Power Engineering (315371)

Model Question Paper — Set C · Model Answers

Q.1

a) One TR and its value in kW. 2 M

One tonne of refrigeration is the rate of heat removal required to freeze one ton (2000 lb ≈ 907 kg) of water at 0 °C into ice at 0 °C in 24 hours (latent heat of ice = 335 kJ/kg).

1 TR = 907 × 335 kJ / (24 × 60 min) = 211.0 kJ/min = 3.517 kJ/s ≈ **210 kJ/min ≈ 3.5 kW**

(Definition 1 M, derivation 1 M)

b) Specific humidity and degree of saturation. 2 M

Specific humidity (humidity ratio) ω: mass of water vapour present per kg of dry air: $\omega = m_v/m_a = 0.622\ p_v/(p - p_v)$ kg/kg d.a.

Degree of saturation μ: ratio of the actual specific humidity to the specific humidity of saturated air at the same DBT: $\mu = \omega/\omega_s$ (0 for dry air, 1 for saturated air).

(1 M each)

c) BSFC and brake thermal efficiency. 2 M

BSFC: mass of fuel consumed per unit brake power per hour: BSFC = $\dot{m}_f$ (kg/h)/BP (kW), in kg/kWh. Lower BSFC means better economy.

Brake thermal efficiency: ratio of brake power to the heat supplied by the fuel: $\eta_{bth} = BP/(\dot{m}_f \times CV)$.

(1 M each)

d) Volumetric and mechanical efficiency of compressor. 2 M

Volumetric efficiency: ratio of the free air delivered per stroke (at intake conditions) to the swept volume of the cylinder.

Mechanical efficiency: ratio of the indicated power (power given to the air in the cylinder) to the shaft power supplied to the compressor: η_m = IP/shaft power.

(1 M each)

e) Purpose of EGR. 2 M

EGR sends a controlled part (about 5–15 %) of the exhaust gas back into the intake manifold. The inert exhaust gas dilutes the charge and lowers the peak combustion temperature and oxygen concentration, so the formation of **oxides of nitrogen (NOx)** is reduced. It helps the engine meet emission norms (BS-VI).

(Purpose 2 M)

f) Applications of vapour absorption systems. 2 M

1. Large central air-conditioning plants using waste heat / steam (hotels, hospitals, factories).

2. Cogeneration and trigeneration plants (using exhaust of DG sets / gas turbines).
3. Domestic and caravan refrigerators working on gas or kerosene (Electrolux).
4. Solar-powered cooling; cold storage and ice plants in process industries with surplus heat.

(Any four, ½ M each)

g) Sources of energy wastage in compressed air system. 2 M

1. Air leakage from joints, hoses, couplings and valves.
2. Pressure drop in undersized piping, choked filters and dryers.
3. Generating air at higher pressure than needed; artificial demand.
4. Hot intake air, poor intercooling; misuse of compressed air for cleaning / cooling; running compressors unloaded.

(Any four, ½ M each)

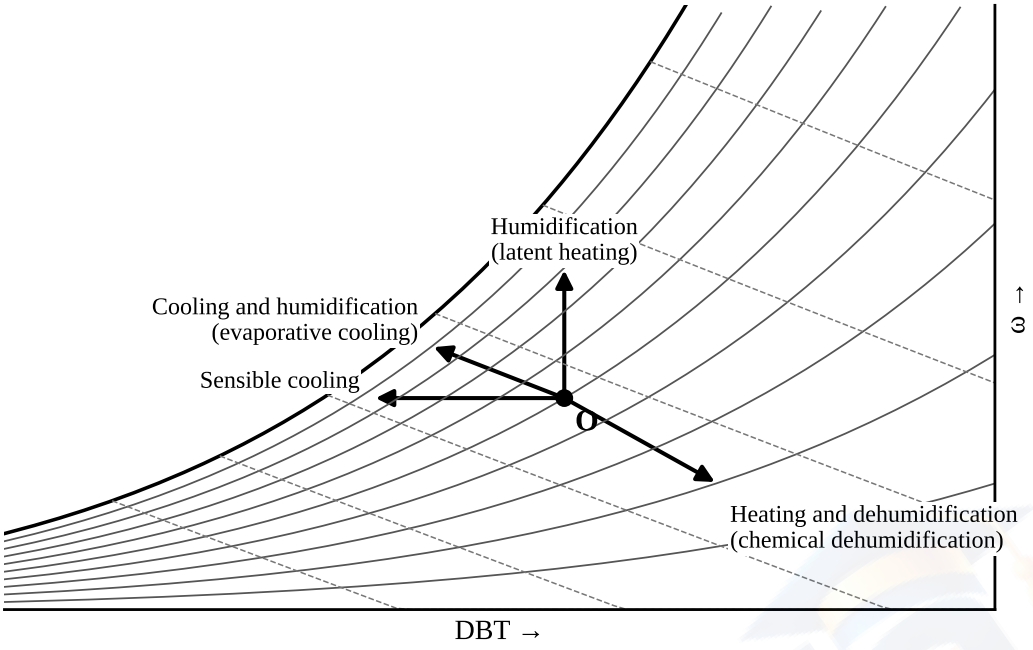
Q.2

a) Desirable properties of an ideal refrigerant. 4 M

1. Low boiling point at atmospheric pressure.
2. High latent heat of vaporisation (large refrigerating effect per kg, small mass flow).
3. Low specific volume of vapour (smaller compressor).
4. Moderate condenser pressure and evaporator pressure above atmospheric.
5. High critical temperature and low freezing point.
6. Non-toxic, non-flammable and non-explosive.
7. Chemically stable, non-corrosive, compatible with oil and materials.
8. Easy leak detection; cheap and easily available.
9. Zero ODP and low GWP (environment friendly).
10. High thermal conductivity and low viscosity (good heat transfer, low pressure drop).

(Any eight, ½ M each)

b) Psychrometric processes. 4 M



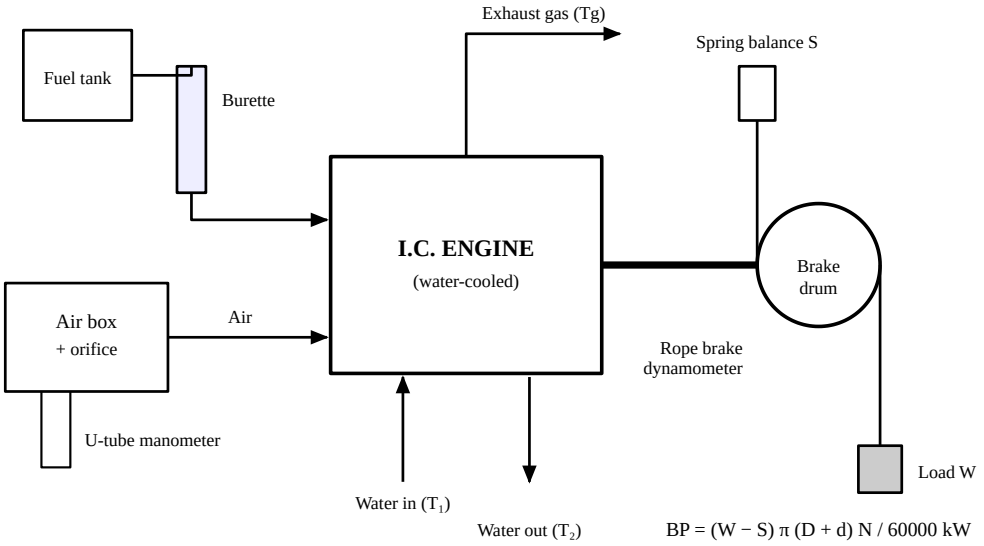
Psychrometric processes shown from an initial state O

- **Sensible cooling (O → left):** DBT falls at constant ω (coil above dew point); RH increases.
- **Humidification / latent heating (O → up):** ω increases at constant DBT (steam added at air temperature).
- **Heating and dehumidification (O → down-right):** air passed over a solid / liquid absorbent (silica gel); moisture removed, latent heat released raises DBT.
- **Cooling and humidification (O → up-left along WBT line):** air passed through water spray at WBT; DBT falls, ω rises (evaporative / desert cooler).

(Each process 1 M)

c) Need of heat balance sheet; measurement of losses. 4 M

Need: It gives a complete account of the heat supplied by the fuel and its distribution into BP, cooling water, exhaust gases and unaccounted losses. It shows the engine's thermal performance, where heat is wasted, guides design of cooling system and scope of waste heat recovery.



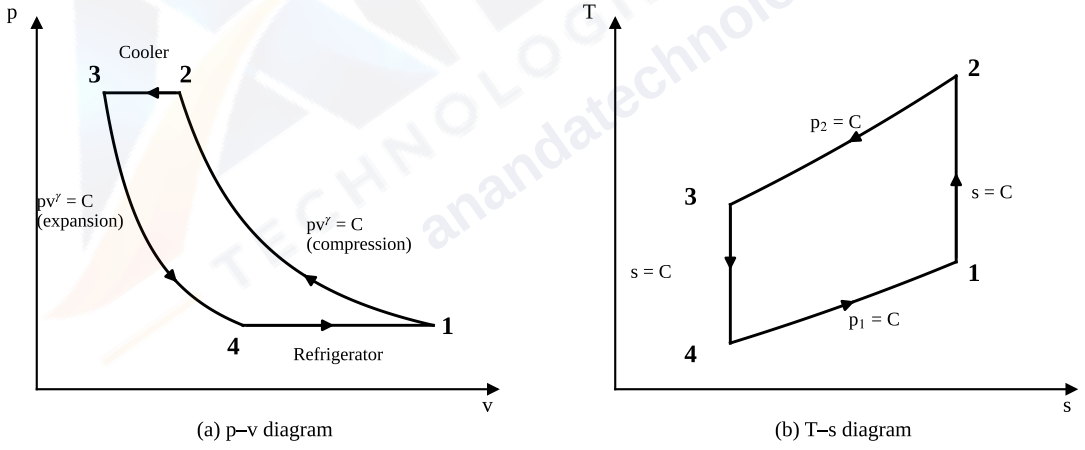
Engine test rig for heat balance (rope brake, fuel, air, water and exhaust measurement)

Heat to cooling water: mass of water flowing through the jacket in a known time is measured (collecting tank / flow meter) and its inlet and outlet temperatures t_1, t_2 are noted: $Q_w = m_w\, c_w(t_2 - t_1)$.

Heat to exhaust gases: mass of exhaust $m_g = m_a + m_f$ (air measured by air box with orifice and manometer); exhaust temperature t_g by thermocouple: $Q_g = m_g\, c_{pg}(t_g - t_{room})$. Alternatively an **exhaust gas calorimeter** is used: the gases heat water in a calorimeter; $m_{gw}c_w\Delta t_w = m_g c_{pg}(t_{g1} - t_{g2})$ gives c_{pg} and the heat.

(Need 1 M, cooling water 1½ M, exhaust 1½ M)

d) Bell-Coleman cycle and applications. 4 M



(a) p–v diagram (b) T–s diagram

Bell-Coleman (reversed Brayton) air refrigeration cycle

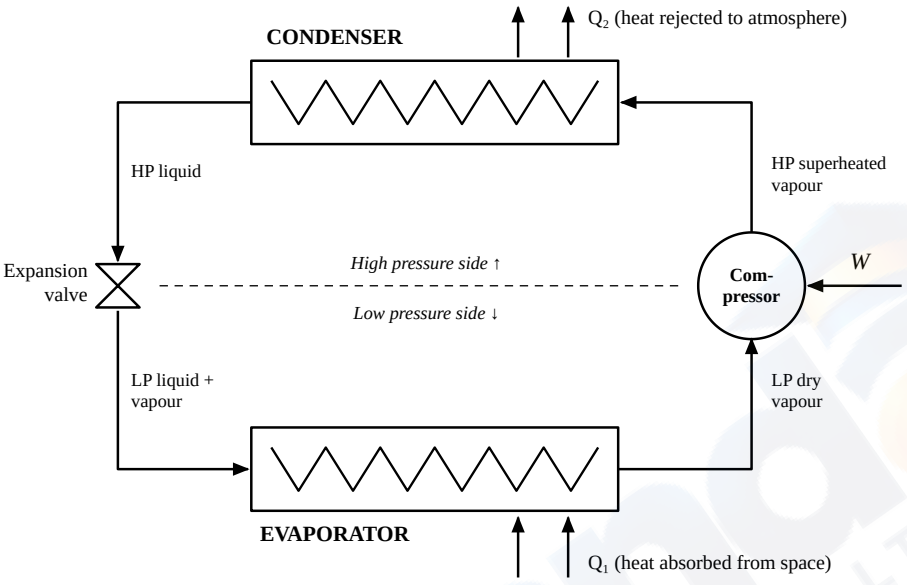
1. 1–2 isentropic compression of air in the compressor.
2. 2–3 constant-pressure heat rejection in the cooler.
3. 3–4 isentropic expansion in the expander (temperature falls far below the cold-chamber temperature).
4. 4–1 constant-pressure heat absorption in the refrigerator — refrigerating effect.

Applications: (1) aircraft cabin cooling (air is light, safe, freely available, and compressed air from the engine compressor is used); (2) liquefaction of gases and some ship / train refrigeration where safety of refrigerant is important.

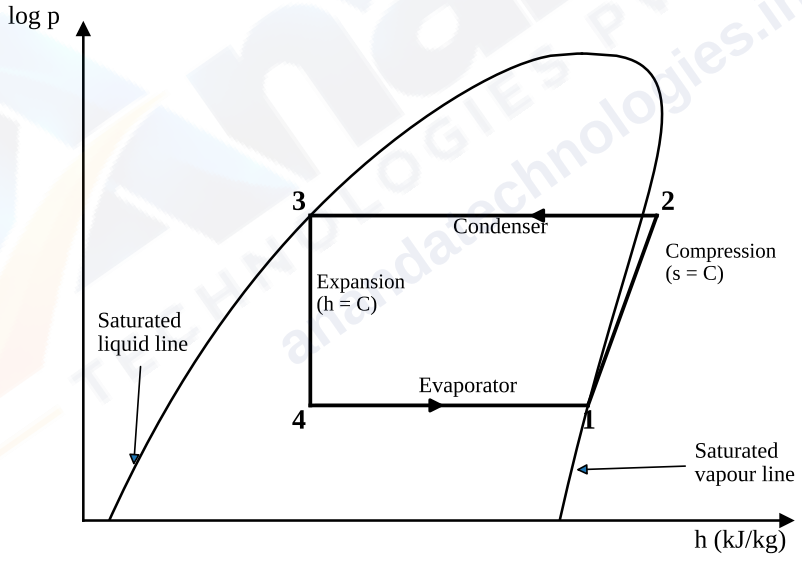
(Diagrams 2 M, processes 1 M, applications 1 M)

Q.3

a) Simple VCR system and p–h diagram. 4 M



Vapour compression refrigeration system (schematic)



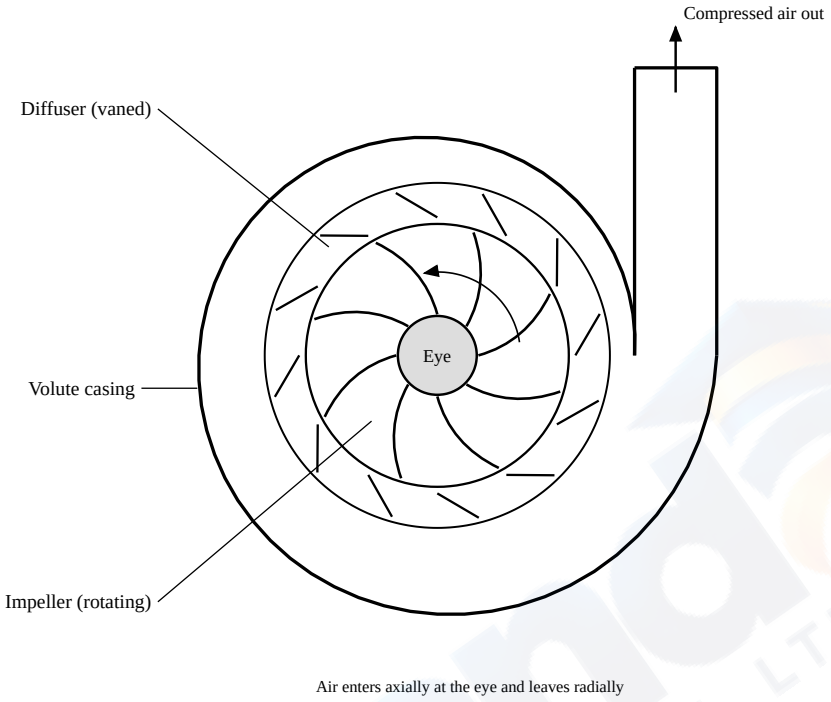
Simple saturated VCR cycle on p–h diagram

- **1–2 Compression (s = C):** dry saturated vapour from the evaporator is compressed to condenser pressure; it becomes superheated.
- **2–3 Condensation (p = C):** vapour gives heat Q_2 to the cooling medium and condenses to saturated liquid.
- **3–4 Expansion (throttling, h = C):** pressure and temperature fall in the expansion valve; some liquid flashes to vapour.

- **4–1 Evaporation (p = C):** low-pressure liquid takes heat Q_1 from the space to be cooled and evaporates. RE = $h_1 - h_4$, W = $h_2 - h_1$, COP = RE/W.

(Sketch 1½ M, p–h 1 M, working 1½ M)

b) Centrifugal compressor. 4 M



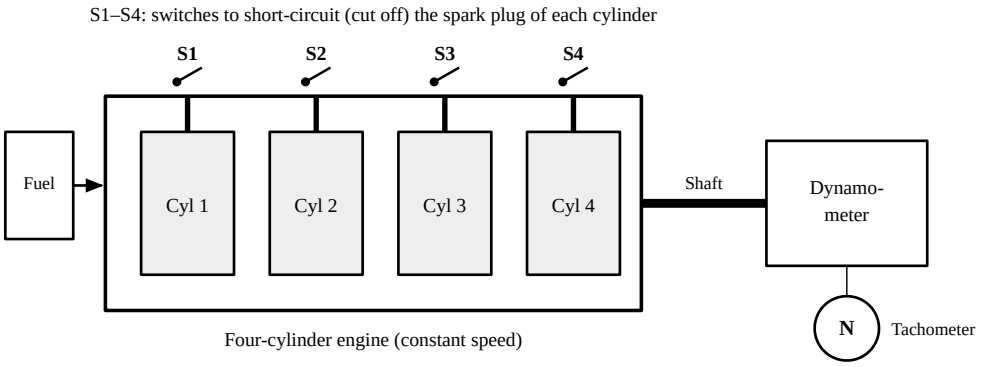
Centrifugal compressor: impeller, diffuser and volute casing (front view)

Construction: It consists of an **impeller** with radial / backward-curved vanes mounted on a shaft, a **diffuser** (vaned or vaneless) surrounding the impeller, and a spiral **volute casing** that collects the air and leads it to the outlet.

Working: Air enters axially at the eye of the impeller. The rapidly rotating impeller throws the air radially outward; its velocity (kinetic energy) and pressure increase due to centrifugal action. In the diffuser the passage area increases, so velocity falls and kinetic energy is converted into pressure. The volute further converts velocity into pressure and delivers the air. Roughly half of the pressure rise occurs in the impeller and half in the diffuser. Pressure ratio per stage about 2–4; continuous, oil-free flow.

(Sketch 2 M, construction 1 M, working 1 M)

c) IP without indicator (Morse test); why not single-cylinder. 4 M



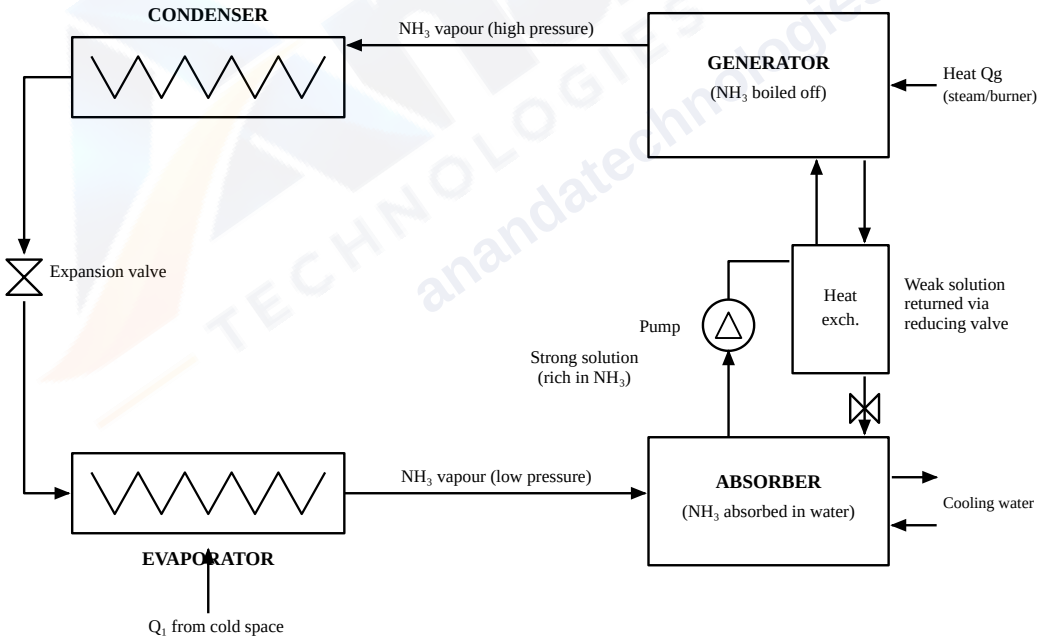
Morse test set-up on a four-cylinder SI engine

1. Run the multi-cylinder engine at constant speed N and measure BP with all cylinders working.
2. Cut off one cylinder (short the spark plug / stop fuel), restore the speed to N by reducing the load and measure BP_n.
3. Friction power is the same at the same speed, so IP of that cylinder = $BP - BP_n$.
4. Repeat for each cylinder; $IP = \Sigma(BP - BP_n)$; $FP = IP - BP$; $\eta_m = BP/IP$.

Why not for a single-cylinder engine: if the only cylinder is cut off the engine stops (it cannot run at the same speed by itself), so BP_n cannot be measured. The method needs the other cylinders to keep the engine running.

(Procedure 3 M, reason 1 M)

d) Aqua-ammonia vapour absorption system. 4 M



Simple vapour absorption (aqua-ammonia) refrigeration system

- The compressor of a VCR system is replaced by an **absorber, pump and generator**.
- **Absorber:** low-pressure NH_3 vapour from the evaporator is absorbed by weak aqua-ammonia solution; heat of absorption is removed by cooling water. A strong solution is formed.

- **Pump and heat exchanger:** the strong solution is pumped to generator pressure and pre-heated by the hot weak solution returning from the generator.
- **Generator:** heat (steam / burner / waste heat) drives NH_3 vapour out of the solution; the weak solution returns to the absorber through a pressure-reducing valve.
- **Condenser, expansion valve, evaporator:** as in VCR — NH_3 vapour condenses, is throttled and evaporates in the evaporator, producing the refrigerating effect. (An analyser and rectifier remove water vapour in practical plants.)

(Sketch 2 M, working 2 M)

Q.4

a) Factors affecting compressed air and RAC systems. 4 M

i) Compressed air system:

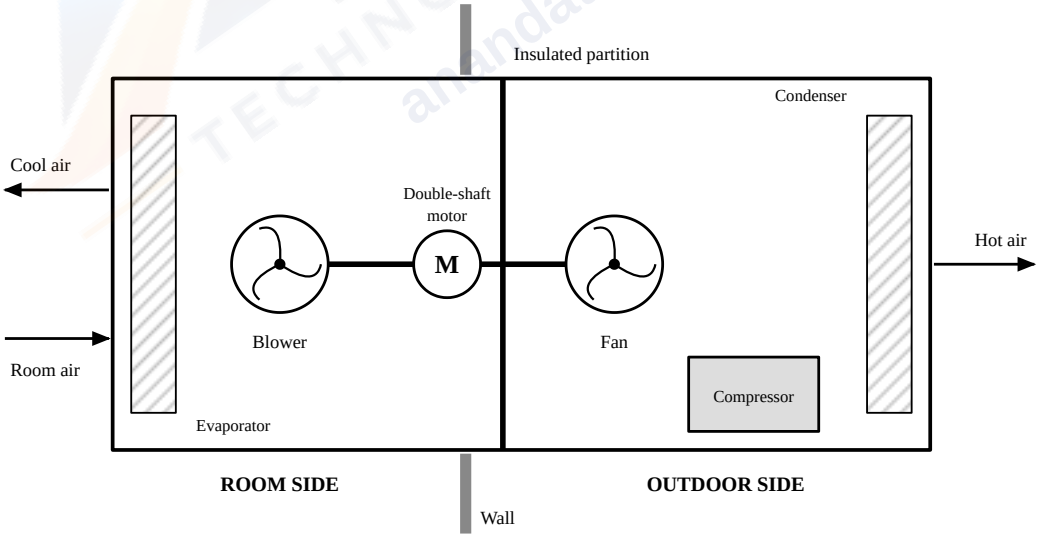
1. Air leaks in pipes, joints, valves, hoses and couplings (often 20–30 % of output is wasted).
2. Pressure drop due to small / long distribution pipes, clogged filters and dryers.
3. Generating air at a pressure higher than required by end-uses.
4. High intake air temperature and dusty intake air (1 °C rise in intake temperature ≈ 0.33 % more power).

ii) Refrigeration and air-conditioning system:

1. Dirty condenser and evaporator coils, choked filters (poor heat transfer).
2. Wrong refrigerant charge (leakage or over-charge).
3. High condensing temperature (poor ventilation of condenser / high ambient) and too low evaporator setting.
4. Heat gain through walls, doors, glass and infiltration; poor insulation of pipes and ducts.

(Four factors each, ½ M each)

b) Window air conditioner. 4 M



Window air conditioner (sectional plan)

Construction: All parts are in one cabinet fitted in a wall / window opening. An insulated partition divides it into the room side (evaporator, blower, air filter, front grille and controls) and the outdoor

side (hermetic compressor, condenser, propeller fan). One double-shaft motor drives both the blower and the condenser fan. A capillary tube is the expansion device.

Working: The blower draws room air through the filter and over the cold evaporator; the air is cooled and dehumidified and returned to the room through adjustable louvers. Condensate drips to the base and flows to the outdoor side (where the fan slinger sprays it on the condenser). On the outdoor side the fan blows atmospheric air over the condenser, removing the heat absorbed in the room plus the compressor work. A thermostat switches the compressor to maintain room temperature.

(Sketch 2 M, working 2 M)

c) Energy saving opportunities for office AC. 4 M

1. Set thermostat at 24–26 °C (each 1 °C rise saves about 3–5 % energy); use timers / occupancy sensors so units switch off when the room is empty.
2. Clean air filters every 2–4 weeks and clean evaporator / condenser coils regularly; dirty coils raise condensing pressure and power.
3. Keep doors and windows closed; seal gaps; provide door closers, air curtains and false ceiling / roof insulation to reduce heat gain.
4. Shade the outdoor unit and give free air flow around it; avoid hot-air recirculation.
5. Use window films, blinds or curtains to cut solar heat; switch off unused lights and computers (internal heat).
6. Replace old units by BEE 5-star / inverter units; use energy-efficient (EC / BLDC) fan motors.

(Any four with reason, 1 M each)

d) Advantages of multistage compression; minimum-work condition. 4 M

1. Work required is less (process nearer to isothermal) — saving in power.
2. Lower delivery temperature per stage: better lubrication, less risk of oil carbonisation, safer.
3. Higher volumetric efficiency, because the pressure ratio per stage is small (less re-expansion of clearance air).
4. Better mechanical balance and more uniform torque; lighter cylinders for HP stages; moisture can be drained in the intercooler.

Condition for minimum work (two stages, perfect intercooling, same n in both stages): the work in each stage is equal and the intermediate pressure is the geometric mean:

$$p_2 = \sqrt{(p_1\, p_3)}, \text{ i.e. } p_2/p_1 = p_3/p_2$$

(Advantages 2 M, condition 2 M)

e) Compressor with clearance — numerical. 4 M

Given: $V_a = 2.5$ m³/min = 0.04167 m³/s, $p_1 = 1.013$ bar = 101.3 kPa, $p_2 = 8$ bar, $C = V_c/V_s = 0.05$, $n = 1.3$; $r = p_2/p_1 = 7.8973$

$$\eta_{vol} = 1 + C - C\,(p_2/p_1)^{1/n} = 1 - C[(p_2/p_1)^{1/n} - 1]$$

$(p_2/p_1)^{1/n} = 7.8973^{0.7692} = 4.9020$

(i) $\eta_{vol} = 1 - 0.05 \times (4.9020 - 1) =$ **0.8049 = 80.49 %**

Swept volume needed = $V_a/\eta_{vol} = 2.5/0.8049 = 3.106$ m³/min

(ii) With clearance, the work depends only on the volume actually sucked ($V_1 - V_4$) = free air delivered:

$$IP = [n/(n-1)]\, p_1(V_1 - V_4)\, [(p_2/p_1)^{(n-1)/n} - 1]$$

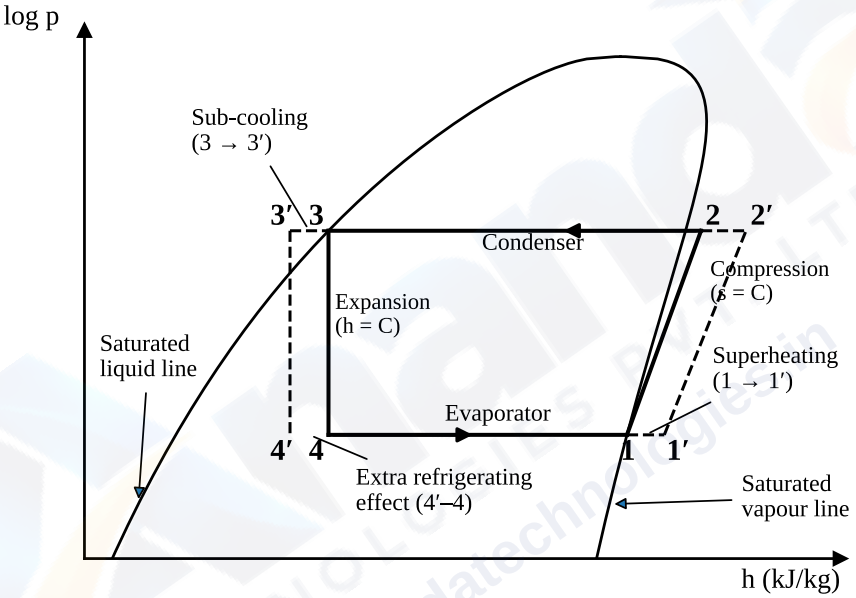
$= 4.3333 \times 101.3 \times 0.04167 \times (1.6111 - 1) =$ **11.18 kW**

(iii) Isothermal power = $p_1 V_a \ln r = 101.3 \times 0.04167 \times 2.0665 = 8.72$ kW; $\eta_{iso} = 8.72/11.18 =$ **78.0 %**

(η_{vol} 1½ M, IP 1½ M, η_{iso} 1 M)

Q.5

a) R-134a cycle with superheating and sub-cooling. 6 M



Cycle 1′-2′-3′-4′ (with superheating and sub-cooling) compared with simple saturated cycle 1-2-3-4

Actual cycle: $h_1 = 396.9$, $h_2 = 431.6$, $h_3 = h_4 = 249.0$ kJ/kg (throttling: $h_4 = h_3$)

(i) Refrigerating effect RE = $h_1 - h_4 = 396.9 - 249.0 =$ **147.9 kJ/kg**

Capacity Q = 2 × 3.5 = 7.00 kW

(ii) $\dot{m} = Q/RE = 7.00/147.9 =$ **0.0473 kg/s = 2.840 kg/min**

Work per kg = $h_2 - h_1 = 431.6 - 396.9 = 34.7$ kJ/kg

(iii) Power = $\dot{m}(h_2 - h_1) = 0.0473 \times 34.7 =$ **1.64 kW**

(iv) COP = RE/W = 147.9/34.7 = **4.262**

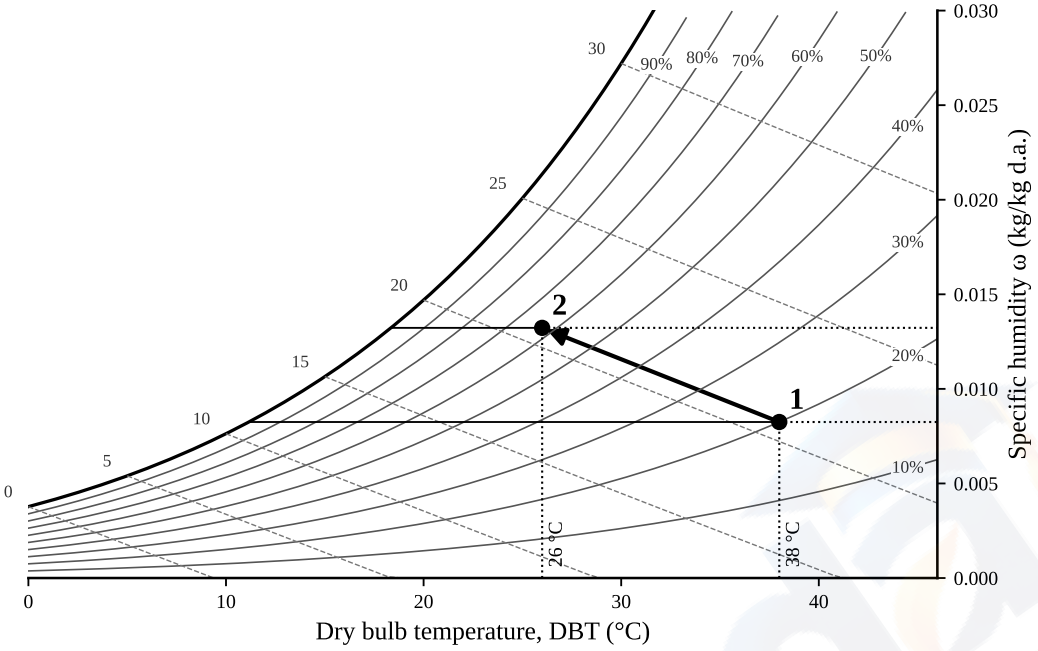
Simple saturated cycle: RE = 392.7 − 256.4 = 136.3 kJ/kg; W = 426.5 − 392.7 = 33.8 kJ/kg; COP = 4.033

(v) Increase in COP = (4.262 − 4.033)/4.033 × 100 = **5.7 %**. Sub-cooling raises the refrigerating effect without extra work; useful superheating inside the evaporator also adds refrigerating effect.

(Heat rejected in condenser = $\dot{m}(h_2 - h_3)$ = 0.0473 × (431.6 − 249.0) = 8.64 kW = Q + W, check.)

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

b) Adiabatic humidifier numerical. 6 M



Adiabatic humidification (evaporative cooling) 1 → 2 along the constant WBT line

Inlet state 1 (38 °C, 20 % RH):

- Saturation pressure at 38 °C (steam table): p_s = 6.6315 kPa
- Vapour pressure $p_v = \varphi\, p_s = 0.200 \times 6.6315 = 1.3263$ kPa
- Specific humidity $\omega = 0.622\, p_v/(p - p_v) = 0.622 \times 1.3263/(101.325 - 1.3263)$ = **0.00825 kg/kg d.a.**
- Dew point temperature = saturation temperature at p_v = **11.2 °C**
- Enthalpy $h = 1.005\, t + \omega(2500 + 1.88\, t) = 1.005 \times 38 + 0.00825(2500 + 1.88 \times 38)$ = **59.40 kJ/kg d.a.**
- Specific volume $v = R_a T/(p - p_v) = 0.287 \times 311.15/(101.325 - 1.3263)$ = **0.8930 m³/kg d.a.**
- Wet bulb temperature (from chart, along constant-WBT line through the state) = **20.8 °C**

In an adiabatic humidifier the water evaporates by taking heat from the air itself, so the process follows the **constant wet bulb temperature line** (enthalpy ≈ constant).

(i) Outlet: DBT = 26 °C, WBT = WBT_1 = **20.8 °C**; from the chart φ_2 = **63 %**, ω_2 = 0.01323 kg/kg d.a., DPT_2 = **18.3 °C**

Mass of dry air $\dot{m}_a = V/v_1 = 150/0.8930 = 167.97$ kg/min

(ii) Water vapour added = $\dot{m}_a(\omega_2 - \omega_1) = 167.97 \times (0.01323 - 0.00825) = 0.836$ kg/min = **50.2 kg/h**

(iii) Change in sensible heat = $\dot{m}_a\, c_{p,humid}\, (t_1 - t_2)$, $c_{p,humid} = 1.005 + 1.88\, \omega_1 = 1.0205$ kJ/kg·K

= 167.97 × 1.0205 × (38 − 26) = 2057.0 kJ/min = **34.28 kW** (sensible heat removed; it re-appears as latent heat of the added vapour, $h_1 \approx h_2$: 59.40 ≈ 59.84 kJ/kg)

Values are calculated from psychrometric relations at p = 1.01325 bar (saturation pressures from steam tables); readings taken from the chart will agree within about ±0.5 °C and ±1 kJ/kg.

(Chart 1 M, outlet properties 2 M, water added 1½ M, sensible heat 1½ M)

c) Engine trial and heat balance sheet. 6 M

A = (π/4) × 0.15² = 0.017671 m²; n = N/2 = 450 (four-stroke)

IP = p_mLAn/60 = 800 × 0.18 × 0.017671 × 450/60 = **19.09 kW**

BP = 2πNT/60000 = 2π × 900 × 160/60000 = **15.08 kW**

η_m = BP/IP = 15.08/19.09 = **79.01 %**

Heat supplied = (4.4/3600) × 42,500 = 51.94 kW; η_{bth} = 15.08/51.94 = **29.03 %** (η_{ith} = 36.74 %)

Heat balance per minute:

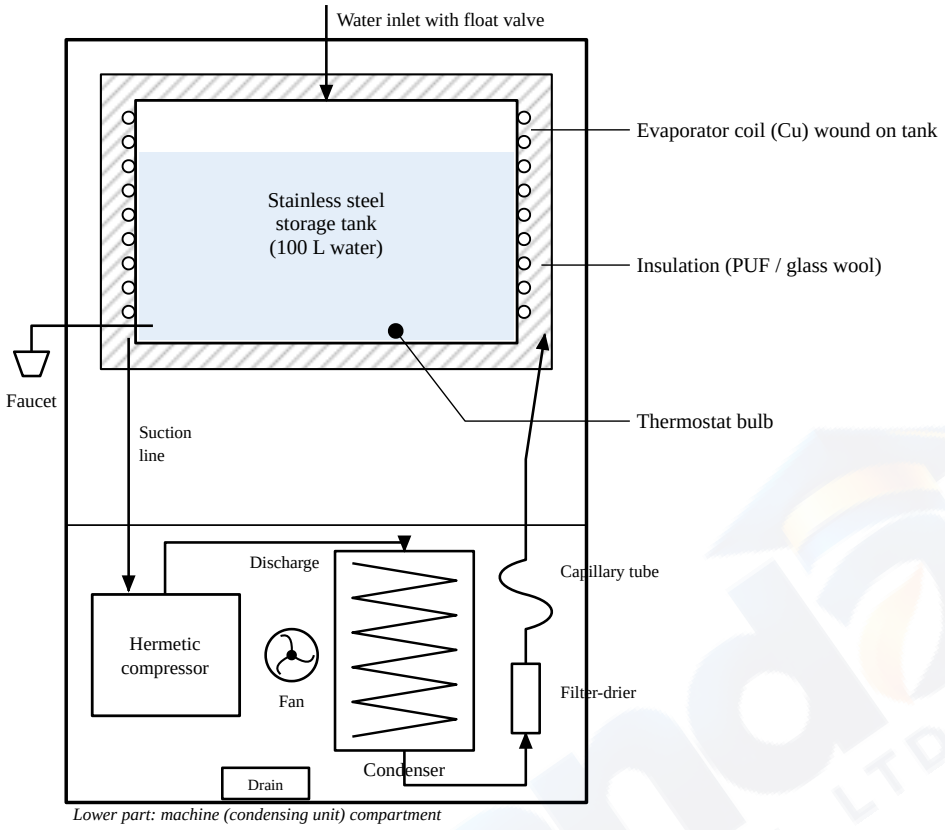
- Heat supplied = (4.4/60) × 42,500 = **3116.67 kJ/min**
- BP = 15.080 × 60 = **904.78 kJ/min**
- Cooling water = 8.5 × 4.187 × 22 = **782.97 kJ/min**
- Exhaust gas = (140 + 4.4)/60 = 2.4067 kg/min → 2.4067 × 1.1 × (380 − 30) = **926.57 kJ/min**
- Unaccounted = **502.35 kJ/min** (by difference)

Heat supplied (kJ/min)			**Heat distributed (kJ/min)**		
Item	**kJ/min**	**%**	**Item**	**kJ/min**	**%**
Heat supplied by fuel (m_f × CV)	3116.67	100.00	Heat equivalent of BP	904.78	29.03
			Heat carried by cooling water	782.97	25.12
			Heat carried by exhaust gases	926.57	29.73
			Heat unaccounted (by difference)	502.35	16.12
Total	**3116.67**	**100.00**	**Total**	**3116.67**	**100.00**

(IP, BP, η_m 2 M; η_{bth} 1 M; heat balance sheet 3 M)

Q.6

a) Storage-type water cooler — components and functions. 6 M



Storage type water cooler — construction (sectional sketch)

Component	Function
Hermetic compressor	Compresses low-pressure refrigerant vapour to high pressure and temperature; motor and compressor sealed in one shell (no leakage).
Air-cooled condenser with fan	Rejects heat of the refrigerant to atmospheric air; vapour condenses to liquid.
Filter-drier	Removes moisture and foreign particles from the liquid refrigerant; prevents choking of capillary.
Capillary tube	Throttling device: reduces pressure and temperature of refrigerant; maintains pressure difference.
Evaporator coil on tank	Refrigerant evaporates, absorbing heat from water through the tank wall — cooling effect.
Stainless-steel storage tank	Stores cooled drinking water hygienically; does not rust.
Insulation (PUF)	Reduces heat gain from surroundings to the cold tank.
Thermostat	Senses water temperature and switches the compressor ON / OFF to hold the set temperature.
Float valve and faucets	Float valve keeps the tank full from the supply line; faucets deliver cold water.

(Sketch 3 M, functions 3 M)

b) Morse test numerical and purpose. 6 M

When one cylinder is cut off, the speed is brought back to the original value by reducing the load. The drop in BP equals the indicated power of the cylinder that was cut off: $IP_n = BP - BP_n$.

Cylinder	IP = BP − BP_n	IP (kW)
1	18.4 − 12.9	5.50
2	18.4 − 12.8	5.60
3	18.4 − 13.0	5.40
4	18.4 − 12.7	5.70
	Total IP	**22.20**

(i) IP of engine = **22.20 kW**

(ii) η_m = BP/IP = 18.4/22.20 = **82.88 %**; friction power = 22.20 − 18.4 = 3.80 kW

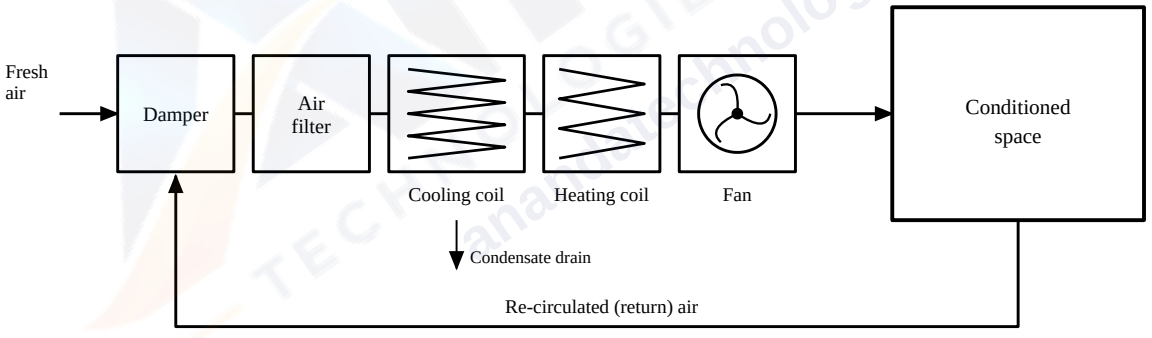
(iii) Heat supplied = (6.2/3600) × 44,000 = 75.78 kW

η_{bth} = 18.4/75.78 = **24.28 %**; η_{ith} = 22.20/75.78 = **29.30 %**; BSFC = 6.2/18.4 = 0.3370 kg/kWh

Purpose of Morse test: to find the indicated power (and hence friction power and mechanical efficiency) of a multi-cylinder engine at a given speed without using an indicator, and to check whether all cylinders develop equal power.

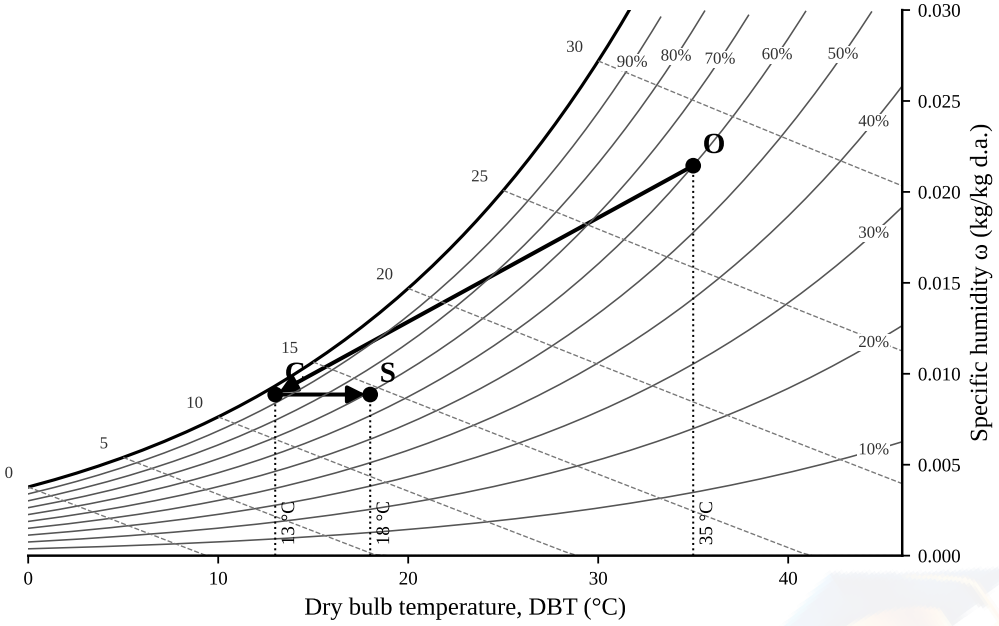
(IP table 3 M, η_m 1 M, efficiencies 1 M, purpose 1 M)

c) Summer air-conditioning system. 6 M



Summer air-conditioning system (cooling and dehumidification with reheat)

Working: Fresh outdoor air and re-circulated room air are mixed in controlled proportion by dampers and passed through the air filter. The mixture flows over the **cooling coil** (evaporator / chilled-water coil) whose surface is below the dew point of the air; the air is cooled and its moisture condenses (cooling and dehumidification). The air leaving the coil is nearly saturated and too cold, so it passes over a **heating (reheat) coil** for sensible heating to the required supply temperature and RH. The fan delivers it through ducts to the conditioned space, where it picks up the sensible and latent heat. Part of the room air is exhausted and the rest re-circulated.



Summer AC on chart: O → C cooling and dehumidification; C → S reheat (sensible heating)

Example states shown: O = 35 °C, 60 % RH; C = 13 °C (near saturation, ω = 0.0089); reheat to S = 18 °C at the same ω (RH ≈ 69 %) before supply.

(Sketch 2 M, working 3 M, chart 1 M)

Note: This is a **model question paper prepared by Ananda Technologies** only for practice and exam preparation, based on the pattern of previous MSBTE question papers and the K-scheme syllabus. It is **not an official MSBTE paper**, and it does **not** mean that these questions will come in the exam. Answers follow standard textbook points; marks shown are indicative. Free for students — anandatechnologies.in