A05 家庭用トラフ型太陽集熱器の設計とその特性
Design and Performance of Trough Type Solar Collector for Domestic use

Nour Chaabane*,1, Yuki Shiono*1, Nobuya Shimizu*1, Hiroshi Sekiya*2
and Masafumi Katsuta*1

*1 Graduate School of Creative Science and Engineering, Waseda University
Okubo 3-4-1, Shinjuku-ku, Tokyo 169-8555, Japan
*2 Graduate School of Environment and Energy, Waseda University
Nishitomita 1011, Honjo-shi, Saitama 367-0035, Japan

Abstract
The aim of this research is to create a Zero Energy House system using insolation. One method to run the system is by using the parabolic trough solar collector. This paper shows the specification and design of the equipment. This technology is made of a parabolic mirror for the collector. A copper pipe installed in the focus collects solar heat and water is used as heat transfer fluid. The trough type collector is available for Rankine cycle power generation systems, and also for a Stirling cycle engine operated by a low-grade heat source.

Key Words: Solar Energy, Parabolic Trough Collector, Power Generation, Low-Grade Heat, Stirling Engine

1. Introduction
In these days when measures for the realization of sustainable low carbon society are considered to be urgent, research and development on system machinery which aimed at clean energy and saving energy becomes indispensable. In addition, the authors hope that renewable energy, such as solar heat, wood biomass, and wastes, is used as a heat source for the power generation system, and our attention to solar heat. Now many solar power generation systems, which are based on Rankine cycle engines or Stirling cycle engines, are operated by the collectors of a tower, trough and dish type.

The authors are developing a solar Stirling engine power generation system, which uses a solar tracking type collector made by a Fresnel lens. Solar energy of 500～550W in our design is concentrated on a heater, so that the Stirling engine generator for performing characteristic examination as a basic system has a power generation output of 200W. A trough collector described in this paper is available to operate the Stirling engine by a low-grade heat source.

Parabolic trough collectors (PTC) are systems in the shape of "U". Their primary purpose is to concentrate the sunlight onto a receiver tube located along the focal line of the reflector. There is a glass tube that surrounds the receiver which acts as an insulator and thus reducing heat loss. These systems often use a single axis or dual axis solar tracking system as shown in Fig.1. Moreover, the Receiver’s temperature can reach in excess of 400 °C.

Fig.1 Parabolic trough collector (PTC)

The PTC application can be divided into 2 main groups. The first group is for the power generation, which is composed of a series of PTC units which are 6 m in width, between 100 and 150 m in length, and have concentration ratios around 60. This system can reach between 300 and 400 °C. The second group is for low temperature heat production between 60 and 250 °C, which is used for industrial heat process, domestic hot water, space heating, air conditioning, and refrigeration.

The aim of this research is to design a parabolic trough collector, test it, and determine its thermal efficiency. Then the authors estimate its potential in the south of Tunisia, where the insolation ratio is high, are intend to compare the experimental results with the theoretical results and other efficiency curves of the same types of solar collectors.

2. Design and construction of PTC
2.1 Design of the parabola
The 3 most important parameters to design the parabolic trough collector as shown in Fig. 2 are:
- Rim Angle: φr [deg]
- Collector aperture: Wa [m]
- Receiver diameter: Dr [m]

Rim angle
The rim angle is the angle from the line normal to the collector passing through the focus to the rim of the collector as
Collector aperture

In order to compare with other collectors with a 1 m² aperture area, the collector aperture should be adapted with the length of the frame L which is 1.042 m. Therefore \( W_a = \frac{1}{1.042} = 0.97 \text{ m} \).

Receiver diameter

This parameter determines the intercept factor which is important for optical efficiency. The intercept factor is the ratio of total energy intercepted by the receiver to the energy reflected by the reflector. Its value depends on receiver diameter, surface angle error of the mirror and solar beam spread. These errors are named random and non-random.

The random errors \( \sigma_{\text{random}} \) are truly random in nature and defined by Rabl [3] as

\[
\sigma_{\text{random}} = \sqrt{\sigma_{\text{sun}}^2 + 4\sigma_{\text{stop}}^2 + \sigma_{\text{mirror}}^2}.
\]

Where \( \sigma_{\text{sun}} \) is apparent changes of the Sun's width, \( \sigma_{\text{stop}} \) is scattering effects caused by random slope-errors, and \( \sigma_{\text{mirror}} \) is scattering effects associated with the reflective surface.

The non-random errors \( \beta \) and \( dr \) are

- The angle \( \beta \): the misalignment between the normal of the collector aperture and the reflected ray from the sun due to the tracking system.
- The distance \( dr \): the receiver misallocation due to the manufacturing.

The expression of the intercept factor \( \gamma \) given by Güven [4]:

\[
\gamma = \frac{1 + \cos \phi}{2 \sin \phi} \left[ \frac{\text{Erf}(1 + \cos \phi)(1 - 2d^* \sin \phi) - (\beta^*(1 + \cos \phi))}{\sqrt{2\pi \sigma_{\text{random}}^2(1 + \cos \phi)}} \right] - \frac{\text{Erf}(-\sin \phi(1 + \cos \phi))(1 + 2d^* \sin \phi) + (\beta^*(1 + \cos \phi))}{\sqrt{2\pi \sigma_{\text{random}}^2(1 + \cos \phi)}} \right] \\
\frac{d\phi}{1 + \cos \phi}
\]

(2)

The authors used MATLAB to numerically evaluate this expression, and then the intercept factor was used to evaluate the thermal and optical efficiency. As input parameters, vectors of different diameters, random errors \( \sigma_{\text{random}} \), non-random errors \( \beta \) and \( dr \) were used by

\[
\sigma_{\text{random}} = \sigma_{\text{random}} \times CR
\]

\[
\beta^* = \beta \times CR
\]

\[
d^* = dr / Dr.
\]

Because this collector is made by hand, therefore developing countries parameters for the errors were used [4]

\[
\sigma_{\text{random}} = 0.0113 \text{ rad}
\beta = 0.0176 \text{ rad}
\]

\[
dr = 6.2 \text{ mm}
\]

For this research case, because everything is used for the first time, the authors do not have the precise estimation of the errors related to the manufacturing, assembling and solar tracking system of the equipment. An objective of this research is to estimate the experimental intercept factor \( \gamma \) and compare it with the theoretical.

2.2 Design of the collector

To build a collector characterized on Table 1 with low cost, strong structure and high accuracy, selection of material is a very important parameter to consider.

<table>
<thead>
<tr>
<th>Table 1 General characterization of the collector</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rim Angle ( \varphi ) [deg]</td>
</tr>
<tr>
<td>Collector aperture ( W_a ) [m]</td>
</tr>
<tr>
<td>Receiver diameter ( d_r ) [mm]</td>
</tr>
<tr>
<td>Height of the parabola ( h_p ) [m]</td>
</tr>
<tr>
<td>Focal distance ( f ) [m]</td>
</tr>
<tr>
<td>Collector concentration ratio ( CR )</td>
</tr>
<tr>
<td>Collector dimension [m]</td>
</tr>
<tr>
<td>Aperture area of PTC ( A_v ) [m²]</td>
</tr>
</tbody>
</table>

The built PTC is shown in Fig.4. 5 parabolic ribs were used to give the shape of the parabola. These ribs are made of plywood and cut by a laser with high accuracy. The dimension of this parabola is 1000 mm in width and 230 mm in height with 4 chamfers of \( 5 \text{ mm} * 45° \) on the corners. A high reflective aluminum sheet (reflection of the collector \( \rho = 0.95 \)) was chosen, light and easy to bend by hand to set firmly on the ribs in order to reduce the degree of technical skills required. This aluminum sheet mirror finish is MIRO 2 made by Alnand which is 0.5 mm in thickness and 1135 mm *1040 mm in dimension. The receiver is a copper tube coated by heat resistant and high absorbance flat black paint. This component has a high heat conductivity and absorption. Its absorbance of the receiver \( \alpha = 0.94 \). 2 brackets were used to hold the receiver, made of plywood of 18 mm thickness (same plywood plate used for the ribs). For the assembling, these components were attached by strong structure and light channels, which were used to attach the ribs together and to fix the reflector on the ribs. 4 channels "L" shaped Aluminum and 1 steel tie bar were used for assembling.
of the whole unit to the ground (movement of the direction of east and west).

For the circulating pump, the heat transfer fluid is water. The choice is according to the mass flow rate needed in order to measure the thermal efficiency (2, 3, 4, 5, 6 L/min) and compare with other PTC units.

3. Theoretical thermal efficiency

The thermal efficiency $\eta$ is estimated by:

Useful energy delivered / Total energy on the collector.

To estimate it, different mathematical models will be used to calculate the optical efficiency $\eta_{op}$ which is characterized by the properties of different materials, overall heat losses coefficient of the receiver $UL$, the heat removal factor $FR$, direct solar radiation $I$, inlet temperature of the fluid $T_i$ and ambient temperature $Te$.

The theoretical thermal efficiency $\eta$ can be calculated by the following equation given by Kalogirou [5].

$$\eta = FR \times \left( \eta_{op} - UL \times \left( \frac{Te - T_i}{T_i + CE} \right) \right)$$

(6)

Which $\eta_{op}$ is the optical efficiency [6]:

$$\eta_{op} = \rho * \alpha * \gamma * \left( \left( 1 - A_r \tan \theta \right) * \cos \theta \right)$$

(7)

$\rho$ is the mirror reflection, $\alpha$ is the absorbance of the receiver, $\gamma$ is the intercept factor, and $A_r$ is the geometric factor, which is the reduction of the effective area of aperture area due to abnormal incidence angle $\theta$. A solar tracking system is connected to this collector, so the incidence angle $\theta$ is 0, therefore

$$\eta_{op} = \rho * \alpha * \gamma$$

(8)

Heat removal factor $FR$ [7]:

$$FR = \frac{M \times C_p}{A_{r,\text{int}} \times UL \times \left( 1 - \exp \left( -A_{r,\text{int}} \frac{UL * F}{M \times C_p} \right) \right)}$$

(9)

Where $A_r$ is the receiver area, $C_p$ is specific heat of the heat transfer fluid and $M$ is Mass flow rate. A subscript int indicates the inside of the receiver. $F^-$ is the collector's efficiency factor given as [7]:

$$F^- = \frac{1}{UL \times \frac{1}{\left( \frac{DR_{\text{ext}}}{DR_{\text{int}}} \right)^{\frac{1}{3}}} - \frac{DR_{\text{ext}}}{DR_{\text{int}}}}$$

(10)

The theoretical results and other efficiency curves of the same types of solar collectors are indicated in Table 2, where $C_p$ is the direct solar irradiance and $\Delta T = T_i - T_e$ is the difference between the inlet fluid temperature and ambient temperature.

The convective heat transfer $h_{conv}$ is calculated depending on the flow inside of the receiver. $K_r$ is conductivity of the copper tube, and a subscript ext is the outside of the receiver. It is the most complex parameter to estimate, because it depends on the flow nature, Reynolds number $Re$, Prandtl number $Pr$, entrance length and so on.

Overall heat loss coefficient $UL$, based on the receiver area $Ar$ as shown on Fig.6 is given by:

$$q_{2,\text{cond}} = \frac{q_{2,\text{conv}} + q_{2,\text{conv,air}} + q_{2,\text{cond,bracket}}}{Ar}$$

(11)

$$q_{2,\text{conv}} = q_{2,\text{conv,air}} + q_{2,\text{rad}} + q_{2,\text{cond,bracket}}$$

(12)

$$q_{\text{heat loss}} = q_{2,\text{conv,air}} + q_{2,\text{rad}} + q_{2,\text{cond,bracket}}$$

(13)

$q_{2,\text{conv,air}}$ and $q_{2,\text{rad}}$ are small compared with convective and radiation losses.

$$UL = q_{2,\text{conv,air}} + q_{2,\text{rad}} + h_w + h_{r-r-a}$$

(14)

The convective heat transfer coefficient $h_w$ between receiver and ambient air is due to the wind [15].

$$h_w = \frac{Nu_a \times K_a}{Dr_{ext}}$$

(15)

$$Nu_a = 0.4 + 0.54 \times Re_a^{0.52}$$

(16)

$$Nu_a = 0.3 \times Re_a^{0.6}$$

(17)

$K_a$ is Conductivity of air and the subscript $a$ is the value of air.

The radiation heat transfer coefficient $h_{r-r-a}$ between absorber tube and the ambient air [14]:

$$h_{r-r-a} = \frac{\epsilon * \sigma \times (T_r + T_a) \times (T_r^2 - T_a^2)}{M \times C_p}$$

(18)

Where $\epsilon$ is emissivity of the receiver, $T_r$ is receiver temperature and $\sigma$ is Stefan–Boltzmann constant ($= 5.6697 \times 10^{-8}$) [ W m$^{-2}$K$^{-4}$].

A program was prepared to estimate these coefficients by including REFPROP library to determine the physical characteristics of water and air. The data shown on Table 3 are the input parameters used on the program to estimate the theoretical thermal efficiency.

Table 2: Thermal efficiency of different type of PTC

<table>
<thead>
<tr>
<th>Efficiency equations</th>
<th>Reference</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\eta = 0.66 - 0.233 \times (\Delta T/G_p)$</td>
<td>[8]</td>
</tr>
<tr>
<td>$\eta = 0.65 - 0.382 \times (\Delta T/G_p)$</td>
<td>[9]</td>
</tr>
<tr>
<td>$\eta = 0.642 - 0.441 \times (\Delta T/G_p)$</td>
<td>[10]</td>
</tr>
<tr>
<td>$\eta = 0.638 - 0.387(\Delta T/G_p)$</td>
<td>[11]</td>
</tr>
<tr>
<td>$\eta = 0.69 - 0.390 \times (\Delta T/G_p)$</td>
<td>[12]</td>
</tr>
<tr>
<td>$\eta = 0.6128 - 2.302(\Delta T/G_p)$</td>
<td>[13]</td>
</tr>
<tr>
<td>$\eta = 0.552 - 2.010(\Delta T/G_p)$</td>
<td>[14]</td>
</tr>
</tbody>
</table>

4. Evaluation of thermal performance

The thermal performance of the collector will be evaluated...
according to ASHRAE standard 93-1986 [16] in order to compare with other collectors. This standard provides 3 methods to determine the thermal performance; instantaneous thermal efficiency, time constant and incidence angle modifier.

4.1 Instantaneous thermal efficiency
According to ASHRAE standard 93-1986 [16], the instantaneous thermal efficiency $\eta_t$ is calculated for a period of 5 minutes or the time constant test, whatever is larger. It is evaluated by considering

$$\eta_t = \frac{MCP(T_0-T_l)}{A_0G_b}.$$  \hspace{1cm} (19)

Where $T_0$ and $T_l$ respectively are the outlet and inlet temperatures on the receiver tube, and the direct solar irradiance $G_b$ on the same time. On the other hand according to Kalogirou the efficiency $\eta_t$ equation for a collector through the first law of thermodynamic [12] is

$$\eta_t = FR \left( \frac{\eta_{op} - \frac{UL}{CR} \left( \frac{\Delta T}{G_b} \right)}{\Delta T} \right).$$  \hspace{1cm} (20)

This equation has the form of $y = ax + b$ which can help to compare the theoretical and experimental results, using the following equations.

$$a = FR \times \eta_{op}$$  \hspace{1cm} (21)

$$b = \frac{FR \times UL}{CR}$$  \hspace{1cm} (22)

$$x = \frac{\Delta T}{G_b}$$  \hspace{1cm} (23)

4.2 Time constant
According to ASHRAE standard, time constant is the time required for a fluid leaving the collector to reach 63.2% of its steady state [16] for the heating process. As initial conditions, before starting the test, the receiver should be defocused and the inlet temperature of the fluid equal to the ambient temperature. Once we focus the receiver, we will serve to this equation

$$\frac{T_{in}-T_l}{T_{in}-T_{eq}} = e^{-\frac{t}{\tau}}$$  \hspace{1cm} (24)

to calculate the time which $T_l$ is the inlet temperature, $T_{eq}$ is the collector-outlet initial temperature and $T_{in}$ is the collector-outlet water temperature after $t$ [sec]. For the cooling process, time constant is the time required for a fluid leaving the collector from quasi-steady state to reach 36.8% of the collector-inlet initial temperature [16]. This test measures the time response of the solar collector [16].

4.3 Incidence angle modifier
The test as mentioned in Ref. [16] determines the drop of the optical efficiency due to the change of the incidence angle $K_g$. It is calculated by using the following equation.

$$K_g = \frac{n(t \geq T_a)}{FR[\eta_{op}]}.$$  \hspace{1cm} (25)

The subscript $a$ in Eq. 25 shows the normal incidence.

5. Summary
As mentioned above, the authors conducted the design of the parabolic trough collector which was used to operate the power generation system by solar heat. Our plan is to put this system using the Rankine or Stirling cycle engine in practical use in Tunisia.

The built PTC has strong and light structure and is low cost. And also its efficiency is higher than another system with the PTC, because of the solar tracking mechanism. The authors are going to examine performance of the PTC according to ASHRAE standard 93-1986, and the experimental results are to be compared with the theoretical result and the efficiency curves of the same types of solar collectors.

References