CHAPTER-3

SOLAR-COAL HYBRID POWER PLANT MODELING

In this chapter, general equations for solar field modeling, thermodynamic modeling, and the economic modeling of the solar-coal hybrid power plant are presented.

3.1. Solar Field Model

The performance parameters and thermal energy output from a PTC solar field is discussed in this section. The parabolic trough concentrating solar power technology is selected in this research work due to its proven technological maturity and ability to integrate into existing or new conventional power generating units. The study assumes steady-state simulation of the solar field for selecting important field design variables. The solar field component essential information flow representation is as shown in Figure 3.1. The input parameters required for the solar field model include the temperature of HTF at field inlet (°C), the DNI over the field (W/m^2), the HTF volumetric flow rate (m^3/s), the ambient temperature (°C), and the wind speed (m/s). At the outlet the solar field model returns the HTF outlet temperature, the rate of heat absorbed (Q_{abs}), heat loss from the field (Q_{hl}), and the collector field efficiency as a whole (Patnode 2006).

3.1.1. Fundamental solar angles

The use of solar energy for practical purposes requires sound knowledge and understanding of the earth's relative motion about its orbital axis and its movement with respect to the Sun. This is because the Sun's position is continuously changes in every hour of the day, directly

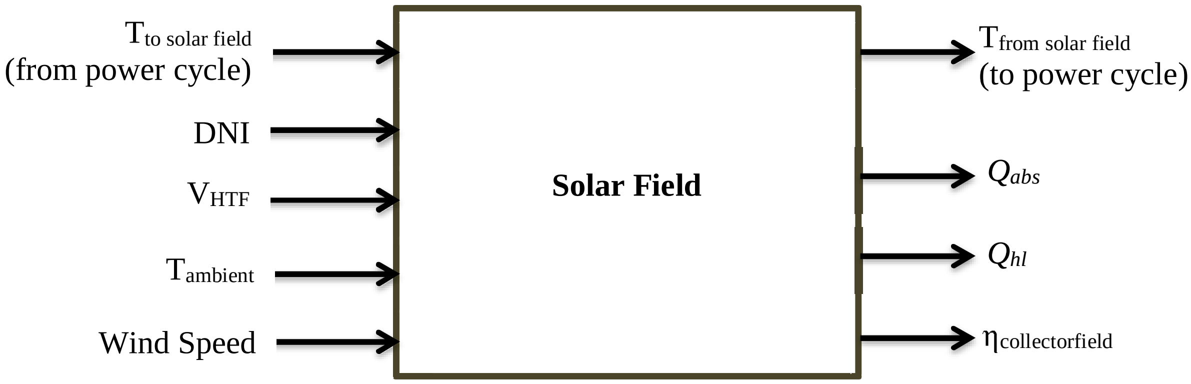

Figure 3.1 The basic information flow representation for solar field (Patnode 2006).

affecting the intensity and direction of solar radiations falling on solar energy collection systems. Therefore, the solar collector must be continuously maneuvered to track the movement of the Sun. In the case of CSP technologies, for a particular location on earth, the position of the Sun is determined by fundamental solar angles. Further, the earth does not revolve around the Sun in a perfectly circular path, but it revolves around in an elliptical path. Consequently, the length of the day of regular 24 hours as measured by a uniform clock does not remain constant. Due to the eccentricity effects and tilt in the earth's rotation axis, the day length varies. To account for this variation, a term known as Equation of Time (*EOT*) is used to rectify this discrepancy. The EOT along with various basic solar angles such as declination angle (δ), hour angle (ω), latitude angle (ϕ), collector tilt angle (β), and incidence angle (θ) can be calculated using the following equations (Duffie and Beckman 2013).

$$\begin{aligned} EOT = {} & 229.18(0.000075+0.001868\cos(B)-0.032077\sin(B) \\ & -0.014615\cos(2B)-0.04089\sin(2B)) \end{aligned} \tag{3.1}$$

Here, B is the day angle in degrees and can be obtained as follows

$$B=\frac{360}{365}(n-1) \tag{3.2}$$

The declination angle (δ) can be calculated as follows:

$$\delta = 23.45 \sin\left(360 \times \frac{284+n}{365}\right) \tag{3.3}$$

Here, in equations 3.2 and 3.3, '*n*' is the particular day of a year.

The hour angle (ω) can be obtained as follows

$$\omega = \left(ST - 12\right) \times 15° \tag{3.4}$$

Here, *ST* denotes solar time in hours.

The collector tilt angle (β) can be determined as (Duffie and Beckman 2013)

$$\tan\beta = \frac{\cos\delta \sin\omega}{\sin\phi \sin\delta + \cos\phi \cos\delta \cos\omega} \tag{3.5}$$

Where the latitude angle (ϕ) of a location is defined as the angle made by the line drawn from the center of the earth to the point of location with the equatorial plane, and it varies from 90° in the North Pole to - 90° in the South Pole in Antarctic circle (Shimeles 2014).

While designing and analyzing the performance of tracking and non-tracking solar collectors, the role of incidence angle is very crucial. The beam radiation is reduced by a factor of the cosine incidence angle in actual solar energy collectors. For a parabolic trough plane rotated about N-S axis with continuous tracking in E-W direction to minimize the incidence angle, θ can be determined as follows (Duffie and Beckman 2013).

$$\cos\left(\theta\right) = \sqrt{\left(\cos^2\theta_z + \cos^2\delta \times \sin^2\omega\right)} \tag{3.6}$$

In actual practice, for a PTC the incidence angle has to be rectified for deviations in the higher angular incidence displacements occurring due to the reflection and absorption losses from the receiver glass envelope. As the incidence angle increases, these losses also increase. The incidence angle modifier (*IAM*) is proposed to correct the deviations in angular displacement in order to account for these losses. For a given type of solar collector, the IAM (k) is unique. The IAM (k) for Luz solar collector of second-generation (*LS-2*) is obtained as (Patnode 2006).

$$IAM = k = \cos(\theta) + 0.000884(\theta) - 0.00005369(\theta)^2 \tag{3.7}$$

For Euro-trough solar collector, the IAM (k) can be determined as (Shimeles 2014)

$$IAM = k = \cos(\theta) - 2.859621 \times 10^{-5} (\theta)^2 - 5.25097 \times 10^{-4} (\theta) \tag{3.8}$$

3.1.2. Parabolic trough collector field

The solar field model is based on a steady-state condition, whose thermal output depends on the absorbed solar radiation incident on the collector reduced by the solar field's losses. The total irradiative solar energy incident on the collector area is calculated as (Hou et al. 2012).

$$Q_{LD} = x.DNI.\cos(\theta).A_c \tag{3.9}$$

Here A_c is the aperture area of the solar collector field (m^2) and x is row shadow factor. The solar energy absorbed by the absorber tubes can be determined using the following relation.

$$Q_{abs} = Q_{LD} \times k \times \eta_o \tag{3.10}$$

where k is the incidence angle modifier (*IAM*) and η_o is the optical efficiency of solar collector field accounting all the losses because of optics and imperfections. According to the literature (Forristall 2003, Patnode 2006) the most suitable value of η_o is 0.7133, including the cleanliness correction factor. While k can be calculated with the help of equations 3.7 or 3.8 depending upon the type of solar collector used.

The total heat loss in the solar field is a combination of both heat losses in the absorbing tube and heat losses in the piping system. These losses can be determined from the corresponding empirical formulas. For calculating absorber tube heat losses per unit aperture area, i.e., the heat losses from heat collection element (*HCE*), the empirical correlation available in literature is used. For reference, it is written below (Patnode 2006, Hou et al. 2012).

$$Q_{loss\_HCE} = \frac{A+B+C}{(T_o - T_i)W} \tag{3.11}$$

Where,

$$A = a_0 (T_o - T_i) \tag{3.12}$$

$$B = \frac{a_1 (T^2{}_o - T_i^2)}{2} + \frac{a_2 (T^3{}_o - T_i^3)}{3} + \frac{a_3 (T^4{}_o - T_i^4)}{4} \tag{3.13}$$

$$C = DNI \left[b_0 (T_o - T_i) + \frac{b_1 (T^3{}_o - T_i^3)}{3} \right] \tag{3.14}$$

The values of coefficients $a_0, a_1, a_2, a_3, b_0, b_1$, and standard deviations for vacuum annulus, are given in Table 3.1 (Patnode 2006).

The solar field piping heat loss per unit aperture area is determined as a function of the difference between the average solar field temperature and the ambient air temperature.

$$Q_{loss\_piping} = 0.01693 \Delta T - 0.0001683 \Delta T^2 + 6.78 \times 10^{-7} \Delta T^3 \tag{3.15}$$

Where

$$\Delta T = \frac{T_o + T_i}{2} - T_a \tag{3.16}$$

Table 3.1:

Coefficients for heat loss in HCE (Patnode 2006)

Coefficients	Values	Standard Deviation
a_0	-9.463033	0.8463850
a_1	0.3029616	0.01454877
a_2	-0.001386833	0.00007305717
a_3	0.000006929243	0.0000001070953
b_0	0.0764961	0.0005293835
b_1	0.0000001128818	0.000000006394787

During solar field operation, the thermal piping losses to and from the solar field are generally small of the order of 10 W/m^2 or even less (Patnode 2006).

The net useful energy collected by the heat transfer fluid (*HTF*) over the field is obtained as the difference between the heat absorbed by the fluid into the absorber tubes and the sum of receiver heat loss and piping heat loss. Thus, net useful energy collected by HTF over the field is determined using the following equation.

$$Q_u = Q_{abs} - \left(Q_{loss\_HCE} + Q_{loss\_piping}\right)A_c \tag{3.17}$$

3.2. Power Cycle Component Model (Thermodynamic Model)

In this section, the schematics of models of essential components such as boiler, turbine, deaerator, condenser, closed feedwater heater, and pump are presented. The general governing equations of mass and energy balance across these essential components are also given in this chapter.

3.2.1. Turbine

The steam turbine produces work output in the successive stages when high pressure and high-temperature steam expand in different HP, IP, and LP turbine stages. The actual work output of the turbine is calculated using a simple control volume energy balance equation. The simple thermodynamic model of the turbine is as shown in Figure 3.2.

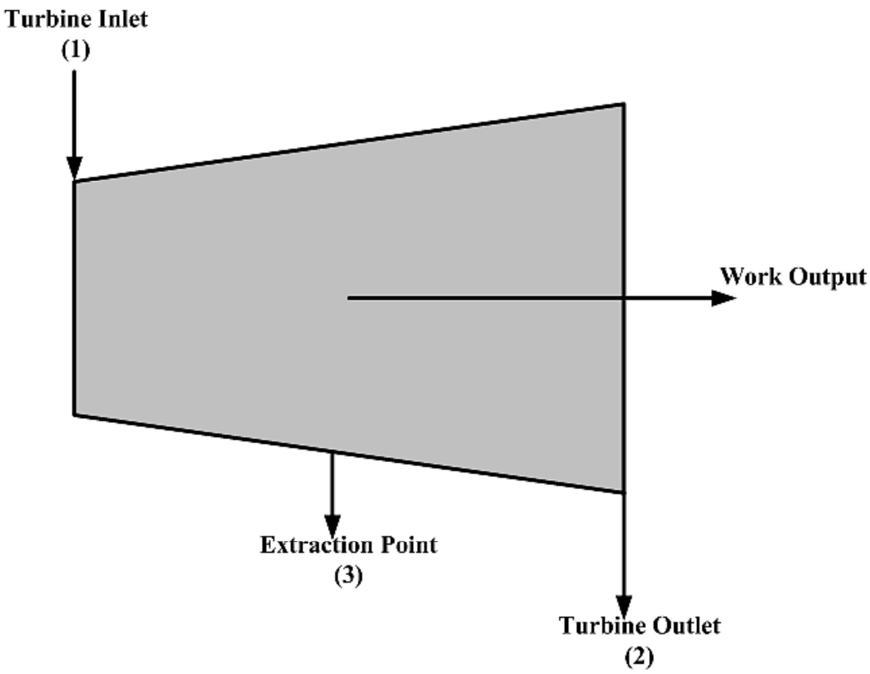

Figure 3.2: Thermodynamic model of turbine inlet and outlet.

$$\dot{W}_{out} = \dot{m}_1 . h_1 - \left(\dot{m}_2 . h_2 + \dot{m}_3 . h_3 \right) \quad (3.18)$$

3.2.2. Boiler

The boiler is the energy input device in the power cycle where two mass flows i.e., the mass flow of feedwater and cold reheat steam, do not mix together. The mass balance across the boiler is given in equation 3.19. Figure 3.3 shows the boiler inlet and outlet model.

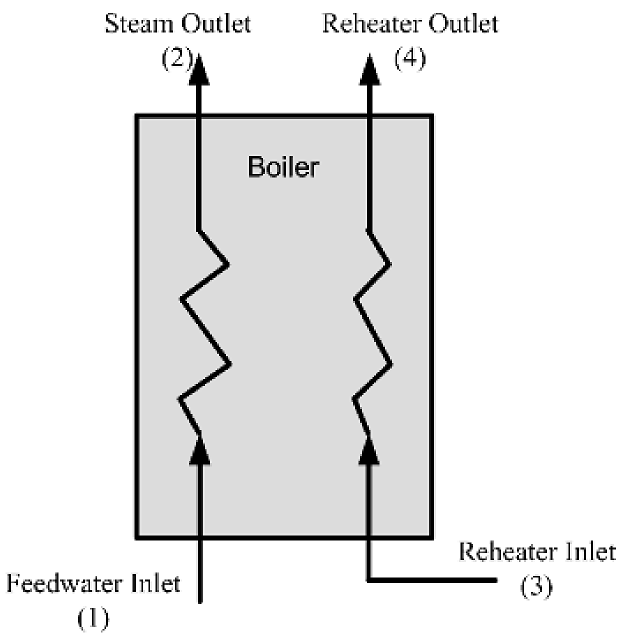

Figure 3.3: Boiler inlet and outlet model.

$$\begin{cases} \dot{m}_1 = \dot{m}_2 \\ \dot{m}_3 = \dot{m}_4 \end{cases} \tag{3.19}$$

The boiler heat load is calculated as follows

$$Q_{boiler} = \dot{m}_1 \times (h_2 - h_1) + \dot{m}_3 \times (h_4 - h_3) \tag{3.20}$$

3.2.3 Deaerator

The deaerator is an open type feedwater heater which is also known as a mixing chamber where there are three inlet streams and one outlet stream. The deaerator model with inlet streams and outlet stream is shown in Figure 3.4. The mass balance in the deaerator can be determined using the following equation.

$$\dot{m}_4 = \dot{m}_1 + \dot{m}_2 + \dot{m}_3 \tag{3.21}$$

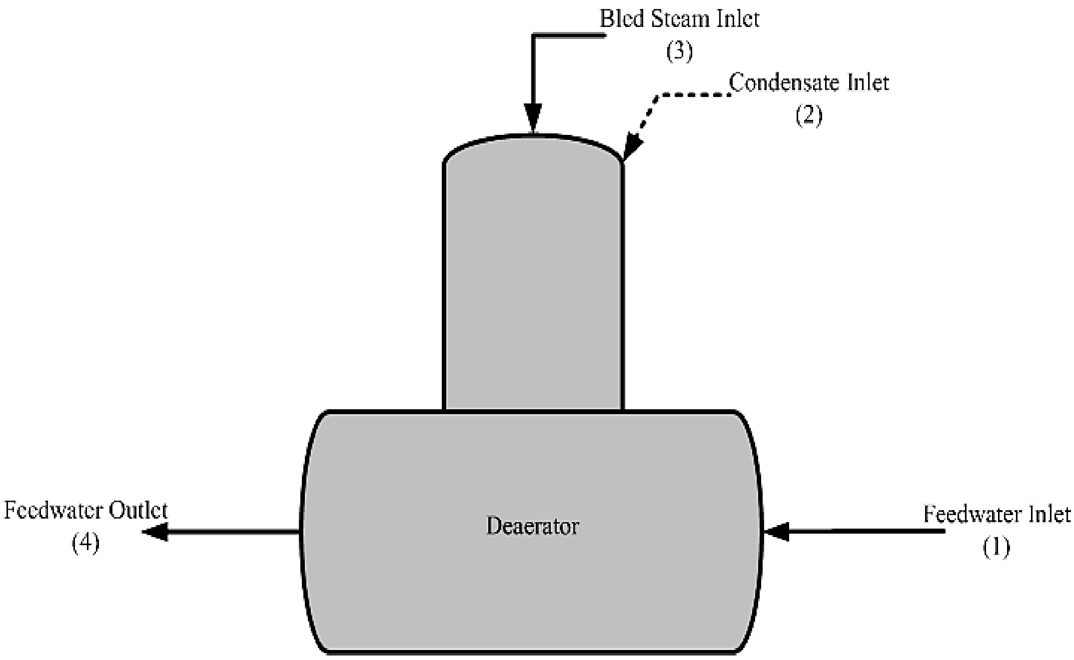

Figure 3.4: Deaerator inlet and outlet model.

3.2.4 Condenser

The steam condenser's purpose is to condense the exhaust steam from the turbine so that the working fluid returns to the liquid state and thus can be pumped back to the boiler. The exhaust steam condensation is carried out using a massive amount of cooling water usually taken from the nearby river or lake. The mass balance in the condenser is obtained as follows.

$$\begin{cases} \dot{m}_1 = \dot{m}_2 \\ \dot{m}_4 = \dot{m}_3 + \dot{m}_5 \end{cases} \tag{3.22}$$

3.2.5 Closed feedwater heater

The closed feedwater heater is a type of heat exchanger in which the bled steam from the turbine is used to preheat the working fluid, thus causing the plant's power output to decrease but increases the cycle efficiency. The inlet and outlet model of the closed feedwater heater is shown in Figure 3.6. The mass balance across the closed feedwater heater can be determined using the following equation with respect to Figure 3.6.

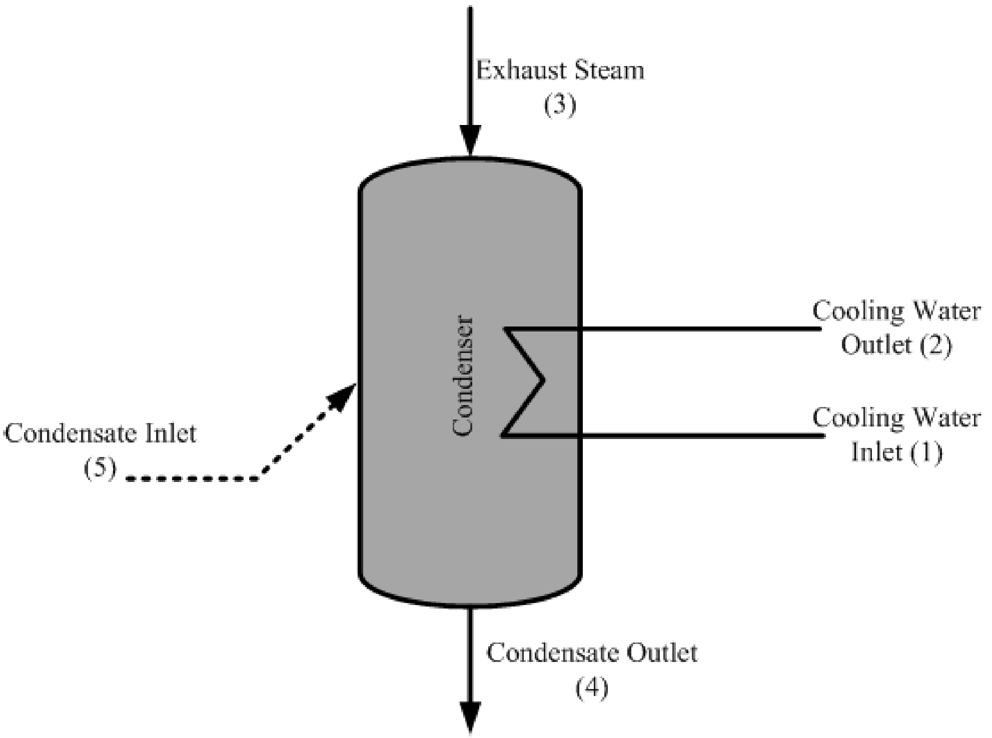

Figure 3.5: Condenser inlet and outlet model.

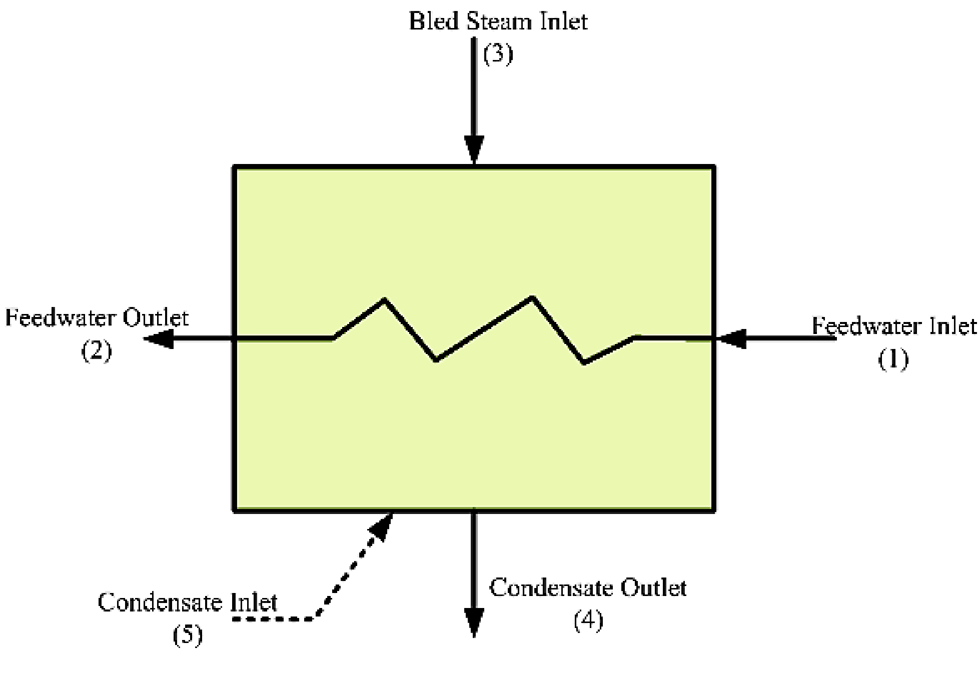

Figure 3.6: Inlet and outlet model of closed feedwater heater.

$$\begin{cases} \dot{m}_3 + \dot{m}_5 = \dot{m}_4 \\ \dot{m}_1 = \dot{m}_2 \end{cases} \tag{3.23}$$

3.2.6 Pump

The pump's function is to increase the pressure of the working fluid in a power plant cycle. The condensate extraction pump provided at the outlet of the condenser increases the pressure of feedwater from condensing pressure to the low feedwater heater pressure. At the same time, the boiler feed pump provided after the deaerator raises the pressure of feedwater up to the boiler inlet pressure. The inlet and outlet model of the pump is shown in Figure 3.7, and corresponding work input to the pump can be obtained as follows.

$$\dot{W}_{1-2} = \dot{m} \times (h_2 - h_1) \tag{3.24}$$

3.3. Solar-Coal Hybrid Plant Performance Parameters

In this study, PTC solar technology is considered for integrating solar energy into a conventional coal-based power plant. The various thermodynamic performance parameters of the solar field are calculated using the equations given below:

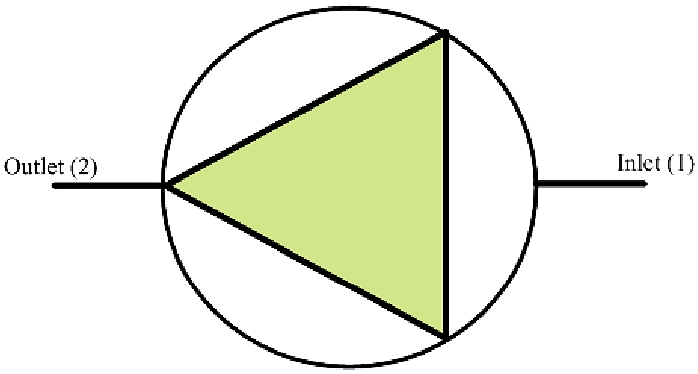

Figure 3.7: Pump inlet and outlet model.

The input energy to the solar field ($\dot{Q}_s$) is evaluated as

$$\dot{Q}_s = \frac{\dot{Q}_c}{\eta_c} \tag{3.25}$$

Here $\dot{Q}_c$ is the output energy of the PTC solar field (MW_{th}) and η_c is the collection efficiency of the solar collector. In this study, the collection efficiency of PTC is taken as 60% (Pai 1991). The output energy of the PTC solar field is calculated as:

$$\dot{Q}_c = \dot{m}.\Delta h \tag{3.26}$$

Here $\dot{m}$ is the feedwater mass flow rate (kg/s) and Δh is the specific enthalpy gain (kJ/kg). The collector area needed to transfer the output energy is evaluated as:

$$A_c = \dot{Q}_c \Big/ (DNI \times \eta_c) \tag{3.27}$$

Here, direct normal irradiance (*DNI*) is in W/m^2. The land needed for the solar collector field is usually three times the collector area (Pai 1991).

Plant energy efficiency (η_I) can be obtained as:

$$\eta_I = \dot{W}_{net} \Big/ \left(\dot{m}_f \times GCV \right) \tag{3.28}$$

Here $\dot{W}_{net}$ is the net electrical output (kW_e) and $\dot{m}_f$ is the mass flow rate of fuel (kg/s). GCV is the gross calorific value of the fuel (kJ/kg).

The energy performance index (*EnPI*) in power boosting mode is defined as:

$$EnPI = \Delta P_{excess} \Big/ \dot{Q}_s \tag{3.29}$$

Solar contribution can be calculated as:

$$\text{Solar Contribution},\ (\%) = \frac{\dot{Q}_s}{Q_b + \dot{Q}_s} \times 100 \tag{3.30}$$

$\dot{Q}_s$ is the input solar energy (MW_{th}) transferred to feedwater and Q_b is the heat input of the steam generator (MW_{th}).

3.4. Economic Model

In this section, the economic model is developed using the methodology followed by previous studies available in the literature. For evaluating the annualized cost of electricity (*ACoE*) and the Levelized cost of electricity (*LCoE*) generation, the procedure followed by Suresh et al. is adopted in this study. The discount rate is considered as 12%, and the power plant life is taken as 25 years. The plant capacity factor (*PCF*) is taken as 0.85 and auxiliary power consumption (*APC*) as 7.5%. The cost of capital/unit (*CC*) can be found as:

$$CC = TCC/P_G \tag{3.31}$$

Here TCC is the total cost of capital (*USD*) and P_G is the generator output (kW).

The capital recovery factor (*CRF*) is evaluated as:

$$CRF = \frac{d \times [1+d]^n}{\{[1+d]^n - 1\}} \tag{3.32}$$

Here d is the discount rate, and n is plant life (years).

The annualized capital cost (*ACC*)/kW can be calculated as:

$$ACC = CC \times CRF \tag{3.33}$$

Net energy generated annually (P_{Net}) can be obtained as:

$$P_{Net} = 8760 \times PCF \times (1 - APC) \tag{3.34}$$

Fixed capital cost/unit (*FCC*) can be evaluated as:

$$FCC = ACC / P_{Net} \tag{3.35}$$

In this research work, the fixed operation & maintenance cost (*FOM*) is considered as 2.5% of the capital cost (Sathaye and Phadke 2006).

Fixed O&M cost/unit can be evaluated as:

$$C_{FOM} = FOM / P_{Net} \tag{3.36}$$

The cost of fuel/unit (C_F) can be obtained as:

$$C_F = (FC \times UHR_{Net}) / GCV \tag{3.37}$$

Here UHR_{Net} is the net unit heat rate. In this work, the variable O&M cost (C_{VOM}) is considered the same as fuel cost per unit of electricity generated.

Total variable cost/unit (C_V) can be calculated as:

$$C_V = C_F + C_{VOM} \tag{3.38}$$

The annualized cost of electricity (*ACoE*) generation can be obtained as

$$ACoE = FCC + C_{VOM} + C_V \tag{3.39}$$

The effect of escalation in annual fuel and O&M cost using levelizing factor (*LF*) is taken into account for calculating ACoE.

Equivalent discount rate with escalation (d_e) can be obtained as:

$$d_e = d - e/1 + e \tag{3.40}$$

The LF is evaluated as:

$$LF = \left[\frac{(1+d_e)^n - 1}{d_e \times (1+d_e)^n}\right]\left[\frac{d \times (1+d)^n}{(1+d)^n - 1}\right] \tag{3.41}$$

An escalation rate (e) of 2% in variable cost and fuel/O&M-fixed is considered for economic investigation (Suresh et al. 2010).

The levelized fuel and O&M cost (C_L) can be calculated as:

$$C_L = LF \times (C_{FOM} + C_V) \tag{3.42}$$

The levelized cost of electricity generation (*LCoE*) can be obtained as:

$$LCoE = FCC + C_L \tag{3.43}$$

The simple payback period (*SPP*) of the plant under study can be obtained as:

$$SPP = CC/(P_{Net} \times ACoE) \tag{3.44}$$

3.5. Problem Description

Based on the information and knowledge acquired from the introduction and literature survey, the present research work focuses on integrating solar energy into existing conventional coal-based power plants to mitigate climate change and address global warming of the planet earth. This research emphasizes the advantages of hybridization of solar energy with conventional coal-fired thermal power plants through fuel-saving and power-boosting approaches. To achieve the present research work's specific objectives stated in the introduction chapter, the two case studies have been performed on solar-coal hybrid power plants in the subsequent chapters, i.e., chapter 4 and chapter 5, followed by chapter 6 on the policy status of solar energy in India.

The detailed description of case study – 1 given in Chapter–4 (Results and discussion part–1) will focus on integrating solar thermal energy for feedwater preheating into an existing 330 MW subcritical coal-based power plant. In this chapter, three different scenarios (mentioned in chapter–4) for substituting extraction steam in reference plant under study by solar energy will be investigated. This case study presents the economic and environmental analysis of a subcritical solar integrated coal-based power plant for all three scenarios considered.

The detailed description of case study – 2 given in Chapter–5 (Results and discussion part–2) will focus on integrating solar thermal energy for feedwater preheating into an existing 660 MW supercritical coal-based power plant. In this chapter, three different schemes (mentioned in chapter–5) for substituting extraction steam in reference plant under study by solar energy are investigated. This case study will present the economic and environmental analysis of a supercritical solar integrated coal-based power plant for all three scenarios considered.

"Policy interventions" regarding solar-thermal hybrid technologies in India will discuss the comparative status of all the states' solar policies in Chapter–6. This chapter's focus is to identify the present status of solar policies promoting the exploitation of concentrated solar power (CSP) technologies in India and highlight the barriers and challenges hindering CSP technologies' progress. This chapter also provides future recommendations for hybridizing solar energy with existing conventional thermal power plants through lucrative policy enablers.