

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: 2P4CEA001 – 2P4CEA016 | 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. An oil-fired package boiler generates 10,000 kg/h of steam (enthalpy 665 kcal/kg) from feed water at 85 °C while firing 720 kg/h of furnace oil of GCV 10,200 kcal/kg. Its direct-method efficiency is above 80 %.

[2P4CEA001 | Chapter 1: Boilers | Direct method efficiency from steam and fuel data]

Answer: FALSE (1 mark)

Given: $Q = 10,000 \text{ kg/h}$; $h_g = 665 \text{ kcal/kg}$ ($\approx 2,784 \text{ kJ/kg}$); $h_f = 85 \text{ kcal/kg}$ ($\approx 356 \text{ kJ/kg}$) (feed water at 85 °C); $q = 720 \text{ kg/h}$; $\text{GCV} = 10,200 \text{ kcal/kg}$ ($\approx 42,705 \text{ kJ/kg}$).

Formula: $\eta (\text{direct}) = Q \times (h_g - h_f) \times 100 / (q \times \text{GCV})$.

Substitution: $\eta = 10,000 \times (665 - 85) \times 100 / (720 \times 10,200) = 58,00,000 \times 100 / 73,44,000$.

Result: $\eta = 78.98 \%$, i.e. below 80 %, so the statement is false. The direct method gives no clue to the individual losses; the indirect method is needed to locate them.

Ref: BEE Guidebook Book 4, Ch. 1 (Boilers).

2. A commercial office building with a built-up area of 12,000 m² consumed 18,00,000 kWh from the grid and generated 60,000 kWh from its DG sets in a year. Its energy performance index (EPI) is 155 kWh/m²/year.

[2P4CEA002 | Chapter 14: Buildings and Commercial Establishments | Energy performance index (EPI) of an office building]

Answer: TRUE (1 mark)

Given: Grid = 18,00,000 kWh; DG = 60,000 kWh; built-up area = 12,000 m².

Formula: $\text{EPI (kWh/m}^2\text{/year)} = \text{total annual electricity used (grid + DG)} / \text{built-up area}$.

Substitution: $\text{EPI} = (18,00,000 + 60,000) / 12,000 = 18,60,000 / 12,000$.

Result: EPI = 155 kWh/m²/year (= 558 MJ/m²/year) – true. DG energy must be included; leaving it out would understate the EPI at 150 kWh/m²/year.

Ref: BEE Guidebook Book 4, Ch. 14 (Buildings and Commercial Establishments).

3. A gas turbine cogeneration plant has an overall plant fuel rate of 0.30 Sm³/kWh on natural gas of calorific value 9,500 kcal/Sm³. Its overall plant heat rate is 2,850 kcal/kWh.

[2P4CEA003 | Chapter 3: Cogeneration Systems | Overall plant heat rate of a gas turbine cogeneration plant]

Answer: TRUE (1 mark)

Given: Fuel rate = 0.30 Sm³/kWh; calorific value = 9,500 kcal/Sm³ (≈ 39.8 MJ/Sm³).

Formula: Overall plant heat rate (kcal/kWh) = fuel rate (Sm³/kWh) $\times$ calorific value (kcal/Sm³).

Substitution: Heat rate = $0.30 \times 9,500$.

Result: Heat rate = 2,850 kcal/kWh ($\approx 11,932$ kJ/kWh) – the statement is true. Equivalent electrical efficiency = $860/2,850 \times 100 = 30.2\%$.

Ref: BEE Guidebook Book 4, Ch. 3 (Cogeneration Systems).

4. A shell-and-tube exchanger with 50 m² heat transfer area transfers 5,04,000 kcal/h at an LMTD of 12 °C. Its overall heat transfer coefficient is about 960 kcal/m² h °C.

[2P4CEA004 | Chapter 4: Heat Exchangers | Overall heat transfer coefficient from duty and LMTD]

Answer: FALSE (1 mark)

Given: Q = 5,04,000 kcal/h (≈ 586 kW); A = 50 m²; LMTD = 12 °C.

Formula: $U = Q / (A \times \text{LMTD})$.

Substitution: $U = 5,04,000 / (50 \times 12) = 5,04,000/600$.

Result: U = 840 kcal/m² h °C (≈ 977 W/m² K) (not 960), so the statement is false. A falling U at the same flows is the usual indicator of fouling.

Ref: BEE Guidebook Book 4, Ch. 4 (Heat Exchangers).

5. A 4-pole, 50 Hz induction motor has a name-plate full-load speed of 1,470 rpm. A tachometer reading of 1,485 rpm (at rated voltage) shows that the motor is loaded to about 50 %.

[2P4CEA005 | Chapter 5: Electric Motors and Variable Speed Drives | Motor loading by the slip method]

Answer: TRUE (1 mark)

Given: $N_s = 120 \times 50/4 = 1,500$ rpm; $N_{FL} = 1,470$ rpm; $N_{measured} = 1,485$ rpm.

Formula: Load (slip method) = $(N_s - N_{measured}) / (N_s - N_{FL}) \times 100$ – valid at rated voltage.

Substitution: Load = $(1,500 - 1,485) / (1,500 - 1,470) \times 100 = 15/30 \times 100$.

Result: Load $\approx 50\%$ – true. The slip method is least reliable at low loads and when the supply voltage deviates from rated, because slip varies inversely with the square of voltage.

Ref: BEE Guidebook Book 4, Ch. 5 (Electric Motors and Variable Speed Drives).

6. For pitot-tube traverse in a rectangular fan duct of 1.4 m $\times$ 0.8 m, the equivalent diameter works out to about 1.10 m.

[2P4CEA006 | Chapter 6: Fans and Blowers | Equivalent diameter of a rectangular duct for traverse points]

Answer: FALSE (1 mark)

Given: Duct width W = 1.4 m; height H = 0.8 m.

Formula: Equivalent diameter $D_e = 2 \times W \times H / (W + H)$.

Substitution: $D_e = 2 \times 1.4 \times 0.8 / (1.4 + 0.8) = 2.24/2.2$.

Result: D_e = 1.018 m, not 1.10 m, so the statement is false. D_e is used to fix the straight-length location (and number of traverse points) for the measurement plane.

Ref: BEE Guidebook Book 4, Ch. 6 (Fans and Blowers).

7. A centrifugal pump delivers $0.05 \text{ m}^3/\text{s}$ of water against a total head of 40 m. The motor draws 30 kW at an efficiency of 90 %. The pump efficiency is about 73 %.

[2P4CEA007 | Chapter 7: Pumps | Pump efficiency from hydraulic power and motor input]

Answer: TRUE (1 mark)

Given: $Q = 0.05 \text{ m}^3/\text{s}$; $H = 40 \text{ m}$; $\rho = 1,000 \text{ kg/m}^3$; motor input = 30 kW; $\eta_{\text{motor}} = 90 \%$.

Formula: Hydraulic power P_h (kW) = $Q \times \rho \times g \times H / 1,000$; pump shaft power = input $\times \eta_{\text{motor}}$; $\eta_{\text{pump}} = P_h / \text{shaft power}$.

Substitution: $P_h = 0.05 \times 1,000 \times 9.81 \times 40 / 1,000 = 19.62 \text{ kW}$; shaft power = $30 \times 0.90 = 27 \text{ kW}$; $\eta_{\text{pump}} = 19.62/27$.

Result: $\eta_{\text{pump}} = 72.7 \%$ – the statement is true.

Ref: BEE Guidebook Book 4, Ch. 7 (Pumps).

8. A screw compressor delivers a free air delivery of $600 \text{ m}^3/\text{h}$ at $7.0 \text{ kg/cm}^2(\text{g})$, with atmospheric pressure of $1.03 \text{ kg/cm}^2(\text{a})$, and draws 55 kW of shaft power. Its isothermal efficiency is about 63 %.

[2P4CEA008 | Chapter 8: Compressors | Isothermal efficiency of an air compressor]

Answer: TRUE (1 mark)

Given: $P_1 = 1.03 \text{ kg/cm}^2(\text{a})$ ($\approx 1.01 \text{ bar(a)}$); $P_2 = 7.0 + 1.03 = 8.03 \text{ kg/cm}^2(\text{a})$ ($\approx 7.87 \text{ bar(a)}$); Q_1 (FAD) = $600 \text{ m}^3/\text{h}$; shaft power = 55 kW.

Formula: Isothermal power (kW) = $P_1 \times Q_1 \times \log_e r / 36.7$ (P_1 in kg/cm^2 absolute, Q_1 in m^3/h , $r = P_2/P_1$); isothermal efficiency = isothermal power / actual shaft power $\times 100$.

Substitution: $r = 8.03/1.03 = 7.796$; isothermal power = $1.03 \times 600 \times \ln 7.796 / 36.7 = 34.58 \text{ kW}$; $\eta_{\text{iso}} = 34.58/55 \times 100$.

Result: $\eta_{\text{iso}} = 62.9 \%$ – the statement is true.

Ref: BEE Guidebook Book 4, Ch. 8 (Compressors).

9. A water-cooled screw chiller operating at a COP of 5.0 has a specific power consumption of about 0.80 kW/TR.

[2P4CEA009 | Chapter 9: HVAC Systems | Chiller COP and specific power]

Answer: FALSE (1 mark)

Given: COP = 5.0; 1 TR = 3.516 kW of refrigeration.

Formula: kW/TR = 3.516 / COP.

Substitution: kW/TR = $3.516 / 5.0$.

Result: kW/TR = 0.703, not 0.80, so the statement is false. (0.80 kW/TR would correspond to COP = $3.516/0.80 = 4.40$.)

Ref: BEE Guidebook Book 4, Ch. 9 (HVAC Systems).

10. A power-plant surface condenser receives cooling water at 32°C and returns it at 40°C , while the saturation temperature at condenser pressure is 45°C . The condenser effectiveness is about 80 % and its terminal temperature difference (TTD) is 5°C .

[2P4CEA010 | Chapter 11: Thermal Power Station | Condenser effectiveness and TTD]

Answer: FALSE (1 mark)

Given: CW inlet = 32°C ; CW outlet = 40°C ; $T_{\text{sat}} = 45^\circ\text{C}$.

Formula: Condenser effectiveness = $(\text{CW outlet} - \text{CW inlet}) / (T_{\text{sat}} - \text{CW inlet}) \times 100$; TTD = $T_{\text{sat}} - \text{CW outlet}$.

Substitution: Effectiveness = $(40 - 32) / (45 - 32) \times 100 = 8/13 \times 100$; TTD = $45 - 40$.

Result: Effectiveness = 61.5 % (not 80 %); TTD = 5 °C. Since one part of the statement is wrong, the statement is false.

Ref: BEE Guidebook Book 4, Ch. 11 (Thermal Power Station).

..... 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

[2P4CEA011] (5 marks)

Chapter 4: Heat Exchangers | Plate heat exchanger in a dairy – LMTD, U-value, fouling and effectiveness

L-1 A milk dairy in Anand cools raw milk in a counter-flow plate heat exchanger using chilled water. Measured data during an energy audit:

Parameter	Value
Milk flow / specific heat	12,000 kg/h / 0.93 kcal/kg °C
Milk inlet / outlet temperature	36 °C / 6 °C
Chilled water flow / inlet temperature	18,000 kg/h / 1.5 °C (specific heat 1.0 kcal/kg °C)
Heat transfer area	22 m ²
Design (clean) overall heat transfer coefficient	2,400 kcal/m ² h °C

Calculate (a) the heat duty and chilled water outlet temperature, (b) the LMTD, (c) the actual overall heat transfer coefficient and its percentage of design value, with the fouling resistance, and (d) the effectiveness of the exchanger.

MODEL ANSWER

a) Heat duty and chilled water outlet (1 mark)

Given: $m_{\text{milk}} = 12,000 \text{ kg/h}$; $C_p = 0.93 \text{ kcal/kg } ^\circ\text{C}$ ($\approx 3.89 \text{ kJ/kg}\cdot\text{K}$); $\Delta T_{\text{milk}} = 36 - 6 = 30 \text{ } ^\circ\text{C}$; $m_{\text{cw}} = 18,000 \text{ kg/h}$.

Formula: $Q = m \times C_p \times \Delta T$; $T_{\text{cw,out}} = T_{\text{cw,in}} + Q/(m_{\text{cw}} \times C_{p_w})$.

Substitution: $Q = 12,000 \times 0.93 \times 30$; $T_{\text{cw,out}} = 1.5 + 3,34,800/18,000$

Result: $Q = 3,34,800 \text{ kcal/h}$ ($\approx 389 \text{ kW}$); $T_{\text{cw,out}} = 20.1 \text{ } ^\circ\text{C}$.

b) LMTD (counter-flow) (1 mark)

Given: $\Delta T_1 = 36 - 20.1 = 15.9 \text{ } ^\circ\text{C}$ (hot inlet – cold outlet); $\Delta T_2 = 6 - 1.5 = 4.5 \text{ } ^\circ\text{C}$.

Formula: $\text{LMTD} = (\Delta T_1 - \Delta T_2) / \ln(\Delta T_1 / \Delta T_2)$.

Substitution: $\text{LMTD} = (15.9 - 4.5) / \ln(15.9/4.5)$

Result: $\text{LMTD} = 9.03 \text{ } ^\circ\text{C}$.

c) Overall heat transfer coefficient and fouling (2 marks)

Given: $Q = 3,34,800 \text{ kcal/h}$ ($\approx 389 \text{ kW}$); $A = 22 \text{ m}^2$; $\text{LMTD} = 9.03 \text{ } ^\circ\text{C}$; $U_{\text{design}} = 2,400 \text{ kcal/m}^2 \text{ h } ^\circ\text{C}$.

Formula: $U = Q/(A \times \text{LMTD})$; fouling resistance $R_f = 1/U_{\text{actual}} - 1/U_{\text{design}}$.

Substitution: $U = 3,34,800/(22 \times 9.03)$

$R_f = 1/1,685 - 1/2,400$

Result: $U = 1,685 \text{ kcal/m}^2 \text{ h } ^\circ\text{C}$ ($\approx 1,960 \text{ W/m}^2 \text{ K}$) = **70.2 % of design**; $R_f = 0.000177 \text{ m}^2 \text{ h } ^\circ\text{C/kcal}$. About 30 % loss of U points to milk-stone / protein fouling – review CIP frequency.

d) Effectiveness (1 mark)

Given: $C_{\text{milk}} = 12,000 \times 0.93 = 11,160 \text{ kcal/h } ^\circ\text{C}$ ($\approx 13.0 \text{ kW/K}$) (C_{min}); $C_{\text{cw}} = 18,000 \text{ kcal/h } ^\circ\text{C}$ ($\approx 20.9 \text{ kW/K}$).

Formula: $\epsilon = \text{actual heat transfer} / \text{maximum possible} = C_{\min} \times (T_{h,\text{in}} - T_{h,\text{out}}) / [C_{\min} \times (T_{h,\text{in}} - T_{c,\text{in}})]$.

Substitution: $\epsilon = (36 - 6) / (36 - 1.5) \times 100$

Result: $\epsilon = 87.0 \%$.

[2P4CEA012] (5 marks)

Chapter 10: Financial Analysis | Waste heat recovery project – payback, NPV, PI, discounted payback and IRR

L-2 A ceramic tile plant in Morbi plans a kiln waste-heat recovery project costing ₹ 1.20 crore (all at the start of year 1). Net annual savings expected are ₹ 32 lakh, ₹ 34 lakh, ₹ 36 lakh, ₹ 36 lakh and ₹ 36 lakh in years 1 to 5, with a salvage value of ₹ 10 lakh at the end of year 5. The company's discount rate is 12 %.

Calculate (a) the simple payback period, (b) the net present value and profitability index, (c) the discounted payback period, and (d) state whether the IRR is above or below 15 % with a supporting calculation.

MODEL ANSWER

a) Simple payback (1 mark)

Given: Investment = ₹ 1,20,00,000; savings yr 1–3 cumulative = ₹ 1,02,00,000; yr 4 = ₹ 36,00,000.

Formula: Simple payback = year before full recovery + unrecovered amount / cash flow of the recovery year.

Substitution: Payback = $3 + (1,20,00,000 - 1,02,00,000) / 36,00,000$

Result: Simple payback = 3.50 years.

b) NPV and profitability index (2 marks)

Given: Cash flows (₹ lakh): 32, 34, 36, 36, 36 + 10 salvage = 46; $i = 12 \%$.

Formula: $PV = CF_t / (1 + i)^t$; $NPV = \sum PV - \text{investment}$; $PI = \sum PV / \text{investment}$.

Substitution: $PV_1 = 32,00,000 / 1.12 = 28,57,143$; $PV_2 = 34,00,000 / 1.12^2 = 27,10,459$; $PV_3 = 36,00,000 / 1.12^3 = 25,62,409$

$PV_4 = 36,00,000 / 1.12^4 = 22,87,865$; $PV_5 = 46,00,000 / 1.12^5 = 26,10,164$; $\sum PV = ₹ 1,30,28,040$

Result: NPV = ₹ 10,28,040 (₹ 10.28 lakh); PI = 1.086 → positive NPV, project acceptable.

c) Discounted payback (1 mark)

Given: Cumulative PV: yr 3 = ₹ 81,30,011; yr 4 = ₹ 1,04,17,876; yr 5 = ₹ 1,30,28,040.

Formula: Discounted payback = last year with negative cumulative PV + shortfall / PV of next year.

Substitution: DPB = $4 + (1,20,00,000 - 1,04,17,876) / 26,10,164$

Result: Discounted payback = 4.61 years (recovery depends on the salvage value).

d) IRR versus 15 % (1 mark)

Given: Same cash flows; trial rate 15 %.

Formula: IRR is the rate at which NPV = 0; if NPV at 15 % > 0, IRR > 15 %.

Substitution: NPV at 15 % = $32/1.15 + 34/1.15^2 + 36/1.15^3 + 36/1.15^4 + 46/1.15^5 - 120$ (₹ lakh) = ₹ 0.66 lakh

Result: NPV(15 %) is positive, so IRR > 15 %; by interpolation IRR ≈ 15.2 %.

..... 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)

[2P4CEA013] (20 marks)

Chapter 2: Furnaces | Forging furnace heat balance and combustion tuning (LDO-fired)

N-1 An oil-fired batch-cum-continuous forging furnace in a crankshaft forging unit at Pune heats steel billets from 35 °C to 1,200 °C. There is no recuperator. The energy auditor collected the following data during a steady 6-hour trial:

Parameter	Value
Fuel: light diesel oil (LDO), ultimate analysis % by mass	C 86.5, H 13.0, S 0.3, O 0.2
GCV of LDO / fuel cost	10,700 kcal/kg / ₹ 78 per kg
Fuel consumption / billet throughput	118 kg/h / 2.4 t/h
Mean specific heat of steel (35–1,200 °C)	0.13 kcal/kg °C
O ₂ in flue gas (dry) / flue gas temperature at furnace exit	8.0 % / 820 °C
Ambient temperature / humidity	35 °C / 0.025 kg moisture per kg dry air
Specific heat of dry flue gas / superheated vapour	0.26 / 0.45 kcal/kg °C
Roof: area / average outer surface temperature	12 m ² / 110 °C
Side walls: area / average outer surface temperature	28 m ² / 95 °C
Surface emissivity; natural-convection factor a (roof / side)	0.75; 2.8 / 2.2
Charging door opening / fraction of time open	0.6 m × 0.3 m / 20 %
Furnace inside temperature / radiation factor ϕ for the opening	1,250 °C / 0.55
Operating hours	5,000 h/year

- Find the theoretical air, percentage excess air and actual air supplied per kg of fuel. **[3 Marks]**
- Determine the heat input and the useful heat per tonne of billets and the direct (thermal) efficiency of the furnace. **[3 Marks]**
- Calculate per tonne of billets: (i) dry flue gas loss, (ii) loss due to H₂ in fuel, (iii) loss due to moisture in air, (iv) wall (roof + side) loss and (v) radiation loss through the door opening. **[8 Marks]**
- Draw up the heat balance (kcal/t and %) and determine the unaccounted loss. **[2 Marks]**
- If combustion tuning with a new burner reduces flue gas O₂ to 3 % (flue gas temperature unchanged), estimate the new fuel consumption and the annual monetary savings. Assume stock heat, wall, opening and unaccounted losses per tonne remain unchanged. **[4 Marks]**

MODEL ANSWER**a) Combustion air (3 marks)**

Given: C = 86.5 %, H = 13.0 %, O = 0.2 %, S = 0.3 %; O₂ in flue gas = 8.0 %.

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

Substitution: Theoretical air = $[11.6 \times 86.5 + 34.8 \times (13.0 - 0.2/8) + 4.35 \times 0.3]/100 = 14.56 \text{ kg/kg}$

EA = $8/(21 - 8) \times 100 = 61.5 \%$; AAS = 1.6154×14.56

Result: Theoretical air = 14.56 kg/kg; EA = 61.5 %; actual air = 23.52 kg/kg fuel.

b) Heat input, useful heat and efficiency (3 marks)

Given: Fuel = 118 kg/h; production = 2.4 t/h; GCV = 10,700 kcal/kg ($\approx 44,799 \text{ kJ/kg}$); steel heated 35 → 1,200 °C, Cp = 0.13.

Formula: Specific fuel (kg/t) = fuel/production; heat input = SFC × GCV; useful heat = $1,000 \times C_p \times (t_2 - t_1)$; $\eta = \text{useful}/\text{input} \times 100$.

Substitution: SFC = $118/2.4 = 49.17 \text{ kg/t}$; heat input = $49.17 \times 10,700 = 5,26,083 \text{ kcal/t}$

Useful heat = $1,000 \times 0.13 \times (1,200 - 35) = 1,51,450 \text{ kcal/t}$

Result: Heat input = 5,26,083 kcal/t (2,203 MJ/t); useful heat = 1,51,450 kcal/t; furnace efficiency = 28.8 %.

c)(i) Dry flue gas loss (2 marks)

Given: AAS = 23.524 kg/kg; theoretical air = 14.562; $T_f - T_a = 820 - 35 = 785$ °C; $C_p = 0.26$.

Formula: Dry flue gas $m = C \times 44/12 + S \times 64/32 + AAS \times 0.77 + (AAS - \text{theoretical air}) \times 0.23$ (kg/kg fuel); $\text{loss} = SFC \times m \times C_p \times (T_f - T_a)$.

Substitution: $m = 0.865 \times 44/12 + 0.003 \times 2 + 23.524 \times 0.77 + (23.524 - 14.562) \times 0.23 = 23.352$ kg/kg

$\text{Loss} = 49.17 \times 23.352 \times 0.26 \times 785$

Result: Dry flue gas loss = 2,34,337 kcal/t (44.5 % of input).

c)(ii) Loss due to hydrogen in fuel (2 marks)

Given: H = 13.0 %; latent heat 584 kcal/kg ($\approx 2,445$ kJ/kg); C_p vapour 0.45.

Formula: $\text{Loss} = SFC \times 9H \times [584 + C_p \times (T_f - T_a)]$.

Substitution: $\text{Loss} = 49.17 \times 9 \times 0.13 \times (584 + 0.45 \times 785) = 49.17 \times 1.17 \times 937.25$

Result: H₂ loss = 53,915 kcal/t (10.2 %).

c)(iii) Loss due to moisture in air (1 mark)

Given: AAS = 23.524 kg/kg; humidity = 0.025 kg/kg dry air.

Formula: $\text{Loss} = SFC \times AAS \times \text{humidity} \times C_p \times (T_f - T_a)$.

Substitution: $\text{Loss} = 49.17 \times 23.524 \times 0.025 \times 0.45 \times 785$

Result: Humidity loss = 10,214 kcal/t (1.94 %).

c)(iv) Wall losses (2 marks)

Given: Ambient $T_a = 308$ K; roof 383 K, $a = 2.8$, 12 m²; side 368 K, $a = 2.2$, 28 m²; $\epsilon = 0.75$.

Formula: Heat loss (kcal/m² h) = $a \times (T_s - T_a)^{1.25} + 4.88 \epsilon [(T_s/100)^4 - (T_a/100)^4]$ (Guidebook Ch. 2); per tonne = $\Sigma (q \times A)/\text{production}$.

Substitution: Roof: $2.8 \times 75^{1.25} + 4.88 \times 0.75 \times (3.83^4 - 3.08^4) = 1,076.2$ kcal/m² h $\times 12 = 12,914$ kcal/h (≈ 15.0 kW)

Side: $2.2 \times 60^{1.25} + 4.88 \times 0.75 \times (3.68^4 - 3.08^4) = 709.2$ kcal/m² h $\times 28 = 19,859$ kcal/h (≈ 23.1 kW)

Per tonne = $(12,914 + 19,859)/2.4$

Result: Wall loss = 13,655 kcal/t (2.60 %).

c)(v) Radiation through door opening (1 mark)

Given: Opening = $0.6 \times 0.3 = 0.18$ m²; furnace 1,523 K; ambient 308 K; $\phi = 0.55$; open 20 % of time.

Formula: Loss (kcal/h) = $4.88 \times [(T_f/100)^4 - (T_a/100)^4] \times A \times \phi \times \text{fraction open}$.

Substitution: Loss = $4.88 \times (15.23^4 - 3.08^4) \times 0.18 \times 0.55 \times 0.20 = 5,190$ kcal/h (≈ 6.03 kW) $\div 2.4$ t/h

Result: Opening loss = 2,162 kcal/t (0.41 %).

d) Heat balance (2 marks)

Item	kcal/t billet	%
Heat input (fuel)	5,26,083	100.00
Heat in stock (useful)	1,51,450	28.79
Dry flue gas loss	2,34,337	44.54
Loss due to H ₂ in fuel	53,915	10.25
Loss due to moisture in air	10,214	1.94
Wall loss	13,655	2.60
Opening loss	2,162	0.41
Unaccounted (scale, cooling, CO, by difference)	60,350	11.47

Result: Unaccounted loss = 60,350 kcal/t (11.5 %). The dry flue gas loss ($\approx 45\%$) dominates – high excess air and no recuperator.

e) Combustion tuning to 3 % O₂ (4 marks)

Given: O₂ = 3 %; T_f = 820 °C; fixed per-tonne heat (stock + wall + opening + unaccounted) unchanged; 5,000 h/year; ₹ 78/kg.

Formula: EA = $3/18 \times 100$; net useful heat per kg fuel = GCV – (dry flue gas + H₂ + humidity losses per kg); SFC_{new} = fixed heat per tonne / net heat per kg.

Substitution: EA = 16.7 %; AAS = $1.1667 \times 14.562 = 16.989$ kg/kg; m = 16.818 kg/kg

Net heat per kg: before = 4,629.5 kcal/kg ($\approx 19,383$ kJ/kg); after = $10,700 - (3,432.5 + 1,096.6 + 150.0) = 6,020.9$ kcal/kg ($\approx 25,208$ kJ/kg)

Fixed heat = 2,27,617 kcal/t $\rightarrow$ SFC_{new} = 2,27,617/6,020.9

Saving = $(49.17 - 37.80) \times 2.4 = 27.27$ kg/h $\times 5,000$ h $\times$ ₹ 78

Result: New SFC = 37.80 kg/t (23.1 % less); fuel saving = 27.3 kg/h; annual savings = ₹ 1,06,34,840 (₹ 106.35 lakh).

Comment: adding a recuperator to preheat combustion air would be the next step; the guidebook rule of thumb is about 1 % fuel saving for every 20 °C rise in combustion air temperature.

[2P4CEA014] (20 marks)

Chapter 7: Pumps | Parallel pumping in a water works – system curve, SEC and VFD versus throttling

N-2 A municipal water works in Nagpur pumps treated water from a clear-water sump to an elevated service reservoir with two identical centrifugal pumps running in parallel into a common header. Both motors are fixed-speed. Field measurements:

Parameter	Value
Combined flow (electromagnetic flowmeter)	2,700 m ³ /h
Suction / discharge pressure (each pump)	0.2 kg/cm ² (g) / 4.6 kg/cm ² (g)
Discharge gauge above suction gauge	0.5 m
Static head (sump level to reservoir inlet)	32 m
Input power per motor / motor efficiency	238 kW / 94.5 %
Pump shut-off head (from manufacturer's curve)	56 m
Tariff	₹ 7.50 per kWh

Assume the pump curve $H = H_0 - b q^2$ (q = flow per pump) and the system curve $H = H_s + k Q^2$ (Q = total flow).

- Determine the total head, hydraulic power per pump and the pump efficiency. **[5 Marks]**
- Find the constants b and k . Estimate the flow and head if only one pump runs. **[5 Marks]**
- Calculate the specific energy consumption (kWh per million litres, ML) and compare it with the theoretical minimum for lifting water through the static head. **[3 Marks]**
- At night the demand is 2,000 m³/h for 10 h/day, which is presently met by throttling the common discharge valve (both pumps running, pump efficiency drops to 70 %). Instead, VFDs on both pumps would match the system curve (pump efficiency 78 %, VFD efficiency 97 %). Find the required speed, power with each option and the annual savings. **[7 Marks]**

MODEL ANSWER

a) Head, hydraulic power and efficiency (5 marks)

Given: P_d = 4.6 kg/cm²(g) (≈ 4.51 bar(g)); P_s = 0.2 kg/cm²(g) (≈ 0.196 bar(g) = 19.6 kPa(g)); Δz = 0.5 m; q = 1,350 m³/h per pump; input = 238 kW; η_m = 94.5 %.

Formula: $H = (P_d - P_s) \times 10 + \Delta z$ ($1 \text{ kg/cm}^2 \approx 10 \text{ m water}$); $P_h \text{ (kW)} = q \text{ (m}^3/\text{s)} \times \rho \times g \times H / 1,000$; $\eta_{\text{pump}} = P_h / (\text{input} \times \eta_m)$.

Substitution: $H = (4.6 - 0.2) \times 10 + 0.5 = 44.5 \text{ m}$

$P_h = (1,350/3,600) \times 1,000 \times 9.81 \times 44.5/1,000 = 163.70 \text{ kW}$; shaft = $238 \times 0.945 = 224.91 \text{ kW}$

Result: $H = 44.5 \text{ m}$; $P_h = 163.7 \text{ kW}$ per pump; $\eta_{\text{pump}} = 72.8 \%$.

b) Curve constants and single-pump operation (5 marks)

Given: $H_o = 56 \text{ m}$; at $q = 1,350 \text{ m}^3/\text{h}$, $H = 44.5 \text{ m}$; $H_s = 32 \text{ m}$ at $Q = 2,700 \text{ m}^3/\text{h}$.

Formula: $b = (H_o - H)/q^2$; $k = (H - H_s)/Q^2$; one pump: $H_o - b Q^2 = H_s + k Q^2 \rightarrow Q = \sqrt{[(H_o - H_s)/(b + k)]}$.

Substitution: $b = (56 - 44.5)/1,350^2 = 6.3100 \times 10^{-6} \text{ m}/(\text{m}^3/\text{h})^2$; $k = (44.5 - 32)/2,700^2 = 1.7147 \times 10^{-6} \text{ m}/(\text{m}^3/\text{h})^2$

$Q = \sqrt{[(56 - 32)/(6.3100 \times 10^{-6} + 1.7147 \times 10^{-6})]}$

Result: One pump alone: $Q = 1,729 \text{ m}^3/\text{h}$ (64 % of two-pump flow) at $H = 37.1 \text{ m}$. Parallel operation adds head (friction) as well as flow, so the second pump adds only about 36 % of the flow.

c) Specific energy consumption (3 marks)

Given: Total input = $2 \times 238 = 476 \text{ kW}$; flow = $2,700 \text{ m}^3/\text{h} = 2.7 \text{ ML/h}$; static head = 32 m .

Formula: $\text{SEC} = \text{kW} / (\text{ML/h})$; theoretical minimum (kWh/ML) = $\rho \times g \times H_s \times 1,000 \text{ m}^3 / 3.6 \times 10^6 \text{ J per kWh} = 9.81 \times H_s/3.6 (= 2.725 \times H_s)$.

Substitution: $\text{SEC} = 476/2.7 = 176.3 \text{ kWh/ML}$

Minimum = $9.81 \times 32/3.6 = 87.2 \text{ kWh/ML}$

Result: $\text{SEC} = 176.3 \text{ kWh/ML}$ vs theoretical $87.2 \text{ kWh/ML} \rightarrow$ overall wire-to-water efficiency at static head = **49.5 %.**

d) Throttling versus VFD at $2,000 \text{ m}^3/\text{h}$ (7 marks)

Given: $Q = 2,000 \text{ m}^3/\text{h}$ ($1,000 \text{ m}^3/\text{h}$ per pump); $10 \text{ h/day} \times 365 \text{ days} = 3,650 \text{ h}$; η_{pump} : 70 % (throttled), 78 % (VFD); $\eta_m = 94.5 \%$; $\eta_{\text{VFD}} = 97 \%$; ₹ 7.50/kWh.

Formula: Throttled head = $H_o - b q^2$; system head = $H_s + k Q^2$; affinity: $H_o n^2 - b q^2 = \text{system head} \rightarrow n = \sqrt{[(H_{\text{sys}} + b q^2)/H_o]}$; power = $Q \times 9.81 \times H/(3,600 \times \eta_{\text{pump}})$.

Substitution: Throttled: $H = 56 - 6.3100 \times 10^{-6} \times 1,000^2 = 49.69 \text{ m} \rightarrow P = 2,000/3,600 \times 9.81 \times 49.69/(0.70 \times 0.945) = 409.4 \text{ kW}$

System head at $2,000 \text{ m}^3/\text{h} = 32 + 1.7147 \times 10^{-6} \times 2,000^2 = 38.86 \text{ m} \rightarrow n = \sqrt{[(38.86 + 6.310)/56]} = 89.8 \%$ speed

VFD: $P = 2,000/3,600 \times 9.81 \times 38.86/(0.78 \times 0.945 \times 0.97) = 296.2 \text{ kW}$

Saving = $409.4 - 296.2 = 113.2 \text{ kW} \times 3,650 \text{ h}$

Result: Speed $\approx 89.8 \%$ (44.9 Hz); throttled **409.4 kW vs VFD **296.2 kW**; saving **4,13,133 kWh/year** = ₹ **30,98,501** (₹ 30.99 lakh) per year. Night SEC falls from 204.7 to 148.1 kWh/ML.**

Note: because the static head is about 70 % of the total head, speed can be reduced only to about 90 %; check that the pumps stay above minimum stable flow and that the reduced-speed head still exceeds the static head at all reservoir levels.

[2P4CEA015] (20 marks)

Chapter 9: HVAC Systems | Direct-fired vapour absorption chiller – COP, combustion efficiency, heat balance and economics

N-3 A shopping mall in Gurugram uses a direct-fired double-effect vapour absorption (lithium bromide–water) chiller, fired with high-speed diesel (HSD) because piped gas was not available when it was installed. The auditor measured the following at steady load:

Parameter	Value
Chilled water flow / inlet / outlet temperature	240 m ³ /h / 13.0 °C / 7.5 °C
Cooling water flow (absorber + condenser in series) / inlet / outlet	520 m ³ /h / 30.0 °C / 34.2 °C
Specific heat of water / density	4.18 kJ/kg K / 1,000 kg/m ³
HSD firing rate / GCV / density	92 kg/h / 10,800 kcal/kg / 0.83 kg/litre
HSD ultimate analysis, % by mass	C 86.0, H 13.4, S 0.3, O 0.2, moisture 0.1
Flue gas O ₂ (dry) / exit temperature / ambient temperature	5.0 % / 195 °C / 34 °C
Specific heat of dry flue gas / superheated vapour	0.24 / 0.45 kcal/kg °C
Auxiliaries: CHW pump / CW pump / cooling tower fans / VAM solution pumps	22 / 41 / 18.5 / 6 kW
HSD price / electricity tariff / operating hours	₹ 90 per litre / ₹ 9.00 per kWh / 3,600 h/year

- a) Calculate the refrigeration effect (kW and TR), the fuel heat input (kW) and the evaporator-side COP. [5 Marks]
- b) Estimate the combustion efficiency of the direct-fired generator by the indirect method (dry flue gas and H₂/moisture losses only). [6 Marks]
- c) Using the cooling-water heat rejection, cross-check the refrigeration effect by heat balance and comment on the measurement consistency. [4 Marks]
- d) Calculate the fuel consumption per TR-h and the operating cost per TR-h (fuel + auxiliaries). Compare with a water-cooled centrifugal chiller of 0.65 kW/TR (same auxiliaries) and find the annual saving on replacement. [5 Marks]

MODEL ANSWER

a) Refrigeration effect, heat input and COP (5 marks)

Given: $m_{chw} = 2,40,000 \text{ kg/h}$; $\Delta T = 13.0 - 7.5 = 5.5 \text{ K}$; $C_p = 4.18 \text{ kJ/kg K}$; fuel = 92 kg/h $\times$ 10,800 kcal/kg ($\approx 45,217 \text{ kJ/kg}$).

Formula: $Q_e \text{ (kW)} = m \times C_p \times \Delta T / 3,600$; $TR = Q_e / 3.516$; $Q_{in} \text{ (kW)} = \text{fuel} \times \text{GCV} \times 4.1868 / 3,600$; $COP = Q_e / Q_{in}$.

Substitution: $Q_e = 2,40,000 \times 4.18 \times 5.5 / 3,600 = 1,532.7 \text{ kW} \rightarrow 1,532.7 / 3.516 = 435.9 \text{ TR} (\approx 1,533 \text{ kW})$

$Q_{in} = 92 \times 10,800 \times 4.1868 / 3,600 = 1,155.6 \text{ kW} (= 9,93,600 \text{ kcal/h})$

Result: Refrigeration = 435.9 TR ($\approx 1,533 \text{ kW}$); heat input = 1,155.6 kW; COP = 1.326 – typical of a direct-fired double-effect machine (1.2–1.4 on GCV).

b) Combustion efficiency (indirect) (6 marks)

Given: C 86.0, H 13.4, S 0.3, O 0.2, M 0.1 %; O₂ = 5 %; $T_f - T_a = 195 - 34 = 161 \text{ °C}$; GCV = 10,800 kcal/kg ($\approx 45,217 \text{ kJ/kg}$).

Formula: Theoretical air = $[11.6 C + 34.8 (H - O/8) + 4.35 S] / 100$; $EA = O_2 / (21 - O_2) \times 100$; dry flue gas $m \approx AAS + 1 - (9H + M)$; $L_1 = m \times C_p \times \Delta T / \text{GCV} \times 100$; $L_2 = (9H + M) \times [584 + C_{p_v} \times \Delta T] / \text{GCV} \times 100$; $\eta = 100 - L_1 - L_2$.

Substitution: Theoretical air = $[11.6 \times 86 + 34.8 \times (13.4 - 0.025) + 4.35 \times 0.3] / 100 = 14.64 \text{ kg/kg}$; $EA = 5 / 16 \times 100 = 31.25 \%$

$AAS = 1.3125 \times 14.64 = 19.22 \text{ kg/kg}$; $m = 19.22 + 1 - (1.206 + 0.001) = 19.01 \text{ kg/kg}$

$L_1 = 19.01 \times 0.24 \times 161 / 10,800 \times 100 = 6.80 \%$; $L_2 = 1.207 \times (584 + 0.45 \times 161) / 10,800 \times 100 = 7.34 \%$

Result: Combustion efficiency $\approx 85.9 \%$ (radiation and other minor losses neglected). Dropping exit gas O₂ to about 3 % and cleaning the generator fire-tubes would raise it.

c) Heat balance cross-check (4 marks)

Given: $m_{cw} = 5,20,000 \text{ kg/h}$; $\Delta T = 4.2 \text{ K}$; heat absorbed from fuel = $1,155.6 \times 0.8586 \text{ kW}$.

Formula: Heat rejected (absorber + condenser) = refrigeration effect + heat from fuel (pump work neglected) →

$$Q_e(\text{balance}) = Q_{cw} - Q_{in} \times \eta_{\text{comb.}}$$

Substitution: $Q_{cw} = 5,20,000 \times 4.18 \times 4.2/3,600 = 2,535.9 \text{ kW}$

$$Q_e(\text{balance}) = 2,535.9 - 992.2 = 1,543.7 \text{ kW}$$

Result: Q_e by heat balance = 1,543.7 kW vs 1,532.7 kW measured → deviation 0.72 % (within ±5 %), so the CHW-side readings are acceptable; COP on balance = 1.336.

d) Operating cost and comparison with an electric chiller (5 marks)

Given: Load = 435.9 TR ($\approx 1,533 \text{ kW}$); HSD 92 kg/h ÷ 0.83 = 110.8 litre/h × ₹ 90; auxiliaries = 87.5 kW; electric option 0.65 kW/TR + same auxiliaries; ₹ 9.00/kWh; 3,600 h.

Formula: Cost per TR-h = hourly cost/TR; annual saving = (VAM cost – electric cost) per hour × hours.

Substitution: Fuel = 92/435.9 = 0.211 kg/TR-h; VAM cost = 110.84 × 90 + 87.5 × 9 = ₹ 9,976 + ₹ 788 = ₹ 10,763/h

Electric: 0.65 × 435.9 + 87.5 = 370.8 kW × ₹ 9 = ₹ 3,338/h

$$\text{Saving} = (10,763 - 3,338) \times 3,600$$

Result: VAM: 0.211 kg HSD/TR-h, ₹ 24.69/TR-h; electric: ₹ 7.66/TR-h. Annual saving on replacement $\approx$ ₹ 2,67,32,946 (₹ 2.67 crore).

Comment: direct-fired absorption chillers are economical only with cheap gas or where waste heat / exhaust (e.g. DG or gas-engine jacket and exhaust heat) is available; on HSD the auditor should recommend replacement or conversion to PNG/waste-heat firing.

[2P4CEA016] (20 marks)

Chapter 11: Thermal Power Station | Answer any ONE: (a) thermal power economiser/APH, (b) integrated steel SEC, (c) GT cogeneration in textile processing, (d) cement raw mill heat balance

N-4 Answer any ONE of the following among the four questions given below (20 marks).

N-4 (a) Thermal power station – economiser, air preheater and air in-leakage

In a 250 MW coal-fired unit in Chhattisgarh, the following data were logged at full load:

Parameter	Value
Feed water flow / economiser water inlet / outlet	760 t/h / 252 °C / 286 °C
Flue gas temperature at economiser inlet / outlet (= APH gas inlet)	465 °C / 372 °C
Secondary air flow through APH / air inlet / air outlet	680 t/h / 36 °C / 318 °C
Flue gas temperature at APH outlet: measured / design	152 °C / 135 °C
O ₂ (dry) at APH gas inlet / outlet	3.6 % / 6.2 %
Flue gas flow at APH outlet (handled by ID fans)	1,250 t/h
ID fan input power (2 fans)	2 × 1,600 kW
Dry flue gas per kg coal / specific heat of flue gas	6.9 kg/kg / 0.24 kcal/kg °C
Coal: GCV / firing rate / price	3,600 kcal/kg / 172 t/h / ₹ 3,900 per t
Present boiler efficiency / operating hours / cost of generation	86.2 % / 7,800 h / ₹ 3.20 per kWh

a) Find the economiser effectiveness and heat pick-up. [4 Marks]

b) Find the APH (air-side) effectiveness and heat pick-up. [4 Marks]

c) Estimate the APH air leakage. [3 Marks]

d) After seal replacement the leakage drops to 7 %. Taking ID fan power proportional to gas flow handled (same pressure rise and efficiency), estimate the annual saving in ID fan energy and cost. [5 Marks]

e) If the APH outlet gas temperature is brought back to design, estimate the change in dry flue gas loss and the annual coal saving (₹). [4 Marks]

OR

N-4 (b) Integrated steel plant – specific energy consumption and improvement options

An integrated steel plant in Odisha (BF–BOF route) produced 3.0 million tonnes of crude steel (tcs) and 3.15 million tonnes of hot metal (thm) in a year. Annual energy data:

Energy stream	Quantity	Calorific value
Coking coal (to coke ovens)	22.5 lakh t	7,000 kcal/kg
Pulverised coal injection (PCI) coal	4.8 lakh t	7,200 kcal/kg
Natural gas	1,200 lakh Sm ³	8,900 kcal/Sm ³
Purchased electricity	16,500 lakh kWh	860 kcal/kWh
Surplus BF gas exported to a neighbouring unit	19,000 lakh Nm ³	750 kcal/Nm ³
Electricity exported to grid	1,800 lakh kWh	860 kcal/kWh
Coke charged to blast furnaces	15.3 lakh t	–

- Calculate the gross energy input and the net specific energy consumption (Gcal/tcs). Compare with the Indian range and the world benchmark given in the guidebook. **[7 Marks]**
- A top-pressure recovery turbine (TRT) on the blast furnaces would generate 1,100 lakh kWh/year, reducing purchased power. Find the new SEC and the annual saving at ₹ 7.10/kWh. **[5 Marks]**
- Raising PCI by 40 kg/thm replaces coke at 0.85 kg coke per kg coal (coke yield from coking coal 0.68). Find the new coke rate, the net change in SEC, and the net annual benefit with coke at ₹ 32,000/t and PCI coal at ₹ 14,500/t. **[8 Marks]**

OR

N-4 (c) Gas turbine cogeneration in a textile processing unit

A textile dyeing and finishing unit in Surat processes 5,200 kg/h of fabric. A gas turbine with a heat recovery steam generator (HRSG, with supplementary firing) meets its power and steam needs. Test data (4-hour average):

Parameter	Value
NG to gas turbine / to HRSG supplementary burner	1,560 Sm ³ /h / 120 Sm ³ /h
Net calorific value of NG	8,900 kcal/Sm ³
GT power output	4,850 kW
Exhaust gas flow / temperature at HRSG inlet / specific heat	16.8 kg/s / 530 °C / 0.26 kcal/kg °C
Steam generated / enthalpy	12,600 kg/h / 665 kcal/kg
Feed water enthalpy	105 kcal/kg
Alternative: grid power / NG-fired boiler efficiency (NCV)	₹ 8.60 per kWh / 88 %
NG price / operating hours	₹ 48 per Sm ³ / 8,000 h/year

- Calculate the overall plant fuel rate and heat rate of the gas turbine and its electrical efficiency. **[5 Marks]**
- Find the HRSG thermal efficiency (with auxiliary firing) and the flue gas exit temperature. **[6 Marks]**
- Determine the energy utilisation factor (EUF). **[3 Marks]**
- Compare the hourly energy cost with separate production (grid power + NG boiler) and find the annual benefit. Also express specific steam (kg/kg fabric) and power (kWh/kg fabric). **[6 Marks]**

OR

N-4 (d) Cement plant – raw mill heat balance and hot gas requirement

A closed-circuit raw mill in a cement plant in Andhra Pradesh grinds and dries limestone-based raw meal using preheater exit gas. Data:

Symbol	Parameter	Value
F _q	Fresh feed	160 TPH

Mf / Mp	Feed / product surface moisture	7.0 % / 1.0 %
Fs	Specific heat of raw material	0.21 kcal/kg °C
Ght / Ghs	Hot gas temperature / specific heat	310 °C / 0.34 kcal/Nm ³ °C
Eg / Egt / Egs	Dedusting (mill exit) gas / temperature / specific heat	2,40,000 m ³ /h / 95 °C / 0.31 kcal/Nm ³ °C
Fa	False air (% of Eg)	12 %
Amb / Ambs	Ambient temperature / specific heat of air	33 °C / 0.30 kcal/Nm ³ °C
Rt	Reference temperature	20 °C
P	Mill motor power (motor 95 %, gearbox 98 %)	3,650 kW
Ra / Rf	Radiation area / loss factor	210 m ² / 50 kcal/m ² °C

- a) Calculate the heat outputs: heat to raw material, heat to dedusting gases, radiation loss, heat to evaporate moisture and heat to false air. **[8 Marks]**
- b) Calculate the heat inputs from fresh feed, grinding power and false air. **[4 Marks]**
- c) Determine the hot gas required (Nm³/h) and its share of the total heat demand, and the mill specific power (kWh/t). **[4 Marks]**
- d) If false air is reduced to 5 % by sealing, find the reduction in hot gas requirement. **[4 Marks]**

MODEL ANSWER

N-4 (a) Solution — Thermal power station

a) Economiser (4 marks)

Given: FW 252 → 286 °C at 7,60,000 kg/h; gas inlet 465 °C.

Formula: Effectiveness = (FW out – FW in)/(gas in – FW in) × 100; heat pick-up = m × 1 × (FW out – FW in) (Guidebook Ex. 11.3).

Substitution: $\epsilon = (286 - 252)/(465 - 252) \times 100 = 34/213 \times 100$

$Q = 7,60,000 \times 1 \times 34$

Result: $\epsilon_{eco} = 16.0 \%$; heat pick-up = 25,840 Mcal/h (30.1 MW).

b) Air preheater (4 marks)

Given: Air 36 → 318 °C at 6,80,000 kg/h; gas inlet 372 °C; Cp air 0.24.

Formula: APH effectiveness = (air out – air in)/(gas in – air in) × 100; heat pick-up = m_{air} × 0.24 × (air out – air in).

Substitution: $\epsilon = (318 - 36)/(372 - 36) \times 100 = 282/336 \times 100$

$Q = 6,80,000 \times 0.24 \times 282$

Result: $\epsilon_{APH} = 83.9 \%$; heat pick-up = 46,022 Mcal/h (53.5 MW).

c) APH leakage (3 marks)

Given: O₂ in = 3.6 %; O₂ out = 6.2 %.

Formula: APH leakage % = (O₂ out – O₂ in)/(21 – O₂ out) × 100.

Substitution: Leakage = (6.2 – 3.6)/(21 – 6.2) × 100 = 2.6/14.8 × 100

Result: APH leakage = 17.6 % – much higher than a well-sealed regenerative APH should show – compare with the design / PG test value.

d) ID fan saving after sealing (5 marks)

Given: Outlet gas = 1,250 t/h at 17.57 % leakage; new leakage 7 %; ID fans 3,200 kW; 7,800 h; ₹ 3.20/kWh.

Formula: Gas entering APH = outlet/(1 + L/100); new outlet = inlet × 1.07; $P_{\text{new}} = P \times (\text{new flow/old flow})$.

Substitution: Gas in = 1,250/1.1757 = 1,063.2 t/h; new outlet = 1,063.2 × 1.07 = 1,137.6 t/h

$P_{\text{new}} = 3,200 \times 1,137.6/1,250 = 2,912.4 \text{ kW}$; saving = 287.6 kW × 7,800 h

Result: ID fan saving = 287.6 kW → 22,43,531 kWh/year = ₹ 71,79,299 (₹ 71.79 lakh) per year.

e) Restoring APH exit gas temperature (4 marks)

Given: m = 6.9 kg/kg; $C_p = 0.24$; $\Delta T = 152 - 135 = 17^\circ\text{C}$; GCV 3,600; $\eta = 86.2\%$; coal 172 t/h; ₹ 3,900/t; 7,800 h.

Formula: $\Delta L = m \times C_p \times \Delta T / \text{GCV} \times 100$; coal_new = coal × $\eta / (\eta + \Delta L)$.

Substitution: $\Delta L = 6.9 \times 0.24 \times 17 / 3,600 \times 100 = 0.782\%$

Coal_new = 172 × 86.2/(86.982) = 170.45 t/h → saving 1.546 t/h × 7,800 h

Result: Boiler efficiency rises by ≈ 0.78 percentage points; coal saving ≈ 12,061 t/year = ₹ 4,70,39,752 (₹ 4.70 crore) per year.

Note: part of the high exit temperature is the cold-end dilution effect of leakage air; APH basket cleaning (soot blowing) and seal repair should be taken up together.

N-4 (b) Solution — Integrated steel plant

a) Gross input and net SEC (7 marks)

Given: Quantities and calorific values as tabulated; production 30,00,000 tcs.

Formula: Energy (Gcal) = quantity × CV; net = gross – exports; SEC = net/tcs.

Substitution: Coking coal = 22,50,000 t × 7,000 × 10³ kcal/t = 1,57,50,000 Gcal; PCI = 34,56,000 Gcal

NG = 12,00,00,000 × 8,900 = 10,68,000 Gcal; power = 1,65,00,00,000 × 860 = 14,19,000 Gcal; gross = 2,16,93,000 Gcal

Exports: BF gas 14,25,000 Gcal + power 1,54,800 Gcal → net = 2,01,13,200 Gcal

Result: Gross SEC = 7.23 Gcal/tcs; net SEC = 6.70 Gcal/tcs – within the Indian range of 6.17–8 Gcal/tcs (Table 12.1) but 49 % above the world level of about 4.5 Gcal/tcs.

b) Top-pressure recovery turbine (5 marks)

Given: TRT generation = 11,00,00,000 kWh/year; 860 kcal/kWh (≈ 3,601 kJ/kWh); ₹ 7.10/kWh.

Formula: Energy credit = kWh × 860; SEC_new = (net – credit)/tcs.

Substitution: Credit = 11,00,00,000 × 860 = 94,600 Gcal

SEC_new = (2,01,13,200 – 94,600)/30,00,000

Result: SEC = 6.673 Gcal/tcs (↓ 0.032); saving = ₹ 78,10,00,000 (₹ 78.10 crore) per year. TRT recovers BF top-gas pressure energy without burning any fuel.

c) Higher PCI (8 marks)

Given: Coke = 15,30,000 t; hot metal = 31,50,000 t; +40 kg PCI/thm; replacement 0.85; coke yield 0.68; ₹ 32,000/t coke; ₹ 14,500/t PCI.

Formula: Coke rate = coke/thm; coke saved = thm × 40 × 0.85; coking coal saved = coke saved/0.68; $\Delta E = \text{coal saved} \times 7,000 - \text{PCI added} \times 7,200$; benefit = coke saved × price – PCI added × price.

Substitution: Coke rate = 15,30,000 × 1,000/31,50,000 = 485.7 kg/thm → new = 485.7 – 34 = 451.7 kg/thm

Coke saved = 31,50,000 × 0.034 = 1,07,100 t; coking coal saved = 1,57,500 t; PCI added = 1,26,000 t

$\Delta E = 1,57,500 \times 7.0 - 1,26,000 \times 7.2 = 1,95,300 \text{ Gcal} \rightarrow \Delta \text{SEC} = 0.065 \text{ Gcal/tcs}$

Benefit = 1,07,100 × 32,000 – 1,26,000 × 14,500

Result: New coke rate = 451.7 kg/thm; SEC (after TRT and PCI) = 6.61 Gcal/tcs; net benefit = ₹ 1,60,02,00,000 (₹ 160.02 crore) per year.

N-4 (c) Solution — GT cogeneration in a textile unit

a) Fuel rate, heat rate, efficiency (5 marks)

Given: GT fuel = 1,560 Sm³/h; NCV 8,900 kcal/Sm³ ($\approx$ 37.3 MJ/Sm³); power 4,850 kW.

Formula: Fuel rate = Sm³/h $\div$ kW; heat rate = fuel rate $\times$ CV; $\eta_e = 860/\text{heat rate} \times 100$.

Substitution: Fuel rate = 1,560/4,850 = 0.3216 Sm³/kWh

Heat rate = 0.3216 $\times$ 8,900

Result: Fuel rate = 0.3216 Sm³/kWh; heat rate = 2,863 kcal/kWh ($\approx$ 11,987 kJ/kWh); $\eta_e = 30.0\%$ (NCV).

b) HRSG efficiency and exit temperature (6 marks)

Given: Exhaust 16.8 kg/s = 60,480 kg/h at 530 °C; Cp 0.26; aux NG 120 Sm³/h; steam 12,600 kg/h; $h_s - h_{fw} = 665 - 105 = 560$ kcal/kg ($\approx$ 2,345 kJ/kg).

Formula: $\eta_{\text{HRSG}} = \text{steam} \times (h_s - h_{fw}) \times 100 / [(\text{exhaust flow} \times C_p \times T_{\text{in}}) + (\text{aux fuel} \times \text{CV})]$ (Guidebook Ch. 3);

$T_{\text{exit}} = [\text{exhaust heat} + \text{aux heat} - \text{steam heat}] / (m \times C_p)$.

Substitution: Numerator = 12,600 $\times$ 560 = 70,56,000 kcal/h ($\approx$ 8,205 kW)

Denominator = 60,480 $\times$ 0.26 $\times$ 530 + 120 $\times$ 8,900 = 94,02,144 kcal/h ($\approx$ 10.9 MW)

$T_{\text{exit}} = (94,02,144 - 70,56,000) / (60,480 \times 0.26)$

Result: $\eta_{\text{HRSG}} = 75.0\%$; gas exit temperature $\approx 149^\circ\text{C}$.

c) EUF (3 marks)

Given: Power 4,850 kW; useful heat = steam heat; total NG = 1,680 Sm³/h $\times$ 8,900.

Formula: EUF = $(P \times 860 + Q_{\text{steam}}) / (\text{fuel} \times \text{CV}) \times 100$.

Substitution: EUF = $(4,850 \times 860 + 70,56,000) / (1,680 \times 8,900) \times 100$

Result: EUF = 75.1 %.

d) Economics versus separate production (6 marks)

Given: NG ₹ 48/Sm³; grid ₹ 8.60/kWh; boiler 88 % on NCV; 8,000 h; fabric 5,200 kg/h.

Formula: Separate cost = grid power + boiler NG for same steam; benefit = separate – cogeneration cost.

Substitution: Cogeneration: 1,680 $\times$ 48 = ₹ 80,640/h

Separate: boiler NG = 70,56,000 / (0.88 $\times$ 8,900) = 900.9 Sm³/h $\rightarrow$ ₹ 43,244/h; grid = 4,850 $\times$ 8.60 = ₹ 41,710/h; total ₹ 84,954/h

Specific steam = 12,600/5,200; specific power = 4,850/5,200

Result: Benefit = ₹ 4,314/h $\rightarrow$ ₹ 3,45,13,013 (₹ 3.45 crore) per year. Specific steam = 2.42 kg/kg fabric; specific power = 0.933 kWh/kg fabric.

Comment: the benefit is sensitive to the gas price and grid tariff; heat recovery from jet-dyeing drain and stenter exhaust (Ch. 15) lowers the steam demand and improves the heat-to-power match.

N-4 (d) Solution — Raw mill heat balance

a) Heat outputs (8 marks)

Given: $F_q = 1,60,000$ kg/h; $E_{gt} = 95^\circ\text{C}$; $R_t = 20^\circ\text{C}$; $E_g = 2,40,000$ m³/h; $F_a = 12\%$; $R_a = 210$ m²; $R_f = 50$.

Formula: $Hop = Fq \times Fs \times (Egt - Rt - 5)$; $Hog = Eg \times Egs \times (Egt - Rt) \times 273 / (Egt + 273)$; $Hor = Ra \times Rf \times (Egt - Rt)$; $W = Fq \times (Mf - Mp) / (100 - Mp)$; $How = W \times (540 + Egt - Amb)$; $Hoa = Eg \times Fa / 100 \times Ambs \times (Egt - Rt)$ (Guidebook Ex. 13.2).

Substitution: $Hop = 1,60,000 \times 0.21 \times 70 = 23,52,000 \text{ kcal/h}$ ($\approx 2,735 \text{ kW}$); $Hog = 2,40,000 \times 0.31 \times 75 \times 273 / 368 = 41,39,511 \text{ kcal/h}$ ($\approx 4,813 \text{ kW}$)

$Hor = 210 \times 50 \times 75 = 7,87,500 \text{ kcal/h}$ ($\approx 916 \text{ kW}$); $W = 1,60,000 \times 6 / 99 = 9,697 \text{ kg/h}$; $How = 9,697 \times (540 + 95 - 33) = 58,37,576 \text{ kcal/h}$ ($\approx 6,788 \text{ kW}$)

$Hoa = 2,40,000 \times 0.12 \times 0.30 \times 75 = 6,48,000 \text{ kcal/h}$ ($\approx 753 \text{ kW}$)

Result: Total heat output $Ho = 1,37,64,587 \text{ kcal/h}$ (16.01 MW): moisture evaporation 42.4 %, dedusting gas 30.1 %, raw material 17.1 %, radiation 5.7 %, false air 4.7 %.

b) Heat inputs (other than hot gas) (4 marks)

Given: $Amb = 33^\circ\text{C}$; $P = 3,650 \text{ kW}$; $\eta_{\text{motor}} 0.95$; $\eta_{\text{gear}} 0.98$.

Formula: $Hif = Fq \times Fs \times (Amb - Rt)$; $Hip = P \times \eta_m \times \eta_g \times 860$; $Hia = Eg \times Fa / 100 \times Ambs \times (Amb - Rt)$.

Substitution: $Hif = 1,60,000 \times 0.21 \times 13 = 4,36,800 \text{ kcal/h}$ ($\approx 508 \text{ kW}$); $Hip = 3,650 \times 0.95 \times 0.98 \times 860 = 29,22,409 \text{ kcal/h}$ ($\approx 3,398 \text{ kW}$)

$Hia = 2,40,000 \times 0.12 \times 0.30 \times 13 = 1,12,320 \text{ kcal/h}$ ($\approx 131 \text{ kW}$)

Result: $Hi = 34,71,529 \text{ kcal/h}$ ($\approx 4,037 \text{ kW}$).

c) Hot gas requirement (4 marks)

Given: $Ho = 1,37,64,587$; $Hi = 34,71,529 \text{ kcal/h}$ ($\approx 4,037 \text{ kW}$); $Ght = 310^\circ\text{C}$; $Ghs = 0.34$.

Formula: $Gh \times Ghs \times (Ght - Rt) = Ho - Hi$; specific power = P / Fq .

Substitution: $Gh = (1,37,64,587 - 34,71,529) / (0.34 \times 290)$

Specific power = $3,650 / 160$

Result: Hot gas required = $1,04,392 \text{ Nm}^3/\text{h}$, supplying 74.8 % of the heat demand; mill specific power = 22.81 kWh/t.

d) Effect of reducing false air to 5 % (4 marks)

Given: $Fa = 5\%$; other data unchanged.

Formula: Recompute Hoa and Hia with $Fa = 5\%$ and solve for Gh again.

Substitution: $Hoa = 2,40,000 \times 0.05 \times 0.30 \times 75 = 2,70,000 \text{ kcal/h}$ ($\approx 314 \text{ kW}$); $Hia = 2,40,000 \times 0.05 \times 0.30 \times 13 = 46,800 \text{ kcal/h}$ ($\approx 54.4 \text{ kW}$)

$Gh = [Ho - Hi (\text{new})] / (0.34 \times 290)$

Result: Hot gas = $1,01,223 \text{ Nm}^3/\text{h}$; reduction = $3,169 \text{ Nm}^3/\text{h}$ ($3,12,480 \text{ kcal/h}$ ($\approx 363 \text{ kW}$)), freeing hot gas for coal mill drying and lowering mill fan load.

..... End of Section - III