

**PRACTICE PAPER – SET 3**  
**NATIONAL CERTIFICATION EXAMINATION FOR**  
**ENERGY MANAGERS & ENERGY AUDITORS**  
**PAPER – 3: ENERGY EFFICIENCY IN ELECTRICAL UTILITIES**  
**ANSWER KEY, EXPLANATIONS AND MODEL ANSWERS**

**Duration: 3 HRS      Max. Marks: 150**

**General instructions:**

- Please check that this question paper contains 64 questions
- The question paper is divided into three sections
- All questions in all three sections are compulsory
- All parts of a question should be answered at one place
- Use of a non-programmable scientific calculator is permitted

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**Section – I: OBJECTIVE TYPE      Marks: 50 x 1 = 50**

1. Answer all 50 questions
2. Each question carries one mark
3. Choose the most appropriate option (a, b, c or d)

**1** A failed float switch lets a transfer pump overflow an overhead tank for 3 h every day of the year. During overflow the pump delivers 40 m<sup>3</sup>/h against a static lift of 26 m plus 4 m of friction. Pump efficiency is 62 % and motor efficiency 88 %. The electrical energy wasted in a year is about:

*[5P3CEM001 | Chapter 6: Pumps and Pumping System | 6.6 Overhead tank overflow from a failed float switch – energy wasted]*

- a) 6,560 kWh
- b) 5,780 kWh
- c) 3,580 kWh
- d) 5,690 kWh

**Answer & explanation**

Correct answer: (a) ≈ 6,560 kWh.

Given:  $Q = 40 \text{ m}^3/\text{h}$ ;  $H = 26 + 4 = 30 \text{ m}$ ;  $\eta_{\text{pump}} 0.62$ ;  $\eta_m 0.88$ ;  $3 \text{ h} \times 365 \text{ days}$ .

Formula:  $\text{input kW} = \rho g Q H / (\eta_{\text{pump}} \times \eta_m)$ ;  $\text{waste} = \text{kW} \times \text{hours}$ .

Substitution:  $\text{hydraulic power} = 1,000 \times 9.81 \times (40/3,600) \times 30 = 3.27 \text{ kW}$ ;  $\text{input} = 3.27 / (0.62 \times 0.88) = 5.99 \text{ kW}$ ;  $\times 1,095 \text{ h}$ .

Result: ≈ **6,560 kWh a year wasted, plus about 43,800 m<sup>3</sup> of treated water.**

Why the others are wrong: (b) ignores motor efficiency; (c) gives only hydraulic energy, ignoring both efficiencies; (d) uses the static lift alone and forgets friction head. Level controllers, overflow alarms and periodic checks cost

little. In large campuses and housing societies the water lost is often worth more than the electricity, so both should be reported.

Ref: BEE Guidebook Paper 3, Ch. 6 (Pumps – energy conservation opportunities in pumping systems).

**2** A building's AHUs move 72,000 m<sup>3</sup>/h of air (density 1.2 kg/m<sup>3</sup>) through their cooling coils, removing 5.5 g of moisture per kg of air on humid days. If the condensate is collected for cooling tower makeup, it yields about:

[5P3CEM002 | Chapter 4: HVAC and Refrigeration System | 4.16 AHU condensate recovery as cooling tower makeup]

- a) 396 litres/h
- b) 570 litres/h
- c) 475 litres/h**
- d) 330 litres/h

#### Answer & explanation

Correct answer: (c)  $\approx$  475 litres/h.

Given:  $Q = 72,000 \text{ m}^3/\text{h}$ ;  $\rho = 1.2 \text{ kg/m}^3$ ;  $\Delta W = 0.0055 \text{ kg/kg}$ .

Formula:  $\dot{m}_{\text{air}} = Q \times \rho$ ; condensate =  $\dot{m}_{\text{air}} \times \Delta W$  (1 kg of water  $\approx$  1 litre).

Substitution:  $\dot{m}_{\text{air}} = 72,000 \times 1.2 = 86,400 \text{ kg/h}$  (24 kg/s);  $86,400 \times 0.0055$ .

Result: **condensate  $\approx$  475 litres/h, i.e. about 11.4 m<sup>3</sup> a day over 24 humid hours.**

Why the others are wrong: (a) multiplies the volume flow by  $\Delta W$  without converting to mass; (b) applies the density twice; (d) divides by density instead of multiplying. This water is close to distilled, cold and low in dissolved solids, so it is ideal tower makeup and also lets the tower run at higher cycles of concentration. In humid Indian cities, large buildings can meet a meaningful share of their tower makeup this way at very low cost.

Ref: BEE Guidebook Paper 3, Ch. 4 (HVAC – psychrometrics, dehumidification); Ch. 7 (Cooling towers – water conservation).

**3** A 50 kVAr, 415 V, 3-phase capacitor unit is tested at 410 V and draws 52 A per phase. Correcting the rated output to the test voltage, the unit is delivering what percentage of its expected kVAr?

[5P3CEM003 | Chapter 1: Electrical Systems | 1.4 Capacitor health check from measured current]

- a) 73.9 %
- b) 72.1 %
- c) 75.7 %**
- d) 43.7 %

#### Answer & explanation

Correct answer: (c)  $\approx$  75.7 %.

Given: rated 50 kVAr at 415 V; test 410 V, 52 A.

Formula: actual kVAr =  $\sqrt{3} \times V \times I / 1,000$ ; expected kVAr at test voltage = rated  $\times (V_{\text{test}} / V_{\text{rated}})^2$ .

Substitution: actual =  $1.732 \times 410 \times 52 / 1,000 = 36.93 \text{ kVAr}$ ; expected =  $50 \times (410/415)^2 = 50 \times 0.9760 = 48.80 \text{ kVAr}$ ;  $36.93 / 48.80$ .

Result: **output  $\approx$  75.7 % of expected; the unit has lost about a quarter of its capacitance and should be replaced.**

Why the others are wrong: (a) compares with the 50 kVAr nameplate without correcting for the lower voltage; (b) corrects the rating in the wrong direction,  $(415/410)^2$ ; (d) uses single-phase  $V \times I$  without  $\sqrt{3}$ . Elements degrade with heat and harmonics, and an APFC panel full of tired units cannot hold its PF set point. Measuring current per step is a quick audit test.

Ref: BEE Guidebook Paper 3, Ch. 1 (Electrical system – capacitors, performance and maintenance).

**4** A campus wants to shift 2,400 TR ( $\approx 8,439$  kW)·h of afternoon cooling to a stratified chilled-water tank charged at night, with a usable temperature difference of 8 K and a figure of merit (usable fraction of theoretical capacity) of 0.90. Taking  $c_p$  of water 4.186 kJ/kg·K and 1,000 kg/m<sup>3</sup>, the tank volume needed is about:

[5P3CEM004 | Chapter 4: HVAC and Refrigeration System | 4.8 Stratified chilled-water thermal storage – tank volume]

- a) **1,008 m<sup>3</sup>**
- b) 907 m<sup>3</sup>
- c) 816 m<sup>3</sup>
- d) 1,120 m<sup>3</sup>

#### Answer & explanation

Correct answer: (a)  $\approx 1,008$  m<sup>3</sup>.

Given: storage 2,400 TR ( $\approx 8,439$  kW)·h;  $\Delta T$  8 K; figure of merit 0.90; 1 TR = 3.516 kW.

Formula: energy = TR·h  $\times$  3.516  $\times$  3,600 kJ; capacity per m<sup>3</sup> = 1,000  $\times$  4.186  $\times$   $\Delta T$  kJ;  $V = \text{energy} / (\text{capacity per m}^3 \times \text{FOM})$ .

Substitution: energy = 2,400  $\times$  3.516  $\times$  3,600 =  $3.038 \times 10^7$  kJ; per m<sup>3</sup> = 33,488 kJ;  $\times 0.90 = 30,139$  kJ;  $3.038 \times 10^7 / 30,139$ .

Result: **tank volume  $\approx 1,008$  m<sup>3</sup>.**

Why the others are wrong: (b) ignores the figure of merit, so the tank would fall short because of the thermocline and mixing; (c) multiplies by the figure of merit instead of dividing; (d) divides by it twice. Chilled-water storage needs roughly six to eight times the volume of ice storage for the same TR·h, but chillers stay at normal evaporator temperatures. Widening the usable  $\Delta T$  is the cheapest way to shrink the tank.

Ref: BEE Guidebook Paper 3, Ch. 4 (HVAC – thermal energy storage and load shifting).

**5** A centrifugal blower develops a total pressure of 6,000 Pa at 75 % efficiency, with its motor outside the air stream. Taking air density 1.2 kg/m<sup>3</sup> and  $c_p$  1.005 kJ/kg·K, and assuming all shaft power ends up as heat in the air, the temperature rise of the air through the blower is about:

[5P3CEM005 | Chapter 5: Fans and Blowers | 5.6 Temperature rise of air across a high-pressure blower]

- a) 4.98 K
- b) 3.73 K
- c) **6.63 K**
- d) 7.96 K

#### Answer & explanation

Correct answer: (c)  $\approx 6.63$  K.

Given:  $\Delta p = 6,000$  Pa;  $\eta = 0.75$ ;  $\rho = 1.2$  kg/m<sup>3</sup>;  $c_p = 1,005$  J/kg·K.

Formula: shaft power per m<sup>3</sup> =  $\Delta p / \eta$  (J/m<sup>3</sup>); temperature rise =  $\Delta p / (\eta \times \rho \times c_p)$ .

Substitution:  $6,000 / (0.75 \times 1.2 \times 1,005) = 6,000 / 904.5$ .

Result: **air temperature rises by about 6.6 K.**

Why the others are wrong: (a) ignores the efficiency, so it counts only the useful air power and not the loss that also heats the air; (b) multiplies by efficiency instead of dividing; (d) leaves out the density, treating each m<sup>3</sup> as 1 kg. This rise is a quick field check of blower efficiency with two thermometers, and it is also a hidden load: in air-conditioned supply systems every pascal of fan pressure must be removed again by the cooling coil. Lowering system pressure therefore saves both fan and chiller energy.

Ref: BEE Guidebook Paper 3, Ch. 5 (Fans and blowers – fan performance, efficiency).

**6** On a Sunday with all users isolated, compressors are stopped and the pressure in the system (receiver 6 m<sup>3</sup> plus piping 2 m<sup>3</sup>) falls from 7.0 to 5.5 bar(g) in 6 minutes. Taking atmospheric pressure as 1.013 bar, the average leakage in free air is about:

[5P3CEM006 | Chapter 3: Compressed Air System | 3.5 Weekend leak test by receiver pressure decay]

- a) 1.48 m<sup>3</sup>/min
- b) 1.97 m<sup>3</sup>/min**
- c) 0.99 m<sup>3</sup>/min
- d) 3.31 m<sup>3</sup>/min

#### Answer & explanation

Correct answer: (b)  $\approx 1.97 \text{ m}^3/\text{min}$ .

Given:  $V = 6 + 2 = 8 \text{ m}^3$ ;  $P_1 = 7.0 \text{ bar(g)}$ ;  $P_2 = 5.5 \text{ bar(g)}$ ;  $t = 6 \text{ min}$ ;  $P_a = 1.013 \text{ bar}$ .

Formula: leakage (free air) =  $V \times (P_1 - P_2) / (P_a \times t)$ , an isothermal mass balance on the stored air.

Substitution:  $8 \times 1.5 / (1.013 \times 6) = 12 / 6.078$ .

Result: **average leakage  $\approx 1.97 \text{ m}^3/\text{min}$  of free air over the 7.0–5.5 bar band.**

Why the others are wrong: (a) uses only the 6 m<sup>3</sup> receiver and ignores the volume held in the pipework; (c) halves the result on a mistaken "average pressure" argument, though the pressure difference already captures the mass lost; (d) adds atmospheric pressure to the pressure drop, which is wrong because only the change in stored mass leaks away. Leaks run at the higher working pressure during production, so the production-time loss is slightly higher still.

Ref: BEE Guidebook Paper 3, Ch. 3 (Compressed air system – leakage test methods).

**7** A centrifugal pump absorbs 22.0 kW shaft power when handling water at its duty point. The same pump is now used, at the same flow and speed, for a brine of specific gravity 1.25 with viscosity close to water. Its shaft power will be about:

[5P3CEM007 | Chapter 6: Pumps and Pumping System | 6.3 Pumping a denser liquid – power and head]

- a) 27.5 kW**
- b) 22.0 kW
- c) 34.4 kW
- d) 17.6 kW

#### Answer & explanation

Correct answer: (a)  $\approx 27.5 \text{ kW}$ .

Given:  $P_{\text{water}} = 22.0 \text{ kW}$ ;  $SG = 1.25$ ; same flow, speed and viscosity.

Formula: a centrifugal pump develops the same head in metres of liquid at a given flow and speed; power =  $\rho g Q H / \eta$ , so power  $\propto SG$ .

Substitution:  $22.0 \times 1.25$ .

Result: **shaft power  $\approx 27.5 \text{ kW}$ ; head in metres is unchanged, but discharge pressure rises by 25 %.**

Why the others are wrong: (b) assumes power is independent of density; (c) squares the specific gravity; (d) divides by it. The practical consequence is motor overload: a pump that ran its motor at 85 % on water would now need about 106 %. When a pump is moved to denser service, check the motor rating and the casing pressure rating before running.

Ref: BEE Guidebook Paper 3, Ch. 6 (Pumps – pump characteristics, power and effect of liquid properties).

**8** An HT consumer's meter records 4,80,000 kWh, 2,10,000 kVAh lagging and 30,000 kVAh leading (night-time over-compensation) in a month. The tariff bills on kVAh and treats leading kVAh as if it were lagging. The average power factor used in the bill is about:

[5P3CEM008 | Chapter 1: Electrical Systems | 1.2 kVAh billing – monthly PF when leading kVAh counts as lagging]

- a) 0.936
- b) 0.916
- c) 0.667
- d) 0.894**

**Answer & explanation**

Correct answer: (d)  $\approx 0.894$ .

Given: 4,80,000 kWh; 2,10,000 kVAh lag; 30,000 kVAh lead, counted as lag.

Formula:  $\text{kVAh} = \sqrt{(\text{kWh})^2 + (\text{kVAh})^2}$  with  $\text{kVAh} = \text{lag} + \text{lead}$ ;  $\text{PF} = \text{kWh} / \text{kVAh}$ .

Substitution:  $\text{kVAh} = 2,40,000$ ;  $\text{kVAh} = \sqrt{(4,80,000)^2 + (2,40,000)^2} = 5,36,656$ ;  $4,80,000 / 5,36,656$ .

Result: **average PF  $\approx 0.894$ ; the bill is raised on about 5.37 lakh kVAh, about 12 % more than kWh.**

Why the others are wrong: (a) nets the leading kVAh off the lagging, which this tariff does not allow; (b) simply ignores the leading kVAh; (c) divides kWh by the arithmetic sum of kWh and kVAh instead of the vector sum.

Here over-compensation at night costs as much as under-compensation by day, so APFC steps must be set to drop out at light load rather than left on.

Ref: BEE Guidebook Paper 3, Ch. 1 (Electrical system – electricity billing and power factor).

**9** A mall entrance admits 3,000 m<sup>3</sup>/h of outdoor air (density 1.2 kg/m<sup>3</sup>) with an enthalpy 30 kJ/kg higher than room air, for 12 h a day on 360 days. An air curtain cuts this infiltration by 70 % and draws 1.5 kW. With the cooling plant at 1.1 kW/TR, the net annual saving is about:

[5P3CEM009 | Chapter 10: Energy Conservation in Buildings and ECBC | 10.4 Air curtain at a busy air-conditioned entrance]

- a) 21,900 kWh**
- b) 28,380 kWh
- c) 17,170 kWh
- d) 43,800 kWh

**Answer & explanation**

Correct answer: (a)  $\approx 21,900$  kWh.

Given:  $Q = 3,000$  m<sup>3</sup>/h;  $\rho = 1.2$  kg/m<sup>3</sup>;  $\Delta h = 30$  kJ/kg; 4,320 h; 70 % reduction; curtain 1.5 kW; 1.1 kW/TR.

Formula: infiltration load =  $Q\rho/3,600 \times \Delta h$  (kW); plant power = load / 3.516  $\times$  kW/TR; net =  $(0.70 \times \text{plant power} - \text{curtain power}) \times \text{hours}$ .

Substitution: load =  $1.0$  kg/s  $\times 30 = 30.0$  kW = 8.53 TR; plant power = 9.39 kW;  $0.70 \times 9.39 = 6.57$  kW; net =  $5.07$  kW  $\times 4,320$ .

Result: **net saving  $\approx 21,900$  kWh a year.**

Why the others are wrong: (b) ignores the air curtain's own fan power; (c) leaves out the air density, treating m<sup>3</sup> as kg; (d) runs the curtain for 24 h, though the entrance is open only 12 h. Air curtains work best on doors kept open for long periods. Vestibules or revolving doors do better where wind pressure is high.

Ref: BEE Guidebook Paper 3, Ch. 10 (Energy conservation in buildings – envelope, infiltration control).

**10** A process exhaust duct carries  $8 \text{ m}^3/\text{s}$  at  $15 \text{ m/s}$  (air density  $1.2 \text{ kg/m}^3$ ) through four sharp unvaned elbows with a loss coefficient  $K = 1.2$  each. They are replaced by vaned elbows with  $K = 0.25$ . With combined fan, drive and motor efficiency of  $65 \%$ , the reduction in input power is about:

[5P3CEM010 | Chapter 5: Fans and Blowers | 5.7 Duct fittings – replacing unvaned elbows with vaned elbows]

- a) 12.63 kW
- b) 6.31 kW**
- c) 4.10 kW
- d) 2.67 kW

**Answer & explanation**

Correct answer: (b)  $\approx 6.31 \text{ kW}$ .

Given:  $Q = 8 \text{ m}^3/\text{s}$ ;  $v = 15 \text{ m/s}$ ;  $\rho = 1.2 \text{ kg/m}^3$ ; 4 elbows;  $K$  from 1.2 to 0.25; overall  $\eta = 0.65$ .

Formula: velocity pressure  $VP = \rho v^2 / 2$ ; fitting loss  $\Delta p = K \times VP$ ; power saved  $= Q \times \Sigma \Delta p_{\text{saved}} / \eta$ .

Substitution:  $VP = 1.2 \times 225 / 2 = 135 \text{ Pa}$ ; saving per elbow  $= 0.95 \times 135 = 128.3 \text{ Pa}$ ;  $\times 4 = 513 \text{ Pa}$ ;  $8 \times 513 / 0.65 = 6,314 \text{ W}$ .

Result: **input power falls by about 6.31 kW.**

Why the others are wrong: (a) forgets the  $\frac{1}{2}$  in the velocity pressure, doubling the loss; (c) gives the air power saved and does not divide by the  $65 \%$  efficiency; (d) multiplies by the efficiency instead of dividing. Fitting losses scale with velocity squared, so in high-velocity ducts, elbows, abrupt transitions and dampers can cost more than the straight runs.

Ref: BEE Guidebook Paper 3, Ch. 5 (Fans and blowers – system resistance, energy saving opportunities).

**11** A roof ventilation fan is driven by a 4/8-pole two-speed motor. At 1,470 rpm the motor draws  $30.0 \text{ kW}$  at  $92 \%$  efficiency. At night it is switched to 735 rpm, where the motor efficiency is  $85 \%$ . Assuming the fan follows the cube law, the night-time input power is about:

[5P3CEM011 | Chapter 2: Electrical Motors | 2.9 Pole-changing (two-speed) motor on a ventilation fan]

- a) 3.75 kW
- b) 8.12 kW
- c) 2.93 kW
- d) 4.06 kW**

**Answer & explanation**

Correct answer: (d)  $\approx 4.06 \text{ kW}$ .

Given: input at high speed  $30.0 \text{ kW}$ ;  $\eta_m$  high  $= 0.92$ , low  $= 0.85$ ; speed ratio  $735/1,470 = 0.5$ .

Formula: fan shaft power  $\propto N^3$ ; input  $=$  shaft power /  $\eta_m$  at that speed.

Substitution: shaft at high speed  $= 30.0 \times 0.92 = 27.6 \text{ kW}$ ; at half speed  $= 27.6 \times 0.5^3 = 3.45 \text{ kW}$ ; input  $= 3.45 / 0.85$ .

Result: **night input  $\approx 4.06 \text{ kW}$ , about 86 % less than at high speed.**

Why the others are wrong: (a) applies the cube law to input power and ignores the lower low-speed efficiency; (b) uses a square law, which holds for pressure, not power; (c) multiplies by the efficiency instead of dividing. A pole-changing motor gives only fixed steps, but for fans that need full and reduced duty only, it captures most of the cube-law saving without a VFD's cost and harmonics.

Ref: BEE Guidebook Paper 3, Ch. 2 (Electric motors – speed control, pole-changing motors).

**12** An office's thermal mass is at 28 °C at night. A night purge blows 20,000 m<sup>3</sup>/h of outdoor air (density 1.2 kg/m<sup>3</sup>, cp 1.005 kJ/kg·K) through the building for 6 h using fans drawing 4.0 kW; outdoor air averages 22 °C and leaves at an average of 26 °C. The cooling removed would otherwise be done next morning by a chilled-water plant at 1.0 kW/TR. The net electricity saved per night is about:

[5P3CEM012 | Chapter 4: HVAC and Refrigeration System | 4.16 Night purge ventilation – net saving after fan energy]

- a) 44.6 kWh
- b) 21.7 kWh**
- c) 45.7 kWh
- d) 14.1 kWh

**Answer & explanation**

Correct answer: (b) ≈ 21.7 kWh.

Given: Q = 20,000 m<sup>3</sup>/h; ρ = 1.2 kg/m<sup>3</sup>; cp = 1.005 kJ/kg·K; air warms 22 → 26 °C (4 K); 6 h; fans 4.0 kW; plant 1.0 kW/TR.

Formula: heat =  $\dot{m} c_p \Delta T \times h$ ; avoided = (kWh / 3.516) × kW/TR; net = avoided – fan kWh.

Substitution:  $\dot{m} = 6.667$  kg/s;  $6.667 \times 1.005 \times 4 = 26.8$  kW;  $\times 6$  h = 160.8 kWh thermal = 45.7 TR·h; avoided 45.7 kWh; fans 24 kWh.

Result: **net saving ≈ 21.7 kWh per night.**

Why the others are wrong: (a) uses the 6 K difference between building and outdoor air, but the air actually warms only 4 K; (c) ignores the fan energy; (d) drops the air density. Night purge pays only when outdoor air is well below building temperature and fan pressure is low, since fan kWh competes with a chiller giving about 3.5 kWh of cooling per kWh.

Ref: BEE Guidebook Paper 3, Ch. 4 (HVAC – energy saving opportunities); Ch. 10 (Energy conservation in buildings).

**13** An AHU fan must move 10 m<sup>3</sup>/s against filter, coil, ductwork and diffuser losses totalling 780 Pa, and the air leaves the system carrying a velocity pressure of 60 Pa. With a fan total efficiency of 70 %, the fan shaft power is about:

[5P3CEM013 | Chapter 5: Fans and Blowers | 5.6 Fan shaft power from system losses and outlet velocity pressure]

- a) 11.1 kW
- b) 8.4 kW
- c) 5.9 kW
- d) 12.0 kW**

**Answer & explanation**

Correct answer: (d) ≈ 12.0 kW.

Given: Q = 10 m<sup>3</sup>/s; system losses 780 Pa; exit velocity pressure 60 Pa;  $\eta_{\text{total}} = 0.70$ .

Formula: fan total pressure =  $\Sigma$  losses + exit velocity pressure; shaft power =  $Q \times \text{FTP} / \eta_{\text{total}}$ .

Substitution: FTP = 780 + 60 = 840 Pa;  $10 \times 840 / 0.70 = 12,000$  W.

Result: **shaft power ≈ 12.0 kW.**

Why the others are wrong: (a) omits the kinetic energy carried out of the system, which the fan must also supply; (b) gives the air power without dividing by efficiency; (c) multiplies by efficiency instead. Using total pressure with total efficiency, or static pressure with static efficiency, consistently is a frequent source of error in audit

reports. A clogged filter adding 150 Pa would raise shaft power by about 2.1 kW, which shows why filter  $\Delta p$  alarms matter.

Ref: BEE Guidebook Paper 3, Ch. 5 (Fans and blowers – fan performance assessment, power and efficiency).

**14** A 1,000 kVA, 11/0.433 kV distribution transformer has a no-load loss of 1.70 kW at rated voltage. The incoming supply stays at 11.6 kV all year because the tap is left on the wrong position. Taking no-load loss as varying with the square of the applied voltage, the extra energy lost in a year is about:

[5P3CEM014 | Chapter 1: Electrical Systems | 1.5 Transformer no-load loss at sustained over-voltage]

- a) 812 kWh
- b) 1,670 kWh**
- c) 2,570 kWh
- d) 1,520 kWh

#### Answer & explanation

Correct answer: (b)  $\approx$  1,670 kWh.

Given: no-load loss at 11.0 kV = 1.70 kW; actual voltage 11.6 kV; transformer energised 8,760 h a year.

Formula: no-load loss  $\propto V^2$ ; extra loss =  $P_o \times [(V/V_r)^2 - 1] \times 8,760$ .

Substitution:  $11.6 / 11.0 = 1.0545$ ; squared = 1.1121; extra =  $1.70 \times 0.1121 = 0.1905$  kW;  $\times 8,760$ .

Result: **extra no-load loss  $\approx$  1,670 kWh a year.**

Why the others are wrong: (a) lets the loss rise only in proportion to voltage, which halves the effect; (c) applies a cube law, which overstates it; (d) uses 8,000 operating hours, but a transformer draws its no-load loss whenever it is energised, whether or not the plant works. The over-voltage also pushes up motor magnetising current and lighting power, so correcting the tap usually saves far more downstream than in the transformer itself.

Ref: BEE Guidebook Paper 3, Ch. 1 (Electrical system – transformer losses).

**15** A conveyor needs 18 kW at the gearbox output. The existing worm reducer is 72 % efficient; a helical-bevel replacement is 95 % efficient. Motor efficiency is 90 % in both cases. Over 6,000 h a year, the energy saved is about:

[5P3CEM015 | Chapter 2: Electrical Motors | 2.5 Transmission losses – worm gear replaced by helical-bevel gear]

- a) 36,320 kWh
- b) 40,350 kWh**
- c) 24,840 kWh
- d) 32,680 kWh

#### Answer & explanation

Correct answer: (b)  $\approx$  40,350 kWh.

Given: output 18 kW;  $\eta$  worm 0.72;  $\eta$  helical 0.95;  $\eta_m$  0.90; 6,000 h.

Formula: motor input = output / ( $\eta_{\text{gear}} \times \eta_m$ ); saving = (input worm – input helical)  $\times$  hours.

Substitution: worm =  $18 / (0.72 \times 0.90) = 27.78$  kW; helical =  $18 / (0.95 \times 0.90) = 21.05$  kW; difference 6.73 kW;  $\times 6,000$ .

Result: **saving  $\approx$  40,350 kWh a year (about ₹ 3.03 lakh at ₹ 7.50/kWh).**

Why the others are wrong: (a) compares gearbox input powers only and forgets that each kW saved there also saves motor losses; (c) multiplies the output by the efficiency difference (0.23), which is not how losses scale; (d) multiplies by motor efficiency instead of dividing. Worm gears are compact and cheap but lose 25–30 % at high ratios, and their efficiency falls further when lightly loaded or poorly lubricated. That makes them a high-yield target in conveyors and mixers.

Ref: BEE Guidebook Paper 3, Ch. 2 (Electric motors – transmission and drive losses).

**16** Lamps cost ₹ 450 with a rated life of 20,000 h. Spot replacement costs ₹ 300 labour per failed lamp. Group relamping at 14,000 h costs ₹ 60 labour per lamp, and 10 % of lamps fail early and must still be spot-replaced (lamp plus ₹ 300 labour). The group-relamping cost per lamp position per 10,000 burning hours is about:

[5P3CEM016 | Chapter 8: Lighting System | 8.7 Group relamping versus spot replacement – cost per 10,000 burning hours]

- a) ₹ 364
- b) ₹ 293
- c) ₹ 386
- d) ₹ 418**

#### Answer & explanation

Correct answer: (d) ≈ ₹ 418.

Given: lamp ₹ 450; spot labour ₹ 300; group labour ₹ 60; group interval 14,000 h; 10 % early failures.

Formula: cost per cycle = lamp + group labour + early-failure fraction × (lamp + spot labour); per 10,000 h = cycle cost × 10,000 / interval.

Substitution:  $450 + 60 + 0.10 \times 750 = ₹ 585$  per 14,000 h;  $585 / 1.4$ .

Result: **≈ ₹ 418 per 10,000 h, against ₹ 375 for pure spot replacement (750 per 20,000 h).**

Why the others are wrong: (a) ignores early failures; (b) spreads the group cycle over the full 20,000 h rated life instead of the 14,000 h interval; (c) counts only the labour for early failures and forgets the lamps. Here group relamping costs more, so it is justified only if the higher maintained lumen output lets fewer luminaires be installed or if access costs (scaffolding, production stoppage) are high.

Ref: BEE Guidebook Paper 3, Ch. 8 (Lighting – maintenance, lamp replacement and light loss factor).

**17** A 50 mm compressed-air header carrying 10 m<sup>3</sup>/min of free air shows a pressure drop of 0.80 bar between the receiver and the far end. If it is replaced by a 65 mm header of the same length and route, the new pressure drop at the same flow will be about:

[5P3CEM017 | Chapter 3: Compressed Air System | 3.8 Distribution header upsizing – pressure drop varies with d<sup>5</sup>]

- a) 0.22 bar**
- b) 0.47 bar
- c) 0.28 bar
- d) 0.36 bar

#### Answer & explanation

Correct answer: (a) ≈ 0.22 bar.

Given:  $\Delta p_1 = 0.80$  bar with  $d_1 = 50$  mm;  $d_2 = 65$  mm; same flow and length.

Formula: for turbulent flow at constant mass flow,  $\Delta p \propto L Q^2 / d^5$ , so  $\Delta p_2 = \Delta p_1 \times (d_1/d_2)^5$ .

Substitution:  $50/65 = 0.7692$ ;  $0.7692^5 = 0.2693$ ;  $0.80 \times 0.2693$ .

Result: **new drop ≈ 0.22 bar, freeing about 0.58 bar of generation pressure.**

Why the others are wrong: (b) scales with  $d^2$ , the area ratio, which governs velocity but not friction loss; (c) uses  $d^4$ ; (d) uses  $d^3$ . Because pressure drop falls so steeply with diameter, modest header upsizing often lets the compressor set point come down by half a bar or more, which is worth roughly 3–4 % of compressor power, while also improving end-use pressure stability.

Ref: BEE Guidebook Paper 3, Ch. 3 (Compressed air system – distribution system design and pressure drop).

**18** During an audit, the LT CT-operated energy meter of a unit advanced 3,600 units in a month. The CTs actually installed are 200/5 A, but the billing system still applies a multiplying factor of 30 from the earlier 150/5 A CTs. The unit's true consumption for the month is about:

[5P3CEM018 | Chapter 1: Electrical Systems | 1.2 Metering audit – wrong CT multiplying factor]

- a) 1,08,000 kWh
- b) 1,44,000 kWh**
- c) 1,92,000 kWh
- d) 81,000 kWh

**Answer & explanation**

Correct answer: (b)  $\approx 1,44,000$  kWh.

Given: meter advance 3,600; installed CT 200/5 A; billing MF 30 (150/5 A).

Formula: true kWh = meter advance  $\times$  true MF, where MF = CT primary / CT secondary (whole-current meter element rated 5 A).

Substitution: true MF =  $200 / 5 = 40$ ;  $3,600 \times 40$ .

Result: **true consumption  $\approx 1,44,000$  kWh; the bill shows only 1,08,000 kWh, so 36,000 kWh a month is unbilled.**

Why the others are wrong: (a) is the billed figure using the outdated MF of 30; (c) applies the correction twice ( $3,600 \times 40 \times 40/30$ ); (d) applies the ratio the wrong way ( $3,600 \times 30 \times 30/40$ ). An energy auditor must reconcile utility bills with the sub-meter totals; a 25 % gap like this surfaces as an impossible specific energy consumption and must be corrected before any baseline or savings is computed.

Ref: BEE Guidebook Paper 3, Ch. 1 (Electrical system – electricity billing and metering).

**19** A 1.0 MWe natural-gas engine generator, at full load, burns  $245 \text{ Sm}^3/\text{h}$  of gas of GCV  $9,500 \text{ kcal/Sm}^3$  ( $\approx 39.8 \text{ MJ/Sm}^3$ ). Taking NCV as 90 % of GCV, its electrical efficiency on the NCV basis (as engine makers quote it) is about:

[5P3CEM019 | Chapter 9: Diesel / Natural Gas Power Generating System | 9.5 Gas engine generator – efficiency on NCV and GCV basis]

- a) 36.9 %
- b) 45.6 %
- c) 33.3 %
- d) 41.1 %**

**Answer & explanation**

Correct answer: (d)  $\approx 41.1$  %.

Given: output 1,000 kW; gas  $245 \text{ Sm}^3/\text{h}$ ; GCV  $9,500 \text{ kcal/Sm}^3$  ( $\approx 39.8 \text{ MJ/Sm}^3$ ); NCV =  $0.90 \times \text{GCV} = 8,550 \text{ kcal/Sm}^3$  ( $\approx 35.8 \text{ MJ/Sm}^3$ );  $1 \text{ kWh} = 860 \text{ kcal}$  ( $\approx 3,601 \text{ kJ}$ ).

Formula:  $\eta = \text{output} \times 860 / (\text{gas} \times \text{CV})$ .

Substitution: NCV basis:  $1,000 \times 860 / (245 \times 8,550) = 8,60,000 / 20,94,750$ .

Result:  **$\eta \approx 41.1$  % on NCV ( $\approx 36.9$  % on GCV).**

Why the others are wrong: (a) is the GCV-basis efficiency; it is correct for an Indian energy balance but is not the figure makers quote; (b) divides the NCV efficiency by 0.9 again, double-counting the conversion; (c) multiplies the GCV efficiency by 0.9, converting in the wrong direction. Gas is billed on GCV, so cost per kWh must use GCV. Mixing bases in a comparison with a grid or diesel option can flatter the engine by about 10 %, which an auditor must catch.

Ref: BEE Guidebook Paper 3, Ch. 9 (DG and gas engine power generation – performance and efficiency).

**20** A dry-type transformer feeds 6-pulse drives. The measured current spectrum, per unit of the fundamental, is: 1st 1.00, 5th 0.35, 7th 0.20, 11th 0.09, 13th 0.06. Defining  $K = \sum(I_h^2 h^2) / \sum(I_h^2)$ , the K-factor needed for a replacement transformer is about:

[5P3CEM020 | Chapter 1: Electrical Systems | 1.10 Harmonics – K-factor of a transformer feeding drives]

- a) 6.48
- b) 7.61
- c) 5.37
- d) 3.48

**Answer & explanation**

Correct answer: (a)  $\approx 6.48$ .

Given:  $I_h$  (pu) = 1.00, 0.35, 0.20, 0.09, 0.06 for  $h = 1, 5, 7, 11, 13$ .

Formula:  $K = \sum(I_h^2 h^2) / \sum(I_h^2)$ .

Substitution: numerator =  $1.000 + 3.0625 + 1.960 + 0.9801 + 0.6084 = 7.611$ ; denominator =  $1 + 0.1225 + 0.04 + 0.0081 + 0.0036 = 1.1742$ ;  $7.611 / 1.1742$ .

Result:  **$K \approx 6.48$ , so a K-7 rated unit (or a standard unit suitably derated) is required.**

Why the others are wrong: (b) forgets to divide by  $\sum I_h^2$ , i.e. normalises to the fundamental only; (c) divides by  $(1 + \text{THD})$  instead of by  $\sum I_h^2$ ; (d) weights the currents by  $h$  rather than squaring both current and order. Eddy-current loss in windings rises roughly with  $h^2$ , so even small high-order currents heat a standard transformer far more than their rms value suggests.

Ref: BEE Guidebook Paper 3, Ch. 1 (Electrical system – harmonics and their effects).

**21** An LED retrofit cuts the lighting load of a fully air-conditioned office by 20 kW for 3,000 h a year. All lighting heat enters the conditioned space, and the chilled-water plant runs at 1.1 kW/TR overall. The total annual electricity saving, including the cooling plant, is about:

[5P3CEM021 | Chapter 8: Lighting System | 8.6 Lighting load reduction – the air-conditioning bonus]

- a) 60,000 kWh
- b) 1,26,000 kWh
- c) 78,770 kWh
- d) 66,000 kWh

**Answer & explanation**

Correct answer: (c)  $\approx 78,770$  kWh.

Given: lighting reduction 20 kW; 3,000 h; plant 1.1 kW/TR; 1 TR = 3.516 kW of cooling.

Formula: lighting saving =  $\text{kW} \times \text{h}$ ; cooling saving =  $(\text{kW} / 3.516) \times \text{kW/TR} \times \text{h}$ ; total = sum.

Substitution: lighting =  $20 \times 3,000 = 60,000$  kWh; cooling load removed =  $20 / 3.516 = 5.69$  TR; plant saving =  $5.69 \times 1.1 \times 3,000 = 18,770$  kWh.

Result: **total  $\approx 78,770$  kWh a year, about 31 % more than the lighting saving alone.**

Why the others are wrong: (a) ignores the cooling plant; (b) treats each kW of heat as needing 1.1 kW of plant power, i.e. confuses kW of heat with TR; (d) counts only that wrong cooling figure and drops the lighting saving. In return-air plenum ceilings part of the heat goes to the return air rather than the room, but it still reaches the coil, so the bonus remains.

Ref: BEE Guidebook Paper 3, Ch. 8 (Lighting – energy saving opportunities); Ch. 4 (HVAC – cooling load).

**22** A 15 kW conveyor drive runs about 6,000 h a year at only 35–45 % of rated torque and at variable speed. Which choice is likely to give the largest energy reduction over the present IE2 induction motor with mechanical speed variation?

*[5P3CEM022 | Chapter 2: Electrical Motors | 2.7 Motor technology choice for a load that runs mostly at part load]*

- a) IE3 induction motor one size larger, to run cooler at this load
- b) IE2 motor rewound with heavier copper and kept on mechanical speed variation
- c) IE4 permanent-magnet synchronous motor fed from a matched variable speed drive**
- d) IE3 induction motor at the same rating, started through a soft starter

**Answer & explanation**

Correct answer: (c).

At 35–45 % load an induction motor's efficiency and PF sag because its magnetising current and core loss stay almost constant while output falls. A permanent-magnet synchronous motor needs no magnetising current from the stator and has no rotor copper loss, so its efficiency curve stays nearly flat down to low load. Fed from a drive, it also replaces the lossy mechanical variator, saving at both the motor and the transmission.

Why the others are wrong: (a) a larger motor would run at an even lower fraction of rating, so part-load losses get relatively worse; (b) rewinding usually lowers efficiency, and the mechanical variator losses remain; (d) a soft starter only limits starting current, and its thyristors add running loss unless bypassed. Rated IE class compares motors at full load. For a duty like this, the auditor should compare efficiency at the actual load point using the maker's curves, together with the drive train.

Ref: BEE Guidebook Paper 3, Ch. 2 (Electric motors – energy-efficient motors, part-load performance, speed control).

**23** A hospital has 220 exit signs, each with an 11 W CFL on an electromagnetic ballast losing a further 5 W. They burn 8,760 h a year. Replacing them with 2.5 W LED signs saves about:

*[5P3CEM023 | Chapter 8: Lighting System | 8.6 Round-the-clock exit signs – CFL to LED]*

- a) 16,380 kWh
- b) 13,010 kWh
- c) 30,840 kWh
- d) 26,020 kWh**

**Answer & explanation**

Correct answer: (d)  $\approx 26,020$  kWh.

Given: 220 signs; existing  $11 + 5 = 16$  W each; LED 2.5 W; 8,760 h.

Formula: saving =  $n \times (W_{\text{old}} - W_{\text{new}}) \times \text{hours} / 1,000$ .

Substitution:  $220 \times (16 - 2.5) \times 8,760 / 1,000 = 220 \times 13.5 \times 8.76$ .

Result: **saving  $\approx 26,020$  kWh a year, about 84 % of the exit-sign load.**

Why the others are wrong: (a) uses lamp watts only and forgets the ballast loss; (b) assumes 12 h a day, but exit signs must stay lit at all times; (c) is the whole existing consumption, as if the LED signs drew nothing. Loads that run 8,760 h a year (exit signs, staircase and basement lights, security lighting) give very fast payback even when small, and LED signs also cut maintenance because lamps last several years.

Ref: BEE Guidebook Paper 3, Ch. 8 (Lighting – energy-efficient lighting, LED retrofits).

**24** A plant runs two fixed-speed 20 m<sup>3</sup>/min base compressors and one VSD compressor that modulates efficiently only between 6 and 18 m<sup>3</sup>/min. Base machines run fully loaded or are stopped. Over which demand band can the VSD NOT hold pressure in its modulating range, forcing inefficient load/unload or VSD start-stop cycling?

[5P3CEM024 | Chapter 3: Compressed Air System | 3.5 Trim compressor sizing – the VSD control gap]

- a) 38 to 46 m<sup>3</sup>/min
- b) 40 to 46 m<sup>3</sup>/min
- c) 38 to 40 m<sup>3</sup>/min
- d) 26 to 46 m<sup>3</sup>/min

**Answer & explanation**

Correct answer: (a).

Given: base units 20 m<sup>3</sup>/min each; VSD range 6–18 m<sup>3</sup>/min.

Formula: band covered with n bases on = 20n + 6 to 20n + 18; a control gap exists wherever consecutive bands do not overlap.

Substitution: one base: 26–38 m<sup>3</sup>/min; two bases: 46–58 m<sup>3</sup>/min. Demand above 38 needs more than one base plus a fully loaded VSD, but switching on a second base leaves the VSD asked for less than its 6 m<sup>3</sup>/min minimum until demand reaches 46.

Result: **control gap from 38 to 46 m<sup>3</sup>/min.**

Why the others are wrong: (b) starts the gap at 40, where the second base comes on, ignoring that the VSD already tops out at 38; (c) stops at 40, forgetting the VSD minimum of 6; (d) wrongly includes the 26–38 band that one base plus the VSD covers. A trim VSD's range should exceed the largest base step (20 m<sup>3</sup>/min).

Ref: BEE Guidebook Paper 3, Ch. 3 (Compressed air system – capacity control and multiple compressor control).

**25** An auditor must measure average illuminance in a hall 24 m × 16 m, with luminaires 3.0 m above the working plane. Using the room-index grid rule — number of points = (x + 2)<sup>2</sup>, where x is the room index rounded up to the next whole number, with x = 4 for all room indices of 3 and above — the minimum number of points is:

[5P3CEM025 | Chapter 8: Lighting System | 8.5 Illuminance survey – number of measuring points from room index]

- a) 36
- b) 25
- c) 49
- d) 16

**Answer & explanation**

Correct answer: (a) 36 points.

Given: L = 24 m; W = 16 m; Hm = 3.0 m.

Formula: room index RI = L × W / [Hm × (L + W)]; points = (x + 2)<sup>2</sup>, x = RI rounded up, capped at 4 for RI ≥ 3.

Substitution: RI = 384 / (3.0 × 40) = 3.2; RI ≥ 3, so x = 4; (4 + 2)<sup>2</sup>.

Result: **minimum 36 measuring points, laid out on an even 6 × 6 grid.**

Why the others are wrong: (b) truncates 3.2 to x = 3, which gives too few points; (c) rounds up to 5, beyond the cap of the rule; (d) uses x = 2, as if the room index were below 2. Too few points can miss dark zones between luminaire rows or near walls and overstate uniformity. The measured average should then be compared with the design maintained illuminance before deciding whether de-lamping or dimming is possible.

Ref: BEE Guidebook Paper 3, Ch. 8 (Lighting – illuminance measurement and lighting performance assessment).

**26** A manufacturer's 20 kW test fan is scaled up to a geometrically similar fan with an impeller 1.25 times larger in diameter, running at the same speed and handling air of the same density. The shaft power of the larger fan at the corresponding point is about:

[5P3CEM026 | Chapter 5: Fans and Blowers | 5.3 Fan laws for size – geometrically similar larger fan]

- a) 39.1 kW
- b) 61.0 kW**
- c) 31.3 kW
- d) 48.8 kW

**Answer & explanation**

Correct answer: (b)  $\approx$  61.0 kW.

Given:  $P_1 = 20$  kW;  $D_2/D_1 = 1.25$ ; same speed and density.

Formula: for geometrically similar fans,  $Q \propto N D^3$ ,  $p \propto \rho N^2 D^2$ , so  $P \propto \rho N^3 D^5$ .

Substitution:  $1.25^5 = 3.052$ ;  $20 \times 3.052$ .

Result: **shaft power  $\approx$  61.0 kW (flow up 1.95 times, pressure up 1.56 times).**

Why the others are wrong: (a) scales power with  $D^3$ , which is the flow law; (c) uses  $D^2$ , which is the pressure law; (d) uses  $D^4$ . The size laws let a designer predict a larger fan from a model test, and let an auditor check whether a replacement impeller of a different diameter on the same shaft will overload the motor. Small diameter changes have large power effects because of the fifth power.

Ref: BEE Guidebook Paper 3, Ch. 5 (Fans and blowers – fan laws).

**27** A glass supplier quotes only a shading coefficient (SC) of 0.32 for a façade glass. The project must show  $SHGC \leq 0.25$  for this orientation. Taking  $SHGC = 0.87 \times SC$ , the glass's SHGC and its compliance status are:

[5P3CEM027 | Chapter 10: Energy Conservation in Buildings and ECBC | 10.3 Converting a quoted shading coefficient to SHGC for ECBC compliance]

- a) 0.278; fails the 0.25 limit**
- b) 0.368; fails the 0.25 limit
- c) 0.320; fails the 0.25 limit
- d) 0.209; meets the 0.25 limit

**Answer & explanation**

Correct answer: (a).

Given:  $SC = 0.32$ ;  $SHGC$  limit 0.25.

Formula: SC is the ratio of a glass's solar gain to that of 3 mm clear glass, whose SHGC is about 0.87; so  $SHGC = 0.87 \times SC$ .

Substitution:  $0.87 \times 0.32 = 0.278$ .

Result:  **$SHGC \approx 0.278$ , above 0.25, so the glass does not comply as specified.**

Why the others are wrong: (b) divides by 0.87 instead of multiplying; (c) treats SC and SHGC as identical, which overstates the gain but happens to reach the same compliance verdict for the wrong reason; (d) also multiplies by a 0.75 frame-area factor, but the SHGC limit applies to the glazing itself, so frames cannot be used to meet it. The designer can switch to a glass of  $SHGC \leq 0.25$ , or add external shading and claim the equivalent-SHGC credit allowed by the code's projection-factor method.

Ref: BEE Guidebook Paper 3, Ch. 10 (Energy conservation in buildings and ECBC – fenestration requirements).

**28** On a high-head water pump delivering 180 m of head, precision probes show the water warming by 0.11 K from suction to discharge. Taking  $g = 9.81 \text{ m/s}^2$  and  $c_p = 4.186 \text{ kJ/kg}\cdot\text{K}$ , and neglecting heat exchange with the surroundings, the pump efficiency is about (the motor efficiency is 88.5 %):

[5P3CEM028 | Chapter 6: Pumps and Pumping System | 6.4 Pump efficiency by the thermodynamic (temperature-rise) method]

- a) 73.9 %
- b) 70.2 %
- c) 88.5 %
- d) 79.3 %**

#### Answer & explanation

Correct answer: (d)  $\approx 79.3 \%$ .

Given:  $H = 180 \text{ m}$ ;  $\Delta T = 0.11 \text{ K}$ ;  $c_p = 4,186 \text{ J/kg}\cdot\text{K}$ .

Formula: useful energy per kg =  $gH$ ; loss per kg =  $c_p \Delta T$ ;  $\eta_{\text{pump}} = gH / (gH + c_p \Delta T)$ .

Substitution:  $gH = 9.81 \times 180 = 1,765.8 \text{ J/kg}$ ;  $c_p \Delta T = 4,186 \times 0.11 = 460.5 \text{ J/kg}$ ;  $1,765.8 / 2,226.3$ .

Result: **pump efficiency  $\approx 79.3 \%$** .

Why the others are wrong: (a) computes  $1 - \text{loss} / \text{useful energy}$ , dividing the loss by the wrong quantity; (b) multiplies by the motor efficiency, which gives a wire-to-water figure, not pump efficiency; (c) simply quotes the motor efficiency. The thermodynamic method needs no flow meter, which is why it suits large, high-head pumps where flow is hard to measure. It needs thermometers resolving about 0.001 K, so it is not used on low-head pumps, where the temperature rise is tiny.

Ref: BEE Guidebook Paper 3, Ch. 6 (Pumps – performance assessment and efficiency).

**29** A 1,600 kVA, 11/0.433 kV transformer has 1.0 % resistance and 5.5 % reactance. It carries 80 % of rated kVA at 0.85 PF lagging. Using the approximate formula, its percentage voltage regulation is about:

[5P3CEM029 | Chapter 1: Electrical Systems | 1.5 Transformer voltage regulation at part load and lagging PF]

- a) 3.0 %**
- b) 3.7 %
- c) 4.2 %
- d) 4.5 %

#### Answer & explanation

Correct answer: (a)  $\approx 3.0 \%$ .

Given:  $\%R = 1.0$ ;  $\%X = 5.5$ ; load fraction  $x = 0.80$ ;  $\cos \phi = 0.85$ , so  $\sin \phi = 0.527$ .

Formula:  $\% \text{ regulation} \approx x \times (\%R \cos \phi + \%X \sin \phi)$ .

Substitution:  $\%R \cos \phi = 1.0 \times 0.85 = 0.850$ ;  $\%X \sin \phi = 5.5 \times 0.527 = 2.897$ ; sum = 3.747;  $\times 0.80$ .

Result: **regulation  $\approx 3.0 \%$** .

Why the others are wrong: (b) is the full-load value; the load fraction of 0.80 was left out; (c) swaps the multipliers, putting  $\sin \phi$  against  $\%R$  and  $\cos \phi$  against  $\%X$ ; (d) uses the whole impedance ( $5.59 \% \times 0.80$ ) as if the load were purely reactive, which overstates the drop at 0.85 PF. Because reactance dominates, raising the PF with capacitors on the LT side cuts the  $\sin \phi$  term and lifts the voltage, which is one more reason to compensate close to the transformer.

Ref: BEE Guidebook Paper 3, Ch. 1 (Electrical system – transformers, voltage and power factor).

**30** A seafood plant freezes 20 t of fish a day from 25 °C to –18 °C in a blast freezer that runs 20 h a day. The fish freezes at –2 °C. Specific heat above freezing 3.7 kJ/kg·K, below freezing 1.9 kJ/kg·K, latent heat 250 kJ/kg. The average product refrigeration load is about:

[5P3CEM030 | Chapter 4: HVAC and Refrigeration System | 4.10 Blast freezer – product freezing load with latent heat]

- a) 10.3 TR
- b) 25.0 TR
- c) 30.0 TR**
- d) 32.3 TR

**Answer & explanation**

Correct answer: (c)  $\approx$  30.0 TR.

Given: 20,000 kg/day; 25  $\rightarrow$  –2 °C at 3.7 kJ/kg·K; latent 250 kJ/kg; –2  $\rightarrow$  –18 °C at 1.9 kJ/kg·K; 20 h/day.

Formula:  $q = c_{p1}\Delta T_1 + L + c_{p2}\Delta T_2$  per kg; load =  $m \times q$  / (running seconds); TR = kW / 3.516.

Substitution:  $q = 3.7 \times 27 + 250 + 1.9 \times 16 = 99.9 + 250 + 30.4 = 380.3$  kJ/kg;  $20,000 \times 380.3 / 72,000 \text{ s} = 105.6$  kW.

Result:  $\approx$  **105.6 kW  $\approx$  30.0 TR of product load, before infiltration, fan and door loads.**

Why the others are wrong: (a) leaves out the latent heat, which is two-thirds of the total; (b) spreads the load over 24 h although the freezer runs only 20 h; (d) applies the unfrozen specific heat all the way to –18 °C. Sizing on the wrong basis either under-freezes product or leaves compressors short-cycling at part load.

Ref: BEE Guidebook Paper 3, Ch. 4 (Refrigeration – cold storage and product loads).

**31** An AHU needs 5.5 kW of air power. The existing belt-driven forward-curved fan has a fan efficiency of 62 %, belt efficiency of 93 % and motor efficiency of 89 %. An EC plug-fan array has a quoted system efficiency (fan, motor and electronics together) of 66 %. Over 8,000 h a year, the energy saved is about:

[5P3CEM031 | Chapter 5: Fans and Blowers | 5.2 Belt-driven AHU fan replaced by an EC plug-fan array]

- a) 13,070 kWh
- b) 19,070 kWh**
- c) 20,890 kWh
- d) 10,830 kWh

**Answer & explanation**

Correct answer: (b)  $\approx$  19,070 kWh.

Given: air power 5.5 kW; existing  $\eta = 0.62 \times 0.93 \times 0.89$ ; EC array system  $\eta = 0.66$ ; 8,000 h.

Formula: input = air power / system efficiency; saving = (input old – input new)  $\times$  hours.

Substitution: old system  $\eta = 0.513$ , input = 10.72 kW; new input =  $5.5 / 0.66 = 8.33$  kW; difference 2.38 kW  $\times$  8,000.

Result: **saving  $\approx$  19,070 kWh a year.**

Why the others are wrong: (a) forgets the belt loss in the old drive; (c) uses 8,760 h instead of the stated 8,000 h; (d) treats the 66 % as fan-only efficiency and divides again by a motor efficiency, although the figure already covers the motor and electronics. Comparing wire-to-air efficiencies on the same basis is essential, and EC arrays also give built-in speed control and redundancy, so further savings come when flow is turned down.

Ref: BEE Guidebook Paper 3, Ch. 5 (Fans and blowers – fan types, efficiency and drive losses).

**32** A frozen-food store has 6 unit coolers, each with 12 kW of electric defrost heaters run 4 times a day for 30 minutes. About 40 % of the defrost heat ends up in the room and must be removed again by the refrigeration plant, whose COP is 1.6. The annual energy attributable to defrost is about:

- a) 52,560 kWh
- b) 85,410 kWh
- c) 65,700 kWh**
- d) 1,64,250 kWh

**Answer & explanation**

Correct answer: (c)  $\approx$  65,700 kWh.

Given:  $6 \times 12$  kW heaters; 4 defrosts  $\times$  0.5 h per day; 40 % of heat re-enters the room; COP 1.6; 365 days.

Formula: heater energy  $E = \text{kW} \times \text{h}$ ; refrigeration penalty  $= 0.40 \times E / \text{COP}$ ; total  $= E + \text{penalty}$ .

Substitution:  $E = 72 \times 2 = 144$  kWh/day; penalty  $= 0.40 \times 144 / 1.6 = 36$  kWh/day; total 180 kWh/day  $\times$  365.

Result: **defrost costs  $\approx$  65,700 kWh a year.**

Why the others are wrong: (a) counts only the heater energy; (b) assumes all the defrost heat must be removed again, ignoring the stated 40 % share; (d) runs each defrost for an hour instead of 30 minutes. Defrost on demand (by coil air-side pressure drop or frost sensing), shorter terminated cycles or hot-gas defrost can cut this by half, and a lower moisture load from door curtains and dock seals reduces how often coils need it.

Ref: BEE Guidebook Paper 3, Ch. 4 (Refrigeration – energy saving opportunities in cold storage).

**33** An LT feeder normally carries 100 A in each phase. Because single-phase loads are badly distributed, it now carries 120 A, 100 A and 80 A (balanced PF,  $120^\circ$  apart). The neutral conductor has the same resistance as a phase conductor. Compared with balanced loading, the total feeder  $I^2R$  loss rises by about:

- a) 2.7 %
- b) 6.7 %**
- c) 4.0 %
- d) 8.0 %

**Answer & explanation**

Correct answer: (b)  $\approx$  6.7 %.

Given: balanced  $3 \times 100$  A; unbalanced 120, 100, 80 A;  $R_{\text{neutral}} = R_{\text{phase}}$ .

Formula: phase loss  $\propto \Sigma I^2$ ; neutral current  $I_n = \sqrt{(I_a^2 + I_b^2 + I_c^2 - I_a I_b - I_b I_c - I_c I_a)}$ ; neutral loss  $\propto I_n^2$ .

Substitution: balanced  $\Sigma I^2 = 30,000$ ; unbalanced phases  $= 14,400 + 10,000 + 6,400 = 30,800$ ;  $I_n^2 = 30,800 - (12,000 + 8,000 + 9,600) = 1,200$ , so  $I_n \approx 34.6$  A; total  $= 32,000$ .

Result: **loss rises by  $2,000/30,000 \approx 6.7$  % for the same total current.**

Why the others are wrong: (a) counts only the extra phase loss and forgets the neutral; (c) counts only the neutral loss; (d) takes the neutral current as the arithmetic difference ( $120 - 80 = 40$  A) instead of the phasor sum. Imbalance also causes voltage unbalance at motors, which raises their losses far more than this.

Rebalancing single-phase loads is a zero-cost first step in any LT distribution audit.

Ref: BEE Guidebook Paper 3, Ch. 1 (Electrical system – distribution losses, load balancing).

**34** A large ID fan and motor rotor have a combined moment of inertia of  $450 \text{ kg}\cdot\text{m}^2$  and run at 990 rpm. For a direct-on-line start on a constant-voltage supply, the energy dissipated in the rotor during acceleration roughly equals the kinetic energy finally stored. That rotor heat per start is about:

- a) 4.84 MJ
- b) 2.47 MJ
- c) 1.21 MJ
- d) 2.42 MJ**

#### Answer & explanation

Correct answer: (d)  $\approx$  2.42 MJ.

Given:  $J = 450 \text{ kg}\cdot\text{m}^2$ ;  $N = 990 \text{ rpm}$ .

Formula:  $\omega = 2\pi N/60$ ; stored kinetic energy  $E = \frac{1}{2} J \omega^2$ ; rotor loss during unloaded DOL acceleration  $\approx E$ .

Substitution:  $\omega = 2\pi \times 990 / 60 = 103.7 \text{ rad/s}$ ;  $E = 0.5 \times 450 \times 103.7^2 = 2.418 \times 10^6 \text{ J}$ .

Result: **rotor heat  $\approx$  2.42 MJ per start ( $\approx$  0.67 kWh); the supply delivers about twice this in total.**

Why the others are wrong: (a) drops the  $\frac{1}{2}$ , or adds stored and dissipated energy, giving the total drawn rather than the rotor heat; (b) uses synchronous speed (1,000 rpm) instead of the running speed; (c) takes the rotor heat as half the stored energy. Each start dumps megajoules into the rotor in seconds, which is why makers limit starts per hour for high-inertia drives. A VFD start avoids this heat because slip stays small throughout.

Ref: BEE Guidebook Paper 3, Ch. 2 (Electric motors – starting, inertia and duty).

**35** Label tables give total losses for two 250 kVA distribution transformers: 3-star, 980 W at 50 % load and 2,930 W at 100 % load; 5-star, 770 W and 2,210 W. Both run 8,760 h a year at an rms load of 60 %. The annual energy saved by the 5-star unit is about:

*[5P3CEM035 | Chapter 1: Electrical Systems | 1.5 BEE star-labelled distribution transformers – annual loss saving]*

- a) 1,840 kWh
- b) 3,920 kWh
- c) 6,310 kWh
- d) 2,495 kWh**

#### Answer & explanation

Correct answer: (d)  $\approx$  2,495 kWh.

Given:  $P_{50} = P_o + 0.25 P_c$  and  $P_{100} = P_o + P_c$  for each unit; rms load 0.60; 8,760 h.

Formula:  $P_c = (P_{100} - P_{50}) / 0.75$ ;  $P_o = P_{100} - P_c$ ; loss at 60 % =  $P_o + 0.36 P_c$ .

Substitution: 3-star  $P_c = 1,950 / 0.75 = 2,600 \text{ W}$ ,  $P_o = 330 \text{ W}$ , loss =  $330 + 936 = 1,266 \text{ W}$ ; 5-star  $P_c = 1,440 / 0.75 = 1,920 \text{ W}$ ,  $P_o = 290 \text{ W}$ , loss =  $290 + 691 = 981 \text{ W}$ ; difference  $284.8 \text{ W} \times 8,760$ .

Result: **saving  $\approx$  2,495 kWh a year per transformer.**

Why the others are wrong: (a) takes the 50 % loss difference (210 W) as if load were fixed at 50 %; (b) scales copper loss linearly with load (0.6) instead of with its square; (c) uses the 100 % loss difference all year. Splitting the label points into no-load and load loss allows a fair comparison.

Ref: BEE Guidebook Paper 3, Ch. 1 (Electrical system – transformer losses; BEE Standards & Labelling).

**36** Split AC outdoor units are packed into a narrow recess, and their condenser inlet air is 6 K hotter than ambient. The maker's data show capacity falling 1.0 % and input power rising 1.5 % per K rise in condenser inlet temperature. The specific power (kW/TR) is worse than in free air by about:

*[5P3CEM036 | Chapter 4: HVAC and Refrigeration System | 4.9 Split AC outdoor units – hot-air recirculation at the condenser]*

- a) 15.0 %
- b) 9.0 %
- c) 16.0 %**

d) 6.4 %

#### Answer & explanation

Correct answer: (c)  $\approx 16.0$  %.

Given: condenser inlet + 6 K; capacity  $-1.0$  %/K; power  $+1.5$  %/K.

Formula: new kW/TR ratio =  $(1 + 0.015 \times \Delta T) / (1 - 0.010 \times \Delta T)$ .

Substitution: capacity factor =  $1 - 0.06 = 0.94$ ; power factor =  $1 + 0.09 = 1.09$ ;  $1.09 / 0.94 = 1.160$ .

Result: **specific power  $\approx 16.0$  % worse.**

Why the others are wrong: (a) adds the two percentages ( $9 + 6$ ), but capacity divides and so compounds; (b) counts only the rise in power and forgets the lost capacity; (d) counts only the capacity loss ( $1/0.94$ ). Condensers that breathe their own exhaust also trip on high pressure in summer, just when cooling is needed. Relocating units, adding discharge ducts or baffles, or spacing them to recommended clearances is a near-zero-cost fix the auditor should check on every rooftop and balcony.

Ref: BEE Guidebook Paper 3, Ch. 4 (HVAC and refrigeration – factors affecting performance, condensing temperature).

**37** An ultrasonic survey of a 7 bar(g) network finds three leaks equivalent to 3 mm holes, five of 2 mm and one of 5 mm. At this pressure a 2 mm hole leaks about  $0.20 \text{ m}^3/\text{min}$  of free air. With a specific power of  $6.5 \text{ kW per m}^3/\text{min}$ , the compressor power wasted is about:

[5P3CEM037 | Chapter 3: Compressed Air System | 3.5 Leak survey – scaling leak flow with hole diameter]

a) 15.6 kW

**b) 23.4 kW**

c) 40.0 kW

d) 11.7 kW

#### Answer & explanation

Correct answer: (b)  $\approx 23.4 \text{ kW}$ .

Given:  $0.20 \text{ m}^3/\text{min}$  per 2 mm hole; holes  $3 \times 3 \text{ mm}$ ,  $5 \times 2 \text{ mm}$ ,  $1 \times 5 \text{ mm}$ ;  $6.5 \text{ kW per m}^3/\text{min}$ .

Formula: at fixed upstream pressure, leak flow  $\propto$  orifice area  $\propto d^2$ ; flow of a hole =  $0.20 \times (d/2)^2$ ; power = total flow  $\times$  specific power.

Substitution: 3 mm  $\rightarrow 0.20 \times 2.25 = 0.45 \text{ m}^3/\text{min}$  each,  $\times 3 = 1.35$ ; 2 mm  $\rightarrow 5 \times 0.20 = 1.00$ ; 5 mm  $\rightarrow 0.20 \times 6.25 = 1.25$ ; total  $3.60 \text{ m}^3/\text{min}$ ;  $\times 6.5$ .

Result: **leak load  $\approx 3.6 \text{ m}^3/\text{min}$ , wasting  $\approx 23.4 \text{ kW}$ .**

Why the others are wrong: (a) scales flow with diameter rather than area; (c) scales with  $d^3$ ; (d) treats every hole as a 2 mm hole. The single 5 mm leak loses as much as six 2 mm leaks, so a leak-tagging programme should rank and fix leaks by estimated flow, not by count.

Ref: BEE Guidebook Paper 3, Ch. 3 (Compressed air system – leakage and its quantification).

**38** A 1,500 kVA DG set's engine-driven radiator fan absorbs a constant 45 kW of shaft power. An electric radiator fan on a VFD controlled by coolant temperature would average 12 kW of electrical power, supplied by the set itself. Alternator efficiency is 95 %, and the set gives 3.6 kWh per litre. Over 3,000 h a year the diesel saved is about:

[5P3CEM038 | Chapter 9: Diesel / Natural Gas Power Generating System | 9.5 DG radiator – engine-driven fan replaced by a VFD electric fan]

**a) 25,630 litres**

b) 27,500 litres

c) 37,500 litres

d) 24,340 litres

#### Answer & explanation

Correct answer: (a)  $\approx$  25,630 litres.

Given: engine fan 45 kW shaft; electric fan 12 kWe;  $\eta_{alt}$  0.95; 3.6 kWh/litre; 3,000 h.

Formula: shaft power to make 12 kWe =  $12 / 0.95$ ; shaft power freed =  $45 - 12/0.95$ ; extra electrical output = freed shaft  $\times 0.95$ ; fuel saved = extra kWe  $\times h /$  (kWh per litre).

Substitution:  $12 / 0.95 = 12.63$  kW; freed =  $32.37$  kW;  $\times 0.95 = 30.75$  kWe;  $30.75 \times 3,000 / 3.6$ .

Result:  $\approx$  **25,630 litres of diesel a year.**

Why the others are wrong: (b) subtracts the electric fan power directly from the shaft power, ignoring alternator loss; (c) forgets the electric fan's own consumption; (d) applies the alternator efficiency twice. A fixed engine-driven fan runs at full air flow even in cool weather or at light load, so temperature-controlled fans pay back quickly on prime-power DG sets.

Ref: BEE Guidebook Paper 3, Ch. 9 (DG sets – energy saving measures, auxiliaries).

**39** A drive bus carries a fundamental current of 400 A with a current THD of 38 %. An active harmonic filter must cancel the harmonic current, and the designer adds a 25 % margin. The minimum filter rating is about:

[5P3CEM039 | Chapter 1: Electrical Systems | 1.10 Harmonics – sizing an active harmonic filter]

a) 152 A

b) 203 A

c) 72 A

**d) 190 A**

#### Answer & explanation

Correct answer: (d)  $\approx$  190 A.

Given:  $I_1 = 400$  A;  $THD_1 = 0.38$ ; margin 25 %.

Formula: harmonic current  $I_h = I_1 \times THD_1$  (THD is referred to the fundamental); rating =  $I_h \times 1.25$ .

Substitution:  $I_h = 400 \times 0.38 = 152$  A;  $152 \times 1.25$ .

Result: **active filter rating  $\approx$  190 A (choose the next standard size above).**

Why the others are wrong: (a) omits the specified margin; (b) first computes the total rms current,  $400 \times \sqrt{1 + 0.38^2} = 428$  A, and multiplies that by THD, but THD is defined against the fundamental; (c) squares THD, which applies to power-type quantities, not to current. An active filter injects equal and opposite harmonic currents, so it is sized on harmonic amps, not on load kVA. It also stops drive harmonics from overloading detuned capacitor banks.

Ref: BEE Guidebook Paper 3, Ch. 1 (Electrical system – harmonics, mitigation by filters).

**40** A cooling tower evaporates 50 m<sup>3</sup>/h and is held at 3 cycles of concentration by an automatic blowdown valve. Drift is 6.0 m<sup>3</sup>/h with old eliminators and the valve bleeds 19.0 m<sup>3</sup>/h. New eliminators cut drift to 0.15 m<sup>3</sup>/h. If the treatment programme keeps COC at 3, the makeup and blowdown will become:

[5P3CEM040 | Chapter 7: Cooling Towers | 7.4 Drift eliminator upgrade under cycles-of-concentration control]

a) Makeup falls to 69.15 m<sup>3</sup>/h; blowdown stays at 19.00 m<sup>3</sup>/h

b) Makeup falls to 66.67 m<sup>3</sup>/h; blowdown is set to 16.52 m<sup>3</sup>/h

**c) Makeup unchanged at 75.0 m<sup>3</sup>/h; blowdown opens to 24.85 m<sup>3</sup>/h**

d) Makeup rises to 80.85 m<sup>3</sup>/h; blowdown opens to 30.70 m<sup>3</sup>/h

### Answer & explanation

Correct answer: (c).

Given:  $E = 50 \text{ m}^3/\text{h}$ ;  $\text{COC} = 3$ ; drift  $D$  from  $6.0$  to  $0.15 \text{ m}^3/\text{h}$ ; blowdown  $B = 19.0 \text{ m}^3/\text{h}$ .

Formula: total bleed needed  $(B + D) = E / (\text{COC} - 1)$ ; makeup  $= E + B + D$ .

Substitution:  $B + D = 50 / 2 = 25 \text{ m}^3/\text{h}$ , fixed by the COC set point; new  $B = 25 - 0.15 = 24.85 \text{ m}^3/\text{h}$ ; makeup  $= 50 + 25 = 75 \text{ m}^3/\text{h}$ .

Result: **makeup stays at  $75 \text{ m}^3/\text{h}$ ; the controller simply opens blowdown to  $24.85 \text{ m}^3/\text{h}$ .**

Why the others are wrong: (a) assumes the drift saved comes off makeup, but the controller replaces it with blowdown to hold COC; (b) uses  $E/\text{COC}$  instead of  $E/(\text{COC} - 1)$ ; (d) adds the drift reduction to blowdown twice.

Drift reduction still matters: it stops chemical-laden mist spreading and recovers treatment chemicals in the basin. Water is saved only if the treatment allows a higher COC afterwards.

Ref: BEE Guidebook Paper 3, Ch. 7 (Cooling towers – water balance, drift and blowdown).

**41** A 55 kW induction motor driving a constant-torque load draws 96 A at 415 V. The plant voltage falls to 380 V. Assuming shaft power, efficiency and power factor stay roughly constant, the stator copper loss changes by about:

[5P3CEM041 | Chapter 2: Electrical Motors | 2.3 Low supply voltage on a fully loaded motor – copper loss rise]

- a) 9.2 % rise
- b) 30.3 % rise
- c) 19.3 % rise**
- d) 16.2 % fall

### Answer & explanation

Correct answer: (c)  $\approx 19.3 \%$  rise.

Given:  $I_1 = 96 \text{ A}$  at  $V_1 = 415 \text{ V}$ ;  $V_2 = 380 \text{ V}$ ; constant output, PF and efficiency.

Formula: for constant power,  $I \propto 1/V$ ; copper loss  $\propto I^2$ , so loss ratio  $= (V_1/V_2)^2$ .

Substitution:  $I_2 = 96 \times 415/380 = 104.8 \text{ A}$ ;  $(415/380)^2 = 1.0921^2 = 1.1927$ .

Result: **stator copper loss rises by about 19.3 %.**

Why the others are wrong: (a) is the rise in current, not in  $I^2R$  loss; (b) cubes the voltage ratio; (d) assumes loss falls with  $V^2$ , which is true of a constant-impedance load, not a motor delivering fixed power. The extra heat raises winding temperature, and roughly every  $10^\circ\text{C}$  above rating halves insulation life. Rotor copper loss also rises because slip increases at lower voltage, so a sustained 8 % low voltage on loaded motors costs efficiency and life.

Ref: BEE Guidebook Paper 3, Ch. 2 (Electric motors – effect of voltage variation on performance).

**42** Two identical 625 kVA, 415 V DG sets run in parallel, sharing an 800 kW, 0.80 PF load equally in kW. Because of an AVR setting mismatch, set A runs at 0.95 PF and set B carries the remaining kVAr. Compared with equal sharing, the current of set B is higher by about:

[5P3CEM042 | Chapter 9: Diesel / Natural Gas Power Generating System | 9.3 Parallel DG sets – unequal kVAr sharing from excitation mismatch]

- a) 23.2 %**
- b) 56.2 %
- c) 46.3 %
- d) 11.6 %

### Answer & explanation

Correct answer: (a)  $\approx 23.2\%$ .

Given: total 800 kW at 0.80 PF, so 600 kVAr; each set 400 kW; set A PF 0.95.

Formula:  $kVar_A = kW \times \tan(\cos^{-1} 0.95)$ ;  $kVar_B = 600 - kVar_A$ ;  $kVA_B = \sqrt{(400^2 + kVar_B^2)}$ ; with equal sharing each set carries 500 kVA.

Substitution:  $kVar_A = 400 \times 0.3287 = 131.5$ ;  $kVar_B = 468.5$ ;  $kVA_B = \sqrt{(1,60,000 + 2,19,500)} = 616.0$ ;  $616.0 / 500$ .

Result: **set B's current is about 23.2 % above its fair share; it runs at 0.65 PF and 98.6 % of rating.**

Why the others are wrong: (b) compares kVAr shares (468.5 vs 300), not current; (c) compares set B's kVA with set A's instead of with equal sharing; (d) halves the correct answer. Circulating reactive current heats alternator windings and leaves no margin for motor starting. Matching AVR droop or using reactive load-sharing control fixes it at no cost.

Ref: BEE Guidebook Paper 3, Ch. 9 (DG sets – parallel operation and load sharing).

**43** A warehouse has 2,000 LED lamps of 18 W each with low-cost drivers of power factor 0.55. They are replaced by lamps of the same wattage with drivers of power factor 0.95. Treating PF as displacement only, the reduction in apparent power drawn by the lighting is about:

[5P3CEM043 | Chapter 8: Lighting System | 8.3 LED driver power factor – kVA drawn by a large lighting load]

a) 14.4 kVA

**b) 27.6 kVA**

c) 29.5 kVA

d) 42.8 kVA

#### Answer & explanation

Correct answer: (b)  $\approx 27.6$  kVA.

Given:  $P = 2,000 \times 18 = 36$  kW;  $PF_1 = 0.55$ ;  $PF_2 = 0.95$ .

Formula:  $kVA = kW / PF$ ; reduction =  $kW/PF_1 - kW/PF_2$ .

Substitution:  $36 / 0.55 = 65.45$  kVA;  $36 / 0.95 = 37.89$  kVA; difference.

Result: **apparent power falls by about 27.6 kVA, a 42 % cut in lighting current.**

Why the others are wrong: (a) multiplies kW by the PF difference, which has no physical meaning; (c) subtracts kW from the original kVA; (d) is the reduction in kVAr,  $36 \times (\tan \phi_1 - \tan \phi_2)$ , not in kVA. Cheap drivers draw distorted current, so real true PF is often worse than displacement PF and adds neutral current from triplen harmonics. Specifying drivers with  $PF \geq 0.9$  and low THD protects cables, transformers and the kVAh bill.

Ref: BEE Guidebook Paper 3, Ch. 8 (Lighting – LED lighting and control gear); Ch. 1 (Electrical system – power factor).

**44** A 2,000 kVA plant transformer has a no-load current of 1.5 % and a leakage reactance of 6 %. On Sundays the plant load is only 10 % of rated kVA, and the utility meter is on the HT side. The fixed capacitor needed on the LT side to supply the transformer's own reactive demand at that load is about:

[5P3CEM044 | Chapter 1: Electrical Systems | 1.4 Fixed compensation for transformer magnetising and leakage kVAr at light load]

a) 30.0 kVAr

b) 42.0 kVAr

**c) 31.2 kVAr**

d) 12.0 kVAr

#### Answer & explanation

Correct answer: (c)  $\approx 31.2$  kVAr.

Given:  $S = 2,000$  kVA;  $I_o = 1.5\%$ ;  $X_l = 6\%$ ; load fraction  $x = 0.10$ .

Formula: transformer kVAr  $\approx S \times (I_o + X_l \times x^2)$ ; magnetising kVAr is constant, leakage kVAr rises with the square of load.

Substitution: magnetising  $= 2,000 \times 0.015 = 30.0$  kVAr; leakage  $= 2,000 \times 0.06 \times 0.10^2 = 1.2$  kVAr; total 31.2 kVAr.

Result: **fixed compensation  $\approx 31.2$  kVAr.**

Why the others are wrong: (a) ignores the leakage-reactance term, small but real; (b) uses  $x$  instead of  $x^2$ , overstating leakage kVAr tenfold; (d) keeps only the leakage term at  $x$  rather than  $x^2$  and drops magnetising kVAr entirely. HT metering sees the transformer's kVAr even when the LT APFC has switched every step off, so a small fixed bank sized this way prevents a poor monthly PF from light-load hours. Oversizing it would push the PF leading.

Ref: BEE Guidebook Paper 3, Ch. 1 (Electrical system – power factor correction, location of capacitors).

**45** A pump delivers  $180 \text{ m}^3/\text{h}$ . The suction gauge reads a vacuum of  $0.25 \text{ kg/cm}^2$  ( $\approx 0.245$  bar) on a 200 mm pipe; the discharge gauge reads  $3.6 \text{ kg/cm}^2(\text{g})$  ( $\approx 3.53 \text{ bar(g)}$ ) on a 150 mm pipe and is mounted 1.2 m above the suction gauge. Taking  $1 \text{ kg/cm}^2 = 10 \text{ m}$  of water, the total head developed is about:

[5P3CEM045 | Chapter 6: Pumps and Pumping System | 6.4 Total head from gauges – vacuum, gauge elevation and velocity head]

a) 35.0 m

b) 38.5 m

**c) 40.0 m**

d) 39.7 m

#### Answer & explanation

Correct answer: (c)  $\approx 40.0$  m.

Given:  $Q = 0.05 \text{ m}^3/\text{s}$ ;  $p_s = -0.25 \text{ kg/cm}^2$  ( $\approx -0.245$  bar);  $p_d = 3.6 \text{ kg/cm}^2(\text{g})$  ( $\approx 3.53 \text{ bar(g)}$ );  $z_d - z_s = 1.2 \text{ m}$ ;  $D_s = 0.200 \text{ m}$ ;  $D_d = 0.150 \text{ m}$ .

Formula:  $H = (p_d - p_s) \text{ head} + (z_d - z_s) + (v_d^2 - v_s^2)/2g$ , with  $v = Q/A$ .

Substitution: pressure head  $= (3.6 + 0.25) \times 10 = 38.5 \text{ m}$ ;  $v_s = 1.59 \text{ m/s}$ ,  $v_d = 2.83 \text{ m/s}$ ; velocity head difference  $= (8.00 - 2.53)/19.62 = 0.28 \text{ m}$ ;  $H = 38.5 + 1.2 + 0.28$ .

Result: **total head  $\approx 40.0$  m.**

Why the others are wrong: (a) subtracts the suction vacuum instead of adding it; (b) counts only the gauge pressure difference and leaves out gauge elevation and velocity heads; (d) adds the elevation but omits the velocity-head correction. Small slips like these change the computed efficiency by several points, enough to wrongly condemn or clear a pump.

Ref: BEE Guidebook Paper 3, Ch. 6 (Pumps – head, flow and efficiency measurement).

**46** An office's split ACs use 4,20,000 kWh a year at a  $20^\circ\text{C}$  set point. Following BEE's advisory, the set point is raised to  $24^\circ\text{C}$ , assuming a saving of 6 % of the baseline per  $^\circ\text{C}$ . To keep comfort, 60 BLDC ceiling fans of 28 W are switched on for 2,500 h a year. The net annual saving is about:

[5P3CEM046 | Chapter 10: Energy Conservation in Buildings and ECBC | 10.3 AC default set point raised to  $24^\circ\text{C}$  with ceiling fans]

a) 1,00,800 kWh

b) 93,300 kWh

c) 71,400 kWh

**d) 96,600 kWh**

### Answer & explanation

Correct answer: (d)  $\approx 96,600$  kWh.

Given: baseline 4,20,000 kWh; 4 °C rise; 6 %/°C; fans  $60 \times 28$  W for 2,500 h.

Formula: AC saving = baseline  $\times 0.06 \times \Delta T$ ; fan energy =  $n \times W \times h / 1,000$ ; net = AC saving – fan energy.

Substitution: AC saving =  $4,20,000 \times 0.24 = 1,00,800$  kWh; fans =  $60 \times 28 \times 2,500 / 1,000 = 4,200$  kWh; net =  $1,00,800 - 4,200$ .

Result: **net saving  $\approx 96,600$  kWh a year.**

Why the others are wrong: (a) forgets the energy of the added fans; (b) uses 50 W conventional fans instead of the 28 W BLDC fans stated; (c) counts a 3 °C rise (21 to 24). Air movement of about 0.5–1 m/s makes 24–26 °C feel comfortable, so fans plus a higher set point give large net savings.

Ref: BEE Guidebook Paper 3, Ch. 10 (Energy conservation in buildings – HVAC set points, thermal comfort); BEE advisory on AC default temperature.

**47** A pharmaceutical clean-room AHU supplies 60 air changes per hour (ACPH) with its fan drawing 75 kW.

Validation allows 40 ACPH when the room is unoccupied for 10 h a day on 330 days. The system resistance is almost all filters and ductwork (pressure  $\propto$  flow<sup>2</sup>) and a VFD is fitted. The annual fan energy saved is about:

[5P3CEM047 | Chapter 4: HVAC and Refrigeration System | 4.16 Clean-room air change rate setback at night]

a) 82,500 kWh

**b) 1,74,200 kWh**

c) 1,37,500 kWh

d) 1,92,600 kWh

### Answer & explanation

Correct answer: (b)  $\approx 1,74,200$  kWh.

Given: fan input 75 kW at 60 ACPH; setback 40 ACPH; 10 h  $\times$  330 days = 3,300 h.

Formula: with a system curve through the origin, fan power  $\propto$  flow<sup>3</sup>; saving =  $(P_1 - P_2) \times$  hours.

Substitution: flow ratio =  $40/60 = 0.667$ ;  $0.667^3 = 0.296$ ;  $P_2 = 75 \times 0.296 = 22.2$  kW; saving =  $52.8$  kW  $\times$  3,300.

Result: **saving  $\approx 1,74,200$  kWh a year, about 70 % of the fan power during setback hours.**

Why the others are wrong: (a) cuts power in proportion to flow (one-third), ignoring the fan laws; (c) uses a square law; (d) counts 365 days instead of the 330 days stated. Clean rooms are often designed with generous ACPH and run that way round the clock. A validated night setback, with pressure cascades kept and recovery time proven, is one of the largest HVAC savings in pharma.

Ref: BEE Guidebook Paper 3, Ch. 4 (HVAC and refrigeration – energy saving opportunities in air handling).

**48** A designer must meet an SHGC limit of 0.25 for vertical glazing and wants the most daylight per unit of solar heat. Options: A (VLT 0.62, SHGC 0.42), B (VLT 0.48, SHGC 0.24), C (VLT 0.30, SHGC 0.22), D (VLT 0.20, SHGC 0.25).

Which glass should be chosen, and what is its light-to-solar-gain ratio (LSG = VLT/SHGC)?

[5P3CEM048 | Chapter 10: Energy Conservation in Buildings and ECBC | 10.3 Choosing glazing by light-to-solar-gain ratio]

**a) Glass B, LSG 2.00**

b) Glass A, LSG 1.48

c) Glass C, LSG 1.36

d) Glass D, LSG 0.80

### Answer & explanation

Correct answer: (a).

Given: SHGC limit 0.25; VLT and SHGC for glasses A–D.

Formula:  $LSG = VLT / SHGC$ ; choose the highest LSG among glasses that satisfy  $SHGC \leq 0.25$ .

Substitution: A:  $0.62/0.42 = 1.48$  but SHGC 0.42 fails the limit; B:  $0.48/0.24 = 2.00$ ; C:  $0.30/0.22 = 1.36$ ; D:  $0.20/0.25 = 0.80$ .

Result: **Glass B, LSG 2.00, the spectrally selective option.**

Why the others are wrong: (b) gives the most light but breaks the SHGC limit and would raise cooling load; (c) meets the limit but admits much less daylight per unit of heat; (d) only just meets the limit and is the darkest, so electric lighting would rise. A high LSG cuts both cooling and lighting energy, which matters in India's cooling-dominated climates. ECBC sets SHGC limits and minimum VLT to stop designers meeting SHGC with dark glass.

Ref: BEE Guidebook Paper 3, Ch. 10 (Energy conservation in buildings and ECBC – fenestration, SHGC and VLT).

**49** A VFD was fitted on a 110 kW cooling-water pump expecting to cut flow, but process changes mean it now runs at 50 Hz all the time. Direct-on-line, the motor drew 98.0 kW. The drive efficiency at full output is 97 %. Over 6,000 h a year, the extra energy taken by keeping the VFD in circuit is about:

[5P3CEM049 | Chapter 2: Electrical Motors | 2.9 VFD left running at full speed – a net energy penalty]

- a) 17,640 kWh
- b) 18,190 kWh**
- c) 26,550 kWh
- d) 20,410 kWh

#### Answer & explanation

Correct answer: (b)  $\approx 18,190$  kWh.

Given: motor input 98.0 kW at 50 Hz; drive efficiency 0.97; 6,000 h.

Formula: input with drive = motor input /  $\eta_{\text{drive}}$ ; extra = (input with drive – motor input)  $\times$  hours.

Substitution:  $98.0 / 0.97 = 101.03$  kW; extra = 3.03 kW;  $\times 6,000$ .

Result: **extra  $\approx 18,190$  kWh a year; a bypass contactor at full speed avoids it.**

Why the others are wrong: (a) takes 3 % of the motor input ( $98 \times 0.03$ ), but drive loss is 3 % of drive input; (c) uses 8,760 h instead of the stated 6,000 h; (d) bases the loss on the 110 kW rating instead of the actual 98 kW. A VFD saves energy only when it reduces speed. At full speed it adds its own losses and harmonics, so audits should check the drive's actual frequency log and fit a bypass or retune flow control where it never slows down.

Ref: BEE Guidebook Paper 3, Ch. 2 (Electric motors – variable speed drives).

**50** A secondary chilled water pump driven by a motor drawing 90 kW (motor efficiency 93 %, pump efficiency 70 %) circulates water in a closed loop. The motor is air-cooled outside the water. The chiller compressor runs at 0.60 kW/TR. The additional chiller compressor power needed only to remove the heat added to the water by this pump is about:

[5P3CEM050 | Chapter 4: HVAC and Refrigeration System | 4.8 Chilled water pump heat – extra chiller load from pumping energy]

- a) 10.0 kW
- b) 15.4 kW
- c) 23.8 kW
- d) 14.3 kW**

#### Answer & explanation

Correct answer: (d)  $\approx 14.3$  kW.

Given: motor input 90 kW;  $\eta_m$  0.93;  $\eta_{\text{pump}}$  0.70; 0.60 kW/TR; 1 TR = 3.516 kW.

Formula: heat into water = shaft power = input  $\times \eta_m$  (pump losses also end up in the water); extra TR = heat / 3.516; extra chiller kW = TR  $\times$  kW/TR.

Substitution:  $90 \times 0.93 = 83.7$  kW;  $83.7 / 3.516 = 23.81$  TR;  $23.81 \times 0.60$ .

Result:  $\approx$  **14.3 kW of extra compressor power, so each pumping kW really costs about 1.16 kW.**

Why the others are wrong: (a) treats only the hydraulic power ( $83.7 \times 0.70$ ) as heat, but friction in the pump and in the pipework also degrades to heat in the water; (b) uses the 90 kW motor input, yet motor losses go to the plant room air, not the loop; (c) reports the 23.8 TR of extra load as if it were kW. Reducing chilled water pumping (VFD, lower head) therefore saves twice.

Ref: BEE Guidebook Paper 3, Ch. 4 (HVAC – energy saving in chilled water systems).

..... End of Section I .....

## Section – II: SHORT DESCRIPTIVE QUESTIONS      Marks: 8 x 5 = 40

4. Answer all Eight questions
5. Each question carries Five marks

**S-1** An engineering unit on kVAh billing installs a 400 kWp rooftop solar plant behind its HT meter. The APFC panel is unchanged. Monthly data:

| Parameter             | Before solar    | After solar     |
|-----------------------|-----------------|-----------------|
| Grid kWh              | 3,00,000        | 2,52,000        |
| Grid kVAh (lagging)   | 1,20,000        | 1,20,000        |
| Tariff                | ₹ 8.00 per kVAh | ₹ 8.00 per kVAh |
| Plant operating hours | 600 h/month     | 600 h/month     |

(a) Average PF before and after solar. (1.5 Marks)

(b) Monthly kVAh and energy bill before and after, and how much of the expected saving is lost because the PF fell. (2 Marks)

(c) Additional capacitor kVAh needed to restore the original PF. (1.5 Marks)

[5P3CEM051 | Chapter 1: Electrical Systems | 1.2 Rooftop solar behind the meter – PF drop and kVAh billing]

### S-1 – Model answer (5 marks)

Given: before 3,00,000 kWh and 1,20,000 kVAh; after 2,52,000 kWh and 1,20,000 kVAh; ₹ 8.00/kVAh; 600 h/month.

(a) Power factor (1.5 Marks)

Formula:  $PF = kWh / \sqrt{(kWh^2 + kVAh^2)}$  (0.5 Mark)

Substitution: before  $3,00,000 / \sqrt{(3,00,000^2 + 1,20,000^2)}$ ; after  $2,52,000 / \sqrt{(2,52,000^2 + 1,20,000^2)}$  (0.5 Mark)

Result: **PF before  $\approx$  0.928; PF after  $\approx$  0.903 – solar displaces kW but the plant's kVAh still comes from the grid** (0.5 Mark)

(b) kVAh and bill (2 Marks)

Substitution: kVAh before = 3,23,110; after = 2,79,113 (0.5 Mark)

Result: **bill before  $\approx$  ₹ 25,84,880; after  $\approx$  ₹ 22,32,904; saving  $\approx$  ₹ 3,51,976 per month** (0.5 Mark)

Had the PF stayed at 0.928: kVAh =  $2,52,000 / 0.9285 = 2,71,412$ ; bill  $\approx$  ₹ 21,71,298 (0.5 Mark)

Result:  **$\approx$  ₹ 61,605 a month (about 15 % of the expected saving) is lost to the lower PF** (0.5 Mark)

(c) Capacitor to restore PF (1.5 Marks)

Formula: target kVArh = kWh\_after  $\times$  tan  $\phi$ \_original, where tan  $\phi$  = 1,20,000 / 3,00,000 = 0.40 (0.5 Mark)

Substitution: target = 2,52,000  $\times$  0.40 = 1,00,800 kVArh; reduction = 1,20,000 – 1,00,800 = 19,200 kVArh;  $\div$  600 h (0.5 Mark)

Result:  $\approx$  **32 kVAr extra (one more APFC step), set to the grid-side CT so the controller sees the net import** (0.5 Mark)

Marking: (a) 0.5 + 0.5 + 0.5; (b) 0.5  $\times$  4; (c) 0.5  $\times$  3.

**S-2** A 37 kW fixed-speed hydraulic pump on a plastic injection moulding machine works a 30 s cycle. Measured motor input over one cycle, and the maker's figures for a servo-motor-driven pump replacement:

| Phase of cycle                     | Duration | Existing input | Servo-pump input |
|------------------------------------|----------|----------------|------------------|
| Injection                          | 4 s      | 32 kW          | 30 kW            |
| Holding pressure                   | 8 s      | 18 kW          | 6 kW             |
| Cooling / mould open (pump idling) | 18 s     | 11 kW          | 0.8 kW           |

The machine runs 6,500 h a year; power costs ₹ 8.00/kWh; the retrofit costs ₹ 7,50,000.

(a) Average input power of the existing drive. (1.5 Marks)

(b) Average input power with the servo pump. (1 Mark)

(c) Annual energy and cost saving, and simple payback. (1.5 Marks)

(d) Why is the saving largest in the holding and idling phases? (1 Mark)

[5P3CEM052 | Chapter 2: Electrical Motors | 2.9 Injection moulding machine – servo-pump retrofit on a fixed-speed hydraulic drive]

### **S-2 – Model answer (5 marks)**

Given: cycle 30 s; existing 32 / 18 / 11 kW for 4 / 8 / 18 s; servo 30 / 6 / 0.8 kW; 6,500 h; ₹ 8.00/kWh; cost ₹ 7,50,000.

(a) Existing average (1.5 Marks)

Formula:  $P_{avg} = \Sigma(P \times t) / \text{cycle time}$  (0.5 Mark)

Substitution:  $(4 \times 32 + 8 \times 18 + 18 \times 11) / 30 = (128 + 144 + 198) / 30 = 470 / 30$  (0.5 Mark)

Result:  **$P_{avg} \approx 15.67$  kW** (0.5 Mark)

(b) Servo average (1 Mark)

Substitution:  $(4 \times 30 + 8 \times 6 + 18 \times 0.8) / 30 = (120 + 48 + 14.4) / 30 = 182.4 / 30$  (0.5 Mark)

Result:  **$P_{avg} \approx 6.08$  kW** (0.5 Mark)

(c) Saving and payback (1.5 Marks)

Substitution:  $15.67 - 6.08 = 9.59$  kW;  $\times$  6,500 h (0.5 Mark)

Result:  $\approx$  **62,313 kWh a year,  $\approx$  ₹ 4,98,507 a year (61 % saving)** (0.5 Mark)

Result: **simple payback  $\approx$  7,50,000 / 4,98,507  $\approx$  1.5 years** (0.5 Mark)

(d) Reason (1 Mark)

A fixed-speed pump delivers full flow all the time; in holding and idling almost all of it is spilled over relief or unloading valves as heat (which also loads the oil cooler). The servo pump slows to the small flow needed to hold pressure and almost stops when no motion is needed, so it supplies only the hydraulic power actually used. (1 Mark)

Marking: (a)  $0.5 \times 3$ ; (b)  $0.5 \times 2$ ; (c)  $0.5 \times 3$ ; (d) 1.

**S-3** A plant uses  $30 \text{ m}^3/\text{min}$  of free air. The main compressors run at 8.0 bar(g) only because one machine ( $2.5 \text{ m}^3/\text{min}$ ) needs 7.5 bar(g) at its inlet; all other users need 6.0 bar(g), and the distribution drop is 0.5 bar. Present specific power is 6.6 kW per  $\text{m}^3/\text{min}$ . Proposal: lower the main header to 6.5 bar(g) and feed the  $2.5 \text{ m}^3/\text{min}$  user through a small booster from 6.5 to 8.0 bar(g), which needs 1.6 kW per  $\text{m}^3/\text{min}$ . Take main compressor power  $\propto [(P_2/P_1)^{(n-1)/n} - 1]$  with  $n = 1.35$ ,  $P_1 = 1.013 \text{ bar(a)}$ ; 8,000 h/year; ₹ 8.00/kWh.

(a) Specific power of the main compressors at 6.5 bar(g). (2 Marks)

(b) Total compressor power before and after the change. (2 Marks)

(c) Annual saving. (1 Mark)

[5P3CEM053 | Chapter 3: Compressed Air System | 3.5 One high-pressure user – booster compressor instead of raising the whole network]

### S-3 – Model answer (5 marks)

Given:  $30 \text{ m}^3/\text{min}$ ;  $8.0 \rightarrow 6.5 \text{ bar(g)}$ ; 6.6 kW per  $\text{m}^3/\text{min}$  at 8.0 bar(g); booster  $2.5 \text{ m}^3/\text{min}$  at 1.6 kW per  $\text{m}^3/\text{min}$ ;  $n = 1.35$ , so  $(n - 1)/n = 0.2593$ ; 8,000 h; ₹ 8.00/kWh.

(a) Main compressor at 6.5 bar(g) (2 Marks)

Formula:  $SP_2 = SP_1 \times [(P_2'/P_1)^{0.2593} - 1] / [(P_1'/P_1)^{0.2593} - 1]$  (0.5 Mark)

Substitution:  $(7.513/1.013)^{0.2593} - 1 = 0.6815$ ;  $(9.013/1.013)^{0.2593} - 1 = 0.7628$ ; ratio = 0.8934 (1 Mark)

Result: **SP at 6.5 bar(g)  $\approx 5.90 \text{ kW per m}^3/\text{min}$  (about 10.7 % lower)** (0.5 Mark)

(b) Power before and after (2 Marks)

Before:  $30 \times 6.6 = 198 \text{ kW}$  (0.5 Mark)

After: main  $30 \times 5.90 = 176.9 \text{ kW}$  (all air, including the booster's inlet air, is compressed to 6.5 bar(g)); booster  $2.5 \times 1.6 = 4.0 \text{ kW}$  (1 Mark)

Result: **after  $\approx 180.9 \text{ kW}$ ; reduction  $\approx 17.1 \text{ kW}$**  (0.5 Mark)

(c) Annual saving (1 Mark)

Result:  **$17.1 \times 8,000 \approx 1,36,800 \text{ kWh a year} \approx ₹ 10,94,400$**  (0.5 Mark). Leakage and unregulated end uses also fall at the lower header pressure, so the real saving will be higher. (0.5 Mark)

Marking: (a)  $0.5 + 1 + 0.5$ ; (b)  $0.5 + 1 + 0.5$ ; (c)  $0.5 + 0.5$ .

**S-4** An air-cooled chiller averages 250 TR ( $\approx 879 \text{ kW}$ ) for 1,500 summer hours at  $40^\circ\text{C}$  dry bulb and  $24^\circ\text{C}$  wet bulb, with a specific power of  $1.30 \text{ kW/TR}$ . Adiabatic pads of 75 % saturation effectiveness are proposed on the condenser air inlets. Data: specific power falls 2.0 % per K drop in condenser inlet air temperature (linear); condenser air flow  $180 \text{ m}^3/\text{s}$  at  $1.15 \text{ kg/m}^3$ ; latent heat of water  $2,430 \text{ kJ/kg}$ ; bleed-off 20 % of evaporation; water ₹ 60/ $\text{m}^3$ ; pad pumps 4 kW; electricity ₹ 9.00/kWh.

(a) Condenser inlet air temperature with pads. (1 Mark)

(b) New specific power and the annual electricity saved by the chiller. (1.5 Marks)

(c) Annual water consumption and its cost. (1.5 Marks)

(d) Net annual saving. (1 Mark)

[5P3CEM054 | Chapter 4: HVAC and Refrigeration System | 4.16 Evaporative pre-cooling of air-cooled chiller condenser air]

### S-4 – Model answer (5 marks)

Given: 250 TR ( $\approx 879$  kW); 1,500 h; DBT 40 °C; WBT 24 °C;  $\epsilon = 0.75$ ; 1.30 kW/TR;  $-2.0$  %/K; air  $180 \text{ m}^3/\text{s} \times 1.15 \text{ kg/m}^3$ ;  $h_{fg}$  2,430 kJ/kg; bleed 20 %; ₹ 60/ $\text{m}^3$ ; pumps 4 kW; ₹ 9.00/kWh.

(a) Inlet temperature (1 Mark)

Formula:  $T_{out} = DBT - \epsilon (DBT - WBT)$ ; substitution:  $40 - 0.75 \times 16$  (0.5 Mark)

Result:  **$T \approx 28.0$  °C, a 12 K drop** (0.5 Mark)

(b) Chiller saving (1.5 Marks)

Substitution: new kW/TR =  $1.30 \times (1 - 0.02 \times 12) = 1.30 \times 0.76$  (0.5 Mark)

Result:  $\approx$  **0.988 kW/TR** (0.5 Mark); saving =  $250 \times (1.30 - 0.988) \times 1,500 =$  **1,17,000 kWh** (0.5 Mark)

(c) Water (1.5 Marks)

Formula: evaporation =  $\dot{m} c_p \Delta T / h_{fg}$ ;  $\dot{m} = 180 \times 1.15 = 207 \text{ kg/s}$  (0.5 Mark)

Substitution:  $207 \times 1.005 \times 12 / 2,430 = 1.027 \text{ kg/s} = 3.70 \text{ m}^3/\text{h}$ ;  $\times 1,500 \text{ h} = 5,548 \text{ m}^3$ ; with 20 % bleed = 6,657  $\text{m}^3$  (0.5 Mark)

Result: **water  $\approx 6,657 \text{ m}^3$  a year, costing  $\approx$  ₹ 3,99,427** (0.5 Mark)

(d) Net saving (1 Mark)

Substitution: net electricity =  $1,17,000 - 4 \times 1,500 = 1,11,000 \text{ kWh} =$  ₹ 9,99,000 (0.5 Mark)

Result: **net  $\approx$  ₹ 9,99,000 – ₹ 3,99,427  $\approx$  ₹ 5,99,573 a year ( $\approx$  ₹  $6.0 \times 10^5$ ); water treatment and pad scaling must be managed** (0.5 Mark)

Marking: (a) 0.5 + 0.5; (b) 0.5  $\times$  3; (c) 0.5  $\times$  3; (d) 0.5 + 0.5.

**S-5** A chiller plant rejects 2,450 kW through its cooling tower. Condenser-water flow is  $700 \text{ m}^3/\text{h}$ , pumped by a VFD pump drawing 55 kW. The proposal is to cut flow to  $470 \text{ m}^3/\text{h}$ , where the pump will draw 22 kW. The chiller maker says the average condenser-water temperature will rise by 0.9 K, and chiller power (390 kW at this load) rises 2.5 % per K. Take  $c_p$  4.186 kJ/kg·K, 1,000 kg/ $\text{m}^3$ , 5,000 h/year and ₹ 8.00/kWh.

(a) Condenser-water range before and after. (1.5 Marks)

(b) Pump power saved. (1 Mark)

(c) Chiller power penalty. (1.5 Marks)

(d) Net annual saving, and whether to proceed. (1 Mark)

[5P3CEM055 | Chapter 7: Cooling Towers | 7.4 Condenser-water flow reduction – pump saving against chiller penalty]

### **S-5 – Model answer (5 marks)**

Given:  $Q = 2,450 \text{ kW}$ ; flow  $700 \rightarrow 470 \text{ m}^3/\text{h}$ ; pump  $55 \rightarrow 22 \text{ kW}$ ;  $\Delta T_{avg} +0.9 \text{ K}$ ; chiller 390 kW,  $+2.5$  %/K; 5,000 h; ₹ 8.00/kWh.

(a) Range (1.5 Marks)

Formula: range =  $Q / (\dot{m} c_p)$  with  $\dot{m} = \text{flow} \times 1,000 / 3,600$  (0.5 Mark)

Substitution: before  $2,450 \times 3,600 / (700 \times 1,000 \times 4.186)$ ; after  $2,450 \times 3,600 / (470 \times 1,000 \times 4.186)$  (0.5 Mark)

Result: **range  $\approx 3.01 \text{ K}$  before and  $\approx 4.48 \text{ K}$  after** (0.5 Mark)

(b) Pump saving (1 Mark)

Result:  **$55 - 22 = 33 \text{ kW}$  saved** (1 Mark)

(c) Chiller penalty (1.5 Marks)

Formula: penalty = chiller kW  $\times 0.025 \times \Delta T$  (0.5 Mark)

Substitution:  $390 \times 0.025 \times 0.9$  (0.5 Mark)

Result:  $\approx$  **8.78 kW extra chiller power** (0.5 Mark)

(d) Net saving (1 Mark)

Result:  $\text{net} \approx 33 - 8.78 = 24.2 \text{ kW}$ ;  $\times 5,000 \text{ h} \approx 1,21,125 \text{ kWh a year} \approx ₹ 9,69,000$  – proceed, checking the tower's minimum wetting flow and condenser tube velocity at  $470 \text{ m}^3/\text{h}$  (1 Mark)

Marking: (a)  $0.5 \times 3$ ; (b) 1; (c)  $0.5 \times 3$ ; (d) 1.

**S-6** A clear-water pump delivers  $150 \text{ m}^3/\text{h}$  from a sump. At present it sits above the sump: static suction lift 6.5 m, suction-line friction including the foot valve 2.0 m, discharge static head 22.0 m, discharge friction 4.5 m. NPSH required is 3.5 m; take atmospheric head 10.1 m and vapour-pressure head 0.43 m. Because it cavitates, its efficiency is only 62 %. It is proposed to move the same type of pump into a dry pit so that it has 0.5 m of positive suction head, a short suction line with 0.6 m friction and no foot valve; its efficiency will then recover to 72 %. Motor efficiency 90 %; 6,000 h/year; ₹ 7.50/kWh.

(a) Total head and input power at present. (1.5 Marks)

(b) Total head and input power after relocation. (1.5 Marks)

(c) NPSH available before and after. (1 Mark)

(d) Annual saving, and why the static lift cannot be "saved". (1 Mark)

[5P3CEM056 | Chapter 6: Pumps and Pumping System | 6.4 Relocating a suction-lift pump to flooded suction – what really changes]

#### S-6 – Model answer (5 marks)

Given:  $Q = 150 \text{ m}^3/\text{h}$ ; present  $H = 6.5 + 2.0 + 22.0 + 4.5$ ;  $\eta = 0.62 \rightarrow 0.72$ ;  $\eta_m = 0.90$ ; 6,000 h; ₹ 7.50/kWh.

(a) Present (1.5 Marks)

Formula:  $H = \text{static lift} + \text{friction}$ ; input  $\text{kW} = \rho g Q H / (\eta_{\text{pump}} \eta_m)$  (0.5 Mark)

Substitution:  $H = 35.0 \text{ m}$ ;  $1,000 \times 9.81 \times 0.04167 \times 35.0 / (0.62 \times 0.90)$  (0.5 Mark)

Result:  **$H = 35.0 \text{ m}$ ; input  $\approx 25.64 \text{ kW}$**  (0.5 Mark)

(b) After relocation (1.5 Marks)

Total static lift, sump water level to delivery, is unchanged at  $6.5 + 22.0 = 28.5 \text{ m}$ ; friction =  $0.6 + 4.5$  (0.5 Mark)

Substitution:  $H = 28.5 + 0.6 + 4.5 = 33.6 \text{ m}$ ;  $1,000 \times 9.81 \times 0.04167 \times 33.6 / (0.72 \times 0.90)$  (0.5 Mark)

Result:  **$H = 33.6 \text{ m}$ ; input  $\approx 21.19 \text{ kW}$**  (0.5 Mark)

(c) NPSH available (1 Mark)

Before:  $10.1 - 6.5 - 2.0 - 0.43 = 1.17 \text{ m}$ , below the 3.5 m required, so it cavitates (0.5 Mark)

After:  $10.1 + 0.5 - 0.6 - 0.43 = 9.57 \text{ m}$ , ample (0.5 Mark)

(d) Saving and explanation (1 Mark)

Result:  **$25.64 - 21.19 = 4.44 \text{ kW}$** ;  $\times 6,000 \text{ h} \approx 26,664 \text{ kWh} \approx ₹ 1,99,980 \text{ a year}$  (0.5 Mark). The water must still be lifted 28.5 m from sump level to the delivery point whatever the pump's position; only the foot-valve friction and the cavitation-related efficiency loss are recovered. Of the 4.44 kW, only about 1 kW comes from lower head; the rest comes from curing cavitation. (0.5 Mark)

Marking: (a)  $0.5 \times 3$ ; (b)  $0.5 \times 3$ ; (c)  $0.5 \times 2$ ; (d)  $0.5 \times 2$ .

**S-7** An off-grid plant runs two 500 kVA (400 kW each) DG sets in parallel to supply a steady daytime load of 520 kW for 8 h a day, 300 days a year. A solar PV plant is added; over the 8 h it averages 180 kW. The DG maker requires each running set to stay above 30 % of rated kW. Specific energy generation (kWh per litre): 3.5 at 65 % load per set; 3.1 at 42.5 % load per set. Diesel costs ₹ 92 per litre.

(a) Maximum PV power the site can absorb with both sets running. (1 Mark)

(b) Daily diesel consumption before PV. (1 Mark)

(c) Daily diesel consumption with the average PV output. (1.5 Marks)

(d) Annual diesel and cost saving. (1 Mark)

(e) Why should both sets keep running even though the DG load (340 kW) could be carried by one set? (0.5 Mark)

[5P3CEM057 | Chapter 9: Diesel / Natural Gas Power Generating System | 9.6 Solar PV added to a DG-powered site – minimum loading limits and fuel saved]

#### S-7 – Model answer (5 marks)

Given: load 520 kW;  $2 \times 400$  kW sets; minimum 30 % per set; PV average 180 kW; 8 h  $\times$  300 days; SEG 3.5 and 3.1 kWh/litre; ₹ 92/litre.

(a) PV absorption limit (1 Mark)

Formula: PV limit = load – (sets running  $\times$  0.30  $\times$  rated kW); Substitution: minimum DG output =  $2 \times 0.30 \times 400 = 240$  kW (0.5 Mark)

Result: **maximum PV  $\approx 520 - 240 = 280$  kW; any more must be curtailed** (0.5 Mark)

(b) Before PV (1 Mark)

Substitution: each set at 260 kW = 65 %;  $520 \times 8 / 3.5$  (0.5 Mark)

Result:  **$\approx 1,188.6$  litres a day** (0.5 Mark)

(c) With PV (1.5 Marks)

Substitution: DG load =  $520 - 180 = 340$  kW, i.e. 170 kW per set = 42.5 % (0.5 Mark);  $340 \times 8 / 3.1$  (0.5 Mark)

Result:  **$\approx 877.4$  litres a day** (0.5 Mark)

(d) Annual saving (1 Mark)

Result:  **$(1,188.6 - 877.4) \times 300 \approx 93,346$  litres a year  $\approx ₹ 85.9$  lakh ( $\approx ₹ 8.59 \times 10^6$ )** (1 Mark). Note that the DG kWh fell by 35 % but fuel only by 26 %, because the sets now run at a poorer part-load SEG.

(e) Reliability (0.5 Mark)

A cloud can remove most of the PV output within seconds; one set alone would then face 520 kW against its 400 kW rating and trip. Spinning reserve equal to the PV output must be kept unless a battery or fast load-shedding scheme is added. (0.5 Mark)

Marking: (a) 0.5 + 0.5; (b) 0.5 + 0.5; (c) 0.5  $\times$  3; (d) 1; (e) 0.5.

**S-8** A 200-room hotel has average occupancy of 70 %. Each room has 300 W of lighting and appliances and a fan-coil unit carrying 0.9 TR ( $\approx 3.16$  kW), served by a chilled-water plant at 1.2 kW/TR. At present, lights and AC stay on while guests are out (about 10 h a day), and the 30 % unsold rooms are kept cooled for 12 h a day to look ready. A guest-room management system (key card plus occupancy-linked thermostat) will switch off lights and reduce AC load by 60 % while guests are out, and will cool unsold rooms only for 1 h before check-in. Electricity ₹ 9.00/kWh; installed cost ₹ 22,000 per room.

(a) Daily saving in occupied rooms while guests are out. (1.5 Marks)

(b) Daily saving in unsold rooms. (1.5 Marks)

(c) Annual energy and cost saving, and payback. (2 Marks)

[5P3CEM058 | Chapter 10: Energy Conservation in Buildings and ECBC | 10.4 Hotel guest-room energy management – key-card and unsold-room setback]

#### S-8 – Model answer (5 marks)

Given: 200 rooms; 70 % occupancy (140 occupied, 60 unsold); 300 W per room; 0.9 TR ( $\approx 3.16$  kW) at 1.2 kW/TR = 1.08 kW per room; 10 h absence; 60 % AC reduction; unsold rooms 12 h  $\rightarrow$  1 h; ₹ 9.00/kWh; ₹ 22,000 per room.

(a) Occupied rooms (1.5 Marks)

Formula: saving = rooms  $\times$  hours  $\times$  (lighting kW + AC kW  $\times$  0.60) (0.5 Mark)

Substitution:  $140 \times 10 \times (0.30 + 1.08 \times 0.60) = 140 \times 10 \times 0.948$  (0.5 Mark)

Result:  $\approx 1,327.2$  kWh a day (0.5 Mark)

(b) Unsold rooms (1.5 Marks)

Formula: saving = rooms  $\times$  hours avoided  $\times$  AC kW (0.5 Mark)

Substitution:  $60 \times (12 - 1) \times 1.08$  (0.5 Mark)

Result:  $\approx 712.8$  kWh a day (0.5 Mark)

(c) Annual saving and payback (2 Marks)

Substitution:  $(1,327.2 + 712.8) \times 365 = 2,040 \times 365$  (0.5 Mark)

Result:  $\approx 7,44,600$  kWh a year  $\approx$  ₹ 67,01,400 ( $\approx$  ₹ 6.70  $\times 10^6$ ) (0.5 Mark)

Cost =  $200 \times 22,000 =$  ₹ 44,00,000 (0.5 Mark)

Result: **simple payback  $\approx$  0.66 year (about 8 months)** (0.5 Mark)

Marking: (a)  $0.5 \times 3$ ; (b)  $0.5 \times 3$ ; (c)  $0.5 \times 4$ .

..... End of Section II .....

### Section – III: LONG DESCRIPTIVE QUESTIONS      Marks: 6 x 10 = 60

6. Answer all Six questions

7. Each question carries Ten marks

**L-1** A textile processing unit's LT bus runs at 433 V because its 2,000 kVA, 11/0.433 kV transformer is on its nominal tap; all equipment is rated 415 V. The auditor proposes changing the off-load tap to give 415 V. Data:

| Item                                                            | Data                                                              |
|-----------------------------------------------------------------|-------------------------------------------------------------------|
| Transformer no-load loss at present voltage                     | 2.4 kW, energised 8,760 h/year (take no-load loss $\propto V^2$ ) |
| Magnetic-ballast lighting (power $\propto V^2$ )                | 120 kW                                                            |
| LED lighting with constant-power drivers                        | 60 kW                                                             |
| Lightly loaded induction motors                                 | 400 kW input; maker's tests show input falls by 1.5 % at 415 V    |
| Thermostat-controlled electric heaters                          | 150 kW                                                            |
| Capacitor banks rated 800 kVAr at 440 V (output $\propto V^2$ ) | in service                                                        |
| Plant operating hours / tariff                                  | 7,000 h/year / ₹ 8.00 per kWh                                     |

(a) Percentage voltage reduction and the saving in transformer no-load loss (kW and kWh/year). (2 Marks)

(b) Saving in magnetic-ballast lighting. (1.5 Marks)

(c) Saving in the motor group. (1 Mark)

(d) Explain why no energy saving is credited for the LED lighting and the heaters. (1.5 Marks)

(e) Capacitor output at 433 V and at 415 V, and the consequence for the PF. (2 Marks)

(f) Total annual energy and cost saving. (2 Marks)

### L-1 – Model answer (10 marks)

Given: V from 433 V to 415 V;  $P_o = 2.4$  kW; lighting 120 kW ( $\propto V^2$ ); motors 400 kW ( $-1.5\%$ ); capacitors 800 kVAr at 440 V; 7,000 h (transformer 8,760 h); ₹ 8.00/kWh.

(a) Voltage reduction and no-load loss (2 Marks)

Formula: reduction =  $1 - 415/433$ ;  $P_o$  saving =  $P_o \times [1 - (415/433)^2]$  (0.5 Mark)

Substitution:  $415/433 = 0.9584$ ;  $(0.9584)^2 = 0.9186$  (0.5 Mark)

Result: **voltage reduction  $\approx 4.16\%$ ; no-load loss saving  $\approx 0.195$  kW** (0.5 Mark);  $\times 8,760$  h =  **$\approx 1,712$  kWh a year** (0.5 Mark)

(b) Magnetic-ballast lighting (1.5 Marks)

Formula: saving =  $120 \times [1 - (415/433)^2]$  (0.5 Mark)

Substitution:  $120 \times 0.0814$  (0.5 Mark)

Result:  **$\approx 9.77$  kW, with a similar fall in light output that must stay within design lux** (0.5 Mark)

(c) Motors (1 Mark)

Result:  **$400 \times 0.015 = 6.0$  kW** – lower voltage cuts magnetising current and core loss of lightly loaded motors (1 Mark)

(d) LED and heaters (1.5 Marks)

LED drivers regulate to constant power, so their input is unchanged at the lower voltage (0.5 Mark). A thermostat-controlled heater draws  $(415/433)^2$  less power, but it stays on longer to deliver the same heat, so energy is unchanged (0.5 Mark). Crediting a  $V^2$  saving to either would overstate the result; voltage optimisation saves only on voltage-sensitive, non-regulated loads (0.5 Mark).

(e) Capacitors (2 Marks)

Formula:  $Q = Q_{\text{rated}} \times (V/V_{\text{rated}})^2$  (0.5 Mark)

Substitution:  $800 \times (433/440)^2$  and  $800 \times (415/440)^2$  (0.5 Mark)

Result:  **$\approx 774.7$  kVAr at 433 V and  $\approx 711.7$  kVAr at 415 V, i.e.  $\approx 63.1$  kVAr less** (0.5 Mark); the PF will fall unless about 63 kVAr more is switched in, so the APFC setting and spare steps must be checked (0.5 Mark).

(f) Total saving (2 Marks)

Substitution:  $(9.77 + 6.0) \times 7,000 = 1,10,387$  kWh; plus 1,712 kWh from the transformer (1 Mark)

Result:  **$\approx 1,12,098$  kWh a year  $\approx$  ₹ 8,96,787 ( $\approx$  ₹  $8.97 \times 10^5$ ) at no capital cost beyond a planned shutdown** (1 Mark)

Marking: (a)  $0.5 \times 4$ ; (b)  $0.5 \times 3$ ; (c) 1; (d)  $0.5 \times 3$ ; (e)  $0.5 \times 4$ ; (f)  $1 + 1$ .

**L-2** A process cooling tower cell has an axial fan that absorbs 30 kW at the fan shaft at full speed. It is driven by an old induction motor (efficiency 91 % at this load) through a right-angle gearbox (efficiency 94 %) and runs at full speed for 8,000 h a year; cold-water temperature is allowed to float. The proposal is a low-speed direct-drive permanent-magnet synchronous motor (efficiency 95 %) on a VFD (efficiency 97 %), with no gearbox, controlled from cold-water temperature. With control, the fan will run at 100 % speed for 3,000 h, 80 % for 3,000 h and 60 % for 2,000 h (fan shaft power  $\propto$  speed<sup>3</sup>). Power ₹ 8.00/kWh; installed cost ₹ 26,00,000.

(a) Electrical input and annual energy of the existing drive. (2 Marks)

(b) Fan shaft power and electrical input at each speed with the new drive. (2.5 Marks)

(c) Annual energy with the new drive and the annual saving (kWh and ₹). (2 Marks)

(d) Split the saving into the part due to the drive train alone (new drive at full speed all year) and the part due to speed control. (2 Marks)

(e) Simple payback, and one non-energy benefit of removing the gearbox. (1.5 Marks)

[5P3CEM060 | Chapter 2: Electrical Motors | 2.7 Cooling-tower fan – gearbox drive replaced by direct-drive permanent-magnet motor with VFD]

### L-2 – Model answer (10 marks)

Given: fan shaft 30 kW at 100 % speed;  $\eta_{\text{motor}}$  0.91,  $\eta_{\text{gearbox}}$  0.94 (existing);  $\eta_{\text{PMSM}}$  0.95,  $\eta_{\text{VFD}}$  0.97 (new); 8,000 h; profile 3,000/3,000/2,000 h at 100/80/60 %; ₹ 8.00/kWh; cost ₹ 26,00,000.

(a) Existing drive (2 Marks)

Formula:  $P_{\text{in}} = P_{\text{shaft}} / (\eta_{\text{gearbox}} \times \eta_{\text{motor}})$  (0.5 Mark)

Substitution:  $30 / (0.94 \times 0.91) = 30 / 0.8554$  (0.5 Mark)

Result:  **$P_{\text{in}} \approx 35.07 \text{ kW}$**  (0.5 Mark);  $\times 8,000 \text{ h} = \approx 2,80,570 \text{ kWh a year}$  (0.5 Mark)

(b) New drive at each speed (2.5 Marks)

Formula:  $P_{\text{shaft}} = 30 \times (N/N_0)^3$ ;  $P_{\text{in}} = P_{\text{shaft}} / (0.95 \times 0.97) = P_{\text{shaft}} / 0.9215$  (0.5 Mark)

Substitution: shaft power 30,  $30 \times 0.512 = 15.36$  and  $30 \times 0.216 = 6.48 \text{ kW}$  (1 Mark)

Result: **inputs  $\approx 32.56 \text{ kW}$ ,  $16.67 \text{ kW}$  and  $7.03 \text{ kW}$**  (1 Mark)

(c) New annual energy and saving (2 Marks)

Substitution:  $32.56 \times 3,000 + 16.67 \times 3,000 + 7.03 \times 2,000$  (1 Mark)

Result:  **$\approx 1,61,736 \text{ kWh a year}$ ; saving  $\approx 1,18,834 \text{ kWh} \approx ₹ 9,50,674 \text{ a year}$  ( $\approx ₹ 9.51 \times 10^5$ )** (1 Mark)

(d) Split of saving (2 Marks)

Substitution: drive train alone =  $(35.07 - 32.56) \times 8,000$  (1 Mark)

Result:  **$\approx 20,126 \text{ kWh from the drive train}$  and  **$\approx 98,709 \text{ kWh from speed control}$  – the cube law, not the motor, gives most of the saving** (1 Mark)**

(e) Payback and benefit (1.5 Marks)

Result: **simple payback  $\approx 26,00,000 / 9,50,674 \approx 2.73 \text{ years}$**  (1 Mark); no gearbox oil changes, oil leaks or gear failures, and less vibration on the fan deck (0.5 Mark)

Marking: (a)  $0.5 \times 4$ ; (b)  $0.5 + 1 + 1$ ; (c)  $1 + 1$ ; (d)  $1 + 1$ ; (e)  $1 + 0.5$ .

**L-3** A pharmaceutical tablet-compression area needs low humidity. Its AHU handles  $25,000 \text{ m}^3/\text{h}$  (density  $1.2 \text{ kg/m}^3$ , cp  $1.005 \text{ kJ/kg}\cdot\text{K}$ ): mixed air at  $27^\circ\text{C}$  is cooled to  $12^\circ\text{C}$  at the chilled-water coil to remove moisture, and electric heaters then reheat it to  $18^\circ\text{C}$  for supply. A wrap-around heat pipe is proposed: its precooling section cools the entering air from  $27^\circ\text{C}$  to  $22^\circ\text{C}$  before the coil, and its reheat section warms the off-coil air from  $12^\circ\text{C}$  to  $17^\circ\text{C}$  with no energy input. The remaining 1 K of reheat stays electric. The heat pipe adds 120 Pa of air-side pressure drop; fan efficiency 60 %. The chilled-water plant runs at 0.85 kW/TR. The AHU runs 8,000 h/year; power ₹ 8.50/kWh; heat pipe installed cost ₹ 22,00,000.

(a) Electric reheat power at present. (2 Marks)

(b) Electric reheat power with the heat pipe, and the reduction. (2 Marks)

(c) Reduction in cooling-coil load (kW and TR) and the chiller power saved. (2 Marks)

(d) Additional fan power due to the heat pipe. (1.5 Marks)

(e) Net annual energy and cost saving, and simple payback. (2.5 Marks)

[5P3CEM061 | Chapter 4: HVAC and Refrigeration System | 4.16 Pharmaceutical AHU – wrap-around heat pipe replacing electric reheat after dehumidification]

### L-3 – Model answer (10 marks)

Given:  $Q = 25,000 \text{ m}^3/\text{h}$ ;  $\rho = 1.2 \text{ kg/m}^3$ ;  $c_p = 1.005 \text{ kJ/kg}\cdot\text{K}$ ; coil  $27 \rightarrow 12 \text{ }^\circ\text{C}$ ; reheat  $12 \rightarrow 18 \text{ }^\circ\text{C}$ ; heat pipe 5 K precool and 5 K reheat;  $\Delta p$  120 Pa;  $\eta_{\text{fan}}$  0.60; 0.85 kW/TR; 8,000 h; ₹ 8.50/kWh; cost ₹ 22,00,000.

(a) Present reheat (2 Marks)

Formula:  $\dot{m} = Q\rho/3,600$ ;  $P = \dot{m} c_p \Delta T$  (0.5 Mark)

Substitution:  $\dot{m} = 25,000 \times 1.2 / 3,600 = 8.333 \text{ kg/s}$ ;  $P = 8.333 \times 1.005 \times 6$  (1 Mark)

Result: **reheat  $\approx 50.25 \text{ kW electric}$**  (0.5 Mark)

(b) Reheat with heat pipe (2 Marks)

Substitution:  $8.333 \times 1.005 \times 1$  (1 Mark)

Result:  **$\approx 8.38 \text{ kW}$ ; reduction  $\approx 41.88 \text{ kW}$**  (1 Mark)

(c) Coil load and chiller saving (2 Marks)

Substitution: precool =  $8.333 \times 1.005 \times 5 = 41.88 \text{ kW}$ ;  $\div 3.516$  (1 Mark)

Result: **coil load falls by  $\approx 11.91 \text{ TR}$  ( $\approx 41.9 \text{ kW}$ ); chiller saving  $\approx 11.91 \times 0.85 \approx 10.12 \text{ kW}$**  – the same sensible heat that once needed electric reheat is now taken from the entering air (1 Mark)

(d) Fan penalty (1.5 Marks)

Formula:  $\Delta P_{\text{fan}} = Q \times \Delta p / \eta_{\text{fan}}$  (0.5 Mark)

Substitution:  $(25,000 / 3,600) \times 120 / 0.60 = 6.944 \times 120 / 0.60$  (0.5 Mark)

Result:  **$\approx 1.39 \text{ kW}$**  (0.5 Mark)

(e) Net saving and payback (2.5 Marks)

Substitution:  $41.88 + 10.12 - 1.39 = 50.61 \text{ kW}$ ;  $\times 8,000 \text{ h}$  (1 Mark)

Result:  **$\approx 4,04,876 \text{ kWh a year} \approx ₹ 34,41,444$  ( $\approx ₹ 3.44 \times 10^6$ )** (1 Mark)

Result: **simple payback  $\approx 22,00,000 / 34,41,444 \approx 0.64 \text{ year}$** ; dew point and supply humidity stay unchanged because the coil leaving temperature is still  $12 \text{ }^\circ\text{C}$  (0.5 Mark)

Marking: (a) 0.5 + 1 + 0.5; (b) 1 + 1; (c) 1 + 1; (d) 0.5  $\times$  3; (e) 1 + 1 + 0.5.

**L-4** A 10-stage mine dewatering pump at the 480 m level delivers  $60 \text{ m}^3/\text{h}$  of clear sump water at  $30 \text{ }^\circ\text{C}$  (density  $1,000 \text{ kg/m}^3$ ). At this flow each stage develops 60 m, so the pump produces 600 m. The system needs only 520 m at the pump discharge (480 m static lift, rising-main friction plus a minimum valve drop), so a discharge valve throttles away the rest. The pump efficiency is 68 % and the motor efficiency 94 %. Two options are considered: A – remove one impeller stage (destaging), giving 540 m at the same flow with unchanged efficiency; B – fit a VFD (efficiency 97 %) and slow the pump until it delivers exactly 520 m. The pump runs 8,000 h/year; power ₹ 7.50/kWh.

(a) Hydraulic power and motor input at present. (2.5 Marks)

(b) Motor input and annual saving with option A. (2.5 Marks)

(c) Speed ratio and motor input with option B (assume efficiency unchanged at the new speed). (2.5 Marks)

(d) Compare A and B and recommend, with one technical caution for destaging. (2.5 Marks)

[5P3CEM062 | Chapter 6: Pumps and Pumping System | 6.6 Mine dewatering pump – removing an impeller stage to eliminate valve throttling]

#### **L-4 – Model answer (10 marks)**

Given:  $Q = 60 \text{ m}^3/\text{h} = 0.01667 \text{ m}^3/\text{s}$ ;  $\rho = 1,000 \text{ kg/m}^3$ ;  $H = 600 \text{ m}$  (60 m per stage); required 520 m;  $\eta_{\text{pump}}$  0.68;  $\eta_m$  0.94;  $\eta_{\text{VFD}}$  0.97; 8,000 h; ₹ 7.50/kWh.

(a) Present (2.5 Marks)

Formula:  $P_h = \rho g Q H$ ; input =  $P_h / (\eta_{\text{pump}} \times \eta_m)$  (0.5 Mark)

Substitution:  $1,000 \times 9.81 \times 0.01667 \times 600$  (0.5 Mark)

Result: **P<sub>h</sub> ≈ 98.10 kW** (0.5 Mark); input =  $98.10 / (0.68 \times 0.94) \approx \mathbf{153.47 \text{ kW}}$  (0.5 Mark). About 80 m ( $\approx 20.5 \text{ kW}$  of motor input) is burnt across the control valve (0.5 Mark).

(b) Option A – destaging (2.5 Marks)

Substitution:  $H = 9 \times 60 = 540 \text{ m}$ ; input =  $1,000 \times 9.81 \times 0.01667 \times 540 / (0.68 \times 0.94)$  (1 Mark)

Result: **input ≈ 138.13 kW; saving ≈ 15.35 kW** (0.5 Mark);  $\times 8,000 \text{ h} \approx \mathbf{1,22,778 \text{ kWh} \approx ₹ 9,20,839 \text{ a year}}$  (1 Mark)

(c) Option B – VFD (2.5 Marks)

Formula: at constant flow,  $H \propto N^2$  approximately;  $N_2/N_1 = \sqrt{(520/600)}$  (0.5 Mark)

Result: **speed ratio ≈ 0.931** (0.5 Mark)

Substitution: input =  $1,000 \times 9.81 \times 0.01667 \times 520 / (0.68 \times 0.94 \times 0.97)$  (1 Mark)

Result: **input ≈ 137.12 kW; saving ≈ 16.35 kW ( $\approx 1,30,795 \text{ kWh a year}$ )** (0.5 Mark)

(d) Comparison (2.5 Marks)

Option B saves only about 1 kW more than A because the drive's own 3 % loss eats most of the extra head reduction (0.5 Mark). Destaging costs little (a blank stage or dummy impeller during the next overhaul) while a medium-voltage VFD costs far more, so A is preferred where the inflow to the sump is steady (0.5 Mark); B is better if inflow varies widely, since the pump can then also track sump level (0.5 Mark). Caution: keep enough head margin for upset conditions (rising-main friction at maximum flow and any plan to deepen the sump), and have the maker check rotor balance and axial thrust after removing a stage (1 Mark).

Marking: (a)  $0.5 \times 5$ ; (b)  $1 + 0.5 + 1$ ; (c)  $0.5 + 0.5 + 1 + 0.5$ ; (d)  $0.5 + 0.5 + 0.5 + 1$ .

**L-5** A 400 kW boiler ID fan absorbs 360 kW of shaft power at full speed. Its speed is controlled by a hydrodynamic fluid coupling. The system resistance is almost entirely friction (system curve through the origin), so fan flow is proportional to fan speed and fan shaft power varies with the cube of speed at constant fan efficiency. Operating profile: 75 % flow for 6,000 h and 60 % flow for 2,000 h a year. In a fluid coupling the output torque equals the input torque, so input power = output power / (output speed ratio). Motor efficiency 95 %; VFD efficiency 97 %; power ₹ 7.00/kWh.

(a) Fan shaft power at 75 % and 60 % flow. (2 Marks)

(b) Motor input with the fluid coupling at each condition, and the slip loss at 75 % flow. (3 Marks)

(c) Motor input with a VFD at each condition. (2 Marks)

(d) Annual energy with each option and the saving from replacing the coupling by a VFD. (2 Marks)

(e) Where is the coupling's slip loss rejected, and what does this imply for maintenance? (1 Mark)

*[5P3CEM063 | Chapter 5: Fans and Blowers | 5.5 Boiler ID fan – fluid coupling slip losses compared with a VFD]*

#### **L-5 – Model answer (10 marks)**

Given: full-speed shaft power 360 kW; flows 0.75 (6,000 h) and 0.60 (2,000 h); coupling  $P_{in} = P_{out} / \text{speed ratio}$ ;  $\eta_m$  0.95;  $\eta_{VFD}$  0.97; ₹ 7.00/kWh.

(a) Fan shaft power (2 Marks)

Formula:  $P = 360 \times (N/N_o)^3$  (0.5 Mark)

Substitution:  $360 \times 0.75^3$ ;  $360 \times 0.60^3$  (0.5 Mark)

Result:  **$\approx 151.9 \text{ kW at 75 \%}$  and  **$\approx 77.8 \text{ kW at 60 \%}$**  (1 Mark)**

(b) Fluid coupling (3 Marks)

Substitution: coupling input =  $151.9 / 0.75 = 202.5 \text{ kW}$ ;  $77.76 / 0.60 = 129.6 \text{ kW}$  (1 Mark)

Result: **motor input**  $\approx 202.5 / 0.95 \approx 213.2 \text{ kW at } 75 \%$ ;  $129.6 / 0.95 \approx 136.4 \text{ kW at } 60 \%$  (1 Mark)

Result: **slip loss at 75 %**  $= 202.5 - 151.9 \approx 50.6 \text{ kW}$  (1 Mark)

(c) VFD (2 Marks)

Formula: motor-plus-drive input = shaft power / ( $\eta_m \times \eta_{\text{VFD}}$ ) (0.5 Mark)

Substitution:  $151.9 / (0.95 \times 0.97)$ ;  $77.76 / (0.95 \times 0.97)$  (0.5 Mark)

Result:  $\approx 164.8 \text{ kW at } 75 \%$  and  $\approx 84.4 \text{ kW at } 60 \%$  (1 Mark)

(d) Annual energy (2 Marks)

Coupling:  $213.2 \times 6,000 + 136.4 \times 2,000 = \approx 15,51,789 \text{ kWh}$  (0.5 Mark)

VFD:  $164.8 \times 6,000 + 84.4 \times 2,000 = \approx 11,57,645 \text{ kWh}$  (0.5 Mark)

Result: **saving  $\approx 3,94,144 \text{ kWh a year} \approx ₹ 27,59,010 (\approx ₹ 2.76 \times 10^6)$**  (1 Mark)

(e) Slip loss (1 Mark)

The slip power appears as heat in the coupling's working oil and is removed by its oil cooler into the cooling-water circuit. Cooler fouling or low oil level raises oil temperature and can trip the fan, so the cooler's heat duty (about 50 kW here) and oil condition must be maintained. A VFD removes this loss and the cooler altogether. (1 Mark)

Marking: (a)  $0.5 + 0.5 + 1$ ; (b)  $1 + 1 + 1$ ; (c)  $0.5 + 0.5 + 1$ ; (d)  $0.5 + 0.5 + 1$ ; (e) 1.

**L-6** A chemical plant uses 12 air-operated double-diaphragm (AODD) pumps for liquid transfer. Each consumes  $0.9 \text{ m}^3/\text{min}$  of free air at 6 bar(g) when running and runs 60 % of the time. Each pump delivers  $4 \text{ m}^3/\text{h}$  against 25 m head (liquid density  $1,000 \text{ kg}/\text{m}^3$ ). Compressed air is generated at a specific power of  $7.0 \text{ kW per m}^3/\text{min}$ . Electric replacements of overall wire-to-water efficiency 40 % (motor, drive and pump) would run at the same duty. Annual period 5,000 h; power ₹ 8.00/kWh.

(a) Average compressed-air demand of the AODD pumps and the compressor power it requires. (2 Marks)

(b) Hydraulic power of one pump and the overall wire-to-water efficiency of the AODD system. (2 Marks)

(c) Average electrical power of the electric replacements. (2 Marks)

(d) Annual energy and cost saving. (2 Marks)

(e) Give two operating reasons why AODD pumps waste even more air in practice than this estimate, and one reason a plant may keep a few of them. (2 Marks)

[5P3CEM064 | Chapter 3: Compressed Air System | 3.7 Air-operated double-diaphragm pumps replaced by electric pumps – wire-to-water comparison]

#### **L-6 – Model answer (10 marks)**

Given: 12 pumps;  $0.9 \text{ m}^3/\text{min}$  each; 60 % running;  $4 \text{ m}^3/\text{h}$  at 25 m;  $7.0 \text{ kW per m}^3/\text{min}$ ; electric overall  $\eta$  0.40; 5,000 h; ₹ 8.00/kWh.

(a) Air demand and compressor power (2 Marks)

Substitution:  $12 \times 0.9 \times 0.60$  (0.5 Mark)

Result:  $\approx 6.48 \text{ m}^3/\text{min of free air}$  (0.5 Mark); compressor power  $= 6.48 \times 7.0 = \approx 45.36 \text{ kW}$  (1 Mark)

(b) Hydraulic power and efficiency (2 Marks)

Formula:  $P_h = \rho g Q H$  (0.5 Mark)

Substitution:  $1,000 \times 9.81 \times (4/3,600) \times 25 = 272.5 \text{ W per pump}$ ; average for all  $= 0.2725 \times 12 \times 0.60 = 1.962 \text{ kW}$  (0.5 Mark)

Result:  **$P_h \approx 0.2725 \text{ kW per pump}$ ; wire-to-water efficiency  $\approx 1.962 / 45.36 \approx 4.3 \%$**  (1 Mark)

(c) Electric replacements (2 Marks)

Substitution:  $0.2725 / 0.40 = 0.681$  kW per pump;  $\times 12 \times 0.60$  (1 Mark)

Result:  $\approx$  **4.91 kW average** (1 Mark)

(d) Saving (2 Marks)

Substitution:  $45.36 - 4.91 = 40.46$  kW;  $\times 5,000$  h (1 Mark)

Result:  $\approx$  **2,02,275 kWh a year  $\approx$  ₹ 16,18,200 ( $\approx$  ₹  $1.62 \times 10^6$ )** (1 Mark)

(e) Discussion (2 Marks)

Extra waste in practice: AODD pumps keep cycling and consuming air against a closed discharge or when run dry, and they are usually supplied at full header pressure without a regulator, so air use per stroke is higher than needed (0.5 Mark each, 1 Mark). A plant may keep some AODD pumps for intrinsically safe duty in flammable areas, for shear-sensitive or solids-laden liquids, or where dry running must be tolerated (1 Mark).

Marking: (a)  $0.5 + 0.5 + 1$ ; (b)  $0.5 + 0.5 + 1$ ; (c)  $1 + 1$ ; (d)  $1 + 1$ ; (e)  $1 + 1$ .

..... End of Section III .....

## Section I – Answer key

| Q  | Ans | Q  | Ans | Q  | Ans | Q  | Ans | Q  | Ans |
|----|-----|----|-----|----|-----|----|-----|----|-----|
| 1  | a   | 11 | d   | 21 | c   | 31 | b   | 41 | c   |
| 2  | c   | 12 | b   | 22 | c   | 32 | c   | 42 | a   |
| 3  | c   | 13 | d   | 23 | d   | 33 | b   | 43 | b   |
| 4  | a   | 14 | b   | 24 | a   | 34 | d   | 44 | c   |
| 5  | c   | 15 | b   | 25 | a   | 35 | d   | 45 | c   |
| 6  | b   | 16 | d   | 26 | b   | 36 | c   | 46 | d   |
| 7  | a   | 17 | a   | 27 | a   | 37 | b   | 47 | b   |
| 8  | d   | 18 | b   | 28 | d   | 38 | a   | 48 | a   |
| 9  | a   | 19 | d   | 29 | a   | 39 | d   | 49 | b   |
| 10 | b   | 20 | a   | 30 | c   | 40 | c   | 50 | d   |