

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

Independent practice paper prepared by Beyond Smart Cities / Beyond Xlearn (learn.beyondsmartcities.in). Not affiliated with, endorsed by or issued by the Bureau of Energy Efficiency (BEE) or the National Productivity Council (NPC). All questions are original and written to the pattern of the National Certification Examination; the official BEE guidebook (Book 3 – Energy Efficiency in Electrical Utilities) remains the authority.

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 Chilled water at 7 °C flows in an insulated pipe through a non-air-conditioned plant room at 34 °C DBT, where the dew point of the air is 26 °C. Which condition prevents condensation and wet insulation on this pipe?

[4P3CEM001 | Chapter 4: HVAC and Refrigeration System | 4.2 Chilled water piping – insulation and dew point]

- a) Pipe metal kept above 34 °C, so that no heat at all flows inwards
- b) Insulation thickness chosen only to limit heat gain to under 5 %
- c) Insulation outer surface kept above 26 °C, with an intact vapour barrier**
- d) Insulation outer surface allowed to fall to 7 °C, matching the water

Answer & explanation

Correct answer: (c).

Moisture condenses on any surface colder than the dew point of the surrounding air, here 26 °C. The insulation must be thick enough that its outer surface temperature stays above 26 °C, with a margin, under worst humid conditions. It also needs a continuous vapour barrier, because water vapour that diffuses into open-cell or damaged insulation condenses inside it. Wet insulation loses most of its insulating value, so heat gain increases and the chiller works harder.

Why the others are wrong: (a) the pipe carries 7 °C water, so its metal can never be above 34 °C; (b) a heat-gain limit alone may give too thin a layer in humid rooms; (d) a surface at 7 °C would be far below the 26 °C dew point and would drip continuously. During audits, sweating pipes, wet lagging and corrosion under insulation indicate both energy loss and a maintenance problem. Closed-cell insulation with sealed joints is the usual remedy.

Ref: BEE Guidebook Paper 3, Ch. 4 (Psychrometrics; HVAC energy conservation – insulation).

2 A 75 kW, 415 V motor has a full-load current of 132 A and a no-load (magnetising) current of 38 A. On a lightly loaded fan it draws 58 A. Treating the no-load current as reactive and in quadrature with the load component, its approximate loading is:

[4P3CEM002 | Chapter 2: Electrical Motors | 2.5 Motor loading from current – correcting for magnetising current]

- a) 43.9 %
- b) 21.3 %
- c) 33.2 %
- d) 34.7 %**

Answer & explanation

Correct answer: (d) $\approx$ 34.7 %.

Given: $I_{FL} = 132$ A; I_o (magnetising) = 38 A; measured $I = 58$ A.

Formula: load component $I_w = \sqrt{I^2 - I_o^2}$; loading $\approx I_w(\text{measured}) / I_w(\text{full load})$.

Substitution: $I_w = \sqrt{58^2 - 38^2} = \sqrt{1,920} = 43.8$ A; $I_{w,FL} = \sqrt{132^2 - 38^2} = \sqrt{15,980} = 126.4$ A; loading = $43.8 / 126.4$.

Result: **loading $\approx$ 34.7 %**.

Why the others are wrong: (a) takes the simple current ratio 58/132, which overstates light-load loading because the magnetising current flows at every load; (b) subtracts the currents arithmetically, but they are in quadrature and must be combined as phasors; (c) corrects the measured current but divides by the uncorrected full-load current. For an accurate load survey, measure input kW with a power analyser and compare it with rated output divided by efficiency. The current method is acceptable only near full load, where the magnetising share is small.

Ref: BEE Guidebook Paper 3, Ch. 2 (Motor load survey; assessment of motor loading).

3 A plant needs 320 TR ($\approx$ 1,125 kW). It can run one 400 TR ($\approx$ 1,407 kW) chiller at 80 % load (0.58 kW/TR at that load) or two identical 400 TR ($\approx$ 1,407 kW) chillers at 40 % load each (0.71 kW/TR at that load); the second option also runs an extra 18.5 kW primary pump. Over 4,000 h a year, the energy penalty of running two chillers is about:

[4P3CEM003 | Chapter 4: HVAC and Refrigeration System | 4.8 Chiller plant sequencing – one machine loaded versus two at part load]

- a) 1,66,400 kWh
- b) 3,14,400 kWh
- c) 2,40,400 kWh**
- d) 1,57,200 kWh

Answer & explanation

Correct answer: (c) $\approx$ 2,40,400 kWh.

Given: load 320 TR ($\approx$ 1,125 kW); one chiller at 0.58 kW/TR; two chillers at 0.71 kW/TR; extra pump 18.5 kW; 4,000 h.

Formula: chiller kW = TR $\times$ kW/TR; penalty = (two-chiller kW + extra pump – one-chiller kW) $\times$ hours.

Substitution: one: $320 \times 0.58 = 185.6$ kW; two: $320 \times 0.71 = 227.2$ kW + 18.5 = 245.7 kW; $\Delta = 60.1$ kW $\times$ 4,000.

Result: **penalty $\approx$ 2,40,400 kWh a year**.

Why the others are wrong: (a) leaves out the extra primary pump that the second chiller brings; (b) adds two pumps, although one pump already runs with the single chiller; (d) applies the kW/TR penalty to only one chiller's 160 TR, but both chillers run at the poorer 0.71 kW/TR. Chiller sequencing should follow measured part-load

curves. Some VFD centrifugal chillers are most efficient at 40–60 % load with low condenser water temperature, so the right rule comes from data, not habit.

Ref: BEE Guidebook Paper 3, Ch. 4 (HVAC – part-load performance and energy saving opportunities).

4 Two 1,000 kVA distribution transformers are compared on total owning cost, $TOC = \text{price} + A \times \text{no-load loss} + B \times \text{full-load load loss}$, with $A = ₹ 3,00,000$ per kW and $B = ₹ 1,00,000$ per kW. Unit X costs ₹ 9,20,000 (losses 1.10 kW no-load, 9.5 kW load); unit Y costs ₹ 11,10,000 (0.60 kW and 8.2 kW). Which has the lower TOC, and by about how much?

[4P3CEM004 | Chapter 1: Electrical Systems | 1.5 Transformer selection – total owning cost with loss capitalisation]

- a) X, lower by about ₹ 1,90,000
- b) Y, lower by about ₹ 2,50,000
- c) Y, lower by about ₹ 2,80,000
- d) Y, lower by about ₹ 90,000**

Answer & explanation

Correct answer: (d).

Given: $A = ₹ 3,00,000/\text{kW}$ (no-load loss); $B = ₹ 1,00,000/\text{kW}$ (load loss); X: ₹ 9,20,000, 1.10 kW, 9.5 kW; Y: ₹ 11,10,000, 0.60 kW, 8.2 kW.

Formula: $TOC = \text{price} + A \times P_o + B \times P_c$.

Substitution: $X = 9,20,000 + 3,30,000 + 9,50,000 = ₹ 22,00,000$; $Y = 11,10,000 + 1,80,000 + 8,20,000 = ₹ 21,10,000$.

Result: **Y has the lower TOC, by about ₹ 90,000.**

Why the others are wrong: (a) compares purchase prices only, which is the classic lowest-bid error; (b) swaps A and B, but the no-load loss carries the higher factor because it occurs for all 8,760 hours whereas load loss scales with the square of the load; (c) compares capitalised losses but forgets the purchase price. The factors A and B are the present worth of one kW of loss over the transformer's life, built from the energy rate, the loss load factor and the discount rate.

Ref: BEE Guidebook Paper 3, Ch. 1 (Transformers – losses and selection).

5 A DG set runs at 700 kW with a specific fuel consumption of 0.22 kg/kWh and an overall air–fuel ratio of 30:1 by mass. With intake air density 1.15 kg/m³, the combustion air that the DG room openings must admit, in addition to ventilation air for heat removal, is about:

[4P3CEM005 | Chapter 9: Diesel / Natural Gas Power Generating System | 9.3 DG room design – combustion air requirement]

- a) 4,020 m³/h**
- b) 1,940 m³/h
- c) 5,310 m³/h
- d) 4,620 m³/h

Answer & explanation

Correct answer: (a) $\approx 4,020 \text{ m}^3/\text{h}$.

Given: output 700 kW; SFC 0.22 kg/kWh; air–fuel ratio 30 (mass); air density 1.15 kg/m³.

Formula: $\text{fuel} = \text{kW} \times \text{SFC}$; $\text{air mass} = \text{fuel} \times A/F$; $\text{air volume} = \text{mass} / \text{density}$.

Substitution: $\text{fuel} = 700 \times 0.22 = 154 \text{ kg/h}$; $\text{air} = 154 \times 30 = 4,620 \text{ kg/h}$; $4,620 / 1.15$.

Result: **combustion air $\approx 4,020 \text{ m}^3/\text{h}$.**

Why the others are wrong: (b) uses the stoichiometric ratio of about 14.5:1, but diesel engines run with large excess air, so this would starve the engine; (c) multiplies by density instead of dividing; (d) reports the mass flow

in kg/h as if it were a volume. Ventilation air to carry away radiated heat from the engine and alternator is several times larger than the combustion air. A hot, poorly ventilated DG room feeds warm, thin air to the turbocharger, which raises fuel consumption and exhaust temperature and can force derating.

Ref: BEE Guidebook Paper 3, Ch. 9 (DG systems – installation and operating practices).

6 A 90 kW motor has a full-load efficiency of 94.0 %. Of its full-load losses, 1.90 kW are fixed (core plus friction and windage); the rest vary with the square of load. Its efficiency when delivering 22.5 kW (25 % load) is about:

[4P3CEM006 | Chapter 2: Electrical Motors | 2.4 Part-load efficiency from fixed and variable losses]

- a) 98.4 %
- b) 94.0 %
- c) 79.7 %
- d) 91.3 %**

Answer & explanation

Correct answer: (d) $\approx$ 91.3 %.

Given: output 90 kW at $\eta = 0.94$; fixed loss 1.90 kW; variable loss $\propto \text{load}^2$; part-load output 22.5 kW ($x = 0.25$).

Formula: full-load losses = output / η – output; part-load losses = fixed + variable $\times x^2$; η = output / (output + losses).

Substitution: input = $90 / 0.94 = 95.745$ kW, losses = 5.745 kW, variable = 3.845 kW; part-load losses = $1.90 + 3.845 \times 0.0625 = 2.140$ kW; $\eta = 22.5 / 24.64$.

Result: **efficiency $\approx$ 91.3 %**.

Why the others are wrong: (a) treats all losses as varying with load², including core loss, which actually stays constant; (b) scales all losses linearly, which leaves efficiency unchanged; (c) keeps the full-load losses constant. Because fixed losses dominate at light load, efficiency falls steeply below about 40 % load, and power factor falls even faster. That is why the guidebook recommends downsizing or operating in star for chronically under-loaded motors.

Ref: BEE Guidebook Paper 3, Ch. 2 (Motor losses; efficiency at part load).

7 Consumers in a state need 850 MU a year at their meters. Transmission and distribution losses are 15 % of the energy sent out from power stations, and station auxiliary consumption is 8 % of gross generation. The gross generation required is about:

[4P3CEM007 | Chapter 1: Electrical Systems | 1.1 Power system – generation needed for a given delivered energy]

- a) 1,087 MU**
- b) 1,046 MU
- c) 1,056 MU
- d) 1,104 MU

Answer & explanation

Correct answer: (a) $\approx$ 1,087 MU.

Given: energy at consumer meters = 850 MU; T&D loss = 15 % of energy sent out; auxiliary consumption = 8 % of gross generation.

Formula: energy sent out = delivered / (1 – T&D loss fraction); gross generation = sent out / (1 – auxiliary fraction).

Substitution: sent out = $850 / 0.85 = 1,000$ MU; gross = $1,000 / 0.92$.

Result: **gross generation $\approx$ 1,087 MU**.

Why the others are wrong: (b) adds the two percentages (850×1.23), but losses are fractions of different bases; (c) multiplies by 1.15 and 1.08, applying each loss to the smaller downstream quantity; (d) divides by $(1 - 0.23)$, which again adds percentages that belong to different stages. Each stage efficiency must be applied in sequence: overall efficiency = $0.92 \times 0.85 = 78.2\%$. Every unit saved at the consumer end therefore saves about 1.28 units of generation, along with the fuel and emissions behind it.

Ref: BEE Guidebook Paper 3, Ch. 1 (Electrical system – generation, T&D losses).

8 A data-centre chiller plant carries an average 500 TR ($\approx 1,758$ kW) with compressors at 0.60 kW/TR. A plate heat exchanger allows water-side free cooling: when the wet bulb is low enough, the tower carries the full load with the chillers off, but tower fans draw 30 kW more and a heat-exchanger pump adds 15 kW. If such hours total 1,800 a year, the annual saving is about:

[4P3CEM008 | Chapter 7: Cooling Towers | 7.5 Water-side free cooling with the cooling tower]

- a) 5,40,000 kWh
- b) 4,86,000 kWh
- c) 5,13,000 kWh
- d) 4,59,000 kWh**

Answer & explanation

Correct answer: (d) $\approx 4,59,000$ kWh.

Given: load 500 TR ($\approx 1,758$ kW); compressor 0.60 kW/TR; extra tower fan 30 kW; extra heat-exchanger pump 15 kW; 1,800 free-cooling hours.

Formula: saving = $(\text{TR} \times \text{kW/TR} - \text{added fan} - \text{added pump}) \times \text{hours}$.

Substitution: compressor power avoided = $500 \times 0.60 = 300$ kW; net = $300 - 30 - 15 = 255$ kW; $\times 1,800$.

Result: **net saving $\approx 4,59,000$ kWh a year.**

Why the others are wrong: (a) counts the avoided compressor power but ignores both added loads; (b) deducts the fan power but forgets the heat-exchanger pump; (c) deducts the pump but forgets that the tower fans must run harder to reach the colder water needed. Free cooling works only when the wet bulb plus the tower approach and the heat-exchanger approach stays below the chilled water supply temperature. Raising the supply set point, where the load allows, therefore extends the free-cooling hours.

Ref: BEE Guidebook Paper 3, Ch. 7 (Cooling towers – energy saving opportunities).

9 A 6-pole, 50 Hz slip-ring motor drives a constant-torque kiln feeder. To slow the feeder, external rotor resistance is added so that the motor runs at 750 rpm while delivering 60 kW at the shaft. The power dissipated in the rotor circuit at this speed is about:

[4P3CEM009 | Chapter 2: Electrical Motors | 2.9 Speed control by rotor resistance – slip power loss]

- a) 15.0 kW
- b) 18.4 kW
- c) 18.8 kW
- d) 20.0 kW**

Answer & explanation

Correct answer: (d) ≈ 20.0 kW.

Given: synchronous speed $N_s = 120 f / P = 120 \times 50 / 6 = 1,000$ rpm; running speed 750 rpm; shaft output 60 kW (ignore friction).

Formula: slip $s = (N_s - N) / N_s$; air-gap power $P_{ag} = P_{out} / (1 - s)$; rotor copper (slip) loss = $s \times P_{ag}$.

Substitution: $s = 250 / 1,000 = 0.25$; $P_{ag} = 60 / 0.75 = 80 \text{ kW}$; $\text{loss} = 0.25 \times 80$.

Result: **slip power $\approx 20.0 \text{ kW}$, mostly burnt in the external resistors.**

Why the others are wrong: (a) multiplies slip by the output instead of the air-gap power; (b) measures slip from the 980 rpm full-load speed rather than synchronous speed; (c) estimates air-gap power as $\text{output} \times (1 + s) = 75 \text{ kW}$ instead of $\text{output} \div (1 - s)$. Rotor-resistance control wastes power in proportion to the speed reduction. A VFD, or slip-power recovery on large machines, avoids most of this loss.

Ref: BEE Guidebook Paper 3, Ch. 2 (Speed control of induction motors).

10 A packaging line has 12 double-acting cylinders, each 63 mm bore and 250 mm stroke (rod volume neglected), each making 15 complete cycles per minute at 6 bar(g); atmospheric pressure is 1.013 bar. The free air consumed and the compressor power it represents (6.5 kW per m^3/min) are about:

[4P3CEM010 | Chapter 3: Compressed Air System | 3.6 End-use air demand – double-acting cylinders]

a) $1.66 \text{ m}^3/\text{min}$ and 10.8 kW

b) $1.94 \text{ m}^3/\text{min}$ and 12.6 kW

c) $0.97 \text{ m}^3/\text{min}$ and 6.3 kW

d) $2.22 \text{ m}^3/\text{min}$ and 14.4 kW

Answer & explanation

Correct answer: (b).

Given: 12 cylinders; bore 0.063 m ; stroke 0.25 m ; 15 cycles/min; 2 strokes per cycle; 6 bar(g) = 7.013 bar(a); atmosphere 1.013 bar; 6.5 kW per m^3/min .

Formula: $\text{free air} = \text{swept volume} \times \text{strokes/min} \times \text{cylinders} \times (P_{abs} / P_{atm})$.

Substitution: $\text{swept volume} = \pi/4 \times 0.063^2 \times 0.25 = 0.000779 \text{ m}^3$; $\times 2 \times 15 \times 12 = 0.2806 \text{ m}^3/\text{min}$; $\times 7.013/1.013 = 6.923$.

Result: **$\approx 1.94 \text{ m}^3/\text{min}$ of free air, i.e. $\approx 12.6 \text{ kW}$ of compressor power.**

Why the others are wrong: (a) uses the gauge ratio 6/1.013 and so leaves out the atmospheric air already in the cylinder; (c) counts one stroke per cycle, as for a single-acting cylinder; (d) adds an extra 1 to the absolute pressure ratio, double-counting the air at atmospheric pressure. Since air demand scales with absolute pressure, regulating each machine to the lowest pressure that does the job, and using smaller bores where force allows, cuts consumption directly.

Ref: BEE Guidebook Paper 3, Ch. 3 (Compressed air – end use and efficient operation).

11 An AHU supplies $12,000 \text{ m}^3/\text{h}$ of air (density 1.2 kg/m^3). It mixes 25 % outdoor air at 36°C with 75 % return air at 25°C , and the coil cools the mixture to 13°C with no moisture removed. Taking $c_p = 1.005 \text{ kJ/kg}\cdot\text{K}$, the coil's sensible load is about:

[4P3CEM011 | Chapter 4: HVAC and Refrigeration System | 4.2 Psychrometrics – mixed-air temperature and coil sensible load]

a) 48.2 kW

b) 59.3 kW

c) 92.5 kW

d) 70.4 kW

Answer & explanation

Correct answer: (b) $\approx 59.3 \text{ kW}$.

Given: $12,000 \text{ m}^3/\text{h} \times 1.2 / 3,600 = 4.0 \text{ kg/s}$; 25 % outdoor air at 36°C ; 75 % return air at 25°C ; off-coil 13°C .

Formula: mixed temperature $t_m = \Sigma(\text{fraction} \times t)$ (mass-weighted); sensible load = $m \times c_p \times (t_m - t_{off})$.

Substitution: $t_m = 0.25 \times 36 + 0.75 \times 25 = 27.75 \text{ }^\circ\text{C}$; load = $4.0 \times 1.005 \times (27.75 - 13)$.

Result: **sensible coil load $\approx 59.3 \text{ kW}$ ($\approx 16.9 \text{ TR}$).**

Why the others are wrong: (a) uses the return-air temperature and leaves out the warmer outdoor air; (c) assumes the whole flow is outdoor air at $36 \text{ }^\circ\text{C}$; (d) takes a simple average of 36 and $25 \text{ }^\circ\text{C}$, ignoring the 25/75 split. Each extra 10 % of outdoor air adds about 4.4 kW of sensible load, so fresh air should follow occupancy.

Ref: BEE Guidebook Paper 3, Ch. 4 (Psychrometrics – mixing of air streams; cooling load).

12 Four screw compressors share a header. Each has its own pressure switch, set in a cascade so that each machine loads at a pressure 0.3 bar lower than the one before. As a result, header pressure swings between 6.6 and 7.8 bar(g), although users need only 6.2 bar(g). The main energy penalty and its best remedy are:

[4P3CEM012 | Chapter 3: Compressed Air System | 3.5 Multiple compressors – cascade pressure-switch settings versus master control]

a) High average pressure and part-loaded units; use a master sequencer

b) Too few running hours on each unit; rotate the lead machine weekly

c) Excess dryer purge at low pressure; switch to a heatless dryer

d) Low intake air density; duct cold outside air to all four inlets

Answer & explanation

Correct answer: (a).

With a pressure-switch cascade, each added machine needs its own pressure band, so the average header pressure rises well above what users need. Several units also tend to run part-loaded or unloaded at the same time. As a rule of thumb, every 1 bar of excess generation pressure costs about 6–10 % in power, and higher pressure also increases leakage and unregulated use. A master sequencer (central controller) holds the header in a narrow band just above the 6.2 bar(g) needed, runs base machines fully loaded and trims with one unit, ideally a VSD.

Why the others are wrong: (b) rotating the lead machine evens out wear but does not lower the pressure band; (c) heatless dryers use more purge air, not less, and dryer purge is not the issue described; (d) cool intake air helps, but it does not address the main waste, which is the wide, high pressure band.

Ref: BEE Guidebook Paper 3, Ch. 3 (Capacity control; efficient operation of compressed air systems).

13 A 6-pulse VFD feeder draws 180 kW. A power analyser shows a displacement (fundamental) power factor of 0.96 and a current THD of 45 %; voltage distortion is negligible. The true (total) power factor and the apparent power drawn are about:

[4P3CEM013 | Chapter 1: Electrical Systems | 1.10 Harmonics – true power factor of a distorted current]

a) 0.960 and 187.5 kVA

b) 0.528 and 340.9 kVA

c) 0.662 and 271.9 kVA

d) 0.875 and 205.6 kVA

Answer & explanation

Correct answer: (d).

Given: $P = 180 \text{ kW}$; displacement PF (DPF) = 0.96; current THD = 0.45; sinusoidal voltage.

Formula: distortion factor = $1 / \sqrt{1 + \text{THD}^2}$; true PF = DPF $\times$ distortion factor; kVA = kW / true PF.

Substitution: distortion factor = $1 / \sqrt{1 + 0.2025} = 1 / 1.0966 = 0.912$; true PF = $0.96 \times 0.912 = 0.875$; kVA = $180 / 0.875$.

Result: **true PF ≈ 0.875 and apparent power $\approx 205.6 \text{ kVA}$.**

Why the others are wrong: (a) uses only the displacement PF read by a conventional PF meter and ignores the harmonic currents; (b) multiplies DPF by $(1 - \text{THD})$, treating THD as a direct fractional loss of PF; (c) divides DPF by $(1 + \text{THD})$ instead of by the rms factor $\sqrt{1 + \text{THD}^2}$. The practical lesson is that capacitors raise only the displacement PF. On a kVAh-billed feeder with large drives, harmonic filters or multi-pulse or active front-end drives are needed to recover the distortion part.

Ref: BEE Guidebook Paper 3, Ch. 1 (Harmonics; power factor).

14 A 120 TR (≈ 422 kW) screw chiller serving a small process load runs for long hours at about 30 % load using hot gas bypass to stop it cycling. From an energy standpoint, this arrangement:

[4P3CEM014 | Chapter 4: HVAC and Refrigeration System | 4.8 Capacity control – hot gas bypass]

- a) Keeps the compressor near full power while delivering only part of the cooling**
- b) Is efficient, as recirculated hot gas reduces compressor work per TR
- c) Saves power, as the bypass lowers the condensing pressure at part load
- d) Equals a VFD, as the compressor slows down when hot gas is bypassed

Answer & explanation

Correct answer: (a).

Hot gas bypass injects discharge gas into the evaporator or suction to create a false load. The compressor keeps pumping almost its full mass flow at nearly full power, while the useful cooling is only about 30 % of capacity. The resulting kW/TR can be two to three times the full-load value. It is one of the least efficient capacity-control methods.

Why the others are wrong: (b) the recirculated gas is compressed again without producing any cooling, so work per useful TR rises; (c) condensing pressure is not reduced by bypassing gas back to the low side; (d) the compressor speed does not change. Better options are a smaller chiller or compressor matched to the base load, slide-valve or VFD capacity control, multiple compressors, or a buffer tank that lets the machine cycle without short-cycling. Long hours on hot gas bypass are a clear audit finding with a quick payback.

Ref: BEE Guidebook Paper 3, Ch. 4 (Refrigeration – capacity control; energy saving opportunities).

15 A large axial-flow cooling fan has to deliver between 60 % and 100 % of design flow. Compared with a fixed-pitch fan controlled by a discharge damper, using variable-pitch (adjustable-in-motion) blades:

[4P3CEM015 | Chapter 5: Fans and Blowers | 5.5 Flow control – variable-pitch axial fans]

- a) Leaves the fan curve unchanged and only shifts the system resistance curve
- b) Saves energy only if the motor speed is reduced at the same time
- c) Raises power at part load because the blades run at higher incidence
- d) Changes the fan curve itself, so power falls closely with the flow needed**

Answer & explanation

Correct answer: (d).

Changing blade pitch in operation alters the fan characteristic itself, much as speed control does. At reduced pitch, the fan delivers less flow at the pressure the system needs while staying at fairly good efficiency, so absorbed power falls closely with flow. A discharge damper, by contrast, adds resistance and moves the operating point up the existing fan curve, burning energy across the damper.

Why the others are wrong: (a) this describes damper control, which raises system resistance; (b) variable pitch saves energy at constant speed, which suits large fans where a high-voltage VFD is costly; (c) correctly reduced pitch lowers both flow and power, and stall occurs only if the duty is pushed beyond the stable region. In audits,

variable-pitch or speed control is the preferred replacement for damper or vane control on large, heavily throttled axial fans, such as cooling tower and mine ventilation fans.

Ref: BEE Guidebook Paper 3, Ch. 5 (Flow control strategies – variable pitch; fan types).

16 A variable-speed chilled water pump serves several AHUs that have two-way control valves. The VFD holds a constant differential-pressure (DP) set point. Where should the DP sensor be located to give the largest pumping energy saving?

[4P3CEM016 | Chapter 6: Pumps and Pumping System | 6.6 Variable-speed pumping – location of the differential-pressure sensor]

- a) Across the pump itself, since this gives the steadiest and most exact control
- b) Across the farthest (index) AHU, so header friction is not held at part load**
- c) At the pump suction, so that the reading is free of pipe friction effects
- d) Across the nearest AHU, as it carries the largest flow with the least loss

Answer & explanation

Correct answer: (b).

The set point must be high enough for the index (hydraulically farthest) circuit to get its design flow. If the sensor is at the pump, the set point must include the full design friction of the mains. The pump then holds that head even at low flow, so speed can fall only a little and savings are small. With the sensor across the index AHU, the set point covers only that coil and valve. As flow falls, the friction in the mains drops with the square of flow, and the pump slows along a curve close to the system curve, giving savings that approach the affinity laws.

Why the others are wrong: (a) a pump-mounted sensor is stable but locks in design friction head at all loads; (c) suction pressure says nothing about the pressure available at the coils; (d) the nearest AHU has the most pressure available, so it would starve remote AHUs.

Ref: BEE Guidebook Paper 3, Ch. 6 (Flow control strategies – variable speed drives).

17 A workshop has 10 impact wrenches ($0.45 \text{ m}^3/\text{min}$ each, usage factor 0.25), 4 grinders ($0.9 \text{ m}^3/\text{min}$ each, usage factor 0.40) and 2 blow guns ($0.6 \text{ m}^3/\text{min}$ each, usage factor 0.15). Allowing 10 % for leakage and then 20 % for future growth on the use-factored demand, the FAD to specify for the compressor is about:

[4P3CEM017 | Chapter 3: Compressed Air System | 3.4 Compressor sizing – usage factors and allowances]

- a) $2.75 \text{ m}^3/\text{min}$
- b) $9.30 \text{ m}^3/\text{min}$
- c) $3.62 \text{ m}^3/\text{min}$**
- d) $3.57 \text{ m}^3/\text{min}$

Answer & explanation

Correct answer: (c) $\approx 3.62 \text{ m}^3/\text{min}$.

Given: wrenches $10 \times 0.45 \times 0.25$; grinders $4 \times 0.9 \times 0.40$; blow guns $2 \times 0.6 \times 0.15$; leakage 10 %; growth 20 %, applied in sequence.

Formula: design FAD = $\Sigma(n \times q \times \text{usage factor}) \times (1 + \text{leakage}) \times (1 + \text{growth})$.

Substitution: $1.125 + 1.440 + 0.180 = 2.745 \text{ m}^3/\text{min}$; $\times 1.10 \times 1.20 = 2.745 \times 1.32$.

Result: **compressor FAD $\approx 3.62 \text{ m}^3/\text{min}$.**

Why the others are wrong: (a) stops at the use-factored demand and makes no allowance for leakage or growth; (b) adds up the connected consumption of every tool as if all ran at once, which would oversize the compressor about three-fold; (d) adds the two allowances (30 %) instead of applying them in sequence. Oversizing is a

common audit finding: a large machine at low load spends long periods unloaded, drawing roughly 25–35 % of full-load power while delivering no air.

Ref: BEE Guidebook Paper 3, Ch. 3 (Compressed air – compressor selection and capacity assessment).

18 Select the WRONG statement about power factor correction in a plant billed on kVAh:

[4P3CEM018 | Chapter 1: Electrical Systems | 1.4 Automatic power factor correction – over-compensation]

- a) APFC steps follow load changes, avoiding a leading PF at light load
- b) Over-correcting to a leading PF lowers the kVAh bill still further**
- c) Capacitors near large motors relieve upstream cables of reactive current
- d) Excess capacitors at light load can raise the bus voltage noticeably

Answer & explanation

Correct answer: (b) is the wrong statement.

On kVAh billing, the meter integrates apparent energy, $\sqrt{(\text{kWh}^2 + \text{kVAh}^2)}$. Most such meters, as regulators specify, treat leading kVAh like lagging kVAh. So once the PF crosses unity into the leading region, kVAh rises again and the bill increases. The lowest kVAh is reached close to unity PF, not by over-correction.

Why the others are true: (a) automatic PF correction (APFC) panels switch capacitor steps in and out with a relay that tracks the reactive demand, which prevents a leading PF when load falls; (c) reactive current then circulates only between the capacitor and the motor, so upstream cables and transformers carry less current and have lower I^2R loss; (d) a fixed bank that is oversized for night load pushes the PF leading and raises the voltage, which increases losses in lamps and lightly loaded motors. Correct sizing plus automatic control gives the best result.

Ref: BEE Guidebook Paper 3, Ch. 1 (Power factor improvement; electricity billing).

19 A cooling tower rejects 12,00,000 kcal/h ($\approx 1,395$ kW) and runs at 4 cycles of concentration; drift is negligible. Evaporation is estimated from the heat load using a latent heat of 580 kcal/kg ($\approx 2,428$ kJ/kg). The metered makeup is 3.6 m³/h. The unaccounted water loss (leaks, overflows) is about:

[4P3CEM019 | Chapter 7: Cooling Towers | 7.4 Water balance – finding unaccounted makeup losses]

- a) 1.53 m³/h
- b) 0.84 m³/h**
- c) 1.01 m³/h
- d) 2.00 m³/h

Answer & explanation

Correct answer: (b) ≈ 0.84 m³/h.

Given: heat rejected 12,00,000 kcal/h ($\approx 1,395$ kW); latent heat 580 kcal/kg ($\approx 2,428$ kJ/kg); COC = 4; metered makeup 3.6 m³/h.

Formula: $E = \text{heat} / \text{latent heat}$; blowdown $B = E / (\text{COC} - 1)$; expected makeup $= E + B$; unaccounted = metered – expected.

Substitution: $E = 12,00,000 / 580 = 2,069$ kg/h = 2.069 m³/h; $B = 2.069 / 3 = 0.690$ m³/h; expected = 2.759 m³/h; 3.6 – 2.759.

Result: **unaccounted loss ≈ 0.84 m³/h.**

Why the others are wrong: (a) treats makeup as evaporation alone and forgets blowdown; (c) takes blowdown as E/COC instead of $E/(\text{COC} - 1)$; (d) assumes one kilogram of water evaporates per 1,000 kcal, ignoring the latent heat given. A gap of this size usually points to basin overflow from a float valve set too high, leaking valves or unmetered drain-offs. Fixing it saves water and chemicals and makes the measured COC meaningful.

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

20 A replacement pump is chosen so that the normal duty lies at about 55 % of its best-efficiency point (BEP) flow, to allow for future expansion. The main consequences, as covered in the guidebook, are:

[4P3CEM020 | Chapter 6: Pumps and Pumping System | 6.2 Pump selection – operating far from the best-efficiency point]

- a) **Lower efficiency, plus higher radial thrust, vibration and seal wear**
- b) Higher efficiency, since a pump runs best when lightly loaded
- c) No energy effect, since power depends only on the flow delivered
- d) Lower head than needed, so the motor will overload at once

Answer & explanation

Correct answer: (a).

A centrifugal pump is most efficient at its BEP. At about half of BEP flow, efficiency is typically 10–20 percentage points lower, so more input power is used for the same hydraulic duty. The flow pattern inside the casing is also unbalanced, which raises radial thrust, shaft deflection, vibration, recirculation and seal and bearing wear.

Why the others are wrong: (b) pump efficiency falls on both sides of the BEP, it does not rise at low flow; (c) input power equals hydraulic power divided by efficiency, so lower efficiency directly raises energy use; (d) at reduced flow, a centrifugal pump develops more head, not less, and power absorbed is usually lower than at rated flow, so overload is not the issue. A better approach is to select for the present duty close to BEP and meet future growth with a larger impeller, a VFD or an additional pump in parallel when the need is real.

Ref: BEE Guidebook Paper 3, Ch. 6 (Pump selection; pump characteristics).

21 An ID fan discharges 40 m³/s through a 1.4 m × 1.4 m outlet straight to the stack. An evasé (diverging outlet) expands the area to 3.0 m² and recovers 70 % of the fall in velocity pressure as static pressure. With gas density 0.95 kg/m³, fan efficiency 75 % and motor efficiency 95 %, the electrical power saved is about:

[4P3CEM021 | Chapter 5: Fans and Blowers | 5.7 Fan outlet – static regain from an evasé]

- a) 6.37 kW
- b) **4.46 kW**
- c) 7.77 kW
- d) 3.18 kW

Answer & explanation

Correct answer: (b) ≈ 4.46 kW.

Given: Q = 40 m³/s; outlet 1.96 m²; evasé exit 3.0 m²; ρ = 0.95 kg/m³; regain 70 %; η_{fan} 0.75; η_{motor} 0.95.

Formula: VP = ρv²/2; regain = 0.7 × (VP₁ – VP₂); saving = Q × regain / (η_{fan} × η_{motor}).

Substitution: v₁ = 20.41 m/s, VP₁ = 197.8 Pa; v₂ = 13.33 m/s, VP₂ = 84.4 Pa; regain = 0.7 × 113.4 = 79.4 Pa; 40 × 79.4 = 3,175 W; ÷ (0.75 × 0.95).

Result: ≈ **4.46 kW saved**.

Why the others are wrong: (a) assumes all of the velocity-pressure fall is regained; (c) applies 70 % to the whole outlet velocity pressure, as if the gas left the evasé at zero velocity; (d) stops at the air-power saving and leaves out the fan and motor efficiencies. An evasé lets the fan run at lower total pressure for the same flow, so a VFD or a re-pitched impeller can then trim the excess.

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

22 A 500 kVA DG set is to supply a building whose load is mostly VFDs, UPS units and LED drivers with 6-pulse or single-phase rectifier inputs. Which precaution is CORRECT?

[4P3CEM022 | Chapter 9: Diesel / Natural Gas Power Generating System | 9.3 DG sets feeding non-linear loads]

- a) Size on kW alone, because harmonic currents carry real power and load the engine
- b) Derate or oversize the alternator, as harmonic currents add heat and distort voltage**
- c) Add plain capacitors at the DG bus to cancel the harmonic currents at their source
- d) Run the set at minimum load, since rectifier loads draw less current from alternators

Answer & explanation

Correct answer: (b).

Rectifier loads draw non-sinusoidal currents. These harmonic currents flow through the alternator windings and add copper and stray losses, so the windings run hotter for the same kW. An alternator also has a much higher source impedance than a grid transformer, so the same harmonic currents cause much more voltage distortion. That can upset the AVR, other electronic loads and the synchronising equipment. Suppliers therefore derate the alternator or recommend a larger frame, a low-reactance alternator, or multi-pulse or filtered drives.

Why the others are wrong: (a) harmonic currents do almost no useful work; they load the alternator thermally, so sizing on kW alone hides the problem; (c) plain capacitors can resonate with the alternator reactance and, at light load, push the set into a leading PF that it cannot control; (d) light loading does not reduce harmonic distortion and also worsens SFC and wet-stacking.

Ref: BEE Guidebook Paper 3, Ch. 9 (DG systems – load matching; Ch. 1 harmonics).

23 A constant-flow primary chilled water loop is converted to a variable primary flow (VPF) system with VFDs on the chilled water pumps. Which statement correctly describes the energy saving and its main limitation?

[4P3CEM023 | Chapter 4: HVAC and Refrigeration System | 4.8 Chilled water distribution – variable primary flow]

- a) Pump power is unchanged, but chillers run at a constant 5 °C delta-T
- b) Chiller power falls sharply, since VPF raises the refrigerant mass flow
- c) Pump power falls with flow, but evaporator flow must stay above its minimum**
- d) Cooling tower fans can be stopped, as VPF removes condenser heat load

Answer & explanation

Correct answer: (c).

In a variable primary flow system, the chilled water pumps slow down as two-way valves at the air handlers close. Following the affinity laws, pump power then falls roughly with the cube of flow, less any static or differential-pressure set point. Separate secondary pumps and a decoupler bypass are no longer needed. The limitation is that each chiller's evaporator has a manufacturer's minimum flow, often about 40–60 % of design. Below that, tube velocity falls, heat transfer degrades and freeze protection can trip, so a minimum-flow bypass valve and careful sequencing are needed.

Why the others are wrong: (a) VPF exists precisely to cut pump power, and delta-T rises at part load as flow follows load; (b) refrigerant flow follows the cooling load, not chilled water flow; (d) heat rejection is unchanged at the same cooling load. VPF works best together with good delta-T discipline at the coils.

Ref: BEE Guidebook Paper 3, Ch. 4 (HVAC energy conservation – chilled water pumping).

24 A 3-phase LT feeder carries 400 A per phase. Its aluminium conductor resistance is 0.050 Ω per phase at 20 °C (temperature coefficient 0.00403 per °C), and in service the conductor runs at 70 °C. The I^2R loss in the feeder at operating temperature is about:

- a) 28.8 kW
- b) 24.0 kW
- c) 30.8 kW
- d) 16.6 kW

Answer & explanation

Correct answer: (a) $\approx$ 28.8 kW.

Given: $I = 400$ A per phase; $R_{20} = 0.050 \Omega$ per phase; $\alpha = 0.00403 / ^\circ\text{C}$; conductor temperature $70 ^\circ\text{C}$.

Formula: $R_t = R_{20} \times [1 + \alpha \times (t - 20)]$; $\text{loss} = 3 \times I^2 \times R_t$.

Substitution: $R_{70} = 0.050 \times (1 + 0.00403 \times 50) = 0.050 \times 1.2015 = 0.0601 \Omega$; $\text{loss} = 3 \times 1,60,000 \times 0.0601 = 28,836$ W.

Result: **feeder loss $\approx$ 28.8 kW.**

Why the others are wrong: (b) uses the $20 ^\circ\text{C}$ catalogue resistance, which understates the loss by about 17 %; (c) applies the coefficient over $70 ^\circ\text{C}$ instead of over the $50 ^\circ\text{C}$ rise above the reference; (d) multiplies by $\sqrt{3}$ instead of 3, confusing the per-phase loss sum with the line-voltage power formula. A cable that runs hot is usually also undersized for its current. Laying a parallel run halves the current per conductor, which cuts I^2R loss and lowers the operating temperature and resistance as well.

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

25 During a cooling tower inspection, the fan blades show tip clearances of 50–75 mm to the fan stack, whereas the maker specifies under about 20 mm. What is the main effect on tower performance?

- a) Fan power rises sharply, because the wider gap increases the blade loading
- b) Drift loss rises, because the gap lets water droplets bypass the eliminators
- c) There is no effect, since the blade angle alone fixes the air that is delivered
- d) Air recirculates around the blade tips, so air flow and cooling capacity fall**

Answer & explanation

Correct answer: (d).

An axial tower fan builds up a pressure difference across its blades. Where the gap between blade tips and stack is large, air leaks from the high-pressure side back to the low-pressure side around the tips. This recirculation reduces the net air flow through the fill, so the L/G ratio rises, approach worsens and cold-water temperature goes up, while the fan still absorbs much the same power. Correcting excessive or uneven tip clearance, together with balancing and setting all blades to the same pitch, is a standard low-cost tower improvement.

Why the others are wrong: (a) recirculation does not raise fan power; the fan simply moves less useful air for about the same power, so efficiency falls; (b) the tip gap lies above the eliminators and does not affect droplet carry-over; (c) blade angle sets the design duty, but tip leakage still reduces the air actually delivered.

Ref: BEE Guidebook Paper 3, Ch. 7 (Cooling towers – energy saving opportunities; fans).

26 A two-stage air compressor takes in air at 1.013 bar(a) and delivers it at 9.0 bar(g). For minimum total work with perfect intercooling, the intercooler should operate at about:

- a) 2.17 bar(g)**

- b) 3.18 bar(g)
- c) 2.01 bar(g)
- d) 4.50 bar(g)

Answer & explanation

Correct answer: (a) ≈ 2.17 bar(g).

Given: $P_1 = 1.013$ bar(a); delivery 9.0 bar(g), so $P_2 = 10.013$ bar(a); perfect intercooling back to intake temperature.

Formula: minimum work when both stages share the same pressure ratio: $P_i = \sqrt{P_1 \times P_2}$ in absolute pressures.

Substitution: $\sqrt{1.013 \times 10.013} = \sqrt{10.143} = 3.185$ bar(a); gauge = $3.185 - 1.013$.

Result: **intercooler pressure ≈ 3.18 bar(a), i.e. ≈ 2.17 bar(g); each stage ratio ≈ 3.14 .**

Why the others are wrong: (b) reports the absolute value as if it were gauge; (c) uses the 9.0 gauge figure as an absolute pressure in the square root; (d) halves the gauge delivery pressure, which loads the stages unequally. If the intercooler pressure drifts, for example through worn valves in one stage, the work splits unequally and the hotter stage wastes energy. A rising intercooler pressure or temperature is therefore a useful diagnostic during audits.

Ref: BEE Guidebook Paper 3, Ch. 3 (Compressor performance – multistage compression).

27 A cooling tower serves a constant heat load with constant water and air flow. As the season changes and inlet wet bulb temperature falls from 28 °C to 18 °C, the auditor should expect:

[4P3CEM027 | Chapter 7: Cooling Towers | 7.2 Effect of wet bulb temperature on approach]

- a) Unchanged outlet water temperature, as range is constant
- b) Colder outlet water but a wider approach than in summer**
- c) A narrower approach, as cold air absorbs heat more readily
- d) Outlet water falling by exactly 10 °C with the wet bulb

Answer & explanation

Correct answer: (b).

At lower wet bulb temperature, outdoor air holds less moisture and its enthalpy changes less per degree. The same mass of air therefore absorbs less heat per degree of temperature rise. With the same heat load, flow and air quantity, cold-water temperature falls, but by less than the fall in wet bulb, so the approach widens.

Why the others are wrong: (a) range is fixed by load and flow, but outlet temperature follows the wet bulb; (c) the approach actually widens at low wet bulb temperatures; (d) the outlet falls by less than the wet bulb change, as explained. Approach measured in winter must therefore not be compared directly with the design approach at summer wet bulb, or the tower may wrongly be judged to have deteriorated. Correct performance to design conditions with the manufacturer's curves.

Ref: BEE Guidebook Paper 3, Ch. 7 (Cooling tower performance – effect of wet bulb temperature).

28 Two identical centrifugal fans run in parallel into a common duct. One is stopped for maintenance while the other keeps running, and neither fan has a non-return or isolating damper. The most likely result is:

[4P3CEM028 | Chapter 5: Fans and Blowers | 5.5 Fans in parallel – one fan stopped]

- a) Air flows back through the idle fan, so useful flow and efficiency fall**
- b) The running fan delivers exactly half of the earlier combined flow
- c) System resistance doubles, so the running fan moves into its stall zone
- d) The idle fan windmills and adds pressure, raising the delivered flow

Answer & explanation

Correct answer: (a).

In parallel operation, both fans discharge into the same plenum. When one fan stops, the plenum pressure produced by the running fan drives air backwards through the idle fan to its inlet. Much of the running fan's output then recirculates instead of reaching the system. The running fan moves to a high-flow, low-pressure point on its curve, and its motor can overload while the useful flow falls. Isolation or back-draft dampers on each fan prevent this.

Why the others are wrong: (b) one fan alone on the same system curve gives more than half the combined flow, because system resistance falls with flow, and it does so only if there is no back-flow; (c) system resistance depends on the ductwork, not on the number of fans running; (d) a reverse-windmilling idle fan absorbs energy rather than adding pressure. Fans in series, by contrast, add pressure at the same flow.

Ref: BEE Guidebook Paper 3, Ch. 5 (Fans in series and parallel; flow control).

29 A process pump delivers 220 m³/h at 42 m head (pump efficiency 72 %, motor efficiency 92 %), but the process uses only 150 m³/h; the rest returns to the suction tank through a bypass valve. For 8,000 h/year, the electrical energy spent pumping the bypass flow is about:

[4P3CEM029 | Chapter 6: Pumps and Pumping System | 6.6 Flow control by bypass (spill-back) – energy wasted]

- a) 64,100 kWh a year
- b) 89,000 kWh a year
- c) 96,800 kWh a year**
- d) 2,07,300 kWh a year

Answer & explanation

Correct answer: (c) $\approx$ 96,800 kWh a year.

Given: bypass flow = 220 – 150 = 70 m³/h; H = 42 m; $\eta_{\text{pump}} = 0.72$; $\eta_{\text{motor}} = 0.92$; 8,000 h/year.

Formula: hydraulic power = $\rho g Q H$; electrical input = hydraulic power / ($\eta_{\text{pump}} \times \eta_{\text{motor}}$); energy = input $\times$ hours.

Substitution: $9.81 \times (70 / 3,600) \times 42 = 8.01$ kW hydraulic; $8.01 / (0.72 \times 0.92) = 12.09$ kW; $12.09 \times 8,000$.

Result: $\approx$ **96,800 kWh a year ($\approx$ ₹ 7.3 lakh at ₹ 7.50/kWh).**

Why the others are wrong: (a) counts only the hydraulic power, leaving out pump and motor losses; (b) leaves out the motor loss; (d) costs the 150 m³/h of useful flow instead of the bypass flow. Bypass control keeps the pump at a high flow and wastes all the energy in the recirculated water. Impeller trimming or a VFD matched to 150 m³/h removes this waste.

Ref: BEE Guidebook Paper 3, Ch. 6 (Flow control strategies – bypass control).

30 An old twin 28 W T5 luminaire uses lamps of 2,600 lm each, draws 62 W including the ballast, and has a light output ratio (LOR) of 0.70. Its LED replacement has a light source of 4,200 lm, draws 36 W including the driver, and has an LOR of 0.85. Their luminaire efficacies (LED and T5) are about:

[4P3CEM030 | Chapter 8: Lighting System | 8.5 Luminaire efficacy – light output ratio and control-gear loss]

- a) 116.7 and 92.9 lm/W
- b) 99.2 and 58.7 lm/W**
- c) 116.7 and 83.9 lm/W
- d) 99.2 and 65.0 lm/W

Answer & explanation

Correct answer: (b).

Given: T5: $2 \times 2,600 = 5,200$ lm, 62 W system, LOR 0.70; LED: 4,200 lm, 36 W system, LOR 0.85.

Formula: luminaire efficacy = lamp lumens $\times$ LOR / total circuit watts.

Substitution: LED = $4,200 \times 0.85 / 36 = 3,570 / 36$; T5 = $5,200 \times 0.70 / 62 = 3,640 / 62$.

Result: **LED $\approx$ 99.2 lm/W; T5 $\approx$ 58.7 lm/W.**

Why the others are wrong: (a) gives bare-source efficacies, using the T5's 56 W lamp rating and ignoring both LOR and ballast loss; (c) includes the T5 ballast loss but still ignores LOR; (d) applies LOR but uses 56 W instead of the 62 W circuit power. The useful light leaving each luminaire is about the same (3,570 against 3,640 lm), so the swap holds lighting levels while cutting power by about 42 %. Lamp-only lm/W figures exaggerate the performance of open or inefficient fittings.

Ref: BEE Guidebook Paper 3, Ch. 8 (Lighting – luminous efficacy, luminaires and control gear).

31 A crusher motor repeats a 15-minute cycle: 75 kW for 3 min, 30 kW for 9 min and 10 kW (idling) for 3 min. The continuous load that heats the motor windings to the same extent, used to select its rating, is about:

[4P3CEM031 | Chapter 2: Electrical Motors | 2.4 Motor sizing for a cyclic load – rms (thermally equivalent) power]

a) **41.0 kW**

b) 35.0 kW

c) 75.0 kW

d) 38.3 kW

Answer & explanation

Correct answer: (a) $\approx$ 41.0 kW.

Given: 75 kW $\times$ 3 min, 30 kW $\times$ 9 min, 10 kW $\times$ 3 min; cycle 15 min.

Formula: copper loss $\propto$ load², so the thermally equivalent load = $\sqrt{[\Sigma(P^2 \times t) / \Sigma t]}$ (the rms load).

Substitution: $(75^2 \times 3 + 30^2 \times 9 + 10^2 \times 3) / 15 = (16,875 + 8,100 + 300) / 15 = 1,685$; $\sqrt{1,685}$.

Result: **rms load $\approx$ 41.0 kW**, so a 45 kW motor will suffice, provided its pull-out torque covers the 75 kW peak.

Why the others are wrong: (b) is the time-weighted average power, which understates heating because losses rise with the square of load; (c) is the peak, and sizing on it gives a motor running at about 47 % average load with poor efficiency and PF; (d) averages the three load levels without time weighting. Correct sizing avoids the chronic under-loading the guidebook warns about, while still protecting the winding temperature.

Ref: BEE Guidebook Paper 3, Ch. 2 (Motor selection; motor loading).

32 A 600 m² office is uniformly lit to 500 lux at 8.0 W/m². The redesign lowers general lighting to 300 lux (power proportional to lux, same luminaires) and adds 60 task lights of 6 W each at workstations. At 2,750 burning hours a year, the annual saving is about:

[4P3CEM032 | Chapter 8: Lighting System | 8.6 Task-ambient lighting – office retrofit savings]

a) **4,290 kWh**

b) 5,280 kWh

c) 3,300 kWh

d) 6,930 kWh

Answer & explanation

Correct answer: (a) $\approx$ 4,290 kWh.

Given: area 600 m²; existing 8.0 W/m² $\rightarrow$ 4,800 W; general lighting scaled to 300/500 $\rightarrow$ 4.8 W/m² $\rightarrow$ 2,880 W; task lights 60 $\times$ 6 = 360 W; 2,750 h/year.

Formula: saving = (existing W – new general W – task W) × hours / 1,000.

Substitution: $4,800 - 2,880 - 360 = 1,560$ W; $1,560 \times 2,750 / 1,000$.

Result: **≈ 4,290 kWh a year (32.5 % of lighting energy).**

Why the others are wrong: (b) leaves out the task lights' own consumption; (c) deducts the task-light load twice; (d) treats the new general lighting load, less the task lights ($2,880 - 360$ W), as the saving. Task-ambient lighting puts high illuminance only where visual work is done and lowers background levels in circulation areas. Lighting power density falls from 8.0 to 5.4 W/m², with no loss of task visibility. Adding occupancy sensors on the task lights would raise the saving further.

Ref: BEE Guidebook Paper 3, Ch. 8 (Lighting design; energy-saving opportunities).

33 A 55 kW conveyor motor is left running with no load between batches for about 2.5 h in each of 3 shifts, 300 days a year. While idling it draws 30 A at 0.15 PF from a 415 V, 3-phase supply. The energy an idle-stop interlock could save each year is about:

[4P3CEM033 | Chapter 2: Electrical Motors | 2.10 Idle running – no-load losses and idle-stop interlocks]

a) 12,610 kWh

b) 7,280 kWh

c) 4,210 kWh

d) 8,860 kWh

Answer & explanation

Correct answer: (b) **≈ 7,280 kWh.**

Given: V = 415 V (line); I₀ = 30 A; PF at idle = 0.15; idle time 2.5 h × 3 shifts × 300 days.

Formula: idle input P₀ = $\sqrt{3} \times V \times I_0 \times \text{PF}$; annual waste = P₀ × idle hours.

Substitution: P₀ = $1.732 \times 415 \times 30 \times 0.15 = 3,235$ W = 3.235 kW; hours = $2.5 \times 3 \times 300 = 2,250$ h; $3.235 \times 2,250$.

Result: **≈ 7,280 kWh a year can be saved.**

Why the others are wrong: (a) multiplies by 3 instead of $\sqrt{3}$, mixing the per-phase and line-voltage formulas; (c) uses the 240 V phase voltage in the line-voltage formula; (d) counts 365 days although the plant works 300 days. Most idle current is magnetising current, so only the kW should be costed. Interlocks that stop idle motors after a set delay, where restarts per hour allow, remove this waste at almost no cost.

Ref: BEE Guidebook Paper 3, Ch. 2 (Motors – energy saving opportunities; idle running).

34 A screw compressor delivers air at 8.0 bar(g) at a coastal plant (atmosphere 1.013 bar). An identical unit is installed at a hill plant where the atmospheric pressure is 0.815 bar and also delivers 8.0 bar(g). At the hill site, its pressure ratio and the mass of air delivered (relative to the coastal unit) are about:

[4P3CEM034 | Chapter 3: Compressed Air System | 3.5 Effect of altitude – compression ratio and mass delivered]

a) 10.8 and about 80 % of the coastal mass

b) 8.9 and about 100 % of the coastal mass

c) 10.8 and about 100 % of the coastal mass

d) 8.9 and about 80 % of the coastal mass

Answer & explanation

Correct answer: (a).

Given: discharge 8.0 bar(g); atmosphere 1.013 bar (coast) and 0.815 bar (hill); same swept volume, so the same intake volume per minute.

Formula: pressure ratio = $(P_{\text{gauge}} + P_{\text{atm}}) / P_{\text{atm}}$; mass delivered $\propto$ intake density $\propto P_{\text{atm}}$ (same temperature).

Substitution: coast: $9.013 / 1.013 = 8.90$; hill: $8.815 / 0.815 = 10.82$; mass ratio = $0.815 / 1.013 = 0.805$.

Result: **pressure ratio $\approx$ 10.8; about 80 % of the coastal mass of air is delivered.**

Why the others are wrong: (b) ignores the lower atmospheric pressure altogether; (c) recognises the higher ratio but assumes the same mass, although thinner intake air carries less mass per m^3 ; (d) reduces the mass but keeps the coastal ratio. So, for the same free air delivery (FAD) in mass terms, a compressor at altitude needs more power, which is why compressors must be selected for site altitude.

Ref: BEE Guidebook Paper 3, Ch. 3 (Factors affecting compressor performance – altitude).

35 A thyristor soft starter on a 110 kW pump motor is wired so that its thyristors carry the motor current all day after the start. Which statement is correct?

[4P3CEM035 | Chapter 2: Electrical Motors | 2.6 Soft starters – running losses and bypass contactor]

- a) Keep the thyristors in circuit, as they cut running kVA by about 30 %
- b) Use the soft starter to vary pump speed and save energy like a VFD
- c) Remove the soft starter, as it raises inrush current above DOL levels
- d) Fit a bypass contactor after ramp-up, as thyristors waste energy as heat**

Answer & explanation

Correct answer: (d).

A soft starter ramps the motor voltage up gradually, which limits inrush current and mechanical shock during starting. Once the motor reaches full speed, it offers no running benefit. Thyristors drop about 1–1.5 V each, so at full current they dissipate roughly 0.5–1 % of load as heat. That heat needs cabinet ventilation and adds harmonics. A bypass contactor closing after ramp-up removes these losses.

Why the others are wrong: (a) at full voltage, motor kVA is set by the load, and the thyristors add losses instead of reducing kVA; (b) a soft starter controls voltage, not frequency, so it cannot vary pump speed efficiently, and reducing voltage on a loaded motor raises slip and copper loss; (c) soft starters reduce inrush current, typically to 2–4 times full-load current against 6–7 times on DOL. Soft starters protect the system during starting, but VFDs are the energy-saving device for variable flow.

Ref: BEE Guidebook Paper 3, Ch. 2 (Soft starters; speed control).

36 In an office AHU system, the supply ducts run through a hot, unconditioned roof space, and a test shows that 20 % of the supply air leaks out before reaching the rooms. Which statement about the energy effect is CORRECT?

[4P3CEM036 | Chapter 5: Fans and Blowers | 5.8 Supply duct leakage – energy consequences]

- a) Only comfort suffers; fan power falls because the leaks lower system resistance
- b) No energy effect results, because leaked air stays inside the building envelope
- c) Cooling load falls, as the leaked air helps cool the hot roof space above
- d) Fan and coil both work harder, moving and cooling air that never reaches rooms**

Answer & explanation

Correct answer: (d).

To deliver the design air to the rooms, the AHU fan must move about 25 % more air ($1 / 0.8$). The extra flow passes through the coil, filters and main ducts, so fan power rises sharply because power climbs much faster than flow. The coil also cools and dehumidifies air that is then dumped into the roof space, so chiller load rises as well.

Sealing joints, testing ducts for leakage and insulating ducts in unconditioned spaces therefore save both fan and refrigeration energy.

Why the others are wrong: (a) leaks do lower resistance downstream, but the fan must still raise its total flow to meet room demand, so power rises, not falls; (b) the leaks are into an unconditioned roof space, outside the cooled zone, so the cooling is lost; (c) cooling a roof space is not useful cooling and only adds load. Where the system has no extra capacity, rooms are simply under-supplied and become uncomfortable.

Ref: BEE Guidebook Paper 3, Ch. 5 (Fans – energy saving opportunities; ductwork).

37 A standard self-ventilated (TEFC) induction motor is fitted with a VFD and will drive a constant-torque conveyor at about 30 % speed for long periods. What is the main technical concern, and the correct remedy?

[4P3CEM037 | Chapter 2: Electrical Motors | 2.9 VFD on a constant-torque load – motor cooling at low speed]

- a) Core loss rises sharply at low frequency; raise the V/f ratio well above rated
- b) Power factor falls at low speed; connect capacitors across the motor terminals
- c) The shaft fan slows and cooling falls; add a separately powered fan or derate**
- d) Motor current falls with speed; a smaller motor can be used on this drive

Answer & explanation

Correct answer: (c).

On a constant-torque load, the motor draws close to rated current at every speed, so its copper losses stay nearly the same. A TEFC motor is cooled by a fan on its own shaft, and at 30 % speed that fan moves only a small fraction of its rated air. The winding temperature therefore rises, and insulation life falls rapidly. The remedy is a separately powered (forced) cooling fan, a derated or larger motor, or a motor rated by its maker for this inverter duty.

Why the others are wrong: (a) with constant V/f the flux stays constant and core loss falls with frequency; raising V/f over-fluxes the motor; (b) capacitors must never be placed on a VFD's output, as its switching pulses would destroy them; the drive's input PF is set by its rectifier, not the motor; (d) on a constant-torque load current does not fall with speed, so downsizing would overload the motor.

Ref: BEE Guidebook Paper 3, Ch. 2 (Speed control – variable frequency drives).

38 A 300-seat auditorium at full occupancy has gains of 70 W sensible and 45 W latent per person, LED lighting of 6 kW and AV equipment of 4 kW. Ventilation-air load is handled separately. The internal cooling load is about:

[4P3CEM038 | Chapter 4: HVAC and Refrigeration System | 4.6 Cooling load – internal gains from occupants and equipment]

- a) 44.5 kW (12.7 TR)**
- b) 31.0 kW (8.8 TR)
- c) 44.5 kW (14.7 TR)
- d) 50.1 kW (14.2 TR)

Answer & explanation

Correct answer: (a).

Given: 300 people × (70 W sensible + 45 W latent); lighting 6 kW; equipment 4 kW; 1 TR = 3.517 kW.

Formula: internal load = n × (sensible + latent per person) + lighting + equipment; TR = kW / 3.517.

Substitution: people = 300 × 115 W = 34.5 kW; total = 34.5 + 6 + 4 = 44.5 kW; 44.5 / 3.517.

Result: **internal load ≈ 44.5 kW ≈ 12.7 TR.**

Why the others are wrong: (b) counts only the sensible gain from people, but the coil must also condense the moisture they add, so latent load is part of the refrigeration duty; (c) divides by 3.024, which is the TR value in

thousands of kcal/h, not in kW; (d) reads the per-person gains as kcal/h instead of watts. Occupancy-based controls such as CO₂-sensed ventilation, LED lighting and scheduled AV equipment cut these internal gains, and each kW of internal gain removed also saves roughly 0.2–0.3 kW of chiller power.

Ref: BEE Guidebook Paper 3, Ch. 4 (HVAC – heat load estimation).

39 An air-conditioned single-storey warehouse in a hot-dry climate plans a 'cool roof' coating. Which combination of surface properties delivers the cool-roof benefit described in ECBC?

[4P3CEM039 | Chapter 10: Energy Conservation in Buildings and ECBC | 10.3 Cool roofs – the properties that matter]

- a) Low solar reflectance with high thermal emittance
- b) High solar reflectance with high thermal emittance**
- c) High solar reflectance with low thermal emittance
- d) Low solar reflectance with low thermal emittance

Answer & explanation

Correct answer: (b).

A cool roof stays close to ambient temperature in the sun for two reasons. First, high solar reflectance (ECBC 2017 asks for at least 0.60 initially on low-slope roofs) bounces most of the incident solar radiation back. Second, high thermal emittance (at least 0.90) lets the roof re-radiate the heat it does absorb. Together these can lower roof surface temperature by 15–30 °C, which cuts heat flow through the roof into the conditioned space and reduces cooling load, especially under large, poorly insulated single-storey roofs.

Why the others are wrong: (a) a dark, low-reflectance roof absorbs most of the sunlight, so high emittance alone cannot keep it cool; (c) a shiny bare metal roof reflects sunlight but has low emittance, so it retains the heat it absorbs and runs hot; (d) a dark, low-emittance roof absorbs heat and cannot shed it. The solar reflectance index (SRI) combines both properties into one comparative figure.

Ref: BEE Guidebook Paper 3, Ch. 10 (ECBC – building envelope; cool roofs).

40 During an audit of a cooling-water pump, which combination of observations MOST clearly shows that the pump is oversized for its system?

[4P3CEM040 | Chapter 6: Pumps and Pumping System | 6.6 Symptoms of an oversized pump]

- a) Suction gauge showing deep vacuum, with noise from cavitation
- b) Motor drawing above its rated current, with the valve fully open
- c) Pump efficiency of 82 % measured at the best-efficiency flow
- d) Discharge valve kept partly closed, with high discharge pressure**

Answer & explanation

Correct answer: (d).

When a pump is selected with too much head or flow margin, operators throttle the discharge valve to bring flow down to what the process needs. The pressure gauge before the valve then reads well above the system requirement, and the pressure difference across the valve is wasted energy. Typical signs are a throttled valve, high discharge pressure, a large pressure drop across the valve and noise or vibration at the valve. Trimming the impeller, fitting a smaller pump or adding a VFD removes the waste.

Why the others are wrong: (a) deep suction vacuum with cavitation points to a suction-side problem (low NPSH available, a choked strainer or a high suction lift), not oversizing; (b) overcurrent with the valve fully open suggests the pump is running out at too high a flow, or has a mechanical problem; (c) operating at the best-efficiency flow with good efficiency is the sign of a well-matched pump.

Ref: BEE Guidebook Paper 3, Ch. 6 (Flow control strategies; energy conservation opportunities).

41 A municipal body plans to install servo-type lighting voltage controllers that reduce supply voltage on street-light feeders after the lamps warm up. Which statement is correct?

[4P3CEM041 | Chapter 8: Lighting System | 8.6 Lighting voltage controllers – where they save energy]

- a) They save the most on LED feeders, as LED power varies with voltage squared
- b) They save nothing on any lamp type, as lamp wattage is fixed by its rating
- c) They save on HPSV and other discharge-lamp circuits, but little on LED feeders**
- d) They raise lamp lumens and power together, so that efficacy rises sharply

Answer & explanation

Correct answer: (c).

With magnetic ballasts, the power drawn by discharge lamps such as HPSV, metal halide and fluorescent falls markedly when supply voltage is lowered. A lighting voltage controller reduces voltage after the arc has stabilised, typically saving 15–25 % with an acceptable lumen drop, and it also cuts lamp and ballast stress from night-time overvoltage. LED luminaires use electronic constant-current drivers that keep power nearly constant over a wide input voltage range, so lowering voltage saves little. LED savings come instead from dimming through the driver's control input, together with timers and astronomical clocks.

Why the others are wrong: (a) LED drivers regulate their output, so their power does not follow voltage squared; (b) discharge-lamp power with magnetic gear does depend on voltage; (d) the controller lowers the voltage, which reduces both lumens and power. Auditors should check the lamp technology before recommending voltage-based energy savers.

Ref: BEE Guidebook Paper 3, Ch. 8 (Lighting controls; energy saving in lighting).

42 In a cooling tower test, measured circulation is 2,760 m³/h with the fan drawing 41 kW, against a design fan power of 45 kW. At the measured wet bulb, range and cold-water temperature, the maker's curves predict 3,000 m³/h at design fan power. Correcting flow by (design fan power / test fan power)^(1/3), the tower capability is about:

[4P3CEM042 | Chapter 7: Cooling Towers | 7.3 Performance test – tower capability with fan-power correction]

- a) 92.0 %
- b) 89.2 %
- c) 94.9 %**
- d) 101.0 %

Answer & explanation

Correct answer: (c) ≈ 94.9 %.

Given: test flow 2,760 m³/h; test fan power 41 kW; design fan power 45 kW; predicted flow 3,000 m³/h.

Formula: adjusted flow = test flow × (P_{design} / P_{test})^(1/3); capability = adjusted flow / predicted flow.

Substitution: (45 / 41)^(1/3) = 1.0315; 2,760 × 1.0315 = 2,847 m³/h; 2,847 / 3,000.

Result: **capability ≈ 94.9 %**.

Why the others are wrong: (a) ignores that the fan was drawing less than design power during the test, so it understates the tower; (b) inverts the power ratio and penalises the tower twice; (d) corrects flow by the plain power ratio, but air flow varies with the cube root of fan power. A capability below 100 % means the tower cannot carry its design duty at design approach. The causes to check are fill scaling, nozzle blockage, uneven distribution and fan problems before blaming the design.

Ref: BEE Guidebook Paper 3, Ch. 7 (Cooling towers – performance assessment).

43 In a vapour compression system, refrigerant leaves the evaporator at 402 kJ/kg and liquid reaches the expansion valve at 256 kJ/kg (saturated at 40 °C). A subcooler cools the liquid by a further 8 K (liquid cp 1.4 kJ/kg·K) with no change in mass flow or compressor work. The percentage gain in refrigerating capacity is about:

[4P3CEM043 | Chapter 4: HVAC and Refrigeration System | 4.3 Refrigeration cycle – capacity gain from liquid subcooling]

- a) 7.1 %
- b) 7.7 %**
- c) 5.5 %
- d) 22.9 %

Answer & explanation

Correct answer: (b) $\approx 7.7\%$.

Given: $h_{\text{evap,out}} = 402$ kJ/kg; $h_{\text{liquid}} = 256$ kJ/kg; subcooling $\Delta T = 8$ K; $c_{p,\text{liquid}} = 1.4$ kJ/kg·K; same mass flow and work.

Formula: refrigerating effect $RE = h_{\text{evap,out}} - h_{\text{liquid}}$; $\Delta RE = c_p \times \Delta T$; gain = $\Delta RE / RE$ (original).

Substitution: $RE = 402 - 256 = 146$ kJ/kg; $\Delta RE = 1.4 \times 8 = 11.2$ kJ/kg; $11.2 / 146$.

Result: **capacity and COP both rise by $\approx 7.7\%$.**

Why the others are wrong: (a) divides the gain by the new refrigerating effect (157.2 kJ/kg) instead of the original; (c) takes 8 K as 8 kJ/kg, ignoring the liquid's specific heat; (d) uses the specific heat of water (4.187 kJ/kg·K), not of liquid refrigerant. Because compressor work is unchanged, the same percentage applies to COP. This is why a well-maintained condenser with some subcooling, or a liquid-suction heat exchanger where suitable, improves plant kW/TR.

Ref: BEE Guidebook Paper 3, Ch. 4 (Vapour compression refrigeration; performance factors).

44 A new office block has 3,600 m² of gross above-grade exterior wall containing 1,620 m² of vertical fenestration, and 2,000 m² of gross roof area containing 120 m² of skylights. Against the ECBC 2017 prescriptive limits, which statement is correct?

[4P3CEM044 | Chapter 10: Energy Conservation in Buildings and ECBC | 10.3 ECBC envelope – window-to-wall ratio and skylight roof ratio]

- a) Only skylights fail: WWR 45 % is below the 50 % limit, SRR 6 % above 5 %
- b) Only glazing fails: WWR 45 % exceeds 40 %, but SRR 6 % is under 10 %
- c) Neither complies: WWR 45 % exceeds 40 % and SRR 6 % exceeds 5 %**
- d) Neither complies: WWR 81.8 % exceeds 40 % and SRR 6.4 % exceeds 5 %

Answer & explanation

Correct answer: (c).

Given: wall 3,600 m², fenestration 1,620 m²; roof 2,000 m², skylights 120 m².

Formula: window-to-wall ratio WWR = fenestration area / gross exterior wall area; skylight roof ratio SRR = skylight area / gross roof area.

Substitution: $WWR = 1,620 / 3,600 = 0.45$; $SRR = 120 / 2,000 = 0.06$.

Result: **WWR = 45 % against the 40 % maximum; SRR = 6 % against the 5 % maximum; neither complies prescriptively.**

Why the others are wrong: (a) quotes a 50 % WWR limit that is not in the code; (b) quotes a 10 % SRR limit that is also not in the code; (d) divides by the opaque area (wall minus glass, roof minus skylights) instead of the gross area. The design must either reduce glazing and skylights or follow the trade-off or whole-building performance

route, where better SHGC, U-values or shading compensate. Large glass areas raise solar heat gain and cooling load.

Ref: BEE Guidebook Paper 3, Ch. 10 (ECBC – building envelope requirements).

45 Water at $0.10 \text{ m}^3/\text{s}$ flows through 1,200 m of 250 mm internal-diameter pipe with a Darcy friction factor of 0.018. The friction head, and the electrical power needed to overcome it (pump efficiency 75 %, motor efficiency 92 %), are about:

[4P3CEM045 | Chapter 6: Pumps and Pumping System | 6.3 System resistance – friction head from the Darcy equation]

- a) 11.3 m and 16.0 kW
- b) 18.3 m and 17.9 kW
- c) 18.3 m and 26.0 kW**
- d) 36.6 m and 52.0 kW

Answer & explanation

Correct answer: (c).

Given: $Q = 0.10 \text{ m}^3/\text{s}$; $L = 1,200 \text{ m}$; $D = 0.25 \text{ m}$; $f = 0.018$; $\eta_{\text{pump}} 0.75$; $\eta_{\text{motor}} 0.92$.

Formula: $v = Q / (\pi D^2/4)$; $h_f = f \times (L/D) \times v^2/2g$; electrical power = $\rho g Q h_f / (\eta_{\text{pump}} \times \eta_{\text{motor}})$.

Substitution: area = 0.0491 m^2 , $v = 2.04 \text{ m/s}$, $v^2/2g = 0.2115 \text{ m}$; $h_f = 0.018 \times 4,800 \times 0.2115 = 18.28 \text{ m}$; $9.81 \times 0.10 \times 18.28 = 17.93 \text{ kW}$; $\div 0.69$.

Result: **friction head $\approx 18.3 \text{ m}$; power $\approx 26.0 \text{ kW}$.**

Why the others are wrong: (a) takes the area as D^2 and drops $\pi/4$, so velocity and head are understated; (b) reports the hydraulic power and leaves out the pump and motor efficiencies; (d) divides v^2 by g instead of $2g$, which doubles the head. Because h_f varies with v^2 , oversized or parallel pipes and smooth, clean bores cut friction power sharply on long lines that run continuously.

Ref: BEE Guidebook Paper 3, Ch. 6 (System characteristics – friction head).

46 A plant draws 1,200 kW at 0.75 lagging PF. A new 300 kW synchronous motor is installed to drive an air compressor and is over-excited to run at 0.80 leading PF. The overall plant power factor becomes about:

[4P3CEM046 | Chapter 1: Electrical Systems | 1.4 Over-excited synchronous motor as a source of reactive power]

- a) 0.760 lagging
- b) 0.821 lagging
- c) 0.910 lagging
- d) 0.874 lagging**

Answer & explanation

Correct answer: (d) ≈ 0.874 lagging.

Given: existing load 1,200 kW at 0.75 lag ($\tan \phi = 0.8819$); synchronous motor 300 kW at 0.80 lead ($\tan \phi = 0.75$).

Formula: $\text{kVar} = \text{kW} \times \tan \phi$ (lagging +, leading -); $\text{PF} = \Sigma \text{kW} / \sqrt{(\Sigma \text{kW}^2 + \Sigma \text{kVar}^2)}$.

Substitution: lagging kVar = $1,200 \times 0.8819 = 1,058.3$; leading kVar = $300 \times 0.75 = 225$; net = 833.3 kVar; total = 1,500 kW; $\text{kVA} = \sqrt{(1,500^2 + 833.3^2)} = 1,716$.

Result: **plant PF $\approx 1,500 / 1,716 \approx 0.874$ lagging.**

Why the others are wrong: (a) adds the motor's kVar as if it were lagging; (b) subtracts the leading kVar but leaves out the motor's own 300 kW; (c) subtracts the motor's full 375 kVA instead of its kVar component. An over-excited synchronous motor both does useful work and supplies reactive power, so it can replace part of a capacitor bank where a large constant-speed drive is needed anyway.

Ref: BEE Guidebook Paper 3, Ch. 1 (Power factor improvement).

47 A process chiller switches its secondary coolant from water to a 30 % ethylene glycol solution so that it can supply $-3\text{ }^{\circ}\text{C}$ coolant. Compared with water at the same heat load, which statement is correct?

[4P3CEM047 | Chapter 4: HVAC and Refrigeration System | 4.8 Secondary coolant – using glycol brine instead of water]

- a) Lower flow suffices, because glycol has a higher specific heat than water
- b) Pump power falls, since glycol is less viscous than water at $-3\text{ }^{\circ}\text{C}$
- c) Higher flow and pump power are needed, and heat transfer becomes weaker**
- d) The chiller kW/TR improves, because the evaporating temperature rises with glycol

Answer & explanation

Correct answer: (c).

A 30 % glycol solution has a specific heat of about $3.7\text{ kJ/kg}\cdot\text{K}$, against $4.19\text{ kJ/kg}\cdot\text{K}$ for water. Its viscosity at sub-zero temperature is several times higher. For the same heat load and temperature difference, more mass must be circulated, pipe friction rises, and heat-exchanger film coefficients fall. Pump power therefore increases, and a lower evaporating temperature is needed for the same coolant temperature.

Why the others are wrong: (a) glycol has a lower specific heat, not a higher one, so more flow is needed; (b) glycol is more viscous than water, especially when cold, so friction and pump power rise; (d) supplying $-3\text{ }^{\circ}\text{C}$ coolant forces a lower evaporating temperature, which raises compressor lift and kW/TR. Glycol strength should therefore be no higher than the freeze protection needs, and coolant should be supplied at the warmest temperature the process accepts. Each degree lower in evaporating temperature costs roughly 2–3 % more compressor power.

Ref: BEE Guidebook Paper 3, Ch. 4 (Refrigeration – secondary refrigerants; factors affecting performance).

48 An 11 kV feeder has a power loss of 60 kW at its peak load. Its annual load factor is 0.55. Using the empirical loss load factor $\text{LLF} = 0.3 \times \text{LF} + 0.7 \times \text{LF}^2$, the annual energy lost in the feeder is about:

[4P3CEM048 | Chapter 1: Electrical Systems | 1.6 Distribution losses – annual energy loss from the loss load factor]

- a) 2,89,100 kWh
- b) 1,59,000 kWh
- c) 1,98,000 kWh**
- d) 5,25,600 kWh

Answer & explanation

Correct answer: (c) $\approx 1,98,000\text{ kWh}$.

Given: peak loss = 60 kW; LF = 0.55; 8,760 h/year.

Formula: $\text{LLF} = 0.3 \times \text{LF} + 0.7 \times \text{LF}^2$; annual loss = peak loss $\times 8,760 \times \text{LLF}$.

Substitution: $\text{LLF} = 0.3 \times 0.55 + 0.7 \times 0.3025 = 0.165 + 0.2118 = 0.3768$; loss = $60 \times 8,760 \times 0.3768$.

Result: **annual feeder loss $\approx 1,98,000\text{ kWh}$.**

Why the others are wrong: (a) multiplies by the load factor itself, but I^2R loss varies with the square of current, so the average loss is not proportional to the average load; (b) uses LF^2 alone, which is correct only for a perfectly flat load and underestimates losses for a peaky load; (d) assumes the peak loss persists all year. The loss load factor always lies between LF^2 and LF. It also explains why flattening the load curve cuts losses: at the same energy, a higher LF lowers the ratio of loss to energy delivered.

Ref: BEE Guidebook Paper 3, Ch. 1 (Electrical systems – distribution losses and load management).

49 A 500 kVA DG set feeding a building has phase currents of 640 A, 520 A and 410 A because many single-phase loads are unevenly connected. Which effect of this unbalance is correct?

[4P3CEM049 | Chapter 9: Diesel / Natural Gas Power Generating System | 9.3 DG operation – unbalanced phase loading]

- a) Fuel use falls, as the average phase current is only 524 A
- b) Alternator heating rises, and the voltage unbalance harms any three-phase motors**
- c) The engine overloads first, since unbalance acts on the crankshaft
- d) There is no effect, since a star-connected alternator shares all load

Answer & explanation

Correct answer: (b).

Unbalanced phase loading drives negative-sequence currents in the alternator. These induce double-frequency currents in the rotor and damper windings, causing extra heating and vibration. The most heavily loaded phase reaches its thermal limit while the set's kW output is still well below rating. Phase voltages also become unequal. Three-phase motors fed from such a supply then carry negative-sequence currents of their own, which raises their copper loss and temperature and cuts their life.

Why the others are wrong: (a) fuel use depends on total kW plus losses, and unbalance only adds losses; (c) the engine sees the total kW, so the alternator and the heaviest phase reach their limits first; (d) star connection does not share load between phases, and the neutral carries the unbalanced current. The guidebook remedy is to balance single-phase loads across the three phases and to check phase currents during DG audits.

Ref: BEE Guidebook Paper 3, Ch. 9 (Factors affecting DG performance – load balancing).

50 A pump with its discharge valve fully closed absorbs 18 kW of shaft power, and all of this power is converted into heat in the 40 litres of water trapped in its casing. Starting from 30 °C, the time for this water to reach 80 °C is about ($c_p = 4.187 \text{ kJ/kg}\cdot\text{K}$):

[4P3CEM050 | Chapter 6: Pumps and Pumping System | 6.4 Pump at shut-off – heating of trapped water]

- a) 7.8 min**
- b) 7.0 min
- c) 12.4 min
- d) 10.9 min

Answer & explanation

Correct answer: (a) $\approx 7.8 \text{ min}$.

Given: shaft power at shut-off 18 kW, all converted to heat; mass 40 kg; $\Delta T = 80 - 30 = 50 \text{ K}$; $c_p = 4.187 \text{ kJ/kg}\cdot\text{K}$.

Formula: $\text{time} = m \times c_p \times \Delta T / \text{power}$.

Substitution: $40 \times 4.187 \times 50 = 8,374 \text{ kJ}$; $8,374 / 18 = 465 \text{ s}$.

Result: $\approx 465 \text{ s}$, i.e. **about 7.8 minutes**.

Why the others are wrong: (b) uses a 20 kW motor input, but motor losses heat the motor, not the water; (c) uses the final 80 °C as the temperature rise instead of $80 - 30 = 50 \text{ K}$; (d) heats the water to 100 °C instead of 80 °C. At shut-off, flow and hydraulic power are zero, so pump efficiency is 0 %, yet the pump still absorbs a large share of its rated power. Dead-heading wastes all this energy and can flash the water, damaging seals; minimum-flow bypasses and interlocks prevent it.

Ref: BEE Guidebook Paper 3, Ch. 6 (Pump performance characteristics; efficient pumping).

..... 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 A DISCOM supplies a cluster of agricultural pump-sets from one 100 kVA, 11/0.433 kV transformer through a long LT network. It plans a High Voltage Distribution System (HVDS): the 11 kV line is extended along the same route and eight 16 kVA single-pump-group transformers replace the 100 kVA unit. Data:

Parameter	Value
Pump-set load	80 kW at 0.80 PF, 415 V, 3-phase
Equivalent lumped LT line resistance (existing)	0.12 Ω per phase
Equivalent lumped 11 kV line resistance (after HVDS)	0.45 Ω per phase
Pumping hours	2,000 h/year
No-load loss: existing 100 kVA unit / eight new 16 kVA units in total	0.26 kW / 0.64 kW (energised 8,760 h/year)
Average cost of power to the DISCOM	₹ 5.50/kWh

- (a) Line current and I^2R loss in the existing LT network (kW and % of load). (1.5 Marks)
- (b) Line current and I^2R loss in the 11 kV extension after HVDS. (1 Mark)
- (c) Net annual energy saving, allowing for the extra no-load loss of the small transformers, and its value. (1.5 Marks)
- (d) State two non-energy benefits of HVDS for agricultural feeders. (1 Mark)

[4P3CEM051 | Chapter 1: Electrical Systems | 1.6 Rural feeder – LT line loss and the HVDS conversion trade-off]

S-1 – Model answer (5 marks)

Given: $P = 80$ kW; $PF = 0.80$; $V_{LT} = 415$ V; $V_{HT} = 11,000$ V; $R_{LT} = 0.12 \Omega$; $R_{HT} = 0.45 \Omega$ per phase; 2,000 pumping h/year; extra no-load loss = $0.64 - 0.26 = 0.38$ kW for 8,760 h; ₹ 5.50/kWh.

(a) Existing LT network (1.5 Marks)

Formula: $I = P / (\sqrt{3} \times V \times PF)$; loss = $3 I^2 R$

Substitution: $I = 80,000 / (1.732 \times 415 \times 0.80)$; loss = $3 \times 139.1^2 \times 0.12$ (1 Mark)

Result: **$I_{LT} \approx 139.1$ A; LT line loss ≈ 6.97 kW, i.e. about 8.7 % of the 80 kW load** (0.5 Mark)

(b) 11 kV extension after HVDS (1 Mark)

Formula: same, at 11 kV

Substitution: $I = 80,000 / (1.732 \times 11,000 \times 0.80)$; loss = $3 \times 5.249^2 \times 0.45$

Result: **$I_{HT} \approx 5.25$ A; line loss ≈ 37.2 W (0.037 kW) – current falls by the voltage ratio 26.5 and loss by its square, so even a thinner conductor loses almost nothing**

(c) Net annual saving (1.5 Marks)

Line-loss saving = $(6.968 - 0.037) \times 2,000 = 13,861$ kWh/year (0.5 Mark)

Extra transformer no-load loss = $0.38 \times 8,760 = 3,329$ kWh/year (0.5 Mark)

Result: **Net saving $\approx 13,861 - 3,329 \approx 10,532$ kWh/year, worth $\approx ₹ 57,926$ a year for one 100 kVA area** (0.5 Mark)

Comment: the no-load loss of the many small transformers removes about a quarter of the gross saving, so energy-efficient (low no-load loss or star-labelled) distribution transformers should be used. The voltage drop also falls from about 29 V (7.0 %) on LT to about 4 V (0.04 %) on the 11 kV line.

(d) Non-energy benefits (1 Mark, any two $\times 0.5$)

1. Hooking and unauthorised connections on bare LT lines are almost eliminated, as there is no LT line to tap, so commercial (AT&C) losses fall.
2. Pump-set terminal voltage improves, so motor burn-outs and transformer failures fall; each small transformer serves few consumers, so faults affect fewer farmers and overload is easier to detect.

Marking: (a) $1 + 0.5$; (b) 1; (c) 0.5×3 ; (d) 0.5×2 .

S-2 An auditor tests a farmer's borewell submersible pump-set. Water is delivered into a rectangular measuring tank $2.0 \text{ m} \times 1.5 \text{ m}$; the level rises by 0.60 m in 2 minutes 30 seconds. The pumping water level in the bore is 38 m below ground; the delivery pressure gauge at the well head, 0.5 m above ground, reads $1.2 \text{ kg/cm}^2(\text{g})$ ($\approx 1.18 \text{ bar}(\text{g})$). Friction in the column pipe below the gauge is already reflected in the reading; velocity head is negligible. The pump-set draws 12.4 kW from the supply. A BEE star-rated pump-set giving the same duty has an overall (wire-to-water) efficiency of 63% . The set runs $1,800 \text{ h/year}$; the energy value to the State is $\text{₹ } 6.50/\text{kWh}$; the new pump-set costs $\text{₹ } 85,000$.

(a) Flow rate in m^3/h and total head. (1.5 Marks)

(b) Hydraulic power and overall efficiency of the existing pump-set. (1.5 Marks)

(c) Input power of the star-rated pump-set, the annual energy and cost saving and the simple payback. (1.5 Marks)

(d) Specific energy (kWh per m^3) before and after. (0.5 Mark)

[4P3CEM052 | Chapter 6: Pumps and Pumping System | 6.5 Borewell submersible pump-set – field test by tank filling and star-rated replacement]

S-2 – Model answer (5 marks)

Given: tank $2.0 \times 1.5 \text{ m}$, rise 0.60 m in 150 s ; pumping water level 38 m ; gauge 0.5 m above ground reading $1.2 \text{ kg/cm}^2(\text{g})$ ($\approx 1.18 \text{ bar}(\text{g})$); input 12.4 kW ; new overall efficiency 0.63 ; $1,800 \text{ h/year}$; $\text{₹ } 6.50/\text{kWh}$; $\text{₹ } 85,000$.

(a) Flow and head (1.5 Marks)

Formula: $Q = \text{tank area} \times \text{level rise} / \text{time}$; $H = \text{pumping water level} + \text{gauge height} + \text{gauge pressure head}$ ($1 \text{ kg/cm}^2 \approx 10 \text{ m of water}$)

Substitution: $Q = 2.0 \times 1.5 \times 0.60 / 150 = 1.8 / 150$; $H = 38 + 0.5 + 1.2 \times 10$ (1 Mark)

Result: **$Q = 0.012 \text{ m}^3/\text{s} = 43.2 \text{ m}^3/\text{h}$; total head $H = 50.5 \text{ m}$** (0.5 Mark)

(b) Hydraulic power and overall efficiency (1.5 Marks)

Formula: $P_h = \rho g Q H$; $\eta_{\text{overall}} = P_h / \text{electrical input}$

Substitution: $P_h = 1,000 \times 9.81 \times 0.012 \times 50.5 / 1,000 = 5.945 \text{ kW}$ (1 Mark)

Result: **$\eta_{\text{overall}} = 5.945 / 12.4 \approx 47.9 \%$** (0.5 Mark)

(c) Star-rated replacement (1.5 Marks)

Formula: new input $= P_h / 0.63$; saving $= (\text{old} - \text{new}) \times \text{hours}$; payback $= \text{cost} / \text{annual saving}$

Substitution: new input $= 5.945 / 0.63 = 9.436 \text{ kW}$; saving $= (12.4 - 9.436) \times 1,800 = 2.964 \times 1,800$ (0.5 Mark)

Result: **New input $\approx 9.44 \text{ kW}$; saving $\approx 5,335 \text{ kWh/year}$, worth $\approx \text{₹ } 34,675$; simple payback $\approx 2.45 \text{ years}$** (0.5 + 0.5 Mark)

(d) Specific energy (0.5 Mark)

Result: **Before: $12.4 / 43.2 \approx 0.287 \text{ kWh/m}^3$; after: $9.436 / 43.2 \approx 0.218 \text{ kWh/m}^3$ (about 24 % lower)**

Comment: the timed-volume method needs no flow meter and is the standard field method for farm pump-sets. Foot-valve, column-pipe and delivery-pipe losses should be checked as well, since frictionless HDPE pipes and correct pipe sizing add to the saving.

Marking: (a) 1 + 0.5; (b) 1 + 0.5; (c) 0.5 × 3; (d) 0.5.

S-3 A coal-fired boiler has VFD-driven FD and ID fans. Flue gas analysis shows 7 % O₂ (dry) at the economiser outlet. After the burners are tuned and air ingress is sealed, the operating O₂ is brought to 4 %. The fans' present input powers are FD 55 kW and ID 90 kW. Assume: excess air = O₂ / (21 – O₂); combustion air and flue gas volumes both change in proportion to (1 + excess air); both fans work against friction-only systems (no static head), so the cube law applies; the gas temperature change is negligible. The boiler runs 7,200 h/year at ₹ 7.20/kWh.

- (a) Excess air before and after tuning. (1 Mark)
- (b) Ratio of new to old air and gas flow. (1 Mark)
- (c) New FD and ID fan input powers and the total power saved. (1.5 Marks)
- (d) Annual energy and cost saving. (1 Mark)
- (e) State one precaution when lowering O₂. (0.5 Mark)

[4P3CEM053 | Chapter 5: Fans and Blowers | 5.5 FD and ID fans – fan-law savings from reducing excess air with VFDs]

S-3 – Model answer (5 marks)

Given: O₂ falls from 7 % to 4 %; fan input FD 55 kW, ID 90 kW; cube law; 7,200 h/year; ₹ 7.20/kWh.

(a) Excess air (1 Mark)

Formula: $EA = O_2 / (21 - O_2)$

Substitution: before 7 / 14; after 4 / 17

Result: **EA before = 50 %; after ≈ 23.5 %** (0.5 + 0.5)

(b) Flow ratio (1 Mark)

Formula: flow ratio = $(1 + EA_2) / (1 + EA_1)$

Substitution: 1.2353 / 1.50

Result: **new flow ≈ 0.8235 of the old flow (about 17.6 % less air and gas)**

(c) New fan powers (1.5 Marks)

Formula: $P_2 = P_1 \times (\text{flow ratio})^3$ (fan law with speed control, friction-only system)

Substitution: $0.8235^3 = 0.5585$; FD = 55×0.5585 ; ID = 90×0.5585 (0.5 Mark)

Result: **FD ≈ 30.7 kW; ID ≈ 50.3 kW; total saving ≈ $145 \times (1 - 0.5585) \approx 64.0$ kW** (1 Mark)

(d) Annual saving (1 Mark)

Result: **$64.0 \times 7,200 \approx 4,60,907$ kWh/year, worth ≈ ₹ 33,18,530 a year** – in addition to the larger boiler fuel saving from lower dry flue gas loss.

(e) Precaution (0.5 Mark)

Lower O₂ only while CO stays within limits (typically below about 200 ppm) and combustion stays stable, and measure O₂ at the furnace exit as well, because air ingress through the economiser and air heater inflates the O₂ reading at the ID fan, and the ID fan still has to handle that leakage air.

Marking: (a) 0.5 × 2; (b) 1; (c) 0.5 + 1; (d) 1; (e) 0.5.

S-4 An indoor badminton hall measures 32 m × 18 m; the luminaires are mounted 8 m above the playing surface (working plane). The target maintained illuminance is 400 lux and the maintenance factor is 0.80. The proposed LED high-bay luminaire gives 19,500 lm at 150 W (including driver). The manufacturer's utilisation factor table (for the given reflectances) is:

Room index	1.00	1.25	1.50	2.00
------------	------	------	------	------

Utilisation factor	0.56	0.62	0.67	0.72
--------------------	------	------	------	------

The existing installation has 20 metal halide luminaires of 400 W (440 W including ballast). Lighting runs 3,000 h/year at ₹ 8.50/kWh.

- (a) Room index and utilisation factor (by linear interpolation). (1.5 Marks)
- (b) Number of LED luminaires needed (choose a practical layout) and the illuminance achieved. (1.5 Marks)
- (c) LPD before and after, and the annual energy and cost saving. (2 Marks)

[4P3CEM054 / Chapter 8: Lighting System / 8.5 Sports hall LED design – room index, interpolated utilisation factor and LPD]

S-4 – Model answer (5 marks)

Given: L = 32 m, W = 18 m, H_m = 8 m; E = 400 lux; MF = 0.80; LED 19,500 lm, 150 W; existing 20 × 440 W; 3,000 h/year; ₹ 8.50/kWh.

(a) Room index and UF (1.5 Marks)

Formula: $RI = L \times W / [H_m \times (L + W)]$

Substitution: $576 / (8 \times 50)$ (0.5 Mark)

Result: **RI = 1.44** (0.5 Mark)

Interpolation between RI 1.25 (0.62) and 1.50 (0.67): $UF = 0.62 + (1.44 - 1.25) / 0.25 \times 0.05$

Result: **UF ≈ 0.658** (0.5 Mark)

(b) Number of luminaires (1.5 Marks)

Formula: $N = E \times A / (\text{lumens per luminaire} \times UF \times MF)$

Substitution: $N = 400 \times 576 / (19,500 \times 0.658 \times 0.80) = 2,30,400 / 10,265$ (0.5 Mark)

Result: **N ≈ 22.4, so 24 luminaires in a 6 × 4 grid** (0.5 Mark)

Result: **Achieved illuminance = $24 \times 10,265 / 576 \approx 427.7$ lux, i.e. about 428 lux** (0.5 Mark)

Note: rounding up to a symmetric grid is good practice for uniformity; taking the UF of 0.62 for RI 1.25 instead of interpolating would give 23.8 luminaires.

(c) LPD and savings (2 Marks)

Result: **LPD before = $8,800 / 576 \approx 15.28$ W/m²; after = $24 \times 150 / 576 = 6.25$ W/m²** (1 Mark)

Formula: saving = (old W – new W) × hours

Substitution: $(8,800 - 3,600) \times 3,000 / 1,000$

Result: **≈ 15,600 kWh/year, worth ≈ ₹ 1,32,600 a year** (1 Mark)

Comment: LED high bays also give instant restrike after a supply dip, which metal halide lamps cannot, and they can be dimmed for practice sessions.

Marking: (a) 0.5 × 3; (b) 0.5 × 3; (c) 1 + 1.

S-5 A hotel's 300 TR (≈ 1,055 kW) hermetic screw chiller averages 75 % load for 16 h/day at 0.70 kW/TR (compressor input; all compressor input enters the refrigerant). A desuperheater between the compressor and the condenser can recover 15 % of the condenser heat. The hotel needs 90,000 litres/day of hot water heated from 28 °C to 55 °C, now produced by a diesel-fired hot water generator (efficiency 80 %; diesel 10,800 kcal/kg (≈ 45,217 kJ/kg), 0.84 kg/litre, ₹ 92/litre). Take 1 TR = 3,024 kcal/h (≈ 3.52 kW).

- (a) Refrigeration effect, compressor heat and condenser heat rejection (kcal/h). (1.5 Marks)
- (b) Heat recoverable per day and the share of the hot-water demand it meets. (1.5 Marks)

(c) Diesel saved per day and the annual cost saving. (2 Marks)

[4P3CEM055 | Chapter 4: HVAC and Refrigeration System | 4.11 Chiller desuperheater heat recovery for hotel hot water]

S-5 – Model answer (5 marks)

Given: load = $0.75 \times 300 = 225$ TR (≈ 791 kW); 0.70 kW/TR; 16 h/day; recovery 15 %; demand 90,000 litres/day $\times$ (55 – 28) °C; heater $\eta = 0.80$; diesel 10,800 kcal/kg ($\approx 45,217$ kJ/kg) $\times$ 0.84 kg/litre; ₹ 92/litre.

(a) Condenser heat (1.5 Marks)

Formula: refrigeration effect = TR $\times$ 3,024; compressor heat = kW $\times$ 860; condenser heat = sum

Substitution: $225 \times 3,024 = 6,80,400$ kcal/h; $225 \times 0.70 \times 860 = 1,35,450$ kcal/h (1 Mark)

Result: **Condenser heat $\approx 8,15,850$ kcal/h (≈ 949 kW)** (0.5 Mark)

(b) Recovery and share of demand (1.5 Marks)

Formula: recovery = 0.15 $\times$ condenser heat $\times$ hours; demand = mass $\times$ ΔT

Substitution: $0.15 \times 8,15,850 = 1,22,378$ kcal/h; $\times 16$ h = 19,58,040 kcal/day; demand = $90,000 \times 1 \times 27 = 24,30,000$ kcal/day (1 Mark)

Result: **Recovered heat meets ≈ 80.6 % of the hot-water demand** (0.5 Mark)

(c) Diesel and cost saving (2 Marks)

Formula: useful heat per litre = 10,800 $\times$ 0.84 $\times$ 0.80; diesel saved = recovered heat / useful heat per litre

Substitution: $10,800 \times 0.84 = 9,072$ kcal/litre; $\times 0.80 = 7,258$ kcal/litre; $19,58,040 / 7,258$ (1 Mark)

Result: **≈ 269.8 litres/day, worth $\approx ₹ 24,821$ a day, i.e. $\approx ₹ 90.6$ lakh a year** (1 Mark)

Comment: removing superheat in the desuperheater also unloads the condenser slightly. A hot-water storage tank is needed because chiller operation and hot-water demand do not coincide. The balance of about 19 % of demand is topped up by the existing heater.

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

S-6 (a) Match each term in Column A with the most appropriate description in Column B (one description is extra). (2.5 Marks)

Column A	Column B
1. Visible light transmittance (VLT)	a. Chiller efficiency weighted over 100, 75, 50 and 25 % load
2. Integrated part load value (IPLV)	b. Ratio of overhang depth to the height of the window it shades
3. Lighting power density (LPD)	c. Annual energy use per square metre of built-up area
4. Energy performance index (EPI)	d. Fraction of visible light passing through glazing
5. Projection factor	e. Installed interior lighting watts per square metre of floor area
	f. Heat flow through 1 m ² of an element per kelvin temperature difference

(b) List five mandatory requirements that ECBC 2017 places on commercial buildings, irrespective of the compliance approach chosen. (2.5 Marks)

[4P3CEM056 | Chapter 10: Energy Conservation in Buildings and ECBC | 10.2 ECBC and building energy terms – matching and mandatory requirements]

S-6 – Model answer (5 marks)

(a) Matching (0.5 Mark each)

1 $\rightarrow$ d (VLT: share of visible light transmitted, which governs daylighting)

2 $\rightarrow$ a (IPLV: single weighted efficiency figure for chiller part-load operation)

3 → e (LPD: W/m^2 of installed interior lighting, compared with code allowances)

4 → c (EPI: $\text{kWh/m}^2\cdot\text{year}$, the benchmark used for star rating of buildings)

5 → b (projection factor: overhang depth ÷ window height; a larger value means more shading)

Description f describes the U-value (thermal transmittance) and is not used.

(b) Mandatory requirements of ECBC 2017 (any five, 0.5 Mark each)

1. Building envelope: fenestration and opaque elements to meet the mandatory air-leakage and sealing provisions, with labelled U-value, SHGC and VLT of glazing.

2. HVAC: equipment efficiency (chillers, unitary units) to meet the code tables, with controls such as time clocks, temperature set-back and economisers where required, and insulated ducts and pipes.

3. Service water heating: a solar water heating system or heat recovery for buildings with a hot-water demand such as hotels and hospitals.

4. Lighting: automatic lighting shut-off, occupancy or daylight controls in specified spaces, and exit signs within the wattage limit.

5. Electrical systems: transformer losses within prescribed limits, energy-efficient motors, power factor correction at the point of supply (to at least 0.97), and check-metering and monitoring of major loads.

6. Renewable-energy-ready roof area kept free for future solar installation (for buildings that must comply).

Note: the prescriptive, trade-off (EPF) and whole-building performance approaches are applied only after these mandatory provisions have been met.

Marking: (a) 5×0.5 ; (b) 5×0.5 .

S-7 The condenser water flow meter of a chiller plant is out of order. During the audit, a 600 TR ($\approx 2,110 \text{ kW}$) hermetic centrifugal chiller (the only chiller in service) is carrying 520 TR ($\approx 1,828 \text{ kW}$) at 0.62 kW/TR . Its cooling tower readings are: hot water 37.2°C , cold water 32.4°C , inlet wet bulb temperature 28.0°C . Design condenser water flow is $450 \text{ m}^3/\text{h}$. The tower runs at 3 cycles of concentration; drift may be neglected. Take $1 \text{ TR} = 3,024 \text{ kcal/h}$ ($\approx 3.52 \text{ kW}$).

(a) Heat rejected in the cooling tower (kcal/h). (1 Mark)

(b) Condenser water flow estimated from the heat balance, as % of design. (1 Mark)

(c) Range, approach and effectiveness. (1 Mark)

(d) Evaporation loss (guidebook relation), blowdown and makeup water per day. (2 Marks)

[4P3CEM057 | Chapter 7: Cooling Towers | 7.2 Cooling tower flow estimated from a chiller heat balance, with water balance]

S-7 – Model answer (5 marks)

Given: 520 TR ($\approx 1,828 \text{ kW}$); 0.62 kW/TR (hermetic, so all input is rejected); $T_{\text{hot}} = 37.2^\circ\text{C}$; $T_{\text{cold}} = 32.4^\circ\text{C}$; WBT = 28.0°C ; design $450 \text{ m}^3/\text{h}$; COC = 3.

(a) Heat rejected (1 Mark)

Formula: heat rejected = $\text{TR} \times 3,024 + \text{kW} \times 860$

Substitution: $520 \times 3,024 = 15,72,480 \text{ kcal/h}$; $520 \times 0.62 \times 860 = 2,77,264 \text{ kcal/h}$

Result: **heat rejected $\approx 18,49,744 \text{ kcal/h}$ ($\approx 2,151 \text{ kW}$)**

(b) Flow from the heat balance (1 Mark)

Formula: flow (m^3/h) = heat rejected / ($1,000 \times \text{range}$)

Substitution: $18,49,744 / (1,000 \times 4.8)$

Result: **flow $\approx 385.4 \text{ m}^3/\text{h}$, i.e. only about 85.6 % of the $450 \text{ m}^3/\text{h}$ design – check strainers, valves and pump condition**

(c) Range, approach, effectiveness (1 Mark)

Result: **Range = $37.2 - 32.4 = 4.8 \text{ }^\circ\text{C}$; approach = $32.4 - 28.0 = 4.4 \text{ }^\circ\text{C}$; effectiveness = $4.8 / 9.2 \approx 52.2 \%$**

(d) Water balance (2 Marks)

Formula: $E (\text{m}^3/\text{h}) = 0.00085 \times 1.8 \times \text{circulation} (\text{m}^3/\text{h}) \times \text{range}$; $B = E / (\text{COC} - 1)$; $\text{makeup} = E + B$

Substitution: $E = 0.00085 \times 1.8 \times 385.4 \times 4.8 = 2.830 \text{ m}^3/\text{h}$; $B = 2.830 / 2 = 1.415 \text{ m}^3/\text{h}$ (1 Mark)

Result: **Makeup $\approx 2.830 + 1.415 \approx 4.245 \text{ m}^3/\text{h} \approx 101.9 \text{ m}^3/\text{day}$** (1 Mark)

Comment: the high range at low flow raises the condensing temperature. Restoring design flow would lower the range to about $4.1 \text{ }^\circ\text{C}$ and the leaving condenser water temperature, improving chiller kW/TR. Raising COC from 3 to 5 would cut makeup by about 17 %.

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

S-8 A foundry receives power at 11 kV. Its simultaneous loads at the time of maximum demand are:

Load	kW	PF (lagging)
Induction melting furnace (with its own correction)	1,500	0.95
Motors (blowers, sand plant, compressors)	900	0.78
Welding rectifiers	300	0.65
Lighting, HVAC and offices	200	0.90

The demand charge is ₹ 475 per kVA per month on the recorded maximum demand.

(a) Total kW, total kVAr and kVA, and the plant power factor. (2 Marks)

(b) Capacitor kVAr needed at the 11 kV / main LT bus to raise the plant PF to 0.99. (1.5 Marks)

(c) Reduction in maximum demand (kVA) and the monthly demand-charge saving. (1.5 Marks)

[4P3CEM058 | Chapter 1: Electrical Systems | 1.4 Multi-load plant – kVAr for 0.99 PF and contract demand benefit]

S-8 – Model answer (5 marks)

Given: loads as tabulated; target PF 0.99; demand charge ₹ 475/kVA/month.

(a) Load summation (2 Marks)

Formula: $\text{kVAr} = \text{kW} \times \tan(\cos^{-1} \text{PF})$; $\text{kVA} = \sqrt{\Sigma \text{kW}^2 + \Sigma \text{kVAr}^2}$; $\text{PF} = \Sigma \text{kW} / \text{kVA}$

Substitution: furnace $1,500 \times 0.3287 = 493.0 \text{ kVAr}$; motors $900 \times 0.8023 = 722.1 \text{ kVAr}$; welding $300 \times 1.1691 = 350.7 \text{ kVAr}$; lighting $200 \times 0.4843 = 96.9 \text{ kVAr}$ (1 Mark)

Result: **$\Sigma \text{kW} = 2,900 \text{ kW}$; $\Sigma \text{kVAr} \approx 1,662.7 \text{ kVAr}$; $\text{kVA} \approx 3,342.8 \text{ kVA}$; plant PF ≈ 0.868** (1 Mark)

Note: the arithmetic sum of the individual kVA values (3,416.5 kVA) overstates the demand, because the loads have different power factors.

(b) Capacitor rating (1.5 Marks)

Formula: $\text{kVAr} = \text{kW} \times (\tan \phi_1 - \tan \phi_2)$

Substitution: $\tan \phi_1 = 0.5733$ (PF 0.868); $\tan \phi_2 = 0.1425$ (PF 0.99); $2,900 \times (0.5733 - 0.1425)$ (1 Mark)

Result: **$\approx 1,249 \text{ kVAr}$, i.e. a **1,250 kVAr bank in APFC steps**** (0.5 Mark)

(c) Demand reduction (1.5 Marks)

Formula: $\text{new kVA} = \text{kW} / 0.99$; $\text{saving} = \Delta \text{kVA} \times \text{demand rate}$

Substitution: $2,900 / 0.99 = 2,929.3$ kVA; $3,342.8 - 2,929.3$ (0.5 Mark)

Result: **MD falls by ≈ 413.5 kVA, saving $\approx ₹ 1,96,430$ a month ($\approx ₹ 23.6$ lakh a year)** (1 Mark)

Comment: much of the reactive demand comes from the welding rectifiers and motors. Fitting part of the correction near these loads also cuts cable and transformer losses. Because the rectifiers generate harmonics, a detuned APFC panel is advisable.

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

..... 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 state DISCOM designs a one-year demand side management (DSM) portfolio. Data:

Programme	Data
A. Residential BLDC fans	2,50,000 conventional 75 W ceiling fans replaced by 28 W BLDC fans; use 10 h/day for 300 days/year; 60 % of fans on at the evening peak; DISCOM subsidy ₹ 600 per fan plus ₹ 50 per fan for administration
B. Commercial air conditioners	8,000 old 1.5 TR (≈ 5.27 kW) units drawing 1.95 kW replaced by 5-star inverter units averaging 1.20 kW; 1,200 h/year; 80 % coincident with peak; subsidy ₹ 8,000 per unit
C. Cold-store load shifting	40 cold stores can each shift 150 kW of pull-down load from 6–10 pm (4 h) to the night; 70 % participate on 300 days/year; incentive ₹ 1.50 per kWh shifted
Power purchase cost	Evening peak ₹ 9.20/kWh; night ₹ 3.60/kWh; average avoided cost for programmes A and B ₹ 5.20/kWh
Grid emission factor	0.72 t CO ₂ /MWh

- (a) Annual energy saving (MU) and peak demand reduction (MW) of programme A. (2 Marks)
- (b) Annual energy saving and peak reduction of programme B. (1.5 Marks)
- (c) Energy shifted per day under programme C, and the DISCOM's net daily and annual benefit. (1.5 Marks)
- (d) Total programme cost for A and B, their annual avoided-cost benefit and the simple payback. (1.5 Marks)
- (e) Annual CO₂ reduction from the portfolio, explaining how programme C is treated. (1.5 Marks)
- (f) State True or False: (2 Marks)
 - i. Free riders – participants who would have bought the efficient product anyway – make net DSM savings lower than gross savings.
 - ii. Programme C's shifted energy should be valued at the average avoided cost of ₹ 5.20/kWh.
 - iii. BLDC fans save energy because they run at lower speed than induction-motor fans.
 - iv. Per unit of energy saved, programme B cuts the evening peak more than programme A does.

L-1 – Model answer (10 marks)

Given: as tabulated.

(a) Programme A – BLDC fans (2 Marks)

Formula: energy saving = number $\times$ $\Delta W \times h/\text{day} \times \text{days}$; peak reduction = number $\times$ coincidence $\times \Delta W$

Substitution: $\Delta W = 75 - 28 = 47 \text{ W}$; $2,50,000 \times 47 \times 10 \times 300 / 1,000 = 3,52,50,000 \text{ kWh}$; $2,50,000 \times 0.60 \times 47 \text{ W}$ (1 Mark)

Result: $\approx 35.25 \text{ MU/year saved}$; **peak reduction $\approx 7.05 \text{ MW}$** (1 Mark)

(b) Programme B – inverter ACs (1.5 Marks)

Substitution: $8,000 \times (1.95 - 1.20) \times 1,200 = 72,00,000 \text{ kWh}$; $8,000 \times 0.75 \times 0.80 = 4,800 \text{ kW}$ (0.5 Mark)

Result: $\approx 7.2 \text{ MU/year saved}$; **peak reduction $\approx 4.8 \text{ MW}$** (1 Mark)

(c) Programme C – load shifting (1.5 Marks)

Formula: shifted energy = stores $\times$ participation $\times \text{kW} \times \text{hours}$; net benefit = shifted $\times$ (peak cost – night cost – incentive)

Substitution: $40 \times 0.70 \times 150 \times 4 = 16,800 \text{ kWh/day}$; $16,800 \times (9.20 - 3.60 - 1.50) = 16,800 \times 4.10$ (1 Mark)

Result: $\approx ₹ 68,880 \text{ a day}$, i.e. $\approx ₹ 2.07 \text{ crore a year}$ ($\approx ₹ 2,06,64,000$) over 300 days (0.5 Mark)

(d) Cost and payback of A and B (1.5 Marks)

Cost: A = $2,50,000 \times (600 + 50) = ₹ 16.25 \text{ crore}$; B = $8,000 \times 8,000 = ₹ 6.40 \text{ crore}$; total ₹ 22.65 crore (0.5 Mark)

Benefit: $(35.25 + 7.2) \text{ MU} \times ₹ 5.20 = ₹ 22.07 \text{ crore a year}$ ($\approx ₹ 22,07,40,000$) (0.5 Mark)

Result: **Simple payback $\approx 22.65 / 22.07 \approx 1.03 \text{ years}$** (0.5 Mark)

(e) CO₂ reduction (1.5 Marks)

Formula: CO₂ = energy saved (MWh) $\times$ emission factor

Substitution: $(35,250 + 7,200) \text{ MWh} \times 0.72$ (1 Mark)

Result: $\approx 30,564 \text{ t CO}_2 \text{ a year from A and B}$. **Programme C only moves 50.4 lakh kWh a year (5.04 MU) to the night; it does not reduce energy, so no CO₂ credit is taken for it at an average emission factor** (0.5 Mark)

Note: combined peak reduction of A and B $\approx 11.85 \text{ MW}$, plus about 4.2 MW (28 stores $\times$ 150 kW) shifted out of the evening peak by C.

(f) True or False (0.5 Mark each)

i. True – free riders' savings would have happened anyway, so evaluation deducts them (net-to-gross ratio) before crediting the programme.

ii. False – shifted energy is still bought, only at night; its value is the peak–night price difference ($₹ 9.20 - ₹ 3.60$), not avoided energy cost.

iii. False – BLDC fans deliver the same air at the same speed; they save energy through a permanent-magnet rotor with no rotor copper loss and efficient electronic commutation.

iv. True – B removes about 0.67 MW per MU saved (4.8 MW for 7.2 MU), against about 0.20 MW per MU for A (7.05 MW for 35.25 MU), because AC use is far more coincident with the peak.

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

L-2 A pharmaceutical plant's process exhaust fan is directly coupled to a 75 kW, 4-pole standard motor. The fan has no damper or speed control. Measured data before replacement:

Parameter	Value
Voltage, current, PF	415 V, 82 A, 0.83
Speed	1,472 rpm
Motor efficiency at this load (from test)	91.0 %

A vendor replaced the motor with a 55 kW IE3 motor (efficiency 94.2 % at the operating load). It claims savings by assuming the fan shaft power stays the same. However, the new motor runs at 1,488 rpm. The fan runs 8,000 h/year; the tariff is ₹ 7.50/kWh; the new motor cost ₹ 3,20,000.

- (a) Input power, fan shaft power and percentage loading of the old motor. (1.5 Marks)
- (b) Savings claimed by the vendor (kW, kWh/year, ₹/year) and the claimed payback. (1.5 Marks)
- (c) Actual fan shaft power at 1,488 rpm, and the new motor input. (2 Marks)
- (d) Actual saving, its payback and the new motor's loading. (1.5 Marks)
- (e) If the extra air is not needed, what frequency should a VFD on the new motor be set at to restore 1,472 rpm, and what would the input then be (VFD efficiency 97.5 %)? Comment on the result. (1.5 Marks)
- (f) State True or False: (2 Marks)
 - i. Savings from a motor replacement should be verified by measuring input kW before and after at the same duty, not from nameplate efficiencies alone.
 - ii. If the process does not need the extra air, the power drawn because of the higher speed is a pure loss.
 - iii. A VFD has no losses of its own, so slowing the fan with it always gives the full fan-law reduction.
 - iv. Energy savings from motor replacement can be estimated as $\text{kW output} \times (1/\eta_{\text{old}} - 1/\eta_{\text{new}})$ only if the shaft load is unchanged.

[4P3CEM060 | Chapter 2: Electrical Motors | 2.7 Validating an IE3 motor replacement on a fan – speed rise, cube law and true savings]

L-2 – Model answer (10 marks)

Given: 415 V, 82 A, PF 0.83, 1,472 rpm, $\eta_{\text{old}} = 0.91$; new $\eta = 0.942$ at 1,488 rpm; 8,000 h; ₹ 7.50/kWh; cost ₹ 3,20,000.

(a) Old motor (1.5 Marks)

Formula: $\text{input} = \sqrt{3} V I \text{ PF}$; $\text{shaft} = \text{input} \times \eta$; $\text{loading} = \text{shaft} / \text{rating}$

Substitution: $1.732 \times 0.415 \times 82 \times 0.83 = 48.92 \text{ kW}$; 48.92×0.91 (1 Mark)

Result: **Input $\approx 48.92 \text{ kW}$; shaft $\approx 44.52 \text{ kW}$; loading $\approx 59.4 \%$ of 75 kW** (0.5 Mark)

(b) Vendor's claim (1.5 Marks)

Formula: $\text{saving} = \text{shaft} \times (1/\eta_{\text{old}} - 1/\eta_{\text{new}})$

Substitution: $44.52 \times (1/0.91 - 1/0.942) = 44.52 \times 0.03733$ (0.5 Mark)

Result: **$\approx 1.66 \text{ kW}$; $\approx 13,295 \text{ kWh/year}$; $\approx ₹ 99,713 \text{ a year}$; claimed payback $\approx 3.21 \text{ years}$** (1 Mark)

(c) Actual shaft power and input (2 Marks)

Formula: for a fan, $P \propto N^3$, so $P_2 = P_1 \times (N_2/N_1)^3$

Substitution: $\text{speed ratio} = 1,488 / 1,472 = 1.0109$; $\text{cube} = 1.0330$; $P_2 = 44.52 \times 1.0330$ (1 Mark)

Result: **Fan shaft power rises to $\approx 45.99 \text{ kW}$ (+3.3 %); new input = $45.99 / 0.942 \approx 48.82 \text{ kW}$** (1 Mark)

(d) Actual saving (1.5 Marks)

Result: **Saving $\approx 48.92 - 48.82 \approx 0.10 \text{ kW}$, i.e. only $\approx 831 \text{ kWh}$ ($\approx ₹ 6,230$) a year; payback $\approx 51 \text{ years}$** (1 Mark)

Result: **New motor loading = $45.99 / 55 \approx 83.6 \%$ – well loaded, but the higher speed cancels almost all of the efficiency gain** (0.5 Mark)

(e) VFD to restore speed (1.5 Marks)

Formula: $f = 50 \times N_{\text{required}} / N_{\text{new}}$ (slip nearly unchanged at the same torque)

Substitution: $50 \times 1,472 / 1,488$ (0.5 Mark)

Result: **$f \approx 49.46 \text{ Hz}$; input = $44.52 / (0.942 \times 0.975) \approx 48.47 \text{ kW}$; saving $\approx 0.45 \text{ kW} \approx 3,590 \text{ kWh a year}$** (0.5 Mark)

Comment: even so, the saving is far smaller than claimed and would not justify the VFD's cost. The correct project would have checked the speed match before purchase, chosen a motor with matching speed, or treated the fan itself (impeller, inlet conditions, system resistance), where the real savings lie. (0.5 Mark)

(f) True or False (0.5 Mark each)

i. True – measured input at the same fan duty captures speed, load and actual efficiency together, which is exactly what the vendor's claim missed.

ii. True – air delivered beyond the process need does no useful work, so the extra input it needs is waste.

iii. False – the drive's own losses (here 2.5 %) are added, so the net saving is the fan-law reduction less the drive losses.

iv. True – the formula assumes the same output; if the speed and hence the load change, the output must be recalculated.

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

L-3 A textile mill's oil-free screw compressor delivers 20 m³/min of free air at 8 bar(g) and draws 120 kW. It runs fully loaded for 8,000 h/year. Air passes through a water-cooled aftercooler with a moisture separator, then a refrigerated dryer that has no air-to-air pre-cooler and is set to a 3 °C pressure dew point. Condensate is removed by six timer-operated solenoid drains. Audit data:

Item	Value
Intake air (35 °C, 70 % RH): water vapour carried	27.7 g per m ³ of free air
Saturated compressed air at 8 bar(g) holds	5.8 g/m ³ of free air at 40 °C; 9.8 g/m ³ at 50 °C; 0.60 g/m ³ at 3 °C
Aftercooler outlet temperature	design 40 °C; measured 50 °C (fouled tubes)
Free air density; cp of air; latent heat of condensation	1.15 kg/m ³ ; 1.005 kJ/kg·K; 2,500 kJ/kg
Refrigerated dryer	rated refrigeration capacity 30 kW; COP 3.0
Timer drains	each opens for 15 s every 4 min and passes 1.5 m ³ /min of free air while open
Zero-loss (level-sensing) drains	₹ 18,000 each, installed
Tariff	₹ 8.00/kWh

(a) At the design aftercooler outlet (40 °C), the water entering with the air, the condensate removed by the aftercooler and by the dryer, and the total condensate per day. (2 Marks)

(b) The dryer refrigeration load (sensible + latent) and its electrical input at design conditions, and its percentage loading. (2 Marks)

(c) Repeat (b) for the fouled aftercooler (50 °C). Find the extra dryer energy and cost per year, and comment. (2.5 Marks)

(d) The compressed air lost through the timer drains, its power and annual cost, and the payback on replacing them with zero-loss drains. (2.5 Marks)

(e) Suggest two further measures for this air-treatment system. (1 Mark)

L-3 – Model answer (10 marks)

Given: free air 20 m³/min = 1,200 m³/h; 120 kW; 8,000 h/year; moisture 27.7 / 5.8 / 9.8 / 0.60 g per m³ of free air; $\rho = 1.15 \text{ kg/m}^3$; $c_p = 1.005 \text{ kJ/kg}\cdot\text{K}$; $h_{fg} = 2,500 \text{ kJ/kg}$; dryer 30 kW, COP 3.0; 6 drains, 15 s open per 240 s, 1.5 m³/min each; ₹ 18,000 per drain; ₹ 8.00/kWh.

(a) Moisture balance at 40 °C (2 Marks)

Formula: water flow = free air flow $\times$ moisture content; condensate at each stage = flow $\times$ (moisture in – moisture out)

Substitution: in = $1,200 \times 27.7 = 33,240 \text{ g/h}$; aftercooler = $1,200 \times (27.7 - 5.8)$; dryer = $1,200 \times (5.8 - 0.60)$ (1 Mark)

Result: **Water in $\approx 33.24 \text{ kg/h}$; aftercooler condensate $\approx 26.28 \text{ kg/h}$; dryer condensate $\approx 6.24 \text{ kg/h}$; total $\approx 32.52 \text{ kg/h} \approx 780 \text{ litres a day}$** (1 Mark)

Note: about 79 % of the water is removed by the aftercooler at no energy cost, so the aftercooler outlet temperature governs the dryer load.

(b) Dryer load at design (2 Marks)

Formula: air mass flow = $1,200 \times 1.15 / 3,600$; sensible = $m \times c_p \times (t_{in} - 3)$; latent = condensate $\times h_{fg} / 3,600$; input = load / COP

Substitution: $m = 0.3833 \text{ kg/s}$; sensible = $0.3833 \times 1.005 \times 37 = 14.25 \text{ kW}$; latent = $6.24 \times 2,500 / 3,600 = 4.33 \text{ kW}$ (1 Mark)

Result: **Refrigeration load $\approx 18.59 \text{ kW}$ ($\approx 62.0 \%$ of 30 kW); electrical input $\approx 6.20 \text{ kW}$** (1 Mark)

(c) Fouled aftercooler at 50 °C (2.5 Marks)

Substitution: dryer condensate = $1,200 \times (9.8 - 0.60) = 11.04 \text{ kg/h}$; sensible = $0.3833 \times 1.005 \times 47 = 18.11 \text{ kW}$; latent = $11.04 \times 2,500 / 3,600 = 7.67 \text{ kW}$ (1 Mark)

Result: **Load $\approx 25.77 \text{ kW}$ ($\approx 85.9 \%$ of capacity); input $\approx 8.59 \text{ kW}$, i.e. $\approx 2.40 \text{ kW}$ more** (0.5 Mark)

Result: **Extra energy $\approx 2.395 \times 8,000 \approx 19,160 \text{ kWh a year}$, worth $\approx ₹ 1,53,280$** (0.5 Mark)

Comment: the moisture load on the dryer rises by about 77 % for a 10 °C rise. The dryer is left with little margin on hot, humid days, when the dew point drifts up and water is carried into the mains. Cleaning the aftercooler is the cheaper fix. (0.5 Mark)

(d) Timer-drain loss (2.5 Marks)

Formula: average air lost = drains $\times$ flow while open $\times$ open time / cycle time; power = air lost $\times$ specific power

Substitution: $6 \times 1.5 \times 15 / 240 = 0.5625 \text{ m}^3/\text{min}$ ($\approx 2.8 \%$ of output); specific power = $120 / 20 = 6.0 \text{ kW per m}^3/\text{min}$; 0.5625×6.0 (1 Mark)

Result: **$\approx 3.38 \text{ kW}$ wasted; $\approx 27,000 \text{ kWh a year}$, worth $\approx ₹ 2,16,000$** (1 Mark)

Result: **Zero-loss drains cost $6 \times ₹ 18,000 = ₹ 1,08,000$; simple payback $\approx 0.5 \text{ year}$ (about 6 months)** (0.5 Mark)

(e) Further measures (1 Mark, any two $\times 0.5$)

Draw intake air from a cooler, shaded location, which lowers both the moisture load and the compressor power; fit an air-to-air pre-cooler/reheater or a cycling (thermal-mass) dryer for part-load operation; check whether every end use needs a 3 °C dew point; recover heat from the aftercooler cooling water; and trend the aftercooler outlet temperature and dryer dew point as routine checks.

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

L-4 A dairy's ammonia refrigeration plant chills ice-bank glycol and serves cold rooms at a steady 150 TR ($\approx 527 \text{ kW}$) for 6,000 h/year. The audit finds a fouled evaporative condenser and an undersized suction header, so the plant runs at -15 °C evaporating and 45 °C condensing. After descaling the condenser, adding one condenser module and enlarging the suction header, it is expected to run at -10 °C evaporating and 35 °C condensing. Refrigerant

enthalpies read from the p-h chart (saturated vapour leaves the evaporator, saturated liquid leaves the condenser, no sub-cooling or superheat) are:

Condition	Vapour at compressor suction, h_1 (kJ/kg)	Liquid at condenser outlet, h_3 (kJ/kg)	After isentropic compression, h_{2s} (kJ/kg)
Present (–15 °C / 45 °C)	1,443	415	1,700
Proposed (–10 °C / 35 °C)	1,450	366	1,630

Compressor isentropic efficiency is 75 % and motor efficiency 92 % in both cases. Take 1 TR = 3.516 kW, electricity ₹ 8.50/kWh, and the project cost ₹ 28 lakh.

(a) For the present condition, find the refrigerating effect, the ammonia mass flow, the compressor shaft and electrical input power, the kW/TR and the COP. (2.5 Marks)

(b) Repeat (a) for the proposed condition. (2.5 Marks)

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

(d) Heat rejected in the condenser in both cases, and why the condenser has spare capacity after the retrofit. (1.5 Marks)

(e) State three operating practices that keep the condensing pressure low and the suction pressure high in such a plant. (1.5 Marks)

[4P3CEM062 | Chapter 4: HVAC and Refrigeration System | 4.3 Ammonia plant in a dairy – suction and condensing conditions from p-h data, compressor power, kW/TR and saving]

L-4 – Model answer (10 marks)

Given: load 150 TR = $150 \times 3.516 = 527.4$ kW; 6,000 h/year; $\eta_{\text{isentropic}}$ 0.75; η_{motor} 0.92; ₹ 8.50/kWh; cost ₹ 28 lakh.

(a) Present condition (2.5 Marks)

Formula: $RE = h_1 - h_3$; $m = Q \div RE$; shaft power = $m \times (h_{2s} - h_1) \div \eta_{\text{is}}$; input = shaft $\div \eta_{\text{motor}}$; COP = $Q \div \text{shaft}$
 Substitution: $RE = 1,443 - 415 = 1,028$ kJ/kg; $m = 527.4 \div 1,028$; isentropic work = $1,700 - 1,443 = 257$ kJ/kg (1 Mark)

Result: **$m \approx 0.513$ kg/s; shaft $\approx 0.513 \times 257 \div 0.75 \approx 175.8$ kW; input ≈ 191.1 kW; ≈ 1.274 kW/TR; COP ≈ 3.00** (1.5 Marks)

(b) Proposed condition (2.5 Marks)

Substitution: $RE = 1,450 - 366 = 1,084$ kJ/kg; $m = 527.4 \div 1,084$; isentropic work = $1,630 - 1,450 = 180$ kJ/kg (1 Mark)

Result: **$m \approx 0.4865$ kg/s; shaft ≈ 116.8 kW; input ≈ 126.9 kW; ≈ 0.846 kW/TR; COP ≈ 4.52** (1.5 Marks)

Comment: the colder liquid raises the refrigerating effect, and the smaller lift cuts the compression work, so both changes help.

(c) Saving and payback (2 Marks)

Formula: saving = (input_present – input_proposed) $\times$ hours

Substitution: $(191.1 - 126.9) \times 6,000 = 64.2$ kW $\times$ 6,000 (1 Mark)

Result: **$\approx 3,85,020$ kWh/year, worth $\approx ₹ 32,72,670$ a year; simple payback $\approx 28,00,000 \div 32,72,670 \approx 0.86$ year (about 10 months)** (1 Mark)

(d) Condenser heat rejection (1.5 Marks)

Formula: heat rejected = refrigeration load + compressor shaft power

Result: **Present** $\approx 527.4 + 175.8 \approx 703.2 \text{ kW}$; **proposed** $\approx 527.4 + 116.8 \approx 644.2 \text{ kW}$ (1 Mark)

The condenser now rejects about 59 kW less heat and has an extra module, so it can hold the 35 °C condensing temperature even on hot, humid days. (0.5 Mark)

(e) Operating practices (1.5 Marks, any three $\times 0.5$)

Purge non-condensable gases from the condenser regularly; keep evaporative-condenser tubes and nozzles scale-free with proper water treatment and bleed; let condensing pressure float down in cool weather instead of holding a fixed head pressure; defrost evaporator coils on demand to keep the suction temperature up; drain oil from evaporators and the surge drum; avoid throttled suction valves and set the cold-room temperature no lower than the product needs.

Marking: (a) 1 + 1.5; (b) 1 + 1.5; (c) 1 + 1; (d) 1 + 0.5; (e) 0.5 $\times$ 3.

L-5 A five-star hotel in a composite climate has large single clear glass windows on its west and south façades. The guest rooms are air-conditioned for 3,600 h/year of cooling-season hours. Retrofit with double-glazed low-e units is proposed. Data:

Item	West façade	South façade
Glazed area	600 m ²	400 m ²
Mean solar irradiance on the glass during cooling hours	350 W/m ²	250 W/m ²

Glass properties:

Glass type	U-value (W/m ² ·K)	SHGC	Visible light transmittance
Existing single clear glass	5.8	0.82	0.88
Proposed double-glazed low-e	1.8	0.25	0.45

Mean outdoor-to-indoor temperature difference during cooling hours is 9 K. The chilled water plant (chillers, pumps and cooling towers) averages 1.1 kW/TR. Take 1 TR = 3.516 kW, electricity ₹ 9.50/kWh, and the installed retrofit cost ₹ 6,500 per m² of glass. Take the ECBC 2017 limits for non-north vertical fenestration as $U \leq 3.0 \text{ W/m}^2\cdot\text{K}$ and $\text{SHGC} \leq 0.27$.

(a) Solar heat gain, conduction heat gain and total heat gain through the glazing for the existing and the proposed glass. (3 Marks)

(b) Reduction in cooling load (kW and TR), and the annual chiller-plant energy and cost saving. (2.5 Marks)

(c) Check whether each glass type meets the ECBC fenestration limits. (1 Mark)

(d) Retrofit cost and simple payback. (1.5 Marks)

(e) Because of the lower visible transmittance, the corridor and lobby lighting is expected to use 12,000 kWh/year more. Find the net saving and the revised payback, and state one way to limit this penalty. (2 Marks)

[4P3CEM063 | Chapter 10: Energy Conservation in Buildings and ECBC | 10.3 Hotel façade glazing retrofit – solar and conduction gain, chiller energy, ECBC fenestration check and net payback]

L-5 – Model answer (10 marks)

Given: west 600 m² at 350 W/m²; south 400 m² at 250 W/m²; ΔT 9 K; 3,600 h/year; 1.1 kW/TR; ₹ 9.50/kWh; ₹ 6,500/m².

(a) Heat gain through the glazing (3 Marks)

Formula: solar gain = $\Sigma (A \times SHGC \times I)$; conduction gain = $U \times A \times \Delta T$

Existing: solar = $600 \times 0.82 \times 350 + 400 \times 0.82 \times 250 = 1,72,200 + 82,000$; conduction = $5.8 \times 1,000 \times 9$ (1 Mark)

Result: **Existing: solar $\approx$ 2,54,200 W; conduction $\approx$ 52,200 W; total $\approx$ 306.4 kW** (0.5 Mark)

Proposed: solar = $600 \times 0.25 \times 350 + 400 \times 0.25 \times 250 = 52,500 + 25,000$; conduction = $1.8 \times 1,000 \times 9$ (1 Mark)

Result: **Proposed: solar $\approx$ 77,500 W; conduction $\approx$ 16,200 W; total $\approx$ 93.7 kW** (0.5 Mark)

Comment: about 83 % of the existing gain is solar, so SHGC matters far more than U-value here.

(b) Cooling load and plant energy (2.5 Marks)

Formula: $\Delta TR = \Delta \text{heat gain} \div 3.516$; energy = $\Delta TR \times \text{kW/TR} \times \text{hours}$

Substitution: $306.4 - 93.7 = 212.7$ kW; $212.7 \div 3.516$ (0.5 Mark)

Result: **$\approx$ 60.5 TR ($\approx$ 213 kW) less cooling load** (0.5 Mark)

Substitution: $60.5 \times 1.1 = 66.5$ kW $\times$ 3,600 h (0.5 Mark)

Result: **$\approx$ 2,39,560 kWh/year, worth $\approx$ ₹ 22,75,817 a year** (1 Mark)

(c) ECBC fenestration check (1 Mark)

Result: **Existing glass fails both limits ($U\ 5.8 > 3.0$; $SHGC\ 0.82 > 0.27$); the proposed glass meets both ($U\ 1.8 \leq 3.0$; $SHGC\ 0.25 \leq 0.27$)** (1 Mark)

(d) Cost and payback (1.5 Marks)

Substitution: $1,000\ m^2 \times ₹\ 6,500$ (0.5 Mark)

Result: **Cost $\approx$ ₹ 65,00,000; simple payback $\approx$ $65,00,000 \div 22,75,817 \approx 2.86$ years** (1 Mark)

(e) Daylight penalty (2 Marks)

Substitution: lighting penalty = $12,000 \times 9.50 = ₹\ 1,14,000$; net = $22,75,817 - 1,14,000$ (0.5 Mark)

Result: **Net saving $\approx$ ₹ 21,61,817 a year; revised payback $\approx$ $65,00,000 \div 21,61,817 \approx 3.01$ years** (1 Mark)

Mitigation (0.5 Mark): choose a spectrally selective low-e glass with a higher light-to-solar-gain ratio (VLT near 0.6 at the same SHGC), add daylight-linked dimming in perimeter zones, or use external shading on the west façade so a clearer glass can be kept.

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

L-6 A plant's captive DG set has an alternator rated 1,250 kVA at 0.8 PF. Its diesel engine is rated 1,450 bhp (1 bhp = 0.746 kW). The alternator efficiency is 95.5 %, and the engine's brake thermal efficiency at rated output is 38 %. Diesel has a calorific value of 10,700 kcal/kg ($\approx$ 44,799 kJ/kg) and a density of 0.835 kg/litre and costs ₹ 90/litre; O&M adds ₹ 1.20/kWh. The plant load on the set is 820 kW at 0.72 PF; grid power costs ₹ 8.50/kWh.

(a) Maximum electrical output the engine can support, and the maximum load PF at which the set can deliver its full 1,250 kVA. (2.5 Marks)

(b) Fuel consumption (kg/h and litres/h) and specific energy generation (kWh/litre) at this maximum output. (2 Marks)

(c) For the present 820 kW load: its kVA and % loading on kVA and kW bases; the capacitor kVAR needed to reach 0.92 PF and the resulting kVA; and the extra load that can then be added, stating which limit applies. (2 Marks)

(d) Cost of DG generation per kWh against the grid. (1.5 Marks)

(e) State True or False: (2 Marks)

- i. A waste heat boiler or silencer with a high exhaust-side pressure drop raises the engine's specific fuel consumption.
- ii. Fuel measured by volume should be converted to mass using the density at its actual temperature before SFC is calculated.
- iii. An alternator is most efficient at very light load, because its copper losses are then small.
- iv. For this set, fuel accounts for less than half of the cost of DG generation per kWh.

[4P3CEM064 | Chapter 9: Diesel / Natural Gas Power Generating System | 9.3 DG set – engine-limited power factor, fuel rate, PF correction headroom and cost of generation]

L-6 – Model answer (10 marks)

Given: 1,250 kVA; engine 1,450 bhp; $\eta_{alt} = 0.955$; BTE = 0.38; diesel CV = 10,700 kcal/kg = 44,799 kJ/kg ($\times 4.1868$); 0.835 kg/litre; ₹ 90/litre; O&M ₹ 1.20/kWh; load 820 kW at 0.72 PF.

(a) Engine-limited output and maximum PF (2.5 Marks)

Formula: engine output = bhp $\times 0.746$; electrical output = engine output $\times \eta_{alt}$; PF_max = kWe / kVA rating

Substitution: $1,450 \times 0.746 = 1,081.7$ kW; $1,081.7 \times 0.955$ (1.5 Marks)

Result: **Maximum electrical output $\approx 1,033$ kW; maximum PF at 1,250 kVA $\approx 1,033 / 1,250 \approx 0.826$** (1 Mark)

(b) Fuel at maximum output (2 Marks)

Formula: fuel heat = engine output / BTE; fuel (kg/h) = heat (kJ/h) / CV

Substitution: $1,081.7 / 0.38 = 2,846.6$ kW = 1,02,47,684 kJ/h; $\div 44,799$ (1 Mark)

Result: **≈ 228.7 kg/h ≈ 274.0 litres/h; SEG $\approx 1,033 / 274.0 \approx 3.77$ kWh/litre** (1 Mark)

(c) Present load and PF correction (2 Marks)

Result: **kVA = $820 / 0.72 \approx 1,138.9$ kVA, i.e. 91.1 % of the kVA rating but only 79.4 % of the engine-limited 1,033 kW** (0.5 Mark)

Formula: kVA = kW $\times (\tan \phi_1 - \tan \phi_2)$

Substitution: $820 \times (0.9639 - 0.4260)$ (0.5 Mark)

Result: **≈ 441 kVA; kVA falls to $820 / 0.92 \approx 891.3$ kVA** (0.5 Mark)

Result: **At 0.92 PF, the 1,250 kVA rating would allow 1,150 kW, but the engine limits output to 1,033 kW, so only about 213 kW of extra load can be added – the engine, not the alternator, now sets the limit** (0.5 Mark)

(d) Cost of generation (1.5 Marks)

Formula: fuel cost per kWh = diesel price / SEG; total = fuel + O&M

Substitution: $90 / 3.771 = ₹ 23.87/\text{kWh}$; + 1.20 (1 Mark)

Result: **$\approx ₹ 25.07/\text{kWh}$, about ₹ 16.57/kWh (nearly three times) more than grid power at ₹ 8.50** (0.5 Mark)

(e) True or False (0.5 Mark each)

- i. True – exhaust back pressure makes the engine push harder to expel gas and hinders scavenging, so more fuel is burnt per kWh; WHR boilers must be designed within the engine maker's limit.
- ii. True – diesel density falls as its temperature rises, so using a fixed density at the wrong temperature misstates the fuel mass and the SFC.
- iii. False – fixed losses (core, friction, windage, excitation) dominate at light load, so efficiency is low; it peaks near the load where variable losses equal fixed losses.
- iv. False – fuel is ₹ 23.87 of ₹ 25.07/kWh, about 95 % of the cost.

Marking: (a) 1.5 + 1; (b) 1 + 1; (c) 0.5 $\times$ 4; (d) 1 + 0.5; (e) 0.5 $\times$ 4.

..... End of Section III

Section I – Answer key

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