HEAT RECOVERY IN COMPACT SOLAR THERMAL SYSTEM

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ABSTRACT

Thermally conductive thin metallic reflector sheet if used as the concentrating surface in a solar collector, the fraction of net solar radiative heat absorbed by the collector can be recovered as additional heat which is normally dissipated as waste heat through the exposed surfaces of the collector & the supporting frame. Thus increased performance factor will be attributed to the compactness of the system. In this paper an analytical study was conducted for approximation of the thermal performance compared with the bench model experimental results.

Keywords:

Polygeneration, spiral, convection, augmentation, insulation
Introduction:

A compact solar thermal collector comprising of a dish concentrator with its reflector made up of lunes of thin metallic sheet integrated together is proposed in this discussion. The extra weight of a supporting structure for the concentrator is replaced by a spiral housing of heat exchanger pipe with its outlet connected to the inlet of the receiver at the focus in the aperture plane covered with a thin glass. To minimize the convective heat loss, concentrator rear surface along with the heat exchanger pipe assembly is perfectly insulated. Although the rim angle more than 80° develops distorted focal image at the aperture plane as depicted by Evan et al.[1] but the maximum recovery of heat is the prime objective of our study. Here the focal length is kept half of the radius of curvature of the concentrator. In this regard, effectiveness of short focal length image formation was analysed earlier by Kenef S, at el [2]. Use of a thin aperture glass for the sustainable reflection & enhanced heating effect as in case of a flat plate solar heater along with specific weight of the reflecting sheet are considered in this proposed system. Such discussion was earlier made by Maria Brogren at el.[3] on the use of Aluminium-polymer laminated steel reflector & protective polyethylene tetrathalate layer for concentrator surface. Besides weathering effect on specular reflectance the disadvantages of using glass mirrors due to the delamination crack, weight & rigidity to deform were also studied by Leonard Da Jaffe et el.[4] in early 1987 on the test results of various concentrator at the Jet Propulsion Laboratory, California, USA. The proposed system takes into consideration the facts above. In order to reduce the heat loss in solar collector, convection suppression using honeycomb or suppression of cellular convection by lateral walls were suggested by Arnold.J,N et al[5] & Edward.D.K et al[6]. Increased collector performance was depicted by Robert P. Friedman et el.[7] using two stage optical design, Yong Shuai at el.[8] using an upside-down pear cavity receiver with directional attributes of focal flux. It may be observed that without fluid motion heat transfer is dominated by conduction & radiation which is another motive to entrap the volume of air below the aperture glass for maximize the conductive heat recovery.

Both the thermal conductivity & the optical property of the thin metallic reflector sheet is used here for solar heat recovery in a single system.

A comparison study has been made in between the proposed system performance and the experimental results of the bench model so that a thermal system can be developed for boosting up the green energy revolution in community relevance like cooking, crop drying, dying, food preservation, pasteurization, laundries, bio-mass pyrolysis, & several other applications.

2.1 Compact Solar Thermal System

The concentrator focal length = 0.3 m and the aperture radius = 0.53 m. The bench model as in Fig.1(a),(b) was tested on sunny days at Guwahati (North East India) at Φ(Latitude) = 26.1838° N (Altitude = 91.7633 E ) with variable ambient conditions and the average results enlisted at the Table 1, were analyzed accordingly.
Fig. 1(a). Geometrical design of the system

Fig. 1(b.) Experimental bench model

Fig. 2

Table 1:

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Aperture area of concentrator, $A_c$</td>
<td>$0.882 \text{ (m}^2\text{)}$</td>
</tr>
<tr>
<td>Specific heat of water, $C_{pw}$</td>
<td>$4200 \text{ (J/kg K)}$</td>
</tr>
<tr>
<td>Tube inner diameter, $d_i$</td>
<td>$0.005 \text{ (m)}$</td>
</tr>
<tr>
<td>Tube outer diameter, $d_o$</td>
<td>$0.006 \text{ (m)}$</td>
</tr>
<tr>
<td>Solar Irradiance intensity, $I_b$</td>
<td>$600 \text{ (W/m}^2\text{)}$</td>
</tr>
</tbody>
</table>
Tube surface to fluid conductance coefficient, \( K = 237 \, \text{W/mK} \)
Total length of the tube, \( L = 12 \, \text{m} \)
Fluid mass flow rate per unit collector area, \( m_f = 2.5 \, \text{Kg/m}^2 \text{ h} \)
Temperature at the focal image, \( T_f \) (k)
Average Inlet Temperature of water, \( T_{wi} = 361.1 \, \text{(k)} \)
Average Outlet Temperature of water, \( T_{wo} \) (k)
Heat loss coefficient from the tube surface to ambient, \( u = 0.0313 \, \text{(W/h m}^2 \text{k)} \)
Fraction of solar insolation lost at the glass, \( (\alpha \tau)_g = 0.55 \, \text{[Klein's curve]} \)
Radius of the aperture, \( R_a = 0.53 \, \text{(m)} \)

<table>
<thead>
<tr>
<th>Date</th>
<th>( T_{wi} )</th>
<th>( T_{wo} )</th>
<th>( T_{wo}/T_{wi} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>25-04-13</td>
<td>31.8</td>
<td>84.2</td>
<td>1.68</td>
</tr>
<tr>
<td>26-04-13</td>
<td>32.0</td>
<td>89.6</td>
<td>1.69</td>
</tr>
<tr>
<td>21-05-13</td>
<td>32.5</td>
<td>90.7</td>
<td>1.72</td>
</tr>
<tr>
<td>21-05-13</td>
<td>32.9</td>
<td>86.1</td>
<td>1.60</td>
</tr>
</tbody>
</table>

**Table 2:**

2.2 Thermal Analysis

Here the improvised idea of harnessing the absorbed heat by the fluid through tubes in the heat exchanger matrix as collector as referred in Fig. 2 is preferred over the heat generated at the receiver. Heat generated at the receiver is more or less a conventional procedure. From the heat interaction between the concentrator surface & the fluid in the tubes of the heat exchanger for heat recovery considering unit mass of fluid,

\[
c_{pw}(T_{wo} - T_{wi}) = [A_g(\alpha \tau)_g] l_b - u(T_{wo} - T_{wi})[1 - \exp\left(-\frac{u k}{m_f c_p(u + k)}\right)] \frac{1}{u} (i)
\]

The tube surface to fluid inside conductance coefficient is given by Whillier et al[14] as,

\[
k = \frac{2.484}{u} \left[ \frac{0.0854}{L/2} \right] \left[ \frac{1 + 0.0684}{L/2} \right]^{2/3}
\]

From (i) & (ii) we can write

\[
c_{pw}T_{wi}(X - 1) = [A_g(\alpha \tau)_g] l_b - uT_{wi}(X - 1)[1 - \exp\left(-\frac{u k}{m_f c_p(u + k)}\right)] \frac{1}{u} (ii)
\]

where \( X = T_{wo}/T_{wi} \)

Putting the values from the Table

We get \( T_{wi}(X - 1) = 929.775 \)

Finally the analytical value of the ratio \( T_{wo}/T_{wi} \) is found to be

\( X = 3.56 \)

Although the proposed system accumulates the primary heat energy at the focal image on the receiver, yet effort is made to recover the waste heat by the water in the heat exchanger to act as the auxiliary heat to augment the net system output. The efficiency of the system may be expressed as

\[
\eta = \left[ \sum mc \right] / \text{(Ib. A\v)} \left[ \frac{dT}{dt} \right]
\]

where
\( \Sigma mc \) is the net heat capacity, \( A_e \) is the effective heat transfer area. So \( \left[ \Sigma mc / (I_b \cdot A_c) \right] \) is constant for a particular thermal system although a maximum average of solar intensity \( I_b \) is considered for the day.

### 3 Bench model experiment

The bench model performance was tested, taking temperature readings for 3 days as transitory, instant parameters for 10 minutes for stagnation in every 30 minutes interval. Few instantaneous fractional deviations due to manual error were interpolated few times to get the acceptable readings for consecutive two axis rotation of the dish for tracking. The average temperatures with & without insulation to the rear surface of the collector have been enlisted in the Table 2. The ratio of the outlet temperature to the inlet temperature water is recorded. Further the effect of rear collector insulation & the outlet temperature of air in the space bounded by the aperture glass & the concentrator is also recorded. The hot air may be considered as an additional product of the single system. The readings are plotted to get the following Temperature-Time curves.

### 4 Observations:

(a) The water tubes collect absorbed heat from the reflector sheet directly.

(b) The considerable amount of heat to rise the water temperature by almost twice the inlet temperature envisage the possibility of preheating of feed water to the receiver above saturation level at constant pressure.

(c) Improvisation of such a design will act as a polygeneration thermal device to give out hot water, hot air & steam of different quality.

(d) The temperature ratio of water in experimental works deviates almost twice the analytical result which is due to the design defect & handling.

From the temperature curves of the bench model experiment it is found that there is always an increase in temperature for outlet air & water, sharing the absorbed heat by the reflector sheet. The 7% rise in average temperature in water in heat exchanger using rear collector surface insulation predicts the opportunity of the reparation of convective heat loss.

### 5 Discussion

There are better scopes for design improvisation for both the collector beneath the concentrator surface as well as the receiver at the focal plane so that the outlet temperature of water leaving the heat exchanger can be raised to the saturation temperature of water at the corresponding pressure. The feed water to the receiver at this temperature can be well utilised in formation of steam of utility brand at regulated temperature & pressure. The use of preheated water will make it possible to increase the rate of steam production in comparison to the stand alone receiver. Better result can be expected on perfect insulation & on reduction of overall heat losses. The design itself suppresses convective heat loss at it’s aperture glass & conductive loss for rear collector insulation which can be enhanced further.
6 Conclusion:

A selective reflecting material with high quality optical & thermal performance is suggestive for a Polygenerative compact solar thermal system so that a high range of irradiative solar heat accumulation is possible from low to medium temperature water during diffusive radiation up to dry or supersaturated steam during intensive insolation.

This can be extended to a parabolic concentrator also. It will definitely increase the ratio of generating capacity to the land surface area. The increased rate of heat generation at the receiver of the proposed model may also be used for intermittent solar radiation in some area experiencing frequent cloud. On eliminating errors in handling and fabrication the system thermal efficiency will be improved.

7 Future scope:

This study has sufficient scope to develop a Polygenerative solar thermal process heat generating system with better sustainability. Reduced volume & weight with precision controlled fully automatic sun tracking system will definitely improve the performance.

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References:


[3] Optical properties, durability, and system aspects of a new aluminium-polymer-laminated steel reflector for solar concentrators Maria Brogren,∗, Anna Helgessonb, Bjørn Karlssonb,c, Arne Roosa, Uppsala University, Sweden


[8] Radiation performance of dish solar concentrator /cavity receiver systems by Yong Shuai, Xin-Lin Xia, He-Ping Tan School of Energy Science and Engineering, Harbin Institute of Technology, Harbin 150001, China

