

PRACTICE PAPER — SET 3

NATIONAL CERTIFICATION EXAMINATION FOR ENERGY MANAGERS & ENERGY AUDITORS

PAPER – 2: ENERGY EFFICIENCY IN THERMAL UTILITIES

Question Code: 3P2CEM — SOLUTIONS & MODEL ANSWERS

Duration: 3 HRS Max. Marks: 150

General instructions:

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

Disclaimer: Independent practice paper prepared by Beyond Smart Cities / Beyond Xlearn (learn.beyondsmartcities.in). It is not affiliated with, endorsed by or issued by the Bureau of Energy Efficiency (BEE) or the National Productivity Council (NPC). All questions are original and follow the NCE pattern; the official BEE Guidebook (Book 2 – Energy Efficiency in Thermal Utilities) is the authority.

Section – I: OBJECTIVE TYPE

Marks: 50 x 1 = 50

(i) Answer all 50 questions

(ii) Each question carries one mark

1. A captive power plant receives a high-moisture coal with proximate analysis (as received): fixed carbon 30%, volatile matter 26%, ash 26% and moisture 18%. No ultimate analysis is available. Using the guidebook correlation $\%C = 0.97C + 0.7(VM - 0.1A) - M(0.6 - 0.01M)$, where C is fixed carbon, the estimated carbon content is about:

[3P2CEM001]

- a) 21.5%
- b) 37.9%**
- c) 45.5%
- d) 53.0%

Correct answer: (b) 37.9%.

Given: FC = 30%, VM = 26%, A = 26%, M = 18% (as received); moisture above 15%, so the correlation applies.

Formula: $\%C = 0.97 \times FC + 0.7 \times (VM - 0.1A) - M \times (0.6 - 0.01M)$.

Substitution: $0.97 \times 30 = 29.10$; $0.7 \times (26 - 2.6) = 16.38$; $18 \times (0.6 - 0.18) = 7.56$; $\%C = 29.10 + 16.38 - 7.56$.

Result: **Carbon $\approx$ 37.9%**, the figure then used for stoichiometric air and CO₂ estimates.

Why not the others: (c) 45.5% drops the moisture correction term, which the relation subtracts for wet coal. (a) 21.5% subtracts the full ash (VM – A) instead of 0.1A, so the volatile term vanishes. (d) 53.0% adds the moisture term instead of subtracting it, a sign error.

Practitioners often equate carbon with fixed carbon (30%), forgetting that volatile matter also carries carbon; the correlation corrects this.

Ref: BEE Guidebook Paper 2, Ch. 1 (Table 1.7, relationship between ultimate and proximate analysis).

Chapter 1: Fuels and Combustion | Carbon content of high-moisture coal from proximate analysis

2. Three non-coking coal lots are offered to a boiler house with as-received GCVs of Lot P 5,050 kcal/kg ($\approx 21,143$ kJ/kg), Lot Q 4,350 kcal/kg ($\approx 18,213$ kJ/kg) and Lot R 3,250 kcal/kg ($\approx 13,607$ kJ/kg). As per the gradation of Indian coal in the guidebook, the grades are: [3P2CEM002]

a) **P grade C; Q grade D; R grade F**

b) P grade B; Q grade D; R grade E

c) P grade C; Q grade E; R grade F

d) P grade D; Q grade E; R grade G

Correct answer: (a) P grade C; Q grade D; R grade F.

Guidebook GCV bands (kcal/kg): grade A above 6,200; B 5,600–6,200; C 4,940–5,600; D 4,200–4,940; E 3,360–4,200; F 2,400–3,360; G 1,300–2,400.

Placing the lots: 5,050 lies in 4,940–5,600, so P is grade C; 4,350 lies in 4,200–4,940, so Q is grade D; 3,250 lies in 2,400–3,360, so R is grade F.

The guidebook notes that grades D, E and F are the ones normally available to Indian industry, so most boilers are designed for 3,300–4,900 kcal/kg coal ($\approx 20,515$ kJ/kg coal).

Why not the others: (b) places P in grade B, which starts only at 5,600 kcal/kg ($\approx 23,446$ kJ/kg), and R in grade E, which ends at 3,360 kcal/kg ($\approx 14,068$ kJ/kg). (c) moves Q into grade E although 4,350 is above the 4,200 boundary. (d) shifts every lot one grade lower, a common slip when the bands are remembered from the bottom.

Ref: BEE Guidebook Paper 2, Ch. 1 (Solid fuels, gradation of Indian coal).

Chapter 1: Fuels and Combustion | Grades of Indian coal by GCV

3. A forging unit fires 1,560 kg/h of furnace oil (specific gravity 0.93) for 20 hours a day. As per the guidebook norm for the minimum storage capacity of fuel oil, the net storage volume to be provided is about: [3P2CEM003]

a) 235 kL

b) 312 kL

c) **335 kL**

d) 403 kL

Correct answer: (c) 335 kL.

Given: oil = 1,560 kg/h; 20 h/day; specific gravity 0.93 (density 930 kg/m³); guidebook norm = at least 10 days' consumption.

Formula: Storage volume = (mass rate $\times$ hours/day $\times$ days)/density.

Substitution: Mass for 10 days = $1,560 \times 20 \times 10 = 3,12,000$ kg; volume = $3,12,000/930 = 335.5$ m³.

Result: **Minimum net storage $\approx$ 335 kL**, plus bund walls and room for annual tank cleaning.

Why not the others: (b) 312 kL treats kilograms as litres, ignoring that furnace oil is lighter than water. (d) 403 kL assumes 24-hour firing although the unit works 20 hours a day. (a) 235 kL sizes for only 7 days, below the 10-day minimum suggested to cover supply disruptions.

Storage tanks should also carry heating coils and level indication, and the guidebook warns that one leaking drop per second loses about 4,000 litres a year.

Ref: BEE Guidebook Paper 2, Ch. 1 (Storage of fuel oil).

Chapter 1: Fuels and Combustion | Furnace oil storage tank sizing

4. In a chain-grate stoker firing coal with 38% carbon and GCV 3,800 kcal/kg ($\approx 15,910$ kJ/kg), flue gas analysis indicates that 3% of the carbon fired burns only to CO instead of CO₂. Using the guidebook heats of reaction, the resulting loss as a percentage of the fuel heat input is about: [3P2CEM004]

a) 0.73%

b) **1.70%**

c) 2.43%

d) 4.46%

Correct answer: (b) 1.70%.

Given: carbon = 0.38 kg/kg coal; fraction burnt to CO = 3%; GCV = 3,800 kcal/kg ($\approx 15,910$ kJ/kg); C $\rightarrow$ CO₂ releases 8,084 kcal/kg ($\approx 33,846$ kJ/kg) C; C $\rightarrow$ CO releases 2,430 kcal/kg ($\approx 10,174$ kJ/kg) C.

Formula: Loss = carbon to CO $\times$ (8,084 – 2,430); % loss = loss/GCV $\times$ 100.

Substitution: carbon to CO = $0.38 \times 0.03 = 0.0114$ kg/kg coal; loss = $0.0114 \times 5,654 = 64.5$ kcal/kg coal; % = $64.5/3,800 \times 100$.

Result: **Loss $\approx$ 1.70% of heat input.**

Why not the others: (a) 0.73% values the loss at 2,430 kcal/kg ($\approx 10,174$ kJ/kg), which is the heat actually released when carbon burns to CO. (c) 2.43% uses the full 8,084 kcal/kg ($\approx 33,846$ kJ/kg), as if that carbon produced no heat at all. (d) 4.46% applies 3% to the whole coal instead of to its carbon fraction.

CO in flue gas therefore signals poor air distribution or mixing.

Ref: BEE Guidebook Paper 2, Ch. 1 (Principles of combustion, heat of reaction).

Chapter 1: Fuels and Combustion | Heat loss from partial combustion of carbon to CO

5. An imported liquid fuel has carbon 85%, hydrogen 11.5% and sulphur 2.5% by mass, the rest being inerts. Using the guidebook heat releases for complete combustion of carbon, hydrogen and sulphur, the theoretical gross heat release per kg of fuel is about: [3P2CEM005]

a) 5,447 kcal/kg ($\approx 22,805$ kJ/kg)

b) 6,927 kcal/kg ($\approx 29,002$ kJ/kg)

c) 9,649 kcal/kg ($\approx 40,398$ kJ/kg)

d) 10,253 kcal/kg ($\approx 42,927$ kJ/kg)

Correct answer: (d) 10,253 kcal/kg ($\approx 42,927$ kJ/kg).

Given: C = 0.85, H = 0.115, S = 0.025 kg/kg; heat releases C $\rightarrow$ CO₂ 8,084 kcal/kg ($\approx 33,846$ kJ/kg), H $\rightarrow$ H₂O 28,922 kcal/kg ($\approx 1,21,091$ kJ/kg), S $\rightarrow$ SO₂ 2,224 kcal/kg ($\approx 9,311$ kJ/kg).

Formula: Heat release = Σ (mass fraction $\times$ heat of combustion).

Substitution: $0.85 \times 8,084 = 6,871$; $0.115 \times 28,922 = 3,326$; $0.025 \times 2,224 = 56$; total = $6,871 + 3,326 + 56$.

Result: **Gross heat release $\approx$ 10,253 kcal/kg ($\approx 42,927$ kJ/kg)**, consistent with furnace oil GCV of about 10,500 kcal/kg ($\approx 43,961$ kJ/kg).

Why not the others: (c) 9,649 kcal/kg ($\approx 40,398$ kJ/kg) deducts the latent heat of water formed ($9H \times 584$), giving a net value. (b) 6,927 kcal/kg ($\approx 29,002$ kJ/kg) omits hydrogen, which contributes nearly one-third of the heat. (a) 5,447 kcal/kg ($\approx 22,805$ kJ/kg) uses 2,430 kcal/kg ($\approx 10,174$ kJ/kg) for carbon, the value for burning only to CO.

Ref: BEE Guidebook Paper 2, Ch. 1 (Principles of combustion; calorific value).

Chapter 1: Fuels and Combustion | Theoretical heat release from elemental composition

6. A process plant needs one transfer pump for LSHS and another for high speed diesel. Which pairing follows the guidebook advice on pumping fuel oils? [3P2CEM006]

a) Gear pump for LSHS; centrifugal pump for HSD

b) Centrifugal pump for LSHS; gear pump for HSD

c) Centrifugal pumps for both, with larger motors

d) Diaphragm pump for LSHS; piston pump for HSD

Correct answer: (a) Gear pump for LSHS; centrifugal pump for HSD.

The guidebook states that heavy fuel oils are best pumped by positive displacement pumps such as gear pumps, because they can get the fuel moving even when it is cold. A centrifugal pump is not recommended for heavy oils, since its efficiency drops sharply and the power required increases as viscosity rises; centrifugal or turbine pumps are best for light fuels such as diesel.

The guidebook also notes that a gear pump handling LDO gives 7,000–10,000 hours of service, and that furnace oil is not easily pumpable below about 25 °C, so tanks and lines need heating.

Why not the others: (b) reverses the logic, putting the viscosity-sensitive centrifugal pump on the heaviest oil. (c) bigger motors do not cure the efficiency collapse of centrifugal pumps on viscous LSHS. (d) diaphragm pumps have shorter service lives, and piston or diaphragm pumps are meant for high-pressure duties, not routine transfer.

Ref: BEE Guidebook Paper 2, Ch. 1 (Pumping of fuel oil).

Chapter 1: Fuels and Combustion | Pump selection for heavy and light fuel oils

7. Select the WRONG statement about draft systems used in fuel-fired equipment: [3P2CEM007]

- a) Natural draft is produced by an ID fan at the chimney base**
- b) Balanced draft keeps a boiler furnace slightly below atmospheric pressure
- c) Forced draft pushes air and products through the unit and stack
- d) Natural draft arises from the weight difference of hot gas and air

Correct answer: (a) Natural draft is produced by an ID fan at the chimney base.

The question asks for the wrong statement. Natural draft is created only by the chimney: the column of hot, light flue gas inside weighs less than the equivalent column of outside air, and this difference drives the flow. No fan is involved; a fan near the chimney base gives induced draft, not natural draft.

The other statements are correct as per the guidebook. Balanced draft uses both an FD fan and an ID fan and keeps a boiler furnace about 0.05–0.10 inch water column below atmosphere, preventing hot gas leakage. Forced draft delivers air through the burners or grate and pushes the products through the unit and up the stack. The stack-effect explanation of natural draft is the standard one.

Why not the others: (b), (c) and (d) are accurate descriptions, so they are not the answer. Confusing natural with induced draft is a frequent exam error.

Ref: BEE Guidebook Paper 2, Ch. 1 (Draft system: natural, mechanical, forced, induced and balanced draft).

Chapter 1: Fuels and Combustion | Draft systems in combustion equipment

8. A 25 TPH oil-fired boiler with widely fluctuating steam demand currently uses High/Low/Off firing control. Which upgrade and reason are consistent with the guidebook? [3P2CEM008]

- a) ON/OFF control, as full firing always gives best efficiency
- b) Modulating control, matching firing rate over the whole range**
- c) No change, as High/Low/Off is ideal for large boilers
- d) Manual damper control, as automatic controls waste fuel

Correct answer: (b) Modulating control, matching firing rate over the whole range.

The guidebook describes three levels of combustion control. ON/OFF control, where the burner fires at full rate or is off, suits only small boilers. High/Low/Off control, with two firing rates, is fitted to medium-sized boilers. Modulating control continuously adjusts the firing rate, and with it the air-fuel ratio, to match load over the whole operating range, and is used on large boilers.

A 25 TPH boiler with fluctuating demand therefore benefits from modulating control: fewer purge cycles, steadier pressure and correct excess air at every load, which improves efficiency.

Why not the others: (a) ON/OFF cycling on a large boiler causes frequent purges, pressure swings and thermal stress. (c) High/Low/Off is a medium-boiler scheme and cannot track a widely varying load. (d) manual damper setting cannot follow load changes and normally leaves excess air high.

Ref: BEE Guidebook Paper 2, Ch. 1 (Combustion controls: ON/OFF, High/Low/Off and modulating control).

Chapter 1: Fuels and Combustion | Choice of combustion control scheme

9. A paper group plans three biomass units: (i) wet bark at about 50% moisture with mixed particle sizes, (ii) fine, dry sander dust from a board plant, and (iii) a 40 MWth plant firing mixed biomass. The guidebook-consistent choice of combustion technology is: [3P2CEM009]

- a) (i) grate furnace; (ii) dust combustion; (iii) fluidised bed**
- b) (i) dust combustion; (ii) grate furnace; (iii) fluidised bed

c) (i) fluidised bed; (ii) grate furnace; (iii) dust combustion

d) (i) grate furnace; (ii) fluidised bed; (iii) dust combustion

Correct answer: (a) (i) grate furnace; (ii) dust combustion; (iii) fluidised bed.

Grate furnaces tolerate fuels with high moisture (bark often has 25–55%), varying particle size and high ash, so they suit wet bark. Dust combustion is meant for small, dry particles such as sander dust and wood flour; it needs an auxiliary burner for start-up but permits quick load changes. Fluidised beds become economical for larger plants, above about 30 MWth, offering fuel flexibility and low excess air, though with higher dust load in the flue gas.

Why not the others: (b) fires wet, mixed-size bark in suspension, which fails because large wet pieces do not burn out in flight. (c) puts bark in a fluidised bed and sander dust on a grate, where fine dust would sift through or blow off. (d) uses dust combustion for a 40 MWth mixed-fuel plant, ignoring its need for fine, dry fuel.

Volatiles supply about 70% of biomass heat, so secondary air is vital.

Ref: BEE Guidebook Paper 2, Ch. 1 (Biomass combustion technologies).

Chapter 1: Fuels and Combustion / Matching biomass fuels to combustion systems

10. A boiler house has no flue gas monitoring at present. As per the guidebook, regular checks with a portable flue gas analyser and draft gauges can cut excess air by up to about X, and a continuous oxygen analyser with local readout can then give a further reduction of about Y. X and Y are: [3P2CEM010]

a) Up to 5% first, then a further 40–50%

b) Up to 20% first, then a further 10–15%

c) Up to 50% first, then no further benefit

d) Up to 10–15% first, then a further 20%

Correct answer: (b) Up to 20% first, then a further 10–15%.

The guidebook lists excess air control options in increasing sophistication. A portable oxygen or CO₂ analyser with draft gauges, used for periodic readings that guide the operator's manual adjustment, can reduce excess air by up to 20%. A continuous oxygen analyser with a local readout, mounted on the stack, lets the operator trim air continuously and can achieve a further 10–15% reduction over the portable approach. Beyond these come remote-controlled damper positioners and automatic oxygen trim, while an automatic stack damper is justified only on large systems.

Since every 1% cut in excess air matters at the stack, these low-cost steps are usually the first boiler ECO.

Why not the others: (a) understates the portable-analyser benefit and overstates the continuous analyser. (c) a 50% cut from spot checks alone is unrealistic, and continuous monitoring still adds benefit. (d) reverses the two figures given in the guidebook.

Ref: BEE Guidebook Paper 2, Ch. 2 (Energy conservation opportunities: excess air control).

Chapter 2: Boilers / Staged excess air control upgrades

11. An oil-fired boiler shows CO and visible smoke at the stack, although oxygen readings indicate normal or even high excess air. According to the guidebook, the most likely cause is: [3P2CEM011]

a) Burner faults: wrong viscosity, worn or carbonised tips

b) Air shortage: the FD damper must be opened further

c) Low stack temperature causing acid dew point condensation

d) Excess sulphur in oil reacting with CO to form smoke

Correct answer: (a) Burner faults: wrong viscosity, worn or carbonised tips.

The guidebook explains that incomplete combustion can arise from a shortage of air or from poor fuel–air mixing. With oil and gas firing, the presence of CO or smoke while excess air is normal or high indicates burner system problems: oil at the wrong temperature or viscosity, worn or damaged burner tips, carbonisation on tips, and deterioration of diffusers or spinner plates. These spoil atomisation, so droplets burn out incompletely even with plenty of oxygen.

The remedy is maintenance and correct oil preheating, not more air, which would only raise stack losses further.

Why not the others: (b) contradicts the measurement, since oxygen already shows normal or high excess air. (c) acid dew point concerns corrosion of cold surfaces, not CO formation. (d) sulphur forms SO₂ and SO₃; it does not convert CO into smoke.

With coal firing, unburnt carbon in ash is the corresponding symptom and can exceed 2% of heat supplied.

Ref: BEE Guidebook Paper 2, Ch. 2 (Energy conservation opportunities: incomplete combustion).

Chapter 2: Boilers | Diagnosing CO and smoke at normal excess air

12. Which statement about supercritical boilers is CORRECT as per the guidebook? [3P2CEM012]

- a) They operate below 221.2 bar with a large steam drum
- b) Their latent heat is highest at the critical point
- c) They suit small process plants owing to low cost
- d) They have only economiser and superheater, no evaporator**

Correct answer: (d) They have only economiser and superheater, no evaporator.

At the critical point, 221.2 bar and 374.15 °C, water changes to steam without boiling and the latent heat becomes zero. A supercritical boiler operates above this pressure, so there is no distinct boiling stage: water is heated in the economiser section and passes directly into the superheater region. Such once-through units have no drum and no separate evaporator.

The guidebook notes that supercritical units reach plant efficiencies of about 40–42%, reduce fuel use and emissions per unit generated, and need special high-strength materials, making them suited to large utility plants.

Why not the others: (a) describes a subcritical drum boiler; operating below 221.2 bar is by definition not supercritical. (b) inverts the physics, since latent heat falls to zero at the critical point. (c) supercritical technology is costly and complex, used for large power stations, not small process plants.

Ref: BEE Guidebook Paper 2, Ch. 2 (Types of boilers: supercritical boilers).

Chapter 2: Boilers | Features of supercritical boilers

13. For a bituminous coal fired in a pulverised fuel boiler, the guidebook specification for grinding fineness is:

[3P2CEM013]

- a) Under 25% above 6 mm; 70–75% below 1 mm
- b) Under 2% above 300 µm; 70–75% below 75 µm**
- c) Under 2% above 75 µm; all of it below 10 µm
- d) Under 10% above 1 mm; 30–40% below 300 µm

Correct answer: (b) Under 2% above 300 µm; 70–75% below 75 µm.

In pulverised fuel firing, coal is ground to a fine powder so that less than 2% is above 300 microns and 70–75% is below 75 microns, for a bituminous coal. The powder is blown with part of the combustion air through burners into the furnace, where combustion takes place at 1,300–1,700 °C depending on coal grade, with a particle residence time of typically 2 to 5 seconds. Tangential firing, with burners at the corners creating a fireball, is a common arrangement.

Grinding too fine wastes mill power; grinding too coarse raises unburnt carbon in ash and furnace slagging.

Why not the others: (a) quotes millimetre sizes typical of stoker or FBC feed, far too coarse for suspension firing. (c) demands ultra-fine grinding that would sharply increase mill power without benefit. (d) is much coarser than the guidebook figure and would leave particles unburnt within the short residence time.

Ref: BEE Guidebook Paper 2, Ch. 2 (Pulverised fuel boiler).

Chapter 2: Boilers | Coal fineness for pulverised fuel firing

14. Why is the spreader stoker preferred over the chain-grate stoker in many industrial boilers that face fluctuating steam demand? [3P2CEM014]

- a) Coal must be of uniform size, so combustion is very stable
- b) More air at the grate feed end aids load following
- c) It burns lumps in a thick bed for long residence

d) Fines burn in suspension, so ignition responds almost instantly

Correct answer: (d) Fines burn in suspension, so ignition responds almost instantly.

In a spreader stoker, coal is continually fed into the furnace above a burning bed. The coal fines burn in suspension while the larger particles fall onto the grate and burn in a thin, fast-burning bed. Because part of the fuel burns in suspension, the firing responds quickly to load swings and ignition is almost instantaneous when the feed rate is increased. This is why the guidebook says the spreader stoker is favoured over other stokers in many industrial applications.

A chain-grate stoker, by contrast, carries a thick bed slowly through the furnace; it needs coal of uniform size and responds slowly.

Why not the others: (a) uniform sizing is a requirement of the chain grate, not an advantage of the spreader. (b) the higher air requirement at the feed end describes chain-grate operation, not load response. (c) a thick, slow bed is the chain-grate characteristic and makes load following slower.

Ref: BEE Guidebook Paper 2, Ch. 2 (Stoker fired boilers: spreader and chain-grate stokers).

Chapter 2: Boilers / Spreader stoker versus chain-grate stoker

15. Raw water analysis for a medium-pressure boiler shows calcium bicarbonate, magnesium sulphate, chlorides and dissolved silica. Which statement on how these impurities behave inside the boiler is CORRECT? [3P2CEM015]

a) Bicarbonate hardness is temporary and forms soft, removable sludge

b) Sulphate and chloride hardness is temporary and removed by boiling

c) Silica forms no scale and never deposits on turbine blades

d) Calcium bicarbonate forms hard scale that blowdown cannot remove

Correct answer: (a) Bicarbonate hardness is temporary and forms soft, removable sludge.

The guidebook distinguishes two forms of hardness. Alkaline or temporary hardness is due to calcium and magnesium bicarbonates; on heating these decompose to carbonates that settle as a soft sludge, which can be removed by blowdown, so this hardness is removed by boiling. Non-alkaline or permanent hardness is due to calcium and magnesium sulphates, chlorides and nitrates; these are not removed by boiling and form hard scale on heating surfaces, which is difficult to remove.

Silica can form hard silicate scale and, at higher pressures, is carried over with steam to deposit on turbine blades, so it is controlled closely.

Why not the others: (b) labels sulphate and chloride hardness as temporary, reversing the definitions. (c) ignores silica scale and its carry-over to turbine blades. (d) attributes hard, adherent scale to bicarbonates, which actually form soft sludge.

External treatment is preferred because internal treatment alone is not recommended for most boilers.

Ref: BEE Guidebook Paper 2, Ch. 2 (Boiler water treatment).

Chapter 2: Boilers / Hardness types and silica in feed water

16. A boiler failure analysis lists three cases: (i) thinning of tubes by fly-ash erosion near soot blowers, (ii) caustic gouging below deposits, and (iii) creep rupture of superheater tubes after overheating. As per the guidebook, these failure causes are respectively: [3P2CEM016]

a) Mechanical, chemical, metallurgical

b) Chemical, mechanical, metallurgical

c) Mechanical, metallurgical, chemical

d) Metallurgical, chemical, mechanical

Correct answer: (a) Mechanical, chemical, metallurgical.

The guidebook groups boiler tube failures into mechanical, chemical and metallurgical causes, together with gaps in quality assurance during manufacture, erection and repair.

Mechanical failures include erosion by fly ash, impingement of soot-blower steam on tubes, vibration and fatigue at supports, so case (i) is mechanical.

Chemical failures arise from water chemistry: caustic gouging under deposits, hydrogen embrittlement, oxygen pitting and stress corrosion cracking, so case (ii) is chemical.

Metallurgical failures involve changes in the metal, such as long-term creep and short-term overheating rupture in superheaters, graphitisation and weld defects, so case (iii) is metallurgical.

Why not the others: (b) swaps erosion and caustic attack. (c) calls caustic gouging metallurgical, although its root cause is concentrated caustic in water. (d) reverses the first and last cases.

Correct classification directs the remedy: soot-blower alignment for (i), water chemistry for (ii), and firing or temperature control for (iii).

Ref: BEE Guidebook Paper 2, Ch. 2 (Boiler tube failures).

Chapter 2: Boilers | Classifying boiler tube failure mechanisms

17. A 12 TPH boiler supplying a dairy runs at $10.5 \text{ kg/cm}^2(\text{g})$ ($\approx 10.3 \text{ bar(g)}$), although the highest process requirement is only $7 \text{ kg/cm}^2(\text{g})$ ($\approx 6.86 \text{ bar(g)}$). The annual fuel bill is ₹ 4.8 crore. As per the guidebook, the first pressure step and the likely saving are: [3P2CEM017]

a) Straight to $7 \text{ kg/cm}^2(\text{g})$ ($\approx 6.86 \text{ bar(g)}$), saving about ₹ 48–96 lakh/yr

b) To about $9.5 \text{ kg/cm}^2(\text{g})$ ($\approx 9.32 \text{ bar(g)}$), saving about ₹ 0.5–1.0 lakh/yr

c) To about $8.4 \text{ kg/cm}^2(\text{g})$ ($\approx 8.24 \text{ bar(g)}$), saving about ₹ 24–48 lakh/yr

d) To about $8.4 \text{ kg/cm}^2(\text{g})$ ($\approx 8.24 \text{ bar(g)}$), saving about ₹ 4.8–9.6 lakh/yr

Correct answer: (d) To about $8.4 \text{ kg/cm}^2(\text{g})$ ($\approx 8.24 \text{ bar(g)}$), saving about ₹ 4.8–9.6 lakh/yr.

Given: operating pressure $10.5 \text{ kg/cm}^2(\text{g})$ ($\approx 10.3 \text{ bar(g)}$); fuel bill ₹ 4.8 crore/yr; guidebook: reduce pressure in stages, no more than about 20% at a time, saving 1–2% of fuel.

Formula: First step = $0.80 \times$ present pressure; saving = $(1\% \text{ to } 2\%) \times$ fuel bill.

Substitution: $0.80 \times 10.5 = 8.4$; $0.01 \times 4,80,00,000 = 4,80,000$ and $0.02 \times 4,80,00,000 = 9,60,000$.

Result: **First step to about $8.4 \text{ kg/cm}^2(\text{g})$ ($\approx 8.24 \text{ bar(g)}$); saving $\approx$ ₹ 4.8–9.6 lakh/yr.**

Lower pressure lowers saturation and flue gas temperatures and radiation and blowdown losses, but raises specific volume, risking carry-over and higher velocity in mains.

Why not the others: (a) jumps directly to process pressure and assumes a 10–20% saving. (b) is only a 10% step with 0.1–0.2% saving. (c) overstates the saving at 5–10%.

Ref: BEE Guidebook Paper 2, Ch. 2 (Energy conservation opportunities: reduction of boiler steam pressure).

Chapter 2: Boilers | Limits of steam pressure reduction

18. A chemical plant needs 40 TPH of steam at $45 \text{ kg/cm}^2(\text{g})$ ($\approx 44.1 \text{ bar(g)}$) for a turbine, and a separate unit needs 8 TPH at $10.5 \text{ kg/cm}^2(\text{g})$ ($\approx 10.3 \text{ bar(g)}$) for process heating. Considering the guidebook limits of fire-tube boilers, the appropriate choice is: [3P2CEM018]

a) Fire-tube for both, as fire-tube shells suit any pressure

b) Water-tube for 40 TPH; three-pass packaged fire-tube for 8 TPH

c) Water-tube for 8 TPH; a fire-tube shell unit for 40 TPH

d) Fire-tube for 40 TPH, as fire-tube limits are only minimums

Correct answer: (b) Water-tube for 40 TPH; three-pass packaged fire-tube for 8 TPH.

The guidebook states that fire-tube boilers are generally used for relatively small steam capacities and low to medium pressures: up to about 25 TPH and 17.5 kg/cm^2 ($\approx 17.2 \text{ bar}$). Packaged three-pass fire-tube boilers, with high heat release rates, a small combustion space and complete factory assembly, are well suited to an 8 TPH, $10.5 \text{ kg/cm}^2(\text{g})$ ($\approx 10.3 \text{ bar(g)}$) duty.

The 40 TPH, $45 \text{ kg/cm}^2(\text{g})$ ($\approx 44.1 \text{ bar(g)}$) requirement exceeds both limits, so a water-tube boiler is needed; water circulates inside the tubes, which can withstand high pressure, and capacities are not limited by shell size.

Why not the others: (a) ignores the pressure and capacity limits of shells, whose thickness would become impractical at 45 kg/cm^2 ($\approx 44.1 \text{ bar}$). (c) reverses the logic, putting the large high-pressure duty on a shell boiler. (d) wrongly reads 25 TPH and 17.5 kg/cm^2 ($\approx 17.2 \text{ bar}$) as minimum values rather than upper limits.

Ref: BEE Guidebook Paper 2, Ch. 2 (Types of boilers: fire-tube, packaged and water-tube boilers).

Chapter 2: Boilers | Selecting fire-tube or water-tube boilers

19. To reduce cost, a project engineer proposes a 150 mm steam main instead of the 200 mm line originally designed, for the same steam flow and length. Using the guidebook relationship between pressure drop and pipe diameter, the pressure drop in the smaller line will be about: [3P2CEM019]

- a) 1.3 times the original
- b) 1.8 times the original
- c) 3.2 times the original
- d) 4.2 times the original**

Correct answer: (d) 4.2 times the original.

Given: $D_1 = 200$ mm, $D_2 = 150$ mm; same flow and length.

Formula: For a given flow, pressure drop varies inversely as the fifth power of diameter: $\Delta P_2/\Delta P_1 = (D_1/D_2)^5$.

Substitution: $D_1/D_2 = 200/150 = 1.333$; $1.333^5 = 1.333 \times 1.333 \times 1.333 \times 1.333 \times 1.333$.

Result: $\Delta P_2 \approx 4.2 \times \Delta P_1$.

A 25% smaller bore therefore more than quadruples the drop, so steam reaches users at lower pressure and the boiler must run higher to compensate, while velocity rises by $(4/3)^2 \approx 1.78$ times, possibly beyond the 30–40 m/s band for saturated steam.

Why not the others: (a) scales linearly with diameter. (b) uses the square of the ratio, which governs velocity, not pressure drop. (c) uses the fourth power, a frequent slip when recalling the relation.

The guidebook stresses sizing by both velocity and pressure drop; undersizing saves little capital but costs energy for years.

Ref: BEE Guidebook Paper 2, Ch. 3 (Steam pipe sizing and design).

Chapter 3: Steam System | Effect of downsizing a steam main on pressure drop

20. A shell boiler without a superheater delivers steam at 5 kg/cm²(a) (≈ 4.90 bar(a)) with a dryness fraction of 0.96. From the steam table, at this pressure $h_f = 152.13$ kcal/kg (≈ 637 kJ/kg), $h_{fg} = 503.90$ kcal/kg ($\approx 2,110$ kJ/kg) and $h_g = 656.03$ kcal/kg ($\approx 2,747$ kJ/kg). The total heat per kg of the steam delivered is about: [3P2CEM020]

- a) 483.7 kcal/kg ($\approx 2,025$ kJ/kg)
- b) 629.8 kcal/kg ($\approx 2,637$ kJ/kg)
- c) 635.9 kcal/kg ($\approx 2,662$ kJ/kg)**
- d) 656.0 kcal/kg ($\approx 2,747$ kJ/kg)

Correct answer: (c) 635.9 kcal/kg ($\approx 2,662$ kJ/kg).

Given: $h_f = 152.13$ kcal/kg (≈ 637 kJ/kg); $h_{fg} = 503.90$ kcal/kg ($\approx 2,110$ kJ/kg); dryness $x = 0.96$.

Formula: Total heat of wet steam $h = h_f + x \times h_{fg}$.

Substitution: $x \times h_{fg} = 0.96 \times 503.90 = 483.74$; $h = 152.13 + 483.74$.

Result: $h \approx 635.9$ kcal/kg ($\approx 2,662$ kJ/kg).

The guidebook notes that a boiler without a superheater delivers, at best, about 95–98% dry steam, so the latent heat available at the process is less than the steam table h_{fg} , and more steam must be supplied for the same duty.

Why not the others: (d) 656.0 kcal/kg ($\approx 2,747$ kJ/kg) is h_g , valid only for completely dry steam. (b) 629.8 kcal/kg ($\approx 2,637$ kJ/kg) multiplies the whole h_g by 0.96, wrongly applying dryness to the sensible heat as well. (a) 483.7 kcal/kg ($\approx 2,025$ kJ/kg) is only the latent heat carried, omitting the sensible heat h_f .

Ref: BEE Guidebook Paper 2, Ch. 3 (Properties of steam; dryness fraction; Table 3.1).

Chapter 3: Steam System | Total heat of wet steam from a shell boiler

21. A product tank held at 60 °C needs 500 kW. Option 1 heats it with saturated steam condensing at 120 °C (overall $U = 3,000 \text{ W/m}^2\text{K}$). Option 2 uses pressurised hot water entering at 140 °C and leaving at 100 °C (overall $U = 1,500 \text{ W/m}^2\text{K}$). The heat transfer area needed for option 2 is about: [3P2CEM021]

- a) 4.17 m²
- b) 5.56 m²
- c) 5.78 m²**
- d) 8.33 m²

Correct answer: (c) 5.78 m².

Given: $Q = 5,00,000 \text{ W}$; product 60 °C; hot water 140 → 100 °C; $U = 1,500 \text{ W/m}^2\text{K}$.

Formula: $A = Q/(U \times \text{LMTD})$; $\text{LMTD} = (\Delta T_1 - \Delta T_2)/\ln(\Delta T_1/\Delta T_2)$.

Substitution: $\Delta T_1 = 140 - 60 = 80$; $\Delta T_2 = 100 - 60 = 40$; $\text{LMTD} = 40/\ln 2 = 57.7 \text{ °C}$; $A = 5,00,000/(1,500 \times 57.7)$.

Result: **$A \approx 5.78 \text{ m}^2$** , against $5,00,000/(3,000 \times 60) = 2.78 \text{ m}^2$ with steam, about 2.1 times larger.

This reflects the guidebook point that condensing steam gives high film coefficients and a constant temperature, so heaters are smaller than hot water units.

Why not the others: (b) uses the arithmetic mean difference of 60 °C. (a) uses only the inlet difference of 80 °C. (d) uses only the outlet difference of 40 °C.

Ref: BEE Guidebook Paper 2, Ch. 3 (Steam versus hot water; steam as heating medium).

Chapter 3: Steam System / Heat transfer area: steam versus hot water heating

22. Select the CORRECT statement on steam trap failure modes: [3P2CEM022]

- a) A trap failed closed wastes no steam but cuts heating capacity**
- b) A trap failed open causes waterlogging but wastes no steam
- c) An inverted bucket trap usually fails closed, suiting turbine drains
- d) A trap failed closed always blows live steam to atmosphere

Correct answer: (a) A trap failed closed wastes no steam but cuts heating capacity.

A trap that fails closed stops discharging condensate. No live steam escapes, so there is no direct steam loss, but condensate backs up into the heat exchanger or coil, the heat transfer surface becomes waterlogged and heating capacity falls; water hammer is also possible. A trap that fails open passes live steam continuously into the condensate system or atmosphere, which is a direct energy loss.

The guidebook notes that inverted bucket traps tend to fail open, which is why they are preferred where a failed-closed trap would be dangerous, such as turbine drains; they also vent air slowly and can lose their water seal on sudden pressure drops.

Why not the others: (b) a failed-open trap leaks live steam; waterlogging comes from failed-closed traps. (c) inverted bucket traps typically fail open, and that is the property that suits turbine drains. (d) describes failed-open behaviour, not failed-closed.

Ref: BEE Guidebook Paper 2, Ch. 3 (Steam traps: types, selection and failure).

Chapter 3: Steam System / Consequences of steam trap failures

23. During a steam trap survey, a test valve downstream of each trap is opened. Which discharge pattern indicates normal operation for the trap types listed? [3P2CEM023]

- a) Continuous for inverted bucket and disc traps; intermittent for float traps
- b) Intermittent for inverted bucket and disc traps; continuous for float traps**
- c) Continuous blue steam jet from every trap type under normal load
- d) No discharge at all from any trap that is working correctly

Correct answer: (b) Intermittent for inverted bucket and disc traps; continuous for float traps.

The guidebook describes visual testing with a test valve. Inverted bucket and thermodynamic (disc) traps open and shut in cycles, so a healthy trap shows an intermittent discharge of condensate followed by flash steam. A float and thermostatic

trap modulates its valve and discharges condensate continuously, accompanied by flash steam, under steady load. Thermostatic traps may show either pattern, depending on load.

Flash steam, a lazy white cloud, must be distinguished from live steam, which is a clear, high-velocity jet near the outlet; a continuous live steam jet indicates a trap failed open.

Why not the others: (a) reverses the normal patterns of the two trap families. (c) a continuous live steam jet is the sign of a leaking or failed-open trap. (d) no discharge means a blocked or failed-closed trap, or no load, not normal working.

Combining sight checks with ultrasonic or temperature testing improves survey accuracy.

Ref: BEE Guidebook Paper 2, Ch. 3 (Steam trap monitoring and testing).

Chapter 3: Steam System / Reading trap test valve discharges

24. A dye vat is heated by direct steam injection, but steam bubbles reach the surface and burst noisily, with visible vapour loss. Which correction is in line with the guidebook? [3P2CEM024]

- a) Raise steam pressure so bubbles reach the tank bottom quickly
- b) Use one large upward nozzle so steam mixes near the surface
- c) Use superheated steam so that the bubbles condense quickly
- d) Keep steam pressure very low and use many small downward holes**

Correct answer: (d) Keep steam pressure very low and use many small downward holes.

In direct injection the steam must condense completely within the liquid. The guidebook recommends that injection should be at low pressure, about 0.5 kg/cm^2 ($\approx 0.490 \text{ bar}$) and not above 1 kg/cm^2 ($\approx 0.981 \text{ bar}$), through a perforated sparger pipe with many small holes of 2–5 mm diameter, facing downward. Small bubbles at low velocity have a large surface area and enough time to condense before reaching the surface, and the liquid temperature should be controlled by a thermostatic valve.

When bubbles break the surface, both latent heat and water are lost to the atmosphere, and the heating becomes noisy and inefficient.

Why not the others: (a) higher pressure gives bigger, faster bubbles that escape before condensing. (b) a single large upward jet sends steam straight to the surface. (c) superheat delays condensation, because the steam must first lose its superheat before condensing.

Ref: BEE Guidebook Paper 2, Ch. 3 (Direct injection of steam; steam utilisation).

Chapter 3: Steam System / Design of direct steam injection spargers

25. On the steam side of a jacketed pan, films of air, condensate and scale may form on the metal wall. According to the guidebook, which ranking of these resistances is correct? [3P2CEM025]

- a) Condensate film resists most; air film is 100–150 times steel
- b) Steel wall resists most; air and condensate films are negligible
- c) Air film resists most; condensate film is 100–150 times steel**
- d) Scale resists most; an air film offers no real resistance

Correct answer: (c) Air film resists most; condensate film is 100–150 times steel.

The guidebook lists the barriers to heat transfer between steam and product. A condensate water film offers 100–150 times the resistance of steel and 500–600 times that of copper. An air film is far worse, offering 1,500–3,000 times the resistance of steel and 8,000–16,000 times that of copper; a layer of air only 0.025 mm thick resists as much as a copper wall about 400 mm thick.

Air therefore must be vented from steam spaces through thermostatic air vents, and condensate must be drained promptly, before scale or fouling is even considered.

Why not the others: (a) assigns air's much higher resistance to condensate and understates air. (b) the steel wall is the smallest resistance in the path. (d) scale is a real resistance but is not the largest, and air is the most serious barrier.

Practitioners often blame the boiler or trap size when the real cause is air blanketing.

Ref: BEE Guidebook Paper 2, Ch. 3 (Barriers to heat transfer; air venting).

Chapter 3: Steam System / Relative resistance of films to heat transfer

26. A dairy plans to pass high-pressure condensate into a flash vessel and use the flash in a low-pressure air heater. Which design condition listed below is **WRONG** according to the guidebook? [3P2CEM026]

- a) The flash system should work at the lowest feasible pressure
- b) Condensate from the flash vessel is best drained by a float trap
- c) Vessel height must prevent water carry-over with the flash
- d) Flash supply should exceed demand so the vessel stays pressurised**

Correct answer: (d) Flash supply should exceed demand so the vessel stays pressurised.

The question asks for the wrong statement. The guidebook states that demand for flash steam should exceed its supply, so that all flash produced is used and none is vented through the safety valve; if supply exceeds demand, the surplus is wasted and the recovery case weakens.

The other conditions are correct. The flash system should work at the lowest practical pressure, because the amount of flash released rises as the receiving pressure falls. Condensate leaving the flash vessel bottom is best removed by a float trap, which discharges continuously as it arrives. The vessel needs adequate height and size so that flash steam separates and water is not carried over into the low-pressure line.

Why not the others: (a), (b) and (c) are guidebook design rules, so they are true. Condensate still holds roughly 18–30% of the heat in steam, which is why flash and condensate recovery matter.

Ref: BEE Guidebook Paper 2, Ch. 3 (Flash steam recovery; condensate recovery).

Chapter 3: Steam System / Conditions for flash steam recovery

27. A rerolling mill reheating furnace charges 9 t/h of billets and burns 540 litres/h of furnace oil. The mill yield (finished product/charged billets) is 94%. The specific fuel consumption on finished-product basis is about: [3P2CEM027]

- a) 56.4 litres/t
- b) 60.0 litres/t
- c) 63.8 litres/t**
- d) 67.9 litres/t

Correct answer: (c) 63.8 litres/t.

Given: charge = 9 t/h; oil = 540 litres/h; yield = 94%.

Formula: Finished output = charge × yield; SFC (finished basis) = oil rate/finished output.

Substitution: Finished output = $9 \times 0.94 = 8.46$ t/h; SFC = $540/8.46$.

Result: **SFC ≈ 63.8 litres per tonne of finished product.**

The guidebook's furnace performance example reports specific fuel consumption on the finished-product basis, because scale loss, crop ends and misrolls consume fuel without becoming saleable output; a low yield therefore raises SFC even when the furnace itself is unchanged.

Why not the others: (b) 60.0 litres/t is on charged-billet basis, ignoring yield. (a) 56.4 litres/t multiplies by the yield instead of dividing, a direction error. (d) 67.9 litres/t divides by the yield twice.

Tracking both bases separates furnace inefficiency from mill losses.

Ref: BEE Guidebook Paper 2, Ch. 4 (Performance evaluation of a typical furnace; specific fuel consumption).

Chapter 4: Furnaces / Specific fuel consumption on finished-product basis

28. In a pusher reheating furnace, the water-cooled skid pipes take 45 m³/h of cooling water, heated from 32 °C to 47 °C. The available heat of the furnace at its operating temperature is 45% of the fuel GCV of 10,000 kcal/kg (≈ 41,868 kJ/kg). The furnace oil consumed only to make up this cooling loss is about: [3P2CEM028]

- a) 30.4 kg/h
- b) 67.5 kg/h
- c) 150 kg/h**
- d) 470 kg/h

Correct answer: (c) 150 kg/h.

Given: water = 45,000 kg/h; rise = $47 - 32 = 15\text{ }^{\circ}\text{C}$; available heat = 45%; GCV = 10,000 kcal/kg ($\approx 41,868\text{ kJ/kg}$).

Formula: $Q = m \times C_p \times \Delta t$; oil = $Q/(\text{available heat fraction} \times \text{GCV})$.

Substitution: $Q = 45,000 \times 1 \times 15 = 6,75,000\text{ kcal/h}$; oil = $6,75,000/(0.45 \times 10,000)$.

Result: **Oil $\approx 150\text{ kg/h}$** , because only 45% of the fuel heat is available inside the furnace to replace heat lost to the skids.

The guidebook lists water-cooled skids and their losses among pusher furnace disadvantages, along with skid marks on the stock; insulated skids reduce this loss.

Why not the others: (b) 67.5 kg/h divides by the full GCV and ignores the flue gas loss. (a) 30.4 kg/h multiplies by 0.45 instead of dividing. (d) 470 kg/h uses the outlet temperature of $47\text{ }^{\circ}\text{C}$ as the rise.

Ref: BEE Guidebook Paper 2, Ch. 4 (Pusher type furnaces; heat losses through cooling media).

Chapter 4: Furnaces | Fuel equivalent of skid cooling water loss

29. Select the CORRECT statement on continuous reheating furnaces: [3P2CEM029]

a) Walking hearth furnaces cannot heat the bottom face of the stock

b) Pusher furnaces allow charging without any discharge at the far end

c) Walking beam furnaces cannot heat the bottom face of the stock

d) Pusher furnaces have negligible water-cooling losses from their skid supports

Correct answer: (a) Walking hearth furnaces cannot heat the bottom face of the stock.

In a walking hearth furnace the stock rests on the hearth itself, which moves it forward in steps; because the pieces lie directly on the refractory, the bottom face cannot be heated by the flame, so heating is slower and less uniform. A walking beam furnace lifts the stock on water-cooled beams clear of the hearth, so firing from below can heat all faces, at the cost of beam cooling losses.

The guidebook lists pusher furnace disadvantages: a piece must be discharged every time a piece is charged, skid marks appear on the stock, water-cooled skids cause heat losses, and the stock cannot be heated uniformly from all sides.

Why not the others: (b) contradicts the pusher principle, where pushing a piece in pushes one out. (c) walking beam furnaces are chosen precisely because they heat the bottom face. (d) skid cooling losses are a recognised pusher disadvantage.

Ref: BEE Guidebook Paper 2, Ch. 4 (Continuous furnaces: pusher, walking beam and walking hearth).

Chapter 4: Furnaces | Comparison of continuous reheating furnaces

30. A bogie hearth annealing furnace uses a car made of 4 t of steel structure ($C_p 0.12\text{ kcal/kg }^{\circ}\text{C}$ ($\approx 0.502\text{ kJ/kg}\cdot\text{K}$)) and 6 t of refractory ($C_p 0.22\text{ kcal/kg }^{\circ}\text{C}$ ($\approx 0.921\text{ kJ/kg}\cdot\text{K}$)). The car heats from $40\text{ }^{\circ}\text{C}$ to a mean of $560\text{ }^{\circ}\text{C}$ each cycle and is pulled out to cool. With 2 cycles a day for 300 days, fuel GCV 10,700 kcal/kg ($\approx 44,799\text{ kJ/kg}$) and available heat 40%, the annual oil spent on heating the car is about: [3P2CEM030]

a) 21.0 t

b) 52.5 t

c) 65.6 t

d) 131 t

Correct answer: (d) 131 t.

Given: steel 4,000 kg $\times 0.12$; refractory 6,000 kg $\times 0.22$; $\Delta t = 520\text{ }^{\circ}\text{C}$; 600 cycles/yr; GCV 10,700 kcal/kg ($\approx 44,799\text{ kJ/kg}$); available heat 40%.

Formula: Heat per cycle = $\Sigma(m \times C_p) \times \Delta t$; oil = annual heat/($0.40 \times \text{GCV}$).

Substitution: $(480 + 1,320) \times 520 = 9,36,000\text{ kcal per cycle}$; $\times 600 = 56,16,00,000\text{ kcal/yr}$; oil = $56,16,00,000/(0.40 \times 10,700) = 1,31,215\text{ kg}$.

Result: **Oil $\approx 131\text{ t per year}$** .

The guidebook classes this under material handling losses: cars, trays and fixtures heated with the charge and then cooled; lighter cars with fibre insulation reduce it.

Why not the others: (c) counts one cycle a day. (b) divides by the full GCV, ignoring available heat. (a) multiplies by 0.40 instead of dividing.

Ref: BEE Guidebook Paper 2, Ch. 4 (Furnace heat losses: material handling losses).

Chapter 4: Furnaces | Heat stored in a bogie hearth car

31. The guidebook lists typical operating temperatures such as 650–750 °C for bogie annealing furnaces, 800 °C for a bar furnace for a sheet mill and about 1,200 °C for slab reheating. An operator runs a bogie annealing furnace at 900 °C 'to be on the safe side'. The consequences are: [3P2CEM031]

- a) Lower scale loss, since oxidation slows at higher temperatures
- b) Higher heat loss, more oxidation and decarburisation, refractory stress**
- c) No change in losses, as the stock reaches the same temperature
- d) Lower wall loss, since hot refractory conducts less heat

Correct answer: (b) Higher heat loss, more oxidation and decarburisation, refractory stress.

The guidebook states that heating the stock above its required temperature only increases fuel consumption and heat losses, and also causes metallurgical damage: excess oxidation (scale), decarburisation of the surface and over-stressing of the refractories. Wall, opening and flue gas losses all rise with furnace temperature; radiation through openings rises with the fourth power of absolute temperature. The remedy is temperature control, preferably automatic, so that the furnace is held at the set value for the particular product, instead of relying on operator judgement.

Why not the others: (a) inverts the chemistry, since scaling accelerates with temperature. (c) the furnace and flue gas temperatures are higher even if the stock is later held, so losses increase. (d) refractory conductivity generally rises with temperature, and the larger temperature difference increases wall loss.

Ref: BEE Guidebook Paper 2, Ch. 4 (Furnace temperature: operating temperatures and avoiding overheating).

Chapter 4: Furnaces | Overheating beyond required furnace temperature

32. A 650 m chilled water header at 7 °C gains 45 kcal/h (≈ 0.052 kW) per metre through wet, damaged insulation, against 8 kcal/h (≈ 0.0093 kW) per metre when properly insulated with a vapour barrier. The chiller plant consumes 0.75 kW/TR, runs 6,000 h/yr and power costs ₹ 8/kWh. The annual saving from re-insulation is about: [3P2CEM032]

- a) ₹ 2.86 lakh**
- b) ₹ 3.48 lakh
- c) ₹ 3.82 lakh
- d) ₹ 10.07 lakh

Correct answer: (a) ₹ 2.86 lakh.

Given: length 650 m; gain 45 kcal/h (≈ 0.052 kW) per metre damaged, 8 kcal/h (≈ 0.0093 kW) per metre restored; 1 TR = 3,024 kcal/h (≈ 3.52 kW); 0.75 kW/TR; 6,000 h/yr; ₹ 8/kWh.

Formula: Heat gain avoided = $\Delta q \times L$; TR = heat/3,024; saving = TR $\times$ kW/TR $\times$ h $\times$ tariff.

Substitution: $(45 - 8) \times 650 = 24,050$ kcal/h; TR = $24,050/3,024 = 7.95$; kW = $7.95 \times 0.75 = 5.96$; ₹ = $5.96 \times 6,000 \times 8$.

Result: **Saving $\approx$ ₹ 2.86 lakh per year.**

The guidebook stresses that cold insulation needs low conductivity, water resistance and an effective vapour barrier, since condensed moisture ruins insulation.

Why not the others: (b) uses the full 45 kcal/h (≈ 0.052 kW) per metre, as if the new insulation gained nothing. (c) treats tonnes of refrigeration as kW, omitting 0.75 kW/TR. (d) divides by 860, converting to kW instead of TR.

Ref: BEE Guidebook Paper 2, Ch. 5 (Cold insulation; insulation economics).

Chapter 5: Insulation and Refractories | Value of restoring cold insulation

33. A ladle lining is changed from fired bricks to a monolithic castable. Which pair gives a guidebook advantage of monolithics and a correct precaution? [3P2CEM033]

- a) No weak joints; keep the calcium aluminate binder dry**

- b) More joints for expansion; store castable outdoors in open
- c) Higher heat storage; no drying needed after casting
- d) Needs many skilled masons; lower spalling resistance overall

Correct answer: (a) No weak joints; keep the calcium aluminate binder dry.

The guidebook lists the advantages of monolithic refractories: elimination of joints, which are an inherent weakness of brick linings; faster application; better spalling resistance; volume stability; easy transport and handling; and lower heat storage when installed as thinner linings, often called a heat saver.

Castables are bonded with calcium aluminate cement, which is hydraulic: it reacts with moisture, so bags must be stored dry and used within shelf life, otherwise setting strength is lost. After casting, controlled curing and a proper drying and heat-up schedule are needed to remove water without explosive spalling.

Why not the others: (b) the absence of joints is the advantage, and outdoor storage spoils the cement. (c) monolithics still require careful drying before service. (d) monolithics reduce dependence on skilled masons and offer better, not worse, spalling resistance.

Ref: BEE Guidebook Paper 2, Ch. 5 (Monolithic refractories; castables).

Chapter 5: Insulation and Refractories | Advantages and precautions for monolithic refractories

34. Select the WRONG statement about refractory materials as per the guidebook: [3P2CEM034]

- a) Silica brick resists softening under load near fusion
- b) High alumina refractories lose refractoriness as alumina rises**
- c) Zirconia insulates well and resists wetting by glass
- d) Magnesite-chromite spalls less than chrome-magnesite brick

Correct answer: (b) High alumina refractories lose refractoriness as alumina rises.

The question asks for the wrong statement. In high alumina refractories, defined as having 45–100% Al_2O_3 , refractoriness increases as alumina content rises; pure alumina products can be used up to about 1,850 °C. Refractoriness does not fall with alumina.

The other statements reflect the guidebook. Silica brick, at least 93% SiO_2 , keeps high strength at temperatures close to its fusion point, a property few other refractories share. Zirconia, stabilised with calcium or magnesium oxide, has low thermal conductivity, strength up to about 1,500 °C and is not easily wetted by molten glass. Magnesite-chromite bricks, with magnesia as the major component, show better spalling resistance than chrome-magnesite bricks.

Why not the others: (a), (c) and (d) are true statements. A frequent practitioner error is to assume more alumina means poorer high-temperature properties because of fireclay behaviour.

Ref: BEE Guidebook Paper 2, Ch. 5 (Types of refractories: high alumina, silica, zirconia, magnesite-chromite).

Chapter 5: Insulation and Refractories | Properties of specific refractory types

35. A hot oil line operates at 250 °C and is to be lagged with mineral wool. At what temperature should the supplier's thermal conductivity value be read, and how does the guidebook classify this duty? [3P2CEM035]

- a) At the hot-face temperature; low-temperature class
- b) At ambient air temperature; high-temperature class
- c) At the mean face temperature; medium-temperature class**
- d) At the cold-face temperature; high-temperature class

Correct answer: (c) At the mean face temperature; medium-temperature class.

Thermal conductivity of insulating materials rises with temperature, so the guidebook specifies it at the mean temperature of the insulation layer, the average of the hot-face and cold-face temperatures. Using a value read at ambient understates conductivity and gives an undersized thickness and a surface hotter than designed.

The guidebook classifies insulation by application temperature: low temperature up to 90 °C (refrigeration, cold and hot water), medium temperature 90–325 °C (low-temperature heating and steam-raising equipment, steam lines, flue ducts) and high temperature above 325 °C (boilers, superheated steam and furnaces). A 250 °C hot oil line is therefore a medium-temperature duty.

Why not the others: (a) the hot-face value overstates conductivity, and 250 °C is not a low-temperature duty. (b) the ambient value understates conductivity, and 250 °C is below the high-temperature band. (d) uses the cold face and the wrong class.

Ref: BEE Guidebook Paper 2, Ch. 5 (Insulation types and classification; thermal conductivity).

Chapter 5: Insulation and Refractories | Temperature basis for insulation conductivity

36. An atmospheric fluidised bed combustion boiler is operated with a static bed height of 1.2 m. Using the guidebook rule of thumb for bed pressure drop, the pressure drop across the bed alone is about: [3P2CEM036]

- a) 30.5 mm WC (≈ 299 Pa)
- b) 120 mm WC (≈ 1.18 kPa)
- c) 1,200 mm WC (≈ 11.8 kPa)**
- d) 3,048 mm WC (≈ 29.9 kPa)

Correct answer: (c) 1,200 mm WC (≈ 11.8 kPa).

Given: bed height = 1.2 m = 1,200 mm; guidebook rule: bed pressure drop is about 1 inch of water column per inch of bed depth.

Formula: ΔP (mm WC) $\approx$ bed depth (mm).

Substitution: $\Delta P \approx 1,200 \text{ mm} \times 1 = 1,200 \text{ mm WC}$.

Result: **$\Delta P \approx 1,200 \text{ mm WC} (\approx 11.8 \text{ kPa})$** , to which the distributor drop must be added when sizing the FD fan.

Typical AFBC beds are 0.9–1.5 m deep, so bed resistance is a major part of fan power. The guidebook favours shallow beds with horizontal tube bundles because they reduce pressure drop and fan power.

Why not the others: (a) 30.5 mm WC (≈ 299 Pa) is the drop for one inch of bed only. (b) 120 mm WC (≈ 1.18 kPa) confuses millimetres with centimetres. (d) 3,048 mm WC (≈ 29.9 kPa) multiplies by 2.54, applying an inch-to-centimetre factor that the like-for-like rule does not need.

Ref: BEE Guidebook Paper 2, Ch. 6 (AFBC: bed depth and bed pressure drop).

Chapter 6: FBC Boilers | Bed pressure drop in a bubbling fluidised bed

37. A plant burns 12 t/h of coal. Using the guidebook figures for solids recycled to the bed (AFBC: about 2–4 kg of solids per tonne of fuel; CFBC: about 50–100 kg of solids per kg of fuel), the recirculation rates are: [3P2CEM037]

- a) AFBC 24–48 t/h; CFBC 600–1,200 kg/h
- b) AFBC 600–1,200 kg/h; CFBC 24–48 t/h
- c) AFBC 2–4 t/h; CFBC 50–100 t/h
- d) AFBC 24–48 kg/h; CFBC 600–1,200 t/h**

Correct answer: (d) AFBC 24–48 kg/h; CFBC 600–1,200 t/h.

Given: coal = 12 t/h; AFBC 2–4 kg per t fuel; CFBC 50–100 kg per kg fuel.

Formula: $\text{Recycle} = \text{fuel rate} \times \text{specific recycle ratio}$.

Substitution: AFBC: $12 \times 2 = 24$ and $12 \times 4 = 48$ kg/h. CFBC: $12,000 \times 50 = 600,000$ and $12,000 \times 100 = 1,200,000$ kg/h.

Result: **AFBC $\approx 24\text{--}48$ kg/h; CFBC $\approx 600\text{--}1,200$ t/h**, about 25,000 times larger.

In CFBC, high velocities of 3.7–9 m/s carry solids out of the furnace; hot cyclones return them, giving long residence, a Ca/S ratio of about 1.5 against 3.2 in AFBC, and no in-bed tubes.

Why not the others: (a) interchanges kg and t for both units. (b) swaps the boiler types. (c) reads the CFBC ratio per tonne and omits the multiplier for AFBC.

Ref: BEE Guidebook Paper 2, Ch. 6 (AFBC and CFBC: solids recycle).

Chapter 6: FBC Boilers | Solids recirculation in AFBC and CFBC

38. Which bubbling bed design choice reduces fan power while keeping uniform fluidisation, as per the guidebook? [3P2CEM038]

- a) Shallow bed with horizontal in-bed tubes and nozzle distributors**

- b) Deep bed with vertical in-bed tubes and plain open holes
- c) Deep bed with no in-bed tubes and a solid distributor plate
- d) Shallow bed with ash drained only at intervals of weeks

Correct answer: (a) Shallow bed with horizontal in-bed tubes and nozzle distributors.

The guidebook states that a shallow bed offers lower pressure drop, and therefore lower fan power, than a deep bed, and that a horizontal tube bundle in a shallow bed is preferred. The air distributor, with nozzles or bubble caps, gives uniform air entry across the bed, prevents bed material falling into the plenum and is protected from the hot bed by refractory, a static layer or water-cooled tubes. Bed material of about 1 mm and controlled continuous ash drain keep fluidisation stable.

Uniform fluidisation is essential, because dead zones lead to clinkers and hot spots, while channelling leaves part of the bed unfluidised.

Why not the others: (b) deep beds raise fan power, and plain holes let bed material sift through. (c) a solid plate cannot distribute air at all. (d) infrequent ash drainage lets the bed grow coarse and defluidise.

Ref: BEE Guidebook Paper 2, Ch. 6 (FBC design: distributor, bed depth and in-bed tubes).

Chapter 6: FBC Boilers / Bed design for low fan power and uniform fluidisation

39. A 1,500 kW natural gas engine generator has an electrical heat rate of 2,650 kcal/kWh ($\approx 11,095$ kJ/kWh). Its exhaust and jacket heat recovery delivers useful heat equal to 50% of the fuel input. The overall cogeneration efficiency is about: [3P2CEM039]

- a) 32.5%
- b) 48.7%
- c) 66.2%
- d) 82.5%**

Correct answer: (d) 82.5%.

Given: heat rate = 2,650 kcal/kWh ($\approx 11,095$ kJ/kWh); 1 kWh = 860 kcal ($\approx 3,601$ kJ); useful thermal recovery = 50% of fuel input.

Formula: Electrical efficiency = $860/\text{heat rate}$; overall efficiency = electrical + thermal recovery fraction.

Substitution: $\eta_e = 860/2,650 = 0.3245$; overall = 32.45% + 50%.

Result: **Overall cogeneration efficiency $\approx 82.5\%$** , inside the guidebook band of 76–86% for larger reciprocating engines with heat rates of 2,400–3,275 kcal/kWh ($\approx 13,712$ kJ/kWh).

Why not the others: (a) 32.5% is only the electrical efficiency. (b) 48.7% applies the 50% to the electrical output instead of the fuel input. (c) 66.2% takes 50% of the rejected heat (67.5% of input) rather than 50% of input.

Engines recover heat from both exhaust and jacket water, so the base of the recovery percentage must be checked.

Ref: BEE Guidebook Paper 2, Ch. 7 (Table 7.4: typical cogeneration performance parameters).

Chapter 7: Cogeneration / Overall efficiency of a gas engine cogeneration unit

40. In a refinery, a gas turbine drives a generator and its exhaust raises high-pressure steam in a heat recovery boiler; the steam then drives a steam turbine generator before being extracted for process. As per the guidebook classification of topping cycles, this is a: [3P2CEM040]

- a) Steam-turbine topping system
- b) Gas-turbine topping system
- c) Combined-cycle topping system**
- d) Bottoming cycle system

Correct answer: (c) Combined-cycle topping system.

The guidebook describes four topping cycle types. In the combined-cycle topping system, a gas turbine or diesel engine produces electrical or mechanical power, and its exhaust raises steam in a heat recovery boiler that then drives a secondary steam turbine. In the steam-turbine topping system, fuel is burnt in a boiler to make high-pressure steam that passes through a steam turbine before supplying process steam. In the third type, heat is recovered from engine exhaust

and jacket water. In the gas-turbine topping system, the turbine exhaust heats a process directly or produces process steam without a further turbine.

Because the refinery's steam passes through a second turbine, the arrangement is combined cycle.

Why not the others: (a) has no gas turbine stage. (b) would send the HRSG steam straight to process. (d) bottoming cycles start from high-temperature process heat.

Ref: BEE Guidebook Paper 2, Ch. 7 (Classification of cogeneration systems: topping cycles).

Chapter 7: Cogeneration | Identifying topping cycle configurations

41. Why does the guidebook say gas turbine cogeneration needs premium fuels, and what exhaust temperature makes it suited to high-grade process heat? [3P2CEM041]

a) Dirty fuels raise turbine efficiency; exhaust is about 150–200 °C

b) Clean fuels protect turbine blades; exhaust is about 450–550 °C

c) Premium fuels cool the blades; exhaust is about 900–1,000 °C

d) Any fuel suits gas turbines; exhaust is about 250–300 °C

Correct answer: (b) Clean fuels protect turbine blades; exhaust is about 450–550 °C.

The guidebook notes that gas turbines need clean, premium fuels such as natural gas, naphtha or distillate oil, because combustion products pass directly through the turbine blades; ash, sodium, vanadium and sulphur cause deposits, erosion and hot corrosion. Turbine inlet gas is about 900–1,000 °C, while the exhaust leaves at about 450–550 °C, high enough to raise high-pressure steam in a heat recovery boiler or to fire supplementary burners, since it still holds plenty of oxygen.

Standalone simple-cycle efficiency is about 35–40%, and the compressor absorbs a large share, 50–60%, of turbine output.

Why not the others: (a) dirty fuels damage blades and do not raise efficiency, and 150–200 °C is the stack after recovery.

(c) 900–1,000 °C is turbine inlet, not exhaust. (d) fuel quality matters greatly, and 250–300 °C understates exhaust temperature.

Ref: BEE Guidebook Paper 2, Ch. 7 (Gas turbine cogeneration; prime movers).

Chapter 7: Cogeneration | Fuel and exhaust characteristics of gas turbines

42. Select the WRONG statement about the relative merits of cogeneration systems as per the guidebook:

[3P2CEM042]

a) A steam turbine with boiler can use low-quality fuels

b) A gas turbine with WHR boiler has low civil construction cost

c) A combined cycle offers quick start-up and high flexibility

d) A diesel engine with WHR gives the highest overall efficiency

Correct answer: (d) A diesel engine with WHR gives the highest overall efficiency.

The question asks for the wrong statement. The guidebook's comparison table shows that back pressure steam turbine systems with a boiler give the highest overall cogeneration efficiency, up to about 84–92%, because all exhaust steam heat is used in the process. Diesel engines give the highest electrical efficiency, but their overall efficiency is lower and part of their heat, from jacket water and lube oil, is of low grade.

The other statements match the table. Steam turbine plants can burn any fuel, including coal, biomass and low-quality fuels, in the boiler. Gas turbine plants with waste heat boilers are compact, with low civil construction cost and short gestation. Combined cycles offer quick start-up and high flexibility in operation.

Why not the others: (a), (b) and (c) are true. Confusing electrical efficiency with overall efficiency is a common error.

Ref: BEE Guidebook Paper 2, Ch. 7 (Relative merits of cogeneration systems; Table 7.2).

Chapter 7: Cogeneration | Relative merits of cogeneration options

43. A paper mill that cannot tolerate steam outages and has abundant bark and black liquor is evaluating cogeneration. Which guidebook considerations apply? [3P2CEM043]

a) Single large gas turbine, as bark suits gas turbines best

b) Modular multi-unit plant, and a steam turbine on the biomass

- c) Single diesel engine, since it needs no standby supply
- d) A grid-independent plant with no process steam extraction at all

Correct answer: (b) Modular multi-unit plant, and a steam turbine on the biomass.

The guidebook lists technical parameters for cogeneration selection. Reliability and availability: where process loss from an outage is high, the plant should be built from modular units or have standby capacity, so that one unit can be maintained while others run. Local fuel availability: the system should exploit the least-cost fuel available on site; for industries such as paper, rice or sugar, biomass and process wastes are best burnt in a boiler feeding a steam turbine, as the guidebook illustrates with a rice mill using husk.

The heat-to-power ratio of the mill, with its large steam demand, also favours a back pressure or extraction turbine.

Why not the others: (a) gas turbines need clean premium fuels, so bark cannot be fired directly, and a single unit weakens reliability. (c) a single engine offers no redundancy. (d) the mill's main need is process steam.

Ref: BEE Guidebook Paper 2, Ch. 7 (Technical options and parameters for cogeneration).

Chapter 7: Cogeneration | Technical parameters in cogeneration planning

44. Select the CORRECT statement on steam turbine performance assessment in cogeneration plants: [3P2CEM044]

- a) Heat rate rises as less steam is sent to the condenser
- b) A pressure survey is needed only once, at commissioning
- c) Cogeneration efficiency is not simply the inverse of heat rate**
- d) Fully condensing turbines are preferred for industrial cogeneration

Correct answer: (c) Cogeneration efficiency is not simply the inverse of heat rate.

The guidebook cautions that heat rate, the heat input per kWh generated, describes only electrical generation. In a cogeneration plant, much of the heat input leaves as useful process steam, so overall efficiency counts both electricity and recovered heat; it is therefore not simply 860 divided by heat rate.

Heat rate improves, falling, when less steam goes to the condenser, because condenser heat is wasted, while back pressure exhaust is used by the process. Back pressure turbines thus give the highest overall efficiencies, up to about 85%, and are preferred in industry; fully condensing machines reject most heat to the cooling tower.

Regular pressure and temperature surveys across turbine stages help track deterioration from deposits and blade wear.

Why not the others: (a) heat rate falls, not rises, as condenser flow drops. (b) surveys are periodic, not one-time. (d) condensing turbines are least suited to industrial cogeneration.

Ref: BEE Guidebook Paper 2, Ch. 7 (Performance of steam turbines; heat rate).

Chapter 7: Cogeneration | Interpreting heat rate and turbine performance

45. Flue gas leaving a forging furnace at 870 °C is cooled to 430 °C in a recuperator that preheats the combustion air. The furnace burns 320 litres/h of furnace oil. Using the guidebook thumb rule for recuperators, the oil saving is about: [3P2CEM045]

- a) 32.0 litres/h
- b) 64.0 litres/h**
- c) 70.4 litres/h
- d) 126.5 litres/h

Correct answer: (b) 64.0 litres/h.

Given: flue gas 870 → 430 °C, a reduction of 440 °C; oil = 320 litres/h; guidebook rule: about 1% fuel saving for every 22 °C reduction in flue gas temperature.

Formula: % saving = $\Delta T/22$; oil saved = % saving × oil rate.

Substitution: $440/22 = 20\%$; 0.20×320 .

Result: **Oil saving ≈ 64.0 litres/h.**

The guidebook illustrates this rule with a recuperator example where reducing flue gas temperature from 900 °C to 180 °C was taken as a 33% fuel saving.

Why not the others: (a) uses 1% per 44 °C, halving the saving. (c) uses 1% per 20 °C, a rule quoted for boiler air preheaters, not this recuperator rule. (d) divides the flue gas inlet temperature of 870 °C by 22, instead of the temperature reduction. Thumb rules are only screening tools; a detailed heat balance should confirm the saving before investment.

Ref: BEE Guidebook Paper 2, Ch. 8 (Recuperators; benefits of waste heat recovery).

Chapter 8: Waste Heat Recovery | Fuel saving from a furnace recuperator by thumb rule

46. Metallic radiation recuperators, made of two concentric metal tubes with flue gas in the inner tube, are usually built for parallel flow although counter flow transfers heat better. The reason is: [3P2CEM046]

a) Parallel flow also cools the exhaust duct, extending its life

b) Parallel flow lets the cold air absorb most gas radiation

c) Parallel flow gives the larger LMTD and the smaller area

d) Parallel flow avoids any need for a stack or ID fan

Correct answer: (a) Parallel flow also cools the exhaust duct, extending its life.

The guidebook describes the simplest recuperator as two concentric metal tubes, with hot flue gas in the inner tube and combustion air in the annulus. Heat passes mainly by radiation from the gas to the inner tube wall. Although counter flow is thermally more efficient, the parallel flow arrangement is often chosen because the coldest air meets the hottest gas, cooling the inner tube where gas is hottest; this protects the metal and also cools the ducting carrying the gases, extending its service life.

Air is practically transparent to infrared radiation, so the air does not absorb gas radiation directly; it is heated by convection from the tube wall.

Why not the others: (b) contradicts the transparency of air to infrared. (c) counter flow, not parallel flow, gives the larger mean temperature difference. (d) flue gas still needs a stack or fan downstream.

Ref: BEE Guidebook Paper 2, Ch. 8 (Recuperators: metallic radiation recuperator).

Chapter 8: Waste Heat Recovery | Flow arrangement in metallic radiation recuperators

47. An evaporator vents 900 kg/h of vapour at 100 °C to the atmosphere. If this vapour is condensed and the condensate cooled to 60 °C in an exchanger preheating feed liquor, the heat recoverable (latent heat 540 kcal/kg ($\approx 2,261$ kJ/kg), Cp of water 1 kcal/kg °C (≈ 4.19 kJ/kg·K)) is about: [3P2CEM047]

a) 41.9 kW

b) 565 kW

c) 607 kW

d) 670 kW

Correct answer: (c) 607 kW.

Given: vapour = 900 kg/h at 100 °C; latent heat = 540 kcal/kg ($\approx 2,261$ kJ/kg); condensate cooled 100 $\rightarrow$ 60 °C; 860 kcal/h = 1 kW.

Formula: $Q = m \times [L + C_p \times (100 - 60)]$.

Substitution: $Q = 900 \times (540 + 40) = 5,22,000$ kcal/h; kW = 5,22,000/860.

Result: **Recoverable heat ≈ 607 kW.**

The guidebook's waste heat quality table grades vapour streams like flue gases, adding that when they are condensed the latent heat is also recoverable; here latent heat is over 90% of the total.

Why not the others: (b) 565 kW counts only latent heat and ignores condensate sub-cooling. (d) 670 kW cools the condensate to 0 °C, below any practical feed temperature. (a) 41.9 kW counts only the sensible cooling of condensate, missing over 90% of the heat.

Ref: BEE Guidebook Paper 2, Ch. 8 (Classification and quality of waste heat; Table 8.1).

Chapter 8: Waste Heat Recovery | Recovering latent heat from vented vapour

48. A process stream is cooled from 62 °C to 22 °C in a coil immersed in a well-stirred tank kept at a constant 10 °C by a chiller. The log mean temperature difference for the coil is about: [3P2CEM048]

- a) 12.0 °C
- b) 27.3 °C**
- c) 32.0 °C
- d) 40.0 °C

Correct answer: (b) 27.3 °C.

Given: hot stream 62 → 22 °C; bath at a constant 10 °C.

Formula: With one fluid at constant temperature, $LMTD = (\Delta T_1 - \Delta T_2) / \ln(\Delta T_1 / \Delta T_2)$, the same for any flow arrangement.

Substitution: $\Delta T_1 = 62 - 10 = 52$ °C; $\Delta T_2 = 22 - 10 = 12$ °C; $LMTD = (52 - 12) / \ln(52/12) = 40/1.466$.

Result: **LMTD ≈ 27.3 °C.**

Because the bath temperature does not change, the parallel and counter flow distinction disappears; the guidebook's worked example of a coil in a constant-temperature bath uses this approach to size the coil length.

Why not the others: (c) 32.0 °C is the arithmetic mean difference, which overstates driving force and undersizes area by about 15%. (a) 12.0 °C is only the cold-end difference. (d) 40.0 °C is the difference of the end differences, without the logarithm.

Ref: BEE Guidebook Paper 2, Ch. 9 (Heat exchanger performance: LMTD).

Chapter 9: Heat Exchangers | LMTD with one fluid at constant temperature

49. In an evaporator calandria, steam condenses at 133 °C while the liquor boils at a constant 100 °C. For a duty of 660 kW and an overall heat transfer coefficient of 1,650 W/m²K, the heat transfer area required is about: [3P2CEM049]

- a) 3.01 m²
- b) 4.00 m²
- c) 12.1 m²**
- d) 24.2 m²

Correct answer: (c) 12.1 m².

Given: Q = 6,60,000 W; U = 1,650 W/m²K; condensing steam 133 °C; boiling liquor 100 °C.

Formula: When both fluids remain at constant temperature, ΔT is the same along the surface, so $LMTD = \Delta T$; $A = Q / (U \times \Delta T)$.

Substitution: $\Delta T = 133 - 100 = 33$ °C; $A = 6,60,000 / (1,650 \times 33) = 6,60,000 / 54,450$.

Result: **A ≈ 12.1 m².**

The guidebook notes this special case of constant temperature difference, as in condensers and evaporators, where the standard log formula would give 0/0 and the temperature difference is simply the terminal temperature difference.

Why not the others: (a) uses the steam temperature of 133 °C as the driving force. (b) uses the liquor temperature of 100 °C as the driving force. (d) halves the difference to 16.5 °C, treating it like an arithmetic mean of 33 and zero.

Ref: BEE Guidebook Paper 2, Ch. 9 (Temperature profiles; special case of constant temperature difference).

Chapter 9: Heat Exchangers | Area of an evaporator with both fluids changing phase

50. Counter flow normally gives the larger mean temperature difference and the smallest area. In which situation does the guidebook indicate that a parallel flow configuration is frequently used? [3P2CEM050]

- a) When a temperature cross between the two outlets is needed
- b) When maximum heat recovery from the hot fluid is the aim
- c) When space limits force the two fluids to flow at right angles
- d) When both fluids should leave at nearly the same temperature**

Correct answer: (d) When both fluids should leave at nearly the same temperature.

In parallel flow, both fluids enter at the same end: the hottest hot fluid meets the coldest cold fluid at one terminal, and the two streams move together so that, at the other terminal, the hot outlet and cold outlet approach each other. The

guidebook states that this configuration is frequently used where it is desired that the two fluids leave at close to the same temperature. It also states that a temperature cross cannot be achieved in parallel flow.

Counter flow, where fluids enter at opposite ends, is the most efficient arrangement, allowing maximum heat recovery and the smallest area.

Why not the others: (a) a temperature cross is impossible in parallel flow; it needs counter flow or multipass designs. (b) maximum recovery is the counter flow advantage. (c) flow at right angles describes cross flow, used where counter flow is impractical.

Ref: BEE Guidebook Paper 2, Ch. 9 (Heat exchanger types by flow design).

Chapter 9: Heat Exchangers | Applications favouring parallel flow

..... End of Section I

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

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

S-1 [3P2CEM051 | Chapter 1: Fuels and Combustion | Blending slack and sized coal to limit fines on a chain grate]

A textile mill's chain-grate boiler can buy two coals. Lot A is a cheap washery slack coal with many fines; Lot B is a sized coal. The stoker manufacturer limits fines (below 6 mm) in the feed to 25%. The boiler needs a fuel heat input of 12 million kcal/h and runs 7,200 h/yr.

Parameter	Lot A (slack)	Lot B (sized)
Fines below 6 mm	40%	10%
GCV (as received)	4,300 kcal/kg ($\approx 18,003$ kJ/kg)	4,100 kcal/kg ($\approx 17,166$ kJ/kg)
Ash	33%	36%
Landed price	₹ 4,600 per tonne	₹ 5,400 per tonne

- a) Calculate the maximum share of Lot A in a blend that meets the fines limit. (1.5 Marks)
- b) Calculate the GCV and ash of that blend. (1 Mark)
- c) Compare the cost of heat (₹ per million kcal) for Lot B alone and the blend, and find the annual saving from blending. (1.5 Marks)
- d) Why does the guidebook limit fines on stokers, and what preparation helps when fines are present? (1 Mark)

S-1 Ans (model answer with marking scheme)

Given: Fines A = 40%, B = 10%; limit 25%. GCV A = 4,300 kcal/kg ($\approx 18,003$ kJ/kg), B = 4,100 kcal/kg ($\approx 17,166$ kJ/kg); ash A = 33%, B = 36%; price A = ₹ 4,600/t, B = ₹ 5,400/t; heat input = 12 million kcal/h; 7,200 h/yr.

a) Maximum share of Lot A (1.5 marks)

Formula: Fines in blend = $x \times 40 + (1 - x) \times 10 \leq 25$, where x = mass fraction of Lot A

Substitution: $40x + 10 - 10x \leq 25 \rightarrow 30x \leq 15$ [1 mark]

Result: $x = 0.50$, i.e. a 50:50 blend by mass [0.5 mark]

b) Blend properties (1 mark)

Formula: Property of blend = Σ (mass fraction $\times$ property)

Substitution: GCV = $0.5 \times 4,300 + 0.5 \times 4,100$; ash = $0.5 \times 33 + 0.5 \times 36$

Result: **Blend GCV = 4,200 kcal/kg ($\approx 17,585$ kJ/kg); ash = 34.5%** [1 mark]

c) Cost of heat and annual saving (1.5 marks)

Formula: Cost of heat (₹ per million kcal) = price per tonne/(GCV $\times 1,000/10^6$); annual saving = (difference) $\times$ million kcal per year

Substitution: Lot B: $5,400/4.1 = ₹ 1,317.1$; blend price = $0.5 \times 4,600 + 0.5 \times 5,400 = ₹ 5,000/t$, cost = $5,000/4.2 = ₹ 1,190.5$; (Lot A alone would be $4,600/4.3 = ₹ 1,069.8$ but cannot be fired) [0.5 mark]; annual heat = $12 \times 7,200 = 86,400$ million kcal; saving = $(1,317.1 - 1,190.5) \times 86,400$

Result: **Cost of heat: Lot B ₹ 1,317 per million kcal, blend ₹ 1,190 per million kcal; annual saving $\approx ₹ 1,09,37,979 \approx ₹ 1.09$ crore** [1 mark]

d) Fines limit and preparation (1 mark)

Excess fines on a chain grate or stoker choke the air passages through the bed, cause uneven burning and channelling, increase unburnt carbon in ash and fly ash carry-over; segregation of fines also gives patchy firing.

The guidebook recommends sizing (screening before crushing), blending to keep fines in limits, and conditioning fines with water, which agglomerates them and improves bed porosity. [1 mark]

Marking note: accept $\pm 1\%$ rounding; award method marks for the mixing rule even if the arithmetic slips; ash must be reported (higher ash than Lot A is an acceptable trade-off).

S-2 [3P2CEM052 | Chapter 2: Boilers | Techno-economics of replacing an inefficient coal boiler]

An 8 TPH coal-fired boiler installed 28 years ago now operates at 66% efficiency (GCV basis). A new boiler of the same capacity is offered with a guaranteed efficiency of 80% on the same coal. Data:

Parameter	Value
Steam generation	8,000 kg/h
Enthalpy gain of steam (steam enthalpy – feed water enthalpy)	580 kcal/kg ($\approx 2,428$ kJ/kg)
GCV of coal	4,000 kcal/kg ($\approx 16,747$ kJ/kg)
Coal price	₹ 6,200 per tonne
Operating hours	7,200 h/yr
Installed cost of new boiler	₹ 1,60,00,000

- Calculate the coal consumption of the existing and the new boiler. (2 Marks)
- Calculate the annual coal and cost saving and the simple payback period. (2 Marks)
- List the conditions under which the guidebook suggests boiler replacement should be considered. (1 Mark)

S-2 Ans (model answer with marking scheme)

Given: Steam = 8,000 kg/h; $\Delta h = 580$ kcal/kg ($\approx 2,428$ kJ/kg); $\eta_{old} = 0.66$; $\eta_{new} = 0.80$; GCV = 4,000 kcal/kg ($\approx 16,747$ kJ/kg); coal = ₹ 6,200/t; 7,200 h/yr; investment = ₹ 1,60,00,000.

a) Coal consumption (2 marks)

Formula: Coal = steam $\times \Delta h / (\eta \times \text{GCV})$

Substitution: Heat to steam = $8,000 \times 580 = 46,40,000$ kcal/h; old = $46,40,000 / (0.66 \times 4,000) = 46,40,000 / 2,640$ [1 mark]; new = $46,40,000 / (0.80 \times 4,000) = 46,40,000 / 3,200$

Result: **Existing boiler $\approx 1,758$ kg/h; new boiler = 1,450 kg/h** [1 mark]

b) Savings and payback (2 marks)

Formula: Saving = (old – new) $\times$ hours; ₹ saving = tonnes $\times$ price; payback = investment/annual saving

Substitution: $1,757.6 - 1,450 = 307.6$ kg/h (17.5% of present coal); $\times 7,200 = 22,14,545$ kg = 2,214.5 t/yr [1 mark]; ₹ = $2,214.5 \times 6,200 = ₹ 1,37,30,182$; payback = $1,60,00,000 / 1,37,30,182$

Result: **Coal saving $\approx 2,215$ t/yr $\approx ₹ 1.37$ crore/yr; simple payback ≈ 1.17 years (about 14 months)** [1 mark]

c) Guidebook conditions for replacement (1 mark)

Replacement is worth evaluating when the existing boiler is old and inefficient, cannot fire a cheaper substitute fuel, is over- or under-sized for present demand, or was not designed for ideal loading conditions; since a boiler lasts more than 25 years, the decision needs a careful study of fuel, steam demand projections, capital, and the option of improving the existing unit first. [1 mark]

Marking note: $\pm 1\%$ tolerance; payback may be written as 1.2 years; deduct $\frac{1}{2}$ mark if the 66% figure is applied to the new boiler's coal.

S-3 [3P2CEM053 | Chapter 3: Steam System | Steam air heater battery operating at part load]

A spray dryer's air heater battery heats 18,000 m³/h of air (density 1.15 kg/m³, Cp 0.24 kcal/kg °C (≈ 1.00 kJ/kg·K)) from 32 °C to 150 °C using saturated steam at 7 kg/cm²(g) (latent heat 490 kcal/kg ($\approx 2,052$ kJ/kg)); condensate

leaves at saturation. For 2,500 h/yr the dryer runs at half throughput, yet the fan and air heater continue at full flow because there is no control. Steam costs ₹ 2,400 per tonne.

- Calculate the heat duty and steam consumption at full flow. (2 Marks)
- If airflow and steam are reduced in proportion to throughput by a fan VFD and temperature-controlled steam valve, calculate the steam and cost saved per year. (2 Marks)
- State the guidebook observation on air heater batteries and two further steam-saving measures for this dryer. (1 Mark)

S-3 Ans (model answer with marking scheme)

Given: Air = $18,000 \text{ m}^3/\text{h} \times 1.15 \text{ kg/m}^3$; $C_p = 0.24 \text{ kcal/kg } ^\circ\text{C}$ ($\approx 1.00 \text{ kJ/kg}\cdot\text{K}$); $32 \rightarrow 150 ^\circ\text{C}$; latent heat 490 kcal/kg ($\approx 2,052 \text{ kJ/kg}$); part-load period 2,500 h/yr at 50%; steam ₹ 2,400/t.

a) Duty and steam at full flow (2 marks)

Formula: $Q = m \times C_p \times \Delta t$; steam = $Q/\text{latent heat}$

Substitution: $m = 18,000 \times 1.15 = 20,700 \text{ kg/h}$; $Q = 20,700 \times 0.24 \times (150 - 32) = 20,700 \times 0.24 \times 118$ [1 mark]; steam = $5,86,224/490$

Result: **$Q = 5,86,224 \text{ kcal/h}$ ($\approx 682 \text{ kW}$); steam $\approx 1,196 \text{ kg/h}$** [1 mark]

b) Saving with flow and temperature control (2 marks)

Formula: Steam saved = (steam at full flow – steam at half flow) $\times$ part-load hours

Substitution: Saved = $(1,196.4 - 598.2) \times 2,500 = 598.2 \times 2,500 = 14,95,469 \text{ kg}$ [1 mark]; ₹ = $1,495.5 \times 2,400$

Result: **Steam saved $\approx 1,495 \text{ t/yr}$ $\approx$ ₹ 35.9 lakh/yr** [1 mark]

c) Observation and further measures (1 mark)

The guidebook notes that air heater batteries without control consume the same steam whether the plant is fully or partly loaded, so load-linked control is the first step. Further measures: preheat inlet air with dryer exhaust heat; return the condensate (about $7 \text{ kg/cm}^2(\text{g})$ ($\approx 6.86 \text{ bar(g)}$) saturation) via a flash vessel; keep coils free of air and condensate with correct traps and air vents; insulate ducts. [1 mark: any two]

Marking note: $\pm 1\%$ tolerance; accept steam saving of 1,490–1,500 t/yr.

S-4 [3P2CEM054 | Chapter 4: Furnaces | Preheating the charge with furnace flue gas]

A forging furnace heats 3 t/h of steel billets to $1,250 ^\circ\text{C}$. A charge preheating chamber is proposed in which flue gas ($2,800 \text{ kg/h}$ at $850 ^\circ\text{C}$ leaving the furnace, $C_p 0.27 \text{ kcal/kg } ^\circ\text{C}$ ($\approx 1.13 \text{ kJ/kg}\cdot\text{K}$)) preheats the billets from $30 ^\circ\text{C}$ to $330 ^\circ\text{C}$ before they enter the furnace. Mean C_p of steel in this range = $0.12 \text{ kcal/kg } ^\circ\text{C}$ ($\approx 0.502 \text{ kJ/kg}\cdot\text{K}$). The available heat in the furnace at its operating temperature is 40% of the fuel GCV of $10,000 \text{ kcal/kg}$ ($\approx 41,868 \text{ kJ/kg}$). Oil costs ₹ 55 per kg; the furnace works 6,000 h/yr; the chamber costs ₹ 60,00,000.

- Calculate the heat supplied to the billets in the preheating chamber and the flue gas exit temperature (neglect chamber losses). (2 Marks)
- Calculate the oil saving per hour and per year, and the annual saving in ₹. (2 Marks)
- Calculate the simple payback and state one precaution. (1 Mark)

S-4 Ans (model answer with marking scheme)

Given: Stock = $3,000 \text{ kg/h}$; $C_p = 0.12 \text{ kcal/kg } ^\circ\text{C}$ ($\approx 0.502 \text{ kJ/kg}\cdot\text{K}$); $30 \rightarrow 330 ^\circ\text{C}$; flue gas $2,800 \text{ kg/h}$, $C_p 0.27 \text{ kcal/kg } ^\circ\text{C}$ ($\approx 1.13 \text{ kJ/kg}\cdot\text{K}$), $850 ^\circ\text{C}$; available heat 40%; GCV $10,000 \text{ kcal/kg}$ ($\approx 41,868 \text{ kJ/kg}$); ₹ 55/kg; 6,000 h/yr; cost ₹ 60,00,000.

a) Heat to stock and gas exit temperature (2 marks)

Formula: $Q = m_s \times C_{p_s} \times \Delta t_s$; gas drop = $Q/(m_g \times C_{p_g})$

Substitution: $Q = 3,000 \times 0.12 \times 300 = 1,08,000 \text{ kcal/h}$ [1 mark]; gas drop $= 1,08,000 / (2,800 \times 0.27) = 142.9 \text{ }^\circ\text{C}$; exit $= 850 - 142.9$

Result: **Heat to stock = 1,08,000 kcal/h ($\approx 126 \text{ kW}$); flue gas leaves the chamber at about $707 \text{ }^\circ\text{C}$** [1 mark]

b) Oil saving (2 marks)

Formula: Oil saved $= Q / (\text{available heat} \times \text{GCV})$, because the furnace would otherwise have to supply this heat at only 40% availability

Substitution: $1,08,000 / (0.40 \times 10,000) = 27 \text{ kg/h}$ [1 mark]; per year $27 \times 6,000 = 1,62,000 \text{ kg}$; ₹ $= 1,62,000 \times 55$

Result: **Oil saving = 27 kg/h = 162 t/yr $\approx$ ₹ 89.1 lakh/yr** [1 mark]

c) Payback and precaution (1 mark)

Result: **Payback = $60,00,000 / 89,10,000 \approx 0.67 \text{ year}$ (about 8 months)** [0.5 mark]

Precaution: keep preheat below the temperature at which scaling starts on the stock surface and ensure uniform gas contact; the chamber must not choke the flue draft (watch furnace pressure). [0.5 mark]

Marking note: $\pm 1\%$ tolerance; award full marks in (b) only if available heat is used (dividing by the full GCV gives 10.8 kg/h and earns method mark only).

S-5 [3P2CEM055 | Chapter 5: Insulation and Refractories | Insulation thickness for a target surface temperature]

A horizontal steam line of 168 mm outside diameter ($r_1 = 84 \text{ mm}$) carries steam at $180 \text{ }^\circ\text{C}$ in a plant where ambient air is $30 \text{ }^\circ\text{C}$. The surface temperature of the aluminium-clad insulation must not exceed $55 \text{ }^\circ\text{C}$. Mineral wool with thermal conductivity $0.048 \text{ W/m}\cdot\text{K}$ at the mean temperature is to be used. For the cladding, take the surface coefficient $h = (A + 0.005 (T_s - T_a)) \times 10 \text{ W/m}^2\text{K}$ with $A = 0.10$, as per the guidebook method.

- Calculate the surface heat transfer coefficient and the heat flux at the outer surface. (1.5 Marks)
- Calculate the thermal resistance of insulation required and the equivalent thickness. (1.5 Marks)
- Determine the actual insulation thickness (by trial), and select a standard thickness from 65, 75 or 80 mm. (1.5 Marks)
- Estimate the heat loss per metre of pipe with the calculated thickness. (0.5 Mark)

S-5 Ans (model answer with marking scheme)

Given: $r_1 = 0.084 \text{ m}$; $T_h = 180 \text{ }^\circ\text{C}$; $T_a = 30 \text{ }^\circ\text{C}$; $T_s = 55 \text{ }^\circ\text{C}$; $k = 0.048 \text{ W/m}\cdot\text{K}$; $A = 0.10$.

a) Surface coefficient and heat flux (1.5 marks)

Formula: $h = (A + 0.005 (T_s - T_a)) \times 10$; $R_s = 1/h$; $q = (T_s - T_a)/R_s = h (T_s - T_a)$

Substitution: $h = (0.10 + 0.005 \times 25) \times 10 = 2.25 \text{ W/m}^2\text{K}$ [0.5 mark]; $R_s = 1/2.25 = 0.444 \text{ m}^2\text{K/W}$; $q = 25/0.444$

Result: **$h = 2.25 \text{ W/m}^2\text{K}$; heat flux $q = 56.25 \text{ W/m}^2$** [1 mark]

b) Resistance and equivalent thickness (1.5 marks)

Formula: The same flux passes through the insulation: $q = (T_h - T_s)/R_I$, so $R_I = (T_h - T_s)/q$; equivalent thickness $E = R_I \times k$

Substitution: $R_I = (180 - 55)/56.25 = 2.222 \text{ m}^2\text{K/W}$ [0.5 mark]; $E = 2.222 \times 0.048 = 0.1067 \text{ m}$

Result: **$R_I \approx 2.22 \text{ m}^2\text{K/W}$; equivalent thickness $E \approx 106.7 \text{ mm}$** [1 mark]

c) Actual thickness by trial (1.5 marks)

Formula: For a pipe, $E = r_2 \ln(r_2/r_1)$, where $r_2 = r_1 + t$

Substitution: $t = 65 \text{ mm}$: $149 \times \ln(149/84) = 85.4 \text{ mm}$ (< 106.7 , insufficient); $t = 75 \text{ mm}$: $159 \times \ln(159/84) = 101.5 \text{ mm}$ (still short); solving gives $r_2 \approx 162.2 \text{ mm}$ [1 mark]; $t = 162.2 - 84$

Result: **Required thickness $\approx 78 \text{ mm}$; select 80 mm standard thickness ($E = 164 \ln(164/84) \approx 109.7 \text{ mm} \geq 106.7 \text{ mm}$)** [0.5 mark]

d) Heat loss per metre (0.5 mark)

Formula: $Q/L = q \times 2\pi r_2$

Substitution: $Q/L = 56.25 \times 2\pi \times 0.1622$

Result: **Heat loss $\approx$ 57.3 W per metre (with 78 mm); slightly less with the 80 mm selected** [0.5 mark]

Marking note: flat-wall thickness (106.7 mm) instead of the cylindrical equivalent-thickness method earns at most 2 marks; ± 2 mm accepted on r_2 .

S-6 [3P2CEM056 | Chapter 6: FBC Boilers | Ash split, ESP performance and ash handling in an AFBC boiler]

An AFBC boiler fires 11 t/h of a high-ash coal containing 42% ash. Bottom (bed) ash is 35% of the total ash and fly ash 65%. About 20% of the fly ash is collected in the convection-bank and economiser hoppers; the rest reaches the electrostatic precipitator (ESP), which has a collection efficiency of 99.2%. Flue gas flow at the ESP is 1,10,000 Nm³/h, and the emission limit is 100 mg/Nm³. The plant runs 24 h/day.

- a) Calculate total, bottom and fly ash generation in t/h and t/day. (1.5 Marks)
- b) Calculate the dust emitted from the stack (kg/h and mg/Nm³) and the ESP efficiency needed to meet the limit. (2 Marks)
- c) How many 20 t trucks are needed per day for ash disposal? (0.5 Mark)
- d) Why is the fly ash fraction high in FBC boilers, and how can fly ash recycling help? (1 Mark)

S-6 Ans (model answer with marking scheme)

Given: Coal = 11 t/h; ash = 42%; bottom ash 35%, fly ash 65%; 20% of fly ash dropped in hoppers; ESP η = 99.2%; gas = 1,10,000 Nm³/h; limit 100 mg/Nm³; 24 h/day.

a) Ash generation (1.5 marks)

Formula: Total ash = coal $\times$ ash fraction; bottom/fly = total $\times$ share

Substitution: Total = $11 \times 0.42 = 4.62$ t/h; bottom = $4.62 \times 0.35 = 1.617$ t/h; fly = $4.62 \times 0.65 = 3.003$ t/h [1 mark]; per day $\times 24$

Result: **Total 4.62 t/h (110.9 t/day); bottom ash 1.617 t/h (38.8 t/day); fly ash 3.003 t/h (72.1 t/day)** [0.5 mark]

b) Stack emission and required ESP efficiency (2 marks)

Formula: Dust to ESP = fly ash $\times (1 - 0.20)$; emission = dust to ESP $\times (1 - \eta)$; concentration = emission/gas flow; η required = $1 - \text{allowed emission/dust to ESP}$

Substitution: Dust to ESP = $3.003 \times 0.80 = 2.4024$ t/h = 2,402.4 kg/h; emission = $2,402.4 \times 0.008 = 19.22$ kg/h [1 mark]; concentration = $19.22 \times 10^6 \text{ mg}/1,10,000 \text{ Nm}^3 = 174.7 \text{ mg/Nm}^3$; allowed = $100 \times 1,10,000/10^6 = 11.0 \text{ kg/h}$; $\eta = 1 - 11.0/2,402.4$

Result: **Emission $\approx$ 19.2 kg/h $\approx$ 175 mg/Nm³ (exceeds 100 mg/Nm³); ESP efficiency needed $\approx$ 99.54%** [1 mark]

c) Trucks (0.5 mark)

Result: **110.9/20 = 5.5, so 6 trucks per day** [0.5 mark]

d) Reason and remedy (1 mark)

In FBC, coal is fed crushed (1–10 mm) and the high fluidising velocity elutriates fine ash and char from the bed, so fly ash is typically 60–70% of total ash and contains some unburnt carbon. Recycling fly ash from the hoppers or cyclone back to the bed increases residence time, burns out the carbon, improves combustion efficiency and limestone utilisation, and reduces the dust load on the ESP. [1 mark]

Marking note: $\pm 1\%$ tolerance; accept 99.5–99.6% for required efficiency.

S-7 [3P2CEM057 | Chapter 7: Cogeneration | Operating schemes for a back pressure cogeneration plant]

A chemical plant is installing a back pressure steam turbine (actual steam rate 8 kg/kWh, exhaust at 4 kg/cm²(g) to process). Process steam demand varies from 24 TPH (night) to 36 TPH (day); electrical demand varies from 3.0 MW (night) to 5.4 MW (day), with day peaks of steam and power occurring together. The plant is grid-connected.

- a) Base thermal load matching: if the turbine is sized for the minimum process steam and run constantly, find its output and the day-time grid import. (1.5 Marks)
- b) Thermal load following: if all process steam passes through the turbine, find the power range and the day-time grid import. (1.5 Marks)
- c) Electrical load matching (stand-alone): find the steam needed to generate the day peak and comment on the result. (1.5 Marks)
- d) Recommend the scheme for this plant. (0.5 Mark)

S-7 Ans (model answer with marking scheme)

Given: Steam rate = 8 kg/kWh; process steam 24–36 TPH; power demand 3.0–5.4 MW; back pressure exhaust to process at 4 kg/cm²(g) ($\approx$ 3.92 bar(g)).

a) Base thermal load (1.5 marks)

Formula: Power = steam flow/steam rate; grid import = demand – generation

Substitution: 24,000/8 = 3,000 kW [0.5 mark]; day import = 5,400 – 3,000

Result: **Turbine output 3,000 kW constant; day-time import 2,400 kW; the extra 12 TPH of day steam goes through a PRDS** [1 mark]

b) Thermal load following (1.5 marks)

Formula: Same relation over the steam range

Substitution: Night 24,000/8 = 3,000 kW; day 36,000/8 = 4,500 kW [0.5 mark]; day import = 5,400 – 4,500

Result: **Power 3.0–4.5 MW following steam; day-time import 900 kW; night generation exactly meets the 3.0 MW demand** [1 mark]

c) Electrical load matching (1.5 marks)

Formula: Steam required = power $\times$ steam rate

Substitution: 5,400 $\times$ 8 = 43,200 kg/h against 36,000 kg/h process demand [0.5 mark]

Result: **43.2 TPH needed; 7.2 TPH more than the process can absorb, which would have to be vented or dumped, so a back pressure machine cannot run stand-alone here; an extraction-condensing turbine or grid support would be required** [1 mark]

d) Recommendation (0.5 mark)

Thermal load following with grid backup: every tonne of process steam produces power at high overall efficiency, no steam is wasted, and only 0.9 MW is imported at day peak. [0.5 mark]

Marking note: candidates must name the guidebook operating schemes (base electrical, base thermal, electrical load matching, thermal load matching); $\pm 1\%$ tolerance.

S-8 [3P2CEM058 | Chapter 9: Heat Exchangers | Overall U from film coefficients and tube material choice]

A lube oil cooler has oil on the shell side (film coefficient 600 W/m²K) and cooling water in the tubes (film coefficient 4,500 W/m²K). The tube wall is 1.5 mm thick. The duty is 400 kW with an LMTD of 28 °C. Two tube materials are compared: stainless steel (k = 16 W/m·K) and admiralty brass (k = 110 W/m·K). Neglect fouling and treat the wall as flat.

- a) Calculate the overall heat transfer coefficient for each material. (2 Marks)
- b) Calculate the area required for each. (1.5 Marks)

c) Identify the controlling resistance and show, by calculation for the stainless steel tubes, whether doubling the oil-side or the water-side coefficient is more effective. (1.5 Marks)

S-8 Ans (model answer with marking scheme)

Given: $h_o = 600 \text{ W/m}^2\text{K}$; $h_w = 4,500 \text{ W/m}^2\text{K}$; wall $\Delta x = 0.0015 \text{ m}$; $k_{SS} = 16$, $k_{brass} = 110 \text{ W/m}\cdot\text{K}$; $Q = 4,00,000 \text{ W}$; $\text{LMTD} = 28^\circ\text{C}$.

a) Overall U (2 marks)

Formula: $1/U = 1/h_o + \Delta x/k + 1/h_w$

Substitution: SS: $1/600 + 0.0015/16 + 1/4,500 = 0.0016667 + 0.0000938 + 0.0002222 = 0.0019826$ [1 mark];

brass: $0.0016667 + 0.0000136 + 0.0002222 = 0.0019025$

Result: **$U_{SS} \approx 504 \text{ W/m}^2\text{K}$; $U_{brass} \approx 526 \text{ W/m}^2\text{K}$** [1 mark]

b) Area (1.5 marks)

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

Substitution: SS: $4,00,000/(504.4 \times 28)$; brass: $4,00,000/(525.6 \times 28)$ [0.5 mark]

Result: **$A_{SS} \approx 28.3 \text{ m}^2$; $A_{brass} \approx 27.2 \text{ m}^2$ (only about 4% less)** [1 mark]

c) Controlling resistance (1.5 marks)

Formula: Share of each resistance = (its $1/h$ or $\Delta x/k$)/(1/U)

Substitution: Oil film share = $0.0016667/0.0019826 = 84\%$; doubling h_o : $1/U = 1/1,200 + 0.0000938 + 0.0002222 = 0.0011493$; doubling h_w : $1/U = 0.0016667 + 0.0000938 + 1/9,000 = 0.0018716$ [0.5 mark]

Result: **Oil film controls ($\approx 84\%$ of resistance). Doubling oil-side h raises U_{SS} to $\approx 870 \text{ W/m}^2\text{K}$ (+72%), whereas doubling water-side h gives only $\approx 534 \text{ W/m}^2\text{K}$ (+6%)** [1 mark]

Hence improve the oil side (baffle spacing, higher shell velocity, finned tubes) rather than pumping more water; tube material matters little here.

Marking note: $\pm 1\%$ tolerance; method marks for correct series-resistance formula.

..... End of Section II

Section – III: LONG DESCRIPTIVE QUESTIONS

Marks: 6 x 10 = 60

(i) Answer all Six questions

(ii) Each question carries Ten marks

L-1 [3P2CEM059 | Chapter 2: Boilers | Condensing economiser on a natural gas fired boiler]

A 10 TPH three-pass shell boiler fires 760 Sm³/h of natural gas (GCV 9,500 kcal/Sm³ ($\approx$ 39.8 MJ/Sm³), NCV 8,550 kcal/Sm³ ($\approx$ 35.8 MJ/Sm³)) with an efficiency of 84% on GCV basis. Flue gas (11,800 kg/h, containing 1,250 kg/h of water vapour from combustion of hydrogen in the gas) leaves at 140 °C. A condensing economiser is proposed to heat cold boiler make-up and process water, cooling the gas to 65 °C and condensing 15% of the water vapour. Take Cp of flue gas = 0.25 kcal/kg °C ($\approx$ 1.05 kJ/kg·K) and latent heat of condensation = 570 kcal/kg ($\approx$ 2,386 kJ/kg). Assume the recovered heat displaces heat that the boiler would otherwise supply at 84% efficiency. Gas costs ₹ 45 per Sm³; operation is 8,000 h/yr; installed cost is ₹ 85,00,000.

a) Calculate the sensible heat, latent heat and total heat recovered. (3 Marks)

b) Calculate the gas saving and the new overall boiler efficiency on GCV basis. (3 Marks)

c) Calculate the annual saving and the simple payback. (2 Marks)

d) Express the new efficiency on NCV basis and explain why condensing economisers are used mainly with natural gas. (2 Marks)

L-1 Ans (model answer with marking scheme)

Given: Gas = 760 Sm³/h; GCV = 9,500 kcal/Sm³ ($\approx$ 39.8 MJ/Sm³); NCV = 8,550 kcal/Sm³ ($\approx$ 35.8 MJ/Sm³); η = 0.84; flue gas 11,800 kg/h, 140 $\rightarrow$ 65 °C; water vapour 1,250 kg/h, 15% condensed; Cp = 0.25 kcal/kg °C ($\approx$ 1.05 kJ/kg·K); L = 570 kcal/kg ($\approx$ 2,386 kJ/kg); ₹ 45/Sm³; 8,000 h/yr; ₹ 85,00,000.

a) Heat recovered (3 marks)

Formula: Sensible = $m_g \times C_p \times \Delta t$; latent = condensed vapour $\times$ L; total = sensible + latent

Substitution: Sensible = $11,800 \times 0.25 \times (140 - 65) = 2,21,250$ kcal/h [1 mark]; latent = $0.15 \times 1,250 \times 570 = 187.5 \times 570 = 1,06,875$ kcal/h [1 mark]

Result: **Total heat recovered = 3,28,125 kcal/h ($\approx$ 382 kW), of which about one-third is latent heat** [1 mark]

b) Gas saving and new efficiency (3 marks)

Formula: Gas saved = $Q/(\eta \times \text{GCV})$; useful output = $\eta \times \text{gas} \times \text{GCV}$ (unchanged); new efficiency = useful output/(new gas $\times$ GCV)

Substitution: Gas saved = $3,28,125/(0.84 \times 9,500) = 41.1$ Sm³/h (5.4% of firing) [1 mark]; useful output = $0.84 \times 760 \times 9,500 = 60,64,800$ kcal/h; new gas = $760 - 41.1 = 718.9$ Sm³/h [1 mark]; $\eta = 60,64,800/(718.9 \times 9,500)$

Result: **Gas saving $\approx$ 41.1 Sm³/h; overall efficiency rises from 84% to $\approx$ 88.8% (GCV basis), close to the guidebook figure of about 5% gain for a condensing economiser cooling flue gas from 140 °C to 65 °C** [1 mark]

c) Economics (2 marks)

Formula: Annual saving = gas saved $\times$ hours $\times$ price; payback = cost/annual saving

Substitution: $41.12 \times 8,000 \times 45 = ₹ 1,48,02,632$ [1 mark]; payback = $85,00,000/1,48,02,632$

Result: **Annual saving $\approx$ ₹ 1.48 crore; payback $\approx$ 0.57 year (about 7 months)** [1 mark]

d) NCV basis and suitability (2 marks)

Formula: $\eta_{\text{NCV}} = \eta_{\text{GCV}} \times \text{GCV}/\text{NCV}$

Substitution: $88.8 \times 9,500/8,550$

Result: $\eta_{\text{NCV}} \approx 98.7\%$; deeper condensation can push NCV-based figures above 100%, because NCV excludes the latent heat that the economiser recovers. Indian practice reports efficiency on GCV basis to avoid this. [1 mark]

Natural gas is sulphur-free and hydrogen-rich: its flue gas has high water vapour (high dew point, more latent heat) and no SO_3 , so condensing on the tubes causes only mild carbonic acidity, handled with stainless steel. With sulphur-bearing oil or coal, condensate becomes sulphuric acid and corrodes the surfaces; a cold water sink below about 55°C is also essential. [1 mark]

Marking note: $\pm 1\%$ tolerance; accept the alternative method using % efficiency gain if the assumption is stated.

L-2 [3P2CEM060 | Chapter 3: Steam System | Steam system efficiency chain from generation to end use]

A process plant's steam system was audited. Data collected over a steady 8-hour period:

Parameter	Value
Coal fired	2,000 kg/h, GCV 4,200 kcal/kg ($\approx 17,585$ kJ/kg)
Steam generated	10,500 kg/h; enthalpy gain 600 kcal/kg ($\approx 2,512$ kJ/kg) (feed water 60°C)
Surface heat loss from uninsulated and damaged mains	3,40,000 kcal/h (≈ 395 kW)
Visible steam leaks	280 kg/h
Steam through failed-open traps	320 kg/h
Useful heat absorbed by products in all users	34,00,000 kcal/h ($\approx 3,953$ kW)
Coal price; operating hours	₹ 6,000 per tonne; 8,000 h/yr

Value the heat of leaked and trap-passing steam at 600 kcal/kg ($\approx 2,512$ kJ/kg).

- Calculate the generation (boiler) efficiency. (1.5 Marks)
- Calculate the distribution losses and distribution efficiency. (2.5 Marks)
- Calculate the utilisation efficiency and the overall steam system efficiency, and compare with the guidebook typical values. (2 Marks)
- If all leaks and failed traps are rectified and 70% of the surface loss is eliminated by insulation, calculate the coal saving per hour and per year (₹), and the new overall efficiency. (3 Marks)
- Where should the next phase of effort focus, and why? (1 Mark)

L-2 Ans (model answer with marking scheme)

Given: Coal 2,000 kg/h $\times$ 4,200 kcal/kg ($\approx 17,585$ kJ/kg); steam 10,500 kg/h $\times$ 600 kcal/kg ($\approx 2,512$ kJ/kg); surface loss 3,40,000 kcal/h (≈ 395 kW); leaks 280 kg/h; traps 320 kg/h; useful heat 34,00,000 kcal/h ($\approx 3,953$ kW); ₹ 6,000/t; 8,000 h/yr.

a) Generation efficiency (1.5 marks)

Formula: $\eta_{\text{gen}} = \text{heat in steam/fuel heat input}$

Substitution: Input = $2,000 \times 4,200 = 84,00,000$ kcal/h; steam heat = $10,500 \times 600 = 63,00,000$ kcal/h [1 mark]; $\eta = 63,00,000/84,00,000$

Result: $\eta_{\text{generation}} = 75\%$ [0.5 mark]

b) Distribution (2.5 marks)

Formula: Loss = surface loss + (leaks + traps) $\times$ 600; heat delivered = steam heat – loss; $\eta_{\text{dist}} = \text{delivered/steam heat}$

Substitution: Steam losses = $(280 + 320) \times 600 = 3,60,000$ kcal/h [1 mark]; total = $3,40,000 + 3,60,000 = 7,00,000$ kcal/h [0.5 mark]; delivered = $63,00,000 - 7,00,000 = 56,00,000$ kcal/h; $\eta = 56,00,000/63,00,000$

Result: **Distribution loss = 7,00,000 kcal/h (≈ 814 kW); $\eta_{\text{distribution}} \approx 88.9\%$** [1 mark]

c) Utilisation and overall (2 marks)

Formula: $\eta_{\text{util}} = \text{useful heat/heat delivered}$; $\eta_{\text{overall}} = \text{useful heat/fuel input} = \eta_{\text{gen}} \times \eta_{\text{dist}} \times \eta_{\text{util}}$

Substitution: $\eta_{\text{util}} = 34,00,000/56,00,000$; $\eta_{\text{overall}} = 34,00,000/84,00,000$ [1 mark]

Result: **$\eta_{\text{utilisation}} \approx 60.7\%$; $\eta_{\text{overall}} \approx 40.5\%$** , against guidebook typical values of about 80% generation, 83% distribution, 47% utilisation and 35% overall; this plant is better than typical in distribution and utilisation, but generation is below par. [1 mark]

d) After rectification (3 marks)

Formula: Heat saved at the steam side = steam losses + $0.7 \times \text{surface loss}$; coal saved = heat saved/ $(\eta_{\text{gen}} \times \text{GCV})$

Substitution: Heat saved = $3,60,000 + 0.7 \times 3,40,000 = 5,98,000 \text{ kcal/h}$ [1 mark]; coal = $5,98,000/(0.75 \times 4,200) = 189.8 \text{ kg/h}$; per year $189.8 \times 8,000 = 1,518.7 \text{ t}$; ₹ = $1,518.7 \times 6,000$ [1 mark]; new input = $84,00,000 - 189.8 \times 4,200 = 76,02,667 \text{ kcal/h}$; $\eta_{\text{overall}} = 34,00,000/76,02,667$

Result: **Coal saving $\approx 190 \text{ kg/h} \approx 1,519 \text{ t/yr} \approx ₹ 91.1 \text{ lakh/yr}$; new overall efficiency $\approx 44.7\%$** [1 mark]

e) Next focus (1 mark)

Utilisation (about 61%) is still the weakest link after generation: end-use measures such as correct steam pressure at users, air venting, proper trapping, process insulation, waste-heat and condensate recovery, and process scheduling act on the largest remaining loss, and every unit saved at end use also avoids its generation and distribution losses. Boiler tuning to raise generation efficiency is the parallel priority. [1 mark]

Marking note: $\pm 1\%$ tolerance; award method marks for chain multiplication even if values differ due to rounding.

L-3 [3P2CEM061 | Chapter 4: Furnaces | Air infiltration in a reheating furnace under negative pressure]

An oil-fired reheating furnace burns 520 kg/h of furnace oil (GCV 10,200 kcal/kg ($\approx 42,705 \text{ kJ/kg}$); theoretical air 14 kg/kg oil). The burners are set at 10% excess air. Because the furnace runs under negative pressure, cold air enters through the charging door, peep holes and wall cracks. The flue gas at the furnace exit (before the recuperator) shows 8% O_2 (dry), and its temperature is 900°C . Ambient temperature is 35°C ; take C_p of air/flue gas as $0.24 \text{ kcal/kg } ^\circ\text{C}$ ($\approx 1.00 \text{ kJ/kg}\cdot\text{K}$). Furnace oil costs ₹ 50,000 per tonne; the furnace runs 6,000 h/yr.

- Calculate the excess air at the furnace exit, the total air per kg of oil and the infiltrated air (kg/kg oil and kg/h). (3 Marks)
- Calculate the heat carried away by infiltrated air (per kg oil, per hour and as % of fuel input). (2 Marks)
- Estimate the oil and cost saving if the furnace is held at slight positive pressure so that infiltration is eliminated (take % fuel saving equal to % loss eliminated). (3 Marks)
- What problems arise if the furnace pressure is set too high, and how should pressure be controlled? (2 Marks)

L-3 Ans (model answer with marking scheme)

Given: Oil = 520 kg/h; GCV = 10,200 kcal/kg ($\approx 42,705 \text{ kJ/kg}$); theoretical air = 14 kg/kg; burner excess air 10%; exit $\text{O}_2 = 8\%$; flue gas 900°C ; ambient 35°C ; $C_p = 0.24 \text{ kcal/kg } ^\circ\text{C}$ ($\approx 1.00 \text{ kJ/kg}\cdot\text{K}$); ₹ 50,000/t; 6,000 h/yr.

a) Excess air and infiltration (3 marks)

Formula: Excess air % = $\text{O}_2/(21 - \text{O}_2) \times 100$; total air = $(1 + \text{EA}) \times \text{theoretical air}$; infiltration = total – burner air

Substitution: EA = $8/(21 - 8) = 61.5\%$ [1 mark]; total air = $1.615 \times 14 = 22.62 \text{ kg/kg}$; burner air = $1.10 \times 14 = 15.40 \text{ kg/kg}$ [1 mark]; infiltrated = $22.62 - 15.40 = 7.22 \text{ kg/kg}$; $\times 520$

Result: **Excess air at exit $\approx 61.5\%$; infiltrated air $\approx 7.22 \text{ kg/kg oil} \approx 3,752 \text{ kg/h}$** [1 mark]

b) Heat lost with infiltrated air (2 marks)

Formula: Loss = $m_{\text{inf}} \times C_p \times (t_{\text{fg}} - t_{\text{a}})$

Substitution: $7.215 \times 0.24 \times (900 - 35) = 1,497.9 \text{ kcal/kg oil}$ [1 mark]; $\times 520 = 7,78,915 \text{ kcal/h}$; % = $1,497.9/10,200 \times 100$

Result: Loss $\approx 1,498$ kcal/kg oil ($\approx 6,272$ kJ/kg oil) $\approx 7,78,915$ kcal/h (≈ 906 kW) $\approx 14.7\%$ of fuel input [1 mark]

c) Saving from pressure control (3 marks)

Formula: Oil saved = % loss $\times$ oil rate; annual = $\times$ hours; ₹ = t $\times$ price

Substitution: $0.1469 \times 520 = 76.4$ kg/h [1 mark]; $76.4 \times 6,000 = 4,58,185$ kg = 458.2 t/yr [1 mark]; $458.2 \times 50,000$

Result: **Oil saving ≈ 76 kg/h ≈ 458 t/yr $\approx ₹ 2.29$ crore/yr**; the guidebook notes that infiltration can account for up to about 40% of losses in poorly sealed furnaces. [1 mark]

d) Excess positive pressure and control (2 marks)

Too high a pressure causes exfiltration: flames and hot gases leak out through doors and openings, overheating door frames, burner blocks and structural steel, burning out ducting and seals, raising heat loss from openings and creating safety hazards for operators. [1 mark]

The furnace should be held slightly above atmospheric pressure at hearth level by an automatic flue damper controlled from a pressure transmitter, with openings kept small, doors sealed and closed promptly, and cracks repaired. [1 mark]

Marking note: $\pm 1\%$ tolerance; candidates using an approximate infiltration estimate from CO₂ readings may earn method marks.

L-4 [3P2CEM062 | Chapter 7: Cogeneration | Bottoming-cycle power from kiln waste heat in a sponge iron plant]

A sponge iron (DRI) plant has four rotary kilns whose waste gases pass through after-burning chambers and waste heat recovery boilers (WHRBs). Together the WHRBs generate 48 TPH of steam at 66 kg/cm²(g) (≈ 64.7 bar(g)), 490 °C (enthalpy 810 kcal/kg ($\approx 3,391$ kJ/kg)) from feed water at 130 °C (enthalpy 130 kcal/kg (≈ 544 kJ/kg)). The steam drives a condensing turbine-generator whose heat rate, based on heat added to the steam, is 3,650 kcal/kWh ($\approx 15,282$ kJ/kWh). Auxiliary consumption is 9% of gross generation; plant load factor is 90%. Power displaced from the grid costs ₹ 7.20 per kWh; the grid's thermal heat rate is 2,450 kcal/kWh ($\approx 10,258$ kJ/kWh). Project cost is ₹ 72 crore.

- a) Calculate the gross and net power output and the turbine steam rate. (3 Marks)
- b) Calculate the annual net generation, the annual saving and the simple payback. (3 Marks)
- c) Calculate the primary energy (toe/yr) avoided at the grid. (2 Marks)
- d) Explain why this is a bottoming cycle and why a condensing rather than back pressure turbine is chosen here. (2 Marks)

L-4 Ans (model answer with marking scheme)

Given: Steam = 48,000 kg/h; $h_s = 810$, $h_{fw} = 130$ kcal/kg (≈ 544 kJ/kg); heat rate = 3,650 kcal/kWh ($\approx 15,282$ kJ/kWh); auxiliaries 9%; PLF 90%; ₹ 7.20/kWh; grid heat rate 2,450 kcal/kWh ($\approx 10,258$ kJ/kWh); cost ₹ 72 crore.

a) Power output (3 marks)

Formula: Heat added to steam = $m \times (h_s - h_{fw})$; gross power = heat added/heat rate; net = gross $\times (1 - 0.09)$; steam rate = steam/gross power

Substitution: Heat added = $48,000 \times (810 - 130) = 3,26,40,000$ kcal/h [1 mark]; gross = $3,26,40,000 / 3,650 = 8,942$ kW [1 mark]; net = $8,942.5 \times 0.91$; steam rate = $48,000 / 8,942.5$

Result: **Gross $\approx 8,942$ kW; net $\approx 8,138$ kW; steam rate ≈ 5.37 kg/kWh** [1 mark]

b) Annual generation and economics (3 marks)

Formula: Annual net kWh = net kW $\times 8,760 \times$ PLF; saving = kWh $\times$ tariff; payback = cost/saving

Substitution: $8,137.6 \times 8,760 \times 0.90 = 6,41,57,184$ kWh [1 mark]; ₹ = $6,41,57,184 \times 7.20 = ₹ 46,19,31,725$ [1 mark]; payback = $72,00,00,000 / 46,19,31,725$

Result: **Net generation ≈ 6.42 crore kWh/yr; saving $\approx ₹ 46.2$ crore/yr; payback ≈ 1.56 years** [1 mark]

c) Primary energy avoided (2 marks)

Formula: toe = kWh × grid heat rate/10⁷

Substitution: 6,41,57,184 × 2,450/10⁷ [1 mark]

Result: ≈ **15,719 toe/yr (≈ 658 TJ/yr) of primary energy avoided at the grid** [1 mark]

d) Bottoming cycle and turbine choice (2 marks)

In a bottoming cycle, the primary fuel produces high-temperature thermal energy for the process (here the kilns reducing iron ore), and the reject heat from the process is used to raise steam for power; it is common in sponge iron, cement, steel and ceramic plants with high-temperature waste gas. No additional fuel is fired for power. [1 mark]

A DRI plant has little process steam demand, so a back pressure turbine would have no use for its exhaust; a condensing turbine extracts the maximum power from the waste-heat steam and rejects the residual heat in the condenser. [1 mark]

Marking note: ±1% tolerance; method marks if heat rate is applied correctly even when auxiliary consumption is overlooked.

L-5 [3P2CEM063 | Chapter 8: Waste Heat Recovery | Heat recovery from hot dye-house effluent to fresh process water]

A textile dye house drains 50 m³/h of hot effluent at 78 °C. At the same time, 45 m³/h of fresh water at 28 °C is heated to 65 °C with steam for dyeing and washing. A gasketed plate heat exchanger with a self-cleaning filter on the effluent side is proposed to heat the fresh water to 58 °C before the steam heater. Take density 1,000 kg/m³ and Cp 1 kcal/kg °C (≈ 4.19 kJ/kg·K) for both streams. Steam is generated in a coal-fired boiler (efficiency 75%, coal GCV 3,800 kcal/kg (≈ 15,910 kJ/kg), coal ₹ 7,000 per tonne); with condensate returned, each kg of steam supplies 500 kcal (≈ 2,093 kJ). The recovery system (heat exchanger, filter, pump; 22 kW extra electrical load at ₹ 8/kWh) costs ₹ 1,10,00,000. Operation: 6,500 h/yr.

a) Calculate the heat recovered and the effluent outlet temperature. (3 Marks)

b) Calculate the steam and coal saved per hour. (3 Marks)

c) Calculate the net annual saving and simple payback. (2 Marks)

d) How does the guidebook grade this waste heat source, and what design precautions and indirect benefits apply? (2 Marks)

L-5 Ans (model answer with marking scheme)

Given: Effluent 50,000 kg/h at 78 °C; fresh water 45,000 kg/h, 28 → 58 °C; Cp = 1 kcal/kg °C (≈ 4.19 kJ/kg·K); steam useful heat 500 kcal/kg (≈ 2,093 kJ/kg); boiler η = 0.75; GCV = 3,800 kcal/kg (≈ 15,910 kJ/kg); coal ₹ 7,000/t; 22 kW at ₹ 8/kWh; ₹ 1,10,00,000; 6,500 h/yr.

a) Heat recovered and effluent outlet (3 marks)

Formula: Q = m_w × Cp × (t_{out} – t_{in}); effluent outlet = t_{in,eff} – Q/(m_{eff} × Cp)

Substitution: Q = 45,000 × 1 × (58 – 28) = 13,50,000 kcal/h [1 mark]; drop = 13,50,000/50,000 = 27 °C [1 mark]; outlet = 78 – 27

Result: **Heat recovered = 13,50,000 kcal/h (≈ 1,570 kW); effluent leaves at 51 °C** [1 mark]

b) Steam and coal saved (3 marks)

Formula: Steam saved = Q/500; coal saved = Q/(η × GCV)

Substitution: Steam = 13,50,000/500 = 2,700 kg/h [1 mark]; coal = 13,50,000/(0.75 × 3,800) = 13,50,000/2,850 [1 mark]

Result: **Steam saving = 2,700 kg/h; coal saving ≈ 474 kg/h** [1 mark]

c) Economics (2 marks)

Formula: Net saving = coal saved $\times$ hours $\times$ price – electricity cost; payback = cost/net saving

Substitution: Coal: $473.7 \times 6,500 = 30,78,947 \text{ kg} = 3,078.9 \text{ t} \times 7,000 = ₹ 2,15,52,632$; electricity: $22 \times 6,500 \times 8 = ₹ 11,44,000$ [1 mark]; net = ₹ 2,04,08,632; payback = $1,10,00,000/2,04,08,632$

Result: **Net saving $\approx$ ₹ 2.04 crore/yr; payback $\approx$ 0.54 year (about 6.5 months)** [1 mark]

d) Grade, precautions and indirect benefits (2 marks)

Table 8.1 grades heat in liquid effluents as poor if heavily contaminated, since alloy exchangers are then needed; dye effluent carries lint, fibres and chemicals, so the design needs a self-cleaning filter, wide-gap or stainless/titanium plates, provision for cleaning (CIP or easy opening) and regular monitoring of approach temperature for fouling. [1 mark]

Indirect benefits: the effluent reaches the treatment plant 27°C cooler, easing biological treatment and reducing cooling needs; steam demand and boiler load fall, reducing ash, emissions and auxiliary power; the steam heater can be smaller. [1 mark]

Marking note: $\pm 1\%$ tolerance; accept the steam route ($2,700 \text{ kg/h} \times \text{enthalpy}/\eta$) if the assumption is stated.

L-6 [3P2CEM064 | Chapter 9: Heat Exchangers | Temperature-cross duty: plate versus shell-and-tube exchanger]

A pressurised process liquor (6.0 kg/s , $C_p 4.0 \text{ kJ/kg}\cdot\text{K}$) must be cooled from 145°C to 60°C by heating boiler feed water (5.5 kg/s , $C_p 4.18 \text{ kJ/kg}\cdot\text{K}$) entering at 30°C . Two options are considered:

Option	Overall U ($\text{W/m}^2\text{K}$)	LMTD correction factor F	Installed cost
Shell-and-tube, four shells in series (a single 1-2 shell cannot meet the duty)	1,100	0.82	₹ 22,000 per m^2
Gasketed plate heat exchanger (true counter flow)	4,500	1.0	₹ 30,000 per m^2

- Calculate the duty and the feed water outlet temperature, and show that a temperature cross exists. (2 Marks)
- Calculate the counter-flow LMTD. (2 Marks)
- Calculate the area and cost of the shell-and-tube option. (2 Marks)
- Calculate the area and cost of the plate option and compare. (2 Marks)
- Under what conditions would the plate exchanger not be suitable? (2 Marks)

L-6 Ans (model answer with marking scheme)

Given: Hot: $6.0 \text{ kg/s} \times 4.0 \text{ kJ/kg}\cdot\text{K}$, $145 \rightarrow 60^\circ\text{C}$; cold: $5.5 \text{ kg/s} \times 4.18 \text{ kJ/kg}\cdot\text{K}$ from 30°C ; $U_{\text{ST}} = 1,100$, $F = 0.82$; $U_{\text{PHE}} = 4,500 \text{ W/m}^2\text{K}$, $F = 1.0$; costs ₹ 22,000 and ₹ 30,000 per m^2 .

a) Duty and outlet (2 marks)

Formula: $Q = m_h \times C_{p_h} \times (T_{\text{in}} - T_{\text{out}})$; $t_{\text{out}} = t_{\text{in}} + Q/(m_c \times C_{p_c})$

Substitution: $Q = 6.0 \times 4.0 \times 85 = 2,040 \text{ kW}$ [1 mark]; $t_{\text{out}} = 30 + 2,040/(5.5 \times 4.18) = 30 + 88.7$

Result: **$Q = 2,040 \text{ kW}$; feed water leaves at $\approx 118.7^\circ\text{C}$, well above the hot outlet of 60°C , so a temperature cross exists** [1 mark]

b) LMTD (2 marks)

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

Substitution: $\Delta T_1 = 145 - 118.7 = 26.3^\circ\text{C}$; $\Delta T_2 = 60 - 30 = 30.0^\circ\text{C}$ [1 mark]; $\text{LMTD} = (30.0 - 26.3)/\ln(30.0/26.3)$

Result: **$\text{LMTD} \approx 28.1^\circ\text{C}$** [1 mark]

c) Shell-and-tube (2 marks)

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

Substitution: $A = 20,40,000/(1,100 \times 0.82 \times 28.09)$ [1 mark]; cost = $80.5 \times 22,000$

Result: **$A_{ST} \approx 80.5 \text{ m}^2$; cost $\approx ₹ 17.7 \text{ lakh}$** [1 mark]

d) Plate exchanger (2 marks)

Formula: $A = Q/(U \times \text{LMTD})$, $F = 1$ for true counter flow

Substitution: $A = 20,40,000/(4,500 \times 28.09)$ [1 mark]; cost = $16.1 \times 30,000$

Result: **$A_{PHE} \approx 16.1 \text{ m}^2$; cost $\approx ₹ 4.8 \text{ lakh}$, about one-fifth of the area and about 27% of the cost of the shell-and-tube option, in a much smaller footprint** [1 mark]

e) Limitations of plate exchangers (2 marks)

The guidebook notes that plate exchangers achieve high U (up to about $8,000 \text{ W/m}^2\text{K}$ against generally below $2,500$ for shell-and-tube) and handle temperature crosses in true counter flow, but gaskets limit their pressure and temperature capability. They are unsuitable for high-pressure or high-temperature duties beyond gasket ratings (brazed or welded plate types may be considered), for fluids with large solids or fibres that block narrow channels, for large volumetric flows of gases or vapours, and where gasket compatibility with the chemical is doubtful. [2 marks: 0.5 per valid point]

Marking note: $\pm 1\%$ tolerance; a candidate who uses a single 1-2 shell and an F factor without recognising the temperature cross loses the method mark in (c).

..... End of Section III

Answer key – Section I

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