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Design and Analysis of Rooftop Linear Fresnel Reflector Solar Concentrator

K. Gouthamraj, K. Jamuna Rani, G. Satyanarayana
Mechanical Engineering Department,K L University, Guntur 522501

Abstract- ***Many applications of thermal energy, both in industrial and power sectors require medium temperatures ranging from 100°C to 250°C. These applications include industrial process heat, refrigeration, air-conditioning and power generation using organic fluids. The flat plate solar collectors are suitable for low temperature applications maximum up to 80^{o}C and parabolic concentrators are suitable for high temperatures applications above 300^{o}C. In this paper efforts are being made to design a rooftop linear Fresnel reflector solar concentrator (LFRSC) system. This paper discusses the design of a rooftop LFRSC module that is suitable to meet the medium temperature requirements, easy to produce and install like flat plate collector. The design includes thermal analysis. Pro-E modeling tool is used to create 2D and 3D views.***

Keywords: **Solar Thermal Energy, Solar Concentrators, Fresnel Reflector, Secondary reflector, Solar Collector.**

I. INTRODUCTION

In recent years, many concentrating collectors have been proposed. There are basically two types of solar collectors, non concentrating or stationary, and concentrating. A non concentrating collector has the same area for intercepting and for absorbing solar radiation, whereas a sun-tracking concentrating solar collector usually has concave reflecting surfaces to intercept and focus the sun's beam radiation to a smaller receiving area. The high temperature concentrating solar thermal systems, like parabolic trough and linear Fresnel, requires large open area and the system engineering is very complex. These systems are used for power generation using high pressure steam. The temperature is around 400^{o}C.

Flat-plate collectors are the most common solar collector for solar water-heating systems in homes and solar space heating. These collectors heat liquid or air at temperatures less than 80°C. These types of systems are used only for low temperatures applications like domestic hot water and space heating.

Applications like air-conditioning, refrigeration, industrial process heat and distributed power generation through Organic Rankine Cycle require temperatures between 100^{o}C to 250^{o}C. Linear Fresnel reflector and compound parabolic concentrators are suitable devices to produce the medium temperatures. The applications are generally in the range of 10 to 200 kW. In the commercial and industrial units to have an open area to accommodate these units is difficult to locate. However, the rooftop area of the buildings can be used effectively to meet the demand. In this paper the design and thermal analysis of medium temperature concentrator using Fresnel reflectors is discussed.

II. WORKING PRINCIPLE

Linear Fresnel reflectors, as shown Fig 1 use long, thin segments of mirrors to focus sunlight onto a fixed absorber located at a common focal point of the reflectors. A secondary concentrator is used to reflect the rays within the accepting angle. This concentrated energy is transferred through the absorber into thermic fluid. Through a heat exchanger energy is extracted to use to generate power or other commercial applications.

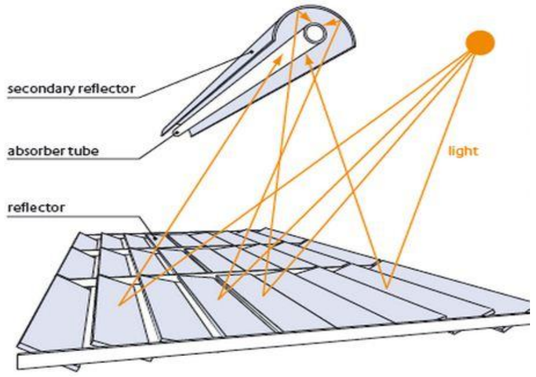

Fig 1. Fresnel Reflector Principle

Rooftop Linear Fresnel Reflector Solar Concentrator (LFRSC) is modular in type and can be interconnected number of units depending on the requirement. They can be connected in parallel–series to achieve given temperature and mass flow rate. The rooftop solar concentrating collector for medium temperature applications using linear Fresnel reflecting mirrors and secondary concentrator with CPC is designed. The collector assembly and components are modeled using PRO-e Package.

The Fresnel reflectors concentrate beam radiation to a stationary receiver. The receiver consists of two stainless steel absorber tubes. Each receiver has a secondary CPC reflector that directs beam radiation on to the absorber tube. The entire optical system is enclosed in a sealed glazed casing.

The module is 2.2 meters long by 1.383 meters wide and 0.351 meters high. Each mirror array with CPC system consists of ten mirrors. Totally the system consists of twenty equal width mirrors that concentrate sunlight on two parallel absorber tubes. The receiver consists of two 25 mm diameter stainless steel absorber tubes. The entire optical system is enclosed in a sealed glass covers as shown in fig 2.

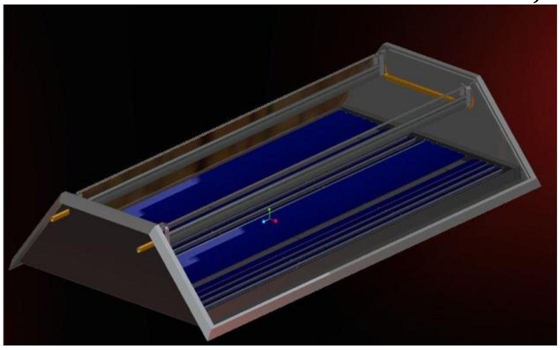

Fig 2. LFRSC

III. DESIGN

In Optical design, tilt of each constituent mirror element is very important. Thin segments of mirrors to focus the sunlight onto a fixed absorber located at common focal point of reflector.

The tilt (θ) of each mirror element is chosen such that a ray normal to the plane of the aperture of the collector and striking the mirror midpoint after reflecting will reach the focal point. The distance between two adjacent mirror elements should be such that they avoid blocking of radiation reflected from any mirror. Tubular absorber of appropriate size placed in the focal plane of LFRSC would intercept all the solar radiations reflected from the constituent mirror elements.The design parameters are:

- Width (W)
- Shift (S)
- Location (Q)
- Aperture plane (XX')
- Tilt with aperture plane (θ)
- Focal distance (F)
- Aperture diameter (D)
- Absorber outer diameter (d_o)
- Absorber inner diameter (d_i)
- Sun subtends angle (ξ_o)

The absorber along with the secondary concentrator will cast a shadow on the aperture plane of the concentrator and hence no mirror element is placed underneath the absorber. The radius of the tubular absorber is taken to be equal to the length of the largest perpendicular dropped from point F on the reflector rays.

While initiating the design of LFRSC with the tubular absorber. The first approximation is taken as the radius of tubular absorber may be taken equal to the half of the width of the mirror element [1] Hence to begin with the first mirror is placed at a distance W/2+f tan (ξ_0) from the centre of aperture plane. The value of the R should be less than the one half of the width of the mirror. Therefore shading will not be occurred. Then the location of the first mirror element Q_1 as

$$Q_1 = R + f \tan\xi_0$$

Here Q_1 is the location of the first mirror

R is the radius of the absorber.

f is the focal distance from the aperture plane.

ξ_0 is the subtends angle of the sun.

Fig 3 gives the geometrical parameters of the mirror elements.

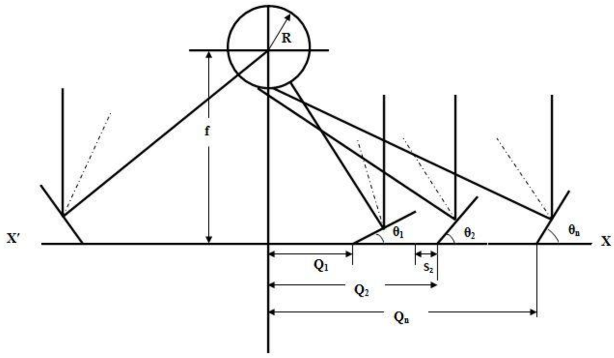

Fig 3: Tubular Absorber

Using fig 3 and geometrical optics relations the tilt of first mirror element is

$$\theta_1 = ½ [(Q_1 + (W/2) \cos\theta_1)/(f-(W/2) \sin\theta_1)]$$

The location of second mirror element and its tilt with aperture plane are chosen such that the ray coming onto it and striking the midpoint of second mirror and after reflection it will reach the focal point. The radiation reflected from the second mirror element is not blocked by the first mirror element. So we have to introduce certain space between the first and second mirror. So this is called Shift associated with the second mirror. The shift of the second mirror is

$$S_2 = W\sin\theta_1 \tan(2\theta_2+\xi_0)$$

The location of the second mirror element and its tilt with aperture plane (XX') is

$$Q_2 = Q_1 + W\cos\theta_1 + S_2$$

On the basis of similar geometrical considerations generalized expressions for the Shift(S) location, and Tilt associated with the n^{th} mirror are

$$Q_n = Q_{n-1} + W \cos\theta_{n-1} + S_n$$

$$S_n = W \sin\theta_{n-1} \tan(2\theta_n + \xi_0)$$

$$\theta_n = ½ [(Q_n + (W/2) \cos\theta_n)/(f-(W/2) \sin\theta_n)]$$

With θ_1=0, S_1=0, and Q_1 = W/2+f tanξ_0 as initial values for iteration and n varies from 1,2,3……….K, K being the total mirror elements placed on either half of the concentrators Considering the values

W = 50mm

d = 25mm

F = 300mm

ξ_o = 16'= 0.00465421 radians

Q_1 = W/2+f tanξ_0

Then Q_1 = 26.39mm

By considering the all above values the values of tilt, shift and location can be calculated by solving the above equations for each mirror element using iteration method. The values are tabulated as follows.

Table1: Reflector Location Parameters

S.no.	Location (mm)	Shift (mm)	Tilt (deg)
1.	26.39	0	4.9

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2.	77.71475	1.50121	9.55
3.	131.4636	4.441751	13.95
4.	188.8325	8.843555	18
5.	251.1333	14.74795	21.7

Based on the above values, the 10 mirrors are placed symmetrically for each absorber. For complete module with 2 absorbers, 20 mirror elements are placed at proper distances. The rooftop LFRSC is modeled using PRO-E package and all dimensions are fixed. The three dimensional view is shown in fig 2.

The secondary concentrator profile is designed using the following relations [8].

$R = 2f_l / (1- \cos(ø))$

$r = R \sin(ø-\theta_{max})-a'$

$z= R \cos(ø-\theta_{max})$

$f_l = a' (1-\cos(90+\theta_{max}))$

$2a' = 2f_l / 1- \cos(90+\theta_{max})$

Here

f_l = focal length of parabolas

θ_{max} = acceptance angle

2a' = radius of the absorber

r and z are spherical co-ordinates

IV. THERMAL ANALYSIS

A steady state thermal analysis of the collector at various solar beam radiation input has been done to estimate the performance of the collector. The usable energy gain of absorber is calculated after deducting the heat losses from absorber. The heat losses are:

1. Radiation Loss from absorber.
2. Convection loss within the space of the collector, i.e. the space between the absorber and reflecting mirror enclosed within the glass covers.

The mode of heat loss is natural convection due to the temperature gradient present in the enclosed space.

3. Convection loss from glass covers to the ambient.

To achieve high efficiency in a concentrating solar collector, there should be minimum thermal losses from the absorber. During operation, the hot absorber tube emits long-wave radiation into the cavity that is absorbed mainly by the bottom wall, which in turn heats up. The heat loss by conduction through enclosed metal box is negligible compared to the radiation and convection losses and hence neglected.

The absorber tube is modeled as an isothermal surface. The absorber tube surface is covered with a selective black chrome sheet. The cover glass at the top and sides of the cavity are modeled as convection boundaries with external heat loss coefficients have calculated. The glasses cover exchanging heat with an ambient temperature of 30^o C. The optical properties and temperatures of components are:

1. Transmissivity of glass cover: 0.95
2. Reflectivity of mirrors : 0.92
3. Absorbivity of absorber tube : 0.92
4. Emissivity of absorber : 0.15
5. Temperature of absorber tube: 200^o C
6. Ambient temperature : 30^o C
7. Cavity Temperature : 80^o C
8. Reflectivity of CPC : 0.90
9. Inlet temperature of thermic fluid: 25°C
10. Glass covers temperature : 50° C
11. Wind velocity : 3 m/s
12. Mass flow rate of thermic fluid: 0.0162kg/sec
13. Specific heat of thermic fluid at mean Operating temperature 1859J/kg K

The heat loss per unit length of absorber due to natural convection and radiation is

$$q_l/L = h_{p-a}\ \pi D_o (T_P - T_c) + \sigma \epsilon\ \pi D_o\ (T_P{}^4 - T_c{}^4)$$

The heat loss from glass cover to ambient is given by due to convection.

$$\backslash q = h_{glass}\ A (T_{glass} - T_a).$$

The useful heat gain is equal to

$$(Q - q_L\text{-}q) = m\ c_p\ (T_{out} - T_{in}).$$

The Q is given by

$$Q = (I_b \times \rho_{reflector} \times \rho_{cpc} \times \tau_{glass})$$

Calculation of Nusselt number is more complex and there are no correlations for calculation of the Nusselt number for the cavity between the absorber and mirrors. From the experimental results [7] and shown fig 4 , a value of

$$N_u = 25$$

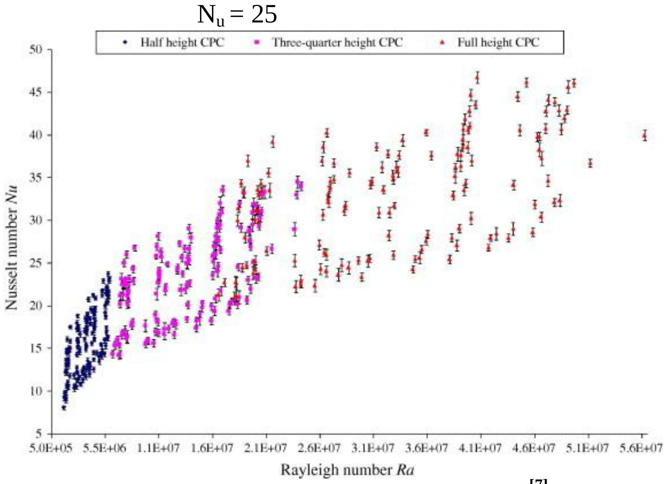

Fig 4: Raleigh number vs. Nusselt number graph [7]

Then the convection heat transfer coefficient is

$$h_{p-c} = 1.445\ w/m^2\ k$$

The convective heat transfer coefficient at the top glass cover is often referred to as the wind heat Transfer coefficient. It has generally been calculated from the following correlation. In this wind speed is taken as 3 m/s.

$$h_{glass} = 8.55 + 2.56\ V\infty \text{ [6]}$$

$$h_{glass} = 16.23$$

Calculating the heat losses based on the above equations the heat losses through convection and radiation within the cavity of the reflector is

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$q_L/L = 38.96$ w/m
$q_L = 155.84$ w/m^2

Heat losses through the convection by the top glass cover is

$q_L = 80$ w/m^2

For various beam radiation data outlet temperatures are calculated. The results are shown in table 2

Table2: Normal Radiation Vs Efficiency

I_b (w/m^2)	T_{out} (°C)	Overall Efficiency
300 w/m^2	31.27°c	34.98%
400 w/m^2	35.97°c	45.98%
500 w/m^2	40.67°c	52.45%
600 w/m^2	45.37°c	56.82%
700 w/m^2	50°c	59.94%
800 w/m^2	54.78°c	62.28%

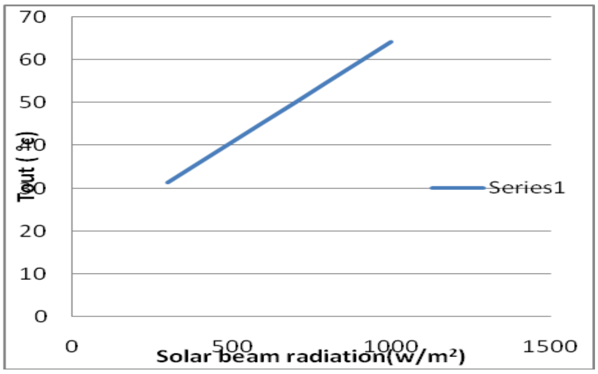

Fig 5: the solar radiation falling on the collector VS the outlet temperature

The efficiency of the solar system is dependent on the amount of solar radiation falling on the collector area. The heat losses depend upon the temperature of the absorber. To reduce the heat losses we have used the selective coating to the absorber tube. The emissivity of this selective coating is very low. The efficiency increase by this selective coating. The efficiency of the collector is defined as the ratio of the useful heat gain to the solar radiation incident on the collector.

V. CONCLUSION

The roof top concentrating collectors will accelerate the development and commercialization of other solar technologies like solar based cooling system. It is a product that can be produced in manufacturing line. Due to the modular nature of the product the system capacity can be increased easily by adding more number of panels. The cost of production is also less compared to site fabrication. The time of installation of the system will be greatly reduced. Maintenance of the system is also use due to the modular in nature. The steady state thermal analysis is done for collector and results are encouraging. In the next phase the concentrator will be tested and compared with simulated results.

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