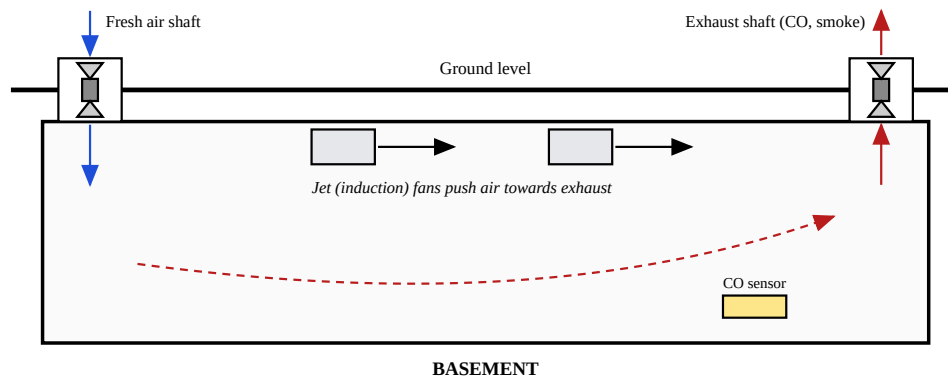


Fig.: Basement (car-park) ventilation — supply, exhaust and jet fans

(Concept 2 M)

f) Classification of air ducts.

2 M

- **On the basis of supply / function:** supply air duct, return air duct, fresh (make-up) air duct, exhaust air duct.
- **On the basis of velocity / pressure:** low velocity (< 10 m/s, low pressure up to 50 mm WG) and high velocity (> 10 m/s, medium/high pressure).
- **On the basis of cross-section (shape):** rectangular, square, circular (round), flat oval.

(Any two bases with types, 1 M each)

g) Factors affecting human comfort.

2 M

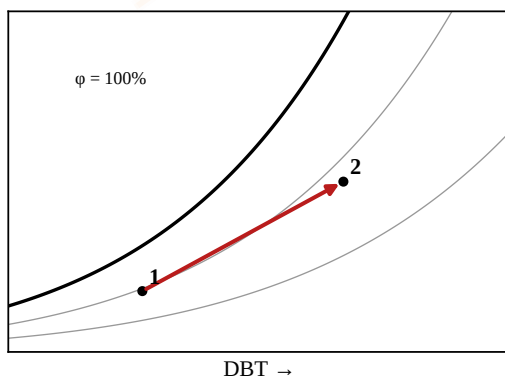
- **Air temperature (DBT):** about 22–26 °C is comfortable in summer for Indian conditions.
- **Relative humidity:** 40–60 %; high humidity prevents evaporation of sweat.
- **Air motion (velocity):** 6–9 m/min in the occupied zone; higher velocity causes draught.
- **Mean radiant temperature** of surrounding surfaces (walls, glass, roof).

(Any four, ½ M each)

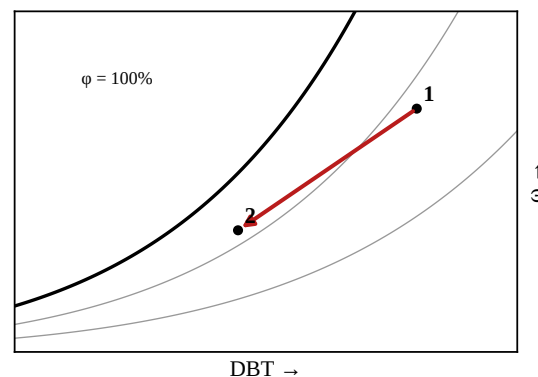
Q.2

a) Heating & humidification and cooling & dehumidification on chart.

4 M



(i) Heating & humidification



(ii) Cooling & dehumidification

Heating and humidification (1–2): Air is heated and moisture is added (heating coil + steam/water spray). DBT and ω both increase; line goes upward to the right. Used in winter air-conditioning.

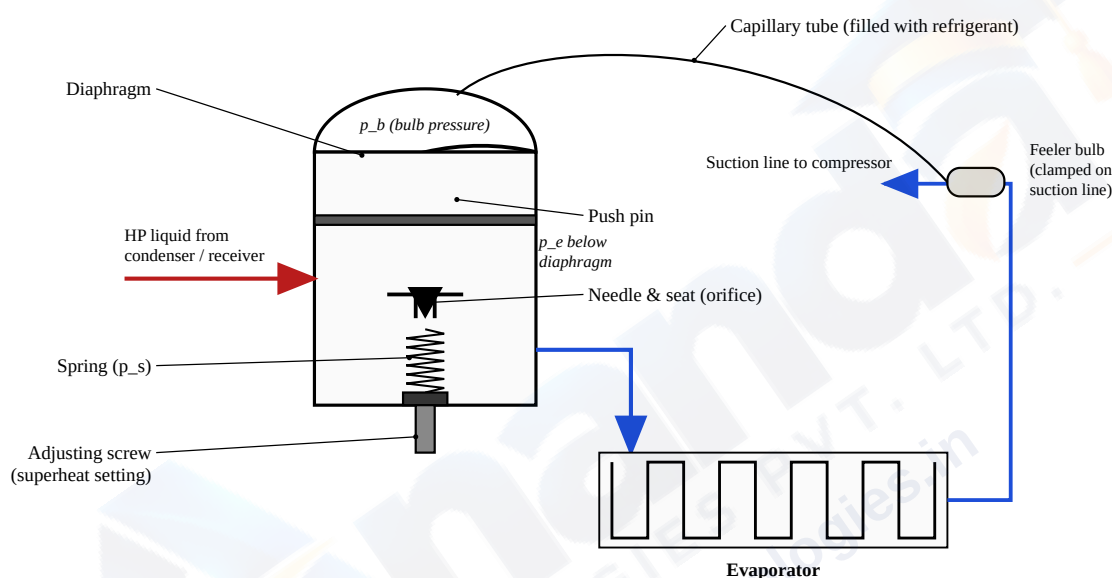
Cooling and dehumidification (1–2): Air is passed over a cooling coil whose surface temperature (ADP) is below the dew point of air. Both DBT and ω fall; moisture condenses and is drained. The process line goes downwards and to the left. Used in all summer air-conditioning. Heat removed = $m(h_1 - h_2)$, moisture removed = $m(\omega_1 - \omega_2)$.

(Each process: sketch 1 M + explanation 1 M)

b) Thermostatic expansion valve.

4 M

The **thermostatic expansion valve (TEV)** controls refrigerant flow into the evaporator so as to keep a **constant degree of superheat** ($\approx 5^\circ\text{C}$) at the evaporator outlet, whatever the load.



Balance: $p_b = p_e + p_s$; more superheat $\rightarrow p_b$ rises $\rightarrow$ valve opens more

Fig.: Thermostatic expansion valve (TEV / TXV)

Construction: valve body with needle and seat; a diaphragm at the top; a feeler bulb clamped on the evaporator outlet (suction line) connected to the space above the diaphragm by a capillary tube; an adjustable spring below the needle.

Working: Three pressures act on the diaphragm — bulb pressure p_b (above, tends to open), evaporator pressure p_e and spring pressure p_s (below, tend to close). At balance $p_b = p_e + p_s$.

- When load increases, refrigerant evaporates earlier, superheat at outlet rises, bulb temperature and p_b rise $\rightarrow$ diaphragm moves down $\rightarrow$ valve opens more $\rightarrow$ more refrigerant flows.
- When load decreases, superheat falls, p_b falls $\rightarrow$ valve closes partly $\rightarrow$ less flow.
- Thus the evaporator is always fed correctly and liquid cannot flood back to the compressor.

(Sketch 2 M, working 2 M)

c) Purpose of insulation; properties of insulating material.

4 M

Purpose of insulation: to reduce heat gain into the conditioned space, ducts, chilled water pipes and refrigerant lines; to prevent condensation (sweating) on cold surfaces; to save energy and

reduce plant capacity; also for sound control.

- Low thermal conductivity ($k \approx 0.02\text{--}0.05 \text{ W/m K}$).
- Low density (light weight) and low specific heat.
- Low water-vapour absorption / should be non-hygroscopic (moisture reduces insulating value).
- Fire resistant / non-inflammable and odourless.
- Should not be attacked by vermin, fungus; chemically stable and durable.
- Easy to apply/fix, available in convenient forms, low cost.
- Should have sufficient mechanical strength and should not settle or shrink.

Common materials: expanded polystyrene (thermocole), polyurethane foam (PUF), glass wool, mineral (rock) wool, nitrile rubber (closed-cell elastomeric) foam, cork, XLPE foam.

(Purpose 1 M, any four properties 3 M)

d) Window air conditioner.

4 M

A **window air conditioner** is a unitary, self-contained unit fitted in a window or wall opening. A partition divides it into a room (indoor) side and an outdoor side.

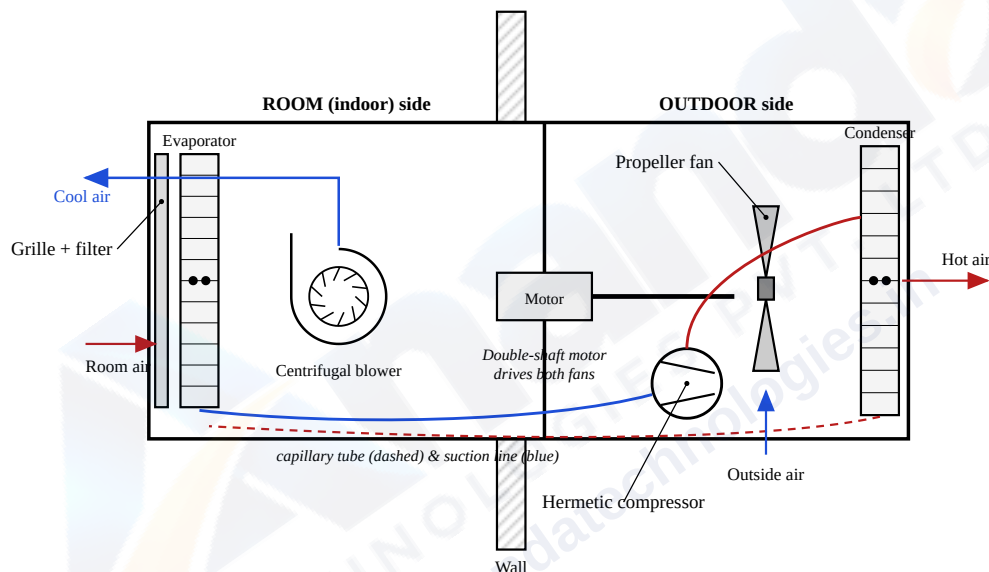


Fig.: Window air conditioner (room air conditioner)

- **Indoor side:** filter and front grille, evaporator (cooling coil), centrifugal blower which draws room air through the filter and coil and throws cooled air back into the room through adjustable louvres.
- **Outdoor side:** hermetic compressor, condenser coil, propeller fan which blows outside air over the condenser.
- One **double-shaft motor** drives both the blower and the propeller fan.
- Refrigerant: compressor → condenser → capillary tube → evaporator → compressor. Moisture condensed on the evaporator drains to the outdoor side and is splashed on the condenser.
- Capacity 0.75–2 TR; controls: thermostat, fan speed selector.

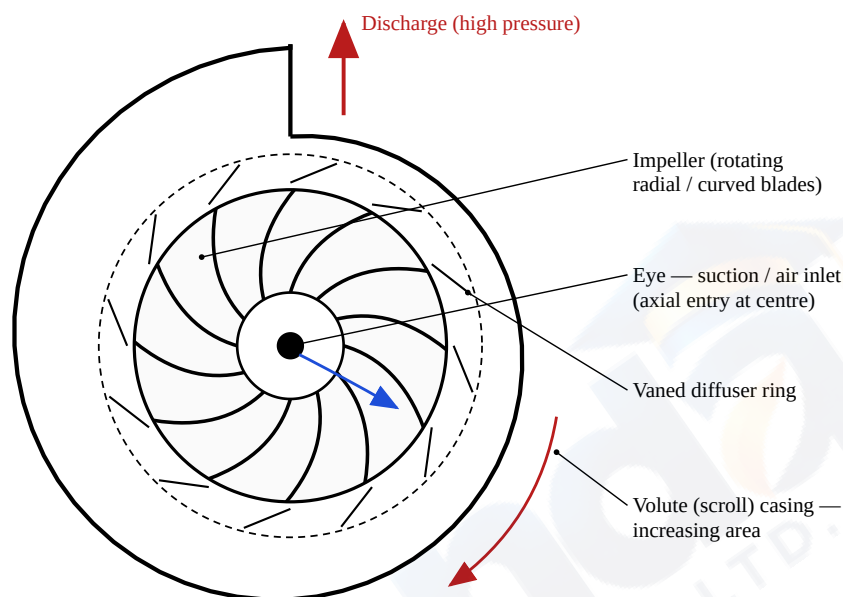
(Sketch 2 M, explanation 2 M)

Q.3

a) Centrifugal compressor.

4 M

Construction: A centrifugal compressor consists of an **impeller** (rotor with radial or backward-curved blades) rotating at high speed (3000–20000 rpm) inside a **casing**; a **diffuser** (vaned or vaneless) around the impeller; and a **volute (scroll) casing** of increasing area that collects the gas. Large units have 2–4 impellers in series on one shaft.



Velocity given to gas by impeller is converted into pressure in diffuser & volute

Fig.: Centrifugal compressor — impeller, diffuser and volute casing

Working:

1. Low pressure refrigerant vapour enters axially at the **eye** of the impeller.
2. The rotating blades throw the vapour radially outward by centrifugal force, increasing its velocity (kinetic energy) and some pressure.
3. In the diffuser the passage area increases, velocity falls and kinetic energy is converted into pressure energy.
4. The volute collects the high-pressure vapour and delivers it to the condenser (or to the next stage).

Applications: large central AC plants and water chillers (≈ 100 –10000 TR) using R-123, R-134a, R-1233zd.

(Sketch 2 M, working 2 M)

b) Summer AC system for hot and humid outdoor conditions.

4 M

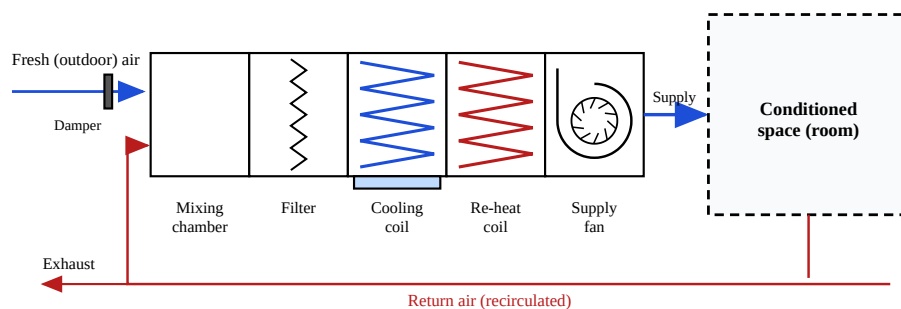


Fig.: Summer air conditioning for hot and humid outdoor conditions

- Outdoor air (hot and humid) is mixed with re-circulated room air and filtered.
- The **cooling coil** (temperature below DPT of air) cools and dehumidifies the air — excess moisture condenses and is drained.
- If the air leaving the coil is too cold (low SHF rooms), a **re-heater** raises its DBT at constant ω to the required supply temperature.
- The fan supplies the air to the room; thermostat controls the coil, humidistat controls re-heat.
- Used in coastal cities (Mumbai, Chennai) where outdoor air is hot and humid.

(Sketch 2 M, explanation 2 M)

c) Central vs unitary AC system.

4 M

Central air conditioning system	Unitary air conditioning system
All equipment (compressor, condenser, chiller, AHU) is at one central plant; conditioned air/chilled water is distributed by ducts/pipes to all rooms.	Factory-assembled self-contained unit (window, split, packaged) placed in or near the room to be conditioned.
Used for large capacity — above about 25 TR (malls, hotels, hospitals, theatres).	Used for small capacity — 0.75 to 25 TR (rooms, small offices, shops).
High initial cost, but lower running cost per TR; long life.	Low initial cost, easy to install; higher running cost per TR.
Needs separate plant room, ducts, skilled operators.	No plant room; ductless or short ducts; simple operation.
Better control of humidity, filtration and fresh air for the whole building.	Control only for the individual room; fresh-air and humidity control limited.
Breakdown affects the whole building.	Breakdown affects only one room.

(Any four points, 1 M each)

d) Losses in ducts.

4 M

Pressure losses in ducts are of two types — **friction (major) losses** and **dynamic (minor) losses**.

- **Friction loss:** due to friction between moving air and duct walls along straight length; $\Delta p_f = f L \rho V^2 / (2m)$ ($m = A/P$). Depends on velocity, length, roughness and size.
- **Loss due to sudden / gradual enlargement:** velocity falls suddenly, eddies form; $\Delta p = \rho(V_1 - V_2)^2 / 2$ for sudden enlargement.
- **Loss due to sudden contraction:** vena contracta forms after entry to smaller duct; $\Delta p = K \rho V_2^2 / 2$ ($K \approx 0.5$ at duct entrance).
- **Loss in bends and elbows:** change of direction causes separation and secondary flow; reduced by splitter / turning vanes and large radius bends.
- **Loss at branches (take-offs), dampers, grilles, filters, coils and exit (velocity pressure) loss at outlet.**

(Any four losses, 1 M each)

Q.4

a) Flooded vs dry expansion evaporator; sketch of flooded evaporator.

4 M

Flooded evaporator	Dry expansion (DX) evaporator
Evaporator is always full of liquid refrigerant ($\approx 75-100\%$).	Only 15–25 % of the coil holds liquid; rest is vapour.
Liquid level controlled by float valve; surge drum needed.	Refrigerant flow controlled by TEV / capillary.
High heat transfer rate; vapour leaving is dry saturated.	Lower heat transfer; vapour leaving is slightly superheated.
Large refrigerant charge, costly.	Small refrigerant charge, cheaper.
Used in large chillers, ammonia plants.	Used in window/split AC, refrigerators, AHU coils.

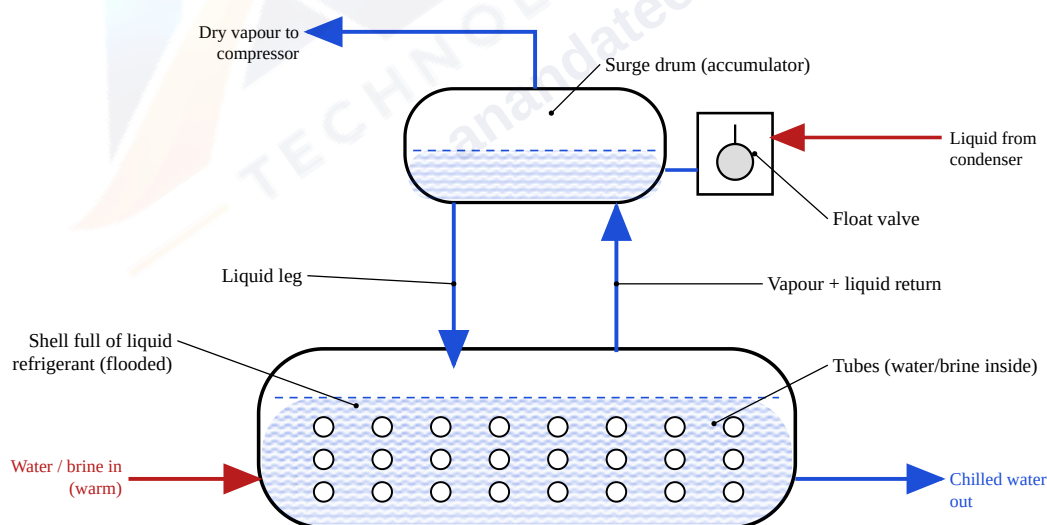


Fig.: Flooded type (shell and tube) evaporator with surge drum and float valve

(Any four differences 2 M, sketch 2 M)

b) Heat gain from occupants.

4 M

Heat released by a person depends on activity level and room temperature. It has a **sensible** part (convection + radiation from body) and a **latent** part (moisture from breathing and sweat).

Occupant load: $Q_s = N \times \text{SHG per person}$; $Q_L = N \times \text{LHG per person}$; where N = number of occupants (from seating capacity or floor area $\div$ area per person), SHG/LHG from tables (ASHRAE/Carrier) for the activity at room temperature. A diversity factor is used when all occupants are not present at the same time.

Activity (room at 25–26 °C)	Sensible heat (W/person)	Latent heat (W/person)
Seated at rest (theatre)	65	30
Seated, light work (office, classroom)	70–75	45–55
Moderately active (laboratory)	75	75
Light bench work / walking	80–90	140
Heavy work (gym)	170	255

Example: classroom of 60 students (light work): $Q_s = 60 \times 75 = 4500 \text{ W}$; $Q_L = 60 \times 55 = 3300 \text{ W}$; total occupant load = 7.8 kW.

(Explanation 2 M, formula/table 1 M, example 1 M)

c) Variable Air Volume (VAV) system.

4 M

Variable Air Volume (VAV) system: an all-air system in which the **temperature of supply air is kept constant** and the **quantity (volume) of air** supplied to each zone is varied according to the load of that zone.

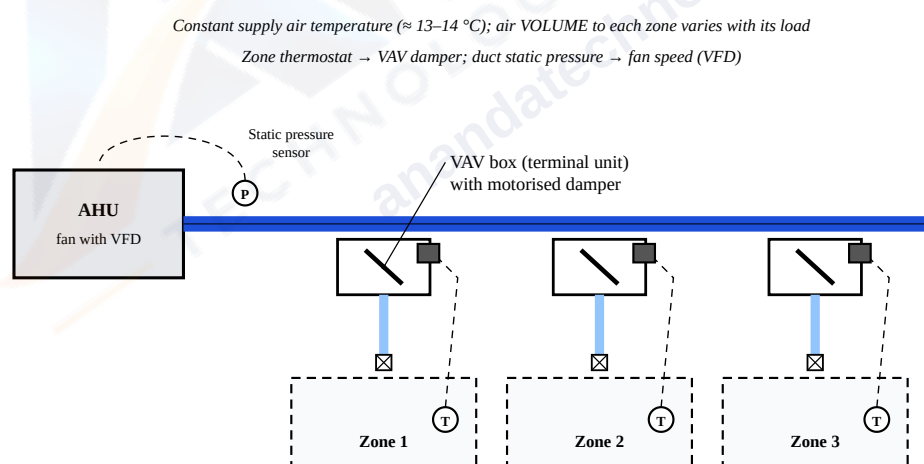


Fig.: Variable Air Volume (VAV) system

Working:

1. The AHU supplies cool air at constant temperature ($\approx 13^\circ\text{C}$) into the main duct.
2. Each zone has a **VAV terminal box** with a motorised damper controlled by the zone thermostat.

3. When zone load falls, the thermostat closes the damper partly — less air enters; when load rises, the damper opens.
4. As boxes close, duct static pressure rises; the pressure sensor signals the **VFD** to reduce fan speed, saving fan energy.

Advantages: individual zone control, large fan energy saving at part load, plant diversity.

Limitation: at very low loads fresh air and air motion may be insufficient (minimum box setting used).

(Concept 1 M, sketch 2 M, working 1 M)

d) Supply air outlets; ceiling diffuser.

4 M

Supply air outlets are the terminal devices that deliver conditioned air into the room and distribute it without draught or noise:

- **Grilles:** rectangular outlets with fixed or adjustable vanes (bars); used for supply and return; no volume control.
- **Registers:** grille combined with a volume control damper behind it — controls both direction and quantity of air.
- **Diffusers:** ceiling outlets (square, round, linear slot) with concentric cones/louvres; spread air in many directions and mix it rapidly with room air.
- **Other outlets:** slot (linear) diffusers, perforated ceiling panels, nozzles / jet diffusers (auditoriums, halls), floor (swirl) diffusers.

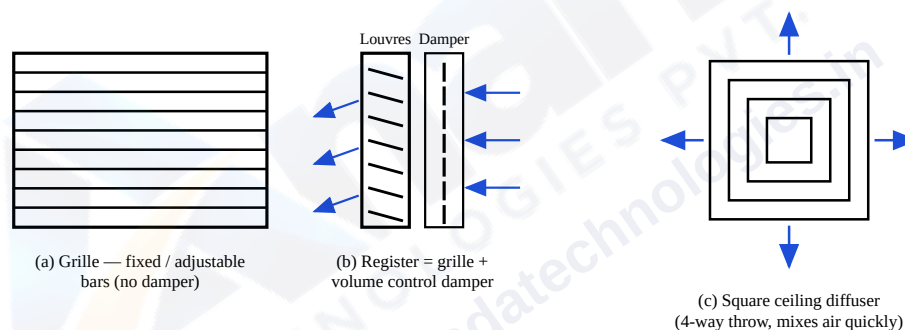


Fig.: Supply air outlets — (a) grille (b) register (c) ceiling diffuser

Functions: (1) deliver and spread supply air uniformly in the occupied zone; (2) control direction (throw) and quantity of air; (3) mix supply air with room air to avoid cold draught; (4) give a neat appearance and keep noise low.

Ceiling diffuser: fitted flush with the false ceiling; consists of a set of concentric square or round cones/louvres. Supply air leaving in 2-, 3- or 4-way pattern along the ceiling induces room air (Coanda effect), so it mixes quickly and reaches the occupied zone at low velocity without draught. Used widely in offices, shops and halls with false ceiling; a damper behind it controls the quantity.

(List 1 M, sketch 1 M, explanation 2 M)

e) Automobile air conditioning system.

4 M

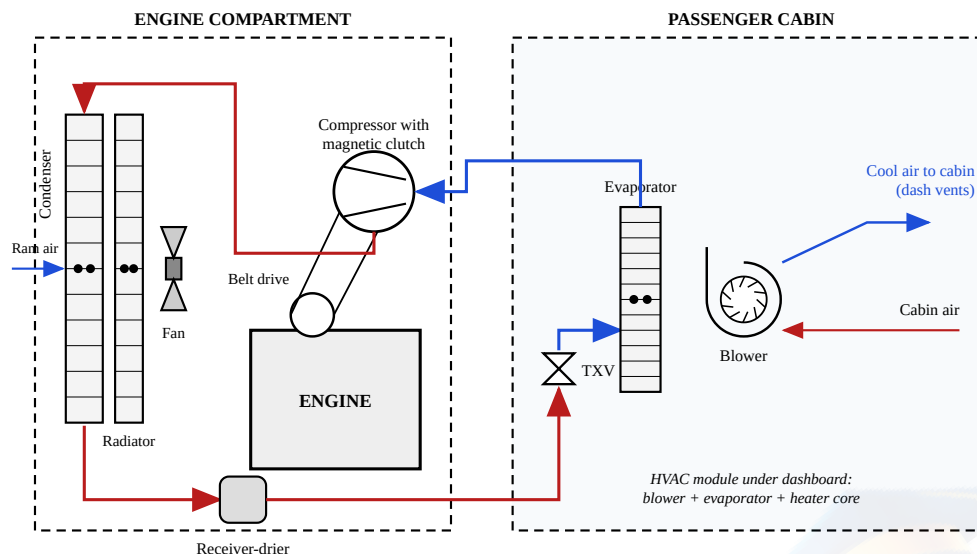


Fig.: Layout of automobile (car) air conditioning system

Components and working:

- **Compressor** (swash-plate/scroll) driven by the engine through a belt and **electromagnetic clutch**; clutch engages when AC is switched on.
- **Condenser** in front of the radiator — cooled by ram air and the radiator fan; vapour condenses to liquid.
- **Receiver-drier** stores liquid and removes moisture and dirt (desiccant + filter).
- **Expansion valve (TXV)** or orifice tube reduces pressure and meters refrigerant to the evaporator.
- **Evaporator** in the HVAC module under the dashboard; the **blower** passes cabin/fresh air over it and cool, dry air enters the cabin through dash vents.
- Refrigerant R-134a / R-1234yf; thermostat/pressure switches cycle the clutch.

(Layout 2 M, explanation 2 M)

Q.5

a) Cooling and dehumidification 35 °C/40 % → 22 °C DBT, 16 °C WBT.

6 M

Given: State 1: DBT = 35 °C, RH = 40 %; State 2: DBT = 22 °C, WBT = 16 °C; $p = 101.325 \text{ kPa}$.

Formulas used ($p = 101.325 \text{ kPa}$; saturation pressures from steam tables):

$$p_v = \phi \cdot p_{vs} \cdot \omega = 0.622 \frac{p_v}{(p - p_v)} \cdot h = 1.005 t + \omega(2500 + 1.88 t) \text{ kJ/kg d.a.}$$

Carrier's equation: $p_v = p_{vw} - (p - p_{vw})(t_d - t_w)/(1544 - 1.44 t_w)$ · DPT = saturation temperature corresponding to p_v

State 1: $p_{vs}(35 \text{ °C}) = 5.6290 \text{ kPa}$; $p_{v1} = 0.4 \times 5.6290 = 2.2516 \text{ kPa}$

$\omega_1 = 0.622 \times 2.2516 / (101.325 - 2.2516) = \mathbf{0.0141 \text{ kg/kg d.a.}}$ (14.14 g/kg)

$h_1 = 1.005 \times 35 + 0.0141(2500 + 1.88 \times 35) = \mathbf{71.45 \text{ kJ/kg d.a.}}$; $\text{DPT}_1 = 19.4 \text{ °C}$

State 2: p_{vs} at WBT $16^\circ\text{C} = 1.8188 \text{ kPa}$

$$p_{v2} = 1.8188 - (101.325 - 1.8188)(22 - 16)/(1544 - 1.44 \times 16) = 1.4263 \text{ kPa}$$

$$\text{DPT}_2 = t_{\text{sat}} \text{ at } 1.4263 \text{ kPa} = \mathbf{12.3^\circ\text{C}}$$

$$\omega_2 = 0.622 \times 1.4263 / (101.325 - 1.4263) = 0.0089 \text{ kg/kg d.a.}; h_2 = 1.005 \times 22 + 0.0089(2500 + 1.88 \times 22) = \mathbf{44.68 \text{ kJ/kg d.a.}}; \text{RH}_2 = 53.9 \%$$

$$\text{Change in enthalpy} = h_1 - h_2 = 71.45 - 44.68 = \mathbf{26.77 \text{ kJ/kg d.a. (heat removed)}}$$

$$\text{Moisture removed} = \omega_1 - \omega_2 = 0.0141 - 0.0089 = 0.0053 \text{ kg/kg d.a. (5.26 g/kg d.a.)}$$

$$\text{Latent heat removed} \approx 2500 \times 0.0053 = 13.14 \text{ kJ/kg}; \text{sensible} = 13.63 \text{ kJ/kg}; \text{SHF} = 13.63/26.77 = 0.51$$

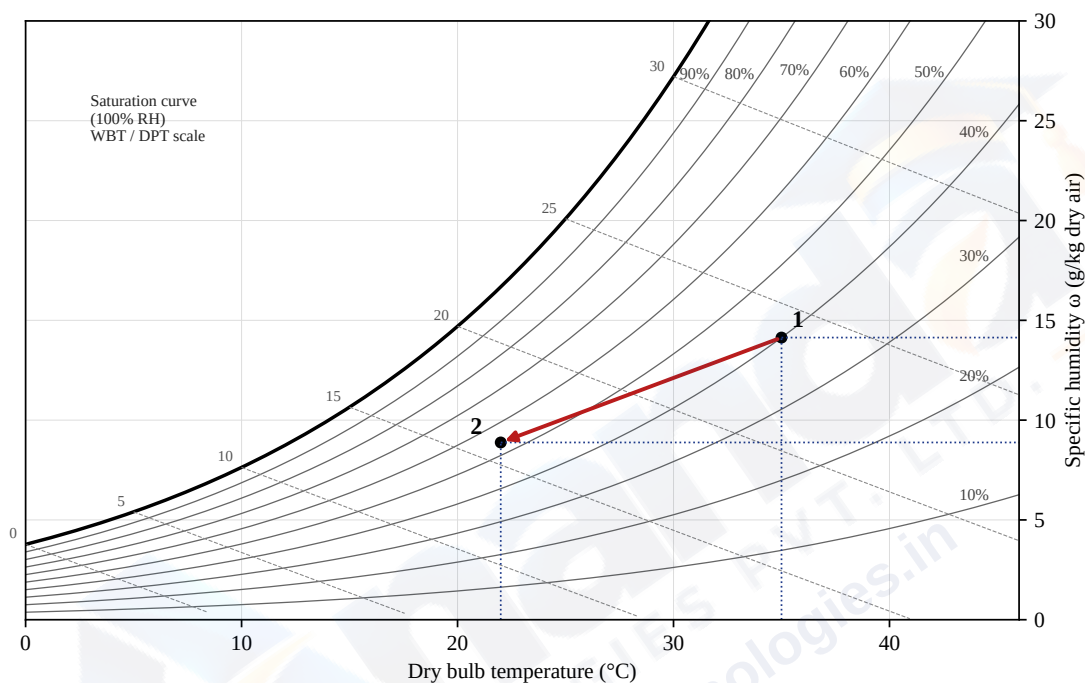


Fig.: Cooling and dehumidification 1 $\rightarrow$ 2 on psychrometric chart (computed)

Answers: (i) $\omega_1 = 0.0141 \text{ kg/kg d.a.} = 14.14 \text{ g/kg d.a.}$ · (ii) DPT at final state = 12.3°C · (iii) Change in enthalpy = 26.77 kJ/kg d.a. · Moisture removed = 5.26 g/kg d.a.

(Chart 2 M, each answer 1 M)

b) Year-round air conditioning system with controls.

6 M

A **year-round air conditioning system** maintains the desired inside condition (e.g. 24°C , 50 % RH) in **all seasons** — it cools and dehumidifies in summer and heats and humidifies in winter, controlled automatically by thermostat and humidistat.

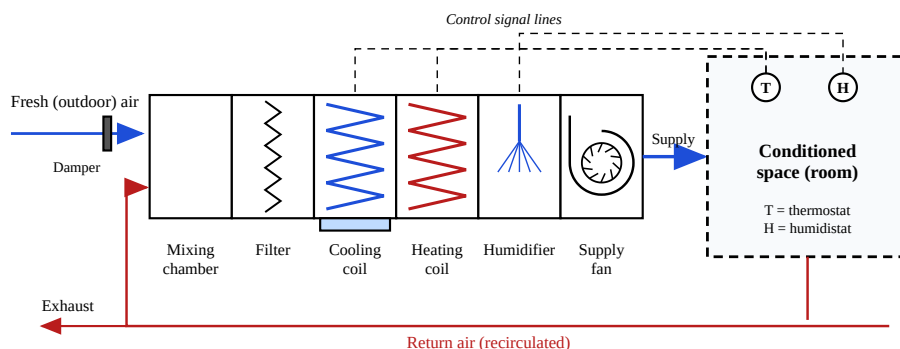


Fig.: Year-round air conditioning system

Working:

1. Return air from the room is mixed with fresh outdoor air in the mixing chamber (dampers control proportion); part of the room air is exhausted.
2. The mixture is cleaned in the **filter**.
3. **Summer:** the cooling coil (chilled water / DX) cools and dehumidifies the air; heating coil works as re-heater only if needed; humidifier is off.
4. **Winter:** the cooling coil is off; the heating coil heats the air and the humidifier (steam/water spray) adds moisture.
5. The supply fan delivers conditioned air through ducts to the room.
6. The room **thermostat** controls the cooling/heating coils and the **humidistat** controls the humidifier (and cooling coil for dehumidification).

Applications: hospitals, hotels, laboratories, computer centres, auditoriums in places with both hot and cold seasons.

(Sketch 3 M, working 3 M)

c) Scroll compressor.

6 M

Construction: A scroll compressor has two interleaved spiral (involute) scrolls: a **fixed scroll** attached to the body, and an **orbiting scroll** driven by an eccentric on the motor shaft. An **Oldham coupling** prevents the orbiting scroll from rotating, so it only orbits. Suction is at the periphery and the discharge port is at the centre of the fixed scroll. The whole unit is usually hermetically sealed.

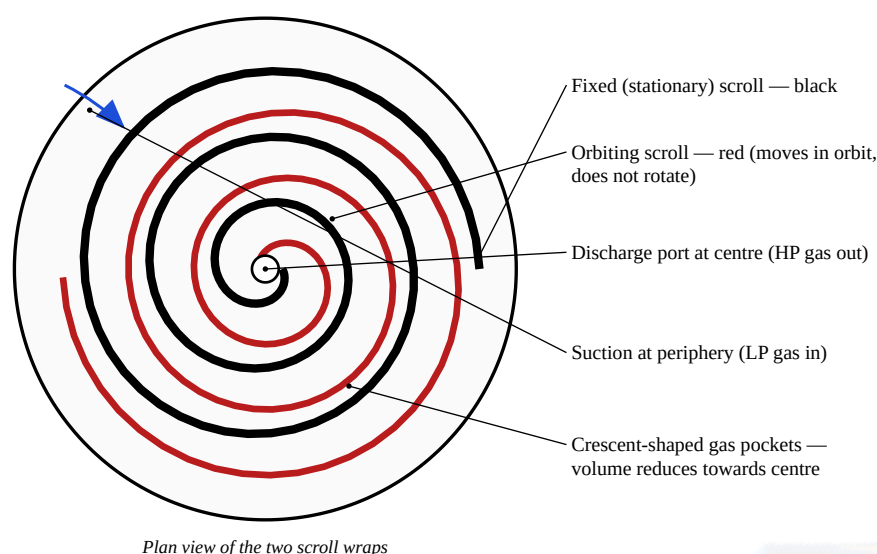


Fig.: Scroll compressor — fixed scroll and orbiting scroll

Working:

1. Suction: as the orbiting scroll moves, gas enters the outer open pockets at the periphery.
2. Compression: the pockets are sealed off and move towards the centre; their crescent-shaped volume continuously reduces, so pressure rises.
3. Discharge: at the centre the compressed gas leaves through the discharge port.
4. Suction, compression and discharge go on simultaneously in different pockets — smooth, continuous flow.

Advantages: no suction/discharge valves, fewer moving parts, high volumetric efficiency, low noise and vibration. **Applications:** split and packaged AC (1–15 TR), heat pumps, automobile AC.

(Construction with sketch 3 M, working 3 M)

Q.6

a) Cooling load for a computer laboratory.

6 M

Assumed data: 33 occupants (32 students + 1 faculty), seated light work — 75 W sensible, 55 W latent each; one projector 300 W; roof not exposed (intermediate floor); ΔT for wall and glass = $38 - 24 = 14$ °C; fresh air 5 L/s per person; infiltration 1 air change/h; $\rho = 1.2$ kg/m³; safety factor 10 %.

Room data (assumed): 9 m × 7 m × 3.2 m (volume 201.6 m³). Outside design: 38 °C DBT, 45 % RH ($\omega_o = 0.0189$ kg/kg); inside design: 24 °C DBT, 50 % RH ($\omega_i = 0.0093$ kg/kg) — ω from $\omega = 0.622p_v/(p - p_v)$.

Source of heat gain	Calculation	Sensible (W)	Latent (W)
Exposed wall	$Q = U A \Delta T = 1.9 \times 25.0 \times 14$	665	—
Glass (conduction)	$Q = U A \Delta T = 5.8 \times 5.0 \times 14$	406	—
Solar gain through glass	$A \times q_{\text{solar}} = 5.0 \times 180 \text{ W/m}^2$	900	—

Occupants (33 persons)	SH = $33 \times 75 \text{ W}$; LH = $33 \times 55 \text{ W}$	2475	1815
Lighting (fluorescent)	$480 \text{ W} \times 1.25$ (ballast factor)	600	—
Computers (CPU + monitor)	$30 \times 150 \text{ W}$	4500	—
Projector	$1 \times 300 \text{ W}$	300	—
Infiltration (1 air change/h)	$\dot{V} = 0.0560 \text{ m}^3/\text{s}$, $\dot{m} = 1.2 \dot{V} = 0.0672 \text{ kg/s}$ SH = $\dot{m} \times 1.0216 \times (38 - 24)$ LH = $\dot{m} \times 2500 \times (0.0189 - 0.0093)$	961	1609
Ventilation (fresh) air 5 L/s per person	$\dot{V} = 0.1650 \text{ m}^3/\text{s}$, $\dot{m} = 1.2 \dot{V} = 0.1980 \text{ kg/s}$ SH = $\dot{m} \times 1.0216 \times (38 - 24)$ LH = $\dot{m} \times 2500 \times (0.0189 - 0.0093)$	2832	4741
Sub-total		13639	8165
Add 10 % safety factor		15003	8981

Total cooling load = $15003 + 8981 = 23984 \text{ W} = 23.98 \text{ kW}$. In tonnes of refrigeration (1 TR = 3.517 kW): **6.82 TR**. Room SHF = $15003/23984 = 0.63 \rightarrow$ select a 7.0 TR unit (next standard size).

Note: computers make the computer lab load mainly sensible (high SHF).

(Listing of loads 2 M, calculations 3 M, total in TR 1 M)

b) Duct system for an L-shaped building.

6 M

For an L-shaped building, an **extended plenum (trunk and branch)** system with the AHU at the **corner** (junction of the two wings) is most suitable.

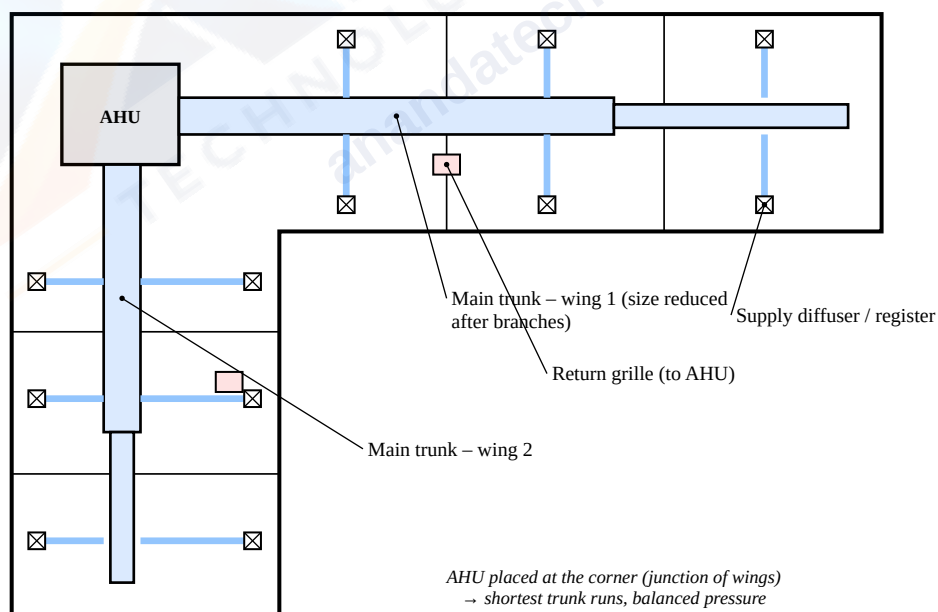


Fig.: Duct system for an L-shaped building (extended plenum along both wings)

- The AHU is placed at the junction of the two wings (or in a plant room near it), so the two trunk ducts are of nearly equal and shortest length.
- One main trunk runs along each wing (over the corridor in the false ceiling); its size is kept constant for some length and then reduced after branches (extended plenum).
- Short branch ducts with volume dampers take air from the trunk to ceiling diffusers / registers in every room.
- Return air is collected through return grilles in corridors / rooms and taken back to the AHU by a return duct or ceiling plenum.
- Advantages: equal static pressure at all branches, easy balancing, fewer fittings, less space and cost; each wing can be zoned with its own damper.
- If the wings are very long, a separate AHU for each wing or a loop (perimeter) duct can be used.

(Selection with reason 1 M, sketch 3 M, explanation 2 M)

c) VCR cycle — 1.5 TR split AC with R-134a.

6 M

Property values (R-134a tables, IIR reference, saturated cycle: dry saturated vapour at compressor inlet, saturated liquid at condenser exit, isentropic compression):

Point	State	p (bar)	T (°C)	h (kJ/kg)	s (kJ/kg K)
1	Dry sat. vapour, compressor inlet	3.50	5.0	401.49	1.7245
2	Superheated, compressor exit ($s_2 = s_1$)	11.60	49.2	426.40	1.7245
3	Sat. liquid, condenser exit	11.60	45.0	263.94	1.2139
4	After throttling ($h_4 = h_3$), dryness $x_4 = 0.294$	3.50	5.0	263.94	1.2299

Refrigerating effect $RE = h_1 - h_4 = 401.49 - 263.94 = 137.55 \text{ kJ/kg}$

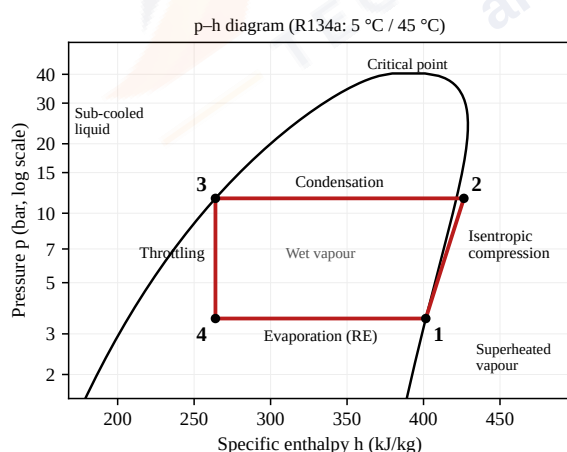
Compressor work $W = h_2 - h_1 = 426.40 - 401.49 = 24.90 \text{ kJ/kg}$

COP (refrigerator / AC) = $RE/W = 137.55/24.90 = 5.52$

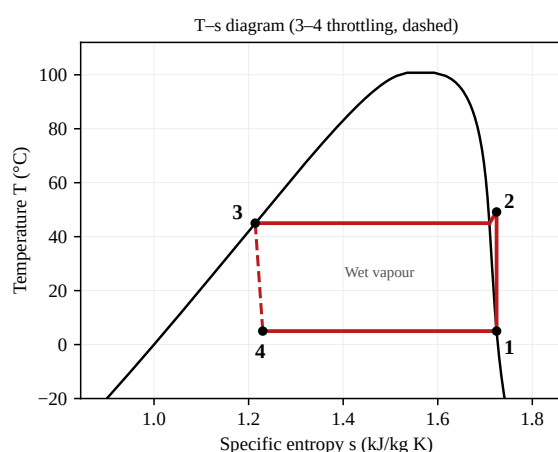
COP (as heat pump) = $(h_2 - h_3)/W = 162.45/24.90 = 6.52 (= COP_R + 1)$

Capacity = 1.5 TR = $1.5 \times 3.517 = 5.276 \text{ kW} \rightarrow \text{mass flow } \dot{m} = 5.276/137.55 = 0.0384 \text{ kg/s}$

Compressor power = $\dot{m} \times W = 0.0384 \times 24.90 = 0.96 \text{ kW}$



p-h diagram (computed)



T-s diagram (computed)

The enthalpies given in the question are the R-134a table values above (rounded).

Answers: RE = 137.55 kJ/kg · COP = 5.52 · $\dot{m}$ = 0.0384 kg/s (2.301 kg/min) · Compressor power = 0.96 kW

(RE 1 M, COP 1 M, $\dot{m}$ 1 M, power 1 M, diagrams 2 M)

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