CHAPTER 3

DESIGN OF THE STUDY

This chapter introduces overall design of the study, which includes the methodology adopted for carrying out the work and various phases of this research. Details of the work done in each phase along with the tools, techniques and models used have also been covered in detail here.

3.1 Methodology

This study has been carried out with the purpose of developing an effective and efficient water desalination system for rural areas of India with low cost. The research has been carried out employing an approach similar to traditional waterfall methodology. The framework consists of requirements, design, development, testing followed by validation as shown in figure 3.1.

![Figure 3.1 Waterfall Model](image)

The study begins by working out the quantity of drinking water required for a family in rural areas. Subsequently, the system has been designed to make available this quantity of water and system components such as solar collector, absorber and condenser have been designed. To ensure an effective and cost efficient design, experimental analysis and simulations were employed. Designed components were then fabricated and tested by varying different parameters such as quantity of water, size of absorber, and intensity of
solar radiation. Finally, a mathematical model was developed and validated using experimental data.

3.2 Overview of the Study

The present study comprises of three major parts namely ‘design’, ‘optimization’ and ‘fabrication’. Design deals with developing a parabolic concentrator, absorber and condenser. Optimization of concentrator is done on the basis of reflectivity of material, quantity of water and intensity of solar radiations whereas optimization of absorber is done on the basis of available focal area and material used by an absorber. The analysis made on both concentrator and absorber is validated using Ray tracing software and by using ANSYS. Condenser is also optimized using theoretical, experimental and CFD analysis. The concentrator, absorber and condenser are then manufactured and tested. Based on the waterfall model methodology, the research work has been carried out in following four phases,

Phase I  Literature review for clarifying the context
Phase II  Design, optimization and fabrication of the concentrator and the absorber
Phase III  Design, optimization and fabrication of the condenser
Phase IV  Testing and validation of the system

3.2.1 Literature review and clarifying the context

First of all, a review of literature on finding different solar desalination techniques has been carried out. It was found that all the commercialized systems are dependent on electricity. By review of available small desalination plants, various techniques and approaches used for design and manufacture of solar systems, design of a small desalination plant using solar energy for a small family for providing safe drinking water was conceptualized.

3.2.2 Design, optimization and fabrication of the concentrator

The parabolic concentrator is designed and manufactured first of all. Then using parabolic concentrator and with the available focal area, an absorber is designed. Following step-by-step approach was used for design, optimization and fabrication of the concentrator.
a. Deciding the size of parabolic concentrator

Size of parabolic concentrator was decided on the basis of heat energy required at the focal area which depends on the quantity of water that is to be evaporated. Other parameters which affect the size of the concentrator are the intensity of solar radiations and reflectivity of the material. The bare minimum requirement of drinking water for a family was considered for designing the system. Heat energy required to evaporate the quantity of water is calculated using equation 3.1.

\[ Q = \frac{m \cdot C_p \cdot \Delta T + m \times \text{Latent heat}}{\text{time}} \]  

Where \( Q \) is the amount of energy required, \( m \) is mass of water, \( \Delta T \) is the increase in the temperature over a time period and \( C_p \) is specific heat of water. If \( I \) is the total solar radiations in W/m², and \( A_c \) is the area of concentrator in m² and considering efficiency of concentrator as 30% to 35% (Amos, 2009; Tay et al., 2012) the net heat energy available, \( Q_n \) at the focal area is calculated by using equation 3.2.

\[ Q_n = (I \times A_c \times \rho) \times 0.3 \text{ W} \]  

Average value of intensity of solar radiations is calculated using theoretical formulae (Sukhatme, 1996; Spencer, 1971). \( n_2 \) is the day angle and can be calculated using equation 3.3.

\[ n_2 = \text{dayangle} = \frac{2\pi}{365}(n_1 - 1) \]  

The equation of time \( t_2 \), is then calculated using equation 3.4.

\[ t_2 = (229.18) \times (0.000075 + 0.001868 \cos n_2) - 0.032077 \sin n_2 - 0.014615 \cos (2n_2) - 0.040849 \sin (2n_2) \]  

After finding the equation of time, then time offset, \( t_5 \) is calculated using equation 3.5.

\[ t_5 = t_2 - 4[\text{longitude}] + 60[\text{time zone}] \]  

Hour angle (\( \omega \)) is calculated using equation 3.6.

\[ \omega = \frac{hr \times 60 + \text{min} + t_5}{4} - 180 \]  

The value of declination angle \( \delta \) was calculated using equation 3.7.

\[ \delta = 23.45 \sin \frac{360[284 + n_1]}{365} \]  

40
$H_o$ is monthly average of the hourly extra-terrestrial radiation on a surface and was calculated by using equation 3.8.

$$H_o = \frac{24}{\pi} \times 1367 \times (\omega \times \sin(\text{latitude}) \times \sin(\delta) + \cos(\text{latitude}) \times \cos(\delta) \times \sin(\omega)) \times \left(1 + 0.033 \times \cos\left(n \times \frac{360}{365}\right)\right)$$  \hspace{1cm} \text{3.8}

Zenith angle $\theta$ is calculated using equation 3.9.

$$\theta = \cos^{-1}\left[\cos(\text{latitude}) \cos(\delta) + \sin(\text{latitude}) \sin(\delta) \cos(\omega)\right]$$  \hspace{1cm} \text{3.9}

Day length which is dependent on the zenith angle is then obtained using equation 3.10.

$$\text{Day length} = \frac{2}{15} \cos^{-1}[\tan(\text{latitude}) \times \tan(\theta)]$$  \hspace{1cm} \text{3.10}

Sunshine factor is calculated using equation 3.11

Sunshine factor = average sunshine / day length

$H_g$ is the monthly average of the daily global radiation on a horizontal surface and is calculated using equation 3.12.

$$H_g = (0.31 + 0.43 \times \text{sunshine factor}) \times 1036$$  \hspace{1cm} \text{3.12}

Coefficient $a$, $b$ and $c$ are determined using equation 3.13 to 3.15.

$$a = 0.409 + 0.5016 \sin\left(\omega - \frac{\pi}{3}\right)$$  \hspace{1cm} \text{3.13}

$$b = 0.6609 - 0.4767 \sin\left(\omega - \frac{\pi}{3}\right)$$  \hspace{1cm} \text{3.14}

$$c = a + b \cos(\omega)$$  \hspace{1cm} \text{3.15}

The value of monthly average of the hourly extra-terrestrial radiation on a horizontal surface, $I_o$ is determined using equation 3.16.

$$I_o = 1367 \left(1 + 0.033 \times \cos\left(n \times \frac{360}{365}\right)\right) \cos\theta$$  \hspace{1cm} \text{3.16}

The values of sunshine factor and zenith angle are used to find the hourly global radiations, $I_g$ using equation 3.17.

$I_g$ is calculated as,$I_g = I_o \times c \times \frac{H_g}{H_o}$  \hspace{1cm} \text{3.17}
MATLAB program was then developed which provided the intensity of radiations I for all months. The average value of solar radiations was then considered for designing. As far as reflectivity of material is considered, a search was made and samples of four anodized aluminum reflectors were collected. These samples were tested for reflectivity in the labs at University of Pune. The cost effective reflective material with good reflectivity was finally selected. The data collected was then substituted in equation 3.2 to find out the area of concentrator. The diameter of the concentrator was calculated using equation 3.18.

\[ d = \sqrt{\frac{4}{\pi}} \times A_c \]  

3.18

**b. Fabrication of parabolic concentrator**

Curve tracing of the concentrator was done using parabolic equation \( x^2 = 4ay \). Aluminum foils were cut in the trapezoidal form and fitted over the metallic frame and the concentrator was then fabricated. Temperature measured at the focal area with the fabricated concentrator was found to be 250\(^0\)C to 260\(^0\)C in good sunshine conditions.

**c. Testing and validation of the concentrator**

The optical efficiency of a concentrator is defined as the ratio of the energy absorbed by the absorber to the energy incident on the concentrator aperture. Optical efficiency factor \( F'\eta_o \) gives, theoretical upper limit of the overall efficiency of the concentrator. Using total thermal capacity of the cooking pot, \( (m \, C_p) \) the value of the heat loss factor \( F'U_L \) of the cooker was evaluated using equation 3.19 (Supple and Thombre, 2013).

\[ F'U_L = \frac{m \, C_p}{A_{abs} \, \tau_o} \]  

3.19

Where, \( \tau_o \) is the time-constant for cooling and \( A_{abs} \) is the area of the cooking pot or absorber. The optical efficiency factor \( F'\eta_o \) of the parabolic concentrator was calculated for each 10-minutes interval of the test using equation 3.20.

\[ F'\eta_o = \frac{(F'U_L) \times A_{abs}}{A_c} \left[ \frac{(t_w - t_a)}{(I)} - (t_{wi} - t_a) \times e^{-\frac{t-w}{\tau_o}} \right] \]  

1 - e^{-\frac{t-w}{\tau_o}} \]  

3.20
Where, \(t_{wi}\) is the temperature of water in the cooking pot at the beginning of the interval in \(^0\text{C}\), \(t_{wi}\) is the temperature of water in the cooking pot at the end of the interval, in \(^0\text{C}\). \(\tau\) is the duration of the interval (10 minutes or 600 s), \(I\) is the intensity of the beam radiation incident on the aperture of the concentrator, averaged during the interval in W/m\(^2\) and \(t_a\) is the ambient air temperature during the interval, in \(^0\text{C}\).

The optical efficiency obtained in the previous section is analyzed in another manner also. The difference between water temperature and ambient temperature \((\Delta T)\) is decremented over an interval of 5 minutes. The value of the instantaneous thermal efficiency is then calculated using equation 3.21.

\[
\eta_i = \frac{m \times C_p \times \Delta T \times t \times A_c \times I}{t \times A_c \times I} \tag{3.21}
\]

The factor \(\Delta T/I\) is determined and plotted against the instantaneous efficiency. A least square regression straight line is fitted through these data points using equation 3.22.

\[
Y = mX + c \tag{3.22}
\]

Where, \(m\) is the slope of the line, \(c\) is the \(Y\) intercept of the line. This equation is compared with the equation 3.23 given below.

\[
\eta_i = \eta_o - (U_{lc}) (\Delta T/I) \tag{3.23}
\]

Where, \(\eta_o\) is the optical efficiency of the system, and \(U_{lc}\) is the heat loss factor of the concentrator. Thus, slope of the line is the heat loss factor of the concentrator and \(y\)-intercept is the optical efficiency of the system (Sukhatme, 1996).

### 3.2.3 Design optimization and fabrication of the absorber

Design of the absorber was taken up by selecting the material first. As the absorber would be using sea water for evaporation, the material should have high corrosion resistance and must withstand high temperature. Commonly used materials were studied and stainless steel, SA-312 Gr 304 was chosen as the material of the absorber. The next task was to find the size of the base of an absorber (ASME- BPVC, 2010).

Experiments were conducted with wooden ply and the size of burned ply indicated the minimum diameter of focal area which was found to be 0.13 m. To determine the travel of focal area, a
steel plate was kept at focal area and tracing of the incident rays was marked on the plate over a period of time. It was observed that for the same tracking position, the area of brightness gets shifted. After 20 minutes of time, the focus got diffused and shifted over a length of 0.3 m. On the basis of the experimentation, the diameter of the absorber was worked out. Validation was done using Tonatiuh software which is employed to trace the sun rays using Monte – Carlo simulation technique. Absorber diameter, parabolic dish of required diameter and focal length, zenith angle and azimuthal angle were specified over a period of 5 minutes. It was observed that after 20 minutes, the rays get diffused and focal region gets shifted completely. After finalizing the diameter the next task was to decide the height of the absorber. Minimum height $H$ is calculated using equation 3.24.

\[
V = \frac{\pi}{4} D^2 H
\]

Where, $V$ is the volume and $D$ is the diameter of the absorber.

Effect of height of the absorber on the quality of condensate water was determined using experimental analysis. Five kg of water was taken in a five liters black coloured stainless steel cooker of 0.3 m diameter which acted as an absorber. Steam coming out of the condenser at $70^\circ$C to $75^\circ$C was condensed using cloth for first hour and 300 ml of water was collected. Then water was heated till it reached a temperature of $100^\circ$C. When the level of water was almost one-third and steam coming out at the condenser was condensed, 300 ml of water was collected. These samples were tested in Hydrotech laboratory, Pune and it was observed that quantity of dissolved solids, salts like chlorides, sulphates and metals like iron were significantly less in sample taken at one-third height. This pointed towards the fact that the absorber should be designed in such a way that two-thirds of the height kept vacant above the water level. The size of the absorber was then finalized with 0.3 m diameter and 0.33 m height. The surface area of the absorber was calculated to be $0.424 \text{ m}^2$.

The concentration ratio, $C$ is defined as ratio of the concentrator surface area to the absorber surface area and indicates the ability to focus radiations. The value of $C$ is calculated using equation 3.25.

\[
C = \frac{A_c}{A_{abs}}
\]
Theoretical value of C ranges from 10 to 40 for good performance and minimum tracking is required for C equal to 10 (Kaddour, 2012; Rubo et al., 2007). For the designed dimensions, concentration ratio was within the range hence dimensions of the absorber were finalized.

Finally, the thickness of the absorber was calculated using concept of cylindrical shell by ASME boiler and pressure vessel code 2010 to ensure that the maximum allowable working pressure of cylindrical shells should be less than the pressure calculated by longitudinal stress and circumferential stress using equations 3.26 and 3.27 (ASME- BPVC, 2010).

For longitudinal stress, \[ t = \frac{PR}{2S+1.4P} \] 3.26

Circumferential stress,

For, \[ t = \frac{PR_0}{S+0.4P} \] 3.27

Before manufacturing, transient thermal analysis was performed at an interval of five minutes and temperature distribution was observed. Material properties of stainless steel and water were provided in engineering data module of ANSYS. Basic geometric model of the absorber was created using design module of ANSYS workbench. Solar radiations were supplied as heat input. Heat loss through convection and radiation losses for analysis, was calculated as under (Keith, 2001).

1) Rayleigh’s number was calculated using equation 3.28.

\[ Ra = \frac{g \times \beta \times \rho \times C_p \Delta T \times L^3}{\nu \times k} \] 3.28

Where \( g \) is acceleration due to gravity, \( \beta \) is coefficient of thermal expansion, \( C_p \) is specific heat at constant pressure, \( \rho \) is density, \( k \) is thermal conductivity, \( \nu \) is viscosity, \( \Delta T \) is rise in temperature and \( L \) is the characteristic length. For vertical cylinder, diameter (d) is considered as the characteristic length whereas, for top and bottom circular plates, \( L \) is equal to \( d/4 \).

2) Nusselt’s Number, \( Nu \) for various geometries is given by equations 3.29 to 3.32.

a) For vertical cylinder \( Nu = 0.56 \times (Ra)^{1/4} \) 3.29

b) For top surface \( Nu = 0.54 \times (Ra)^{1/4} \) 3.30

c) For bottom flat surface \( Nu = 0.27 \times (Ra)^{1/4} \) 3.31

3) Heat transfer coefficient was then calculated using the formula 3.32.
The overall heat loss rate from the absorber was calculated by considering the heat loss from convection, \( Q_c \), and heat loss from radiation. Heat loss to surroundings occurs from top, bottom and vertical sides by natural convection. Heat flow by convection was calculated using equation 3.33.

\[
Q_c = h \times A_{abs} \times \Delta T
\]  

3.33

Where, \( h \) is the heat transfer coefficient and \( \Delta T \) is the change in temperature. The net heat transfer through convection is given as under:

\[
Q_c = \text{Heat loss from vertical surface} + \text{Heat loss from top surface} + \text{Heat loss from bottom surface}
\]

Heat also radiates from the vertical cylindrical surface and top and bottom circular areas of the absorber to the surroundings which was computed using equation 3.34.

\[
Q_r = \varepsilon \times \sigma \times A_{abs} \times \left( T_s^4 - T_a^4 \right)
\]  

3.34

Where, \( \varepsilon \) is emissivity, \( \sigma \) is Stefan Boltzmann constant, \( T_s \) is the final surface temperature and \( T_a \) is the ambient temperature. The overall heat loss rate \( Q_1 \) was calculated by considering the heat loss from convection and heat loss from radiation using equation 3.35.

\[
Q_1 = Q_c + Q_r
\]  

3.35

Hence the net heat energy available at the focal area is the sum of heat energy utilized by the absorber and the heat losses taking place from the surface of the absorber. For the total surface area of the absorber, the heat flux is calculated using equation 3.36.

\[
\text{Heat flux} = \frac{\text{Available heat energy}}{\text{Surface Area}}
\]  

3.36

The given concentrator has focal length \( f \), and the aperture diameter \( d \). The rim angle \( \phi \) of the concentrator and the focal spot radius \( r \), were calculated using equations 3.37 and 3.38 (Stine and Harrigan, 1986).

\[
\phi = 2 \tan^{-1} \left( \frac{d}{4f} \right)
\]  

3.37

\[
r = \frac{2f \tan \theta_h}{\cos \phi (1 + \cos \phi)}
\]  

3.38
3.2.4 Design, optimization and fabrication of the condenser

Steam formed in the absorber has to be passed through a condenser where it gets condensed and purified water is collected at the outlet. As operation of the condenser should not depend on electricity, it was conceived to be a water tank with copper tube immersed in it. Maximum heat energy in the month of May was considered for designing the condenser using equation 3.2. This was the highest value of heat energy considered for deciding the size of the water tank. Steam formation starts distinctly around 55° C, hence restricting the temperature rise of water from the room temperature to 25° C. Energy absorbed by the tank was calculated using equation 3.39.

\[ Q = m \times C_p \times dT \]  

Where \( m \) is the mass flow rate, and \( dT \) is the temperature rise in the tank water. Heat transfer across the pipe, \( Q_p \) was calculated by using equation 3.40.

\[ Q_p = U \times A_p \times \Delta T_m \]  

Where, \( U \) is the overall heat transfer coefficient, whose value ranges from 1000 to 5000 W/m²K, \( A_p \) is the area of the pipe and \( \Delta T_m \) is the log mean temperature difference (LMTD) which is given by the equation 3.41.

\[ \Delta T_m = \frac{(T_{hi} - T_{co}) - (T_{ho} - T_{ci})}{\ln((T_{hi} - T_{co}) / (T_{ho} - T_{ci}))} \]  

Where, \( T_{hi} \) is the inlet temperature of steam, \( T_{ho} \) is the outlet temperature of steam, \( T_{ci} \) is the initial temperature of water and \( T_{co} \) is the final temperature of water (Halman, 1990; Rathore, 2008).

Different standard sizes of pipes were considered and CFD analysis was made on these pipes to find out most effective diameter of the pipe. The pipe fitting in a tank with single, double and triple turns was also analyzed in CFD and the optimum shape of the pipe was found out. The manufactured tank was then tested for condensation and it was observed that, the outlet temperature of condensate water increased slowly and finally reached a temperature of 32° C. Tank temperature showed different values at different heights, because the heating is done from the top to bottom. Experimental readings differed from the theoretical analysis as the tank was not insulated and the heat losses were not considered. Effective size of tank was measured and heat losses through convention and radiations were calculated and new optimized tank was then
designed. Multiphase analysis using CFD was carried out on the revised water tank and the analysis showed almost the same condensate temperature as that achieved using theoretical analysis.

To increase the efficiency of the condenser, so that it can condense more quantity of water than designed, the length of copper tube was increased. CFD analysis was performed and it showed that, increase in the length of pipe reduces the temperature of condensate water from 50$^0$C to 40$^0$C. This analysis was validated using theoretical analysis and finally more effective optimized condenser was manufactured.

### 3.2.5 Testing and validation of the system

The designed system was tested at different time periods and it was observed that a maximum of 9.5 liters of water can be evaporated in good sunshine condition whereas 6 liters of water can be evaporated when the solar radiations are low, if tracking is done after 20 minutes and if the system works for 7 hours in a day. The quantity of water evaporated can be increased by increasing the frequency of tracking. In good sunshine conditions and with 15 minutes tracking time, 11 liters of water can be evaporated. Designed water tank was found to be suitable for condensation. The last hour condensate temperature for 11 liters of water was found to be 46$^0$C. No wet steam escaped from the condensate outlet and condensate remained in the liquid form.

Readings were taken by changing quantity of water, size of absorber and by varying the day and month. The developed model is made to predict the amount of water that is to be evaporated and the time for operation which depends on intensity of solar radiations, size and material of the concentrator and the absorber (Dym, 2004). Buckingham’s theorem provides basic principles of mathematical modeling using dimensional analysis by reducing the number of parameters in the equations, making qualitative studies easier (Langhaar, 1951).

The expression for time of evaporation $t$ in hours is obtained as under.

$$
\therefore t = \frac{1236.3 \times \varepsilon A_{\text{abs}}^{0.2025} (Q^{0.395} - 1)}{(\rho)(1^{0.21}d^{0.22})(y\theta)^{0.185}} \text{ hours}
$$
Where, factors which affect the evaporation are, emissivity ($\varepsilon$), reflectivity ($\rho$), specific heat ($C_P$), density ($\rho$), area of the absorber ($A_{abs}$), ambient temperature ($\theta$) and aperture diameter of the concentrator ($d$).

The equations established through mathematical modeling can be validated using all the observed experimental readings. The error in experimentally observed value and calculated value of mass flow rate was 6%. Correlation coefficient was determined for experimental verses calculated values for time of evaporation using the model and was found to be 0.93.

### 3.3 Concluding Remarks

In this chapter the methodology adopted along with the step-by-step approach employed for carrying out the research work has been described. Tools, techniques and software employed for design and optimization of various components of water desalination system have also been briefly described. The next chapter covers the detailed analysis employing the methodology described.