

PRACTICE PAPER — NATIONAL CERTIFICATION EXAMINATION FOR ENERGY MANAGERS & ENERGY AUDITORS

PAPER – 4: ENERGY PERFORMANCE ASSESSMENT FOR EQUIPMENT AND UTILITY SYSTEMS

(For Energy Auditors only — Open book: only the official BEE Guidebook Book 4 is permitted)

Question code: 1P4CEA001 – 1P4CEA016 | SOLUTIONS & MODEL ANSWERS

Duration: 2 HRS Max. Marks: 100 Pass: 50% marks in the paper

General instructions:

- Please check that this question paper contains 16 questions
- The question paper is divided into three sections
- All questions in all three sections are compulsory (N-4: answer any ONE alternative)
- All parts of a question should be answered at one place
- Time management guidance: Section - I & II: 10 minutes each, Section - III: 100 minutes
- Use of a non-programmable scientific calculator is permitted; show Given, Formula, Substitution with units and Result for every numerical

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Section - I: BRIEF QUESTIONS

Marks: 10 x 1 = 10

- (i) Answer all Ten questions
- (ii) Each question carries One mark
- (iii) State whether each statement is True or False

1. During an indirect-method efficiency test on a coal-fired boiler, the flue gas O_2 is brought down from 6 % to 3 % while the exit flue gas temperature stays unchanged. The dry flue gas loss reduces, even though the CO_2 percentage in the flue gas rises.

[1P4CEA001 | Chapter 1: Boilers | Indirect method – excess air and dry flue gas loss]

Answer: TRUE (1 mark)

Given: O_2 (before) = 6 %; O_2 (after) = 3 %; $T_f - T_a$ unchanged; illustrative theoretical air = 5.4 kg/kg coal.

Formula (Guidebook Ch. 1): $EA \% = O_2 / (21 - O_2) \times 100$; $AAS = (1 + EA/100) \times \text{Theoretical air}$; $L_1 = m \times C_p \times (T_f - T_a) / GCV \times 100$, where m (dry flue gas) rises with AAS.

Substitution: $EA = 6 / (21 - 6) \times 100 = 40.0 \% \rightarrow EA = 3 / (21 - 3) \times 100 = 16.7 \%$. $AAS = 1.40 \times 5.4 = 7.56 \text{ kg/kg} \rightarrow 1.167 \times 5.4 = 6.30 \text{ kg/kg}$.

Result: Air (and hence dry flue gas mass m) falls by about **16.7 %**, so L_1 falls in the same proportion at the same ΔT . The CO_2 % rises only because the same CO_2 is diluted by less excess air – a higher CO_2 % is a sign of better combustion, not of a higher loss.

Ref: BEE Guidebook Book 4, Ch. 1 (Indirect method, Steps 1–6).

2. For a back-pressure steam turbine with fixed steam flow, inlet condition and exhaust pressure, an improvement in turbine stage (isentropic) efficiency increases the heat-to-power ratio of the cogeneration system.

[1P4CEA002 | Chapter 3: Cogeneration Systems | Back-pressure turbine – isentropic efficiency and heat-to-power ratio]

Answer: FALSE (1 mark)

Given (illustration): steam 20 TPH = 20,000 kg/h; $h_1 = 790$ kcal/kg ($\approx 3,308$ kJ/kg); isentropic exhaust enthalpy $h_{2s} = 640$ kcal/kg ($\approx 2,680$ kJ/kg); condensate return 100 kcal/kg (≈ 419 kJ/kg); $\eta_{is} = 70\%$ vs 80% .

Formula: $\eta_{is} = (h_1 - h_2)/(h_1 - h_{2s})$; Power (kW) = $m \times (h_1 - h_2)/860$; Process heat = $m \times (h_2 - h_{cond})$; Heat-to-power ratio = process heat / (power $\times 860$).

Substitution: $\eta = 70\%$: $\Delta h = 105$ kcal/kg (≈ 440 kJ/kg), $P = 2,442$ kW, $h_2 = 685$ kcal/kg ($\approx 2,868$ kJ/kg), heat = 1,17,00,000 kcal/h (≈ 13.6 MW), HPR = 5.57. $\eta = 80\%$: $\Delta h = 120$ kcal/kg (≈ 502 kJ/kg), $P = 2,791$ kW, $h_2 = 670$ kcal/kg ($\approx 2,805$ kJ/kg), heat = 1,14,00,000 kcal/h (≈ 13.3 MW), HPR = 4.75.

Result: Better isentropic efficiency converts more enthalpy into power and leaves less in the exhaust, so the heat-to-power ratio **falls (5.57 $\rightarrow$ 4.75)**; the statement is false.

Ref: BEE Guidebook Book 4, Ch. 3 (Steam turbine performance; EUF).

3. In a counter-flow water-to-water heat exchanger, hot water is cooled from 80°C to 50°C while cold water is heated from 30°C to 60°C . The LMTD of this exchanger is 20°C and its effectiveness is 60% .

[1P4CEA003 | Chapter 4: Heat Exchangers | Counter-flow exchanger – LMTD and effectiveness]

Answer: TRUE (1 mark)

Given: $T_{h,in} = 80^\circ\text{C}$, $T_{h,out} = 50^\circ\text{C}$, $T_{c,in} = 30^\circ\text{C}$, $T_{c,out} = 60^\circ\text{C}$, counter-flow.

Formula (Guidebook Ch. 4): $\text{LMTD} = (\Delta T_1 - \Delta T_2)/\ln(\Delta T_1/\Delta T_2)$, with $\Delta T_1 = T_{h,in} - T_{c,out}$ and $\Delta T_2 = T_{h,out} - T_{c,in}$; when $\Delta T_1 = \Delta T_2$, $\text{LMTD} = \Delta T$. Effectiveness $\epsilon = \text{actual heat transfer} / \text{maximum possible} = (T_{c,out} - T_{c,in})/(T_{h,in} - T_{c,in})$ when the cold fluid has the lower (here equal) heat capacity rate.

Substitution: $\Delta T_1 = 80 - 60 = 20^\circ\text{C}$; $\Delta T_2 = 50 - 30 = 20^\circ\text{C} \rightarrow \text{LMTD} = 20^\circ\text{C}$ (limiting value). Both streams change by 30°C , so heat capacity rates are equal; $\epsilon = (60 - 30)/(80 - 30) = 30/50$.

Result: LMTD = 20°C ; $\epsilon = 60\%$ – the statement is true.

Ref: BEE Guidebook Book 4, Ch. 4 (LMTD, effectiveness).

4. A 75 kW, 4-pole induction motor with a rated full-load efficiency of 94% draws 52.0 kW (measured with a power analyser) while driving a process pump. The motor is operating at about 65% of its rated load.

[1P4CEA004 | Chapter 5: Electric Motors and Variable Speed Drives | Motor loading from measured input power]

Answer: TRUE (1 mark)

Given: Rated output = 75 kW; $\eta_{FL} = 94\%$; measured input $P_i = 52.0$ kW.

Formula (Guidebook Ch. 5): Rated input $P_{ir} = \text{rated output} / \eta_{FL}$; % Load = $P_i / P_{ir} \times 100 = P_i \times \eta_{FL} / \text{rated output} \times 100$.

Substitution: $P_{ir} = 75/0.94 = 79.79$ kW; % Load = $52.0/79.79 \times 100$.

Result: % Load $\approx 65.2\%$ ($\approx 65\%$), so the statement is true. Note the method assumes efficiency at this load is close to the full-load value, which holds for standard motors between about 60% and 100% load.

Ref: BEE Guidebook Book 4, Ch. 5 (Determining motor loading).

5. In a closed chilled-water loop with no static head, reducing the speed of a centrifugal pump through a VSD from 1,480 rpm to 1,184 rpm reduces the pump shaft power to about 64% of its original value.

[1P4CEA005 | Chapter 7: Pumps | VSD in a closed loop – affinity laws]

Answer: FALSE (1 mark)

Given: $N_1 = 1,480$ rpm; $N_2 = 1,184$ rpm; closed loop, system curve passes through origin ($H \propto Q^2$), so the affinity laws apply along the system curve.

Formula (Guidebook Ch. 7 / Ch. 5 VSD): $Q_2/Q_1 = N_2/N_1$; $H_2/H_1 = (N_2/N_1)^2$; $P_2/P_1 = (N_2/N_1)^3$.

Substitution: $N_2/N_1 = 1,184/1,480 = 0.80$; $(0.80)^2 = 64.0\%$; $(0.80)^3 = 51.2\%$.

Result: Shaft power falls to **about 51 %**, not 64 % (64 % is the head ratio, from the square law). Statement is false.

Ref: BEE Guidebook Book 4, Ch. 7 (Pumps) and Ch. 5 (Application of VSD).

6. During a free air delivery (pump-up) test, if the energy auditor underestimates the combined volume of the receiver and the connecting pipeline, the computed compressor capacity will be lower than the actual capacity.

[1P4CEA006 | Chapter 8: Compressors | Pump-up (FAD) test – effect of receiver volume]

Answer: TRUE (1 mark)

Given (illustration): pressure ratio $(P_2 - P_1)/P_0 = 6.5$ (dimensionless); pump-up time $T = 4$ min; actual receiver + pipeline volume $V = 8.0 \text{ m}^3$; volume assumed by the auditor $= 7.2 \text{ m}^3$.

Formula (Guidebook Ch. 8, pump-up method): $Q = [(P_2 - P_1)/P_0] \times V/T \text{ Nm}^3/\text{min}$, where V = receiver + pipeline volume (m^3), T = time to pump up from P_1 to P_2 (min), P_0 = atmospheric pressure ($\text{kg/cm}^2 \text{ a}$).

Substitution (illustration): with $(P_2 - P_1)/P_0 = 6.5$ and $T = 4$ min, $V = 8.0 \text{ m}^3$ gives $Q = 13.0 \text{ Nm}^3/\text{min}$, but an underestimated $V = 7.2 \text{ m}^3$ gives only $11.7 \text{ Nm}^3/\text{min}$.

Result: Q is directly proportional to V , so **an underestimated volume understates FAD** (here by 10 %), wrongly suggesting that the compressor has deteriorated. Statement is true.

Ref: BEE Guidebook Book 4, Ch. 8 (Measurement of FAD by pump-up method).

7. In a water-cooled centrifugal chiller plant, raising the chilled water supply set point from 7°C to 9°C , at the same cooling load and condenser water temperature, generally increases the chiller specific power consumption (kW/TR).

[1P4CEA007 | Chapter 9: HVAC Systems | Chilled water set point and kW/TR]

Answer: FALSE (1 mark)

Concept (Guidebook Ch. 9): $\text{kW/TR} = \text{compressor input power (kW)} / \text{refrigeration load (TR)}$; $\text{COP} = \text{refrigeration effect (kW)} / \text{input (kW)} = 3.516/(\text{kW/TR})$.

Reasoning: A higher chilled water set point raises the evaporator (suction) temperature and pressure. The compressor works across a smaller lift (condensing minus evaporating temperature), so it needs less work per TR.

Result: kW/TR **decreases** (BEE's electrical-utilities guidebook (Book 3) puts the saving at roughly 3 % of compressor power per 1°C rise in evaporator temperature). The statement is false. The set point may be raised only if AHU coils can still meet the load and humidity needs.

Ref: BEE Guidebook Book 4, Ch. 9 (Performance evaluation of VCR systems).

8. An energy conservation project costing ₹ 50 lakh gives net annual savings of ₹ 15 lakh for 5 years. Since its NPV at a 12 % discount rate is positive, the IRR of the project must be lower than 12 %.

[1P4CEA008 | Chapter 10: Financial Analysis | NPV and IRR relationship]

Answer: FALSE (1 mark)

Given: Investment = ₹ 50,00,000; annual saving = ₹ 15,00,000; life $n = 5$ years; discount rate $i = 12\%$.

Formula (Guidebook Ch. 10): $\text{NPV} = \sum \text{CF}_t / (1 + i)^t - \text{Investment}$; IRR = discount rate at which $\text{NPV} = 0$.

Substitution: $\sum 1/1.12^t$ ($t = 1-5$) = 3.6048; $\text{NPV} = 15 \times 3.6048 - 50 = ₹ 4.07 \text{ lakh}$ (positive).

Result: A positive NPV at 12 % means the IRR is **higher** than 12 % (IRR $\approx 15.2\%$). Statement is false.

Ref: BEE Guidebook Book 4, Ch. 10 (Discounted cash flow methods).

9. An open-plan office floor of 600 m^2 has an average maintained illuminance of 300 lux with an installed lighting load of 7.2 kW. The target lux/W/m^2 for its room index is 42. The ILER of this installation falls in the 'review suggested' band.

[1P4CEA009 | Chapter 14: Buildings and Commercial Establishments | Installed Load Efficacy Ratio (ILER)]

Answer: TRUE (1 mark)

Given: Area = 600 m²; E_avg = 300 lux; installed load = 7,200 W; target = 42 lux/W/m².

Formula (Guidebook Ch. 14): Installed W/m² = load/area; actual lux/W/m² = E_avg/(W/m²); ILER = actual lux/W/m² ÷ target lux/W/m². Bands: ≥ 0.75 satisfactory to good; 0.51–0.74 review suggested; ≤ 0.50 urgent action required.

Substitution: W/m² = 7,200/600 = 12.0; lux/W/m² = 300/12 = 25.0; ILER = 25.0/42 = 0.595.

Result: ILER ≈ 0.60 → 'review suggested' (0.51–0.74), so the statement is true. Indicative annual wastage at 3,000 h = (1 – ILER) × 7.2 kW × 3,000 h ≈ 8,743 kWh.

Ref: BEE Guidebook Book 4, Ch. 14 (Performance assessment of lighting system, Tables 14.4–14.6).

10. A stenter processes 800 kg/h of fabric (bone-dry basis), reducing moisture from 0.90 to 0.06 kg/kg bone-dry fabric while the fabric temperature rises from 30 °C to 80 °C. It uses 1,200 kg/h of steam at 660 kcal/kg, and the condensate leaves at 90 °C. The dryer efficiency of this stenter is higher than 60 %.

[1P4CEA010 | Chapter 15: Textile Industry | Stenter (hot-air dryer) efficiency]

Answer: FALSE (1 mark)

Given: W = 800 kg/h bone-dry; m_in = 0.90; m_out = 0.06 kg/kg BD; T_in = 30 °C; T_out = 80 °C; steam 1,200 kg/h, h_s = 660 kcal/kg (≈ 2,763 kJ/kg), condensate 90 °C; latent heat 540 kcal/kg (≈ 2,261 kJ/kg).

Formula (Guidebook Ch. 15): Dryer efficiency % = $W \times (m_{in} - m_{out}) \times \{(T_{out} - T_{in}) + 540\} / Q_{in} \times 100$, where Q_{in} = steam flow × (h_s – h_condensate).

Substitution: Useful heat = 800 × 0.84 × (50 + 540) = 3,96,480 kcal/h (≈ 461 kW); Q_{in} = 1,200 × (660 – 90) = 6,84,000 kcal/h (= 795.3 kW).

Result: Dryer efficiency = 58.0 %, i.e. below 60 %, so the statement is false. Exhaust-air humidity control and condensate/flash heat recovery are the usual improvement options.

Ref: BEE Guidebook Book 4, Ch. 15 (Performance evaluation of stenter – hot air dryer).

..... End of Section - I

Section - II: SHORT NUMERICAL QUESTIONS

Marks: 2 x 5 = 10

- (i) Answer all the Two questions
- (ii) Each question carries Five marks

[1P4CEA011] (5 marks)

Chapter 8: Compressors | Pump-up FAD test with temperature correction and leakage test

L-1 A forging unit in Ludhiana runs a two-stage reciprocating air compressor rated at 13.5 m³/min FAD at 7 kg/cm²(g). During a Sunday shutdown, with all air users isolated, the energy auditor carried out a pump-up test followed by a leakage test with the plant's distribution network connected:

Parameter	Value
Receiver volume / pipeline and after-cooler hold-up volume	7.5 m ³ / 0.4 m ³
Pump-up: initial / final receiver pressure	0.5 / 7.2 kg/cm ² (g)
Atmospheric pressure	1.026 kg/cm ² (a)
Time to pump up	4 min 6 s
Inlet air temperature / air temperature in receiver	32 °C / 44 °C
Compressor power during loading	82 kW
Leakage test: average load time / unload time per cycle	18 s / 52 s
Annual operating hours / tariff	7,200 h / ₹ 8.50 per kWh

Calculate (a) the actual FAD corrected for temperature and the capacity shortfall, (b) the specific power consumption, and (c) the leakage quantity, percentage leakage and the annual cost of the leakage.

MODEL ANSWER

Given: $V = 7.5 + 0.4 = 7.9 \text{ m}^3$; $P_2 - P_1 = 7.2 - 0.5 = 6.7 \text{ kg/cm}^2 (\approx 6.57 \text{ bar})$; $P_o = 1.026 \text{ kg/cm}^2(\text{a}) (\approx 1.01 \text{ bar(a)})$; $T = 4 \text{ min}$ $6 \text{ s} = 4.1 \text{ min}$; $t_1 = 32^\circ\text{C}$; $t_2 = 44^\circ\text{C}$.

Formula (Guidebook Ch. 8, pump-up method): $Q = [(P_2 - P_1)/P_o] \times V/T \text{ m}^3/\text{min}$; $Q_{\text{actual}} = Q \times (273 + t_1)/(273 + t_2)$; specific power = $\text{kW}/(\text{m}^3/\text{min})$; leakage = $Q_{\text{actual}} \times T_{\text{load}}/(T_{\text{load}} + t_{\text{unload}})$.

(a) Substitution: $Q = (6.7/1.026) \times 7.9/4.1 = \mathbf{12.58 \text{ m}^3/\text{min}}$; $Q_{\text{actual}} = 12.58 \times 305/317 = \mathbf{12.11 \text{ m}^3/\text{min}}$ (726.4 m^3/h). Shortfall = $(13.5 - 12.11)/13.5 \times 100 = \mathbf{10.3 \%}$ (2 marks)

(b) Specific power = $82/12.11 = \mathbf{6.77 \text{ kW per m}^3/\text{min}}$ ($= 0.1129 \text{ kWh}/\text{m}^3$) (1 mark)

(c) % leakage = $18/(18 + 52) \times 100 = \mathbf{25.71 \%}$; leakage = $12.11 \times 0.2571 = \mathbf{3.11 \text{ m}^3/\text{min}}$. Power lost = $3.11 \times 6.77 = 21.09 \text{ kW} \rightarrow 1,51,817 \text{ kWh/year} \rightarrow \mathbf{₹ 12,90,446 (\text{₹ 12.90 lakh}) per year}$ (2 marks)

Result/comment: The compressor delivers about 10 % below rating (check valves, piston rings and inter-cooler) and a quarter of its output is lost as leakage. A leakage-arresting drive should come first, as it is a no-cost or low-cost measure.

[1P4CEA012] (5 marks)

Chapter 6: Fans and Blowers | Pitot-tube traverse, flow and static efficiency of a PA fan

L-2 The primary air fan of a 150 TPH coal-fired boiler in a Gujarat chemical plant draws air at 45°C through a 1.8 m diameter round duct. A 10-point pitot-tube traverse (pitot coefficient 0.85) gave the following velocity pressures in mmWC:

Point	1	2	3	4	5	6	7	8	9	10
Δp (mmWC)	18.6	20.4	22.1	23.5	24.2	24.0	23.1	21.8	20.0	18.3

Fan suction static pressure = -28 mmWC ; discharge static pressure = $+165 \text{ mmWC}$. Motor input power = 118 kW ; motor efficiency at this load = 93% ; fan is directly coupled. Take air density at 0°C as $1.293 \text{ kg}/\text{m}^3$. Calculate the average air velocity, volume flow (m^3/s and Nm^3/h) and the static efficiency of the fan.

MODEL ANSWER

Given: $D = 1.8 \text{ m}$; $t = 45^\circ\text{C}$; $C_p = 0.85$; $\Delta P_s = 165 - (-28) = 193 \text{ mmWC} (\approx 1.89 \text{ kPa})$; motor input = 118 kW ; $\eta_{\text{motor}} = 93 \%$; transmission efficiency = 1 (direct).

Formula (Guidebook Ch. 6): $\gamma = 1.293 \times 273/(273 + t)$; velocity at each point $V = C_p \times \sqrt{2 \times 9.81 \times \Delta p/\gamma}$; average velocity = arithmetic mean of point velocities; $Q = A \times V$; static efficiency = $Q \times \Delta P_s / (102 \times \text{fan shaft power}) \times 100$.

Substitution: $\gamma = 1.293 \times 273/318 = \mathbf{1.110 \text{ kg}/\text{m}^3}$. Point velocities (m/s): 15.41, 16.14, 16.80, 17.32, 17.58, 17.51, 17.18, 16.69, 15.98, 15.29. Mean velocity = $\mathbf{16.59 \text{ m/s}}$ (2 marks). (Using the average Δp instead gives 16.61 m/s – a small but avoidable error.)

Result: $A = \pi \times 1.8^2/4 = 2.5447 \text{ m}^2$; $Q = 2.5447 \times 16.59 = \mathbf{42.21 \text{ m}^3/\text{s}}$ = $1,51,971 \text{ m}^3/\text{h}$; at NTP = $1,51,971 \times 273/318 = \mathbf{1,30,466 \text{ Nm}^3/\text{h}}$ (1 mark)

Result: Fan shaft power = $118 \times 0.93 = 109.74 \text{ kW}$; static efficiency = $42.21 \times 193 / (102 \times 109.74) \times 100 = \mathbf{72.8 \%}$ (2 marks)

Comment: A static efficiency in the low 70s is acceptable for a backward-curved PA fan. Compare the measured flow against coal-mill requirement before considering inlet damper to VFD conversion.

..... End of Section - II

Section - III: LONG NUMERICAL QUESTIONS

Marks: 4 x 20 = 80

(i) Answer all the Four questions

- (ii) Each question carries Twenty marks
- (iii) In N-4, answer any ONE of the four alternatives (a) to (d)

[1P4CEA013] (20 marks)

Chapter 1: Boilers | Indirect method efficiency of a coal-fired boiler with combustion tuning

N-1 A 40 TPH travelling-grate, coal-fired water-tube boiler in a kraft paper mill in Tamil Nadu supplies steam at 32 kg/cm²(g) and 400 °C to a back-pressure turbine. The accredited energy auditor carried out an 8-hour indirect-method test. The average test data are given below:

Parameter	Value
Ultimate analysis of coal (as fired), % by mass	C 41.2, H ₂ 2.8, N ₂ 1.1, O ₂ 8.6, S 0.5, Moisture 8.4, Ash 37.4
GCV of coal (as fired)	3,950 kcal/kg
O ₂ in flue gas at air-preheater outlet (dry basis)	5.4 %
CO in flue gas	Negligible (< 50 ppm)
Flue gas temperature at air-preheater outlet	178 °C
Ambient (dry bulb) temperature	31 °C
Humidity in ambient air	0.0185 kg/kg dry air
Specific heat of dry flue gas / superheated vapour	0.24 / 0.45 kcal/kg °C
Ratio of bottom ash to fly ash (by mass)	20 : 80
GCV of bottom ash / fly ash	650 / 380 kcal/kg
Surface (radiation & convection) and unaccounted loss (water-tube boiler)	2.2 %
Steam generation / steam enthalpy	40,000 kg/h at 771 kcal/kg
Feed water temperature	105 °C
Operating hours / landed cost of coal	8,000 h/year / ₹ 5,200 per tonne

- a) Determine the theoretical air requirement, % excess air and actual mass of air supplied per kg of coal. **[4 Marks]**
- b) Determine the actual mass of dry flue gas per kg of coal from its constituents. **[3 Marks]**
- c) Calculate all the boiler losses as per the BEE indirect method. **[7 Marks]**
- d) Find the boiler efficiency by the indirect method, the coal consumption (kg/h) and the evaporation ratio. **[3 Marks]**
- e) Combustion tuning and soot-blowing bring the O₂ down to 3.5 % and the flue gas temperature down to 165 °C (all other data unchanged). Estimate the new efficiency and the annual coal and cost savings for the same steam output. **[3 Marks]**

MODEL ANSWER**a) Theoretical air, excess air and actual air (4 marks)**

Given: C = 41.2 %, H₂ = 2.8 %, O₂ = 8.6 %, S = 0.5 %, N₂ = 1.1 %; flue gas O₂ = 5.4 %.

Formula: Theoretical air = $[11.6 C + 34.8 (H_2 - O_2/8) + 4.35 S]/100$ kg/kg coal; EA % = $O_2 \% / (21 - O_2 \%) \times 100$; AAS = $(1 + EA/100) \times$ theoretical air.

Substitution: TA = $[11.6 \times 41.2 + 34.8 \times (2.8 - 8.6/8) + 4.35 \times 0.5]/100 = [477.92 + 60.03 + 2.18]/100 = 5.401$ kg/kg coal (1.5 marks)

EA = $5.4/(21 - 5.4) \times 100 = 34.62$ % (1 mark)

AAS = $(1 + 34.62/100) \times 5.401 = 7.271$ kg/kg coal (1.5 marks)

b) Mass of dry flue gas (3 marks)

Formula: m = mass of CO₂ + SO₂ + N₂ in fuel + N₂ in air supplied + O₂ in flue gas = $C \times 44/12 + S \times 64/32 + N_2 + AAS \times 0.77 + (AAS - TA) \times 0.23$ (per kg coal, fractions).

Substitution: $\text{CO}_2 = 0.412 \times 44/12 = 1.5107$; $\text{SO}_2 = 0.005 \times 64/32 = 0.0100$; $\text{N}_2 (\text{fuel}) = 0.0110$; $\text{N}_2 (\text{air}) = 7.271 \times 0.77 = 5.5986$; $\text{O}_2 (\text{excess}) = (7.271 - 5.401) \times 0.23 = 0.4300$.

Result: $m = 7.560 \text{ kg dry flue gas/kg coal}$ (3 marks)

c) Boiler losses (7 marks)

Given: $T_f - T_a = 178 - 31 = 147^\circ\text{C}$; $\text{GCV} = 3,950 \text{ kcal/kg}$ ($= 16,538 \text{ kJ/kg}$).

Loss	Formula (Guidebook Ch. 1)	Substitution	Result
L ₁ Dry flue gas	$m \times C_p \times (T_f - T_a) / \text{GCV} \times 100$	$7.560 \times 0.24 \times 147 / 3,950 \times 100$	6.75 %
L ₂ H ₂ in fuel	$9 \text{ H}_2 \times \{584 + C_p(T_f - T_a)\} / \text{GCV} \times 100$	$9 \times 0.028 \times (584 + 0.45 \times 147) / 3,950 \times 100$	4.15 %
L ₃ Moisture in fuel	$M \times \{584 + C_p(T_f - T_a)\} / \text{GCV} \times 100$	$0.084 \times (584 + 66.15) / 3,950 \times 100$	1.38 %
L ₄ Moisture in air	$\text{AAS} \times \text{humidity} \times C_p \times (T_f - T_a) / \text{GCV} \times 100$	$7.271 \times 0.0185 \times 0.45 \times 147 / 3,950 \times 100$	0.23 %
L ₅ CO (partial combustion)	Negligible ($\text{CO} < 50 \text{ ppm}$)	–	0.00 %
L ₆ Surface & unaccounted	Given (water-tube boiler)	–	2.20 %
L ₇ Unburnt in fly ash	$\text{Fly ash/kg coal} \times \text{GCV}_{\text{fa}} / \text{GCV} \times 100$	$(0.374 \times 0.80) \times 380 / 3,950 \times 100$	2.88 %
L ₈ Unburnt in bottom ash	$\text{Bottom ash/kg coal} \times \text{GCV}_{\text{ba}} / \text{GCV} \times 100$	$(0.374 \times 0.20) \times 650 / 3,950 \times 100$	1.23 %
Total losses			18.82 %

Marking: L₁ 2 marks; L₂, L₃ 1 mark each; L₄ 1 mark; L₇ + L₈ 1.5 marks; total 0.5 mark.

d) Efficiency, coal consumption and evaporation ratio (3 marks)

Formula: $\eta = 100 - (L_1 + \dots + L_8)$; $\text{coal (kg/h)} = Q \times (h_g - h_f) / (\eta \times \text{GCV})$; $\text{ER} = \text{steam/coal}$.

Substitution: $\eta = 100 - 18.82 = 81.18 \%$ (1 mark)

$\text{Coal} = 40,000 \times (771 - 105) / (0.8118 \times 3,950) = 2,66,40,000 / 3,206.7 = 8,308 \text{ kg/h (8.31 TPH)}$ (1 mark). $\text{Heat output} = 2,66,40,000 \text{ kcal/h} = 30,982 \text{ kW}$.

$\text{ER} = 40,000 / 8,308 = 4.81 \text{ kg steam/kg coal}$ (1 mark)

e) Effect of combustion tuning (3 marks)

Given: $\text{O}_2 = 3.5 \%$, $T_f = 165^\circ\text{C}$ ($\Delta T = 134^\circ\text{C}$), other data unchanged.

Substitution: $\text{EA} = 3.5 / 17.5 \times 100 = 20.0 \%$; $\text{AAS} = 1.20 \times 5.401 = 6.481 \text{ kg/kg}$; $m = 1.5107 + 0.0100 + 0.0110 + 4.9908 + 0.2485 = 6.771 \text{ kg/kg}$.

$L_1 = 5.51 \%$, $L_2 = 4.11 \%$, $L_3 = 1.37 \%$, $L_4 = 0.18 \%$; L_6, L_7, L_8 unchanged $\rightarrow \eta_{\text{new}} = 82.51 \%$ (1.5 marks)

$\text{New coal} = 2,66,40,000 / (0.8251 \times 3,950) = 8,173 \text{ kg/h} \rightarrow \text{saving} = 134.1 \text{ kg/h} \times 8,000 \text{ h} = 1,073 \text{ t/year}$, worth ₹ 55.78 lakh/year (₹ 55,78,271) (1.5 marks)

Comment: dry flue gas loss is the largest controllable loss; unburnt-in-ash losses ($\approx 4.1 \%$) are next, so grate speed, coal sizing and fly-ash carbon should be tracked in the monthly energy audit.

[1P4CEA014] (20 marks)

Chapter 3: Cogeneration Systems | Extraction-cum-back-pressure turbine cogeneration – stage efficiency, EUF and economics

N-2 A 300 TPD integrated paper mill in Uttarakhand runs a coal-fired boiler feeding an extraction-cum-back-pressure steam turbine. Medium-pressure (MP) steam is extracted for the digester and low-pressure (LP) exhaust steam goes to the paper machine dryers. Condensate from both users returns to the deaerator at 90°C . Average test data over 8 hours:

Parameter	Value
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Turbine inlet steam: 64 kg/cm ² (a), 485 °C, flow	72 TPH, $h_1 = 811.0$ kcal/kg
MP extraction at 11 kg/cm ² (a): flow / actual enthalpy	28 TPH / 707.5 kcal/kg
Isentropic enthalpy at extraction pressure (from inlet state)	690.4 kcal/kg
LP exhaust at 4.5 kg/cm ² (a): flow / actual enthalpy	44 TPH / 668.2 kcal/kg
Isentropic enthalpy at exhaust pressure (from extraction state)	652.0 kcal/kg
Gear box × mechanical efficiency / generator efficiency	98 % / 96.5 %
Boiler feed water temperature / boiler efficiency (GCV)	115 °C / 78 %
GCV of coal / landed cost	3,800 kcal/kg / ₹ 5,800 per tonne
Grid tariff (avoided cost) / annual operation	₹ 7.90 per kWh / 8,000 h
Alternative: process steam from a low-pressure boiler	Efficiency 82 % on the same coal

- Determine the stage (isentropic) efficiency of the HP (inlet to extraction) and LP (extraction to exhaust) sections. **[4 Marks]**
- Calculate the turbine shaft power and the generator output. **[4 Marks]**
- Estimate the coal consumption, the overall plant fuel rate (kg/kWh) and the overall plant heat rate (kcal/kWh) as defined in the guidebook. **[4 Marks]**
- Determine the Energy Utilisation Factor (EUF) and the heat-to-power ratio of the cogeneration system. **[4 Marks]**
- Compare the annual fuel-plus-power cost with separate generation (grid power + low-pressure boiler for the same process heat) and find the simple payback if the cogeneration package cost ₹ 95 crore more than the low-pressure boiler. **[4 Marks]**

MODEL ANSWER

a) Stage (isentropic) efficiency (4 marks)

Given: $h_1 = 811.0$ kcal/kg ($\approx 3,395$ kJ/kg), $h_x = 707.5$ kcal/kg ($\approx 2,962$ kJ/kg), $h_{xs} = 690.4$ kcal/kg ($\approx 2,891$ kJ/kg), $h_e = 668.2$ kcal/kg ($\approx 2,798$ kJ/kg), $h_{es} = 652.0$ kcal/kg ($\approx 2,730$ kJ/kg).

Formula (Guidebook Ch. 3): $\eta_{\text{stage}} = \text{actual enthalpy drop} / \text{isentropic enthalpy drop} \times 100$.

Substitution: HP section = $(811.0 - 707.5) / (811.0 - 690.4) \times 100 = 103.5 / 120.6 \times 100 = \mathbf{85.82 \%}$ (2 marks)

Result: LP section = $(707.5 - 668.2) / (707.5 - 652.0) \times 100 = 39.3 / 55.5 \times 100 = \mathbf{70.81 \%}$ (2 marks). The LP section is weak (worn nozzles/blades or seal leakage) and should be inspected at the next shutdown.

b) Turbine shaft power and generator output (4 marks)

Given: $m_1 = 72,000$ kg/h through HP section; $m_e = 44,000$ kg/h through LP section; $\eta_{\text{gearbox} \times \text{mech}} = 0.98$; $\eta_{\text{gen}} = 0.965$; $1 \text{ kW} = 860 \text{ kcal/h}$.

Formula: $P_{\text{shaft}} (\text{kW}) = [m_1 (h_1 - h_x) + m_e (h_x - h_e)] / 860$; $P_{\text{gen}} = P_{\text{shaft}} \times \eta_{\text{gearbox}} \times \eta_{\text{gen}}$.

Substitution: HP: $72,000 \times 103.5 = 7,452,000 \text{ kcal/h}$ ($\approx 8,665 \text{ kW}$); LP: $44,000 \times 39.3 = 1,729,200 \text{ kcal/h}$ ($\approx 2,011 \text{ kW}$); total = $9,181,200 \text{ kcal/h} \div 860 = \mathbf{10,676 \text{ kW}}$ (2.5 marks)

Result: $P_{\text{gen}} = 10,676 \times 0.98 \times 0.965 = \mathbf{10,096 \text{ kW} \approx 10.10 \text{ MW}}$ (1.5 marks)

c) Coal consumption, fuel rate and plant heat rate (4 marks)

Given: feed water 115°C ($h_{\text{fw}} \approx 115 \text{ kcal/kg}$ ($\approx 481 \text{ kJ/kg}$)); $\eta_{\text{boiler}} = 78 \%$; GCV = $3,800 \text{ kcal/kg}$ ($15,910 \text{ kJ/kg}$).

Formula: Fuel heat input = $m_1 (h_1 - h_{\text{fw}}) / \eta_{\text{boiler}}$; overall plant fuel rate = coal (kg/h)/power (kW); overall plant heat rate = fuel rate $\times$ GCV.

Substitution: $Q_{\text{fuel}} = 72,000 \times (811 - 115) / 0.78 = \mathbf{6,42,46,154 \text{ kcal/h}}$ ($= 74,705 \text{ kW}$; 269.0 GJ/h) (1 mark)

Result: Coal = $6,42,46,154 / 3,800 = \mathbf{16,907 \text{ kg/h (16.91 TPH)}}$ (1 mark)

Result: Fuel rate = $16,907/10,096 = 1.675 \text{ kg/kWh}$; plant heat rate = $1.675 \times 3,800 = 6,363 \text{ kcal/kWh} (\approx 26,641 \text{ kJ/kWh})$ (2 marks). (The high figure is expected: most of the fuel heat leaves as process steam, so heat rate alone is not a fair index for a back-pressure cogeneration plant; EUF is.)

d) EUF and heat-to-power ratio (4 marks)

Given: condensate return 90°C ($h_c \approx 90 \text{ kcal/kg} (\approx 377 \text{ kJ/kg})$); $m_x = 28,000 \text{ kg/h}$; $m_e = 44,000 \text{ kg/h}$.

Formula: Process heat = $m_x (h_x - h_c) + m_e (h_e - h_c)$; EUF = $(P_{\text{gen}} \times 860 + \text{process heat})/\text{fuel heat input}$;
HPR = $\text{process heat}/(P_{\text{gen}} \times 860)$.

Substitution: Process heat = $28,000 \times (707.5 - 90) + 44,000 \times (668.2 - 90) = 1,72,90,000 + 2,54,40,800 = 4,27,30,800 \text{ kcal/h} (= 49,687 \text{ kW})$ (1.5 marks)

Result: EUF = $(10,096 \times 860 + 4,27,30,800)/6,42,46,154 \times 100 = 80.0 \%$ (1.5 marks)

Result: HPR = $4,27,30,800/(10,096 \times 860) = 4.92$ (1 mark)

e) Economics versus separate generation (4 marks)

Given: coal ₹ 5,800/t; grid ₹ 7.90/kWh; 8,000 h/year; LP boiler efficiency 82 %; extra investment ₹ 95 crore.

Formula: annual cost = fuel (t/h) $\times$ ₹/t $\times$ h (+ grid kWh $\times$ tariff for separate case); simple payback = investment/net annual benefit.

Result: **Cogeneration:** coal = $16.9069 \text{ t/h} \times ₹ 5,800 \times 8,000 \text{ h} = ₹ 78.45 \text{ crore/year}$ (1 mark)

Result: **Separate:** LP-boiler coal = $4,27,30,800/(0.82 \times 3,800) = 13,713 \text{ kg/h} \rightarrow ₹ 63,62,99,461$; grid power = $10,096 \text{ kW} \times 8,000 \text{ h} \times ₹ 7.90 = ₹ 63,80,74,611$; total = ₹ 127.44 crore/year (1.5 marks)

Result: **Net benefit** = ₹ 48.99 crore/year; **simple payback** = $95/48.99 = 1.94 \text{ years} \approx 23 \text{ months}$ (1.5 marks). (O&M and capital charges are excluded, as in a first-cut audit estimate.)

[1P4CEA015] (20 marks)

Chapter 9: HVAC Systems | Chiller plant performance with chilled water pump and VFD retrofit

N-3 An automobile paint shop near Pune is cooled by a 900 TR water-cooled centrifugal chiller with a constant-speed secondary chilled water pump. To hold the design flow, the pump discharge valve is kept partly closed. An energy auditor recorded the following steady-state readings (ultrasonic flow meters, calibrated RTDs, power analysers):

Parameter	Value
Chilled water flow / return / supply temperature	420 m ³ /h / 12.6 °C / 7.2 °C
Chiller compressor input power	432 kW
Condenser water flow / inlet / outlet temperature	520 m ³ /h / 30.5 °C / 35.7 °C
Chilled water pump: discharge / suction pressure (gauge)	4.2 / 0.3 kg/cm ² (g)
Elevation of discharge gauge above suction gauge	0.8 m
Pressure drop across the throttled discharge valve	1.2 kg/cm ² (12 m)
Pump motor: rating / measured input / efficiency at this load	90 kW / 68.5 kW / 93.5 %
Operating hours / electricity tariff	7,200 h/year / ₹ 8.40 per kWh

Take density of water = 1,000 kg/m³, specific heat = 1 kcal/kg °C, 1 TR = 3,024 kcal/h and 1 kW = 860 kcal/h.

- Determine the refrigeration load (TR), specific power consumption (kW/TR) and COP of the chiller. **[5 Marks]**
- Check the heat balance of the chiller across the evaporator and condenser and state whether the test data are acceptable ($\pm 5 \%$ criterion). **[3 Marks]**
- Determine the pump head, hydraulic power, pump efficiency and percentage loading of the pump motor. **[5 Marks]**

d) The auditor proposes to open the valve fully and fit a VFD (efficiency 97 %) to deliver the same flow at the reduced head. Assuming pump and motor efficiencies remain the same, estimate the new motor input, annual energy savings and monetary savings. **[4 Marks]**

e) If the chilled water supply set point is raised from 7.2 °C to 9.0 °C after a trial confirms the paint booths stay within limits, estimate the annual compressor energy and cost savings (assume 3 % reduction in compressor power per °C rise). **[3 Marks]**

MODEL ANSWER

a) Refrigeration load, kW/TR and COP (5 marks)

Given: $Q_{chw} = 420 \text{ m}^3/\text{h} = 4,20,000 \text{ kg/h}$; $\Delta T = 12.6 - 7.2 = 5.4 \text{ }^\circ\text{C}$; $P = 432 \text{ kW}$.

Formula (Guidebook Ch. 9): Refrigeration load (TR) = $Q \times \rho \times C_p \times \Delta T / 3,024$; kW/TR = P / TR ; COP = refrigeration effect (kW)/input (kW).

Substitution: Load = $4,20,000 \times 1 \times 5.4 = 22,68,000 \text{ kcal/h}$ ($= 2,637.2 \text{ kW}$; $= 9,496 \text{ MJ/h}$) $\rightarrow \text{TR} = 22,68,000 / 3,024 = 750.0 \text{ TR} (\approx 2,637 \text{ kW})$ (2 marks)

kW/TR = $432 / 750.0 = 0.576 \text{ kW/TR}$ (1.5 marks); COP = $2,637.2 / 432 = 6.10$ (1.5 marks)

b) Heat balance check (3 marks)

Given: condenser water $520 \text{ m}^3/\text{h}$, $30.5 \rightarrow 35.7 \text{ }^\circ\text{C}$; refrigeration load and compressor input from (a).

Formula: Heat rejected in condenser $\approx$ refrigeration load + compressor work; error % = (measured – expected)/expected $\times 100$.

Substitution: Measured = $5,20,000 \times (35.7 - 30.5) = 27,04,000 \text{ kcal/h}$ ($\approx 3,144 \text{ kW}$); expected = $22,68,000 + 432 \times 860 = 26,39,520 \text{ kcal/h}$ ($\approx 3,069 \text{ kW}$).

Result: Error = **+2.44 %**, within $\pm 5 \%$, so the measured flows, temperatures and power are consistent and the kW/TR result can be relied on. (3 marks)

c) Pump head, efficiency and motor loading (5 marks)

Given: $P_d = 4.2 \text{ kg/cm}^2(\text{g}) (\approx 4.12 \text{ bar(g)})$, $P_s = 0.3 \text{ kg/cm}^2(\text{g})$ ($1 \text{ kg/cm}^2 \approx 10 \text{ m water} \approx 98.1 \text{ kPa}$); gauge elevation difference 0.8 m ; $Q = 420 \text{ m}^3/\text{h}$; motor input 68.5 kW ; $\eta_{\text{motor}} 93.5 \%$; rating 90 kW .

Formula (Guidebook Ch. 7): Head $H = (P_d - P_s) \times 10 + \text{elevation correction (m)}$; hydraulic power $P_h (\text{kW}) = Q (\text{m}^3/\text{s}) \times \rho \times g \times H / 1,000$; pump efficiency = $P_h / (\text{motor input} \times \eta_{\text{motor}}) \times 100$; motor load = shaft power / rated output $\times 100$.

Substitution: $H = (4.2 - 0.3) \times 10 + 0.8 = 39.8 \text{ m}$ (1 mark)

$P_h = (420 / 3,600) \times 1,000 \times 9.81 \times 39.8 / 1,000 = 45.55 \text{ kW}$ (1.5 marks)

Shaft power = $68.5 \times 0.935 = 64.05 \text{ kW}$; $\eta_{\text{pump}} = 45.55 / 64.05 \times 100 = 71.12 \%$ (1.5 marks)

Motor loading = $64.05 / 90 \times 100 = 71.2 \%$ (1 mark)

d) Valve opening with VFD (4 marks)

Given: same flow $420 \text{ m}^3/\text{h}$; head required = $39.8 - 12.0 = 27.8 \text{ m}$; $\eta_{\text{pump}} = 71.12 \%$, $\eta_{\text{motor}} = 93.5 \%$, $\eta_{\text{VFD}} = 97 \%$.

Formula: New input = $Q \times \rho \times g \times H_{\text{new}} / (\eta_{\text{pump}} \times \eta_{\text{motor}} \times \eta_{\text{VFD}})$.

Substitution: $P_{h,\text{new}} = (420 / 3,600) \times 9.81 \times 27.8 = 31.82 \text{ kW} \rightarrow \text{input} = 31.82 / (0.7112 \times 0.935 \times 0.97) = 49.33 \text{ kW}$ (2 marks)

Savings = $68.5 - 49.33 = 19.17 \text{ kW} \times 7,200 \text{ h} = 1,38,049 \text{ kWh/year}$, worth **₹ 11,59,611 (₹ 11.60 lakh) per year** (2 marks)

e) Raising the chilled water set point (3 marks)

Given: set point $7.2 \rightarrow 9.0$ °C; 3 % compressor power reduction per °C; $P = 432$ kW; 7,200 h; ₹ 8.40/kWh. **Formula:** power saved = $P \times 0.03 \times \Delta t$; annual saving = kW $\times$ h $\times$ tariff.

Substitution: rise = $9.0 - 7.2 = 1.8$ °C $\rightarrow$ reduction = $3 \% \times 1.8 = 5.4 \%$; power saved = $432 \times 0.054 = \mathbf{23.33 \text{ kW}}$ (1 mark)

Annual savings = $23.33 \times 7,200 = \mathbf{1,67,962 \text{ kWh/year}}$, worth ₹ **14,10,877 (₹ 14.11 lakh) per year** (2 marks). New kW/TR ≈ 0.545 at the same load.

[1P4CEA016] (20 marks)

Chapter 13: Cement Industry | Answer any ONE: (a) cement plant, (b) thermal power unit, (c) steel reheating furnace, (d) commercial building
N-4 Answer any ONE of the following among the four questions given below (20 marks).

N-4 (a) Cement plant – kiln, electricity and preheater leaking air

A 5,000 TPD clinker dry-process plant in Rajasthan (5-stage preheater with precalciner) recorded the following annual data:

Parameter	Value
Clinker produced / cement dispatched	16,00,000 t / 22,40,000 t
Coal fired in kiln + precalciner (GCV 5,450 kcal/kg)	2,20,000 t
Grid power imported	400 lakh kWh
CPP gross generation / auxiliary consumption	1,050 lakh kWh / 9 %
WHR power gross generation / auxiliary consumption	380 lakh kWh / 6 %
Power supplied to colony and township	22 lakh kWh
O ₂ at precalciner outlet / at preheater (PH) outlet	2.4 % / 4.8 % (dry)
PH fan: gas flow at PH outlet / input power (VFD-driven)	3,45,000 Nm ³ /h / 2,650 kW
Operating hours / power cost	7,800 h / ₹ 7.20 per kWh

a) Calculate the specific thermal energy consumption (kcal/kg clinker) and the clinker factor. **[3 Marks]**

b) Calculate the net electricity used for cement production and the specific electrical energy consumption (kWh/t cement). Comment against a typical value of about 77 kWh/t cement. **[5 Marks]**

c) Determine the percentage leaking (false) air across the preheater string as per the guidebook. **[3 Marks]**

d) After sealing PH ports and kiln inlet, leaking air falls to 5 %. With the same gas leaving the precalciner and the PH fan on VFD (assume fan power $\propto \text{flow}^3$), find the new PH outlet O₂, the new fan power and the annual energy and cost savings. **[6 Marks]**

e) Compute the overall specific energy consumption in kcal/kg cement, converting electricity at 860 kcal/kWh. **[3 Marks]**

OR

N-4 (b) 210 MW coal-fired unit – heat rate and condenser performance

A 210 MW sub-critical unit in Chhattisgarh was tested over 4 hours at steady load:

Parameter	Value
Gross generation (average)	205 MW
Coal consumption / GCV	138 t/h / 3,750 kcal/kg
Boiler efficiency (indirect method, GCV basis)	86.8 %
Auxiliary power consumption	8.6 % of gross generation
Condenser CW flow (measured, ultrasonic)	25,200 m ³ /h
CW inlet / outlet temperature	32.4 °C / 41.6 °C
Condenser back pressure / saturation temperature	0.11 ata / 47.4 °C
Coal cost / annual operating hours	₹ 4,300 per tonne / 7,500 h

Guidebook Table 11.7 (typical 210 MW turbine cycle): heat rate 2,000 kcal/kWh at 0.11 ata; 1,975 kcal/kWh at 0.09 ata. Assume 96 % of (turbine heat rate – 860) kcal/kWh is rejected in the condenser.

- Calculate the gross heat rate, turbine heat rate, net heat rate and turbine cycle efficiency. **[6 Marks]**
- Determine the condenser TTD, initial temperature difference, LMTD, effectiveness and heat load (kcal/h and kcal/kWh). **[5 Marks]**
- Estimate the CW flow implied by the turbine heat balance and comment on the measured CW flow. **[4 Marks]**
- Cooling tower repairs bring the back pressure down to 0.09 ata. Using the relative improvement from Table 11.7, estimate the new turbine and gross heat rates and the annual coal and cost savings at the same generation. **[5 Marks]**

OR

N-4 (c) Steel re-rolling mill – billet reheating furnace with recuperator

A 30 TPH pusher-type billet reheating furnace in a secondary steel plant at Mandi Gobindgarh is fired with furnace oil. A metallic counter-flow recuperator preheats the combustion air. Test data:

Parameter	Value
Billet throughput / charging / discharge temperature	30,000 kg/h / 40 °C / 1,180 °C
Mean specific heat of billets	0.12 kcal/kg °C
Furnace oil consumption / GCV	1,350 kg/h / 10,200 kcal/kg
Theoretical air for furnace oil	14.1 kg/kg oil
O ₂ in flue gas (dry)	7.5 %
Flue gas temperature at recuperator inlet / outlet	980 °C / 540 °C
Combustion air temperature at recuperator inlet	35 °C
C _p flue gas / C _p air	0.26 / 0.24 kcal/kg °C
Heat loss from recuperator shell	10 % of heat given up by flue gas
Overall heat transfer coefficient (gas-to-air)	20 kcal/h m ² °C
Operation / furnace oil cost	6,000 h/year / ₹ 52,000 per tonne

- Determine the furnace efficiency by the direct method and the specific fuel consumption. **[4 Marks]**
- Determine the excess air, actual air and flue gas quantity (kg/h). **[4 Marks]**
- Determine the heat recovered by the recuperator and the combustion-air preheat temperature. **[4 Marks]**
- Determine the LMTD, effectiveness and heat transfer area of the recuperator. **[4 Marks]**
- Considering only the sensible heat in flue gas leaving the recuperator, estimate the stack loss now and if O₂ is reduced to 3 % (same temperatures), and the resulting fuel and cost savings. **[4 Marks]**

OR

N-4 (d) IT office building – EPI, lighting ILER and HVAC interaction

A 12-floor IT office campus building in Bengaluru (temperate climate zone) has the following data:

Parameter	Value
Built-up area	32,500 m ²
Annual grid (EB) energy / DG energy	41,60,000 kWh / 2,40,000 kWh
Operation	12 h/day, 300 days/year
Typical floor: area / average maintained illuminance	1,850 m ² / 380 lux
Installed lighting load per floor (T8 fluorescent)	21.5 kW
Target lux/W/m ² for the room index (Table 14.4)	46
LED retrofit per floor: installed load / illuminance	12.4 kW / 400 lux
Chiller plant SEC including auxiliaries	1.05 kW/TR
Commercial tariff / LED retrofit cost (12 floors)	₹ 9.20 per kWh / ₹ 58 lakh

- a) Calculate the EPI (kWh/m²/year) and AAHEPI (Wh/h/m²). **[4 Marks]**
- b) Calculate the ILER of a typical floor, state its assessment band and estimate the annual energy wastage for all 12 floors as per the guidebook. **[6 Marks]**
- c) After the LED retrofit, find the new ILER and the annual lighting energy and cost savings for 12 floors. **[5 Marks]**
- d) All lighting heat ends up as cooling load. Estimate the reduction in cooling load (TR), the additional chiller-plant energy and cost savings, the simple payback of the retrofit and the new EPI. **[5 Marks]**

MODEL ANSWER

N-4 (a) Solution — Cement plant

a) Specific thermal energy consumption and clinker factor (3 marks)

Given: Coal = 2,20,000 t; GCV = 5,450 kcal/kg (22,818 kJ/kg); clinker = 16,00,000 t; cement = 22,40,000 t.

Formula: STEC (kcal/kg clinker) = coal (kg) × GCV/clinker (kg); clinker factor = clinker/cement.

Substitution: STEC = $2,20,000 \times 1,000 \times 5,450 / (16,00,000 \times 1,000) = 11,99,00,00,00,000 / 1,60,00,00,000$; clinker factor = $16,00,000 / 22,40,000$.

Result: STEC = **749.4 kcal/kg clinker** (= 3,137 kJ/kg = 3.137 GJ/t) (2 marks); **clinker factor = 0.714** (cement/clinker = 1.40) (1 mark).

b) Net electricity and SEEC (5 marks)

Given: Grid = 400 lakh kWh; CPP = 1,050 lakh kWh gross, APC 9 %; WHR = 380 lakh kWh gross, APC 6 %; colony = 22 lakh kWh; cement = 22,40,000 t.

Formula: Net generation = gross × (1 – APC); net for cement = grid + net CPP + net WHR – colony; SEEC = net kWh/t cement.

Substitution: Net CPP = $1,050 \times 0.91 = 955.5$ lakh kWh; net WHR = $380 \times 0.94 = 357.2$ lakh kWh (2 marks)

Net for cement = $400 + 955.5 + 357.2 - 22 = 1690.7$ lakh kWh = 16,90,70,000 kWh (1 mark)

SEEC = $16,90,70,000 / 22,40,000$ (1 mark)

Result: SEEC = **75.48 kWh/t cement**. Comment (1 mark): close to the typical 77 kWh/t but well above the best Indian plants (about 66 kWh/t). Raw-mill, cement-mill and kiln-fan circuits are the main targets.

c) Leaking (false) air across the preheater (3 marks)

Given: O₂ at precalciner (PC) outlet = 2.4 %; O₂ at PH outlet = 4.8 %.

Formula: % leaking air = $(O_2 \text{ at PH outlet} - O_2 \text{ at PC outlet}) / (21 - O_2 \text{ at PH outlet}) \times 100$ (Guidebook Ch. 13).

Substitution: $(4.8 - 2.4) / (21 - 4.8) \times 100 = 2.4 / 16.2 \times 100$

Result: Leaking air = **14.81 %** of the gas leaving the precalciner.

d) PH fan after sealing (6 marks)

Given: PH outlet flow = 3,45,000 Nm³/h; leaking air now 14.81 %, after sealing 5 %; fan power 2,650 kW; $P \propto Q^3$; 7,800 h; ₹ 7.20/kWh.

Formula: Gas from PC = PH flow / (1 + leak); new PH flow = gas from PC × (1 + 0.05); $O_{2, \text{new}} = (V_{\text{PC}} \times O_{2, \text{PC}} + V_{\text{leak}} \times 21) / V_{\text{new}}$; $P_2 = P_1 \times (Q_2 / Q_1)^3$.

Substitution: $V_{\text{PC}} = 3,45,000 / 1.1481 = 3,00,484$ Nm³/h (1 mark); $V_{\text{new}} = 3,00,484 \times 1.05 = 3,15,508$ Nm³/h (1 mark)

$O_{2, \text{new}} = (3,00,484 \times 2.4 + 15,024 \times 21) / 3,15,508 = 3.29 \%$ (1 mark)

$P_2 = 2,650 \times (3,15,508 / 3,45,000)^3 = 2,650 \times 0.7648 = 2,027$ kW (1 mark)

Result: Saving = 623 kW → 48,60,629 kWh/year → ₹ 3.50 crore/year (₹ 3,49,96,530) (2 marks). Less leaking air also cuts heat loss in PH exhaust, which improves STEC too.

e) Overall specific energy consumption (3 marks)

Given: Thermal = 2,20,000 t × 5,450 kcal/kg ($\approx$ 22,818 kJ/kg); electrical = 16,90,70,000 kWh at 860 kcal/kWh ($\approx$ 3,601 kJ/kWh); cement = 22,40,000 t.

Formula: Overall SEC (kcal/kg cement) = (thermal kcal + kWh × 860)/cement (kg).

Substitution: (1,19,90,00,00,000 + 16,90,70,000 × 860)/(2,24,00,00,000)

Result: Overall SEC = 600.2 kcal/kg cement (= 2513 kJ/kg).

N-4 (b) Solution — 210 MW thermal unit

a) Heat rates and cycle efficiency (6 marks)

Given: Gross generation = 205 MW = 2,05,000 kW; coal = 1,38,000 kg/h; GCV = 3,750 kcal/kg (15,701 kJ/kg); η_{boiler} = 86.8 %; APC = 8.6 %.

Formula: GHR = coal × GCV/gross kW; unit (gross) heat rate = turbine heat rate/ η_{boiler} → THR = GHR × η_{boiler} ; NHR = GHR/(1 – APC); turbine cycle efficiency = 860/THR × 100.

Substitution: GHR = 1,38,000 × 3,750/2,05,000 = 2,524.4 kcal/kWh ($\approx$ 10,569 kJ/kWh) (2 marks)

THR = 2,524.4 × 0.868 = 2,191.2 kcal/kWh ($\approx$ 9,174 kJ/kWh) (1.5 marks)

NHR = 2,524.4/(1 – 0.086) = 2,761.9 kcal/kWh ($\approx$ 11,564 kJ/kWh) (1.5 marks); η_{cycle} = 860/2,191.2 × 100 (1 mark)

Result: GHR = 2,524.4 kcal/kWh (10,569 kJ/kWh); THR = 2,191.2 kcal/kWh ($\approx$ 9,174 kJ/kWh); NHR = 2,761.9 kcal/kWh ($\approx$ 11,564 kJ/kWh); turbine cycle efficiency = 39.25 %.

b) Condenser performance (5 marks)

Given: T_{sat} = 47.4 °C (0.11 ata); CW in = 32.4 °C; CW out = 41.6 °C; CW flow = 25,200 m³/h.

Formula: TTD = T_{sat} – $T_{\text{CW,out}}$; ITD = T_{sat} – $T_{\text{CW,in}}$; LMTD = (ITD – TTD)/ln(ITD/TTD); effectiveness = CW rise/ITD; heat load = $m \times C_p \times \Delta T$.

Substitution: TTD = 47.4 – 41.6 = 5.8 °C; ITD = 47.4 – 32.4 = 15.0 °C; CW rise = 9.2 °C

LMTD = (15.0 – 5.8)/ln(15.0/5.8) = 9.2/0.9502; ϵ = 9.2/15.0 × 100

Heat load = 25,200 × 1,000 × 1 × 9.2 kcal/h ($\approx$ 0.011 kW); per kWh = 23,18,40,000/2,05,000

Result: TTD = 5.8 °C; ITD = 15.0 °C; LMTD = 9.68 °C; effectiveness = 61.3 %; heat load = 23,18,40,000 kcal/h (= 269.6 MW) = 1,131 kcal/kWh ($\approx$ 4,735 kJ/kWh).

c) CW flow implied by the turbine heat balance (4 marks)

Given: THR = 2,191.2 kcal/kWh ($\approx$ 9,174 kJ/kWh); 96 % of (THR – 860) rejected to condenser; CW rise = 9.2 °C.

Formula: Q_{cond} = 0.96 × (THR – 860) × kW; CW flow (m³/h) = $Q_{\text{cond}}/(\Delta T \times 1,000)$.

Substitution: Q_{cond} = 0.96 × (2,191.2 – 860) × 2,05,000 = 26,19,74,400 kcal/h ($\approx$ 305 MW); flow = 26,19,74,400/(9.2 × 1,000)

Result: Expected CW flow $\approx$ 28,475 m³/h; the measured 25,200 m³/h is about 11.5 % lower. Either the meter under-reads (cross-check with another method) or the CW pumps deliver less than design (impeller wear, choked tubes/screens). Low CW flow is also consistent with the high TTD and back pressure.

d) Benefit of lower back pressure (5 marks)

Given: Table 11.7 (210 MW): 2,000 → 1,975 kcal/kWh ($\approx$ 8,269 kJ/kWh) for 0.11 → 0.09 ata; THR = 2,191.2; η_{boiler} = 86.8 %; 2,05,000 kW; GCV 3,750; 7,500 h; ₹ 4,300/t.

Formula: $\text{THR}_{\text{new}} = \text{THR} \times 1,975/2,000$; $\text{GHR}_{\text{new}} = \text{THR}_{\text{new}}/\eta_{\text{boiler}}$; $\text{coal}_{\text{new}} = \text{GHR}_{\text{new}} \times \text{kW}/\text{GCV}$; $\text{saving} = (\text{coal}_{\text{old}} - \text{coal}_{\text{new}}) \times \text{hours} \times \text{cost}$.

Substitution: Relative improvement = $25/2,000 = 1.25\%$; $\text{THR}_{\text{new}} = 2,191.2 \times 0.9875 = 2,163.8 \text{ kcal/kWh}$ ($\approx 9,059 \text{ kJ/kWh}$) (1 mark)

$\text{GHR}_{\text{new}} = 2,163.8/0.868 = 2,492.8 \text{ kcal/kWh}$ ($\approx 10,437 \text{ kJ/kWh}$) (1 mark); $\text{coal}_{\text{new}} = 2,492.8 \times 2,05,000/3,750 = 136.275 \text{ t/h}$ (1 mark)

Result: Coal saving = **1.725 t/h = 12,938 t/year, worth ₹ 5.56 crore/year (₹ 5,56,31,250)** (2 marks); new NHR $\approx 2,727.4 \text{ kcal/kWh}$ ($\approx 11,419 \text{ kJ/kWh}$).

N-4 (c) Solution — Reheating furnace with recuperator

a) Direct efficiency and specific fuel consumption (4 marks)

Given: Stock = 30,000 kg/h; $C_p = 0.12 \text{ kcal/kg } ^\circ\text{C}$ ($\approx 0.502 \text{ kJ/kg}\cdot\text{K}$); $40 \rightarrow 1,180 ^\circ\text{C}$; oil = 1,350 kg/h; GCV = 10,200 kcal/kg (42,705 kJ/kg).

Formula: $\eta = \text{heat in stock}/\text{heat input} \times 100 = m \times C_p \times (t_2 - t_1)/(\text{fuel} \times \text{GCV}) \times 100$; $\text{SFC} = \text{fuel}/\text{throughput}$.

Substitution: Heat in stock = $30,000 \times 0.12 \times 1,140 = 41,04,000 \text{ kcal/h}$ ($\approx 4,772 \text{ kW}$); heat input = $1,350 \times 10,200 = 1,37,70,000 \text{ kcal/h}$ ($\approx 16,012 \text{ kW}$)

$\text{SFC} = 1,350/30$

Result: $\eta = 29.80\%$; **SFC = 45.0 kg oil/t billet** ($= 459 \text{ Mcal/t} = 1.92 \text{ GJ/t}$).

b) Excess air, actual air and flue gas (4 marks)

Given: $\text{O}_2 = 7.5\%$; theoretical air = 14.1 kg/kg oil; oil = 1,350 kg/h.

Formula: $\text{EA} = \text{O}_2/(21 - \text{O}_2) \times 100$; $\text{AAS} = (1 + \text{EA}/100) \times \text{TA}$; flue gas = $\text{AAS} + 1 \text{ kg/kg fuel}$.

Substitution: $\text{EA} = 7.5/13.5 \times 100 = 55.56\%$; $\text{AAS} = 14.1 \times 1.5556 = 21.933 \text{ kg/kg}$

Air = $21.933 \times 1,350$; flue gas = $22.933 \times 1,350$

Result: **EA = 55.6 %; air = 29,610 kg/h; flue gas = 30,960 kg/h.**

c) Recuperator duty and air preheat (4 marks)

Given: Flue gas 30,960 kg/h, $980 \rightarrow 540 ^\circ\text{C}$, $C_p 0.26$; air 29,610 kg/h at $35 ^\circ\text{C}$, $C_p 0.24$; shell loss 10 %.

Formula: $Q_{\text{gas}} = m_g \times C_{pg} \times (T_1 - T_2)$; $Q_{\text{air}} = 0.90 \times Q_{\text{gas}}$; $T_{\text{air,out}} = T_{\text{air,in}} + Q_{\text{air}}/(m_a \times C_{pa})$.

Substitution: $Q_{\text{gas}} = 30,960 \times 0.26 \times 440 = 35,41,824 \text{ kcal/h}$ ($\approx 4,118 \text{ kW}$); $Q_{\text{air}} = 0.9 \times 35,41,824 = 31,87,642 \text{ kcal/h}$ ($\approx 3,707 \text{ kW}$)

$T_{\text{air,out}} = 35 + 31,87,642/(29,610 \times 0.24)$

Result: Heat recovered = **31,87,642 kcal/h (= 3,707 kW, 23.1 % of fuel input); air preheat = 483.6 °C.**

d) LMTD, effectiveness and area (4 marks)

Given: Counter-flow; gas $980 \rightarrow 540 ^\circ\text{C}$; air $35 \rightarrow 483.6 ^\circ\text{C}$; $U = 20 \text{ kcal/h}$ ($\approx 0.023 \text{ kW}$) $\text{m}^2 ^\circ\text{C}$.

Formula: $\text{LMTD} = (\Delta T_1 - \Delta T_2)/\ln(\Delta T_1/\Delta T_2)$; $\epsilon = Q/[C_{\text{min}} \times (T_{\text{h,in}} - T_{\text{c,in}})]$; $A = Q/(U \times \text{LMTD})$.

Substitution: $\Delta T_1 = 980 - 483.6 = 496.4 ^\circ\text{C}$; $\Delta T_2 = 540 - 35 = 505 ^\circ\text{C}$; $\text{LMTD} = (496.4 - 505)/\ln(496.4/505)$

$C_{\text{gas}} = 8,049.6 \text{ kcal/h } ^\circ\text{C}$ ($\approx 9.36 \text{ kW/K}$), $C_{\text{air}} = 7,106.4 \text{ kcal/h } ^\circ\text{C}$ ($\approx 8.26 \text{ kW/K}$) $\rightarrow C_{\text{min}} = \text{air}$; $\epsilon = 31,87,642/(7,106.4 \times 945)$

$A = 31,87,642/(20 \times 500.7)$

Result: **LMTD = 500.7 °C; effectiveness = 47.5 %; area = 318 m².**

e) Stack loss and excess-air reduction (4 marks)

Given: Flue gas leaves recuperator at 540 °C; ambient 35 °C; C_p 0.26; O_2 reduced to 3 %; 6,000 h; ₹ 52,000/t oil.

Formula: Stack loss % = (flue gas kg/kg) $\times C_p \times (T_g - T_a)/GCV \times 100$; fuel_new = fuel_old $\times (1 - L_{old})/(1 - L_{new})$.

Substitution: $L_{old} = 22.933 \times 0.26 \times 505/10,200 \times 100 = 29.52 \%$

At 3 % O_2 : EA = $3/18 \times 100 = 16.67 \%$; flue gas = $14.1 \times 1.1667 + 1 = 17.450$ kg/kg; $L_{new} = 22.46 \%$

fuel_new = $1,350 \times (1 - 0.2952)/(1 - 0.2246) = 1,227.1$ kg/h

Result: Saving = 122.9 kg/h (9.1 %) = 737.4 t/year, worth ₹ 3.83 crore/year (₹ 3,83,43,020).

N-4 (d) Solution — IT office building

a) EPI and AAHEPI (4 marks)

Given: EB = 41,60,000 kWh; DG = 2,40,000 kWh; built-up area = 32,500 m²; hours = 12 $\times$ 300 = 3,600 h.

Formula: EPI = (EB + DG energy)/built-up area; AAHEPI (Wh/h/m²) = EPI/annual operating hours $\times$ 1,000 (Guidebook Ch. 14).

Substitution: EPI = 44,00,000/32,500; AAHEPI = EPI/3,600 $\times$ 1,000

Result: EPI = 135.4 kWh/m²/year (487 MJ/m²/year); AAHEPI = 37.6 Wh/h/m².

b) ILER and annual wastage (6 marks)

Given: Floor = 1,850 m²; 380 lux; 21.5 kW; target = 46 lux/W/m²; 12 floors; 3,600 h.

Formula: W/m² = W/area; lux/W/m² = lux/(W/m²); ILER = actual/target; wastage = (1 – ILER) $\times$ kW $\times$ hours.

Substitution: W/m² = 21,500/1,850 = 11.62; lux/W/m² = 380/11.62 = 32.70

ILER = 32.70/46 = 0.711 (3 marks); wastage = (1 – 0.711) $\times$ 21.5 $\times$ 3,600 $\times$ 12

Result: ILER = 0.71 $\rightarrow$ 'review suggested' (0.51–0.74) (1 mark); wastage = 2,68,591 kWh/year (2 marks).

c) LED retrofit (5 marks)

Given: New load = 12.4 kW/floor at 400 lux; ₹ 9.20/kWh.

Formula: As in (b); savings = $\Delta kW \times$ floors $\times$ hours $\times$ tariff.

Substitution: lux/W/m² = 400/(12,400/1,850) = 59.68; ILER = 59.68/46

ΔkW = (21.5 – 12.4) $\times$ 12 = 109.2 kW; energy = 109.2 $\times$ 3,600

Result: New ILER = 1.30 (satisfactory to good); savings = 3,93,120 kWh/year = ₹ 36,16,704 (₹ 36.17 lakh)/year.

d) HVAC interaction, payback and new EPI (5 marks)

Given: Lighting reduction = 109.2 kW of heat; 1 TR = 3,024 kcal/h ($\approx$ 3.52 kW); plant SEC = 1.05 kW/TR; investment = ₹ 58 lakh.

Formula: TR reduction = $\Delta kW \times 860/3,024$; HVAC kW saved = TR $\times$ kW/TR; payback = investment/annual savings; EPI_new = (energy – savings)/area.

Substitution: TR = 109.2 $\times$ 860/3,024 = 31.06 TR ($\approx$ 109 kW); HVAC saving = 31.06 $\times$ 1.05 = 32.61 kW $\times$ 3,600 h = 1,17,390 kWh (₹ 10,79,988)

Total = 36,16,704 + 10,79,988 = ₹ 46,96,692; payback = 58,00,000/46,96,692

EPI_new = (44,00,000 – 3,93,120 – 1,17,390)/32,500

Result: Cooling load cut = 31.06 TR ($\approx$ 109 kW); total savings = ₹ 46.97 lakh/year; payback = 1.23 years ($\approx$ 15 months); new EPI = 119.7 kWh/m²/year.

..... End of Section - III