Experimental and Theoretical Validation of Numerical code to Analyze the Performance of Thermosiphon Flat Plate Solar Water Heater

Arunachala U. C., Siddhartha Bhatt M., and Sreepathi L. K.

Abstract— Thermosiphon Flat plate Solar water heater still remains as one of the most interesting technologies for exploitation of solar energy. Its remarkable efficiency, combined with simplicity of construction, autonomous operation, absence of moving parts and hence the minimization of necessary maintenance, make it an interesting alternative to forced circulation systems. In view of the analysis of Thermosiphon system under different working condition as well as study related to deterioration at different degree of scaling, a numerical code has been developed. To validate the code, specification and working conditions of two systems quoted in the research articles are incorporated and the result shows good agreement with both experimental and theoretical energy efficiency parameters as revealed in two cases.

Keywords— FLATSCALE, H-W-B equation, mass flow rate, Thermosiphon Flat plate Solar water heater

I. INTRODUCTION

The process of transition from fossil sources to purely renewable energy began on an organized and planned scale in the mid 20th century. The quantification of percentage of renewable energy to be used (e.g., 5 %, 10 % etc. of the total energy) has begun in the 1980’s. Presently these targets in different countries are to replace 20 % to almost 100 % fossil energy through mainly solar and wind sources.

Now a days both solar electric and thermal systems are popular. In thermal route particularly low and medium thermal systems have wide publicity due to simple design and low operating and maintenance cost. These are meant for water heating, steam generation, refrigeration etc. Solar thermal heating provides an elegant direct route of energy conversion without going through the route of coal thermal plant - electrical energy - transmission - distribution - resistance heating.

Hot water is an essential requirement in industries as well as in the domestic sector. Use of sun’s energy to heat water is not a new idea. More than one hundred years ago, black painted water tanks were used as simple Solar Water heaters (SWH) in a number of countries. Today there are more than 30 million m² of SWH have been installed around the globe. A 100 litres capacity SWH can replace an electric geyser for residential use and may save approximately 1500 units of electricity annually. The use of 1000 SWHs of 100 litres capacity can contribute to a peak load saving of approximately 1 MW [1].

II. THERMOSIPHON FLAT PLATE SOLAR WATER HEATER (TFS)

A Flat Plate Collector (FPC) is a heat exchanger that transforms incident solar radiation into heat by accumulating it within the enclosure through greenhouse effect. The total system with FPC, highly insulated water storage tank and well insulated pipes connecting the two is called as TFS. TFS still remain as one of the most interesting technologies for exploitation of solar energy. Its remarkable efficiency combined with simplicity of construction, autonomous operation, absence of moving parts and hence the minimization of necessary maintenance, make it an interesting alternative to forced circulation systems.

It is generally observed that in tropical areas, the performance of FPCs deteriorate within five to twelve years of their installation due to factors related to manufacturing, operating conditions, and lack of maintenance etc. Out of various factors viz. degradation of gasket and absorber coating, collector fogging, oxide coating on inner glass surface etc., problem due to scaling is significant as it is based on the quality of water used. Whereas the remaining factors are system dependent and due to the stringent compliances of BIS (Bureau of Indian Standard) imposed by MNRE (Ministry for New and Renewable Energy), the quality of TFS is improved in India.

Hence to study the performance deterioration of FPC with different degree of scaling, numerical code called FLATSCALE has been developed (algorithm in MS Excel) which can be used in case of clean as well as scaled FPC. In this paper, results obtained through FLATSCALE by including specification and working condition of TFS (without scaling) as cited in the research articles are compared with the outcome of articles.

III. DEVELOPMENT OF NUMERICAL CODE

The various parameters to be considered in the performance analysis are discussed here.
A. Performance Parameters of FPC
The instantaneous efficiency (\(\eta_i\)) of FPC based on energy accumulation is given by

\[ \eta_i = \frac{q_u}{I_A c} \]  

(1)

Here, ‘\(q_u\)’ represents the useful heat gain (W) from the system with absorber area ‘\(A_p\)’ (m\(^2\)) and solar radiation level(W/m\(^2\)). ‘\(I\)’ The above equation can be modified in terms of other collector parameters. viz., collector heat removal factor and collector efficiency factor.

Collector efficiency factor (‘\(F\)’) is given by [2], [3]

\[ F = \frac{1}{WU_1\left[\frac{1}{U_1|W-d_o|\phi+ d_o} + \frac{1}{\pi d_s h} + \frac{2\pi k_r}{\ln(d_o/d_1)} + \frac{2\pi k_s}{\ln(d_1/d_2)}\right]} \]  

(2)

Where, ‘\(W\)’ is the riser pitch(m), ‘\(d_o, d_i\)’ are the inner, outer and scaled flow diameter of the riser (m), ‘\(h\)’ water side convective coefficient (W/m\(^2\)K). ‘\(k_r\)’ and ‘\(k_s\)’ are the thermal conductivities of riser and scale material (W/mK).

Referring (2), the last term in the denominator accounts for the thermal resistance of the scale.

Collector heat removal factor (‘\(F_r\)’) is

\[ F_r = \frac{M_C p}{U_1 A_p} \left[1 - \exp\left(-\frac{F'U_l A_p}{M_C p}\right)\right] \]  

(3)

Where, ‘\(M\)’ is the mass flow rate(kg/s) and ‘\(U_1\)’ is the overall loss coefficient (W/m\(^2\)K) which is defined as the sum of loss coefficients for sides ‘\(U_1\)’, bottom surface ‘\(U_b\)’, and the top glass cover ‘\(U_t\)’ [4]. Hence

\[ U_1 = U_t + U_b + U_s \]  

(4)

B. H-W-B Equation
Hottel-Whillier-Bliss equation is widely used in performance analysis of FPC. An energy balance on the absorber plate yields the following equation for a steady state.

\[ Q_u = A_p S - U_1 A_p \left(T_p - T_a\right) \]  

(5)

Equation (1) is rewritten by including (5) as

\[ \eta_i = \frac{F R A_p [S - U_1 (T_1 - T_a)]}{I A_c} \]  

(6)

\[ \eta_i = \frac{F R A_p S - U_1 (T_1 - T_a)}{I} \]  

(7)

Here, ‘\(S\)’ is the solar irradiation absorbed by absorber plate (W/m\(^2\)) and ‘\(A_p\)’ is the collector area (m\(^2\)). ‘\(T_p\), ‘\(T_1\)’ and ‘\(T_a\)’ are the absorber plate mean, water inlet and surrounding temperature (K) respectively.

Equation (7) is written based on the absorber plate area as

\[ \eta_i = a_0 - a_1 \frac{(T_1 - T_a)}{I} \]  

(8)

Where ‘\(a0\)’ is Intercept (-) and ‘\(a1\)’ is slope of H-W-B equation (W/m\(^2\)K).

Equation (8) is widely known as H-W-B equation [2]. Here, the value of ‘I’, ‘\(T_1\)’ and ‘\(T_a\)’ are constant and only ‘\(a0\)’ and ‘\(a1\)’ values are going to change due to degradation.

C. Pressure drop/Pressure gain in the System
The hydrodynamic effects of scaling are restriction in flow area inside the collector and associated increased flow resistance. In this section, the pressure gain and drop relationship of the TFS are determined as functions of mass flow rate and then solved to find the mass flow rate.

The pressure gain (\(\Delta P_{\text{gain}}\)) responsible for the flow can be considered in two parts (refer Figure 1), one representing the pressure gain across the collector (\(\Delta P_c\)) and the other between the top and bottom of the tank (\(\Delta P_{\text{tank}}\)) [5]. Therefore

\[ \Delta P_{\text{gain}} = \Delta P_c + \Delta P_{\text{tank}} \]  

(9)

Let \(T(x)\) is the temperature at distance \(x\) from the inlet of the collector, the pressure due to buoyancy forces (\(\Delta P_c\) and \(\Delta P_{\text{tank}}\)) can be evaluated by the following equations [6].

\[ \Delta P_c = \frac{g \beta \rho_2 \sin \theta}{2} \int_0^L (T(x) - T) dx \]  

(10)

Hence (9) is written as

\[ \Delta P_{\text{gain}} = \frac{g \beta \rho_2 \sin \theta}{2} \int_0^L (T(x) - T) dx + g H_w (T_2 - T_1) \]  

(11)

The variation of temperature in the risers can be described by a linear equation [7].

\[ T(x) - T_1 = \frac{T_2 - T_1}{L_r} x \]  

(12)

By integrating (12)

\[ \Delta P_{\text{gain}} = g \beta \rho_2 \sin \theta \left[\frac{L_r \sin \theta}{2} + H_w\right] \]  

(13)

By incorporating H-W-B equation in (13)

\[ \Delta P_{\text{gain}} = g \beta B I A_p \left[0.5 H_r + H_w\right] \left[a_0 - a_1 \frac{(T_1 - T_a)}{I}\right] \]  

(14)

Where \(\beta\) is coefficient of cubical expansion (K\(^{-1}\)), \(\rho_2\)’ is density of water at FPC outlet (kg/m\(^3\)), FPC orientation ‘\(\theta\)’ (°). Riser length ‘\(L_r\)’ (m), ‘\(H_w\)’ is head of water in tank, ‘\(T_2\)’ is temperature of water at FPC exit (K), ‘\(B\)’ is Constant relating density to temperature (kg/m\(^2\)K). Number of risers in FPC is ‘\(N\)’ (-), ‘\(m\)’ is mass flow rate in riser (kg/s) and ‘\(H_r\)’ is the height of riser (m).

Fig. 1 Thermosiphon head in TFS

The total frictional pressure drop (\(\Delta P_{\text{drop}}\)) in the circuit can be separated into the contribution from the collector (\(\Delta P_c\)) and that from the piping components outside the collector (\(\Delta P_{\text{ep}}\)) as shown in Figure 2.
\[ \Delta P_{\text{drop}} = \Delta P_c + \Delta P_{pc} \] (15)

The various losses within the collector \( (\Delta P_c) \) is the sum of friction loss in the risers \( (\Delta P_{rf}) \), friction loss in the header and footer \( (\Delta P_{hf}) \) and losses due sudden contraction and expansion \( (\Delta P_{ce}) \) in the footer-riser-header assembly.

\[ \Delta P_c = \Delta P_{rf} + \Delta P_{hf} + \Delta P_{ec} \] (16)

Friction loss in the risers is

\[ \Delta P_{rf} = \frac{12 \beta_0 \theta r \tau}{\pi d_i^4} \] (17)

Losses due sudden contraction and expansion \( (\Delta P_{ce}) \) in the footer-riser-header assembly is

\[ \Delta P_{ec} = \frac{12 m^2 N d_i}{\rho_r \pi^2 d_i^4} \] (18)

Where, \( \theta_r \) and \( \rho_r \) are kinematic viscosity \( (m^2/s) \) and density \( (kg/m^3) \) of water in riser, \( d_i \) is flow diameter of riser \( (m) \).

Similarly, different losses encountered in the piping components \( (\Delta P_{pc}) \) are the friction loss in connecting pipes \( (\Delta P_{cpf}) \), loss due bends fitted to connecting pipe \( (\Delta P_{cpb}) \) and entry and exit losses at the tank \( (\Delta P_{tank}) \).

\[ \Delta P_{pc} = \Delta P_{cpf} + \Delta P_{cpb} + \Delta P_{tank} \] (19)

Referring laminar flow throughout the system and including various pressure loss terms in (15) \[5\]

\[
\Delta P_{\text{drop}} = \left[ \Delta P_{rf} \left( 1 + \left( \frac{h_k}{h_r} \right) \left( \frac{d_i}{d_i} \right)^4 + \left( \frac{h_k}{h_r} \right) \left( \frac{d_i}{d_i} \right)^4 \right) \right] + \\
\left[ \Delta P_{ec} \left( 1 + \left( \frac{d_i}{d_i} \right)^4 + \left( \frac{d_i}{d_i} \right)^4 \right) \right] (20)
\]

For any flow system, if both the pressure gain and the pressure drop characteristics are known, the flow rate from the two relationships can be solved \[8\].

D. Heat transfer rate calculations

Useful heat flow in the FPC is given by

\[ Q_u = IA_p \eta_l \] (21)

Heat loss from the collector ‘\( Q_t \)’ (W) to the surrounding is \[9\]

\[ Q_t = U_h A_p (T_p - T_a) \] (22)

Flowing fluid convective coefficient is derived by the combined mode heat transfer correlation mentioned below as the value of \( Gr/Re^2 \approx 1 \) all the time \[10\].

\[ Nu = 1.75 \left( \frac{Gr}{Re} \right)^{0.14} \left( G\theta + 0.012 (G\theta (Gr \cos \theta)^{0.333})^{1.333} \right) \] (23)

IV. STUDY AND VALIDATION OF NUMERICAL CODE

The complete analysis of TFS is done using FLATSCALE for clean/scaled condition. The flow chart given in Figure 3 explains different steps involved in the performance analysis.

For the validation of numerical code, experimental and/or theoretical data obtained by Chuawittayawuth & Kumar \[11\] and Belessiotis & Mathioulakis \[12\] are incorporated in FLATSCALE and the analysis is then compared with results available in the quoted papers. The TFS specification used in the analysis is given in Table I.

1. Friction loss in the riser
2. Friction loss in connecting pipes
3. Friction loss in Header and Footer
4. Pressure loss due to bends
5. Pressure loss at tank entry
6. Pressure loss at tank exit
7. Pressure loss due to sudden contraction
8. Pressure loss due to sudden expansion

Fig. 2 Various pressure losses in TFS

Where, ‘\( L_h \)’ is the length of header/footer \( (m) \), diameter of header/footer is ‘\( d_h \)’ \( (m) \) and ‘\( d_{cp} \)’ is diameter of connecting pipe \( (m) \).

Fig. 3 Flow chart for the TFS

A. Case - I

It includes the experimental and theoretical (using TRNSYS software) values incorporated by Chuawittayawuth and Kumar.
[11] and values obtained by FLATSCALE (refer Table II).

Figure 4 shows the variation between experimental and analytical (FLATSCALE) water mean temperature. The maximum deviation noticed is 2.77%. Figure 5 indicates slightly higher analytical absorber plate temperature compared to experimental one. The maximum deviation is 15.55%.

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<td>Collector area</td>
<td>$A_p$</td>
<td>m²</td>
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<td>2.0</td>
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<td>Inner diameter of the riser</td>
<td>$d_r$</td>
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<td>0.008</td>
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<td>3</td>
<td>Diameter of the connecting pipe</td>
<td>$d_{cp}$</td>
<td>m</td>
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<td>0.027</td>
<td>0.022</td>
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<tr>
<td>4</td>
<td>No. of risers</td>
<td>$N$</td>
<td></td>
<td>--</td>
<td>9</td>
<td>17</td>
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<tr>
<td>5</td>
<td>Collector tilt</td>
<td>$\Theta$</td>
<td>°</td>
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<td>15</td>
<td>45</td>
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<tr>
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<td>Riser length</td>
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<td>1.8</td>
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<td>7</td>
<td>Tank height</td>
<td>$H$</td>
<td>m</td>
<td>0.40</td>
<td>0.40</td>
<td>0.45</td>
</tr>
<tr>
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<td>Riser pitch</td>
<td>$W$</td>
<td>m</td>
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<td>0.115</td>
<td>0.11</td>
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<tr>
<td>9</td>
<td>Bottom insulation thickness</td>
<td>$\delta_b$</td>
<td>m</td>
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<td>0.03</td>
<td>0.040</td>
</tr>
<tr>
<td>10</td>
<td>Connecting pipe length</td>
<td>$L_{cp}$</td>
<td>m</td>
<td>------</td>
<td>1.80</td>
<td></td>
</tr>
</tbody>
</table>

Table I SPECIFICATION OF TFS USED IN THE STUDY

<table>
<thead>
<tr>
<th>S. No.</th>
<th>Solar radiation level $I$ (W/m²)</th>
<th>Ambient temp. $T_a$ (K)</th>
<th>Water inlet temp. $T_{i}$ (K)</th>
<th>Water mean temperature $T_m$ (K)</th>
<th>Absorber plate mean temperature $T_p$ (K)</th>
<th>Mass flow rate $M$ (kg/m²h)</th>
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<td>298</td>
<td>303</td>
<td>306</td>
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<td>305</td>
<td>313</td>
<td>316</td>
<td>319</td>
<td>326</td>
</tr>
</tbody>
</table>

Table II ENERGY EFFICIENCY PARAMETERS

Figure 6 shows the difference between experimental and analytical mass flow rate from the unit area of FPC. The maximum variation between the two is found to be 66%. This is due to the assumptions made in the analysis. Figure 7 shows the deviation of collector efficiency between two sets of readings. The maximum percentage deviation calculated is 44.70%. This is high as both the outlet water temperature and flow rate are strong functions of collector efficiency.

B. Case - II

In this section, comparison of the data given by Belessiotis and Mathioulakis [12] with FLATSCALE results is presented. Since a few important data were not directly available, they were found by referring the plots given in the research paper.
Figure 8 shows the variation between experimental and analytical (FLATSCALE) water outlet temperature. The maximum deviation noticed is 13.6%. Figure 9 explains the difference between experimental and theoretical mass flow rate from the unit area of FPC. Here the maximum variation calculated is 20.18%. Figure 10 delivers the deviation of collector efficiency between two set of readings. The maximum difference calculated here is 25.73%. This is high since both outlet water temperature and flow rate are the strong functions of collector efficiency.

V. CONCLUSION

By referring two cases, it is clear that all the energy efficiency parameters given in the literature and computed by FLATSCALE are in good agreement. Hence, the numerical code can be further included to predict the performance deterioration of TFS due to scaling.

REFERENCES