Chapter 6

Exergoeconomic Optimization of Alternative Options

Exergoeconomic optimization of two alternative options available with the fertilizer plant in terms of fuel (steam) source is examined in this chapter. Firstly, the AAVAR system is simulated in combination with 8 MW gas turbine power plant, instead of the independent boiler for steam source, which is also the part of fertilizer company infra-structure. In this case, the steam generated at HRSG is considered as heat source. The system is optimized exergoeconomically and the cost of steam generated at HRSG is estimated for minimum cost of power generation. The optimum cooling cost for AAVAR system is estimated considering steam generated at HRSG as heat source. Section 6.1 deals with the details of the alternative option.

Next alternative option available is tapped steam from the 50 MW steam turbine power plant which is the major source electricity for the fertilizer plant. Section 6.2 describes the exergoeconomic optimization of the AAVAR system using tapped steam from a certain stage of the steam turbine of the plant as heat source. The losses in various components are identified and the cost of steam tapped from the steam turbine is estimated for the minimum cost of power generation by the steam turbine. This tapped steam is utilized as a heat source for AAVAR system and the cost of cooling generated by the system is estimated.

6.1 Steam Generated at HRSG as Heat Source

The existing AAVAR system is equipped with an independent boiler generating saturated steam at 15 bar for the purpose of using it as fuel in the generator. With this arrangement, it is found that the generation cost of steam is 900 ₹/1000 kg. It will be worthwhile to consider other options of steam generation and its utilization as fuel to AAVAR system as some of them are readily available in the fertilizer unit. Keeping this in mind, the present study is carried out to try two additional sources of heat energy available in the plant. The first among them is the partial use of steam generated in the Gas Turbine- Heat Recovery Steam Generator (GT-HRSG) plant as heat source for AAVAR system. It should be noted that GT-HRSG plant acts as a captive power plant catering to the need of power requirement in the fertilizer plant. This section examines the option for the reduction of cost of brine chilling using AAVAR system through exergoeconomic optimization using the steam generated from HRSG partially. It is expected that if the steam generated by the use of waste heat at the GT-HRSG plant as fuel for AAVAR system, there could be significant reduction in the steam cost and hence the cost of cooling.

The following sections give the step by step procedure adopted for the exergoeconomic optimization scheme employed earlier for the existing system. As the details of the scheme are presented earlier, the following section may not repeat the same.

6.1.1 System Simulation

Using the steady state online data, the system simulation is carried out and the missing data are generated. The assumptions underlying the GT-HRSG system model include the following:

- The GT-HRSG system operates at steady state.
- Laws of ideal gas mixture apply for the air and the combustion products.
- The combustion in the combustion chamber is complete.
- Heat loss from the combustion chamber is 2 % of the fuel LHV.

In the GT-HRSG plant model, two types of independent variables are identified, decision variables and parameters. The decision variables are varied in optimization studies, but the parameters remain fixed. All other variables are dependent variables and their values are calculated using the thermodynamic model.

The compressor pressure ratio (p_2 / p_1), isentropic compressor efficiency η_c, effectiveness of air preheater χ_{APH}, isentropic turbine efficiency η_T, temperature of air entering the combustion chamber T_3 and temperature of the combustion product entering the turbine T_4 are considered as decision variables. The dependent variables include the mass flow rates of the air, combustion products and fuel, the power required by the compressor, the power developed by the turbine and pressure and temperature of plant components as follows:

Air compressor	p_2, T_2
Air preheater	p_3, p_6, T_6
Combustion chamber	p_4
Gas turbine	p_5, T_5
HRSG	T_7

Parameters are independent variables whose values are specified. They are kept fixed in optimization study. In this model, the following parameters that are fixed are identified.

- System Products
  - The net power generated by the system is 8 MW
  - Saturated water vapour supplied by the system at $p_9 = 15$ bar
- Air Compressor
  - $T_1 = 298.1\text{K}$, $p_1 = 1.013$ bar
  - Air molecular analysis (%): 77.48 (N_2), 20.59 (O_2), 0.03 (CO_2), 1.90 (H_2O).
- Air Preheater
  - Pressure drop: 3 % on gas side and 5 % on the air side.
- Heat Recovery Steam Generator
  - $T_8 = 298.1\text{K}$, $p_8 = 15$ bar, $p_7 = 1.013$ bar
  - Pressure drop: 5 % on gas side.
- Combustion Chamber
  - $T_{10} = 298.1\text{K}$, $P_{10} = 12$ bar

- Pressure drop: 5 %
- Temperature of combustion product $T_4 = 1520K$

Using the assumptions listed, a standard set of governing equations are available in literature. This involves consideration of several individual control volumes identified with reference to various components of the plant.

Air Compressor

The temperature of the air inlet to compressor, $T_1 = 298.1\text{K}$. At this temperature, the enthalpies of all the constituents, nitrogen, oxygen, carbon dioxide and water vapour are taken from Table F1 of Appendix F while these properties at the temperature other than reference temperature are calculated with the help of Table F2 of Appendix F. Then the enthalpies of all the constituents are added on molar basis and the enthalpy of the air inlet to compressor is calculated on molar basis.

$$h_1' = 0.7748 h_{\mathrm{N_2}}(T_1) + 0.2059 h_{\mathrm{O_2}}(T_1) + 0.0003 h_{\mathrm{CO_2}}(T_1) + 0.019 h_{\mathrm{H_2O}}(T_1) \tag{6.1}$$

The molecular weight of the air inlet to compressor is calculated using

$$M_a = 0.7748 M_{\mathrm{N_2}} + 0.2059 M_{\mathrm{O_2}} + 0.0003 M_{\mathrm{CO_2}} + 0.019 M_{\mathrm{H_2O}} \tag{6.2}$$

Using these values, the enthalpy of air on mass basis is calculated using

$$h_1 = h_1' / M_a \tag{6.3}$$

The temperature at the end of compression is calculated using

$$T_2 = T_1 \left\{ 1 + \frac{1}{\eta_{AC}} \left[\left(\frac{p_2}{p_1} \right)^{\frac{\gamma_a - 1}{\gamma_a}} - 1 \right] \right\} \tag{6.4}$$

Where isentropic efficiency of the compressor $\eta_{AC} = 86\ \%$ and pressure ratio $(p_2 / p_1) = 10$. At this temperature, the enthalpy of the air leaving the compressor (h_2) is calculated following the same procedure as applied for T_1.

Air Preheater

The pressure drop on the air side of the air preheater is considered as 5 % as suggested by Tsatsaronis et al. [114]. The pressure of the air coming out of the air preheater is estimated using

$$p_3 = p_2(1-\Delta p_{a,APH}) \quad \text{with} \quad \Delta p_{a,APH} = 0.05 \tag{6.5}$$

The temperature of the air coming out of the air preheater (T_3) is calculated using the effectiveness of the air preheater:

$$\chi_{APH} = \frac{T_3 - T_2}{T_5 - T_2} \tag{6.6}$$

For the base case, the effectiveness of air preheater is considered as, $\chi_{APH} = 75\ \%$ which will give the temperature of the air (T_3) at the exit of air preheater. The enthalpy of the air coming out of the air preheater (h_3) is calculated following the same procedure as applied for air at temperature T_1.

By energy balance across the air preheater, the temperature of the gas (T_6) leaving from the air preheater is calculated.

$$m_a C_{P,a}(T_3 - T_2) = m_g C_{p,g}(T_5 - T_6) \tag{6.7}$$

The specific heat of air and gas is taken from Tsatsaronis et al [114].

$C_{P,a} = 1.005$ kJ/kgK, $C_{P,g} = 1.17$ kJ/kgK

Combustion Chamber

Denoting the air fuel ratio on molar basis as λ, the molar flow rates of the fuel, air and the combustion product are related by

$$\frac{\dot{n}_f}{\dot{n}_a} = \lambda, \qquad \frac{\dot{n}_P}{\dot{n}_a} = 1 + \lambda, \tag{6.8}$$

Where f, p and a denote fuel, combustion product and air, respectively. For complete combustion of methane the chemical equation takes the form

$$\begin{aligned} &\lambda CH_4 + [0.7748N_2 + 0.2059O_2 + 0.0003CO_2 + 0.019H_2O] \\ &\quad \rightarrow [1+\lambda][x_{N_2}N_2 + x_{O_2}O_2 + x_{CO_2}CO_2 + x_{H_2O}H_2O] \end{aligned} \tag{6.9}$$

Using the temperature of the combustion product from the combustion chamber ($T_4 = 1520K$) and the energy balance across the combustion chamber, air fuel ratio (λ) and enthalpy of combustion product (h_4) are estimated.

$$(1+\lambda)*h_4 = 0.7748*h_{N_2}(T_4)+(0.2059-2\lambda)*h_{O_2}(T_4)+(0.0003+\lambda)*h_{CO_2}(T_4)+ (0.019+2\lambda)*h_{H_2O}(T_4) \quad (6.10)$$

$$-0.02\lambda*\mathrm{LHV}+h_3+\lambda*h_f-(1+\lambda)*h_4=0 \quad (6.11)$$

Where $h_f = -74872$ kJ/kmol and LHV=802361 kJ/kmol and 2 % loss is considered as suggested by Bejan et al. [155]. Solving Eqs. 6.10 and 6.11 for λ and enthalpy of combustion product on molar basis are calculated. Their values are $\lambda = 0.03006$ and $h_4 = 10921$ kJ/kmol. Once the air fuel ratio is calculated, the molar analysis of the product can be decided by balancing mole fractions of carbon, oxygen and nitrogen of the combustion product in Eq. 6.9. Table 6.1 gives the mole fraction of the constituent gases in the combustion products estimated.

$$x_{N_2} = \frac{0.7748}{1+\lambda}, \qquad x_{O_2} = \frac{0.2059-2\lambda}{1+\lambda}$$
$$x_{CO_2} = \frac{0.0003+\lambda}{1+\lambda}, \qquad x_{H_2O} = \frac{0.019+2\lambda}{1+\lambda} \quad (6.12)$$

Table 6.1 Molar Analysis of the Combustion Product

component	N_2	O_2	CO_2	H_2O
Mole fraction	0.7522	0.1415	0.02947	0.07681

Using the mole fractions of the constituents, the molecular weight of the combustion product is calculated as

$$M_P = x_{N_2}*M_{N_2}+x_{O_2}*M_{O_2}+x_{CO_2}*M_{CO_2}+x_{H_2O}*M_{H_2O} \quad (6.13)$$

Enthalpy of combustion product on mass basis (kJ/kg) is calculated using

$$h_4 = h_4^{'}/M_p \quad (6.14)$$

The pressure drop in the combustion chamber is considered as 5% [114] then the pressure of the combustion product is calculated using

$$p_4 = p_3(1-\Delta p_{CC}) \quad \text{with} \quad \Delta p_{CC} = 0.05 \tag{6.15}$$

Gas Turbine

Combustion product from the combustion chamber at temperature T_4 and P_4 enters the gas turbine and expands to the final pressure $p_5 = 1.099$ bar. The temperature of the gas at the exit of the gas turbine (T_5) is calculated using

$$T_5 = T_4\left\{1-\eta_{GT}\left[1-\left(\frac{p_4}{p_a}\right)^{\frac{1-\gamma_g}{\gamma_g}}\right]\right\} \tag{6.16}$$

Where, $\eta_{GT} = 0.86$ for the base case. At T_5, the enthalpy of exhaust gas h_5, is calculated in terms of kJ/kmol using

$$h_5' = x_{N_2}h_{N_2}(T_5) + x_{O_2}h_{O_2}(T_5) + x_{CO_2}h_{CO_2}(T_5) + x_{H_2O}h_{H_2O}(T_5) \tag{6.17}$$

$$h_5 = \frac{h_5'}{M_p} \tag{6.18}$$

Considering the control volume enclosing the compressor and turbine

$$\dot{W}_{CV} = \dot{n}_a(h_1' - h_2') + \dot{n}_p(h_4' - h_5') \tag{6.19}$$

$$\frac{\dot{W}_{CV}}{\dot{n}_a} = (h_1' - h_2') + (1+\lambda)(h_4' - h_5') \tag{6.20}$$

Here all the enthalpies are in kJ/kmol. Converting to a mass rate basis and solving, the mass flow rate of air is

$$\dot{m}_a = \frac{M_a * \dot{W}_{CV}}{(1+\lambda)(h_4' - h_5') + (h_1' - h_2')} \tag{6.21}$$

After calculating the mass flow rate of air, mass flow rate of fuel is found using

$$\dot{m}_f = \lambda\left(\frac{M_f}{M_a}\right)\dot{m}_a \tag{6.22}$$

Then the mass flow rate of gas trough turbine is

$$\dot{m}_g = \dot{m}_a + \dot{m}_f \tag{6.23}$$

The gas leaving from the turbine passes through air preheater and is used for preheating the air going to the combustion chamber. The temperature of the gas leaving from the air preheater (T_6) is calculated by energy balance through Eq. 6.7 and enthalpy at the same temperature is calculated using

$$h_6' = x_{N_2} h_{N_2}(T_6) + x_{O_2} h_{O_2}(T_6) + x_{CO_2} h_{CO_2}(T_6) + x_{H_2O} h_{H_2O}(T_6) \tag{6.24}$$

$$h_6 = \frac{h_6'}{M_p} \tag{6.25}$$

On the gas side of air preheater, pressure drop is considered as 3 % [114], then

$$p_6 = p_5(1 - \Delta p_{g,APH}) \quad \text{with} \quad \Delta p_{g,APH} = 0.03 \tag{6.26}$$

Heat Recovery Steam Generator

The energy of the exhaust gas is utilized in HRSG for steam generation at 15 bar saturated. Owing to the presence of sulphur in natural gas, corrosive sulphuric acid can be formed when the products of combustion are sufficiently cooled. This can be guarded against by maintaining the temperature T_7 above 450 K. By energy balance

$$\dot{m}_s(h_9 - h_8) = \dot{m}_g C_{p,g}(h_6 - h_7) \tag{6.27}$$

Thus, solving Eq. 6.27, it is seen that the steam generation rate in HRSG is $\dot{m}_s = 3.25$ kg/sec which is quite closer to the requirement of steam (fuel) in AAVAR system. Allowing a pressure drop of 5 % in HRSG [114], the pressure of the gas leaving the HRSG is

$$p_7 = p_6(1 - \Delta p_{HRSG}) \text{ with } \Delta p_{HRSG} = 0.05 \tag{6.28}$$

The air inlet to compressor is at T_{ref} and p_{ref} and is considered as ideal gas mixture. The entropy of all the components at temperature T_{ref} and p_{ref} ($s_k^0(T)$) is taken from Table F1 of Appendix F and entropy at other temperature and pressure is calculated with the help of Table F2 of Appendix F. After calculating the entropy of all the component of the air, the entropy of the air inlet to compressor is calculated in terms of kJ/kmol

$$s_1' = 0.7748 s_{N_2}(T_1) + 0.2059 s_{O_2}(T_1) + 0.0003 s_{CO_2}(T_1) + 0.019 s_{H_2O}(T_1) \tag{6.29}$$

$$s_1 = \frac{s_1'}{M_a} \text{ (kJ/kg)} \tag{6.30}$$

The air at stations 2 and 3 is at pressure other than p_{ref}. Then entropy of air at temperature T_2 and P_2 is calculated as

$$s_2' = 0.7748\left[s_{N_2}(T_2) - R\ln\left(\frac{0.7748 * p_2}{p_1}\right)\right] + 0.2059\left[s_{O_2}(T_2) - R\ln\left(\frac{0.2059 * p_2}{p_1}\right)\right] + \\ 0.0003\left[s_{CO_2}(T_2) - R\ln\left(\frac{0.0003 * P_2}{p_1}\right)\right] + 0.019\left[s_{H_2O}(T_2) - R\ln\left(\frac{0.019 * P_2}{p_1}\right)\right] \tag{6.31}$$

$$s_2 = \frac{s_2'}{M_a} \text{ (kJ/kg)} \tag{6.32}$$

Here $p_1 = p_{ref}$. Similarly entropy of air at T_3 and p_3 (s_3) is also calculated.

At station 4, combustion product is considered as ideal gas mixture. The mole fraction of all the components can be estimated using Eq. 6.12. Using these mole fractions, the entropy of combustion product at station 4 is found using following relations in terms of kJ/kmol.

$$s_4' = x_{N_2}\left[s_{N_2}(T_4) - R\ln\left(\frac{x_{N_2} * p_4}{p_1}\right)\right] + x_{O_2}\left[s_{O_2}(T_4) - R\ln\left(\frac{x_{O_2} * p_4}{p_1}\right)\right] + \\ x_{CO_2}\left[s_{CO_2}(T_4) - R\ln\left(\frac{x_{CO_2} * p_4}{p_1}\right)\right] + x_{H_2O}\left[s_{H_2O}(T_4) - R\ln\left(\frac{x_{H_2O} * p_4}{p_1}\right)\right] \tag{6.33}$$

$$s_4 = \frac{s_4'}{M_p} \text{ (kJ/kg)} \tag{6.34}$$

Similarly, entropy at stations 5, 6 and 7 is also calculated.

6.1.2 Exergy Analysis

The exergy of the working substance (streams) in all the components of GT-HRSG system possesses physical and chemical components, physical exergy and chemical exergy are estimated using the procedure given in Sections 6.1.2.1 and 6.1.2.2.

6.1.2.1 Physical Exergy

As discussed in Chapter 4, the physical exergy component is associated with the work obtainable in bringing a matter from its initial state to a state that is in thermal and mechanical equilibrium with the environment. The air inlet to compressor at station 1 is at T_{ref} and p_{ref}. Therefore $h_1 = h_{01}$ and $s_1 = s_{01}$ then from Eq. 4.2, physical exergy at station 1 will be zero. Applying the same Eq. 4.2, the physical exergy at stations 2 and 3 are calculated.

To calculate the exergy of combustion product and exhaust gas from the turbine, it is considered that they are reduced to $T_{ref} = 25^oC$ and $p_{ref} = 1.01325$ bar. At this temperature, some condensation of water will occur and gas phase containing saturated water vapour in equilibrium with saturated liquid water phase. On the basis of 1 kmol of combustion products formed, the gas phase at 25°C would consists of 0.9232 kmol of dry products (0.7522 N_2, 0.1415 O_2, 0.02947 CO_2) plus n_v kmol of water vapour. The partial pressure of water vapour would be equal to the saturation pressure, $p_g(25^oC) = 0.0317$ bar. The amount of water vapour is estimated using

$$p_v = x_v p \tag{6.35}$$

$$\text{i.e., } 0.0317 \text{ bar} = \frac{n_v}{0.9232 + n_v}(1.01325 \text{ bar}) \tag{6.36}$$

Solving Eq. 6.36, $n_v = 0.0298$ kmol. Thus for the case of combustion products given in Table 6.1, the composition of the combustion product at 25°C and 1 atm reads

$\underline{0.7522\ N_2,\ 0.1415\ O_2,\ 0.02947\ CO_2\ ,0.02982\ H_2O\ (g)}$, 0.04699$H_2O$ (l)

Where, the underline identifies the gas phase. Using these values, enthalpy of combustion product at 25°C and 1 atm as

$$h_{04}^{'} = 0.7522N_2 + 0.1415O_2 + 0.02947CO_2 + 0.02982H_2O(g) + 0.04699H_2O(l) \tag{6.37}$$

$$h_{04} = \frac{h_{04}^{'}}{M_p} \text{ kJ/kg} \tag{6.38}$$

The physical exergy at station 4 of Fig.3.3 is calculated using

$$\dot{E}_4^{PH} = \dot{m}_4\left[\left(h_4 - h_{04}\right) - T_0(s_4 - s_{04})\right] \tag{6.39}$$

Using the same value of enthalpy at reference condition, physical exergy at stations 5, 6 and 7 are calculated. The feed water inlet to HRSG at station 8 is maintained at 15 bar and considered at reference temperature. At given temperature and pressure, its enthalpy and entropy is calculated using inbuilt subroutine of EES software. Similarly the enthalpy and entropy of feed water to HRSG at reference temperature and pressure are calculated using inbuilt subroutine of EES software. Using enthalpy and entropy at actual condition and at reference condition, the exergy of feed water to HRSG is calculated using Eq.6.39. At station 9, steam is generated at 15 bar saturated. The enthalpy and entropy of steam at 15 bar saturated and at reference condition are calculated using inbuilt subroutine of EES software and using Eq. 6.39, its physical exergy is calculated. Natural gas, maintained at 10 bar pressure, is used as a fuel in the combustion chamber of gas turbine. The physical exergy of fuel at station 10 is calculated as

$$\dot{E}_{10}^{PH} = \dot{m}_{10}\left[h_{10} - h_0 - T_0\left(s_{10} - s_0\right)\right]$$

Since $T_{10} = T_0$, the above equation reduces to

$$\dot{E}_{10}^{PH} = \dot{m}_{10}\, RT_0 \ln\frac{p_{10}}{p_0} \tag{6.40}$$

6.1.2.2 Chemical Exergy

At stations 1, 2 and 3, air is stable with environment so its chemical exergy is considered as zero. At dead state corresponding to the mixture at station 4 consists of liquid water phase and a gas phase. The new mole fraction of a gas phase is calculated using Eq.6.41 and found as

$$y_{N_2} = \frac{x_{N_2}}{x_{N_2} + x_{O_2} + x_{CO_2} + n_v} \qquad y_{O_2} = \frac{x_{O_2}}{x_{N_2} + x_{O_2} + x_{CO_2} + n_v}$$

$$y_{CO_2} = \frac{x_{CO_2}}{x_{N_2} + x_{O_2} + x_{CO_2} + n_v} \qquad y_{H_2O(g)} = \frac{n_v}{x_{N_2} + x_{O_2} + x_{CO_2} + n_v} \tag{6.41}$$

$y_{N_2} = 0.7893$, $y_{O_2} = 0.1485$, $y_{CO_2} = 0.03093$, $y_{H_2O(g)} = 0.03129$.

Now the chemical exergy for the k[th] component is calculated and added together to find total chemical exergy using following equation

$$e^{CH} = \sum y_k e_k^{CH} + RT_0 \sum y_k \ln y_k \tag{6.42}$$

This is the chemical exergy of gas portion. The chemical exergy of liquid portion is separately calculated and added together to find total chemical exergy. The chemical exergy of individual component (e_k^{CH}) is given in Appendix G.

$$\dot{E}^{CH} = \dot{m}_g\left[\left(x_{N_2} + x_{O_2} + x_{CO_2} + n_v\right)e^{CH} + x_{H_2O(l)} * e^{CH}_{H_2O(l)}\right] \qquad (6.43)$$

The chemical exergy calculated using Eq. 6.43 is expressed in kJ/kmol which can then be converted in to kJ/kg by dividing it by molecular mass of the combustion product. The chemical exergy at stations 5, 6 and 7 will remain the same. Chemical exergy of water, steam and natural gas (CH_4) at stations 8, 9 and 10, respectively, are taken from of Appendix G. The total exergy flow at stations 4 to 10 will be the sum of physical and chemical exergy. Table 6.2 gives the properties and parameters at each stations of the GT-HRSG plant along with the estimated values of exergy.

Table 6.2 State Properties for Gas Turbine Power Plant

Stations	Mass flow rate kg/sec	Pres-Sure. bar	Temp. K	Specific Enthalpy kJ/kg	Specific Entropy kJ/kgK	Physical Exergy kW	Chemical Exergy kW	Total Exergy MW
1	24.09	1.013	298.10	-164.60	6.7860	0	0	0
2	24.09	10.13	621.30	179.90	7.0700	6257	0	6.257
3	24.09	9.63	894.50	488.20	7.4960	10627	0	10.63
4	24.49	9.15	1520	386.30	8.3370	25177	83.99	25.26
5	24.49	1.099	985.50	-279.10	8.4210	8260	83.99	8.344
6	24.49	1.066	754.70	-554.80	8.1120	3768	83.99	3.852
7	24.49	1.013	450	-904.30	7.5350	915.50	83.99	0.9995
8	3.25	15	298.10	106.10	0.3666	4.56	8.13	0.01269
9	3.25	15	471.50	2791	6.4440	2845	8.13	2.853
10	0.41	12	298.10	---	---	155.3	20896	21.05

6.1.2.3 Definition of Fuel, Product and Loss for Various Processes

Using the exergy estimated at each station as given in Table 6.2, fuel, product and loss are to be calculated for all the components in a similar manner as that used in Chapter 5, Section 5.1.2. The step by step procedure adopted is given in Section 5.1.2.1 of Chapter 5. It should be recollected that GT-HRSG is one option to act as the source of heat energy (fuel) for AAVAR system. As such there are six components for the GT-HRSG plant for which fuel, product and loss are estimated as per the requirement of exergoeconomic analysis. Table 6.3 summarises the same and its numerical values are given in Table 6.4.

Table 6.3 Component-wise Fuel, Product and Loss of GT-HRSG Power Plant

Component	Fuel ($\dot{E}_F$)	Product ($\dot{E}_P$)	Loss ($\dot{E}_L$)
AC	$\dot{E}^{W}_{11}$	$\dot{E}_2-\dot{E}_1$	0
APH	$\dot{E}_5-\dot{E}_6$	$\dot{E}_3-\dot{E}_2$	0
CC	$\dot{E}_3+\dot{E}_{10}$	$\dot{E}_4$	0
GT	$\dot{E}_4-\dot{E}_5$	$\dot{E}_{11}+\dot{E}_{12}$	0
HRSG	$\dot{E}_6-\dot{E}_7$	$\dot{E}_9-\dot{E}_8$	0
System	$\dot{E}_{10}$	$(\dot{E}_9-\dot{E}_8)+\dot{E}_{12}$	$\dot{E}_7$

Table 6.4 Exergy Analysis Result for Gas Power Plant

Component	$\dot{E}_F$ kW	$\dot{E}_P$ kW	$\dot{E}_L$ kW	$\dot{E}_D$ kW	Y_D %	Y_L %	Y_D^* %	ε %
AC	8298	6257	0	2041	9.693	0	22.15	75.41
APH	4491	4369	0	122.3	0.5808	0	1.327	97.28
CC	31678	25261	0	6417	30.48	0	69.67	79.74
GT	16917	16299	0	618.4	2.938	0	6.713	96.34
HRSG	2853	2840	0	12.57	0.0597	0	0.1364	99.56
System	21052	10841	999.50	9211	43.76	4.75	100	51.50

6.1.2.4 Results and Discussions

The outcome of the exergy analysis of the gas turbine power plant is given in Table 6.4. The total exergy supplied to the system is 21052 kW, out of which 10841 kW (51.50 %) is converted to useful product, 9211 kW (43.76 %) exergy is destroyed and 999.50 kW (7.75%) is lost to the environment.

The maximum exergy destruction is found in combustion chamber. To reduce the exergy destruction in combustion chamber, inlet temperature T_3 should be increased. It can be achieved by increasing the air preheater's effectiveness or compression ratio of the compressor. The effectiveness of APH is already high, the compression ratio of the compressor should be increased or isentropic efficiency of the compressor should be improved.

The next highest exergy destruction is in air compressor which can be reduced by improving its isentropic efficiency. The next is the gas turbine in this category. The exergy destruction in the combustion chamber and gas turbine can be reduced by increasing the inlet temperature T_4 and improving the isentropic efficiency but at the same time the investment cost of gas turbine and combustion chamber will increase which can increase the product cost. So the optimum temperature should be selected. During implementation of all these improvements, optimum condition should be considered.

6.1.3 Exergoeconomic Analysis

The essence of the economic analysis is the identification and inclusion of various cost heads incurred in the estimation of the total cost for the production. In the present case, the total cost involved in the power generation by the gas turbine consists of many cost heads. Thus, in general, the economic analysis of the system requires the estimation of levelized O&M cost of component ($\dot{Z}_k$) and fuel cost rate ($\dot{C}_f$). $\dot{Z}_k$ should be estimated for each component for GT-HRSG plant using TCI, β, γ and τ (Refer Eq.4.18). The fuel cost rate ($\dot{C}_f$) is governed by the source of heat energy used for the system. The estimation of $\dot{Z}_k$ and $\dot{C}_f$ are explained in the following section.

6.1.3.1 Levelized O&M cost ($\dot{Z}_k$)

The purchase equipment costs of each component are calculated using the cost model of each component given in Appendix H for the year 1994. Using the M&S cost index, they are converted for the year 2009. Using Table 4.1, the TCI related to each component is found. The levelized O&M cost of each component is found using Eq. 4.18. Considering the plant life of 8000 hours, Capital Recovery Factor (β) = 0.1061, O&M cost (γ) = 1.092 % of total investment cost of the component [114], the values of levelized O&M cost for each components are determined. The estimated values of Levelized O&M cost for all the components of GT-HRSG are given in Table 6.5. It can be seen that for each components of the GT-HRSG plant, a number of cost heads are involved in the

Table 6.5 Estimation of Levelized O & M Cost for the Components of GT-HRSG

Component	PEC (₹) (1)	Installation Cost 45 % of PEC (2)	Piping Cost 66 % of PEC (3)	Instru.& Control cost 20 % of PEC (4)	Electrical Equipment 11 % of PEC (5)	On Site Cost (ONSC) 1+2+3+4+5	Land 10 % of PEC (6)	Civil Work 60 % of PEC (7)	Service 65 % of PEC (8)	Of Site Cost (OFSC) (6+7+8)	Direct Cost (DC) ONSC + OFSC
AC	49292450	22181603	32533017	9858490	5422170	119287730	4929245	29575470	32040093	66544808	185832538
APH	428100	192645	282546	85620	47091	1036002	42810	256860	278265	577935	1613937
CC	4454150	2004368	2939739	890830	489957	10779044	445415	2672490	2895198	6013103	16792147
GT	49275800	22174110	32522028	9855160	5420338	119247436	4927580	29565480	32029270	66522330	185769766
HRSG	31017100	13957695	20471286	6203420	3411881	75061382	3101710	18610260	20161115	41873085	116934467

Continue Table 6.5

Engg. & Supervision 30% of PEC (9)	Construction Cost 15 % of DC (10)	Contingency 20 % of FCI (11)	Indirect Cost (IC) (9+10+11)	Fixed Capital Investment (FCI) (DC+IC)	Startup Cost 10% of FCI (12)	Working Capital 15 % of TCI (13)	Allowance For Funds 10% of PEC (14)	Other Outlays (12+13+14)	TCI (FCI+Other Outlays)	Levelized O&M Cost ₹/hr ($\dot{Z}_k$)
14787735	27874880	31158430	73821045	259653583	25965358	51273209	4929245	82167812	341821395	5000
128430	242091	270608	641129	2255065	225506	445303	42810	713619	2968684	43.42
1336245	2518822	2815529	6670596	23462743	2346274	4633135	445415	7424824	30887567	451.80
14782740	27865465	31147905	73796110	259565876	25956588	51255890	4927580	82140058	341705934	4998
9305130	17540170	19606332	46451632	163386099	16338610	32263486	3101710	51703806	215089905	3146

estimation of TCI. TCI consists of FCI and Other Outlays. DC and IC constitute FCI, while Other Outlays consists of start up cost, working capital cost and allowance for funds. DC consists of on-site (ONSC) and off-site (OFSC) costs while IC consists of engineering & supervision, construction and contingency costs.

6.1.3.2 Fuel Cost

The plant uses natural gas (methane) as a fuel. The market prize of methane for the year 2009 is considered as 4.3 \$/mm BTU (1mm BTU = 1055.06 MJ). If the LHV of methane is considered as 50 MJ/kg, then the cost of fuel will be 0.2 \$/kg or 10 ₹/kg.

6.1.3.3 Cost Flow

To calculate the exergy cost flow at each station of the gas power plant, the cost balance equations are modelled as explained below.

Air Compressor

$$\dot{c}_1 \dot{E}_1 - \dot{c}_2 \dot{E}_2 + \dot{c}_{11} \dot{E}_{11} + \dot{Z}_{AC} = 0 \qquad (6.44)$$

Air Preheater

$$\dot{c}_2 \dot{E}_2 - \dot{c}_3 \dot{E}_3 + \dot{c}_5 \dot{E}_5 - \dot{c}_6 \dot{E}_6 + \dot{Z}_{APH} = 0 \qquad (6.45)$$

$$\dot{c}_5 = \dot{c}_6 \qquad (6.46)$$

Combustion Chamber

$$\dot{c}_3 \dot{E}_3 - \dot{c}_4 \dot{E}_4 + \dot{C}_f + \dot{Z}_{CC} = 0 \qquad (6.47)$$

Gas Turbine

$$\dot{c}_4 \dot{E}_4 - \dot{c}_5 \dot{E}_5 - \dot{c}_{11} \dot{E}_{11} - \dot{c}_{12} \dot{E}_{12} + \dot{Z}_{GT} = 0 \qquad (6.48)$$

$$\dot{c}_4 = \dot{c}_5 \qquad (6.49)$$

$$\dot{c}_{11} = \dot{c}_{12} \qquad (6.50)$$

Heat Recovery Steam generator

$$\dot{c}_6 \dot{E}_6 - \dot{c}_7 \dot{E}_7 + \dot{c}_8 \dot{E}_8 - \dot{c}_9 \dot{E}_9 + \dot{Z}_{HRSG} = 0 \tag{6.51}$$

$$\dot{c}_6 = \dot{c}_7 \tag{6.52}$$

Out of these variables, $c_1 ... c_9, c_{11}, c_{12}$ and C_f, the last is known which is the cost of fuel in combustion chamber. The cost of air at compressor inlet and cost of water at inlet to HRSG ($\dot{c}_1$ and c_8) are considered as zero. The remaining 9 are calculated by solving Eqs. 6.44 to 6.52 using EES software. The cost per unit exergy (₹/MJ) and cost flow rate (₹/sec) for each flow of the system are calculated and shown in Table 6.6. For this calculation, known values of $\dot{E}_1$ to $\dot{E}_{12}$ are used.

Table 6.6 Unit Exergy Cost and Cost Flow Rate for Gas Power Plant

Flows	Unit exergy cost ₹/MJ	Exergy flow MW	Cost flow rate ₹/sec
1	0	0	0
2	1.1850	6.2570	7.414
3	0.9598	10.6300	10.200
4	0.6175	25.2600	15.600
5	0.6175	8.3440	5.152
6	0.6175	3.8520	2.379
7	0.6175	0.9995	0.617
8	0	0.0127	0
9	0.9238	2.8530	2.635
10	0.2000	21.0500	4.210
11	0.7261	8.2980	6.025
12	0.7261	8.0000	5.809

6.1.4 Exergoeconomic Evaluation

After calculating the cost rates at each station of the plant using cost rate of fuel ($\dot{C}_{F,k}$) as an input the cost rate of product ($\dot{C}_{P,k}$), cost rate of fuel per unit exergy ($c_{F,k}$), cost rate of product per unit exergy ($c_{P,k}$), cost rate of exergy destruction ($\dot{C}_{D,k}$), cost rate of exergy loss ($\dot{C}_{L,k}$), the relative cost difference (r_k) and exergoeconomic factor (f_k) for each component are calculated using Eq. 4.20 to Eq. 4.27 and given in Table 6.7. Based on these results, the system is exergoeconomically evaluated following the methodology suggested by Bejan et al. [155] and discussed in Section 4.2.3.

Table 6.7 Results of Exergoeconomic Analysis for Gas Power Plant

Component	$c_{F,k}$ ₹/MJ	$c_{p,k}$ ₹/MJ	$\dot{C}_{D,k}$ ₹/hr	$\dot{C}_{L,k}$ ₹/hr	$\dot{Z}_k$ ₹/hr	f_k %	r_k %	ε %
AC	0.7261	1.1850	5334	0	5000	48.38	63.18	75.41
APH	0.6175	0.6375	271.8	0	43.42	13.78	.3.25	97.28
CC	0.4549	0.6175	10509	0	451.80	4.12	35.74	79.74
GT	0.6175	0.7261	1375	0	4998	78.43	17.60	96.34
HRSG	0.6175	0.9279	27.94	0	3146	99.12	50.28	99.56
System	0.2000	0.7790	6632	719.70	13640	64.98	289.50	51.50

6.1.4.1 Results and Discussions

The following observations are made from the results shown in Table 6.7.

(i) The r value for the compressor is highest among all the components, indicates that, for this design configuration, particular attention should be paid to air compressor. The air compressor has the lowest exergetic efficiency and second largest rate of exergy destruction cost. Therefore, it would be cost effective to reduce the exergy destruction in compressor by increasing the isentropic efficiency. By using multi stage compressor and providing the Intercooling between the stages, the power consumption by the compressor can be reduced.

(ii) The HRSG has the second highest r value but having high exergetic efficiency and low rate of cost of exergy destruction. So this component is working properly.

(iii) The combustion chamber has the next highest r value. This is due to the very high exergy destruction costs and extremely low f value. The logical conclusion would be to try to decrease the exergy destruction in the combustion chamber by increasing the air preheating temperature T_3.

6.1.5. Exergoeconomic Optimization

The exergoeconomic optimization of the system requires a thermodynamic model and a cost model. The thermodynamic model gives the performance prediction of the system with respect to some thermodynamic variables such as exergy destruction, exergy loss and exergetic efficiency. The cost model permits detailed calculation of cost values for a given set of the thermodynamic variables. For each component, it is expected that the investment cost increases with increasing capacity and increasing exergetic efficiency.

6.1.5.1 Estimation of B_k, n_k and m_k

Using the value of cost flow at each station and the results of exergoeconomic evaluation, the exergoeconomic optimization of the system is carried out at component level using Eq. 4.29. To solve this equation for local optimum by curve fitting technique, the equivalent power law is found and the required value of B_k and n_k for each component are determined for the selected value of m_k as explained below.

Air Compressor

For Air Compressor, efficiency of the compressor (η_{AC}) and compression ratio (r_c) are considered as decision variables. For the variation of isentropic efficiency of air compressor from 0.85 to 0.89, the necessary data are generated as explained in Chapter 4, Section 4.3. For the generation of data, value of $m_{AC} = 0.95$ as suggested by Bejan et al. [155] is taken. Table 6.8 gives the generated data for carrying out regression fit to obtain B_{AC} and n_{AC}.

Table 6.8 Generated Data Using Investment Cost Equation for Air Compressor

η_{AC}	r_c	$\dot{E}_{P,AC}$	$\dot{E}_{D,AC}$	$\dot{E}_{P,AC}/\dot{E}_{D,AC}$	$TCI_{AC}/\dot{E}_{P,AC}^{0.95}$
0.85	8	5.266	1.998	2.635	117239
0.86	9	5.575	1.936	2.88	159992
0.87	10	5.853	1.879	3.114	231363
0.88	11	6.102	1.827	3.339	374238
0.89	12	6.328	1.778	3.559	803096

From Fig. 6.1, the values of B_{AC} and n_{AC} for the selected value of m_{AC} of 0.95 are found to be 251.88 and 6.17, respectively.

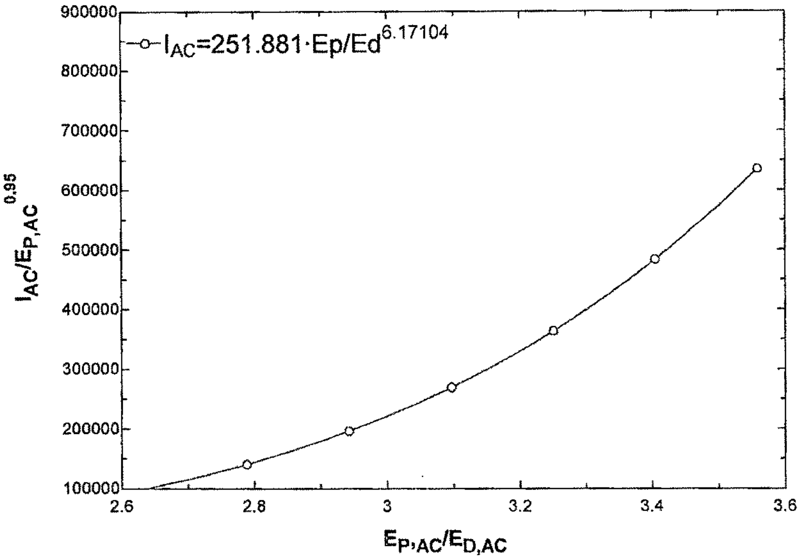

Fig. 6.1 Plot of Investment cost v/s Exergetic Efficiency for Air Compressor

Air Preheater

Air preheater is a device used to recover waste heat of exhaust gas from the gas turbine. By varying the effectiveness of air preheater, the amount of heat recovered can be varied so the effectiveness of air preheater is considered as decision variable. For the range of values of effectiveness from 0.70 to 0.79, data related to exergy of product and destruction are generated and is given in Table 6.9.

Table 6.9 Generated Data Using Investment Cost Equation for Air Preheater

χ_{APH}	$\dot{E}_{P,APH}$	$\dot{E}_{D,APH}$	$\dot{E}_{P,APH}/\dot{E}_{D,APH}$	$TCI_{APH}/\dot{E}_{p,APH}^{0.6}$
0.7	3.97	0.2067	19.18	3223
0.71	4.03	0.1938	20.81	3271
0.72	4.10	0.1805	22.73	3321
0.73	4.17	0.1669	25.00	3373
0.74	4.24	0.1528	27.75	3428
0.75	4.31	0.1383	31.15	3486
0.76	4.38	0.1235	35.46	3548
0.77	4.45	0.1083	41.10	3612
0.78	4.52	0.0927	48.78	3681
0.79	4.59	0.0766	59.89	3754

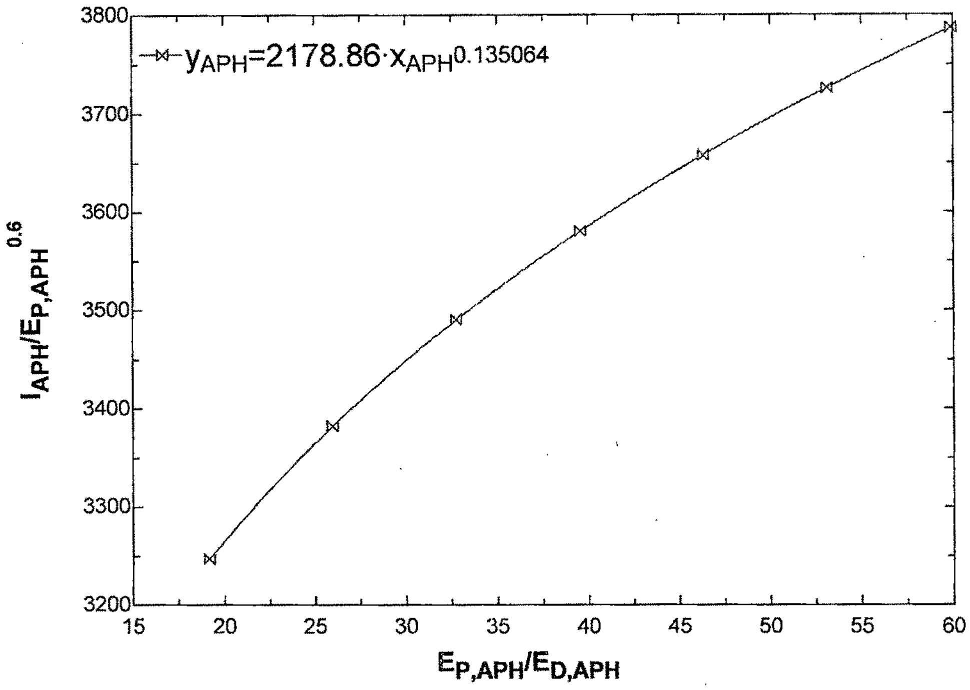

Fig. 6.2: Plot of Investment cost v/s Exergetic Efficiency for Air Preheater

For air preheater with effectiveness (χ_{APH}) of it as decision variable, the Fig. 6.2 shows that the value of B_{APH} and n_{APH} are found to be 2178.86 and 0.135, respectively for the selected value of m_{APH} of 0.6

Table 6.10 Generated Data Using Investment Cost Equation for Combustion Chamber

T_4	$\dot{E}_{P,CC}$	$\dot{E}_{D,CC}$	$\dot{E}_{P,CC}/\dot{E}_{D,CC}$	$TCI_{CC\pi}/\dot{E}_{p,CC}$
1500	24.97	6.02	4.19	2776
1501	24.95	6.03	4.14	2792
1501	24.93	6.03	4.13	2808
1502	24.91	6.04	4.12	2824
1502	24.90	6.05	4.12	2841
1503	24.88	6.06	4.11	2857
1503	24.86	6.06	4.10	2874
1504	24.84	6.07	4.09	2891
1504	24.83	6.08	4.09	2908
1505	24.81	6.09	4.08	2925

Combustion Chamber

For combustion chamber, the temperature of combustion product (T_4) is considered as decision variable. For the variation in T_4 from 1500 K to 1505 K, the required data are generated and given in Table 6.10.

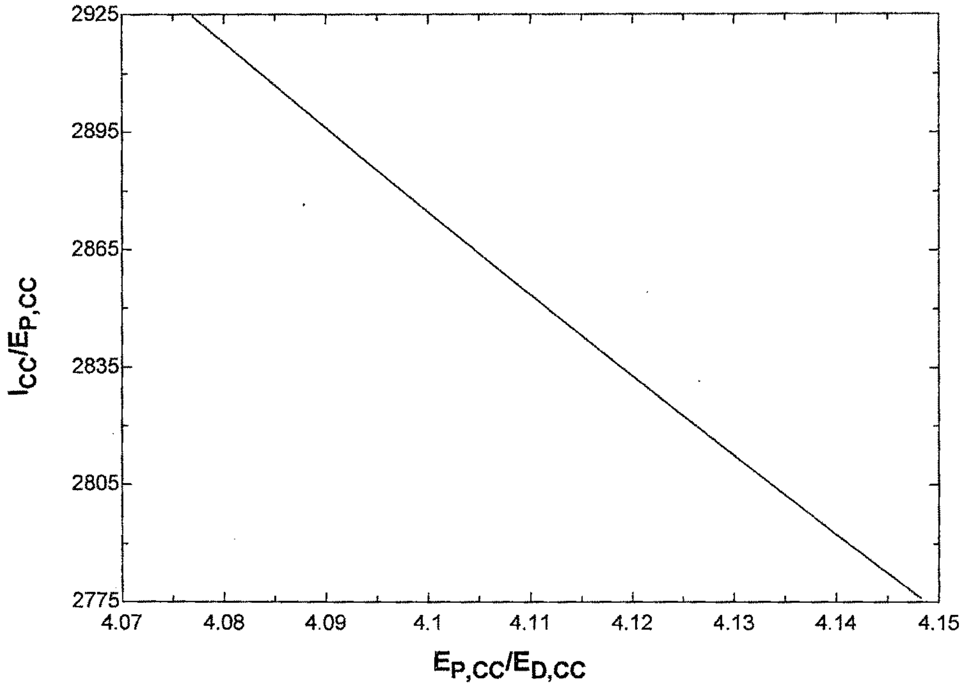

Fig. 6.3 Plot of Investment cost v/s Exergetic Efficiency for Combustion Chamber

For combustion chamber, the slope of the graph is negative so the value of n_{CC} can not be defined. So it is assumed as unity as suggested by Bejan et al. [155]. Fig. 6.3 shows that the value of B_{CC} and n_{CC} are found to be 1001 and 1, respectively for the selected value of m_{CC} of 1.

Gas Turbine

For gas turbine, isentropic efficiency (η_{GT}) and temperature of combustion product (T_4) are considered as decision variable. The generated data are shown in Table 6.11. The graph, given in Fig. 6.4 shows that the value of B_{GT} and n_{GT} are found to be 404.59 and 1.828 for the selected value of m_{GT} of 0.65.

Table 6.11 Generated Data Using Investment Cost Equation for Gas Turbine

η_{GT}	T_4	$\dot{E}_{P,GT}$	$\dot{E}_{D,GT}$	$\dot{E}_{P,GT}/\dot{E}_{D,GT}$	$TCI_{GT}/\dot{E}_{p,GT}^{0.65}$
0.84	1515	16.79	0.776	21.63	113159
0.85	1517	16.54	0.696	23.77	132019
0.86	1519	16.31	0.6198	26.32	157485
0.87	1521	16.09	0.5473	29.4	193539
0.88	1523	15.88	0.4781	33.22	248144

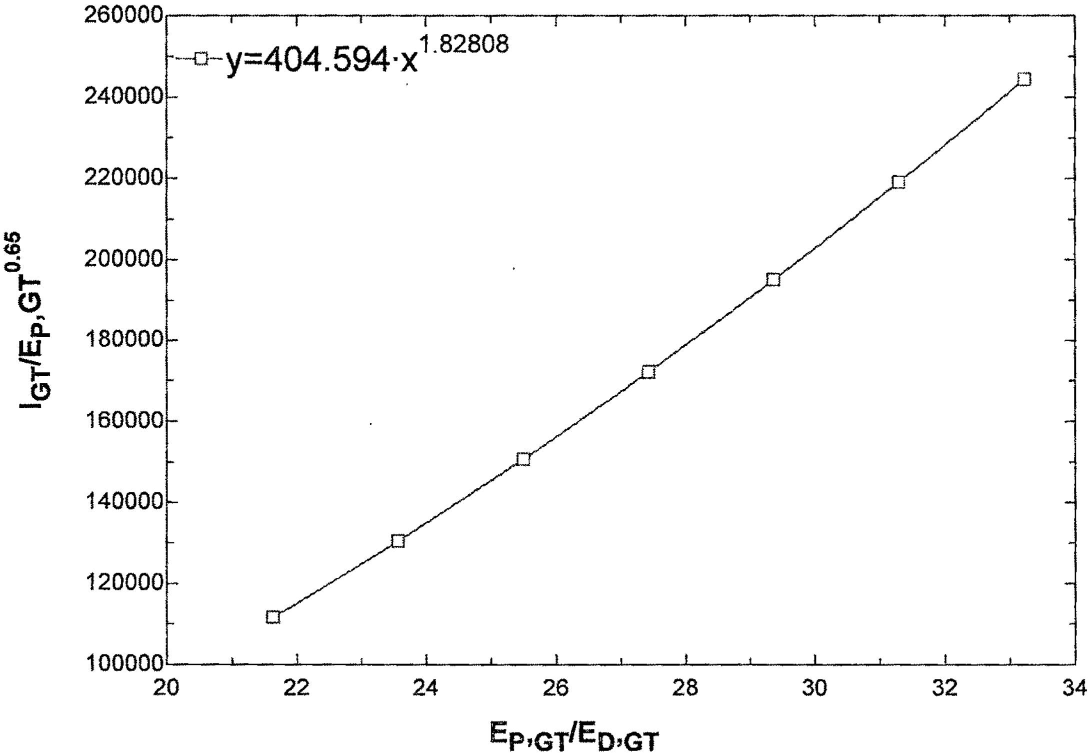

Fig.6.4 Plot of Investment Cost v/s Exergetic Efficiency for Gas Turbine

Heat Recovery Steam Generator

For HRSG, temperature of the exhaust gas coming out of the turbine (T_6) is considered as decision variable. The data generated for the variation in T_6 are given in Table 6.12. The Fig. 6.5 shows that the value of B_{HRSG} and n_{HRSG} are found to be 245553 and 0.0077, respectively for the selected value of m_{HRSG} of 0.85.

Table 6.12 Generated data through investment cost equation for HRSG

T_6	$\dot{E}_{P,HRSG}$	$\dot{E}_{D,HRSG}$	$\dot{E}_{P,HRSG}/\dot{E}_{D,HRSG}$	$TCI_{HRSG}/\dot{E}_{p,HRSG}^{0.85}$
768.2	3.46	0.1363	25.39	251105
766.0	3.34	0.1127	29.66	251791
763.7	3.23	0.0905	35.73	252491
761.5	3.13	0.0694	45.06	253206
759.2	3.03	0.0494	61.21	253934
757.0	2.93	0.0305	96.04	254677
754.7	2.84	0.0126	226.20	255433

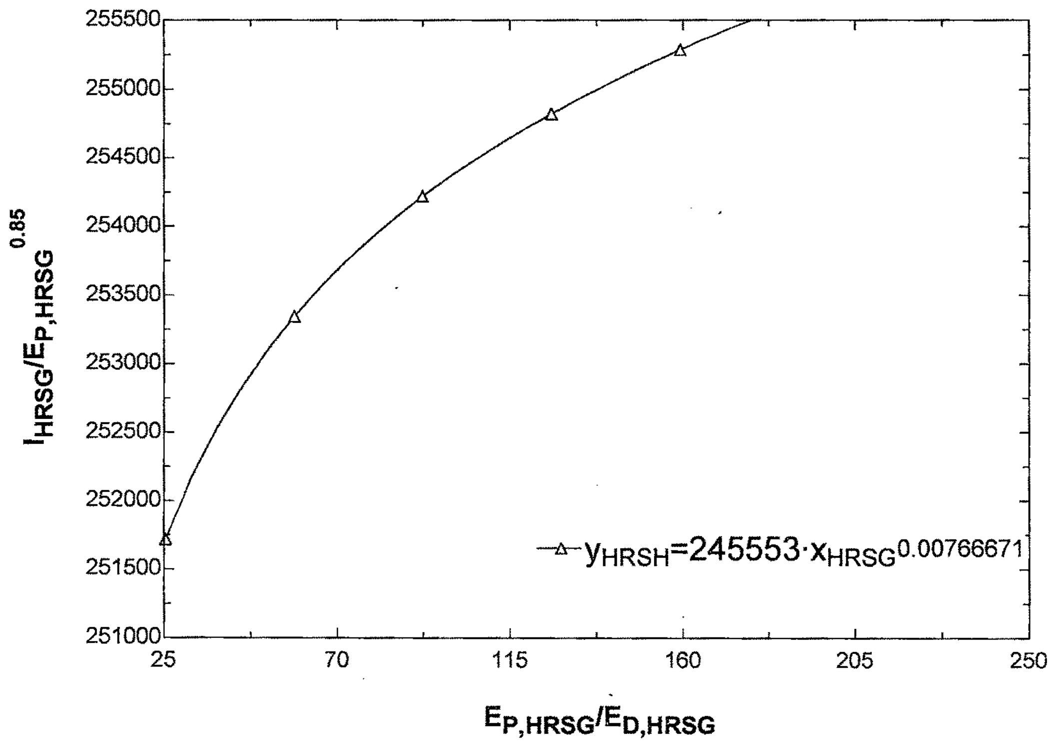

Fig. 6.5: Plot of Investment cost v/s Exergetic Efficiency for HRSG

Table 6.13 summarises the component-wise parameters, B_k, n_k and m_k estimated along with the decision variable.

Table 6.13 Values of B_k, n_k and m_k for various components

Component	Decision variable	B_k	n_k	m_k
AC	η_{AC} & r_c	251.88	6.17	0.95
APH	χ_{APH}	2178.86	0.135	0.6
CC	T_4	1001	1	1
GT	η_{GT} & T_4	404.59	1.828	0.65
HRSG	T_6	245553	0.0077	0.85

6.1.5.2 Optimisation Through Case by Case Iterative Procedure

Optimum values of exergetic efficiency (ε_k^{OPT}), the capital investment (Z_k^{OPT}), the relative cost difference (r_k^{OPT}) and the exergoeconomic factor (f_k^{OPT}) can be calculated using Eqs. 4.37, 4.45, 4.46 and 4.47, respectively. Through an iterative optimization procedure, optimum solution can be achieved, with the help of calculated values of $\dot{C}_{P,tot}, \dot{C}_{D,tot}, \dot{C}_{L,tot}$ and OBF and the guidance provided by the values of $\Delta\varepsilon_k$ and Δr_k, calculated using Eqs. 4.50 and 4.51.

Table 6.14 summarizes the results obtained from the case-by-case iteration carried out starting from the base case (base case is the case evaluated using the data of the existing system) to the optimum case. A total of seven iterative cases are presented and the resulting cases are given as cases I to VII out of which the case IV is found to be the optimum. Each of these cases is obtained through a series of study of positive or negative effects on $\dot{C}_{p,tot}$ and $\dot{C}_{D+L}$ by varying each decision variable. The change in the decision variables are governed by $\Delta\varepsilon_k$ and Δr_k. The details of the case by case iterative procedure for exergoeconomic optimization of AAVAR system is discussed in the following paragraph and the output given in Table 6.14. In the base case, the unit product cost of electricity is 2.61 ₹/kWh and production cost of steam is 810 ₹/1000 kg and total generation of steam is 3.25 kg/sec.

From base case to case-I:

The highest value of Δr_{APH} shows that the product cost of air preheater is very high. It also suggests that the effectiveness of air preheater should be increased. The effectiveness of APH is increased from 0.75 to 0.8. With this, the cost of electricity is reduced to 2.51 ₹/kWh and cost of steam is 830 ₹/1000 kg. But the major heat is recovered from the exhaust gas; the rate of steam generation is reduced to 2.9 kg/sec which is not sufficient for absorption refrigeration system. Therefore this parametric variation is kept pending for later stages.

From case-I to case-II:

The next highest Δr_{HRSG} suggest that the exergy destruction in HRSG can be reduced by decreasing T_6. It can be achieved by increasing the effectiveness of APH which is already checked in the previous iteration. By reducing the air compressor pressure ratio, the heat recovery at APH can be increased and temperature T_6 can be reduced. The air compressor pressure ratio is reduced from 10 to 9. With this, Δr_{HRSG} is reduced but not sufficient reduction in the product cost is achieved. In this condition, the rate of steam generation is 3.15 kg/sec and the cost of steam is 840 ₹/1000 kg.

From case-II to case-III

The next highest Δr_{CC} value suggests that the exergy destruction in the combustion chamber should be reduced. The highest exergy destruction cost can be observed in the combustion chamber in Table 6.6 also. The exergy destruction in combustion chamber can be reduced by increasing the temperature T_3 which can be achieved by increasing the compressor efficiency up to 0.87. But the compressor investment and maintenance cost is so sensitive with compressor efficiency. The increase in compressor efficiency results in increase in the Δr_{AC} and therefore, increase in the product cost.

From case-III to case-IV

The decrease in the compressor efficiency up to 0.85 will give reduction in the product cost. The cost of electricity generation will be 2.56 ₹/kWh and that of steam will be 810 ₹/1000 kg with steam generation rate 3.2 kg/sec.

From case-IV to case-V

The next highest production cost is observed with gas turbine. From Table 6.6, it is observed the investment cost of turbine is very high which can be reduced by decreasing the efficiency of the gas turbine up to 0.85. But doing so, is resulting in increase in production cost

From case-V to case-VI

Opposite to the above step, increase in the gas turbine efficiency up to 0.87 results in increase in the investment cost and subsequently increase in the product cost.

Table 6.14 Variables Obtained Exergoeconomic Optimization of Gas Turbine Power Plant (From Base Case to Optimum Case)

Variable	Base Case		Case-I		Case-II	
p_2 / p_1	10		10		9	
η_{AC}	0.86		0.86		0.86	
χ_{APH}	0.75		0.80		0.75	
η_{GT}	0.86		0.86		0.86	
T_4	1247 °C		1247 °C		1247 °C	
Component	$\Delta\varepsilon(\%)$	$\Delta r(\%)$	$\Delta\varepsilon(\%)$	$\Delta r(\%)$	$\Delta\varepsilon(\%)$	$\Delta r(\%)$
AC	-9.944	179.7	-9.851	183.2	-11.38	236.4
APH	-2.719	70029	-0.9494	29519	-3.399	89538
CC	-20.02	5876	-17.56	5156	-18.78	5568
GT	-2.758	1115	-2.764	1140	-2.641	1117
HRSG	-0.4388	32989	-4.629	37430	-1.479	34598
$\dot{C}_{L,tot}$	719.7 ₹/hr		1158 ₹/hr		978.3 ₹/hr	
$\dot{C}_{D,tot}$	6632 ₹/hr		6038 ₹/hr		6370 ₹/hr	
$\dot{C}_P$	20912 ₹/hr		20088 ₹/hr		21348 ₹/hr	
$OBF = \dot{C}_P + \dot{C}_{L,tot} + \dot{C}_{D,tot}$	28263.7 ₹/hr		27284 ₹/hr		28696 ₹/hr	

Table 6.14 (continued)

Variable	Case-III		Case-IV		Case-V	
p_2 / p_1	9		9		9	
η_{AC}	0.87		0.85		0.85	
χ_{APH}	0.75		0.75		0.75	
η_{GT}	0.86		0.86		0.85	
T_4	1247 °C		1247 °C		1247 °C	
Component	$\Delta\varepsilon(\%)$	$\Delta r(\%)$	$\Delta\varepsilon(\%)$	$\Delta r(\%)$	$\Delta\varepsilon(\%)$	$\Delta r(\%)$
AC	-11.16	284.8	-11.64	205.3	-11.73	197.6
APH	-3.54	98057	-3.259	83341	-3.249	84579
CC	-18.91	5697	-18.66	5478	-18.05	5322
GT	-2.662	1089	-2.628	1132	-2.996	1392
HRSG	-2.139	36072	-2.509	34808	-3.089	34367
$\dot{C}_{L,tot}$	946.1 ₹/hr		1012 ₹/hr		1149 ₹/hr	
$\dot{C}_{D,tot}$	6309 ₹/hr		6469 ₹/hr		6527 ₹/hr	
$\dot{C}_P$	22777 ₹/hr		20484 ₹/hr		21362 ₹/hr	
$OBF = \dot{C}_P + \dot{C}_{L,tot} + \dot{C}_{D,tot}$	30032 ₹/hr		27965 ₹/hr		29038 ₹/hr	

Table 6.14 (continued)

Variable	Case-VI		Case-VII		Case-VIII	
p_2 / p_1	9		9		9	
η_{AC}	0.85		0.85		0.85	
χ_{APH}	0.75		0.75		0.80	
η_{GT}	0.87		0.85		0.85	
T_4	1247 °C		1227 °C		1227 °C	
Component	$\Delta\varepsilon(\%)$	$\Delta r(\%)$	$\Delta\varepsilon(\%)$	$\Delta r(\%)$	$\Delta\varepsilon(\%)$	$\Delta r(\%)$
AC	-11.92	194.6	-11.68	195.6	-11.6	199.3
APH	-3.28	87673	-3.139	79880	-1.249	35675
CC	-19.28	5743	-16.81	4977	-14.04	4231
GT	-2.297	1717	-3.1	1401	-3.104	1436
HRSG	-2.069	35495	-3.159	34364	-1.829	36152
$\dot{C}_{L,tot}$	881.9 ₹/hr		1511 ₹/hr		2027 ₹/hr	
$\dot{C}_{D,tot}$	6417 ₹/hr		6364 ₹/hr		5594 ₹/hr	
$\dot{C}_P$	23544 ₹ /hr		20808 ₹/hr		19955 ₹/hr	
$OBF = \dot{C}_P + \dot{C}_{L,tot} + \dot{C}_{D,tot}$	30843 ₹/hr		28683 ₹/hr		27576 ₹/hr	

From case-VI to case-VII

To reduce the exergy destruction in combustion chamber and gas turbine, the temperature T_4 can be increased but it will increase the investment cost of gas turbine and combustion chamber. So the optimum temperature should be decided. The reduction in T_4 up to 1227 °C results in decrease in the production cost. The cost of electricity generation is 2.6 ₹/kWh and cost of steam generation is 800 ₹/1000 kg with 3.3 kg/sec steam flow rate.

From case-VII to case-VIII

Combining all the favourable parameters, including effectiveness of air preheater, the production cost is reduced. The cost of electricity generated is 2.49 ₹/kWh and cost of steam 790 ₹/1000 kg with steam flow rate 3 kg/sec. Here minimum product cost is achieved but the rate of steam generation is less than the requirement. The required steam is 3.14 kg/sec. If the rate of steam generation is to be maintained above 3.14 kg/sec, then the case-IV is to be considered as optimum one.

Considering case IV as optimum, the steam generated at HRSG will have the cost 810 Rs/1000 kg and using this steam as fuel in AAVAR system, the cooling cost for the cooling generated at evaporator will be reduced from 1.35 ₹/sec to 1.07 ₹/sec.

6.1.5.3 Results and Discussions

Various data generated during the optimization procedure using a case by case approach adopted in the present study of gas turbine power plant with HRSG is given in Table 6.14. The comparison of optimum case with the base case is given in Table 6.15. From the study, it can be seen that the cost of electricity is reduced by 4.60 % (2.61 ₹/kWh to 2.49 ₹/kWh) with corresponding decrease in exergy destruction of 15.65 % (6632 ₹/hr to 5594 ₹/hr). The cost of steam generated at HRSG is also reduced from 810 ₹/1000 kg to 790 ₹/1000 kg. When AAVAR system is associated with the HRSG, the cost of cooling at evaporator is reduced to 4853 ₹/hr to 3910 ₹/hr. It shows that the steam generated at HRSG is more economical compared to steam generated at independent boiler as a fuel for AAVAR system. The table shows that this improvement is achieved at the slight reduction in exergetic efficiency by 2.75 %. It should be observed that the rate of power generation and rate of fuel consumption are maintained constant.

Table 6.15: Comparison between base case and the optimum case for GT-HRSG

Properties	Base Case	Optimum Case	% Variation
Fuel Cost $\dot{C}_{F,tot}$	0.2 ₹/MJ	0.2 ₹/MJ	0
Product Cost $\dot{C}_P$	20912 ₹/hr	19955 ₹/hr	4.58 %
Cost of Electricity	2.61 ₹/kWh	2.49 ₹/kWh	4.60 %
Destruction $\dot{C}_{D,tot}$	6632 ₹/hr	5594 ₹/hr	15.65 %
Exergetic Efficiency ε %	51.50 %	50.08 %	2.75 %
Generated Steam Cost	810 ₹/1000 kg	790 ₹/1000 kg	2.47 %

6.2 Tapped Steam as Heat Source

As mentioned earlier, the cost of steam generated in the independent boiler is about 900 ₹/1000 kg (Chapter 5) and steam generated in HRSG of gas turbine power plant is 790 to 810 ₹/1000 kg (Section 6.1 of this chapter). The cost of cooling produced using the above mentioned sources of fuel (steam) for the existing and alternative option of steam generation in GT-HRSG is calculated and presented. As a second option for the reduction of cost of brine chilling, the use of tapped steam from steam turbine of existing steam power plant is analysed through exergoeconomic optimization in this section.

6.2.1 System Simulation

From the available online data, the system is simulated through energy balance and mass balance for all the components and missing data are generated. The following assumptions are considered.

- The power plant system operates at steady state.
- Ideal gas mixture principles apply for the air and the combustion product in the boiler
- The combustion in the combustion chamber of the boiler is complete.
- Super heater and economizer are not considered as independent part.
- Efficiency of the draught fan 80%

In the steam turbine plant model, two types of independent variables are identified, decision variables and parameters or independent variable. The decision variables are varied in the optimization study, while the parameters remain fixed. All other variables are dependent variables and their values are calculated using thermodynamic analysis.

Temperature of the combustion product in the boiler furnace T_{24}, pressure of the steam generated P_1, isentropic efficiency of the turbine η_T and condenser pressure P_6 are considered as decision variables (refer Fig. 3.4)

The following are the fixed parameters used in the present optimisation:

- *System product*
  - ➢ The net power generated by the system is 50 MW

- *Boiler*
  - ➢ FD fan draught 472 mmWC
  - ➢ ID fan draught 230 mmWC
  - ➢ Air molecular analysis (%): 77.48 (N_2), 20.59 (O_2), 0.03 (CO_2), 1.90 (H_2O).
  - ➢ Gas side pressure drop in the boiler 170 mm WC approx.
- *Steam turbine*
  - ➢ First extraction pressure and flow rate 17 bar and 7.67 TPH
  - ➢ Second extraction pressure and flow rate 7 bar and 6.6 TPH
  - ➢ First extraction pressure and flow rate 4 bar and 10 TPH
- *Surface condenser*
  - ➢ Design temperature 100°C
  - ➢ Cooling water flow 92 m^3/hr

The dependent variables include the mass flow rates of the air, combustion products and fuel, the power consumption by the pump and draught fan moreover turbine exit temperature and pressure. Based on the assumption listed, several control volumes considered and set of governing equations developed are given below:

Steam Turbine

For the power generation of 50 MW by the steam turbine, the pressure and temperature at station 1 are given and pressure for stations 2 to 5 are given. Considering the isentropic efficiency of turbine during expansion of steam as 80 %, the actual enthalpies at stations 1 to 5 are calculated. Then by energy balance,

$$\dot{m}_1(h_1 - h_2) + (\dot{m}_1 - \dot{m}_2)(h_2 - h_3) + (\dot{m}_1 - \dot{m}_2 - \dot{m}_3)(h_3 - h_4) + \dot{m}_5(h_4 - h_5) = \dot{W}_T \tag{6.53}$$

where $\dot{m}_5 = \dot{m}_1 - \dot{m}_2 - \dot{m}_3 - \dot{m}_4$

Condenser

$$\dot{m}_6(h_5 - h_6) = \dot{m}_{28} C_w (T_{29} - T_{28}) \tag{6.54}$$

Open Heater

$$\dot{m}_8 h_8 = \dot{m}_4 h_4 + \dot{m}_7 h_7 + \dot{m}_{10} h_{10} \tag{6.55}$$

Closed Heater-I

$$\dot{m}_3 h_3 + \dot{m}_{11} h_{11} + \dot{m}_{14} h_{14} = \dot{m}_{12} h_{12} + \dot{m}_9 h_9 \tag{6.56}$$

Closed Heater-II

$$\dot{m}_{12}(h_{15} - h_{12}) = \dot{m}_{16}(h_{16} - h_{13}) \tag{6.57}$$

Solving Eqs. 6.53 to 6.57, the mass flow rates of steam at stations 1 to 15 are obtained. Fig. 6.6 illustrates the various station points from 1 to 15 of the steam cycle of the plant on T-S diagram.

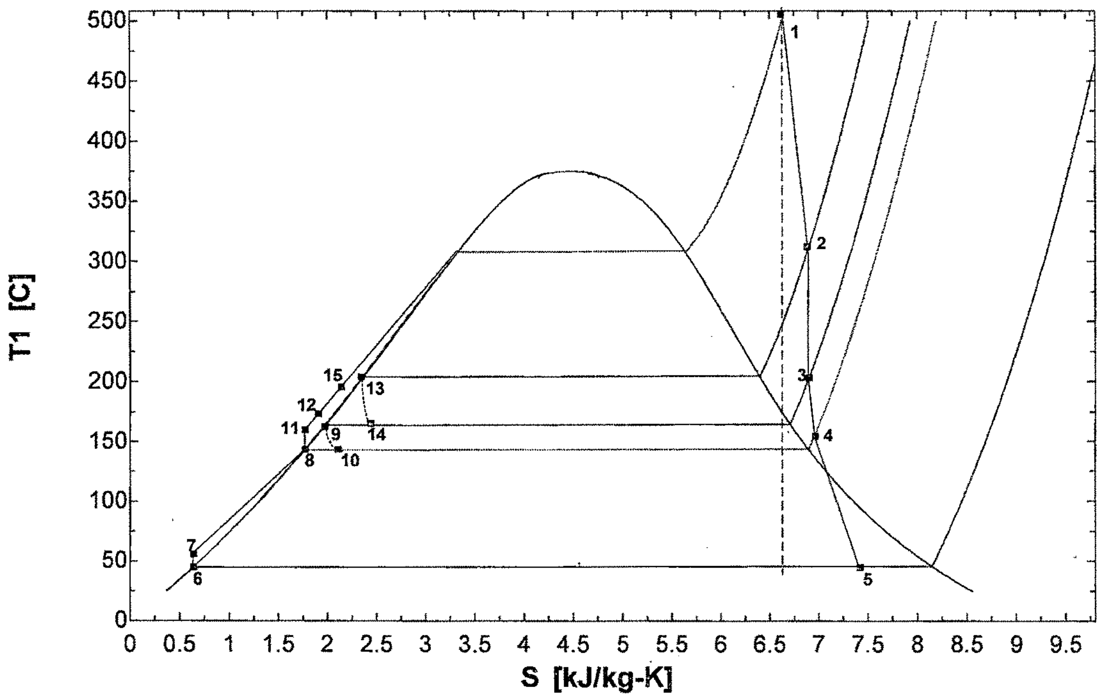

Fig. 6.6 T-S Diagram of Steam Flow Through Steam Turbine

Steam flow from tapping from steam turbine at 17 bar (stream 2) is distributed in two flow, 3.2 kg/sec steam is proposed to divert to AAVAR system as a fuel (station 17) while the remaining steam is supplied to open heater (station 16). Steam from station 17 is throttled to 15 bar (station 18) which is the designed pressure of steam as a fuel for AAVAR system. It is assumed that all the latent heat of steam is consumed in the generator of AAVAR system and condensate comes out at station 19. The condensate is pressurized up to the pressure of 133 bar (station 20) with the help of pump-3 and mixed with the stream flow 15. Mixing of both the streams gives stream 21 which is supplied back to the boiler. From the known properties of steam, the enthalpy and entropy of steam at stations 16 to 21 is estimated with the help of EES software.

Boiler

The ultimate analysis of the coal used in the boiler is given in Table 6.16. Using the percentage of each element, the stoichiometric air fuel ratio is calculated. Considering 20 % excess air supply, the actual air fuel ratio is calculated as explained below.

$$\begin{aligned} &5.925C + 2.53H_2 + 0.281O_2 + 0.05N_2 + 0.181S + Ash + a(O_2 + 3.76N_2) \\ &\rightarrow bCO_2 + cH_2O + dSO_2 + eN_2 \end{aligned} \tag{6.58}$$

From Eq. 6.58

$C: 5.925 = b$

$H_2: 2.53 = c$

$S: 0.181 = d$

$O: 0.281 \times 2 + a \times 2 = 2 \times b + c + 2 \times d$

$N: 0.05 \times 2 + a \times 2 \times 3.76 = 2 \times e$

Table 6.16 Analysis of the Coal Used

Element	% m_i kg	M_i	$n_i = m_i / M_i$
C	71.1	12.00	5.925
H_2	5.1	2.016	2.530
O_2	9.0	32.00	0.281
N_2	1.4	28.01	0.050
S	5.8	32.06	0.181
Ash	7.6	—	—
	100		

Solving the above relations, $a = 7.09$ and $e = 26.71$. Then the combustion equation with stoichiometric air will be

$$\begin{aligned} &5.925C + 2.53H_2 + 0.281O_2 + 0.05N_2 + 0.181S + Ash + 7.09(O_2 + 3.76N_2) \\ &\rightarrow 5.925CO_2 + 2.53H_2O + 0.181SO_2 + 26.71N_2 + Ash \end{aligned} \tag{6.59}$$

With 20% excess air, the combustion analysis will be

$$5.925C + 2.53H_2 + 0.281O_2 + 0.05N_2 + 0.181S + Ash + 7.09 \times 1.2(O_2 + 3.76N_2)$$
$$\rightarrow bCO_2 + cH_2O + dSO_2 + eN_2 + fO_2 \tag{6.60}$$

From the above equation

$C: 5.925 = b$

$H_2: 2.53 = c$

$S: 0.181 = d$

$O: 0.281 \times 2 + 7.09 \times 2 \times 1.2 = 2 \times b + c + 2 \times d + 2 \times f$

$N: 0.05 \times 2 + 7.09 \times 2 \times 3.76 \times 1.2 = 2 \times e$

Solving the above relations, $f = 1.419$ and $e = 32.04$. Then the combustion equation with 20 % excess air will be

$$5.925C + 2.53H_2 + 0.281O_2 + 0.05N_2 + 0.181S + Ash + 8.51(O_2 + 3.76N_2)$$
$$\rightarrow 5.925CO_2 + 2.53H_2O + 0.181SO_2 + 1.414O_2 + 32.04N_2 \tag{6.61}$$

Using Eq. 6.61, the mole fraction of each element of combustion product is calculated and is given in Table 6.17.

Table 6.17 Analysis of Combustion Product

Element	Mole n	Mole fraction $x = n / \sum n$	Molecular weight, M	Mass of element $m = n \times M$
CO_2	5.925	0.1408	44	260.70
H_2O	2.530	0.0601	18	45.54
SO_2	0.181	0.0043	64	11.58
O_2	1.414	0.0337	32	45.44
N_2	32.040	0.7611	28	897.12
$\sum n$	**42.096**	**1.0000**		**1260.38**

The mass of air supplied for 100 kg coal is given by

$m_{air} = 8.51 \times (32 + 3.76 \times 28) = 1168.25kg$

Air fuel ratio

$A / F = m_{air} / m_F$

$A / F = 1168.25 / 100 = 11.68$

Mass flow rate of exhaust gas =1260.38 kg for 100 kg coal.

The gross calorific value (GCV_{coal}) of the coal, measured using calorimeter is found to be 23 MJ/kg.

Forced Draught Fan

The forced draught fan used in the boiler creates draught of 472 mmWC. The temperature of environment air (T_0) and isentropic efficiency of FD fan are taken as 298.1K and 80%, respectively. The temperature of air at the exit of FD fan can be estimated using

$$T_{23} = T_0 \left\{ 1 + \frac{1}{\eta_{FD}} \left[\left(\frac{p_{23}}{p_0} \right)^{\frac{\gamma_a - 1}{\gamma_a}} - 1 \right] \right\} \tag{6.62}$$

At this temperature, T_{23}, the enthalpies of all the constituents, nitrogen, oxygen, carbon dioxide and water vapour are calculated using the Eq. F2 of Appendix F. Then, enthalpies of all the constituents are added on molar basis and the enthalpy of the air inlet to compressor is calculated on molar basis using

$$h_{23}^{'} = 0.7748 h_{N_2}(T_{23}) + 0.2059 h_{O_2}(T_{23}) + 0.0003 h_{CO_2}(T_{23}) + 0.019 h_{H_2O}(T_{23}) \tag{6.63}$$

The enthalpy of air on mass basis is, then, calculated using

$$h_{23} = h_{23}^{'} / M_a \tag{6.64}$$

For the existing case, the temperature of combustion product is taken as 1500 K. Then the enthalpy of combustion product is estimated using

$$h_{24} = x_{N_2} h_{N_2}(T_{24}) + x_{O_2} h_{O_2}(T_{24}) + x_{CO_2} h_{CO_2}(T_{24}) + x_{H_2O} h_{H_2O_g}(T_{24}) + x_{SO_2} h_{SO_2}(T_{24}) \tag{6.65}$$

The combustion product traverse through evaporation zone during which 170 mmWC pressure drops takes place and temperature of gas at the exit of evaporation zone is found to be 160°C during normal operation of the plant. The enthalpy of combustion product at station 25 is given by

$$h_{25} = x_{N_2} h_{N_2}(T_{25}) + x_{O_2} h_{O_2}(T_{25}) + x_{CO_2} h_{CO_2}(T_{25}) + x_{H_2O} h_{H_2O_g}(T_{25}) + x_{SO_2} h_{SO_2}(T_{25}) \tag{6.66}$$

Induced Draught Fan

The ID fan creates draught of 230 mmWC. The temperature and enthalpy of the gas at the exit of ID fan is found for isentropic efficiency of ID fan, $\eta_{ID} = 80\%$, at 80 % using

$$T_{26} = T_{25}\left\{1+\frac{1}{\eta_{ID}}\left[\left(\frac{p_{26}}{p_{25}}\right)^{\frac{\gamma_a-1}{\gamma_a}}-1\right]\right\} \tag{6.67}$$

$$h_{26} = h_{25} + W_{P2} \tag{6.68}$$

The work done by the FD fan and ID fan are estimated using

$$W_{\text{FDfan}} = \frac{\Delta P_{FDfan} \times V_0 \times T_0 \times A/F \times m_f}{273.15 \times \eta_{FDfan}} \tag{6.69}$$

$$W_{\text{IDfan}} = \frac{\Delta P_{\text{IDfan}} \times V_0 \times T_{25} \times A/F \times (m_f + 1)}{273.15 \times \eta_{\text{IDfan}}} \tag{6.70}$$

where, $V_0 = 0.7835 m^3$ is the volume of air at NTP and $T_0 = 298.1K$

Using the enthalpy balance in the evaporation zone of the boiler, the mass flow rate of gas (m_g) in the boiler is calculated for the given rate of power generation.

$$m_1(h_1 - h_{21}) = m_g(h_{24} - h_{25}) \tag{6.71}$$

After calculating the mass flow rate of gas, mass flow rate of fuel can be found as the mass of exhaust gas for 100 kg coal combustion is available from Table 6.17. Using the value of Air Fuel ratio and flow rate of fuel, flow rate of air can be estimated for the given rate of power generation.

The entropy of steam and water at stations 1 to 21 is calculated using the in built subroutine of the EES software at the given temperature and pressure. The air at station 23 is at pressure other than p_{ref}. Then entropy of air at temperature T_{23} and p_{23} is calculated using

$$s'_{23} = 0.7748\left[s_{N_2}(T_{23}) - R*\ln\left(\frac{0.7748*p_{23}}{p_0}\right)\right] + 0.2059\left[s_{O_2}(T_{23}) - R\ln\left(\frac{0.2059*p_{23}}{p_0}\right)\right] + 0.0003\left[s_{CO_2}(T_{23}) - R\ln\left(\frac{0.0003*p_{23}}{p_0}\right)\right] + 0.019\left[s_{H_2O_g}(T_{23}) - R\ln\left(\frac{0.019*p_{23}}{p_0}\right)\right] \tag{6.72}$$

$$s_{23} = \frac{s_{23}^{'}}{M_a} \text{ (kJ/kg)} \tag{6.73}$$

At station 24, combustion product is considered as ideal gas mixture. Table 6.17 gives the mole fraction of all the constituents. Using the mole fractions, the entropy of combustion product at station 24 is found using following relation in terms of kJ/kmol.

$$\begin{aligned} s_{24}^{'} = {} & x_{N_2}\left[s_{N_2}(T_{24}) - R\ln\left(\frac{x_{N_2} * p_{24}}{p_0}\right)\right] + x_{O_2}\left[s_{O_2}(T_{24}) - R\ln\left(\frac{x_{O_2} * p_{24}}{p_0}\right)\right] + \\ & x_{CO_2}\left[s_{CO_2}(T_{24}) - R\ln\left(\frac{x_{CO_2} * p_{24}}{p_0}\right)\right] + x_{H_2O_g}\left[s_{H_2O_g}(T_{24}) - R\ln\left(\frac{x_{H_2O_g} * p_{24}}{p_0}\right)\right] + \\ & x_{SO_2}\left[s_{SO_2}(T_{24}) - R\ln\left(\frac{x_{SO_2} * p_{24}}{p_0}\right)\right] \end{aligned} \tag{6.74}$$

$$s_{24} = \frac{s_{24}^{'}}{M_p} \text{ (kJ/kg)} \tag{6.75}$$

Similarly, the entropy at stations 25 and 26 are calculated at corresponding temperature and pressure. It should be noted that station 22 represents fuel (coal) at boiler inlet, station 27 represents the rate of power generation while the stations 28 and 29 represents cooling water inlet and exit to condenser.

6.2.2 Exergy Analysis

The theoretical description of exergy and its components is given in Chapter 4. In this section, the estimation of the two components of exergy, viz. physical and chemical exergy for each station is given.

6.2.2.1 Physical Exergy

For the stations 1 to 21, the working fluid is either steam or water. At $T_{ref} = 25^o C$ and $p_{ref} = 1.01325$ bar, their enthalpy h_0 and entropy s_0 are found using EES software. Then the physical exergy at stations 1 to 21 is found using Eq 4.2.

Station 23 represents the exit condition of air at FD fan which is the inlet to the combustion chamber of the boiler. To calculate the exergy of air at station 23, its enthalpy and entropy at T_0 and P_0 are found using the following:

$$h'_{0a} = 0.7748 h_{N_2}(T_0) + 0.2059 h_{O_2}(T_0) + 0.0003 h_{CO_2}(T_0) + 0.019 h_{H_2O}(T_0) \quad (6.76)$$

The molecular weight of the air inlet to combustion chamber of the boiler is calculated using

$$M_a = 0.7748 M_{N_2} + 0.2059 M_{O_2} + 0.0003 M_{CO_2} + 0.019 M_{H_2O} \quad (6.77)$$

Using these values, the enthalpy of air on mass basis is calculated using

$$h_{0a} = h'_{0a} / M_a \quad (6.78)$$

$$s'_{0a} = 0.7748 s_{N_2}(T_0) + 0.2059 s_{O_2}(T_0) + 0.0003 s_{CO_2}(T_0) + 0.019 s_{H_2O}(T_0) \quad (6.79)$$

$$s_{0a} = \frac{s'_{0a}}{M_a} \text{ (kJ/kg)} \quad (6.80)$$

Using enthalpy and entropy of air at exit of FD fan and at reference state, the exergy at station 23 is found using Eq. 4.2

To calculate the exergy of combustion product and exhaust gas from the boiler, it is considered that they are reduced to $T_{ref} = 25^oC$ and $p_{ref} = 1.01325$ bar. At this temperature, some condensation of water will occur and gas phase containing saturated water vapour in equilibrium with saturated liquid water phase. On the basis of 1 kmol of combustion products formed, the gas phase at 25°C would consists of 0.9399 kmol of dry products (0.7611 N_2, 0.0337 O_2, 0.1408 CO_2, 0.0043 SO_2) plus n_v kmol of water vapour. The partial pressure of water vapour would be equal to the saturation pressure, $p_g(25^oC) = 0.0317$ bar. The amount of water vapour is found using

$$p_v = x_v p \quad (6.81)$$

$$0.0317 \text{ bar} = \frac{n_v}{0.9399 + n_v}(1.01325 \text{ bar}) \quad (6.82)$$

Solving Eq. 6.82, $n_v = 0.03035$ kmol.

Thus, the composition of the combustion product as given in Table 6.17 is to be modified for the condition at 25°C and 1 atm and is given as under:

$\underline{0.7611\ N_2,\ 0.0337\ O_2,\ 0.1408\ CO_2,\ 0.0043 SO_2,\ 0.03035\ H_2O\ (g)}$, $0.02975 H_2O$ (l).

The underline indicates the gas phase. Enthalpy of combustion product at 25°C and 1 atm is given

$$h'_{024} = x_{N_2} h_{N_2}(T_0) + x_{O_2} h_{O_2}(T_0) + x_{CO_2} h_{CO_2}(T_0) + x_{SO_2} h_{SO_2}(T_{25}) + x_{H_2O} h_{H_2O_g}(T_0) + x_{H_2O} h_{H_2O_l}(T_0) \quad (6.83)$$

$$h_{024} = \frac{h'_{024}}{M_p} \text{ kJ/kg} \quad (6.84)$$

$$s'_{024} = (x_{N_2} s_{N_2}(T_0) + x_{O_2} s_{O_2}(T_0) + x_{CO_2} s_{CO_2}(T_0) + x_{H_2O_g} s_{H_2O_g}(T_0) + x_{H_2O_l} s_{H_2O_l}(T_0))$$

$$s_{024} = \frac{s'_{024}}{M_p} \text{ kJ/kgK} \quad (6.85)$$

Now, the physical exergy at station 24 is calculated using Eq. 4.2. Using the same value of enthalpy and entropy at reference condition, physical exergy at station 25 and 26 is also calculated.

6.2.2.2 Chemical Exergy

At the station 1 to 21, the working fluid is steam or water. When it is brought to the equilibrium with the atmosphere, it will be in liquid state. Chemical exergy of water as selected from Appendix G.

$$e^{CH}_{water} = 45 \text{ kJ/kmol} \quad (6.86)$$

At station 23, air is stable with environment so its chemical exergy is considered as zero. At dead state corresponding to the mixture at stations 24 to 26 consists of liquid water phase and a gas phase. The new mole fraction of a gas phase is calculated as,

$$y_{N_2} = \frac{x_{N_2}}{x_{N_2} + x_{O_2} + x_{CO_2} + x_{SO_2} + n_v} \qquad y_{O_2} = \frac{x_{O_2}}{x_{N_2} + x_{O_2} + x_{CO_2} + x_{SO_2} + n_v}$$

$$y_{CO_2} = \frac{x_{CO_2}}{x_{N_2} + x_{O_2} + x_{CO_2} + x_{SO_2} + n_v} \qquad y_{H_2O(g)} = \frac{n_v}{x_{N_2} + x_{O_2} + x_{CO_2} + x_{SO_2} + n_v} \quad (6.87)$$

$$y_{SO_2} = \frac{x_{SO_2}}{x_{N_2} + x_{O_2} + x_{CO_2} + x_{SO_2} + n_v}$$

They are found as

$y_{N_2} = 0.7844,\ y_{O_2} = 0.03473,\ y_{CO_2} = 0.1451, y_{SO_2} = 0.004432,\ y_{H_2O(g)} = 0.03129$

Table 6.18 State Properties for Steam Power Plant

Stations	Mass flow rate kg/sec	Pres-sure. bar	Temp. °C	Specific Enthalpy kJ/kg	Specific Entropy kJ/kgK	Physical Exergy kW	Chemical Exergy kW	Total Exergy MW
1	56.86	96.00	500.00	3379.00	6.6210	80161.0	142.10	80.30
2	7.08	17.00	286.40	3000.00	6.7970	6928.00	17.70	6.95
3	3.11	7.00	205.20	2856.00	6.9100	2488.00	7.77	2.50
4	5.20	4.00	160.00	2775.00	6.9810	3633.00	13.01	3.65
5	41.47	0.10	45.82	2353.00	7.4250	5973.00	103.70	6.08
6	41.47	0.10	45.79	191.70	0.6489	116.90	103.70	0.22
7	41.47	4.00	45.88	192.10	0.6489	133.30	103.70	0.24
8	53.66	2.00	120.20	504.70	1.5300	2850.00	134.10	2.98
9	6.99	6.00	158.80	670.60	1.9310	694.60	17.47	0.71
10	6.99	4.00	143.60	669.20	1.9310	684.90	17.47	0.70
11	53.66	135.00	123.60	518.80	1.5300	3607.00	134.10	3.74
12	53.66	134.00	154.00	657.50	1.8680	5641.00	134.10	5.78
13	3.88	15.00	198.30	844.80	2.3150	617.80	9.70	0.63
14	3.88	7.00	165.00	838.50	2.3150	593.60	9.70	0.60
15	53.66	133.00	190.00	813.30	2.2190	8396.00	134.10	8.53
16	3.88	17.00	286.40	3000.00	6.6210	3999.00	9.70	4.01
17	3.20	17.00	286.40	3000.00	6.6210	3299.00	8.00	3.31
18	3.20	15.00	198.30	2791.00	6.4440	2800.00	8.00	2.81
19	3.20	15.00	198.30	844.80	2.3150	509.70	8.00	0.52
20	3.20	133.00	201.60	858.40	2.3150	553.30	8.00	0.56
21	56.86	133.00	190.60	815.80	2.2240	8948.00	142.10	9.09
22	8.99	1.06	25.00	--	--	0	206863	206.90
23	105.10	1.06	29.90	-159.60	6.9640	528.50	0	0.53
24	113.40	1.013	1227.00	-951.10	8.6080	97601	9263	106.90
25	113.40	1.00	160.00	-2237.00	7.1370	1565.00	9263	10.83
26	113.40	1.019	163.50	-2233.00	7.14	1879	9263	11.14
27	--	--	--	--	--	--	--	50
28	2555.00	1.013	33.00	138.30	0.4777	1141	6388	7.53
29	2555.00	1.013	41.38	173.30	0.5906	4666	6388	11.05

Now the chemical exergy for the k^{th} component is calculated and added together to find total chemical exergy using following equation

$$e^{CH} = \sum y_k e_k^{CH} + RT_0 \sum y_k \ln y_k \qquad (6.88)$$

This is the chemical exergy of gas portion. The chemical exergy of liquid portion is separately calculated and added together to find total chemical exergy. The chemical exergy of individual component (e_k^{CH}) is taken from Appendix G.

$$\dot{E}^{CH} = \dot{m}_g\left[\left(x_{N_2} + x_{O_2} + x_{CO_2} + x_{SO_2} + n_v\right)e^{CH} + x_{H_2O(l)} * e_{H_2O(l)}^{CH}\right] \tag{6.89}$$

This is the chemical exergy in kJ/kmol. It is then converted in kJ/kg by dividing it by molecular mass of the combustion product. The chemical exergy at stations 24 to 26 will remain same. Standard chemical exergy of coal is taken equal to its GCV. The total exergy flow at all the stations will be the sum of physical and chemical exergy. Table 6.18 gives state properties and total exergy along with its components for various stations from 1 to 29.

6.2.2.3 Definition of Fuel, Product and Loss for Various Processes

For all the components of the steam turbine power plant, fuel, product and loss are defined as given in Chapter 4, Section 4.1.1. They are summarized in Table 6.19 and calculated values are given Table 6.20.

Table 6.19 Fuel, Product and Loss for various Components of Steam Power Plant

Component	Fuel ($\dot{E}_F$)	Product ($\dot{E}_P$)	Loss ($\dot{E}_L$)
Boiler Furnace	$\dot{E}_{22} + \dot{E}_{23} + \dot{E}_{26} - \dot{E}_{25}$	$\dot{E}_{24}$	0
Boiler HX	$\dot{E}_{24} - \dot{E}_{25}$	$\dot{E}_1 - \dot{E}_{21}$	0
Steam Turbine	$\dot{E}_1 - (\dot{E}_2 + \dot{E}_3 + \dot{E}_4 + \dot{E}_5)$	$\dot{E}_{27}^{W}$	0
Turbine Cond. Assly.	$\dot{E}_1 - \dot{E}_2 - \dot{E}_3 - \dot{E}_4 - \dot{E}_6$	$\dot{E}_{27}^{W}$	0
Condenser	—	—	0
Overall System	$\dot{E}_{22} + \dot{W}_{FD} + \dot{W}_{ID}$	$\dot{E}_{27}^{W}$	$\dot{E}_{26} + (\dot{E}_{29} - \dot{E}_{28})$

Table 6.20 Exergy Analysis of Steam Power Plant

Component	$\dot{E}_F$ MW	$\dot{E}_P$ MW	$\dot{E}_L$ MW	$\dot{E}_D$ MW	Y_D %	Y_L %	Y_D^* %	ε %
Boiler Furnace	207.70	106.90	0	100.80	48.53	0	70.46	51.45
Boiler HX	96.04	71.21	0	24.82	11.95	0	17.34	74.15
Steam Turbine	61.14	50.00	0	11.14	5.36	0	7.78	81.78
Turbine Cond. Assly.	67.00	50.00	0	17.00	8.18	0	11.87	74.63
Condenser	--	--	0	5.86	2.82	0	4.09	60.20
Overall System	207.80	50.00	14.67	143.10	68.88	7.06	100	24.06

6.2.2.4 Results and Discussions

The outcome of the exergy analysis of steam turbine power plant is given in Table 6.20. The total exergy supplied to the system is 207.80 MW. Out of which 50 MW (24.06 %) is converted to useful product. 143.10 MW (68.88 %) exergy is destroyed and 14.67 MW (7.06 %) is lost to the environment. The maximum exergy destruction is observed in boiler furnace. To reduce the exergy destruction in boiler furnace, the furnace temperature should be increased. For that, turbulence can be created and better air preheater can improve the performance. The next component in this category is boiler heat exchanger. To reduce the exergy destruction in heat exchanger, effectiveness of the same can be improved.

6.2.3 Exergoeconomic Analysis

The economic analysis of thermal system requires the identification and inclusion of various cost heads incurred in the estimation of the total cost for the production. In the present case, the total cost involved in the power generation of steam turbine consists of many cost heads. Thus, in general, the economic analysis of the system requires the estimation of levelized O & M cost of component ($\dot{Z}_k$) and fuel cost rate ($\dot{C}_f$). $\dot{Z}_k$ should be estimated for each component for steam power plant using TCI, β, γ and τ (Refer Eq.4.18). The fuel cost rate ($\dot{C}_f$) is governed by the source of heat energy used for the system. The estimation of $\dot{Z}_k$ and $\dot{C}_f$ are explained in the following section.

6.2.3.1 Levelized O&M Cost

For estimation of the cost of boiler, turbine, condenser and pumps, the cost models suggested by Silveira et al. [139] are used and are given in Appendix H. These cost models gives the total capital investment including the installation cost, electrical equipment cost, control system cost, piping cost and local assembly cost. Using the Marshall & Swift cost index, they are converted for the year 2009. The operation and maintenance cost of each component is found using Eq. 4.18 in which the plant life is considered as 8000 hours, Capital Recovery Factor (β) = 0.1061, Operation and Maintenance cost, $\gamma = 1.092$ % of total capital investment. The values of operation and maintenance cost ($\dot{Z}_k$) for each component are given in Table 6.22.

6.2.3.2 Fuel Cost

The plant uses coal as a fuel. The market price of coal for the year 2009 was ₹ 3000 per 1000 kg. So cost of fuel is considered as ₹ 3/kg coal.

6.2.3.3 Cost Flow

Applying the formulation of cost balance equations and the definition of fuel, product and loss (Refer Table 6.20); the exergoeconomic cost balance equations for each component of steam power plant are formulated in the following forms:

Considering boiler, turbine and turbine condenser assembly as a control volume, following cost balance equations are modelled.

Boiler

$$c_{22}\,\dot{E}_{22} + c_{23}\,\dot{E}_{23} + c_{23}(\dot{E}_{26} - \dot{E}_{25}) - c_{25}\,\dot{E}_{25} - c_1\,\dot{E}_1 + \dot{Z}_{BL} = 0 \qquad (6.90)$$

$$\dot{c}_{22} = \dot{c}_{25} \qquad (6.91)$$

$$\dot{c}_{24} = \dot{c}_{25} \qquad (6.92)$$

$\dot{c}_{25} = \dot{c}_{26}$ (6.93)

$\dot{c}_{23} = \dot{c}_{27}$ (6.94)

Steam Turbine

$\dot{c}_1 \dot{E}_1 - \dot{c}_2 \dot{E}_2 - \dot{c}_3 \dot{E}_3 - \dot{c}_4 \dot{E}_4 - \dot{c}_5 \dot{E}_5 - \dot{c}_{27} \dot{E}_{27} + \dot{Z}_{ST} = 0$ (6.95)

$\dot{c}_1 = \dot{c}_2$ (6.96)

$\dot{c}_1 = \dot{c}_3$ (6.97)

$\dot{c}_1 = \dot{c}_4$ (6.98)

$\dot{c}_1 = \dot{c}_5$ (6.99)

Out of these variables, $\dot{c}_1 ... \dot{c}_5$ and $\dot{c}_{22} ... \dot{c}_{27}$, the fuel cost $\dot{c}_{22}$ is known. The remaining 10 are calculated by solving Eqs. 6.90 to 6.99 using EES software. The cost per unit exergy (₹/MJ) and cost flow rate (₹/sec) for each flow of the system are calculated and shown in Table 6.21. For this calculation, known values of $\dot{E}_1$ to $\dot{E}_5$ and $\dot{E}_{22}$ to $\dot{E}_{27}$ are used.

Table 6.21 Unit Exergy Cost and Cost Flow Rate for Steam Power Plant

Flows	Unit exergy cost ₹/MJ	Exergy flow MW	Cost flow rate ₹/sec
1	0.4025	80.300	32.320
2	0.4025	6.945	2.796
3	0.4025	2.495	1.004
4	0.4025	3.646	1.468
5	0.4025	6.077	2.446
22	0.1319	204.500	26.980
23	0.5540	0.529	0.293
24	0.1319	106.900	14.100
25	0.1319	10.830	1.429
26	0.1319	11.140	1.470
27	0.5540	50.000	27.700

6.2.4 Exergoeconomic Evaluation

Solution of the cost balance equations will give the cost flow rates at each station of the plant and cost rate of product ($\dot{C}_{p,k}$) using cost rate of fuel as an input ($\dot{C}_{F,k}$). After that, cost rate of fuel per unit exergy ($c_{F,k}$), cost rate of product per unit exergy ($c_{P,k}$), cost rate of exergy destruction ($\dot{C}_{D,k}$), cost rate of exergy loss ($\dot{C}_{L,k}$), the relative cost difference (r_k) and exergoeconomic factor (f_k) for each components are calculated using Eqs. 4.20 to 4.27 and given in Table 6.22.

Table 6.22 Results of Exergoeconomic Analysis

Component	$c_{F,k}$ ₹/MJ	$c_{p,k}$ ₹/MJ	$\dot{C}_{D,k}$ ₹/hr	$\dot{C}_{L,k}$ ₹/hr	$\dot{Z}_k$ ₹/hr	f_k %	r_k %	ε %
Boiler furnace	0.13	0.13	47167	0	22695	27.80	241.30	34.68
Boiler HX	0.13	0.45	11790	0				
Turbine	0.40	0.55	16143	0	11117	40.80	37.62	81.78
Turbine Condenser Assembly	0.40	0.55	24710	0	11187	31.20	37.17	74.63
Condenser	--	--	8567	0	70	0.80	--	60.20
System	0.48	0.55	242144	25227	34328	11.40	15.95	24.34

6.2.4.1 Results and Discussions

The following observations are made from the exergoeconomic analysis of steam power plant with regeneration shown in Table 6.22.

(i) The r value for the boiler is found highest among all the components. The boiler has lowest exergetic efficiency. In combustion chamber of the boiler, the maximum exergy destruction is observed from the Table 6.20. It suggests that the temperature of the combustion product should be increased by modifying the boiler design.

(ii) In the evaporation zone of the boiler, the next highest exergy destruction is observed from the Table 6.20. It suggests that the boiler pressure and temperature should be increased. The turbine is having the next highest r value and exergy destruction cost. It suggests that the isentropic efficiency of steam turbine should be increased by increasing the investment cost.

(iii) The condenser is having very low f value and higher exergetic efficiency. It suggests that the condenser of the plant working properly as it is having less investment cost and less exergy destruction.

6.2.5 Exergoeconomic Optimization

The exergy analysis suggests improvement in the thermal system which is associated the increase in investment and Operation and maintenance cost. These two are conflict in nature. The exergoeconomic optimization provides optimum condition between improvement in thermal performance of the system and increase in the cost.

6.2.5.1 Estimation of B_k, n_k and m_k

Using the value of cost flow at each station and the results of exergoeconomic evaluation, the exergoeconomic optimization of the system is carried out at component level using Eq. 4.29. To solve this equation for local optimum by curve fitting technique, the equivalent power law is found and the required value of B_k and n_k for each component are determined for the selected value of m_k as explained below.

Boiler

For boiler, the temperature and pressure of the steam generated by the boiler are considered as the decision variables. With the variation of temperature and pressure of steam generated in the boiler, the variation of exergetic efficiency of the boiler and total capital investment are generated in the form explained in section 4.3 and given in Table 6.23. The required graph is plotted as shown in Fig. 6.7. By curve fitting technique, the required power law is developed as shown in the Fig. 6.7. The figure shows that the value of B_{BL} and n_{BL} are found to be 1.36×10^7 and 4.5598 for the selected value of m_{BL} of 0.78 as suggested by Bejan et al. [155].

Table 6.23 Generated Data Using Investment Cost Equation for Boiler

p_{ST} bar	T_{ST} °C	$\dot{E}_{P,BL}/\dot{E}_{D,BL}$	$TCI_{BL}/\dot{E}_{p,BL}^{0.78}$
94	490	0.5734	1078000
96	494	0.5760	1100000
98	498	0.5786	1123000
99	502	0.5804	1139000
100	504	0.5817	1151000

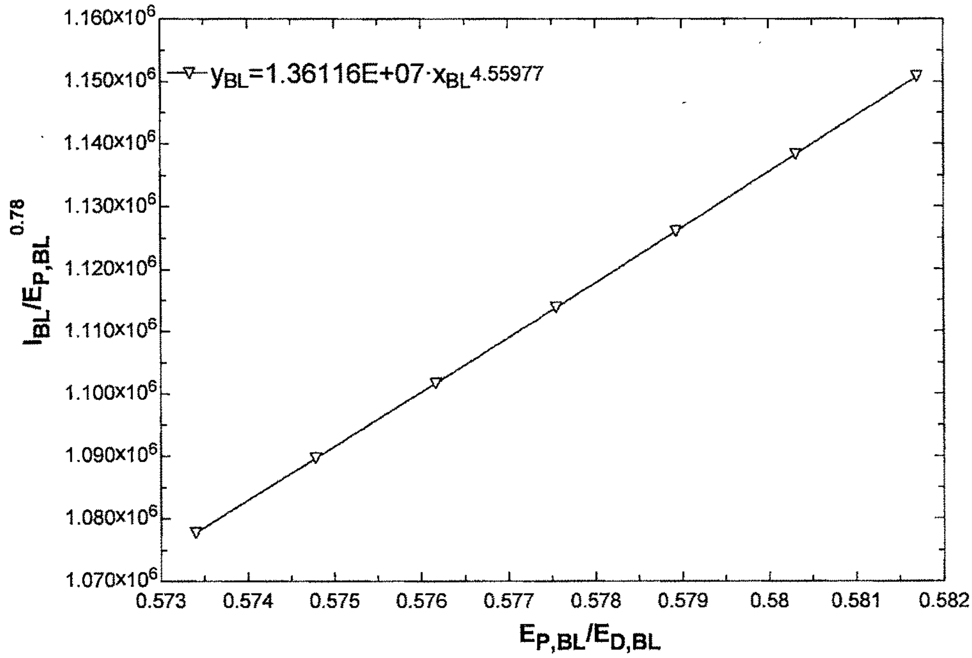

Fig. 6.7 Plot of Investment cost v/s Exergetic Efficiency for Boiler

Steam turbine

For steam turbine, the isentropic efficiency is considered as the decision variable. Parametric variation of various properties with respect to isentropic efficiency is carried out and the following Table 6.24 is generated and the graph of investment cost v/s exergetic efficiency is plotted for the steam turbine as shown in Fig. 6.8 with the required power law through curve fitting technique. The figure shows that the value of B_{ST} and n_{ST} are found to be 364648 and 0.1384, respectively for the selected value of m_{ST} of 0.9 as suggested by Bejan et al. [155].

Table 6.24 Generated Data Using Investment Cost Equation for Steam Turbine

η_{ST}	$\dot{E}_{P,ST}$	$\dot{E}_{D,ST}$	$\dot{E}_{P,ST}/\dot{E}_{D,ST}$	$TCI_{ST}/\dot{E}_{p,ST}^{0.9}$
0.75	14.85	50	3.3660	430252
0.76	14.07	50	3.5530	434163
0.77	13.31	50	3.7560	438057
0.78	12.57	50	3.9780	441935
0.79	11.85	50	4.2210	445798
0.80	11.14	50	4.4880	449645
0.81	10.45	50	4.7840	453477
0.82	9.78	50	5.1120	457294
0.83	9.13	50	5.4790	461096
0.84	8.49	50	5.8900	464883

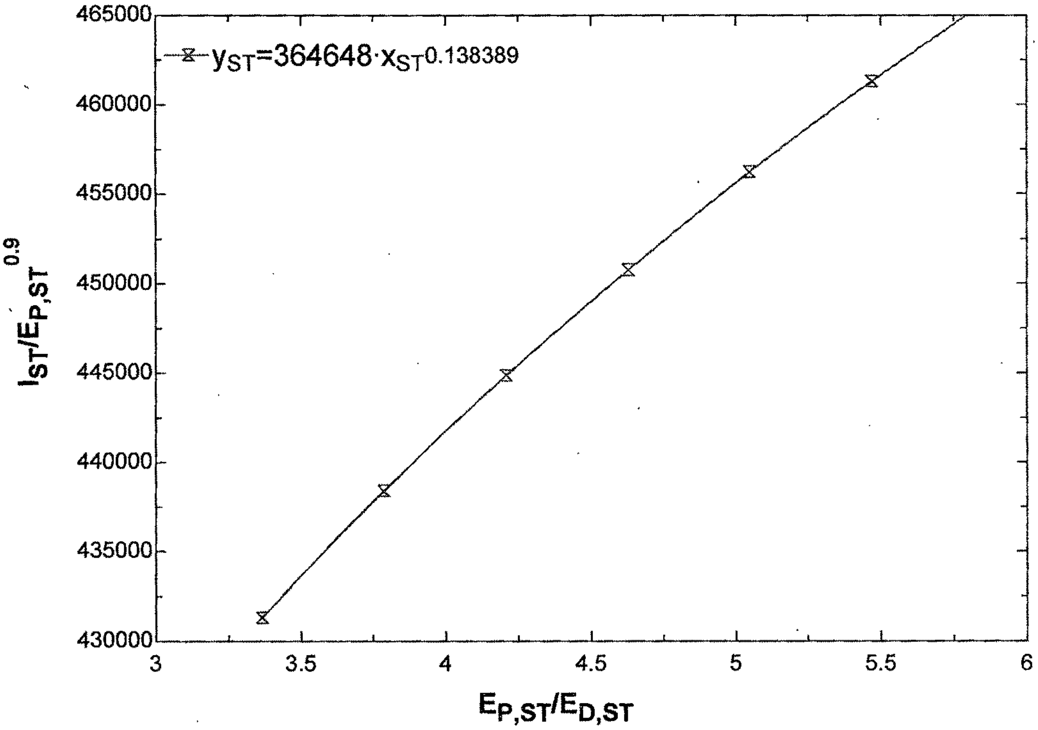

Fig. 6.8 Plot of Investment Cost v/s Efficiency for Steam Turbine

Table 6.25 summarises the component-wise parameters, B_k , n_k and m_k estimated along with the decision variable.

Table 6.25 Values of B_k, n_k and m_k

Component	Decision variable	B_k	n_k	m_k
Boiler	p_{BL} & T_{BL}	1.36 x 10^7	4.5598	0.78
Steam Turbine	η_{ST}	364648	0.1384	0.90

6.2.5.2 Optimisation Through Case by Case Iterative Procedure

Optimum values of exergetic efficiency (ε_k^{OPT}), the capital investment (Z_k^{OPT}), the relative cost difference (r_k^{OPT}) and the exergoeconomic factor (f_k^{OPT}) can be calculated using Eqs. 4.37, 4.45, 4.46 and 4.47, respectively. Through an iterative optimization procedure, optimum solution can be achieved, with the help of calculated values of $\dot{C}_{P,tot}, \dot{C}_{D,tot}, \dot{C}_{L,tot}$ and OBF and the guidance provided by the values of $\Delta\varepsilon_k$ and Δr_k, calculated using Eqs. 4.50 and 4.51.

Table 6.26 summarizes the results obtained from the case-by-case iteration carried out starting from the base case (base case is the case evaluated using the data of the existing system) to the optimum case. A total of seven iterative cases are presented and the resulting cases are given as cases I to VII out of which the case VI is found to be the optimum. Each of these cases is obtained through a series of study of positive or negative effects on $\dot{C}_{p,tot}$ and $\dot{C}_{D+L}$ by varying each decision variable. The change in the decision variables are governed by $\Delta\varepsilon_k$ and Δr_k. The details of the case by case iterative procedure for exergoeconomic optimization of AAVAR system is discussed in the following paragraph and the output given in Table 6.26. In the base case, the unit product cost of electricity is 1.99 ₹/kWh and production cost of steam is 395 ₹/1000 kg.

From base case to case-I

The highest value of Δr_{ST} shows that the product cost of air preheater is very high. It suggests that the isentropic efficiency of the steam turbine should be increased. The isentropic efficiency of the steam turbine is increased from 80% to 85%. With this the cost of electricity is reduced to 1.91 ₹/kWh and cost of steam extracted from the turbine and proposed to be utilized in absorption refrigeration system is reduced to 391 ₹/1000kg steam.

From case-I to case-II

The highest Δr_{BL} suggest that the exergy destruction can be reduced in the evaporation zone of the boiler by increasing the temperature of steam generated. Higher rate of exergy destruction in the evaporation zone of the boiler can be identified from the Table 6.20. In this regards, the temperature of steam is increased from 500°C to 505°C. This will result in the increase in the investment cost of boiler and subsequently the cost of electricity and the cost of steam extracted from the turbine are slightly increased. But the higher temperature of steam reduces the cost of exergy destruction which results in reduction of objective function (OBF). Increase in the temperature beyond this is not so effective.

From case-II to case-III

More rises in the steam temperature gives adverse effect on the product cost and on the objective function.

From case-III to case-IV

Further reduction in the exergy destruction in the evaporation zone of the boiler can be carried out by increasing the steam pressure. The steam pressure is increased from 96 bar to 98 bar. With this, the product cost and objective function is slightly reduced

From case-IV to case-V

Further increase in the pressure from 98 bar to 100 bar gives slight increase in the product cost but reduction in the objective function as the cost of exergy destruction is reduced. So this pressure is accepted as optimum one.

From case-V to case-VI

From Table 6.20, it is observed that the exergy destruction is very high in the combustion chamber of the boiler. This exergy destruction can be reduced by the increase in the temperature of the combustion product. Increasing the temperature of combustion product from 1500°C to 1510°C, the cost of electricity generated is reduced to 1.91 ₹/kWh and the cost of steam extracted will be 389.3 ₹/1000 kg. Beyond this temperature in the boiler furnace, the ace melting temperature is achieved so accepting this temperature of the combustion product as optimum one.

From case-VI to case-VII

To reduce the temperature difference between combustion product and steam generated in a boiler to reduce the exergy destruction, the temperature of steam is increased to 510°C. But it is giving adverse effect on the performance of a system. Hence *case VI* is found to be optimum one. With this optimum configuration of steam power plant (*case-VI*), the cost of steam at station 2 is found to be 389 ₹/1000 kg. Using this steam as fuel in AAVAR system, the cost of cooling at evaporator can be reduced to 0.68 ₹/sec.

Table 6.26 Variables Obtained During Exergoeconomic Optimization of Steam Turbine Power Plant (from Base Case to Optimum Case)

Variable	Base case		Case-I		Case-II		Case-III	
p_1	96 bar		96 bar		96 bar		96 bar	
T_1	500°C		500°C		505°C		510°C	
η_{ST}	0.80		0.85		0.85		0.85	
T_{24}	1500°C		1500°C		1500°C		1500°C	
Component	$\Delta\varepsilon(\%)$	$\Delta r(\%)$	$\Delta\varepsilon(\%)$	$\Delta r(\%)$	$\Delta\varepsilon(\%)$	$\Delta r(\%)$	$\Delta\varepsilon(\%)$	$\Delta r(\%)$
Boiler	-32.46	108.9	-32.38	109	-32.31	109.2	-32.21	109.5
Turbine	-18.15	4936	-13.52	4150	-13.5	4149	-13.5	4151
$\dot{C}_{L,tot}$	25227 ₹/hr		22528 ₹/hr		22513 ₹/hr		22500 ₹/hr	
$\dot{C}_{D,tot}$	242144 ₹/hr		215425 ₹/hr		215287 ₹/hr		215162 ₹/hr	
$\dot{C}_P$	99720 ₹/hr		95580 ₹/hr		95688 ₹/hr		95832 ₹/hr	
$OBF = \dot{C}_P + \dot{C}_{L,tot} + \dot{C}_{D,tot}$	367091 ₹/hr		333533 ₹/hr		333488 ₹/hr		333494 ₹/hr	

Table 6.26 *Continue*

Variable	Case-IV		Case-V		Case-VI		Case-VII	
P_1	98 bar		100 bar		100 bar		100 bar	
T_1	505°C		505°C		505°C		510°C	
η_{ST}	0.85		0.85		0.85		0.85	
T_{24}	1500°C		1500°C		1510°C		1510°C	
Component	$\Delta\varepsilon(\%)$	$\Delta r(\%)$	$\Delta\varepsilon(\%)$	$\Delta r(\%)$	$\Delta\varepsilon(\%)$	$\Delta r(\%)$	$\Delta\varepsilon(\%)$	$\Delta r(\%)$
Boiler	-32.19	109.5	-32.05	109.8	-31.38	107.5	-31.29	107.9
Turbine	-13.5	4151	-13.5	4155	-13.52	4143	-13.5	4143
$\dot{C}_{L,tot}$	22469 ₹/hr		22431 ₹/hr		22113 ₹/hr		22100 ₹/hr	
$\dot{C}_{D,tot}$	214806 ₹/hr		214393 ₹/hr		210019 ₹/hr		209897 ₹/hr	
$\dot{C}_P$	95796 ₹/hr		95904 ₹/hr		95256 ₹/hr		95400 ₹/hr	
$OBF = \dot{C}_P + \dot{C}_{L,tot} + \dot{C}_{D,tot}$	333071 ₹/hr		332728 ₹/hr		327388 ₹/hr		327397 ₹/hr	

6.2.5.3 Results and Discussions

The results of the exergoeconomic optimization of steam power plant are given in Table 6.26. Table 6.27 represents a comparative study of the final cost optimal configuration with that of the existing configuration (base case). It is seen that the overall exergoeconomic cost of the product (electricity) is decreased by about 4.02 % (1.99 ₹/kWh to 1.91 ₹/kWh) with corresponding 4.17 % decrease (0.48 ₹/MJ to 0.46 ₹/MJ) in the fuel cost which resulted from the reduction in consumption of fuel. The cost of tapped steam is reduced from 395 ₹/1000 kg to 389.3 ₹/1000kg. The cost of exergy destruction is also decreased by 13.27 % and that of exergy loss is decreased by 12.34 %. Overall improvement in the system performance is realized by the increase in the exergetic efficiency by 7.64 %. If the taping steam is used a fuel for VAR system then the cooling cost will be reduced from 4853 ₹/hr to 2448 ₹/hr.

Table 6.27 Comparison between Base Case and Optimum Case for Steam Power Plant

Properties	Base Case	Optimum Case	% Improvement
Fuel Cost $\dot{C}_{F,tot}$	0.48 ₹/MJ	0.46 ₹/MJ	4.17
Product Cost $\dot{C}_P$	1.99 ₹/kWh	1.91 ₹/kWh	4.02
Steam Cost $\dot{C}_S$	395 ₹/1000 kg	389.30 ₹/1000 kg	1.45
Loss $\dot{C}_{L,tot}$	25227 ₹ /hr	22113 ₹/hr	12.34
Destruction $\dot{C}_{D,tot}$	242144 ₹/hr	210019 ₹/hr	13.27
Exergetic Efficiency ε	24.34 %	26.18 %	7.64

6.3 Comparison

A one to one comparison of the outcome of the exergoeconomic optimization of the existing AAVAR system using steam from the independent boiler as heat source, the first option of switch over of heat source to steam from HRSG of GT-HRSG system and the second option of switch over of heat source to tapped steam from steam power plant is carried out. The cost of steam generated in independent boiler is found to be 900 ₹/1000kg and thereby the cooling cost of AAVAR system is 1.36 ₹/sec. The alternative first option for steam generation such as GT-HRSG and tapped steam from steam turbine are identified in the fertilizer industry itself and compared in the Table 6.28.

Table 6.28 Comparison of Cost of Cooling for Options of Heat Sources

	GT-HRSG (Option – 1)	Tapped steam (Option – 2)
Steam cost ₹/1000kg	790	389
Cooling cost of AAVAR ₹/sec	1.086	0.68
Cost associated with exergy loss ₹/sec	0.617	1.470
Cost associated with exergy loss ₹/MWs	0.077	0.029

Table 6.28 compares the cooling cost of Option 1 and Option 2 examined in the present study. It is seen that the tapped steam from steam turbine is quite economical as fuel for AAVAR system compared to steam generated at GT-HRSG. The reason behind the difference is the cost of exergy loss from the system. In case of GT-HRSG, the exergy loss takes place in the form of exhaust gas at 177°C (station 7). The unit exergy cost associated with exergy loss is 0.617 ₹/sec (Refer Table 6.6). As the power generation capacity of GT-HRSG is 8 MW, the cost associated with loss per unit power generation is 0.077 ₹/MWs. While in the case of steam power plant, the exergy loss takes place in the form of exhaust gas from the boiler at 163.5 °C (station 26). The unit exergy cost associated with exergy loss is 1.47 ₹/sec (Refer Table 6.21). As the power generation capacity of steam power plant is 50 MW, the cost associated with loss per unit power generation is 0.029 ₹/MWs. The low exergoeconomic loss in steam power plant reduces the cost of power generation and tapped steam from steam turbine.

Since the second option of switch over form the existing heat source of the independent boiler to tapped steam of steam power plant is found to be the best techno-economically, it is proposed to switch over from the existing heat source of steam from independent boiler to tapped steam from 50 MW steam power plant. The saving in the steam cost per 1000 kg steam will be 511 ₹/1000 kg. The annual steam consumption in

AAVAR system is 90403200 kg/year. Therefore, the annual saving in the monetary term will be ₹ 46196035/-.

The switch over is possible only by laying down steam pipe to transport steam from the steam power plant to AAVAR plant which is about 1 km apart. To associate AAVAR system with steam power plant which is about 1 km far from AAVAR system, it is required to establish steam pipe line from steam power plant to AAVAR system. The tapping at steam turbine stage at 17 bar is made up of 6 inch carbon steel pipe of A106 Grade-B Seamless Schedule 40 IBR. It is suggested to extend same pipe line up to AAVAR system. The material cost of pipe is ₹ 1008 per meter length (Appendix-I) which includes supporting systems and bends. Therefore, the total cost of pipe for one km will be ₹ 1008000. The insulation cost will be ₹ 450 per meter length of pipe. Therefore, the total cost of insulation on 1 km pipe line will be ₹ 450000. The total installation cost including pipe material cost and insulation cost will be ₹ 1458000. The total saving in steam cost indicates that this installation cost can be recovered in 12 days only.