Paper 4 Code :Pink

22nd NATIONAL CERTIFICATION EXAMINATION
FOR
ENERGY MANAGERS & ENERGY AUDITORS - JULY, 2022

PAPER – 4 : ENERGY PERFORMANCE ASSESSMENT FOR EQUIPMENT AND UTILITY SYSTEMS
Date : 31.07.2022 Timings : 14:00-16:00 HRS Duration : 2 HRS Max. Marks : 100

Section - I: BRIEF QUESTIONS **Marks: 10 x 1 = 10**

(i) Answer all **Ten** questions
(ii) Each question carries **One** mark

1	Ideally the flow capacity of a forced draft fan of a pulverised fuel boiler operating on balanced draft when compared to an Induced draft fan is _________.	Lower/ Higher/ Same	**Lower**
2	If waste heat delivered to the heat pump is 3440 kcal/hr and power consumed by the heat pump compressor is 1.5 KW then the heat developed by heat pump is 6.6 kW.	True/False	**False**
3	Lower the terminal temperature difference in a steam condenser of a turbine _________ is the heat transfer rate between steam and cooling water.	Lower/Higher	**Higher**
4	In an extraction back pressure cogeneration system, higher the steam flow through the turbine extractions, lower is the energy utilisation factor.	True/False	**False**
5	Rated power of motor is power consumed by motor.	True/False	**False**
6	NPSH required of centrifugal pump increases with flow.	True/False	**True**
7	The lower the dew point of air the higher is the moisture in air.	True/False	**False**
8	In a boiler, higher the % of CO_2 in flue gas, the better is the combustion efficiency.	True/False	**True**
9	Air infiltration in an air-conditioned building, will increase both the latent heat and sensible heat load.	True/False	**True**
10	The turbine cycle efficiency of a thermal power plant will increase with decrease in inlet cooling water temperature to the condenser.	True/False	**True**

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

Section - II: SHORT NUMERICAL QUESTIONS **Marks: 2 x 5 = 10**

L1 A process plant is importing 25 TPH of steam at 20 bar(g) pressure and reduces to 5 bar(g) through PRDS. The plant is also operating a motor driven gas compressor drawing a power of 900 kW. During an energy audit, it was suggested to evaluate the scheme of installing back pressure steam turbine instead of PRDS for driving the gas compressor.

Calculate the steam required for operating the back pressure turbine and hourly monetary savings.

Turbine power generation per unit of inlet steam: 0.045 kWh/kg steam

Power cost : Rs.9.0/kWh

Import steam cost : Rs.3500/MT

<table>
<tr><td>L1 Sol</td><td>

Steam required for back pressure turbine	= (900/0.045)/1000 = 20 TPH
Hourly monetary saving	= (900 x 9) = Rs.8,100

</td></tr>
<tr><td>L2</td><td>A medium size edible oil plant planning to install a thermic fluid heating system for their process requirements. The suggested operating parameters of thermic fluid system are given below:

<u>Heat output data:</u>

Flow rate of thermic fluid : 60 m^3/h
Inlet temperature of thermic fluid : 210 °C
Outlet temperature of thermic fluid : 230 °C
Flue gas exit temperature of thermic fluid heater: 275 °C
Specific heat of thermic fluid : 2.223 kJ/kg °C
Density of the thermic fluid : 826 kg/m^3

The plant has to choose either oil or briquettes as a fuel. The following is the additional data:

Efficiency of oil fired thermic fluid heater: 80%
Efficiency of briquette fired thermic fluid heater: 65%
GCV of fuel oil : 10000 kcal/kg
Cost of oil :Rs.70/kg
GCV of briquettes : 3200 kcal/kg
Cost of briquettes : Rs.8/kg

a) Calculate the heat load of the system in kcal/hr — 3 marks
b) As an energy auditor, based on operating cost, which system will you recommend? — 2 marks</td></tr>
<tr><td>L2-Sol</td><td>Solution:

Heat load of the system:
Mass of thermic fluid = 60 x 826
= 49,560 kg/hr
Q = mcpΔT
= 49,560 x 2.233/4.186 x (230-210)
= 528750.50 kcal/hr
Operating Cost:
1. Fuel oil quantity required = 528750 / (10000 x 0.8) = 66.09 kg/hr
Operating cost with fuel oil = 66.09 x 70 = Rs.4626.3 /hr

2. briquettes required = 528750.50 / (3200 x0.65) = 254.21 kg/hr
Operating cost with briquettes = 254.21 x 8 = Rs.2033.68/hr

As an energy auditor based on the operating cost, I recommend briquettes as fuel for thermic fluid heater system.</td></tr>
</table>

.... **End of Section - II** ....

Section - III: LONG NUMERICAL QUESTIONS **Marks: 4 x 20 = 80**

(i) Answer all the **<u>Four</u>** questions

(ii) Each question carries **<u>Twenty</u>** marks

N-1	**(N1-A)** A large water tube boiler was assessed for its performance. The operating conditions and design conditions of the boiler are given below.

Parameter	Unit	Design conditions	Operating conditions
Steam generation	Tons/hr	210	210
Steam pressure	Kg/cm^2	100	100
Steam temperature	^0c	440	440
Coal firing rate	Tons/hr	43	--
Flue gas temp, at the inlet of Air preheater	^0C	370	--
Flue gas temp, at the outlet of Air preheater	^0C	145	165
Excess air at Air preheater inlet	%	20%	25%
Excess air at preheater outlet	%	--	44.8%
% Dry flue gas heat loss	%	4.67	--
GCV of coal	Kcal/kg	3585	3240
Specific heat of flue gas	Kcal/kg k	0.23	0.23
Dry flue gas flow at air preheater outlet	Tons/hr	--	324
Ambient temperature	^0C	36	37

Using the above data calculate the following:

i) Heat loss in dry flue gas in kcal /hr at design conditions **(4 Marks)**

ii) Heat loss in dry flue gas in kcal/hr at operating conditions **(3 Marks)**

iii) Increase in operating coal consumption in Tons/hr w.r.t design due to higher dry flue gas loss considering boiler efficiency of 86%. **(2 Marks)**

iv) Additional expenditure to meet increased heat loss towards coal for 7000 hours of operation of boiler in a year at a coal cost of Rs 9500/Ton. **(3 Marks)**

v) Air leakage in Air preheater in **%** during operation **(3 Marks)**

(N1-B) Calculate the following for the given operating parameters

i) Effectiveness of air preheater **(3 Marks)**

ii) Actual mass of air supplied in kg of air/kg of fuel the following operating parameters: **(2 Marks)**

S.NO	Description	Design	Operating
1.	Generation	500 MW	440 MW
2.	Flue Gas temperature inlet (°C)	356	315
3.	Flue Gas temperature outlet (°C)	147	178
4.	Air temperature inlet (°C)	36	40
5.	Air temperature outlet (°C)	316	294
6.	Measured O_2 % in flue gas	3.56	4.56
7.	Theoretical air kg of air/kg of fuel	5.0	5.0

N1-S

A.

i) Calculation of dry flue gas heat loss at design conditions

We have,

% Heat loss in dry flue gas= 4.67% = m Cp (Tf—Ta)/GCV of coal--------------(i)

m= mass of dry flue gas, kg/kg of coal

Cp=Specific heat of flue gas, kcal/kg k=0.23

Tf=temp of flue gas deg,C=145

Ta=Ambient temp. Deg,C=36

GCV=3585 kcal/kg

Substituting the values in equation (i)

4.67/100= m x 0.23 x(145-36)/(3585)

m= 6.68 kg of dry flue gas/kg of coal (or) m= 6.67 kg of dry flue gas/kg of coal

coal firing rate= 43x 1000 kg/hr

mass flow rate of dry flue gas=6.68 x 43x1000 =287240 kg/hr=M

Heat loss in dry flue gas= M x Cp x (Tf-Ta)=287240x0.23x(145—36)

=7201106.8 kcal/hr (or) 7190326 kcal/hr

Alternate solution:

= 4.67 x 3585/100

Flue gas loss = 167.41 kcal/ kg

Design fuel firing rate = 43TPH

= 43x1000 x 167.41

= 7198630 kcal/hr

ii) Flue gas Heat loss at operating conditions =M x Cp x (Tf-Ta)-----------(ii)

M=324x 1000 kg/hr

Cp=0.23 kcal/kg K

Tf=165 deg C

Ta=37 deg.C

Substituting the values in equation (ii)

Flue gas heat loss= 324x1000x 0.23 x(165-37)=9538560 kcal/hr-------ANS

iii) Increase in dry flue gas heat loss=(9538560—7201106.8)

=2337453.2 kcal/h------ANS

Coal equivalent of heat loss= 2337453.2/3240kcal/kg=721.4 kg/h

Considering boiler efficiency of 86%,

Coal equivalent= 721.4/0.86=838.8 kg/hr =0.8388 Tons/hr--------ANS

iv) Additional expenditure to be incurred due increase in dry flue gas heat loss/year=

= 0.8388 x 7000 x 9500 Rs/ton=Rs 557.80 Lakhs ---------ANS

v) Air preheater leakage=(O_2 % in flue gas leaving APH—O_2 % in flue gas entering APH)/

(21—O_2 % in flue gas leaving APH)---------------(iii)

25% and 44.8 % excess air corresponds to 4.2% O_2 and 6.5% O_2 respectively as calculated from the equation, Excess Air = %O_2 /(21-%O_2)

From equation (iii)

% APH leakage=(6.5—4.2)/(21-6.5)x100 =15.86% ----------ANS

N1-B-S

Solution:

i).APH Effectiveness (4 Marks)

= (Air temp APH out - Air Temp APH in)/(Flue Gas temp. APH in - Air temp. APH in) x 100

= (294-40)/(315-40) x100

= 92.36 %

ii). Actual mass of air supplied in kg of air/kg of fuel (1 Mark)

Excess air = O_2%/(21-O_2%) = 4.56 % / (21-4.56%) x 100 = 27.73 %

AAS = (1+EA/100) x Theoretical Air

= (1+0.2773) x 5

= 6.386 kg of air/ kg of fuel

N-2	a) Two stage reciprocating belt driven air compressor is designed for the following conditions. Air inlet pressure : 1.033 kg/cm² (a) Air inlet temperature : 30 °C Compressor discharge pressure : 7.0 kg/cm² (g) Isothermal efficiency : 60% Free air delivery : 11.67 m³/min Motor efficiency : 85% Belt power transmission efficiency: 97%

Air inlet pressure : 1.033 kg/cm^2 (a)
Air inlet temperature : 30 °C
Compressor discharge pressure : 7.0 kg/cm^2 (g)
Isothermal efficiency : 60%
Free air delivery : 11.67 m^3/min
Motor efficiency : 85%
Belt power transmission efficiency: 97%

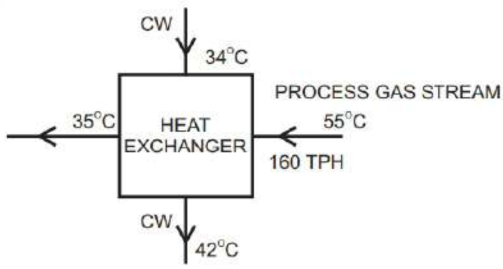
Based on the above data calculate:

(i) Power input to the motor (5 Marks)
(ii) Specific power consumption of the compressor in kW/m^3/hr (1 Mark)
(iii) If the discharge air pressure is reduced to 6 kg/cm^2 g, calculate the reduction in kW (2 Marks)

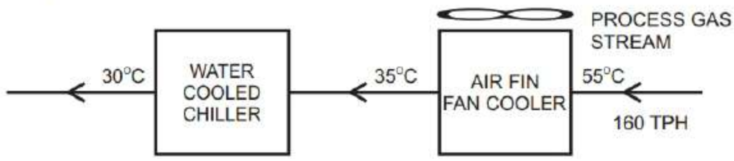
b) In a chemical plant, a 160 TPH process gas stream is being cooled from 55 deg.C to 35 deg.C through cooling water exchanger. The plant team would like to further cool this process gas stream upto 30 deg.C as the envisaged monetary saving potential is Rs.200 lakhs per annum due to process improvement.

Due to water scarcity, the plant team proposes to completely avoid the usage of existing water cooled heat exchanger and proposes to use air fin fan cooler followed by a water followed by a water-cooled chiller as shown in the schematic.

EXISTING CASE :



PROPOSED CASE :



Specific heat of process stream : 0.5 kcal/kg-deg.C
COP of chiller : 4.2
Chiller compressor motor efficiency : 94%
Range of the chiller cooling tower : 5°C
Power consumption of Cooling Tower Pump and Fan: 11 kW
Power cost : Rs. 9/kWh
CW cost : Rs.1.2/m^3
Volumetric flow rate through air fin fan cooler : 300 m^3/sec
Fan differential static pressure : 100 mmWC
Fan efficiency : 70%
Fan motor efficiency : 95%
Operating hours in year : 8000 hrs

Calculate the following:

	i) Cooling water flow (m^3/hr) of existing cooling water exchanger for reducing process gas stream temperature from 55 deg.C to 35 deg.C. (1 Mark) ii) Proposed chiller capacity (TR) (3 Marks) iii) Circulating cooling water flow for the proposed chiller (m^3/hr). (3 Marks) iv) Power consumption (kW), for the proposed water-cooled chiller and air fin fan cooler (3 Marks) v) As an energy auditor economically evaluate the new proposal and give your recommendation. (2 Marks)

N2-S

(A)

(i) Power input to motor

We have Isothermal efficiency = Isothermal power/compressor shaft power = 0.6 -----(1)

Isothermal power =P1 X Qf x $\log_e$ r /36.7 kw---------------(2)

Where,

P1 = Compressor inlet air pressure kg/cm^2 a =1.033

Qf = FAD m^3 /min =11.67x60 m^3/hr

r=P2/P1=(7+1.033)/1.033 =7.776

Substituting the values in equation(2) we get

Isothermal power =1.033 x 11.67 x 60 x $\log_e$ 7.776/36.7 =40.4 kw

From equation(1), 0.6= 40.4/compressor shaft power

Compressor shaft power= 40.4/0.6=67.33 kW

Input power to motor= 67.33/(0.85 x 0.97)=81.66 kW -------ANS

(ii) Specific power consumption=Input power to motor, kW/m^3/hr

= 81.66/(11.67 x 60)

=0.1166 kW/(m^3/hr) –ANS

(iii) calculation of % Saving in power when discharge pressure reduced to 6 kg/cm^2 (g)

NOW, P2= 6 + 1.033 kg/cm^2 a

P1= 1.033 KG/CM^2 a

r= (6+1.033)/1.033=6.8

FAD=11.67 m^3 /min

Isothermal power=(1.033 x 11.67 x 60 x $\log_e$ 6.8)/(36.7)

=37.8 kw

From equation (1) above,

Isothermal efficiency= 0.6 = 37.8/compressor shaft power

Compressor shaft power=37.8/0.6=63 kW

Motor input power= 63/(0.97x0.85)=76.4 kW

Saving in power=(81.66—76.41)

= 5.25 kW

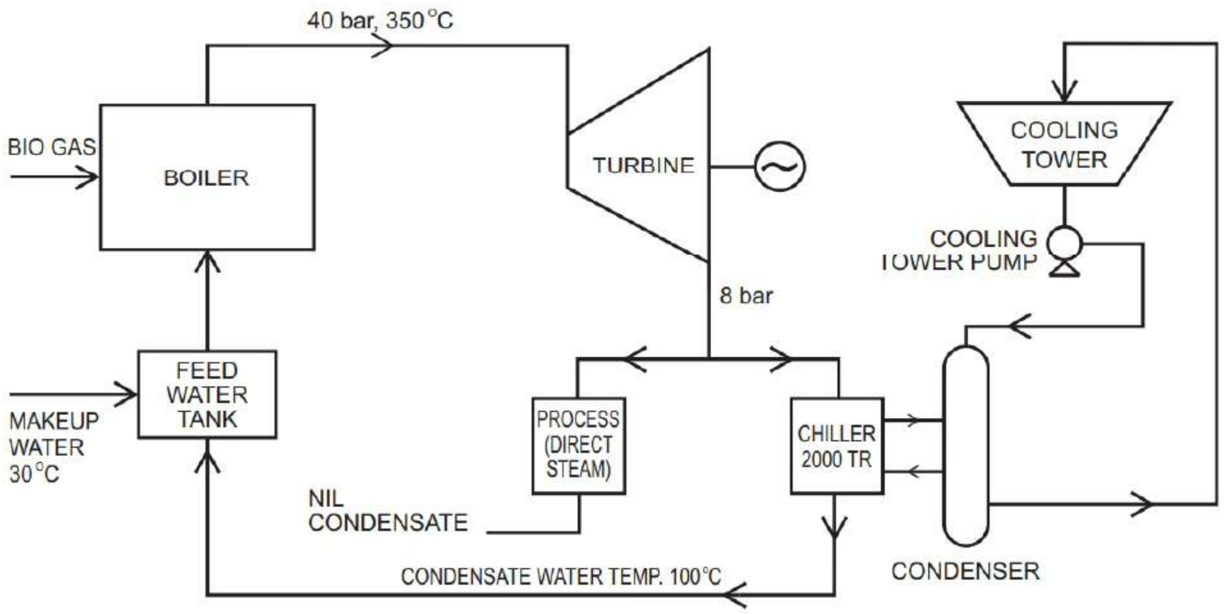
b)

i)	Cooling water flow rate to existing cooler	= 160 x 0.5 x (55-35)/(42-34) **= 200.0**	m^3/hr
ii)	Total required duty in new scheme	= 160 x 1000 x 0.5 x (55-30) /10^6 = 2.00	Gcal/hr
	Heat Duty of Air Fin fan cooler	= 160 x 1000 x 0.5 x (55-35) /10^6 = 1.6	Gcal/hr
	Residual duty required through chiller	= (2.0 – 1.6) x 10^6/3024 **= 132**	TR
iii)	Chiller condenser duty	= 132 x (1+ 1/4.2) = 163	TR
		= 163 x 3024 / 10^6 = 0.49	Gcal/hr

	Required CW flow for new chiller condenser	= (0.49 x 10^6)/{1.0 x (5)} /1000 **= 98**	m^3/hr
iv)	Centrifugal chiller power consumption	= (132 x 3024/4.2/860) /0.94 = 117.6	kW
	Fan power consumption of Air fin fan cooler	= 300 x100/(102x0.70 x 0.95) = 442	kW
	Power consumption of cooling tower pump and fan	= 11 kW	
	Total additional power consumption	= 117.6 + 442 +11 = 570.6	kW
v)	Power cost for the proposed scheme	= 570.6 x 9 x 8000 /10^5 = 410.832	Rs. Lakhs PA
	Process benefit of new scheme	= 200.0	Rs. Lakhs PA
	Potential monetary loss of new scheme	= (200 – (410.832)) **= -210.83**	Rs. Lakhs PA

Recommendation: Proposed scheme is not viable.

N-3 An effluent treatment plant is treating effluent from nearby industries and treated effluent is recycled back to nearby industries as raw water. The biogas generated from the effluent plant is used in a back pressure co-generation system. The operation details are shown in the schematic and tabulated below:



Steam enthalpy at generation pressure and temperature: 720 kcal/kg
Enthalpy of back pressure steam : 660 kcal/kg
Turbine efficiency : 85%
Generator efficiency :97%
Gear box efficiency : 98%
Power generated : 900 kW
Boiler efficiency : 75%
Bio gas GCV : 7000 kcal/sm^3
Capacity of absorption chiller : 2000 TR
COP of absorption chiller : 0.8
Efficiency of cooling tower pump : 75%
Efficiency of cooling tower pump motor : 93%
Head developed by the pump : 15 meter

	Cooling tower range : 5°C **Calculate the following:** i) Steam flow to the turbine in TPH 4 Marks ii) Steam flow to the chiller in TPH 4 Marks iii) Steam flow to the process in TPH 2 Marks iv) Bio gas input to the boiler in sm³/hr 4 Marks v) Calculate the input power drawn by the cooling tower pump in kW 6 Marks
N-3-S	Solution:

i)	Heat Output of generator	= 900 kW x 860 kcal = 774000/(0.97x0.98x0.85) = 957909 kcal/hr Or = 900 kW/ (0.97x0.98x0.85) =1113.8 kW x 860 = 957909 kcal/hr
	Enthalpy drop across the turbine	= 720-660 = 60 kcal/ kg
	Steam flow to the turbine (m1)	= 957909 kcal/hr / 60 kcal/ kg = 15965 kg/ hr or 15.96 TPH
ii)	Heat required for the chiller	= 2000 x 3024 / (0.8) = 7560000 kcal/hr
	Steam flow to the chiller in TPH	= 7560000 kcal/hr / (660-100) kcal/kg = 13500 kg/h or 13.5 TPH
iii)	Steam flow to the process in TPH	= 15.96 -13.5 = 2.46 TPH
vi)	**Biogas consumption:**	
	When two liquids of different temperature are mixed in a tank, to calculate the final temperature	= ((m1xS1xT1) + (m2xS2xT2))/ (m1xS1 + m1xS1) = (13.5x100 + 2.46 x30)/ (13.5+2.46) = 89.2 °C
	The biogas flow rate,q	= 15.96 x 1000 x (720-89.2) / (0.75 x 7000) = 1917.6 sm³/hr
		(or)
		= (13.5 x1000 (720-100))+ (2.46 x1000x (720-30)) / (0.75 x 7000) = 1917.6 sm³/hr
v)	input power drawn by the cooling tower pump	
	Heat rejected by VAM chiller	= (2000 x 3024) + 13.5 x 1000 x (660-100) = 6048000 + 7560000 = 13608000 kcal/hr
	Pump flow rate (cooling tower range 5°C)	= 13608000 / 5 = 2721600/1000 = 2721.6 m³/hr
	Power drawn by the motor	= ((2721.6 /3600) x15x 9.81))/(0.75x0.93) = 159.5 kW

<table>
<tr><td></td><td>Answer any ONE of the following among four questions given below:</td></tr>
<tr><td>N4
A</td><td>The utility data for the upcoming commercial building is as follows

Total Power required for the whole building including centrifugal chiller load : 1000 kW
Building cooling load : 15,12,000 kCal/hr
Centrifugal chiller power consumption : 0.45 kW/TR
Cost of grid power including demand charges : Rs.10.35 /kWh

The management is considering the following two options for operating the building loads
Option 1: Operating all the loads through grid power
Option 2: Installing an 850 kW natural gas engine generator operating at full load with a WHRB and steam operated absorption chiller to meet part of the cooling load. Details of gas engine cogeneration system are given below:

Total power generated from gas engine co-gen plant : 850 kW
Gas engine efficiency : 38 %
Heat absorbed for steam generation in WHRB (as a % of heat input to gas engine) : 22 %
Specific steam consumption for VAM : 4.2 kg/TR
Calorific value of Natural Gas : 8500 kcal/sm^3
Cost of Natural Gas : Rs.45/sm^3
Total enthalpy of steam : 660 kCal/kg
Feed water temperature to WHRB : 60 ^{0}C

Calculate the following:
i) Cost of generating one unit of electricity from the gas engine? (3 marks)
ii) TR generated from Vapour Absorption Chiller driven by WHRB generated steam? (6 Marks)
iii) Total energy cost per hour for option 1 & option 2. (10 Marks)
iv) Which option is economically viable? (1 mark)</td></tr>
<tr><td>N4
A-S</td><td>i) Cost of generating one unit of electricity from gas engine?
Fuel Consumption = 850 kW X 860 / (0.38 X 8500)
= 226.31 sm^3/hr
Cost per unit of electricity from gas engine = (226.31 sm^3/hr X 45 Rs./ sm^3) / 850 kW
= Rs.11.98/ kWh
ii) TR generated from VAM driven by WHRB generated steam

TR required by the building = 1512000/3024
= 500 TR
Heat absorbed by WHRB for Steam generation = 22% x (226.31 x 8500)
= 4,23,199.7 kcal /hr
Amount of steam generated = 423199.7/(660-60)
= 705.33 kg/hr
TR generated by VAM = 705.33/4.2 = 167.94 TR

iii) Total energy cost per hour for option 1 & option 2.
Cost of electricity from Grid (Option 1) = 1000x 10.35 = 10,350 Rs./hr

Total energy cost (Option 2)</td></tr>
</table>

	Additional TR required = (500-167.94) = 332.06 TR Power required from the grid for the additional TR = 332.06 x 0.45 = 149.43 kW Cost of additional TR required = 149.43x 10.35 = 1546.60 Rs/hr Cost of NG for Electricity = 226.31 sm^3/hr X 45 Rs./ sm^3 = **10,184 Rs./hr** = 10184+1546.60 = **11,730.6 Rs./hr** iv) Which option is economically viable? Option -1 is economically viable.
	Or
N4 B	During an energy audit of 5 stage Pre-heater (PH) kiln cement plant, following data were collected:

S. No.	Description	UoM	Value
1.	Clinker output	Tph	55
2.	Return dust in PH gas (% of Kiln feed)	%	9.4
3.	Reference Temperature	Deg C	0
4.	Reference pressure and the barometric pressure	mmWC	10336
5.	NCV of coal	kcal/kg	5500
6.	Cost of coal	Rs./ MT	9500
7.	Power cost	Rs./kWh	5.5
8.	Kiln annual running hours	Hrs/annum	7200

S.No	Flow measurements		PH exit/fan inlet	Cooler exhaust air
1.	Average dynamic pressure	mmWC	16	17.13
2.	Static pressure	mmWC	-440	-28
3.	Temperature	Deg C	355	365
4.	Density of gas at Ref. temperature and pressure	kg/Nm3	1.4	1.293
5.	Pitot tube constant		0.85	
6.	Fan input power	kW	333	62
7.	Fan efficiency	%	80	78
8.	Fan motor efficiency	%	95	95
9.	Diameter of measuring point	M	2	1.6
10.	Specific heat of PH gas	kcal/kg °C	0.248	0.245

Calculate the following:

i. Heat loss in PH exit gas (kcal/kg clinker) [5 marks]

ii. Heat loss in cooler vent air (kcal/kg clinker) [5 marks]

iii. The recommendations made by energy auditor are as follows:

a) The 5 stage pre-heater may be upgraded to 6 stage, which will result in reduction in PH exit temperature by 50 °C and increase in pressure drop by 150 mm WC. Calculate the net annual monetary energy savings for proposed modification considering other parameters same as pre-modification. [8 marks]

b) To increase the cooler recuperation efficiency by reducing cooler exhaust temperature by 75 °C. Calculate the reduction in heat loss in cooler vent air (kcal/kg clinker) [2 marks]

N4 B-S

Density of Pre-heater gas at PH Fan Inlet at prevailing temp., pressure conditions:

$$\rho_{T,P} = \rho_{STP} \text{ X } \frac{273 \text{ X} (10336 + P_S)}{(273 + T) \text{ X} 10336}$$

$$\rho_{T,P} = 1.40 X \frac{273 X (10336 - 440)}{(273+355) X 10336} = 0.583 \text{ kg/m}^3$$

Velocity of PH gas

$$v = P_t \sqrt{\frac{2 g P_d}{\rho_{T,P}}}$$

$$v = 0.85 \sqrt{\frac{2 X 9.8 X 16}{0.583}} = 19.7 \text{ m/sec}$$

Volumetric flow rate of PH gas = velocity X duct cross-sectional area
= 19.7 X (3.14 x $(1)^2$)
= 61.858 m^3/sec
= 61.858 X 3600
= 222688 m^3/hr

Specific volume of PH gas = 222688.8 X 0.583/1.4
= 92733.97 Nm^3/hr
= 92733.97/55000 = **1.686 Nm^3/kg clinker**

i. Heat loss in PH exit gas

$Q1 = m_{ph}\ c_p\ \Delta T$ (C_p of PH gas = 0.248 kcal/kg ºC)

Q1= 1.686 X 1.4 X 0.248 (355-0)
= **207.81 kcal/kg clinker**

[5 marks]

Similarly density of cooler vent air at cooler vent air fan Inlet at prevailing temp., pressure conditions:

$$\rho_{T,P} = \rho_{STP} X \frac{273 X (10336 + P_S)}{(273+T) X 10336}$$

$$\rho_{T,P} = 1.293 X \frac{273 X (10336 - 28)}{(273+365) X 10336} = 0.5518 \text{ kg/m}^3$$

Velocity of cooler vent air in the fan inlet duct

$$v = P_t \sqrt{\frac{2 g P_d}{\rho_{T,P}}}$$

$$v = 0.85 \sqrt{\frac{2 X 9.8 X 17.13}{0.5518}} = 20.96 \text{ m/sec}$$

Volumetric cooler vent air = velocity X duct cross-sectional area
= velocity x (π x d^2)/4
= 20.96 X (3.14×$(1.6)^2$) /4
= 42.12 m^3/sec
= 42.12 X 3600
= 151636 m^3/hr

Specific volume of cooler vent air = 151636 X 0.5518/1.293
= 64712 Nm^3/hr
= 64712/55000 = **1.176 Nm^3/kg clinker**

ii. Heat loss in cooler vent air

$Q2 = m_{CA}\ c_p\ \Delta T$

Q2= 1.176 x 1.293 x 0.245 x (365-0)
= **136 kcal/kg clinker**

[5 marks]

iii.

a) After up-gradation from 5 stage to 6 stage Pre-heater

Reduction in PH exit gas heat loss

$Q = m_{ph}\ c_p\ \Delta T$ (C_p of PH gas = 0.248 kcal/kg ºC)

Q1= 1.686 X 1.4 X 0.248 X 50
= **29.16 kcal/kg clinker**

	Equivalent coal savings (kg /hr) = $\text{Reduction heat loss } (kcal/kg\ clinker)\ X \dfrac{clinker\ output\ (kg/hr)}{NCV\ coal\ (kcal/kg\ coal)}$ Equivalent coal saving $= 29.16\,X \dfrac{55000}{5500}$ kg coal/hr = 291.6 kg coal/hr Annual coal savings = $291.6\,X \dfrac{7200}{1000}$ = 2099.52 MT Annual monetary savings (Thermal) = 2099.52 X 9500 = Rs. 1,99,45,440 per annum **Increase in PH fan power due to increase in pressure drop by 150 mm WC** **Fan power** = $\dfrac{PH\ gas\ flow\ (m^3/hr)\ X\ Rise\ in\ pressure\ drop\ (mm\ WC)}{102\ X\ fan\ efficiency\ (\%)/100\ X\ motor\ efficiency(\%)/100}$ kW $= \dfrac{61.585\,X\,150}{102\,X\,80/100\,X\,95/100}$ = 119.16 kW **Increase in Annual Electrical energy cost = 119.16 X 7200 X 5.5** **= Rs. 47,18,974 per annum** **Net annual monetary savings = 1,99,45,440 - 47,18,974 = Rs. 1,52,26,466 per annum** b) **Improving cooler efficiency** Reduction in cooler vent air heat loss Q = $m_{cooler\,vent}\ c_p\ T$ (C_p of cooler vent air = 0.245 kcal/kg °C) Q1= 1.176 X 1.293 X 0.245 X 75 = 27.94 kcal/kg clinker
	Or
N4 C	Sponge iron is processed in a steel melting shop for production of ingots. The daily sponge iron production in the steel plant is 300 tons. The plant has a coal fired captive power station to meet the entire power demand of the steel plant. The base year (2020) and current year (2021) energy consumption data are given below:

Parameter	UoM	Base year (2020)	Current Year (2021)
Sponge Iron production	T/day	300	300
Specific coal consumption	T/T	1.3	1.15
Specific power consumption	kWh/T	110	95
Yield	%	85	88
SEC of Steel Melting Shop	kWh/ton	850	830
Captive power station heat rate	kcal/kWh	3300	3100
GCV of Coal	kCal/kg	5000	5200

Calculate the following:

i) Specific energy consumption of the plant in Million Kcals/ tonne of finished product for base year as well as for the current year. **15 Marks**

ii) Reduction in coal consumption per day in current compared to base year for the plant **5 Marks**

N4 C-S

i) Specific energy consumption of the plant For Base Year

Specific energy consumption for sponge iron	= 1300 kg x 5000 + 110 kWh x 3300 = 6.863 million kcal/ Ton of SI
Total energy consumption for sponge iron /day	= 300X 6.863 = 2059 million kcal

Actual production considering 85% yield from sponge iron to ingot conversion	= 300 Tons x 0.85 = 255 Tons/ day
Specific energy consumption for ingot	= 850 kWhx 3300 = 2.81 million kcal/ ton of ingot
Total energy consumption for ingot production per day	= 2.81 X 255 = 716.55 million kcal
Plant specific energy consumption for production of finished product (ingot) during base year	= (2059+716.55) / 255 = 10.89 million kcal/ ton

Specific energy consumption of the plant For Current Year

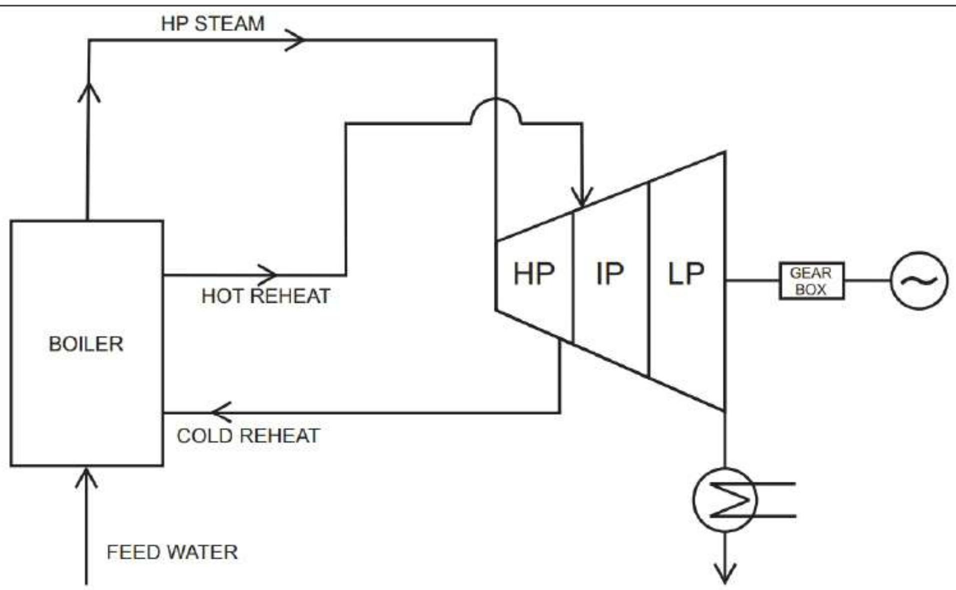
Specific energy consumption for sponge iron	= 1150 kgx 5200 + 95 kWhx 3100 = 6.28 million kcal Ton of SI
Total energy consumption for sponge iron /day	= 300 x 6.28 = 1884 million kcal
Actual production considering 88% yield from sponge iron to ingot conversion	= 300 T X 0.88 = 264 Tons / day
Specific energy consumption for ingot	= 830Kwh x 3100 = 2.573 million kcal/ ton of ingot
Total energy consumption for ingot production per day	2.573 X 264 = 679.27 million kcal
Plant specific energy consumption for production of finished product (ingot) during current year	= (1884+679.27)/264 = 9.70 million kcal/ ton

ii) Reduction in coal consumption

Energy saving in sponge iron plant = (6.863-6.28) x300 = 175 million kcals/day
Energy saving in steel melting plant = (2.81*255-2.57*264) = 38 million kcal/day
Total energy saving = 175 + 38 = 213 million kcal/day
Equivalent coal reduction (saving) = 213 x 10^6/5200 = 40.96 Tons per day

Or

N4 D A thermal power plant is equipped with boiler and reheat steam turbine with the following details



PARAMETER	UNITS	HP TURBINE (INLET)	HP TURBINE (OUTLET/ CRH)	IP TURBINE (INLET/ HRH)	LP TURBINE INLET	CONDENSER
Pressure	**Kg/cm^2**	**145**	**36.7**	**33**	**7**	**0.125**
Temperature	**°C**	**516**	**340**	**516**	**322**	**49**
Enthalpy	**Kcal/kg**	**813**	**735**	**834**	**741**	**610**
Flow	**TPH**	**684**	**635**	**635**	**545**	**545**

LP turbine exhaust dryness fraction : 0.98
Isentropic efficiency of HP turbine : 79.6%
Isentropic enthalpy at LP turbine exhaust pressure : 559 kcal/kg
Boiler feed water temperature : 241 °C
Gear box efficiency : 98%
Generator efficiency : 97%

Calculate:

i) Isentropic enthalpy of HP turbine outlet steam. (4 Marks)

ii) Isentropic efficiency of LP turbine (5 Marks)

iii) Power generated from HP, IP and LP turbine in MW (6 Marks)

iv) Turbine heat rate and station Gross heat rate (kcal/kWh) (3 Marks)

v) If the auxiliary power consumption is 8%, calculate the Net heat rate of the power plant in kcal/kWh. (2 Marks)

N4 D-S

i)	**Isentropic efficiency of HP turbine**	= 79.6 %
	Isentropic enthalpy of HP turbine outlet steam	(813-x) = (813-735)/ 0.796 = 98 X = 813 -98 = 715 kcal/ kg
ii)	Enthalpy of exhaust of LP turbine	= 49+0.98 x (610-49) = 599 kcal/kg
	Isentropic Efficiency of LP turbine	= (741-599)/ (741-559) **= 78 %**
iii)	Power generated from HP, MW	= Mass flow rate x Enthalpy drop across HP turbine/860 = (684x1000x(813-735))/(860x1000)

		= 62.03 MW
	Power generated from IP, MW	= Mass flow rate x Enthalpy drop across IP turbine/860 = (635x1000x(834-741))/(860x1000) = 68.6 MW
	Power generated from LP, MW	= (545x1000x(741-599))/(860x1000) = 89.98 MW
iv)	Gross heat rate (kcal/kWh)	
	Total Power generated by Turbine in MW	= 62.03+68.6+89.98 = 220.67 MW
	Power generated by generator	= 220.67 MW x 0.98 x 0.97 = 209.76 MW
	Turbine heat rate	= (684 x (813-241) + 635*(834-735))/209.76 = 454113/209.76 = 2164.9 kcal/kWh
	Station Gross heat rate	SGHR cannot be calculated since boiler efficiency not given Or Any candidate calculates SGHR, assuming boiler efficiency between 85-88% marks are awarded (Refer guide book-4, page no 175,181 & 193)
v)	Station Net heat rate	Station net heat rate cannot be calculated since station gross heat rate could not be calculated Or Any candidate calculates SNHR by calculating SGHR on assuming boiler efficiency between 85-88% marks are awarded

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