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Energy and exergy efficiency of a daily heat storage unit for buildings heating

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Abstract *- This paper deals with a daily solar storage system conceived and built in Laboratoire de Maîtrise des Technologies de l'Energie (LMTE, Borj Cedria). This system consists mainly of (i) a heat storage unit which consists of a wooden case with dimension of 5 m^3 (5 m x 1 m x 1 m) filed with fin sand, (ii) a heat collection unit which consists of 5 m^2 of south-facing solar collector mounted at a 37° tilt angle and (iii) a network of a polypropylene capillary heat exchanger with an aperture area equal to 5 m² buried inside the heat storage unit and connected to the heat collection unit. The circulation of the heated water between the heat collection unit and the polypropylene capillary heat exchanger allows heating the sand inside the heat storage unit. In order to evaluate the system efficiency during the charging period (during the day) and discharging period (during the night) an energy and exergy investigation were applied. Results showed that during the charging period, the average daily rates of thermal energy and exergy stored in the heat storage unit were 400 and 2.8 kW, respectively. It was found that the net energy and exergy efficiencies in the charging period were 32 % and 22 %, respectively. During the discharging period, the average daily rates of the thermal energy and exergy recovered from the heat storage unit were 2 kW and 2.5 kW, respectively. The recovered heat was compared to the total heat requirements of a tested room (4 m x 3 m x 3 m). The results showed that 30 % of the total heating requirement of the tested room can be achieved from the heat storage system during the whole night until in cold seasons.*

Résumé *- Au sein du Laboratoire Maîtrise des Technologies de l'Energie (LMTE, Borj Cedria), nous avons conçu et construit un système de stockage de l'énergie thermique d'origine solaire. Ce système de stockage est constitué par (i) une cuve en bois de capacité égale à 5 m^3 (5 m x 1 m x 1 m), (ii) un capteur solaire, de 5 m^2 de surface de captation orienté vers le sud et monté à un angle de 37° par rapport à l'horizontal et (iii) un échangeur de chaleur sous forme capillaire fabriqué en polypropylène enterré à 40 cm de profondeur à l'intérieur de la cuve et connecté au capteur solaire. La circulation de l'eau chaude entre le capteur solaire et l'échangeur enterré permet de chauffer le sable contenu dans la cuve et de stocker ainsi de l'énergie thermique sous forme de chaleur sensible. Pour évaluer les performances thermiques du système de stockage de l'énergie solaire pendant la phase du stockage (pendant le jour) et pendant la phase de déstockage (pendant la nuit), une étude énergétique et exergétique a été effectuée. Les résultats ont montré que pendant la période du stockage, le taux journalier moyen d'énergie thermique et exergétique stockée dans le système de stockage varient de 400 à 2.8 kW, respectivement. Les résultats ont aussi montré que, pendant la phase de stockage, l'efficacité énergétique et exergétique sont respectivement égales à 32 % et 22 %. D'autre part, pendant la phase de déstockage, les taux journaliers moyens de l'énergie et de l'exergie récupérée de l'unité de stockage sont respectivement 2 kW et 2.5 kW. L'énergie thermique récupérée de l'unité du stockage a été comparée à l'énergie thermique nécessaire au chauffage de l'air dans une salle (4 m x 3 m x 3 m). Les résultats ont*

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montré que le système de stockage permet de couvrir près de 30 % des besoins thermiques en chauffage de la salle pendant la nuit.

Keywords: Stockage thermique - Performance - Etude énergétique - Etude exergétique - Unité de stockage - Déstockage - Bâtiments.

1. INTRODUCTION

Currently wide efforts have been undertaken to alleviate global warming of earth caused by the emission of carbon dioxide in atmosphere generated by intensive burning of fossil fuels. Due to these problems, tendency towards renewable energy resources, particularly solar energy which is an abundant, clean and safe source, offers wide range of exceptional benefits especially for household air-conditioning. However, its intermittent character constitutes the main impediment of solar energy use. To reduce the time or rate mismatch between energy supply and energy demand it is necessary to store the excess of solar heat for short-or long-term storage. The short-or long-term storage of the sensitive heat in soil seems to be plausible from a technical and economic view point. Indeed, different techniques of storage of the sensitive heat in soil have been used [1-3]. The performances of these techniques are influenced by the transfer of the heat, of the humidity in soil and the thermal efficiency of the heat exchanger used for storing solar heat in soil. These systems consist mainly of two 10 cm-diameter PVC tubular heat exchanger connected. The first heat exchanger is placed inside the greenhouse. The second heat exchanger is buried at 30 cm-depth within the underground of the same greenhouse. The heat collected from the first heat exchanger is transferred to the second heat exchanger buried in the underground. The results showed that the thermal transfer in the ground due to the pressure gradients of humidity in soil is insignificant when compared to the heat transfer due to the temperature of gradients. Results showed that the storage of heat by through the buried air/soil heat exchanger was an interesting method that can be used for the daily solar heat storage. In fact, the thermal efficiency of the solar heat storage system was evaluated at 50 %.

In this context, we have built in our laboratory (LMTE, Borj Cedria), a prototype for sensible solar energy storage. The experimental system consists mainly of (i) a solar storage unit which consists of a wooden casing whit dimension of 5 m^3 filed with fin sand. Inside the solar storage unit was buried a network of a polypropylene capillary heat exchanger with an aperture area equal to 5 m^2 and (ii) a solar collector unit with an aperture area of 5 m^2 connected to a buried heat exchanger. An experimental study was carried out in order to evaluate the thermal performances of the storage system. Performances of the heat storage system was evaluated by a theoretical analyzes based on first and second law of thermodynamics.

From a first law perspective, the efficiency of a thermal energy storage system was assessed in terms of how much thermal energy the system can store. This approach produces workable designs, but not necessarily those with the highest possible thermodynamic efficiencies. It has been shown in recent years that the design of thermodynamically efficient heat transfer equipment must be based on the second law of thermodynamics in addition to the first law [4]. Exergy analysis, derived from both the first and second laws of thermodynamics, provide greater insight to cost effective design and management of complex processes [5]. Unfortunately, there are only a few publications related to exergetic efficiency of the large-scale heat storage applications. The endeavour of this paper is to appraise the performances of this low cost storage system studied under various Tunisian climatic conditions using energy and exergy analyzes. The energy and the exergy analyzes established in this paper allowed the estimation of the maximal value of heat stored inside the heat storage unit.

2. EXPERIMENTAL INVESTIGATION

2.1 Experimental device

Figure 1 illustrates a schematic diagram of the solar heat stored system. The system consists of three units: (i) heat collection unit, (ii) heat storage unit and (iii) data acquisition unit.

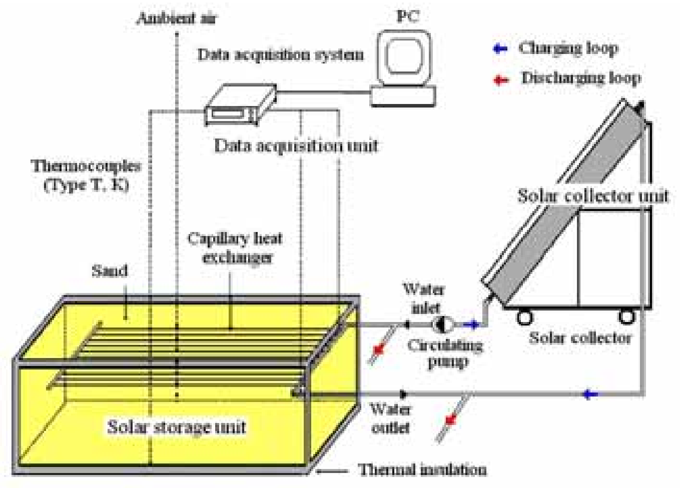

Fig. 1: Solar heat storage system

Fig. 2: Heat collector unit

2.1.1 Heat collection unit

The heat collection unit is composed of a solar collector with an aperture area of 5 m^2 mounted at 37° towards the south. The absorber of the solar collector was made with

40-cm thick concrete slab painted black. The concrete matrix was used to increase the storage capacity of the heat collection unit. A capillary polypropylene heat exchanger (Fig. 2) with an aperture area equal to 5 m^2 was being integrated inside the absorber matrix to carry the heat transfer fluid inside the solar storage collector. The thermophysical and geometrical proprieties of the absorber and the heat exchanger were given in **table 1** and **table 2** respectively.

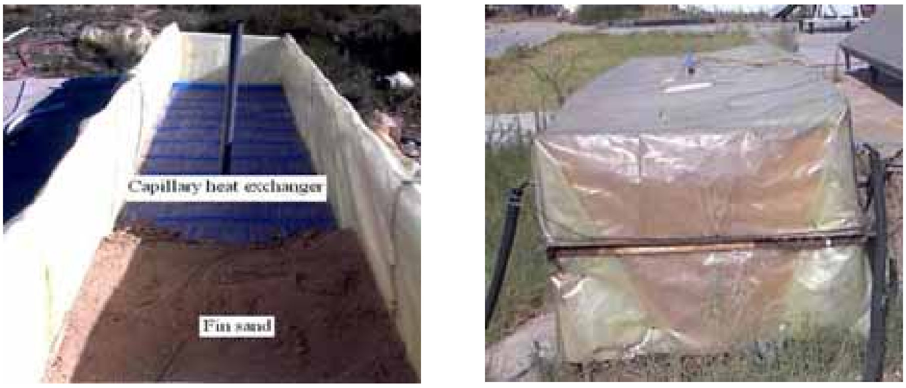

Fig. 3: Heat storage unit

Table 1: Physical and geometrical proprieties of the heat collection unit

Parameters	Value
Dimensions	1m x 5 m
Collector price	90 ($/1m)
Absorber material	Concrete
Concrete mass	200 kg
Concrete density	2800 kg/m^3
Concrete thermal conductivity	0.81 W/m.K
Concrete specific heat	880 J/kg.K
Toxic effect	No

Table 2: Physical and geometrical proprieties of the capillary heat exchanger

Parameters	Value
Dimensions	1m x 5 m
Inner capillary	3 mm
Outer diameter	4.2 mm
Heat exchanger price	15 ($/1m)
Heat exchanger material	Polypropylène
Polypropylene density	950 kg/m^3
Polypropylene thermal Conductivity	0.22 W/m.K
Toxic effect	No

2.1.2 Heat storage unit

A wooden case of a 5m length, 1m width and 1 m depth, field with fin sand, was used as a heat storage unit (Fig. 3). The sand characterized by a high storage capacity per unit volume was used as a heat storage material. The whole surface of the heat storage unit was insulated with 30-mm of polyurethane foam. A network of a polypropylene capillary heat exchanger with 5 m^2 aperture area is buried within the sand inside the case at 40 cm-depth. In fact, preliminary experiment was carried out on the storage unit in order to determine the optimal depth in which the heat exchanger will be buried. The test consisted of following the variation of the sand temperature at different depths vs time.

By analyzing the results we noticed that the superior layers of sand undergoes easily with the external climatic variation. Beyond a certain depth, the sand temperature does not vary between the day and the night. Thus sand inside the storage unit can be divided into two zones (upper and lower).

The thickness of the upper zone is about 20 cm. In this zone, the sand temperature fluctuates seriously with external climatic conditions. Under this depth the sand temperature dose not oscillates with external climatic conditions. Thus, less than 20 cm-depth is considered to be a long term thermal storage section.

2.1.3 Data acquisition unit

Measurements made during the experimental studies are divided into to groups:

- The first group measurements are related with the heat collection unit. Both the collector outlet temperature T_0 and collector inlet temperature T_i have been measured by T-type thermocouples.

- Second group measurements are concerned with the sand in the wooden case and the temperature value of the surrounding ground. Thus, copper-constantan T-type thermocouples were placed at different depths inside the sand at 10, 20, 30, 40, 50, 60, 70, 80, 90 and 100 cm depth on the central line.

The ambient air temperature was measured by another thermocouple placed out of the system. A pyrometer has been used to measure the global radiation coming towards the horizontal surface. The data measured were recorded every 15 mm with a total accuracy assumed to be about 5% by a HP data logger and then transferred to a PC for further evaluation.

2.2 Experimental protocol

An experimental investigation were conduced in Borj Cedria (North of Tunisia): Latitude 36° 50’ N, Longitude 10° 44’ E. The experimental study was elaborated in two complementary stages:

(i) First stage (a storage phase)

The experiments were carried out at the same time periods between 9.00 and 17.00 of the days for a mass flow rate maintained equal to 0.0416 kg/s by using the sliding valve integrated at the experimental loop. During this phase the heated water (at the temperature rang of 40-50°C) supplied by the heat collection unit, is conduced thought the polypropylene heat exchanger buried at 40 cm-depth inside the sand. The hot water inside the heat exchanger dissipates its thermal energy to the sand by conduction.

(ii) Second stage (a restoration period)

The heat stored in the accumulator during the charging period was recovered during the nights (20:00–03:00 h). During the discharging phase cold water, at a temperature and masse flow equal to 20°C and to 0.0416 kg/s respectively, replenish thought the heat exchanger buried in the sand. The hot sand inside the case dissipates the stored solar energy to the buried heat exchanger.

3. NUMERICAL STUDY

3.1 Solar collector unit

The energy gain (daily heat output) of the solar collector unit, Q_u (W), can be represented by the following empirical equation [6]:

$$Q_u = \alpha_1 H + \alpha_2 (T_{a,av} - T_{e,av}) + \alpha_3 \quad (1)$$

α_1, α_2 and α_3 are constants for a system, determined from the test results (**Table 1**).

The overnight heat loss coefficient, U_c (W/K.m²), of the solar collector unit is determined by measuring the temperature loss of the water during a 12 h nocturnal period [7]. The formula used is:

$$U_c = \frac{M_c\, C_{p,c}}{A_c\ .\ \Delta t}\ .\ \ln \frac{T_1 - T_{a,av}}{T_F - T_{abs,av}} \quad (2)$$

The daily thermal efficiency of the solar collector unit, $\overline{\eta}_j$, is given by the expression:

$$\overline{\eta}_j = \frac{\int_{t1}^{t2} Q_u(t)\, dt}{\int_{t1}^{t2} A_c . H(t)\, dt} \quad (3)$$

3.2 Heat storage unit

3.2.1 Energy analysis for charging and discharging period

The rate of heat transfer $Q_{Charging}(t)$ from the solar collector unit into the heat storage accumulator was calculated during the charging period by using the following equation:

$$Q_{Charging}(t) = \dot{m}_w\, C_p\, (T_i(t) - T_0(t)) \quad (4)$$

The rate of heat recovered from the heat accumulator during the discharging period. $Q_{Discharging}(t)$ was calculated by:

$$Q_{Discharging}(t) = \dot{m}_w\, C_p\, (T_0(t) - T_i(t)) \quad (5)$$

The rate of heat stored in the heat storage unit $Q_{Storage}(t)$ was determined with respect to the heat transfer rate into the storage unit and heat losses for the charging period.

$$Q_{Storage}(t) = Q_{Charging}(t) - Q_{Losses}(t) \quad (6)$$

$Q_{Losses}(t)$ represents the rate of heat loosed from the whole surface area of the heat storage unit. It is given by the relation:

$$Q_{Losses}(t) = \lambda_s . A_s . (T_m(t) - T_s(t)) \quad (7)$$

The evaluation of the sand temperature $T_s(t)$ during the charging and the discharging periods, is obtained by the resolution of the equation of the heat conduction [8]:

$$\rho.C.\frac{\partial T_s}{\partial t} = \frac{\partial}{\partial x}\left(\lambda_s \frac{\partial T_s}{\partial x}\right) + \frac{\partial}{\partial y}\left(\lambda_s \frac{\partial T_s}{\partial xy}\right) \tag{8}$$

To resolve the equation (8), the following simplifying assumptions were taking into account; (i) The sand was regarded as being homogeneous and (ii) The convective coefficient, h, was supposed to be constant throughout the heat exchanger embedded inside the sand. The integration of the heat conduction equation (between t at $t+\Delta t$) gives the equation:

$$a_P T_P = a_E T_E + a_W T_W + a_N T_N + a_S T_S + a_W^0 T_W^0 \tag{9}$$

Where:

$$a_E = \frac{\lambda_s)_e.\Delta y}{\partial x_e},\; a_W = \frac{\lambda_s)_W.\Delta y}{\partial x_W},\; a_N = \frac{\lambda_s)_n.\Delta y}{\partial x_n},\; a_S = \frac{\lambda_s)_s.\Delta y}{\partial x_s},$$

$$a_P^0 = \frac{\rho C \Delta x.\Delta y}{\partial t} \text{ and } a_P = a_E + a_W + a_N + a_S + a_P^0$$

The corresponding boundary conditions are:

$$-\lambda_s \frac{\partial T_S}{\partial x}\Big|_{x=0} = 0,\; -\lambda_s \frac{\partial T_S}{\partial x}\Big|_{x=L} = 0,\; -\lambda_s \frac{\partial T_S}{\partial x}\Big|_{y=0} = 0,$$

$-\lambda_s \frac{\partial T_S}{\partial x}\Big|_{x=k} = h(T_{water} - T_m)$, k is the position of the heat exchanger within sand and h represents the heat transfer coefficient between the sand particles and the water in the heat exchanger. The form of the correlation used for the evaluation of the heat transfer coefficient h was developed by [9]:

$$Nu = 0.023 . Pr^{0.33} . Re^{0.8} . \left[1 + \left(\frac{d}{L}\right)^{0.7}\right]$$

The energy efficiency for the charging period was defined as the ratio of the heat stored in the heat storage unit to the heat transfer from the solar heat collector. Then, the total energy efficiency during the charging period $\eta_{Charging}(t)$ in % was formulated as follows:

$$\eta_{Charging}(t) = \frac{Q_{Storage}(t)}{Q_{Charging}(t)}.100 \tag{10}$$

3.2.2 Exergy analysis for the charging and discharging periods

The flow rate of exergy transferred from the solar collector unit to the storage unit crossing the capillary heat exchanger was calculated by:

$$\dot{X}_c(t) = \dot{m}_w \left[(h_0 - h_i) - T_{ex}.(s_0 - s_i)\right]$$

where $h_0 - h_i = C_p.(T_i(t) - T_0(t))$ is the variation of the specific enthalpy, $h_0 - h_i = C_p.Ln\frac{T_i(t)}{T_0(t)}$ is the variation of the specific entropy and T_{ex} is the reference outside temperature.

The rate of thermal exergy transfer from the heat collection unit into the heat storage unit, $\dot{X}_c(T)$, was calculated during the charging period from the following equation (10).

$$\dot{X}_c(t) = Q_{Charging}(t) - T_{ex}\,\dot{m}_w\,C_p.Ln\frac{T_i(t)}{T_0(t)} \tag{11}$$

The rate of the thermal exergy stored in the heat storage unit $\dot{X}_s(t)$ was determined in relation to the exergy transfer rate into the heat storage unit and the exergy loosed from the heat storage unit for the charging period:

$$\dot{X}_s(t) = \dot{X}_c(t) - \dot{X}_l(t) \tag{12}$$

where $\dot{X}_l(t)$ is the rate of thermal exergy loosed from the heat storage unit during the charging period. $\dot{X}_l(t)$ was evaluated as follows:

$$\dot{X}_l(t) = Q_{Charging}(t).\left[1 - \frac{T_s(t)}{T_m(t)}\right] \tag{13}$$

During the charging, the exergy efficiency was determined by:

$$\Psi_c(t) = \frac{\dot{X}_s(t)}{\dot{X}_l(t)}.100) \tag{14}$$

The rate of thermal exergy, $\dot{X}_R(t)$, recovered from the heat storage unit during the discharging period was evaluated by:

$$\dot{X}_R(t) = Q_{Discharging}(t) - T_{ex}.\dot{m}_w\,C_p.Ln\frac{T_0(t)}{T_i(t)} \tag{15}$$

3.3 Heat requirement of the tested room

The total heating load can be calculated by considering into account the effect of various parameters according [11, 12]:

$$Q_{Discharging} = Q_{Tansmission} + Q_{Solar} + Q_{Internal} + Q_{Airflow} \tag{16}$$

$Q_{Transmission}$ represents the heat transfer from exterior walls, windows, door and envelopes [13]

$$Q_{Transmission} = h_i \sum_{j=1}^{Nw} A_{wj}(T_{wi} - T_a)_j + A_{gj}(T_g - T_a)_j + A_{dj}(T_d - T_a)_j + h_{iR} \sum_{j=1}^{NR} A_{Rj}(T_{Ri} - T_a)_j \quad (17)$$

Q_{Solar} represents the heat due to sun radiation transmitted from windows. Thermal heat due to this phenomenon is calculated by using:

$$Q_{Solar} = \tau_G H + \tau_D D \quad (18)$$

$Q_{Internal}$ represents internal heat generated by lighting system, persons in the building and home appliances can be calculated by using [14]:

$$Q_{Internal} = Q_{Linghting} + Q_{Persons} + Q_{Appliances} \quad (19)$$

Heat generated by lighting system was calculated by:

$$Q_{Linghting} = P \times F \quad (20)$$

Heat generated by persons and appliances were achieved from appropriate tables.

$Q_{Airflow}$ is the heat due to airflow into the building (sensible and latent heat). Heat load due to outside air flow infiltration can be calculated by [15]:

$$Q_{Airflow} = Q_S + Q_L \quad (21)$$

Where

-Sensible heat by entering air

$$Q_S = \dot{V}.\rho.C_{p,a}.(T_{a,0} - T_{a,i}) = \dot{m}_a.C_{p,a}.(T_{a,0} - T_{a,i})$$

-Latent heat by entering air

$$Q_L = \dot{V}.\rho.(W_i - W_0).\lambda$$

4. RESULTS AND DISCUSSION

The efficiency of the heat storage system depends on thermal and physical properties of the heat storage material (*Sand*), heat storage temperature and performances of the heat exchanger embedded inside the sand. In view of evaluating efficiency of the heat storage system, an energetic and an exergetic analyses were evaluated for the following charging/discharging process.

4.1 Charging process

During experimental tests water inlet mass flow rates and temperature were 0.0416 $kg.s^{-1}$ and 20 °C respectively. The solar radiation and the solar collector unit outlet water temperature increase during a clear sky day were represented in figure 4. As expected, the outlet water temperature of the solar collector unit depends on solar radiation. It increases to a maximum value of 50 °C at 14:00 pm in the noon and remains almost constant during 3 h before it starts to decrease later in the afternoon.

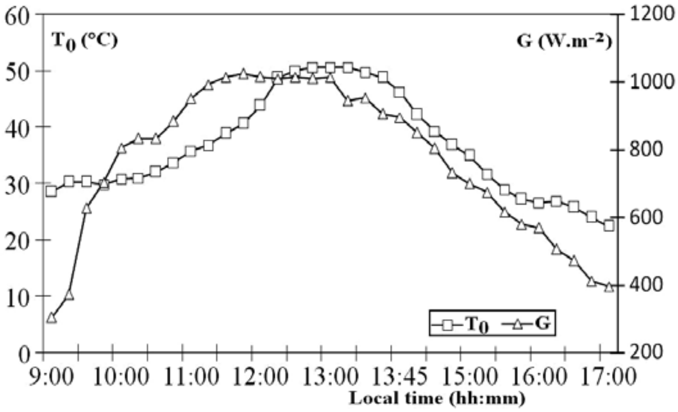

Fig. 4: Outlet water temperature increase us a function of day times for mass flow rates and average ambient temperature equal to 0.0416 kg/s and 25 °C

A simple and suitable way of presenting the results is given through the (Input/Output) method which offers an alternative to predict the solar collector unit long term performances. α_1, α_2 and α_3 {Eq. (1)} were determined from the test results by using the least square method and using a MATLAB program. We established also that the solar collector unit enjoy a daily efficiency ($\overline{\eta}_j$) of 32 % achieved at an average solar radiation level of 750 W.m^{-2} and an ambient temperature of 25 °C. The overnight heat loss coefficient, U_c, determined by using {Eq. (2)}, is assumed to be equal to 14 W.°C^{-1}. The solar collector parameters were determined during an experimentally investigation. Results are shown in **Table 3**.

Table 3: Collector parametric data

Parameters	Value
Collector overall energy loss coefficient U_c (W/m^2.K)	12
Optical yield, η_0	0.86
α_1 (m^{-2})	30
α_2 (W/K)	0.35
α_3	-12.7
$\overline{\eta}_j$ (%)	32

The water supplied by the solar collector unit was circulated within the buried heat exchanger. In figure 5 is represented the numerical and experimental variation of the sand temperature at 50 cm-depth during the storage phase (between 9:00 and 17:00). It was found that the temperature of the sand, T_S, inside the case increased gradually with the increased of water temperature supplied by the solar collector (T_0). At the end of the charging process the sand temperature, at 50 cm-depth, reached a maximum value unit was not affected by the decrease of the temperature of the water supplied by the solar collector unit. Figure 5 shows a good agreement between numerical and experimental values of the sand temperature.

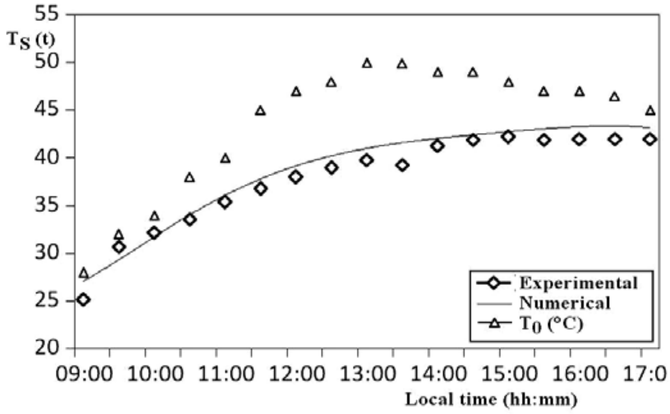

Fig. 5: Evolution of the sand temperature at 50 cm-depth during the charging phase

During the charging phase the rates of heat and thermal exergy stored in the heat storage unit were calculated by using equations (6) and (12), respectively. The results of energy and exergy analyzes during the charging period are shown in Figure 6. We noted that the rate of the heat stored in the heat storage unit ranged from 400 W to 2.8 kW, whereas the rate of the thermal exergy stored in the heat storage unit was in the range of 380 W and 800 W.

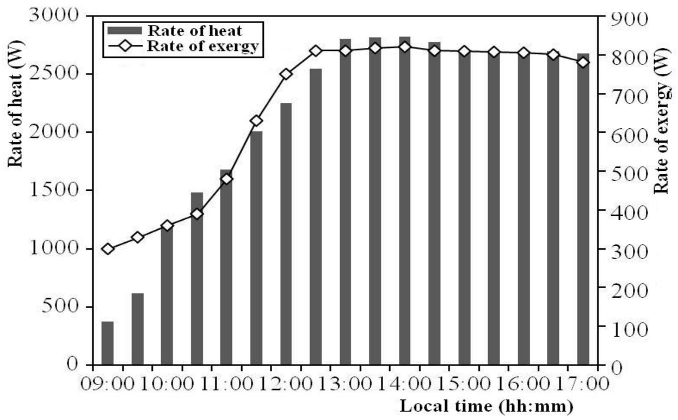

Fig. 6: Rates of heat and thermal exergy stored in the heat storage unit during the charging period

During this phase the average daily rates of the heat and thermal exergy stored in the heat storage unit were 515 W and 1.5 kW, respectively. We noted that the thermal energy stored in the heat storage unit was higher compared with the thermal exergy. The maximum values of energy and exergy rates (2.6 kW and 830 W respectively) were obtained at 14:00 when the sand temperature is in its maximum value.

The energy and exergy efficiencies of the heat storage system during the charging period were calculated by using equations (10) and (14), respectively. The changes of the average hourly energy and exergy efficiencies were showed as a function of time in Figure 7.

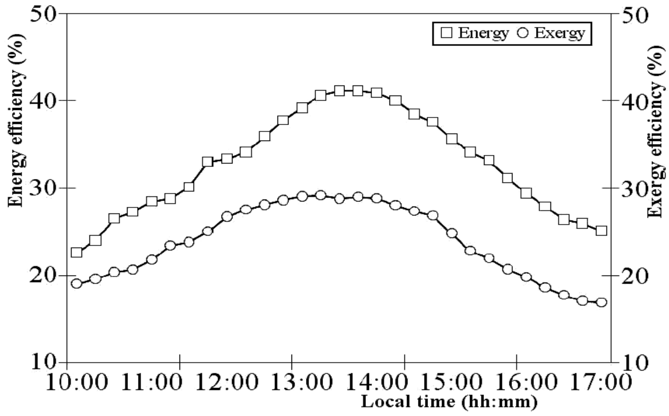

Fig. 7: Change of the net energy and exergy efficiencies for the charging period

The energy and exergy efficiencies increased with the temperature increasing of the water supplied by the solar collector unit. The result showed that during the charging period (09:00 h –17:00 h) the energy efficiency ranged from 23 to 41 %, while the net exergy efficiency was in the range of 17 and 28 %. The energy and exergy efficiencies reach their maximum values (41 % and 17 % respectively) at 14:00. We established that the average daily net energy and exergy efficiencies were 32 and 22.5 %, respectively.

4.2 Discharge process

During the discharge process, the temperature distribution of the sand indicated by the 12 thermocouples fixed at different equal intervals depths inside the case. Figure 8 showed that the temperature of the sand at 50 cm-depth decreases vs time.

During the energy discharging, the sand temperature in the storage heat unit decrease quickly in the first two hours. At the end of the discharging process the sand temperature at 50-cmdepth reached a minimum value about 21 °C. The decrease of the sand temperature is due to the release of sensible heat stored in sand.

Figure 8 shows also a good agreement between numerical and experimental results. The rates of heat and thermal exergy recovered from the heat storage unit during thedischarging period were calculated by using Equations (5) and (15), respectively. The changes of the average hourly rates of heat and thermal exergy recovered from the heat storage unit during the discharging period were showed as a function of time in figure 9.

The amount of heat recovered from the heat storage unit was in the range of 2 kW and 2.5 kW. However, the thermal exergy recovered from the heat storage unit ranged from 550 W to 900 W. During this phase the average daily heat and thermal exergy recovered from the heat storage unit were 2.25 kW and 725 W, respectively. It was also found that the difference between energy and exergy analyzes is significant. In fact, since exergy is a measure of the quality of energy, exergy efficiency is more significant than energy efficiency, and that exergy analysis should be considered in the evaluation and comparison of the thermal energy storage systems.

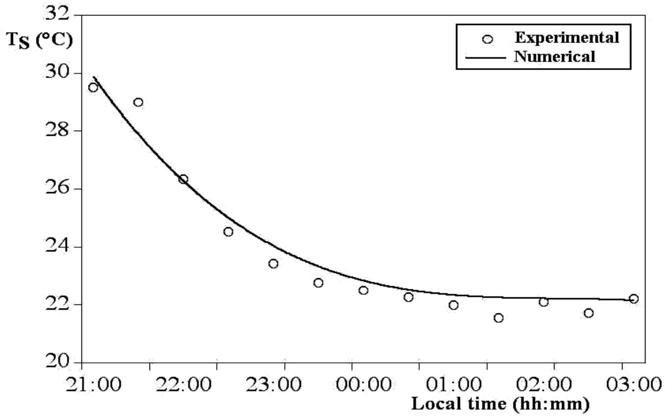

Fig. 8: Evolution of the sand temperature
at 40 cm-depth during the discharging phase

Exergy analysis clearly takes into account the loss of availability of heat in storage operations and, hence, it more correctly reflects the thermodynamic and economic value of the storage operation. The optimization of the design and exploitation of the thermal energy storage systems can be made by means of the exergy analysis. When optimizing the efficiency of a thermal energy storage system, both design and operational parameters must be considered.

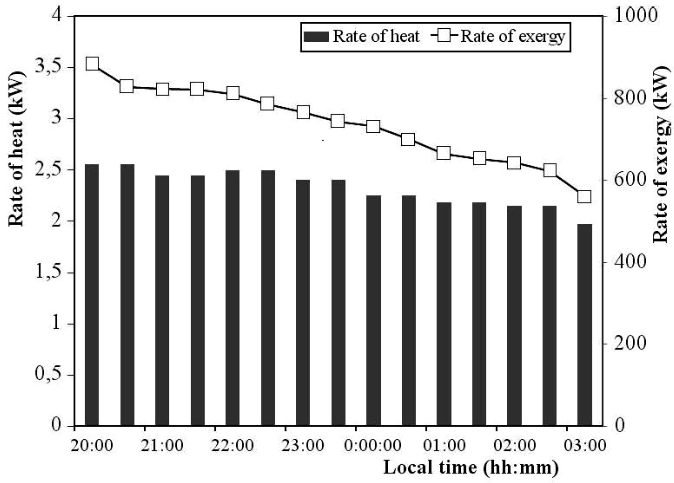

Fig. 9: Rates of heat and thermal exergy recovered
from the heat storage unit for the discharging period

4.3 Heat requirement of the tested room

The total heat recovered from the heat storage unit during the discharging period was compared to the total heat requirements of a tested room (4 m × 3 m × 3 m) calculated from equation (16). In figure 10 was represented the variation of the total heat requirement of the tested room and the amount of heat recovered from the heat storage unit during the discharging period.

We noted that the total heat requirement of the room ranged from 7.5 to 9 kW, while the rate of heat recovered from the heat storage unit to the tested household was in the range of 2 and 2.5 W. We established that about 30 % of the total heat requirement of the tested room was obtained from the heat storage unit.

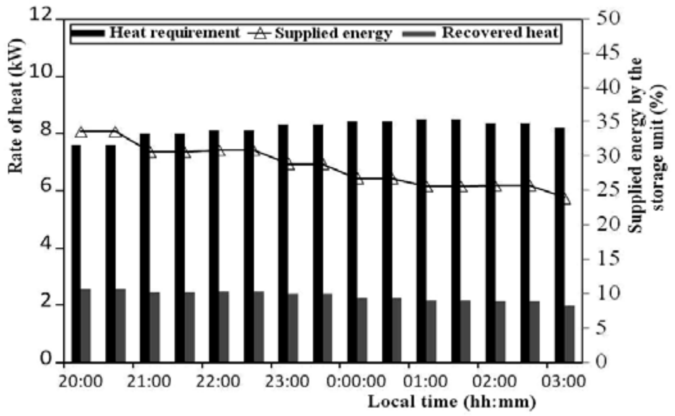

Fig. 10: Changes of the total heat requirement of the tunnel greenhouse and the rate of heat recovered from the heat storage unit during the discharging period

5. CONCLUSION

The performance of a newly designed heat storage system has been presented. Results showed, that the heat storage system attained a considerable rate of heat stored (about 2.8 kW) with an energy efficiency and exergy efficiency equal to 41 % and 28 %, respectively. We noted that the amount of heat recovered from the heat storage unit was in the range of 2 kW and 2.5 kW during the discharging period. However, it was found that the thermal exergy recovered from the heat storage unit ranged from 380 W to 800 W.

The results showed also that the difference between energy and exergy analyzes is significant. Since exergy analysis clearly takes into account the loss of availability of heat in storage operations and, hence, it more correctly reflects the thermodynamic and economic value of the storage operation.

During the discharging period, the total heat requirement of the room was about 8.6 kW, while the rate of heat recovered from the heat storage unit to the tested household evaluated by the numerical study was about 2.5 kW. We established that the heat storage system would be a promising solution for air heating buildings during the night-time. In fact, about 30 % of the total heat requirement of the tested room was obtained from the heat storage unit.

NOMENCLATURE

A_c : Surface area of the collector, m^2

A_d : Surface area of doors, m^2

A_g : Surface area of glasses of windows, m^2

A_s : Surface area of the solar storage unit, m^2

A_w : Surface area of walls, m^2

A_g : Surface area of glass, m^2

A_{Ri} : Internal roof surface, m^2

$C_{p,a}$: Specific heat of air at constant pressure, kJ/(kg.K)

C : Water specific heat, J/(kg.K)

d : Diameter of the capillary heat exchanger, M

H : Solar irradiation at the collector aperture, W/m^2

h_i : Heat transfer coefficient, $W/(m^2.°C)$

M_c : Concrete mass, kg

P :Power consumption for lighting system, W

$Q_{Charging}$: Rate of heat supplied from the collector, W

Q_d : Rate of heat recovery from the accumulator,W

$Q_{Storage}$: Rate of heat stored into the accumulator, W

S_S : Accumulator surface area, m^2

$T_{Abs,av}$: Average concrete temperature, °C

$T_{e,av}$: Average cold-water inlet temperature

T_0 : Water outlet temperature, °C

T_S : Soil temperature inside the greenhouse °C

T_{Ri} : Temperature of the internal roof surface, °C

t : Time, s

$\eta_{Charging}$: Net energy efficiency for charging period, %

C_p : Water specific heat, J/(kg.K)

$C_{p,c}$: Specific heat of concrete absorber, J/(kg.K)

D : Diffusive intensity of radiation

F : Lighting coefficient (F = 1.2 for florescent lamps and 1 for others)

h_{iR} : Heat transfer coefficient of the air under the roof, $W/(m^2.°C)$

L : Length of the capillary tube, m

$\dot{m}_a$: Air mass flow rate, kg/s

$\dot{m}_w$: Flow rate of the charging water, m^3/s

$Q_{Discharging}$: Rate of heat supplied fromthe accumulator, W

Q_{Loss} : Rate of heat loss from the accumulator, W

T_F : Final temperature of the water at the solar storage exit after 12 h, °C

T_a : Ambient air temperature, °C

$T_{a,av}$: Average ambient air temperature, °C

T_i : Water in let temperature, °C

T_m : Soil temperature inside the accumulator

T_{wi} : Temperature of the interior surface of the wall, °C

T_I : Initial temperature of the water in the tank, °C

$\dot{V}$: Volumetric flow rate of air, kg/s

τ_D : Transmission coefficient of glass for diffuse solar radiation

Greek letters

α : Absorption coefficient

τ : Transmittance for solar radiation

λ_s : Equivalent heat conduction coefficient, W/(m.K)

ρ : Water density, kg/m^3

ν : kinematic fluid viscosity

τ_G : Transmission coefficient of glass for direct solar radiation

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REFERENCES

[1] H.H. Öztûrk and A. Basçetinçelik, '*Thermal Energy Storage Techniques*', The Union of Turkish Chambers of Agriculture, Publication Number 230, Ankara, 2002.

[2] T. Boulard and A. Baille, '*Simulation and Analysis of Soil Heat Storage Systems for a Solar Greenhouse, II: Simulation*', Energy in Agriculture, Vol. 5, N°4, pp. 285 – 293, 1986.

[3] C. Gauthier, M. Lacroix and H. Bernier, '*Numerical Simulation of Soil Heat Exchanger-Storage Systems for Greenhouses*', Solar Energy, Vol. 60, N°6, pp. 333 - 346, 1997.

[4] R.J. Krane, '*A Second Law Analysis of The Optimum Design and Operation of Thermal Energy Storage Systems*', International Journal of Heat and Mass Transfer, Vol. 30, N°1, pp. 43 - 57, 1987.

[5] D.L. Larson and L.A.B. Cortez, '*Exergy Analysis: Essential to Effective Energy Management*', Transactions of ASAE, Vol. 38, N°4, pp. 1173 - 1178, 1995.

[6] E. Hikmet, '*Experimental Energy and Exergy Analysis of a Double-Flow Solar Air Heater Having Different Obstacles on Absorber Plates*', Building and Environment, Vol. 43, N°6, pp. 1046 - 1054, 2008.

[7] M. Souliotis and Y. Tripanagnostopoulos, '*Experimental Study of CPC type ICS Solar Systems*', Solar Energy, Vol. 76, N°4, pp. 389 - 408, 2004.

[8] S.V. Patankar, '*Numerical Heat Transfer and Fluid Flow*', Edition Mc Graw-Hill, New York, 1980.

[9] R. Kübler, M. Bierer and E. Hahne, '*Heat Transfer from Finned and Smooth Tube, Heat Exchanger Coils in Hot Water Stores*', International Journal of Heat and Mass Transfer, Vol. 30, N°14, 1987.

[10] H.H Öztûrk, '*The Research on Storage of Solar Energy in Phase Change Material (PCM) for Greenhouse Heating*', Ph.D Thesis, Department of Agricultural Machinery, Institute of Natural and Applied Sciences, University of -Cukurova, Turkey, 1997.

[11] T. Hong and Y. Jiang, '*A New Multizone Model for the Simulation of Building Thermal Performance*', Building and Environment, Vol. 32, N° 2, pp. 123 – 128, 1997.

[12] M.S. Hatamipour, H. Mahiyar and M. Taheri, '*Evaluation of Existing Cooling Systems for Reducing Cooling Power Consumption*', Energy and Buildings, Vol. 39, N°1, pp. 105 - 112, 2007.

[13] A. Sharian, B. Shalabi a, A. Rousan and B. Tashtoush, '*Effects of the Absorptance of External Surfaces on Heating and Cooling Loads of Residential Buildings in Jordan*', Energy Conversion and Management, Vol. 39, N°3/4, pp. 273-284, 1998.

[14] Y. Jiang, '*State Space Method for Analysis of the Thermal Behavior of Rooms and Calculation of Air-Conditioning Load*', *ASHRAE Transactions*. Vol. 88, pp. 122–132, 1981.

[15] E. Shaviv, A. Yezioro and I.G. Capeluto, '*Thermal Mass and Night Ventilation as Passive Cooling Design Strategy*', Renewable Energy. Vol. 24, N°3-4, pp. 445 - 452, 2001.