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# Optimization of Thermosyphon Solar Water Heaters Using TRNSYS. Part1: Improved Model development and Validation

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**Abstract.** TRNSYS is a standard simulation environment that can be successfully used to evaluate thermal performance of the thermosyphon solar water heaters through the use of component Type 45. However, it is possible for errors to be incurred in the case of performing optimization of some of the design parameters of a thermosyphon system. This is thought to be due to the fixing of the values of collector performance characteristics and tank overall heat loss coefficient throughout the optimization process in the case of changing collector and tank parameters. The suggestion in this study is to add two new components in order to account theoretically for this situation as well as to change the characteristic performance of the collector, pipe and tank loss coefficients from being parameters to instead being inputs to the current version of Type 45. The study has shown that linking these two components with the modified thermosyphon-collector component Type 245 (referred to as 'Modified TRNSYS Model') gives results that agree to within an error of less than 7% with the traditional way of evaluating system performance (solar fraction) by using Type 45 (referred to as 'Original TRNSYS Model'). Further a more, the modified TRNSYS model eliminates restrictions on design parameter optimization and gives a wider choice for conducting parametric studies.

**Keywords:** thermosyphon solar water heater, TRNSYS, new TRNSYS types, validation

## 1. Introduction

Welcome There have been many attempts in the literature to model thermosyphon solar water heaters [**[1]**-**[11]**] and most of them have failed to accurately represent the system. Recently, there appears to have been no attempt to improve on, or to create new models that treat the system more rigorously. The only program available and used by most researchers in this field is the TRNSYS Type45. This model has been validated by a number of researchers [**[12]**,**[13]**] and has been used frequently to investigate, evaluate and optimise the thermal performance and the design parameters of thermosyphon solar water heaters [**[14]**-**[20]**].

The thermosyphon component in the TRNSYS program, Type45 requires a number of experimentally-determined items of information (namely, $F_R\tau\alpha$, $F_RU_L$, $b_o$, $U1$, $U2$, $UA_t$). This, in turn, requires that evaluation or optimisation of a system should be carried out on an already existing system where the components are already tested and reported. No doubt, evaluating the thermal performance of the system in the presence of experimentally-determined information will give more accurate results than calculating that information theoretically. However, in the case of optimisation, keeping that information fixed despite the fact that its values change throughout the optimisation process will probably lead to inaccurate results. Therefore, recourse to theoretical models to re-calculate those values during the optimisation process would be better for obtaining more accurate results.

Some of the aforementioned studies have used TRNSYS for conducting parametric studies on thermosyphon solar water heaters in order to see the effect on the thermal performance of the system without

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considering the effect on the collector characteristics or tank heat loss. A theoretical model for a flat plate collector used in this study shows that changing riser diameter from 5mm to 15 mm for three different collectors will change the value of $F_R\tau\alpha$ by an average of 1% and the value of $F_R U_L$ by 7%. Furthermore, changing the number of risers in the collector from 5 to 15 results in more than an 11% increase in $F_R\tau\alpha$ and a 13% increase in $F_R U_L$. Changing the collector aspect ratio and the dimensions will, of course, change the collector performance characteristics dramatically. Similarly, changing the dimension and volume of the insulated tank will change the overall heat loss coefficient of the tank. Combination of these changes might have a significant effect on the evaluation of the system performance.

In this study, an attempt is made to achieve a better estimation of the effect of changing design parameters on the thermal performance of a thermosyphon system. Two components were added to the TRNSYS model to account for the information that is determined experimentally. These components are: collector characteristics (Type210) and pipe-tank losses coefficients (Type211)

# 2. TRNSYS New and Modified Components

## 2.1. Collector characteristics component (Type210)

The flat plate solar collector is considered to be the most important part of a thermosyphon solar water heating system. It is a very special type of heat exchanger that converts solar radiation into thermal energy that heats the working fluid passing through it. The collector consists of many parts that need to be sized properly for efficient collector design. This, of course, requires accurate knowledge of collector thermal analysis. In fact, the thermal analysis of the flat plate solar collector remains a very difficult problem due to the structure and complexity of the collector, and hence the assumptions imposed to simplify the analysis. The literature has revealed a great deal of work devoted to the study and modelling of flat plate solar collectors. From the literature, this study examines four models, with the aim of identifying the best one for further analysis and modelling. The models under consideration are: i) CoDePro program **[23]** based on the Hottel-Whillier-Bliss (HWB) equations; ii) the model of Kirchhoff and Billups incorporating back and sides losses coefficients as they were neglected in the original work this is denoted here as (Model_1) **[22]**; iii) the model of Prabhakar et al **[21]**, which includes the effect of back loss from the absorber to the ambient, as well as the energy balance equation to calculate the cover temperature as given in **[22]** this model is denoted here as (Model_2); iv) the last model comprises the same equations used by model_1 but a different boundary condition between the absorber plate and riser tubes as shown in figure (1) (this model is denoted here as Model_3).
The boundary condition derived between the fin and water in the tube is:

$$2\left(-k_f\delta_f\left.\frac{\partial T_f}{\partial x}\right|_{(0,y)}\right)=U_{fw}\pi D\left(T_f-T_w\right) \tag{1}$$

Where $T_f$ is the fin temperature, $T_w$ is the water temperature and $U_{fw}$ is the heat transfer coefficient between the fin and the water in tubes.

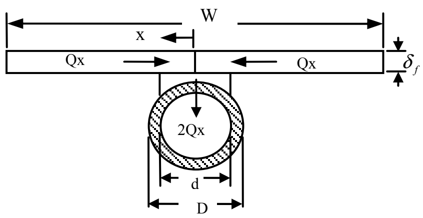


Figure 1: Fin and tube cross section

'CoDePro' is a software program provided by Wisconsin University that is available online **[23]**, its model being based on the Hottel, Whillier, and Bliss equation. The other three models (models 1,2 and 3 referred to earlier) consist of differential equations governing the temperature distribution of the plate, glazing cover, and fluid, these being solved numerically by using the finite difference technique and implicit scheme of time in which $N \times M$ simultaneous algebraic equations are obtained for N nodes. The algebraic equations obtained are solved iteratively by using the Gauss Seidel iterative method. The equations are encoded (compiled) by using Fortran 90, and runs on Intel Visual Fortran 9. The program shows very good stability for different mesh size of collector fins, and the best agreement compared to the experimental ones was obtained when the number of nodes in the X direction of the fin (as shown in figure (1)) is taken as equal to or greater than 4 according to the fin width at $\Delta X = 0.0125$ m.

The resulting predictions from these models are compared with data determined experimentally for three collectors tested according to standard EN 12975-2. These comparisons are shown in figures (2-4), and the main specifications of these collectors are listed in table1.

Table 1: Main collector specifications

| Parameter | Collector 1 [[25]] | Collector 2 [[26]] | Collector 3 [[27]] |
|---|---|---|---|
| Brand name | EWF-T2 | Titan Series2 | Derya-DSC25 |
| Collector area (m2) | 2.272 | 1.99 | 2.489 |
| Number of risers | 8 | 13 | 10 |
| Riser diameter (mm) | 6.4 | 9.52 | 7 |
| $F_R \tau\alpha$ | 0.779 | 0.802 | 0.766 |
| $F_R U_L$ | 4.336 | 4.083 | 4.592 |

From figures 2 to 4 and the calculation of the average root mean square (RMS) error of the results, Model_3 shows good agreement with the experimental data to within an RMS error of less than about 1.06%. This compares with errors RMS of the other three models as 2.21%, 2.7% and 1.21 % for CoDePro, Model_1 and Model_2, respectively. Therefore Model_3 was chosen to represent the characteristic performance of the flat plate solar collector (Type210) in this study.

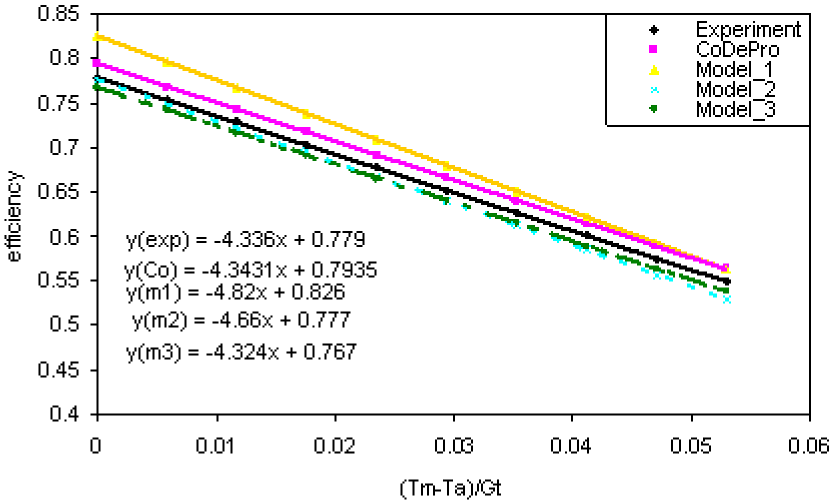


Figure 2: Experimental and theoretical efficiency curves for collector 1

## 2.2. Pipe-Tank Heat Losses Coefficients (Type211)

After the solar collector, the next most important part of a thermosyphon system is the storage tank. It is an essential requirement for storing hot water and keeping it warm for as long a time as possible in order to provide the user with sufficient hot water. To do this effectively, one requires the correct volume and level of insulation of the storage tank, taking into consideration the cost effectiveness. Hence, tank volume and overall heat loss coefficient of the tank are very important parameters in determining the thermal performance of the solar system. Altering the tank volume, aspect ratio or the insulation thickness is likely to change the value of heat loss coefficient of the tank. In Type 45 model, the overall heat loss coefficient of the tank, is considered as a parameter (fixed value) and hence, changing the volume and the height of the tank, of course, will affect the overall loss coefficient of the tank and must be considered in performing any parametric study.

In this study, the overall tank heat loss coefficient is calculated theoretically to account for changes of tank dimension and insulation material and thickness during the optimization process. The calculation is made according to ISO 9459-2.**[28]**

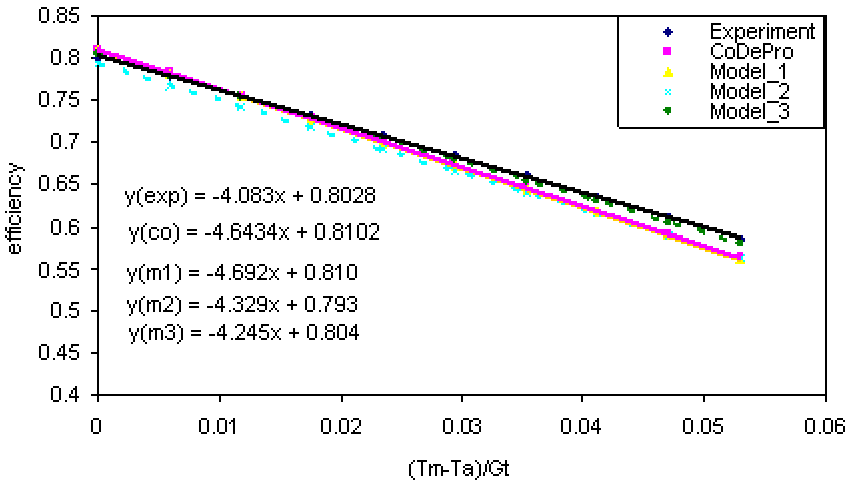


Figure 3: Experiment and theoretical efficiency curves for collector 2

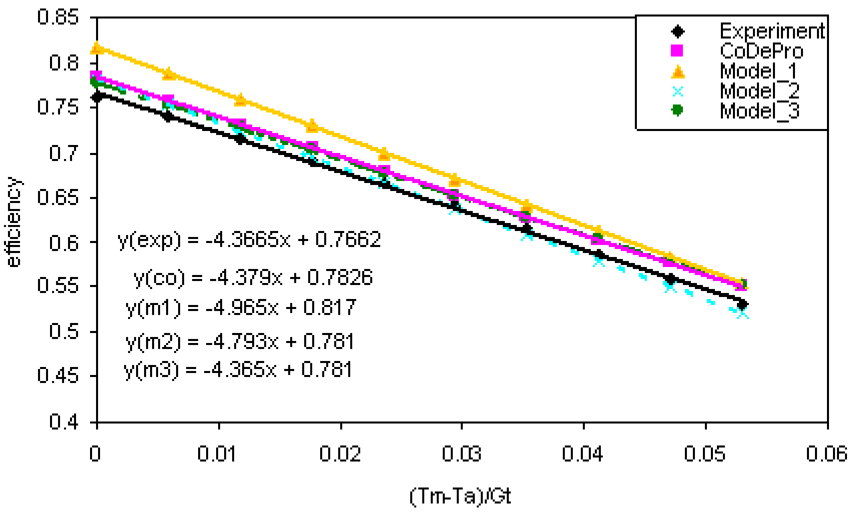


Figure 4: Experimental and theoretical efficiency curves for collector 3

In the theoretical procedure, the tank is divided into three parts: top, bottom and sides. Heat loss from the tank can be estimated as:

$$Q_{t,Loss} = Q_T + Q_B + Q_S \tag{2}$$

Ignoring the effect of stratification in the tank and assuming the water in the tank to be well mixed at temperature ($T_{ave}$). Equation 1 can be rewritten as :

$$(UA)_t = (UA)_T + (UA)_B + (UA)_S \tag{3}$$

The procedure for determining the heat loss coefficient of the tank in ISO 9459-2 prescribes that the wind should flow freely around the tank with a velocity in the range 3 – 5 m/s. Hence, correlations for forced convection are considered appropriate for determining convection heat transfer coefficients between the outer surface of the tank and the ambient environment.

Many empirical correlations are available in the literature to calculate heat transfer coefficients from different surfaces and shapes. In this study, the best agreements compared to the experiment are obtained by using the following correlations.

For the top and bottom of the cylinder, correlations for forced convection over flat plates are used **[24]**:

$$Nu = 0.45 + \left(0.6774\phi^{0.5}\right)\left\{1 + \frac{(\phi/12{,}500)^{\frac{3}{5}}}{\left[1 + (\phi_{um}/\phi)^{\frac{7}{2}}\right]^{\frac{2}{5}}}\right\}^{\frac{1}{2}} \tag{4}$$

and

$$\phi = \mathrm{Re}\,\mathrm{Pr}^{\frac{2}{3}}\left[1 + \left(\frac{0.0468}{\mathrm{Pr}}\right)^{\frac{2}{3}}\right]^{-\frac{1}{2}} \tag{4-a}$$

$$\phi_{um} \approx 1.875\phi(\mathrm{Re} = \mathrm{Re}_u) \tag{4-b}$$

$\mathrm{Re}_u$ is the Reynolds number at the end of the turbulent transition region. The above equation can be used either for constant wall temperature and uniform heat flux cases. The correlation as well can predict heat transfer from laminar region through transition and into turbulence regions **[24]**.

For the sides of the cylinder, the following correlations are used **[24]**.

$$Nu = 0.3 + \frac{0.62\,\mathrm{Re}^{0.5}\,\mathrm{Pr}^{1/3}}{[1 + (0.4/\mathrm{Pr}^{2/3}]^{0.25}}\left[1 + \left(\frac{\mathrm{Re}}{282000}\right)^{5/8}\right]^{4/5} \tag{5}$$

The above equation under predicts by 20% in the range of $20{,}000 < \mathrm{Re} < 400{,}000$. Therefore, the recommended equation in this particular range is **[24]**:

$$Nu = 0.3 + \frac{0.62\,\mathrm{Re}^{0.5}\,\mathrm{Pr}^{1/3}}{[1 + (0.4/\mathrm{Pr}^{2/3}]^{0.25}}\left[1 + \left(\frac{\mathrm{Re}}{282000}\right)^{1/2}\right] \tag{6}$$

In addition, the radiant heat exchange between the outer surface of the tank and the surroundings are considered also. An iterative method is used to calculate the outer surface temperature of the tank. Using the above correlations as well as correlations for air and water properties as function of temperature, a computer program was written in Fortran 90 and run on Intel Visual Fortran 9. The program is validated by carrying out experiments for two different tanks and comparing the measured values for $UA_t$ with the predictions from the program (Type 211). The comparisons are shown in table 2 .

The predictions are in a good agreement with experiment, with an average error of less than 5.7 %. It is possible that, some of the error might be the result of irregularity in the tank insulation thickness. The experiment values are slightly higher than the measured values. This is due to the fact that, the heat dissipated from a circulating pump and connecting pipes between the inlet and outlet of the tank that is used to mix the water in the tank at beginning and ending of the experiment are not considered in the above values. The heat losses via the connecting pipes and the circulating pump can be estimated from their relative areas. The relative area of Tank 1 is about 3 % of the total surface area of the tank, hence, the modified value of the overall heat loss coefficient from the Tank 1 will change from 1.762 (Table 2) to 1.70 with a new estimated error of less than 1.0%. The relative area of the Tank 2 is about 4% of the total surface area of the tank and hence, the modified value of the overall heat loss coefficient from Tank 2 will become 1.88 with an error less than 3.7%.

Table 2: Measured and predicted heat losses for two hot water cylinders.

| Tank | Volume (Lit) | Aspect ratio (L/D) | Insulation (cm) | | | UAt (W/K) | | Error % |
|---|---|---|---|---|---|---|---|---|
| | | | Top | Bottom | Side | Exp. | Type211 | |
| 1 | 126 | 1.78 | 3.2 | 3.7 | 3.5 | 1.762 | 1.694 | 3.9% |
| 2 | 90 | 1.3 | 2.1 | 3.2 | 2.5 | 1.958 | 1.810 | 7.5% |
| | | | | Average error is | | | | 5.7% |

Furthermore, model Type 211 also includes a calculation of the inlet and outlet heat loss coefficients of the connecting pipes in the thermosyphon system.

## 2.3. Modified Thermosyphon-Collector Component Type 245

The thermosyphon-collector component model Type 45 has been slightly modified to accept the output of the new components mentioned above in section 2-1 and 2-2 as an input. In fact, no modifications are made in the main body of this component, the only change is to alter the six parameters ($F_R\tau\alpha$, $F_R U_L$, $b_o$, $U1, U2$, $UA_t$) in type 45 from being parameters (in the TRNSYS terminology) to instead be inputs. This will enable their values to vary during the simulation.

## 3. Testing the Model

In this study, two TRNSYS models are examined, the first model, which is referred to as the 'Original TRNSYS Model' (OM) is the TRNSYS usual procedure to model thermosyphon solar water heaters. The original TRNSYS model is comprises the following main components: thermosyphon-collector component type 45, weather data component type109, load profile, in addition to a flow mixer and diverter, output and utility components as shown in Figure 5a. The second model is referred to as 'Modified TRNSYS Model' (MM), and comprises the thermosyphon-collector component type245 as modified in this study, collector characteristics components type210, pipe-tank heat loss type211, weather data component type109, load profile, in addition to a flow mixer and diverter, output and utility components as depicted in Figure 5b.

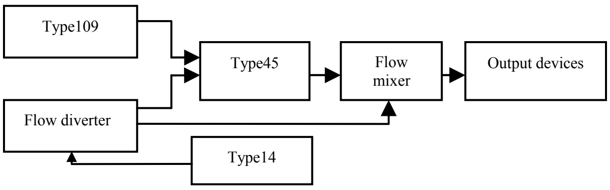


Figure (5 a) Schematic diagram of the major components of the original model

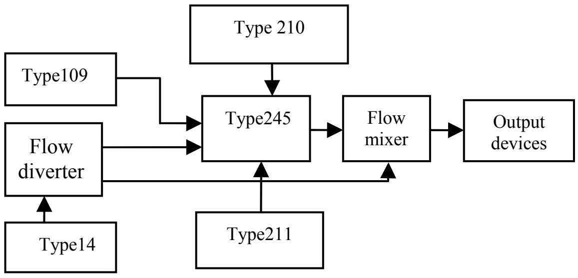


Figure (5 b) Schematic diagram of the major components of the modified model

## 4. Results and Discussion

Comparison between the modified TRNSYS model and the original TRNSYS model was made on two thermosyphon systems (system 1 and system 2) each having specifications as described in Table 3. In the comparison, it is assumed that the daily quantity of hot water withdrawn is 180 litres at 60 ºC, and is withdrawn according to the simple load pattern as shown in Figure 6. Weather data for Tripoli Airport, Libya, as provided by TRNSYS, is used in this study.

Table 3 system features

| System 1 | | | | System 2 | | | |
|---|---|---|---|---|---|---|---|
| $Ac$ | 2.272 m$^2$ | $V_t$ | 95 lit | $Ac$ | 2.489 m$^2$ | $V_t$ | 125 lit |
| $F_R\tau\alpha$ | 0.679 | $UA_t$ | 7.13 kJ/h | $F_R\tau\alpha$ | 0.6791 | $UA_t$ | 6.52 kJ/h |
| $F_R U_L$ | 13.77 kJ/h m$^2$ | $U1, U2$ | 8.7946 kJ/h m$^2$ k | $F_R U_L$ | 14.6052 kJ/h m$^2$ | $U1, U2$ | 8.7946 kJ/h m$^2$ k |
| $G_{test}$ | 72 kg/h m$^2$ | $V_{load}$ | 130 Lit/day | $G_{test}$ | 72 kg/h m$^2$ | $V_{load}$ | 130 Lit/day |
| $Dr$ | 6.4 mm | $H_t$ | 0.60m | $Dr$ | 7.0 mm | $H_t$ | 0.81m |
| $D_h$ | 22 mm | $H_r$ | 0.55 m | $D_h$ | 22 mm | $H_r$ | 0.75 m |
| $Nr$ | 8 | $H_{th}$ | 0.50m | $Nr$ | 10 | $H_{th}$ | 0.70m |
| $D_i$, $D_o$ | 22 mm | $H_{aux}$ | 0.45 m | $D_i$, $D_o$ | 22 mm | $H_{aux}$ | 0.58 m |
| $Hc$ | 1.38 m | $NB1$, $NB2$ | 4 | $Hc$ | 1.3 m | $NB1$, $NB2$ | 4 |
| $H_o$ | 1.54 m | $P_{aux}$ | 10 MJ/h | $H_o$ | 1.45 m | $P_{aux}$ | 10 MJ/h |
| $L_h$ | 1.087 | $T_{main}$ | 20 ºC | $L_h$ | 1.24 | $T_{main}$ | 20 ºC |
| $L_i$ | 2.5 m | $T_{set}$ | 60 ºC | $L_i$ | 2.5 m | $T_{set}$ | 60 ºC |
| $L_o$ | 1.5 m | $\beta$ | 43 deg | $L_o$ | 1.75 m | $\beta$ | 43 deg |

The monthly and yearly solar fraction has been used as the measure for thermal performance of the thermosyphon system for both cases (original and modified models). The characteristic performances of the collectors listed in Table3 are based on the difference in temperature between inlet fluid and ambient divided by incident solar radiation on the collector plan as should be used in TRNSYS.

Figure 7 shows the monthly solar fraction of System1. It is clear that, there is little difference between the original and modified models. The modified model predicts slightly higher values of solar fraction (by maximum of 4.5%) as compared to the original model whereas the error in estimating yearly performance is less that 3.1%.

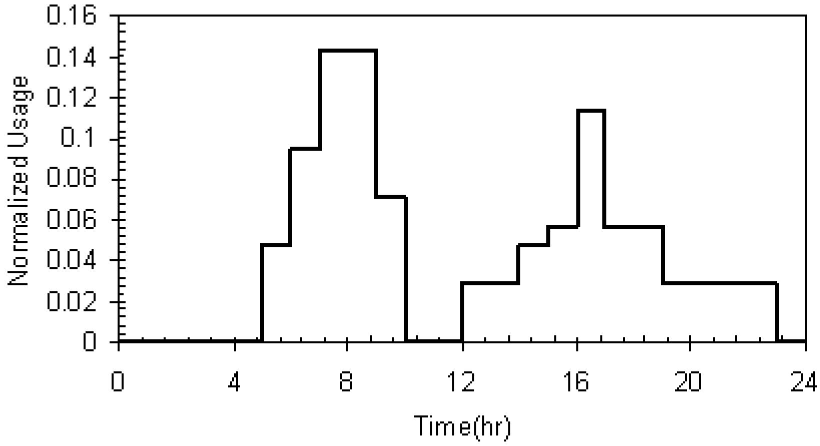

Figure 6: Hot water load pattern used for the study

In a similar way, the agreement between original and modified TRNSYS models in case of System 2 is also very good as can be seen in Figure 8, with slightly higher predictions of monthly solar fraction by the modified model of no more than 7.5%, and the yearly error prediction of less than 4.9%.

The study has shown that the errors incurred by calculating theoretically the characteristic performance of the collector and overall heat loss coefficient of the tank instead of measuring them experimentally can be considered acceptable for the purpose of evaluating and optimising thermosyphon systems. Therefore, the modified TRNSYS model (MM) can be used for conducting parametric study of the design parameters of thermosyphon solar systems. This is described in Part 2 of this paper.

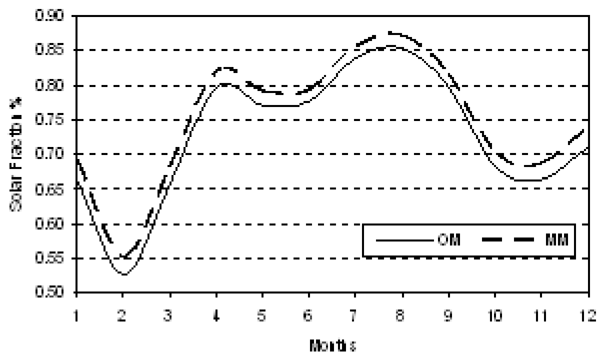

Figure 7 Monthly solar fraction of system 1

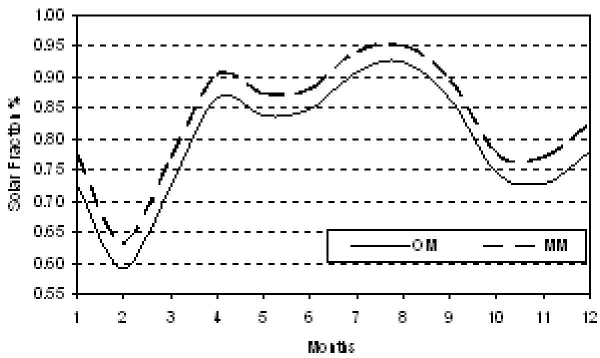

Figure 8 Monthly solar fraction of system 2

## 5. Conclusions

Two new components have been added to TRNSYS for use with a or the modified thermosyphon-collector (Type245) to account for the experimentally determined information ($F_R\tau\alpha$, $F_RU_L$, $b_o$, $U1$,$U2$, $UA_t$). The new components were validated against experimental data and good agreement was obtained between the models predictions and experiment. The collector characteristics component type210 shows very good agreement with three different sets of experimental data, with an average RMS error of 1.06%. The pipe-tank heat loss coefficients model type211 also gives good agreement with the data from two experiments conducted for validation purposes. This gave an average relative error less than 5.7%. These models were incorporated into a modified TRNSYS thermosyphon model (MM) the latter giving close agreement with the results of the original TRNSYS thermosyphon model (OM) when tested with two different thermosyphon systems, system 1 and system 2. Yearly errors of less than 3.1% and 4.9%, respectively were obtained on comparison of predictions. From original and modified models, it is concluded that the modified TRNSYS model developed in this paper can be used for parametric investigations of thermosyphon systems. The model offering the ability to vary collector characteristics as predicted during the calculation, as apposed to using fixed values determined from measurement a parametric investigation of this kind is presented in Part 2.

# 6. Nomenclature

| | | | |
|---|---|---|---|
| $Ac$ | Collector area m$^2$ | $P_{aux}$ | Auxiliary energy input to tank (KJ/hr) |
| $b_o$ | Incidence angle modifier coefficient. | Pr | Prandtl number |
| $D_h$ | Header diameter (m) | $Q_B$ | Bottom heat loss from tank (KJ/hr) |
| $D_i$, $D_o$ | Inlet and outlet connecting pipes diameters (m) | $Q_S$ | Side heat loss from tank (KJ/hr) |
| $Dr$ | Riser diameter (m) | $Q_T$ | Top heat loss from tank (KJ/hr) |
| $F_R\tau\alpha$ | Intercept of the collector efficiency curve | $Q_{t,Loss}$ | Total heat loss from tank (KJ/hr) |
| $F_RU_L$ | Slope of the collector efficiency curve | Re | Reynolds number |
| $G_{test}$ | Collector flow rate at test condition (kg/s m$^2$) | $T_f$ | Local fin temperature (°C) |
| $H_{aux}$ | Auxiliary heater position height (m) | $T_{main}$ | Temperature of water supplied from the main (°C) |
| $Hc$ | Collector perpendicular height (m) | $T_{set}$ | Auxiliary heater setting temp (°C) |
| $H_o$ | Height from datum to the tank bottom (m) | $T_w$ | Local water temperature in the riser (°C) |
| $H_r$ | Upriser height from the tank bottom (m) | $UA_t$ | Overall UA value for tank (KJ/hr ° C) |
| $H_t$ | Tank height (m) | $U1, U2$ | Loss coefficients for inlet and outlet pipes (KJ/hr m$^2$ ° C) |
| $H_{th}$ | Thermostat position height (m) | $U_{fw}$ | Heat transfer coefficient between water and fin (W/m$^2$K) |
| $k_f$ | Thermal conductivity of the plate (W/mK) | $V_{load}$ | Hot water load (Lit/day) |
| $L_h$ | Header length (m) | $V_t$ | Tank volume (Lit) |
| $L_i$, $L_o$ | Lengths of inlet and outlet pipes (m) | $\beta$ | collector tilt angle (deg) |
| $NB1$, $NB2$ | Number of equivalent right angle bends in inlet and outlet connecting pipes. | $\delta_f$ | Fin material thickness (m) |
| $Nr$ | Number of risers | $\Delta X$ | Step thickness in the X-direction (m) |
| $Nu$ | Nusselt number | $\Delta Y$ | Step thickness in the Y-direction (m) |

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