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Optimization of Thermosiphon Solar Water Heaters working in a Mediterranean Environment

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***Abstract*-- Cyprus is currently the leading country in the world with respect to solar water heaters installations. The increase of the thermosiphon solar water heaters usage over the last years makes this subject to be very important for further investigation. The main objective of this study is to investigate the parameters affecting the system's performance in order to find the optimum design to increase the performance of the system. For this purpose, a number of riser and header tube diameters were considered ranging from 6 mm to 35 mm, slopes from latitude from 20° to 90°, distances between the top of the collector to the bottom side of the storage tank ranging from -30 to +20 cm and the vertical or horizontal tank position. The system is modelled using TRNSYS and simulated with the Typical Meteorological Year (TMY) of Nicosia, Cyprus.**

It was observed that, the current typical system is not the optimum case and its operation can be further improved. The optimum system obtained from the simulations has a flat plate collector with header pipe of 22 mm and 20 riser pipes of 8 mm diameter, sloped at 40° and the distance between the top of the collector and the bottom of the storage tank is -20 cm. These findings should prove valuable for the collector and systems designers and manufacturers.

***Index Terms*— thermosiphon, flat plate collector, riser tubes, solar water heater, optimization, solar thermal.**

I. Nomenclature

A_c	Area of the collector (m^2)
F_R	Heat removal factor
I	Solar radiation (W/m^2)
Q_u	Rate of useful energy (W)
T_a	Ambient temperature (K)
T_{fi}	Inlet fluid temperature (K)
U_L	Overall heat loss coefficient (W/m^2K)
Greek	
τ	Transmittance of the cover plate
α	Absorbance of the absorber plate

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II. Introduction

As the energy crisis escalates and the price of gas and electricity increase, there is an inevitable shift to solar energy, which can be considered to be a unique opportunity to create a new clean energy economy. There are several solar energy technologies such as the solar thermal systems for heating, cooling and water heating, solar photovoltaics for electricity generation, solar architecture and artificial photosynthesis. Solar technologies tap directly into the infinite power of the sun and use that energy to produce heat and power. Lately, solar power systems are experiencing a rapid growth worldwide and mostly in the countries with high amount of solar radiation. Studies show that the worldwide leader country for the use of solar water heating systems per capita is Cyprus [1].

Solar radiation maps from all over the world show that Cyprus is between the countries with the highest amount of solar radiation. The mean daily global solar radiation varies from about 2.3 kWh/m^2 in the cloudiest months of the year December and January, to about 7.2 kWh/m^2 in July [1]. The graph in Figure 1 presents the solar thermal capacity of glazed water collectors in operation per 1000 inhabitants by the end of 2011 in the world. As can be observed, Cyprus is the leader country in terms of installed capacity of water collectors in operation per 1000 inhabitants. Between the EU countries, Cyprus had the highest installed collector capacity with 541.2 kWth/1000 inhabitants and then Austria and Greece with 355.7 kWth/1000 inhabitants and 268.2 kWth/1000 inhabitants, respectively. On the other hand, UK seems not to use those systems very much, as their installed capacity in 2011 was only 7.3 kWth 1000 inhabitants [1].

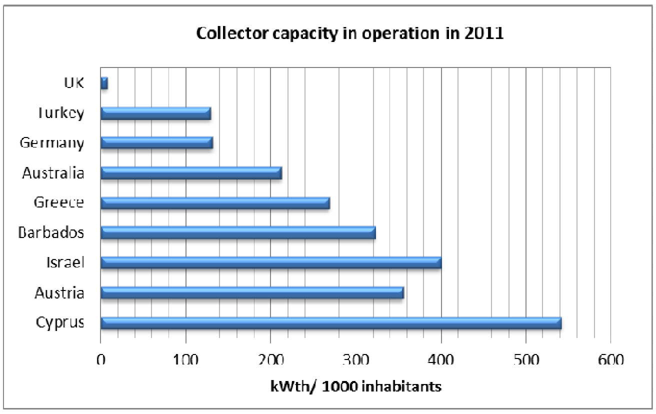

Fig. 1. Solar thermal capacity in operation per 1000 inhabitants in 2011 in Europe [1]

III. System under Investigation

The majority of the systems installed in Cyprus are of the thermosiphon type. Consequently, this study is focused on the performance of SWH systems, which operate thermosiphonically with natural circulation of water, in the Mediterranean environment of Cyprus. A typical system in Cyprus uses 3 m^2 of flat plate collectors, 160 lt storage tank, its collectors are usually inclined at 45° from horizontal. The collector has 10 copper riser tubes of 15 mm diameter, header tubes with a diameter of 28 mm and its absorber plate is also made from copper [2]. The storage tank was used to be vertically installed in the past few years while in the new systems is installed horizontally.

The characteristics of the flat plate collector considered in this study for the typical thermosiphon system, are shown in Table I.

TABLE I
THE CHARACTERISTICS OF A FLAT PLATE COLLECTOR FOR THERMOSIPHON SYSTEM IN CYPRUS.

Parameter	Characteristics
Riser pipe diameter	15 mm
Riser tubes material	copper
Number of riser pipes	10
Header pipe diameter	28 mm
Absorber plate thickness	0.5 mm
Glass type	4 mm low iron glass
Collector insulation	Fiberglass 30 mm sides
	Fiberglass 50 mm back
Glazing	Low iron glass
External casing material	Galvanized sheet

The general aim of this study was to investigate the design parameters affecting the performance of the SWH systems operating thermosiphonically. However, the main objective of this paper is to investigate through modeling and simulation possible configurations of the thermosiphonic system, which will optimize the performance of the system. For this purpose, the following parameters are investigated:

- Number of riser tubes
- Riser tube diameters from 6 mm to 28 mm
- Header tube diameters from 8 mm to 35 mm
- Slopes from 20-90°, 5° step size
- Distances between the top of the collector and the bottom side of the storage tank, from -30 to +20 cm
- Horizontal or vertical storage tank
- Length of the riser and header pipes in different collector's shapes

The investigation of the thermosiphonic system will allow us to propose a new improved system design with higher efficiency than the existing systems.

Initially, the system was modelled using TRNSYS and simulated with the Typical Meteorological Year (TMY) of Nicosia, Cyprus. To achieve the objectives mentioned above, each of the design parameters under investigation needs to be examined separately. This is because in this way, the effect of each parameter on the system's efficiency will be recorded.

IV. The Performance of the Collector

The basic performance parameter is the collector thermal efficiency. The solar collector thermal efficiency is defined as the ratio of the useful thermal energy leaving the collector to the usable solar irradiance falling on the aperture area given by:

$$\eta = \frac{\dot{Q}_u}{A_c x I} \quad (1)$$

Where $\dot{Q}_u$ = Rate of useful energy output (W)
A_c= Area of the collector (m^2)
I= Solar irradiance falling on the collector's area (W/m^2)

However, it is more convenient to express the collector performance in terms of the fluid inlet temperature. This equation is known as the classical Hottel-Whillier-Bliss (HWB) equation to predict the collector performance as follows:

$$\dot{Q}_u = A_c F_R \left[(\tau\alpha) I - U_L (T_{fi} - T_a) \right] \quad (2)$$

Where τ= Transmittance of the cover plate
α= Absorptance of the absorber plate
T_a= Ambient temperature (°C)
U_L= Overall heat loss coefficient ($W/m^2 \cdot K$)
T_{fi}: Inlet fluid temperature (°C)
F_R: Heat removal factor

From Eq. (2), the collector efficiency in terms of the heat removal factor is given by:

$$\eta = \frac{\dot{Q}_u}{A_c I} = F_R \left[(\tau\alpha) - U_L \frac{T_{fi} - T_a}{I} \right] \quad (3)$$

Heat removal factor of the collector, F_R is affected only by the solar collector characteristics, the fluid type and the flow rate through the collector. Consequently, from the equation of fluid temperature distribution, the heat removal factor can be expressed as:

$$F_R = \frac{\dot{m} C_p}{A_c U_L} \left[1 - \exp\left(\frac{A_c U_L F'}{\dot{m} C_p} \right) \right] \quad (4)$$

Where F′is the collector efficiency factor, a measure of how enhanced the heat transfer is between the fluid and the absorber plate. The collector efficiency factor F' is given by the following equation and all the dimensions are illustrated in Figure 2.

$$F' = \frac{\frac{1}{U_L}}{W \left[\frac{1}{U_L (D + (W - D) F)} + \frac{1}{C_b} + \frac{1}{\pi D_i h_{fi}} \right]} \quad (5)$$

Where F is the standard fin efficiency given by:

$$F = \frac{\tanh\left[m\frac{(W-D)}{2}\right]}{m\frac{(W-D)}{2}} \tag{6}$$

Where C_b= Bond conductance (W/m K)
k= thermal conductivity of the absorber (W/m K)
h_{fi}=convective heat transfer coefficient between the fluid and tube wall (W/m^2·K)

And the factor m is given by:

$$m = \sqrt{\frac{U_L}{k\delta}} \tag{7}$$

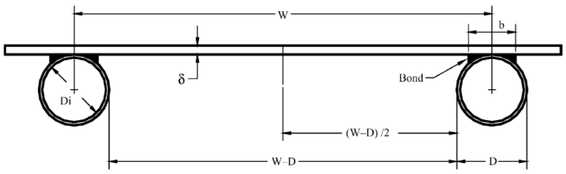

Fig. 2. Flat plate and tube configuration [3]

Morrison and Braun [4] studied the modeling and operation characteristics of thermosiphon solar water heaters with vertical or horizontal storage tanks. They found that the system performance is maximized when the daily collector volume flow is approximately equal to the daily load flow, and the system with horizontal tank did not perform as well as that with a vertical one. This model has also been adopted by the TRNSYS simulation program. According to this model, a thermosiphon system consisting of a flat-plate collector and a stratified tank is assumed to operate at a steady state. The system is divided into N segments normal to the flow direction, and the Bernoulli's equation for incompressible flow is applied to each segment. For steady-state conditions the sum of pressure drop at any segment is:

$$\Delta P_i = \rho_i g h_{fi} + \rho_i g H_i \tag{8}$$

And the sum of the pressure changes around the loop is 0; that is,

$$\sum_{i=1}^{N} \rho_i h_{fi} = \sum_{i=1}^{N} \rho_i H_i \tag{9}$$

where
ρ_i = density of any node calculated as a function of local temperature (kg/m^3).
h_{fi} = friction head drop through an element (m).
H_i = vertical height of the element (m).
For each time interval the thermosiphon flow rate must uniquely satisfy Eq. (9).

The friction head loss in pipes is given by:

$$H_f = \frac{fLv^2}{2dg} + \frac{kv^2}{2g} \tag{10}$$

where
d = pipe diameter (m).
v = fluid velocity (m/s).
L = *length* of pipe (m).
k = fitting loss coefficient.
f = friction factor.

The friction factor f, is equal to:

$$f = 64 / Re \text{ for } Re < 2000 \qquad f = 0.032 \text{ for } Re > 2000 \tag{11}$$

An estimate of the thermosiphon head may be found based on the relative positions of the tank and the flat-plate collectors as shown in Fig. 3. The thermosiphon head generated by the differences in density of fluid in the system may be approximated by making the following assumptions:

1. There are no thermal losses in the connecting pipes.
2. Water from the collector rises to the top of the tank.
3. The temperature distribution in the tank is linear.

Therefore, according to the dimensions indicated in Fig. 3, the thermosiphon head generated is given by:

$$h_T = \frac{1}{2}(S_i - S_o)\left[2(H_3 - H_1) - (H_2 - H_1) - \frac{(H_3 - H_5)^2}{(H_4 - H_5)}\right] \tag{12}$$

Where S_i and S_o is the specific gravity of the fluid at the collector inlet and outlet respectively. Here only direct circulation thermosiphon systems are considered in which water is the collection fluid. The specific gravity according to the temperature (in °C) of water is given by:

$$S = 1.0026 - 3.906 \times 10^{-5} T - 4.05 \times 10^{-6} T^2 \tag{13}$$

Therefore, as can be observed from the above equations, the performance of a SWH system depends on many factors such as the construction of the collector and the arrangement of the system, mainly with respect to the distance between the top of the solar collector and the bottom of the storage tank and the solar collector slope, which affects both the energy collected and the hydrostatic pressure of the system.

V. Technical Work Preparation

As already mentioned, the considerations of the parameters that affect the performance of the thermosiphon solar water heater have been made using TRNSYS simulation program through the use of component Type 45. There have been many attempts in the literature to model thermosiphon solar water heaters and most of them have failed to accurately represent the system [5]. However, the only program available and used by most researchers in this field is the TRNSYS Type 45. One such validation which concerns Cypriot manufactured units is from Kalogirou and Papamarcou in 2000 [6].

The various configurations evaluated, concerning header and riser pipes diameters are shown in Table II. As can be seen, riser pipe diameters vary from very small up to one size smaller pipe diameter than the header pipe.

TABLE II
HEADER AND RISER PIPE DIAMETER VARIATIONS

Header diameter (mm)	Riser diameters (mm)
15	6, 8, 10, 12
22	6, 8, 10, 12, 15
28	6, 8, 10, 12, 15, 22
35	6, 8, 10, 12, 15, 22, 28

Although all the parameters affecting the system's performance were conducted using TRNSYS environment, there was a need to define previously some inputs needed for the program for each case under investigation. For the collector, those parameters are the heat removal factor of the collector F_R, the collector heat loss coefficient U_L, the slope of performance curve ($F_R \cdot U_L$) and the intercept of the efficiency curve F_R ($\tau\alpha$). After the estimation of the inputs needed for the program, the design parameters were examined one by one and their effect in the performance of the system was recorded.

It is important to mention that the effect of the variations of a parameter on the system's efficiency was conducted by comparing the efficiency of the system every time that a change in the design was simulated.

A. Calculation of the Heat Removal Factor - F_R

Initially the collector performance characteristics were used to estimate the heat removal factor (F_R), the overall heat loss coefficient (U_L) and he transmittance absorptance product ($\tau\alpha$) of the collector. The heat removal factor depends on the collector efficiency factor, which depends on the heat loss coefficient, diameter of the riser tubes, distance between the riser tubes, collector fin efficiency and the internal riser pipe convection heat transfer coefficient. The heat transfer coefficient depends on the flow rate and is estimated from the Nusselt number, which is a function of the Reynolds number that determines the type of flow in the riser tubes.

Subsequently, the new heat removal factor is estimated for the corresponding riser pipe internal diameter as well as the number of riser tubes, which is determined by trial and error so as the resulting type of flow (Reynolds number) to remain the same in order to have an adequate circulation of water in the collector. By using this new heat removal factor the input parameters $F_R(\tau\alpha)$ and F_RU_L were calculated as well as the number of riser tubes, which are used in the various simulations. The values obtained from these calculations are shown in Table III. The bold letters show the specifications of the collectors at the typical thermosiphonic system with vertical storage tank.

TABLE III
MODIFIED COLLECTOR CHARACTERISTICS AND NUMBER OF RISER TUBES USED IN SIMULATIONS

Riser Diameter (mm)	Number of riser tubes	F_R ($\tau\alpha$)	F_R U_L (kJ/hr m^{2o}C)
6	29	0,836	25,263
8	20	0,825	24,94
10	16	0,817	24,685
12	13	0,807	24,395
15	**10**	**0,792**	**23,94**
22	7	0,765	23,13
28	5	0,731	22,103

Subsequently, the simulations started firstly by defining the efficiency of the typical system without any modifications. This will then be compared to the efficiency of the system each time that a different parameter is tested.

B. Typical system with vertical storage tank

The system set as typical in Cyprus that simulated at the beginning has a total aperture are 3 m^2 (2 panels), with storage tank capacity 160 lt installed vertically. The collector slope angle for this system was taken to be 45°. The collector for the typical system has 10 riser pipes made of copper with 15 mm diameter and header pipes with diameter 28 mm. As mentioned before, the slope of performance curve (F_RU_L) and the intercept of the efficiency curve F_R ($\tau\alpha$) for this system, are shown in bold letters in Table III.

The simulations carried out for the typical system with vertical tank, showed that the efficiency of the system was 40.13%.

This system is going to be the basis for comparison for the next simulations. Every parameter under investigation was applied on this system, one each time, and the best result selected to be remain constant for the next simulation to find the optimum design at the end.

Then, the typical system was simulated with the various combinations of the header and riser pipes diameters in order to identify the best combination of riser and header pipes diameter and the required number of riser diameters that gives the higher system's performance.

C. Riser and Header pipes diameter and number of risers

The data from Table III were used as inputs to the program for each size of header diameter and its related riser pipes number and diameter as presented in Table II in order to record the efficiency of the system for each situation. The simulations have been separated in four sets of runs for header diameters 15 mm, 22 mm, 28 mm and 35 mm for the appropriate sizes of riser tubes in each group.

It is important to mention that the tested system was the typical one with 45° slope of the collector and the bottom of the tank was in the same height with the top of the collector as shown in Fig. 3. The only variables in each simulation were the size of the riser and header tubes and the number of riser tubes to keep constant the flow of the water. Consequently, the performance for each simulation was recorded and the highest one was selected to be kept constant for the next simulations.

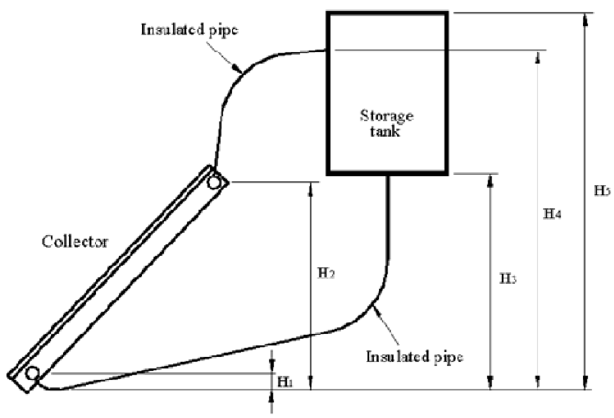

Fig. 3. Schematic diagram of a thermosiphon solar water heater.

In this way we were able to find the combination of the number of riser pipes and their diameter size as well as the header diameter that give higher performance of the system.

The results from the simulations are shown in Fig. 4 and as can be observed, a small riser diameter of 8 mm is the optimum. The smallest pipe of 6 mm is not selected because the systems employed in Cyprus are usually open, circulating the water supplied to the user directly into the collector, which may create scaling problems. A header diameter of 22 mm is selected in terms of performance because this is the same diameter as the connecting pipes between the collector and storage tank, which makes the installation easier.

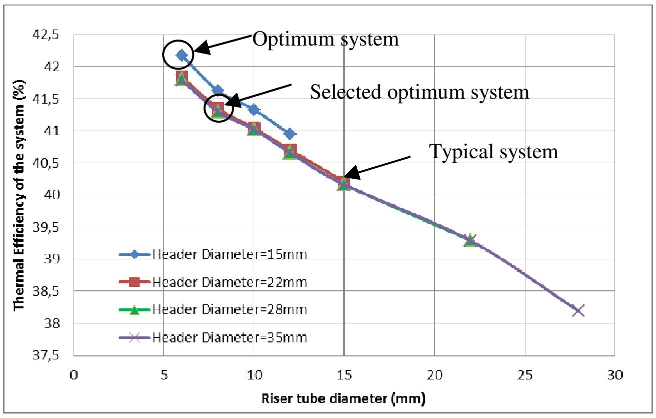

Fig. 4. Performance of the system for the tested collector's construction variations.

The improved system with the selected parameters has 2% higher efficiency than the typical system presented in the previous section.

After the selection of the optimum design parameters of the collector, those parameters were kept constant in order to investigate other parameters of the system such as the slope of the collector or the height of the storage tank. Step-by-step improvements on the system's design will allow us to have a new improved system at the end of the project.

D. Thermosiphonic system with horizontal storage tank

In the past years the storage tank of the thermosiphonic systems used to be vertically installed while in the recent years is being installed horizontally for the purpose of reducing the overall height of the unit. The two configurations are shown in Fig. 5.

Fig. 5. SWH system with horizontal storage tank (left) and SWH system with vertical storage tank (right)

After the definition of the dimensions of the connecting pipes and tank configuration in the program, the system with the horizontal tank was simulated and its efficiency was found to be 38.94%. It was observed that, the typical system with the vertical tank was 2% more efficient than the system with the horizontal tank. This is the same conclusion as Morrison and Braun found in 1985 [4].

As done for the typical system with the vertical tank, the selected parameters for the collector concerns the number of riser tubes and the diameter of the riser and header pipes were applied for this system and simulated. The outcomes from the simulation showed that the efficiency of the system was improved by 2.1%.

From now on, in the investigation of the rest of parameters the two systems are going to be presented the same time in order to have the opportunity to compare them and see the change in their efficiency for every modified parameter.

E. Slope of the collector

When constructing and installing a SWH it has to be assured that circulation of the heat carrying fluid (water) does occur. The circulation of the water in the collector depends in many factors such as the material of the pipes, the diameter of the pipes and the height that the water needs to be transferred. This height is depended with the slope of the collector. The lower is the height more easily the water moves.

The slope of the collector is very important for the system's performance so the system was simulated and tested in slopes from 20-90° at steps of 5°.

Figure 6 shows the results from simulations for slopes 20-90° for the selected number of risers and tubes diameters (from the previous section) for vertical storage tank configuration, for vertical height between the collector and the tank to be zero.

As can be observed, the optimum angle for this system is not the 45º that is used for the typical systems in real life so far. The chart in Fig. 6 shows that the typical systems with vertical tank perform slightly better in slopes of the collector around 35º-40º.

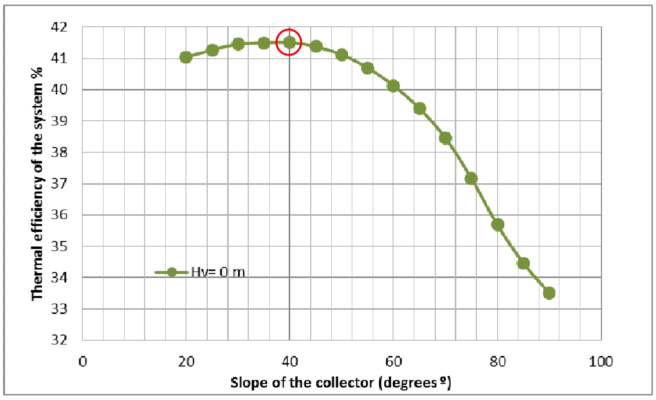

Fig. 6. Performance of the typical system with the improved collector characteristics for slopes 20-90°

Figure 7 shows the outcomes of the simulations for angles of the collector from 15º to 90º for a system with horizontal tank and the collector characteristics as selected from section C.

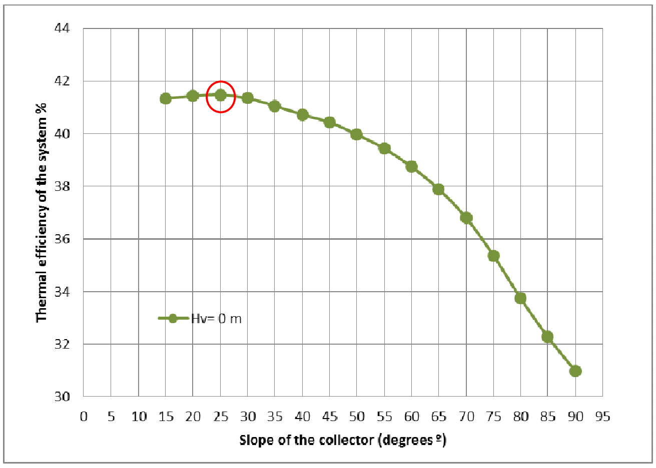

Fig. 7. Performance of the system with horizontal storage tank with the improved collector characteristics for slopes 15-90°

Figure 8 shows a comparison of the system performance between the vertical and horizontal tank configurations. A height between the collector top and the storage tank bottom of -20 cm is chosen as it shows more clearly the difference between the two arrangements. It is obvious that this system performs better in lower slope of the collector around 25º-35º. This outcome is very interesting if we consider that in real life, those systems are being installed in smaller angles but nobody ever said the reason for this. This outcome can somehow confirm the installation of those systems. From the results presented here it can be concluded that the system with horizontal tank performs better in lower inclination of the collector and this is the first written evidence which ensures that there is a reason for the way of their installation. It is now proved that their installation in lower slope, which was probably selected by experience and the outcomes from the simulations, agree.

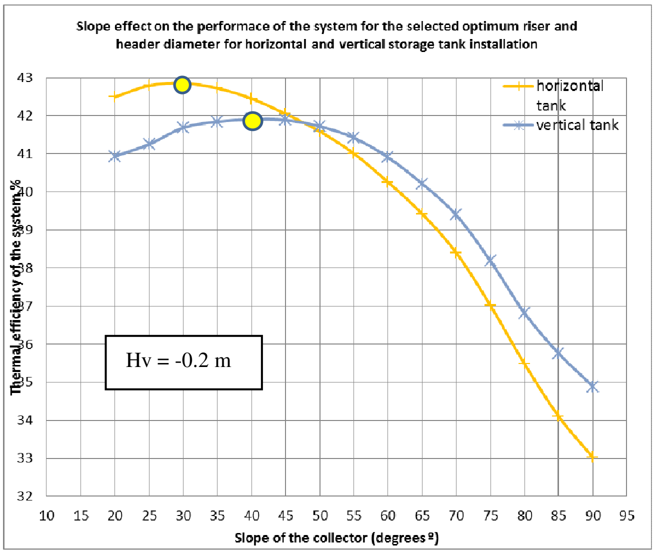

Fig. 8. Comparison of the performance of the system with vertical and horizontal storage tank

It is important to mention that the curves in Fig. 8 are not for the typical system with the tank and the collector at the same height. For the typical systems with the tank's bottom to be at the same height with the collector's top, the system with vertical tank is more efficient than the system with horizontal tank and the efficiencies of both of them are not higher than 41% as presented in previous sections. The graphs in Fig. 8 show an example of the slope's effect in systems with the bottom of the tank to be 20 cm below the top of the collector.

As can be observed, the bigger is the slope of the collector, the lower is the performance of the system. This is logical because we know that the circulation of the water depends on the dynamic forces ρ (density), g (gravity of earth), h (height). Consequently, the lower is the slope the lower is the height, thus the lower are the forces that need to circulate the water and the higher is the efficiency.

F. Height between the collector top and the tank bottom

Figure 3, presented before, shows schematically the typical thermosiphon system showing the bottom of the tank and the top of the collector to be at the same height. In this study, some different cases of this vertical distance were investigated, for the tank to be below or above the collector's top.

Six cases of this height difference were investigated, for vertical distance between the top of the collector and bottom of the tank -30 cm, -20 cm, -10 cm, 0 cm, 10 cm and 20 cm. The negative sign means that the tank's bottom is below the collector's bottom. Subsequently, the positive sign means that the bottom of the tank is above the top of the collector while zero distance means that they are in the same height level.

Figure 9 shows the results from 96 simulations for a thermosiphonic system with vertical storage tank for the six cases of heights discussed before, for slopes of the collector from 20º-90º. The results illustrated in Figure 9, show that the system with vertical tank performs better when the vertical distance between the top of the collector and the bottom of the tank is negative. That means that the system's efficiency is higher when the bottom of the tank is placed below the collector's top surface level. However, this doesn't mean that the tank should be placed on the ground level because in that case there will be a problem in the circulation of the water.

Additionally, as can be observed, the slope of 40° is the optimum for all the cases so this is the selected value to simulate the new selected parameters.

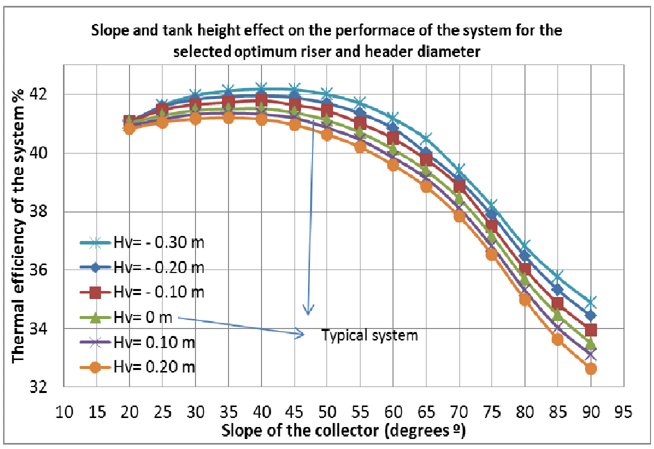

Fig. 9. Efficiency of the thermosiphon system with vertical tank in terms of the collector's slope and the height difference of the collector's top and the tank's bottom.

The efficiency of the system with vertical storage tank was improved by 0.5% with the changes in the slope of the collector and the height of the tank. In addition with the changes made in the collector's design characteristics presented in section C there is a totally 2.4% improvement from the typical system.

Figure 10 shows the outcomes from 96 simulations for 6 cases of vertical height between the collector and the tank for slopes between 15º and 90º for the system with horizontal storage tank. As can be observed the lower is the tank placed above the collector's top, the higher is the system's efficiency.

Additionally, it is clearly shown that the system with horizontal tank performs better in lower slopes of the collector than the one with vertical tank as shown previously in Fig. 8.

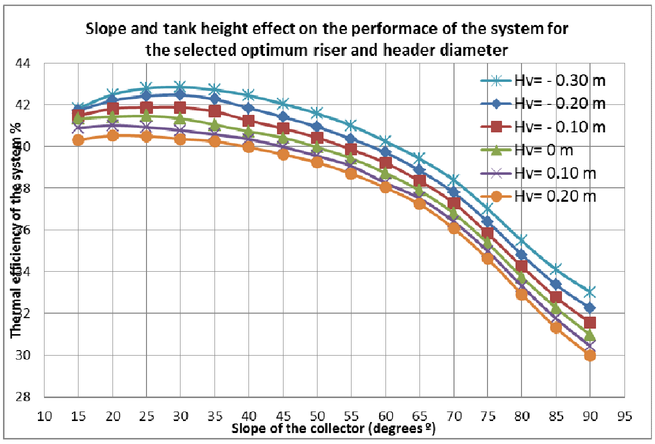

Fig. 10. Efficiency of the thermosiphon system with horizontal tank in terms of the collector's slope and the height difference of the collector's top and the tank's bottom

Accordingly for the system with horizontal storage tank, the slope of 30º was selected as optimum, and the height of the tank selected to be the 20 cm below the top of the collector. Then the selected parameters so far for the system with horizontal tank were simulated and the outcomes from the simulations showed that the performance of the system was 43.29% while before applying these changes it was 41.04%. This means that the changes in the collector achieved a 2.1% efficiency increase as mentioned in section D and the changes in the slope of the collector and the height of the tank improved the efficiency of the system by more 2.3%. In total we achieved to improve the system with the horizontal tank by 4.4 %.

G. *Different shape of the collector*

The design characteristics of the collector were investigated before but the shape of the collector was the same for every parameter tested so the length of the header and riser pipes was always the same. This section will show some other shapes that the collector could be manufactured. The aim is to prove whether those shapes are more efficient than the typical collector's shape. Consequently, the typical system with collector slope 45° and collector outlet to tank outlet vertical distance 0 m was taken as the basis in order to modify the shape of the collector and the related properties of its components to examine the system's efficiency in different shapes of the collector.

The length of the riser and header pipes depends on the shape geometry of the collector panel. Accordingly, a different shape of the collector for the same area (long distance to be the width instead of height that is the usual case), would require different length of riser and header pipes as well as different size diameters of the tubes in order to keep the flow constant.

This section, will describe the consideration of the dimensional characteristics of the collector's components and every shape tested will be compared with the collector of the typical system with 10 riser tubes of 15 mm and header tubes 28 mm. The typical collector has area of 1.5 m^2 where its length is 1.5 m and its width is 1 m which means that the length of the riser pipes was around 1.5 m each and accordingly the length of the header pipes was 2 m (2 pipes x 1 m each).

Four cases of different shapes of collector have been investigated in this study as follow:

1. Length= 1.5 m, Width= 1 m, Risers= 10 (typical)
2. Length= 1.15, Width= 1.30 m, Risers=13
3. Length= 1.875, Width= 0.8 m, Risers=8
4. Length= 1, Width= 1.5 m, Risers=15

The number of risers is estimated by considering that the distance between the riser tubes remains equal to 10 cm, as in the typical system, where the width is 1 m and there were 10 risers.

As can be observed, the Case 1 represents the typical system as described in section B with efficiency 40.13%. The results from the simulations of using a modified collector in the typical system are presented in Fig. 11.

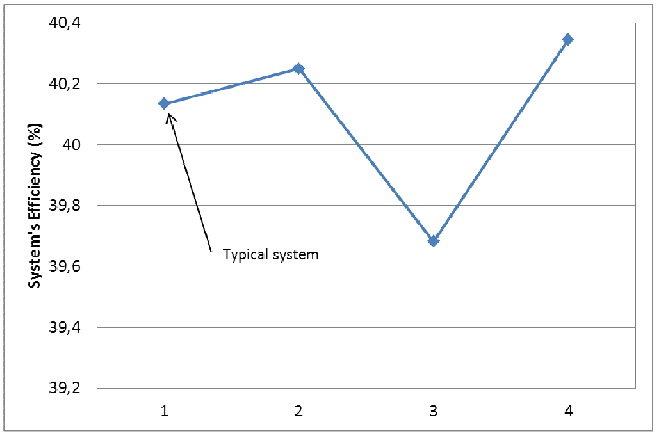

Fig. 11. Performance of the system in the four cases of collector's shapes

It is clearly shown that the Cases 2 and 4 are more efficient than the typical system but the difference between their efficiency is minimal. The efficiency of the system with collector from Case 1 differs from the efficiency of the Case 2 by less than 0.2% while the lower from the higher efficiencies presented in the diagram (Case 4) differs only by 0.66%. Case 3 with the taller collector seems to be the worst case.

Consequently, these parameters do not worth to be changed in the typical system, as their effect on the system's efficiency is almost zero.

VI. General outcomes

After the examination of all the parameters and the analysis of their effect on the thermosiphonic system's performance, this section will recapitulate the outcomes of the changes and their effects.

Figure 12 shows that in the typical form the system with vertical tank is more efficient than the system with the horizontal tank. Firstly, the collector's design characteristics have been changed and they improved the efficiency of the system by 2% for the system with vertical tank and by 2.1% for the system with horizontal tank.

Afterwards, the changes selected to be done on the height of the tank and the slope of the collector improved the system with the vertical tank by 0.5% and the system with horizontal tank by 2.3%.

These last changes overturned the results and the system with the horizontal tank is more efficient than the system with the vertical tank. This has the added advantage of reduced overall system height and should be the preference of the manufacturers.

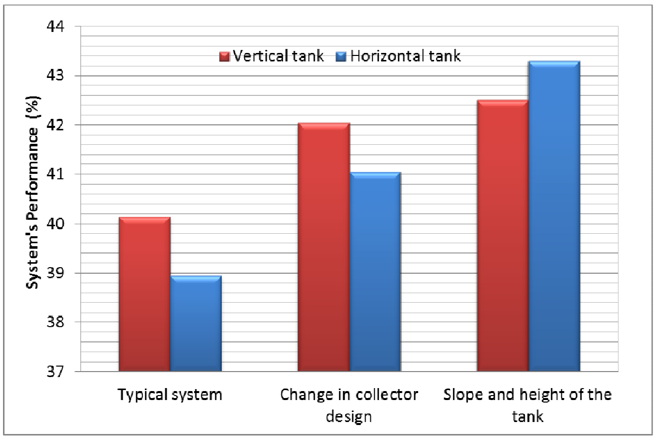

Fig. 12. Step by step improvements made on the systems with vertical and horizontal storage tank.

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VIII. Biographies

Soteris Kalogirou was born in Trachonas, Nicosia, Cyprus on November 11, 1959. He is a Senior Lecturer at the Department of Mechanical Engineering and Materials Sciences and Engineering of the Cyprus University of Technology, Limassol, Cyprus. He received his HTI Degree in Mechanical Engineering in 1982, his M.Phil. in Mechanical Engineering from the Polytechnic of Wales in 1991 and his Ph.D. in Mechanical Engineering from the University of Glamorgan in 1995. In June 2011 he received from the University of Glamorgan the title of D.Sc. He is Visiting Professor at Brunel University, UK and Adjunct Professor at the Dublin Institute of Technology (DIT), Ireland. For more than 25 years, he is actively involved in research in the area of solar energy and particularly in flat plate and concentrating collectors, solar water heating, solar steam generating systems, desalination and absorption cooling.

He has 41 books and book contributions and published 264 papers; 109 in international scientific journals and 155 in refereed conference proceedings. Until now, he received more than 4000 citations on this work and his h-index is 35. He is Deputy Editor-in-Chief of Energy, Associate Editor of Renewable Energy and Editorial Board Member of another eleven journals. He is the editor of the book Artificial Intelligence in Energy and Renewable Energy Systems, published by Nova Science Inc., co-editor of the book Soft Computing in Green and Renewable Energy Systems, published by Springer and author of the book Solar Energy Engineering: Processes and Systems, published by Academic Press of Elsevier

Rafaela Agathokleous was born in Limassol, Cyprus on June 1990. She graduated from the Ayias Fylaxeos Lyceum in 2008 and studied Mechanical Engineering and Materials Science Engineering at Cyprus University of Technology until 2012. In 2012 she participated as a last year undergraduate student in a competition organized by the Research Promotion Foundation in Cyprus and she earned first prize praise. By 2013 she studied for the degree of Masters of Science in Sustainable Energy Technologies and Management at Brunel University London.

Rafaela is a student member at the Chartered Institution of Building Services Engineers and member at the Cyprus Scientific and Technical Chamber.

She is currently employed as a Research Assistant at Cyprus University of Technology, and soon she will do her PhD in the field of Energy. Her special fields of interest are sustainability, sustainable and renewable energy technologies, sustainable building design, energy efficient buildings, and solar energy.