AN EXPERIMENTAL INVESTIGATION OF A NEW HEAT EXCHANGER FOR A SOLAR DOMESTIC HOT WATER SYSTEM (Part 1)

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Abstract
Renewable energy sources have the capacity to play a significant role in replacing conventional fuels in four fields, such as electric power production, hot water production, transportation of fuels, and countryside (off-grid) power services fuels in four fields. One of the most widely known in solar thermal applications is solar water heating system .
In the present paper a comparison of the thermal performance of two designs of heat exchanger of solar domestic hot water system was achieved. The comparison was carried out in terms of the heat exchanger overall heat transfer coefficient (UA) product, maximum tank temperature, thermal stratification. The tested heat exchangers were, an immersion heat exchanger (Immhx1) and an external shell-and-coil heat exchanger (S&Chx). The results showed that the heat exchanger of (S&Chx) achieved a higher value of overall heat transfer coefficient (UA), higher thermal stratification, and higher temperature in the upper layers of the tank compared with heat exchanger (Immhx1). Regarding the energy obtained from solar radiation, collected and transferred to the collecting tank, the performance of the heat exchanger (S&Chx) was better compared with heat exchanger (Immhx1)
Key words: Heat exchanger, solar water heating system, overall heat transfer coefficient

Nomenclature

G_T	Total solar irradiance on the collector (W/m²)
A_c	Collector area (m^2)
$\eta_{col\text{-}max}$	Maximum instantaneous collector efficiency (%)
F_R	Collector heat removal factor
$\tau\alpha$	Collector cover transmittance – absorptance product
U_L	Collector overall heat loss coefficient (W/m^2. C)
T_{coll}	Collector inlet temperature (^{o}C)
T_{air}	Ambient air temperature (^{o}C)
Q_{t1}	Energy stored in the tank at t_1 (W)
Q_{t2}	Energy stored in the tank at t_2 (W)
m_w	Mass of water in the storage tank (kg)
$C_{p,w}$	Specific heat for the water (J/kgK)
T_{avrT}	Storage tank average temperature (^{o}C)
$T_{i,\ tank}$	Initial temperatures of the storage tank (^{o}C)
Δt	Elapsed time (sec)
Q_{end}	heat stored at a particular time during the day (MJ)
Q_{start}	heat stored at the beginning of the day (MJ)

1. INTRODUCTION AND LITRATURE SURVEY

Solar domestic hot water (*SDHW*) systems perform three basic operations: collecting energy by a solar collector, transferring the energy to the water through a heat exchanger, and storing the energy in a storage tank for domestic use.

Collection process based on the "greenhouse effect", where sunlight is collected and converted to heat energy by a solar collector. The solar collector is mounted on or near the house, faced to the south. Sunlight is passing through glass to be absorbed by the collector flat plate. The plate converts the sunlight into heat, which is prevented from escaping by the glazing of the solar collector. Transferring the thermal energy is carried by circulating the fluid through the solar collector and then transferred the heat to the tank by using a heat exchanger. Storing process involves the storage of heated water in an insulated tank. Solar water heaters usually have a slightly larger hot water storage capacity than other water heaters. This is because solar heat is available only during the day and enough hot water must be collected to meet evening and early morning needs **[1]**.

Two types of heat exchangers can be used to transfer the heat from the hot fluid to the cold water. The first type is called an external heat exchanger, where the heat transfers between the hot fluid coming from the solar collector and the cold water occurs outside the storage tank.

The second type is an internal heat exchanger, where the heat transfers between the hot fluid and the cold water occurs inside the storage tank, which is called an immersion heat exchanger.

MacLeod and Allen carried an experimental work under a topic of Evaluation of components in Solar Water Heaters with Photovoltaic Powered Pumps. They stated that the collector flow rate and PV power were found to vary linearly with insolation (solar radiation level), and the flow was found to be laminar in the supply line and turbulent in the return line at full sun conditions. The study suggested two ways to increase the system flow rate using the same PV power. One was involving turning up the linear booster input voltage to maximize PV power output, and the other method involved reducing the hydraulic losses in the system by increasing the diameter in the return line**[2]**.

Sandnes, Rekstad designed and built a PV/T test collector using single-crystal silicon cells in combination with a solar heat absorber in polymer plastics. The system was tested experimentally to determine its thermal and photovoltaic performance, the efficiency of different collector configuration were compared with PV performance, and temperature readings were discussed. A comparison of PV/T absorber with pure thermal absorber showed a low thermal efficiency for the PV/T system. Significant cooling of the PV cells was achieved by low-temperature operation of the heat collector which resulted in improved PV efficiency. Good agreement was obtained between simulation and experimental results**[3]**

Jardany and Allen built and tested five configurations of immersion heat exchangers. Each one was tested for several weeks. The overall heat transfer coefficient (UA) was used to characterize and compare the performance of each heat exchanger in three different configurations of the SDHW system. The performance of each system configuration was analyzed in terms of heat loses, system efficiency, and tank thermal stratification. They concluded that the SDHW system with immersion heat exchanger of 0.170 m^2 in area was 34% more efficient than the identical system operated with side-arm heat exchanger of area 0.6 m^2. Moreover, they created five semi-empirical models to predict the UA values and to obtain information on the magnitude of the natural convection heat transfer coefficients for each of the immersion heat exchangers they tested**[4]**.

P.Ganesh Kumar et al., presented an innovative hybrid system that serves the dual purpose of heating air and water simultaneously. Based on the results of the experimental investigation, it was inferred that the collector efficiency is directly proportional to the volume percentage of the nanomaterial. The average temperature difference of 14.54°C was achieved in the solar collector, whereas a maximum temperature of 18.32°C was obtained for 0.2 volume percentage of MWCNT at a mass flow rate of 0.01 kg/s. Moreover, the maximum thermal efficiency of 51.03% was obtained for a 0.2 volume percentage SG/MWCNT **[5]**.

Jian Yao et al., suggested a residential heating system using Borehole Heat Exchanger (BHE) coupled with solar assisted PV/T heat pump with further performance analysis. The simulation results showed that a larger mass flow rate could increase the heat extract capacity of BHE's and also increases the flow resistance and pump power under nominal conditions. The circulating water could not extracted heat from rock-soil when the inlet temperature exceeds 48.5°C. Furthermore, the maximum water temperature from hybrid system could reach 40.8°C with solar fraction of 67.5% at PV/T area of 1000 m^2, and the solar irradiation is 600 W/m^2 and depth of the BHE is 2500m. In the meantime, the heating C.O.P. of the hybrid system could reach 7.4 and the system could operate independently without power input from electrical grid **[6]**.

T. Mohapatra, et al., presented a study concerned, the thermal performance of a three fluid heat exchanger (TFHE) used in solar flat plate collector system is studied. The TFHE was an improved version of double pipe heat exchanger, where a helical tube was inserted between two concentric straight tubes for better performance. The study presents a new technique of simultaneous air and water heating in TFHE using solar energy. They reported that the heating cost can be minimized considerably by supplying the hot water or thermal fluid from a solar flat plate collector through the helical tube of TFHE to heat incoming cold air and water in the inner most pipe and outer annulus. The TFHE was investigated experimentally and validated by comparing the result of experimental approach with literature. Decent agreements between the experimental and literature values were observed. The purpose of the study was to determine the effect of Reynolds number and Dean number on performance of the TFHE in steady-state for both flow configurations. The overall heat transfer coefficients, varies directly and effectiveness, varies inversely with hot water volume flow rate, however hot water flow rate have least or no effect on coil side Nusselt number. The effect of inner Dean Numbers on overall heat transfer coefficients, and effectiveness, is negligible in parallel flow configuration **[7]**.

2. THE EXPERIMENTAL WORK

The experimental set-up was designed and constructed, as shown in Figure 1. It can be seen that the heat fluid (Glycol mixture) flowing through the coil in order to heat the water entering the heat exchanger from the bottom to top. The water temperature was measured along the height of the heat exchanger as shown in the Figure. The experiment was initially run to ensure that there is no any human error in the installation process, such as leakage. Initial readings were taken as expected values. To simulate the daily actual hot water consumption, hot water was drawn from the top of the tank at early morning and late afternoon

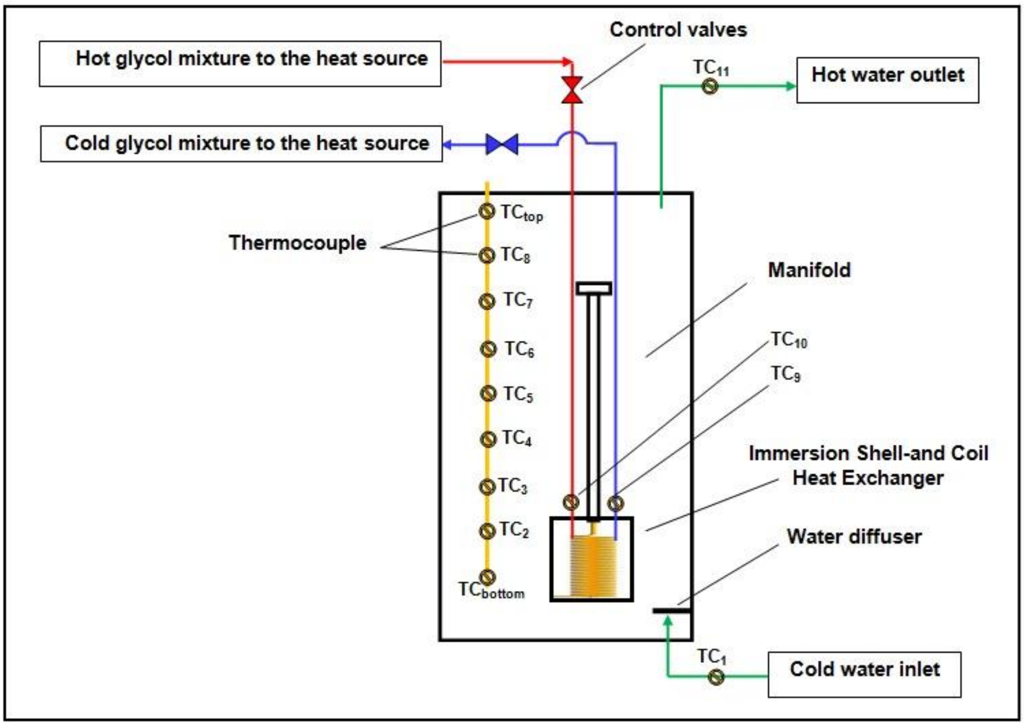

Fig. 1 Schematic diagram of *SDHW* system with one immersion shell-and-coil heat exchanger

An Immersion Heat Exchanger Test

The SDHW system used an immersion heat exchanger (*Immhx1*), as shown in Figure 2. The *Immhx1* consists of two finned bayonet immersion heat exchangers connected together at a 75° angle. It is a bayonet type to facilitate the insertion of the heat exchanger in the storage tank. The purpose of the angle 75° is to facilitate the rotation of the heat exchanger. The outer tube is a finned tube with a diameter of 19.05 mm OD, while the inner tube was made of copper with a diameter of 15.88 mm OD

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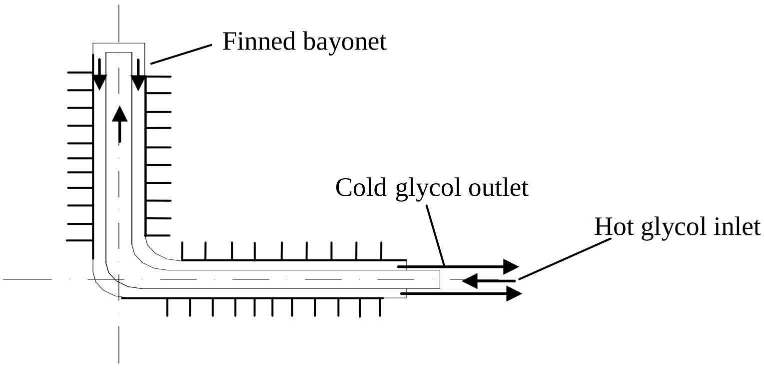

Fig. 2 schematic diagram of first immersion heat exchanger (*Immhx1*)

The glycol enters the tube of the heat exchanger, flows to the end of the inner tube, and then rotates with an angle of 180^o, and then flows through the annulus back to the heat exchanger exit. The annulus is sized to produce relatively high velocity in order to create low resistance of heat transfer inside the tube.

2.2 An external shell-and-coil heat exchanger (S&Chx) Test

The S&Chx is a counter flow type consists of four concentric helical copper coils enclosed in a copper shell. The entire heat exchanger was insulated with a foamed rubber of thickness 50 mm. The solar collector coolant was circulated using a DC pump, while the water circulated naturally by convection through the shell side. The shell-and-coil heat exchanger was connected with the storage tank from the bottom and the top creating natural circulated loop. Figure 3 shows a schematic diagram of shell-and-coil heat exchanger (S&Chx)

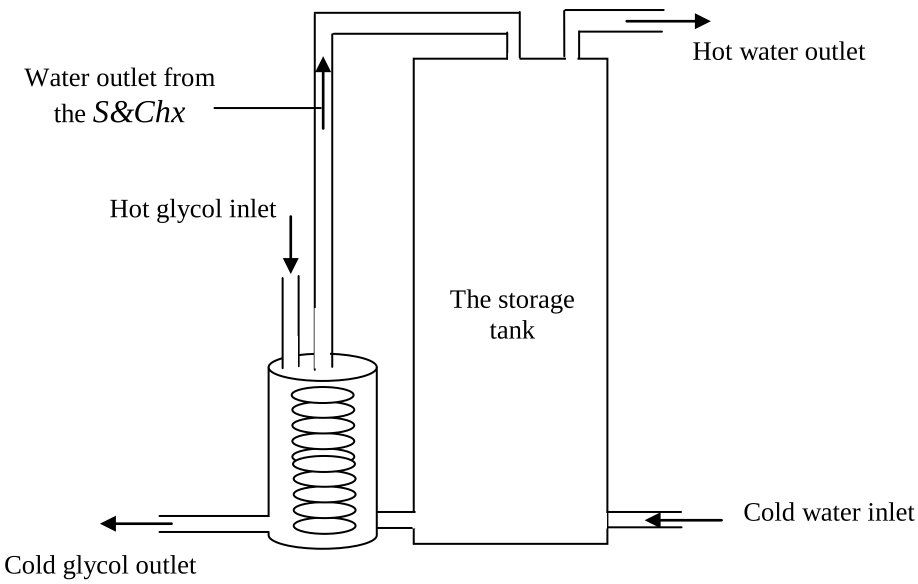

Figure 3 schematic diagram of shell-and-coil heat exchanger (*S&Chx*)

3. THE EXPERIMENTAL PROCEDURE

The Solar Domestic Hot Water system was installed and modified to work with two heat exchangers independently by using three valves to control the hot and cold glycol paths. The solar collector was placed on the rooftop of the laboratory building.

A computer data acquisition system was used to monitor and record the data to analyses the thermal performance system.

Computer was operated to control two solenoid valves. The first valve was installed in the water supply line to drain the warm water inside the building pipes. The second valve was used to drain the hot water from the top of the tank. This process at the beginning and the end of day to draw about 100 L from the storage tank each time, in order to simulate the actual hot water consumption on a daily basis.

A water diffuser was installed at the bottom of the storage tank to reduce the rate of the cold water velocity inlet to the storage tank, thereby; reducing the early mix between the cold and hot water in the tank.

The system is provided with a pressure relief valve to remove the air from the top of the tank, and at the same time acts as a safety valve to protect the system from overheating in case of hot sunny day.

The flow, temperature and pressure were recorded continuously at intervals not exceeding a few minutes by means of a computer.

3.1 System Analysis

The data acquisition system creates two data files for each day of the year; one for the system parameters, and one for the storage tank temperatures. These data are required for system performance analysis. The following equations are used to calculate system performance parameters.

- The glycol mass flow rate is calculated from the measured rate of rotation of the pump (RPM) by using the equation

$$GPM_R = \frac{GPH}{60} * \frac{RPM}{1725}$$

GPM_R : Gallons per minute based on revolution
GPH : 70 GPH (constant for this pump)
RPM : Revolutions per minute
1725 : Constant for the used pump (maximum RPM of the pump)

- The collector heat transfer rate is calculated by equation

$$q_{coll} = G_T A_C \eta_{coll-\max} \quad (1)$$

$$\eta_{col\text{-}max} = F_R(\tau\alpha) - F_R\ U_L \frac{(T_{coll} - T_{air})}{G_T} \quad (2)$$

- The rate of heat loss from the storage tank is calculated by equation:

$$q_{tank} = \frac{Q_{t2} - Q_{t1}}{t_2 - t_1} = \frac{m_w C_{p_w}(T_{avr\ T} - T_{i,tank})}{\Delta t} \quad (3)$$

1. The accumulated heat addition to the tank, Q_T, is calculated with respect to the energy contained in the storage at the beginning of the day, Therefore; the variations in the $T_{avr,T}$ from one day to another are factored into the computation of η_{sys}. Q_T is calculated by equation:

$$Q_T = Q_{end} - Q_{start} \quad (4)$$

Both Q_{end} and Q_{start} are computed using,

$$Q = m_w C_{p\,w}(T_T - T_{cw}) \quad (5)$$

4. RESULTS AND DISCUSSION

It was mentioned previously that, two models of heat exchangers (Immhx1) and (S&Chx) were tested separately. The two models will be discussed in detail as follows:

4.1 Thermal Analysis of the First and Second Tests:

Figures 4 and 5 show the solar flux change on May 22 using heat exchanger Immhx1 and on May 23 using heat exchanger S&Chx. The values of solar radiation, the sun position relative to the Earth, and the length of daylight of 22th and 23th May were almost identical. From Figure 4, it becomes clear that there is a fluctuation in the value of the solar flux during the period from 10:00 AM to 11:00 AM due to the cloud period, where in Figure 5 it can seen that no fluctuation was observed except early morning.

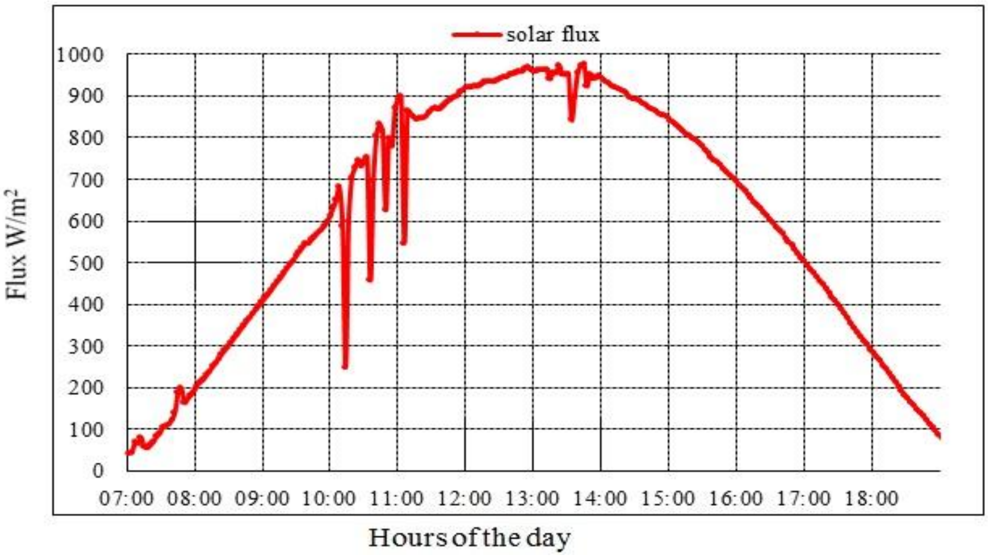

Fig. 4 Variation of solar flux during the day for heat exchanger *Immhx1* (22 May)

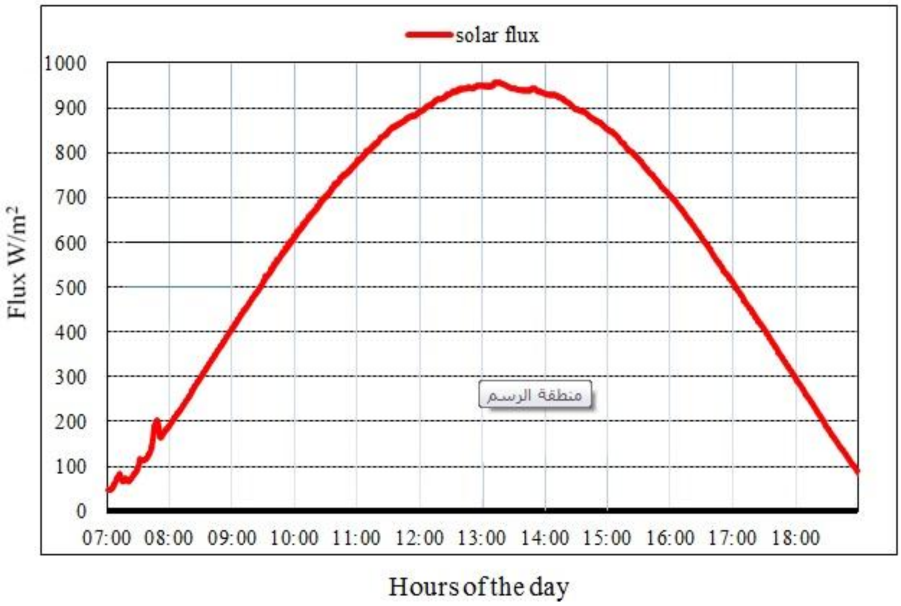

Fig. 5 Variation of solar flux during the day for heat exchanger *S&Chx* (23 May)

4.2 Overall Heat Transfer Coefficient (UA) and Heat Transfer Rate (qhx) for Immhx1

Figure 6 shows the change of overall heat transfer coefficient (UA values) with the heat transfer rate (qh_x) for Immhx1 on the morning and afternoon of May 22. In the afternoon, the UA values of the Immhx1 were always higher than those in the morning for the same heat transfer rate (qh_x). This is mainly occurred due to the fact that in the morning the water temperatures were well below the glycol temperatures as compared to the afternoon. This means that $(\Delta T_{lm})_{AM}$ is higher than $(\Delta T_{lm})_{PM}$, where $q_{AM} = (UA)_{AM} (\Delta T_{lm})_{AM}$.

Figure 7 shows the change of overall heat transfer coefficient versus the heat transfer rate of the S&Chx on May 23. The UA values for qh_x for the S&Chx were much higher in the afternoon than in the morning also due to the same reason mentioned above. The peak UA value on the same day was 95 W/K at qh_x=1490 W, which is 62% higher than the Immhx1UA peak.

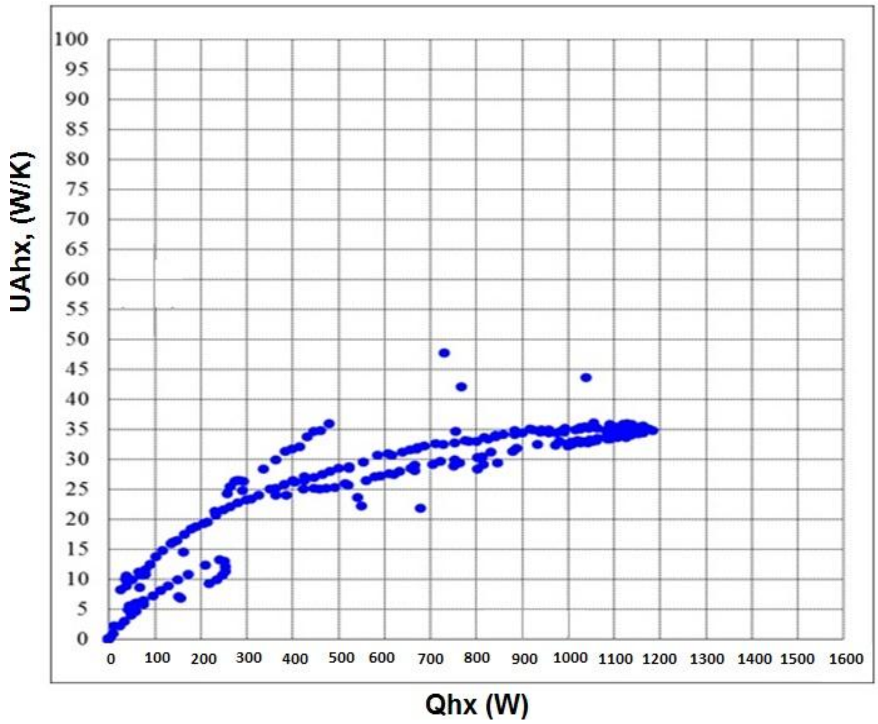

Fig. 6 Variation of overall heat transfer coefficient (UAhx) with heat transfer rate (qhx) of the *Immhx1*

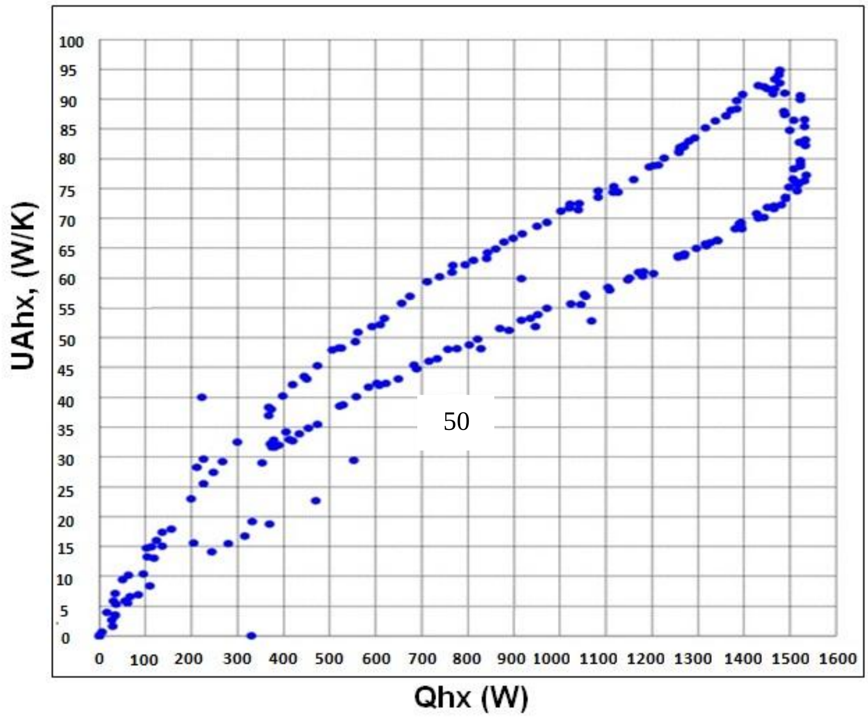

Fig. 7 Variation of overall heat transfer coefficient (UAhx) with heat transfer rate (qhx) of the *S&Chx*

4.3 Storage Tank Temperature Distribution for immersion heat exchanger

Figure 8 shows the tank stratification on May 22, using Immhx1 heat exchanger. During the day, the tank temperatures at the top and the bottom were changed from 43^{o}C and 13^{o}C to 41^{o}C and 15^{o}C respectively.

This indicates that a large energy was extracted during the day. The decreasing of tank top temperature was attributed to the losses through the insulation and heating the water at the bottom of the tank by conduction. About 100 L of water was drawn in the morning, in order to reduce the average storage tank temperature. The temperature difference between the top and the bottom after drawing the water was 24ºC and the main reason for this is cold and warm temperature of glycol at the top and bottom respectively.

At noon, only two thermal zones can be distinguished; one at the level of the Immhx1 and the other in the portion of the tank above the Immhx1. In essence, there is no thermal stratification in the tank, due to the nature of heat delivery from Immhx1

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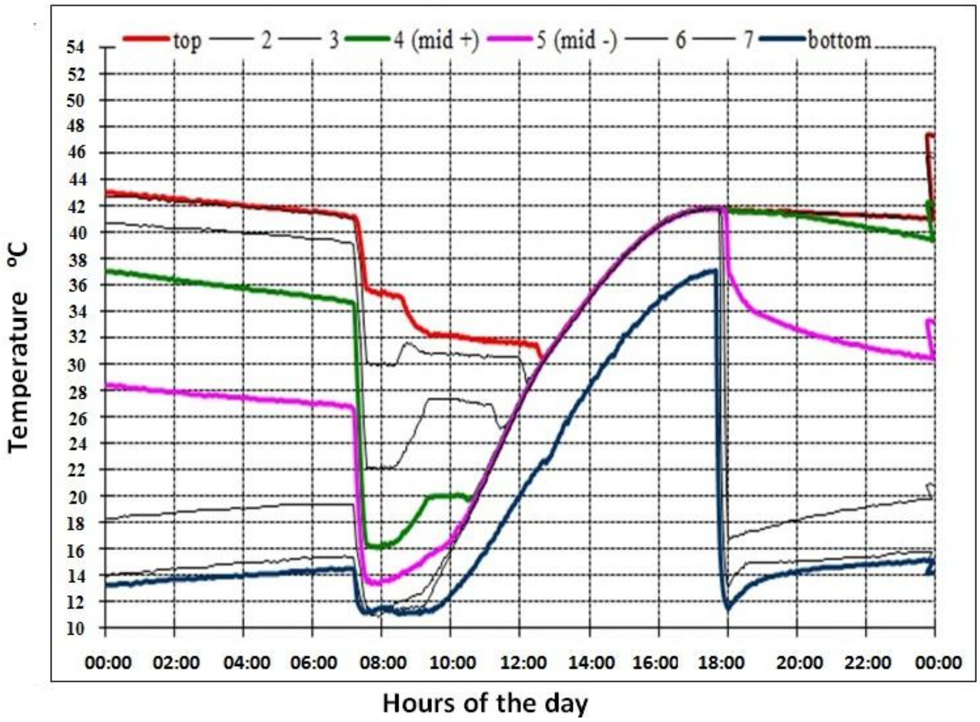

Fig.8 The temperature distribution in the storage tank for the *Immhx1*

4.4 Storage Tank Temperature Distribution for the S&Chx

Figure 9 shows the temperature profile of the tank on May 23. By comparing these results with that obtained in Figure 8 it can be seen that Figure 9 clearly shows a high degree of 8 thermal stratification in the tank (due to the delivery of hot water directely to the top of the tank). This stratification can not be seen in the second half of May 22 in the t case of (Immhx1).

The maximum temperature at the top of the tank of (S&Chx) was (T_{top} = 51°C), while in (Immhx1) condition was (T_{top} = 42°C). The results show that the (S&Chx) achieved better stratification and better solar system efficiency compared with (Immhx1)

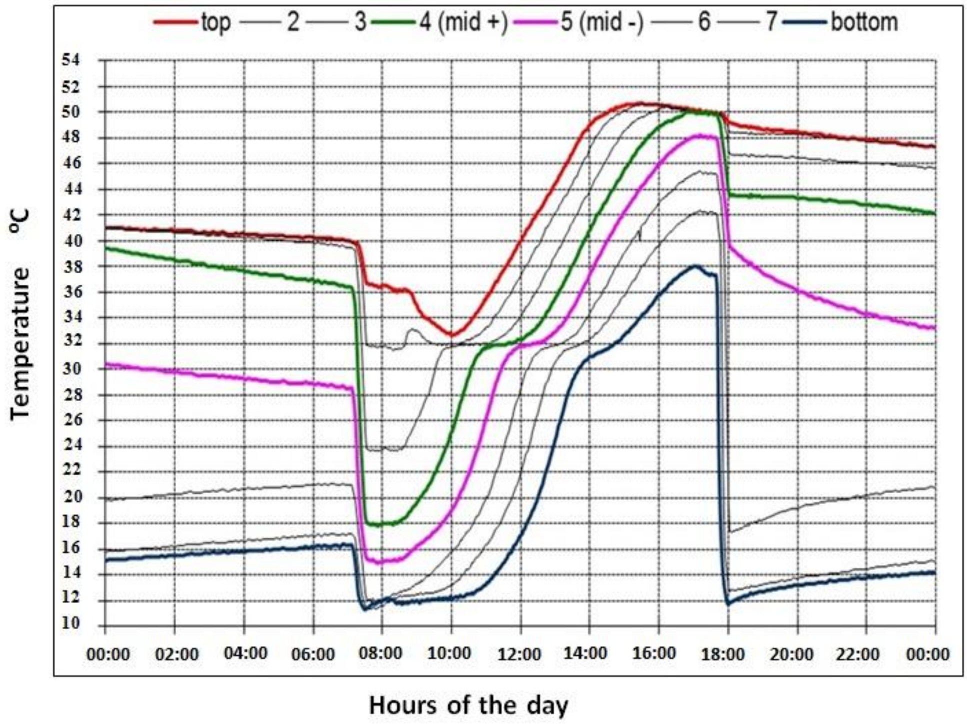

Fig. 9 The temperature distribution in the storage tank for the S&CHx

4.5 Thermal performance of the SDHW system for both heat exchangers Immhx1 and S&Chx

Figures 10 and 11 show the thermal performance of the SDHW system for both heat exchangers Immhx1 and S&Chx respectively. Based on Figure 10 the heat collected by (qcol) was 1400W, while the heat transferred to the storage tank through Immhx1 was 1175W. This means that 15% of the collected heat was lost. These high losses are attributed to the reduction of the heat transfer area of Immhx1. In Figure 11 heat collected by (qcol) was 1680W, while the heat transferred to the storage tank through S&Chx was 1530W. This means that 8.9% of the collected heat was lost. Unstable of heat losses was noticed in S&Chx heat exchanger and this mainly was occurred due to several varibles such as the variation of the average tank temperature, and the glycol flow rate, which is the most important variable to calculate the rate of heat transfer.

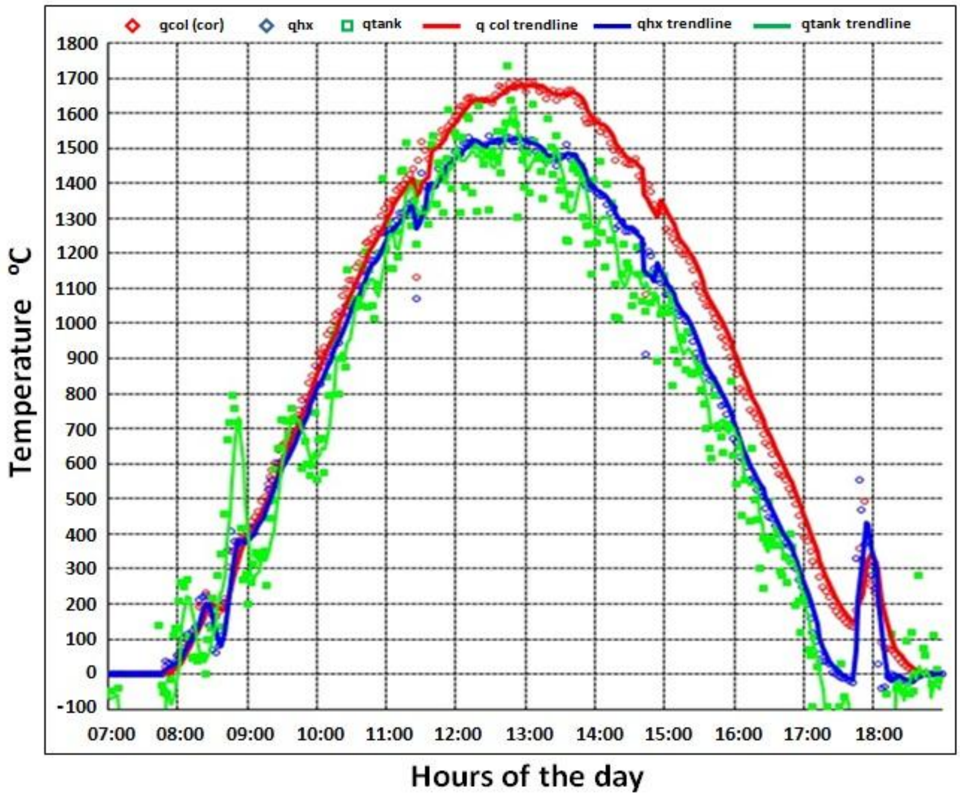

Fig. 10 variation of heat with time using *Immhx1* on May 22

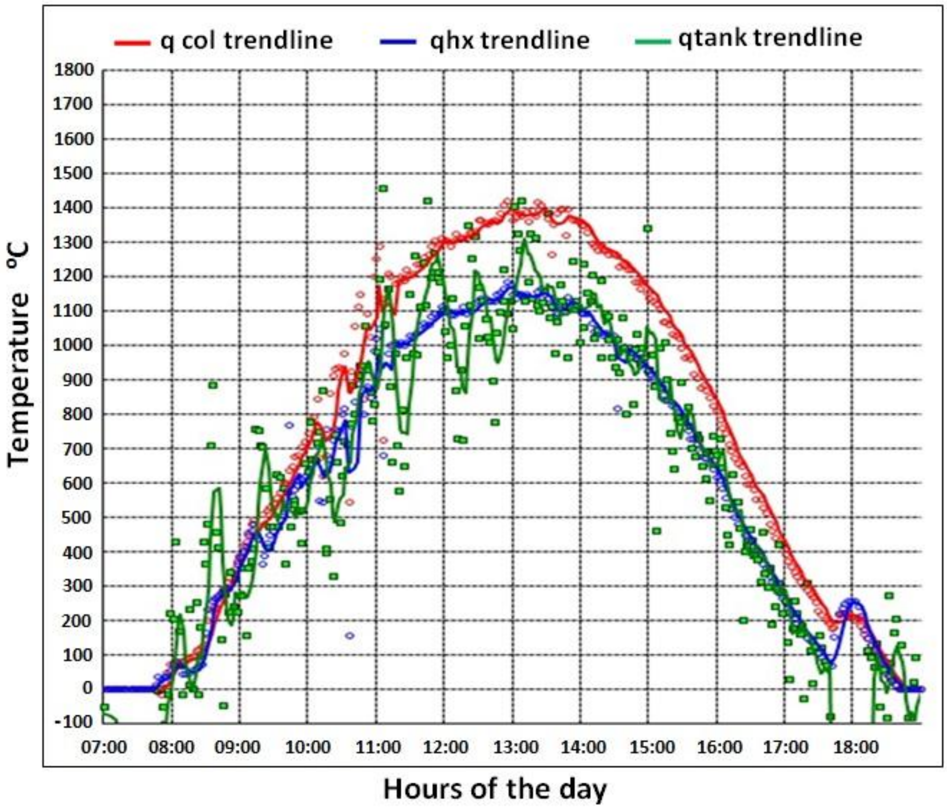

Fig. 11 variation of heat with time using *S&Chx* on May 23

5. CONCLUSIONS

Based on the obtained results and the discussion the following point were conclude and summarized as follow.

1. The external shell-and-coil heat exchanger (*S&Chx*) achieved higher overall heat transfer coefficient (UA) and higher temperature at the top of the tank compared with immersion heat exchanger *(Immhx1)*.
2. The external shell-and-coil heat exchanger (*S&Chx*) achieved higher thermal stratification compared with immersion heat exchanger *(Immhx1).*
3. The external shell-and-coil heat exchanger (*S&Chx*) transferred higher value of heat to the storage tank.
4. The *Immhx1* provided a good challenge during the installation processes due to its length accompanied with the 75° elbow.
5. Using an external heat exchanger (*S&Chx*) requires more pipe connections that must be thermally insulated in addition to the more allocated space and, therefore, increase the costs.

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