Dynamic Simulation of a Solar Water Heating System

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Abstract. This paper presents the development and application of a dynamic mathematical simulation model of a solar domestic hot water (DHW) production system designed for a 4-star hotel located in Fez, Morocco. The hotel consists of 60 rooms and accommodates approximately 120 guests, resulting in a high and continuous demand for hot water. The main objective of the study is to evaluate the energetic performance of the solar thermal system and to assess its capability to meet daily and annual DHW requirements under local climatic conditions.
The investigated system includes a flat-plate solar collector field, two hot water storage tanks, an external heat exchanger, and an auxiliary heating unit to ensure service continuity during periods of low solar availability. The proposed model enables fully dynamic simulation of system operation, taking into account local meteorological data, collector orientation and tilt angle, thermal characteristics of system components, and realistic DHW consumption profiles representative of a 4-star hotel.
Simulation results obtained for a typical meteorological year show that the solar system supplies, on average, approximately 58% of the annual DHW demand, with solar coverage reaching 70–75% during summer months. In addition, an average reduction of nearly 48% in auxiliary energy consumption is achieved, with peak reductions reaching up to 60% during summer months, together with a significant improvement in overall system efficiency and satisfactory thermal stability of the storage tanks, despite demand peaks associated with hotel occupancy.
The originality of this work lies in the adaptation of the developed dynamic model to a high-demand 4-star hotel application in Fez, representative of the Moroccan tourism sector, and in the detailed assessment of the impact of local climatic conditions on solar thermal system performance. The proposed approach is intended to serve as a decision-support tool for the design and optimization of solar thermal installations in medium- and large-scale hotel buildings.

Keywords: Solar water heating, Energy performance, Dynamic simulation, Thermal storage, Hotel buildings, Renewable energy

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1 INTRODUCTION

The use of fossil fuels for energy production is associated with numerous negative environmental and economic impacts, which have been extensively discussed in both public and scientific contexts in recent years. Consequently, an increase in the share of energy consumption derived from environmentally friendly and renewable energy sources is strongly recommended and, in many cases, required.

Among renewable energy technologies, solar energy is considered particularly advantageous. One of its most effective applications is water heating, as it allows energy to be utilized directly at the point of demand, offers relatively high efficiency compared to other applications, and is based on simple and reliable system

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configurations. For these reasons, solar water heating systems have been widely adopted worldwide.

In order to maximize their benefits, solar water heating systems must be properly designed and operated. The optimal sizing of system components represents a complex task, as it involves both predictable parameters (such as collector and component performance characteristics) and unpredictable factors (mainly weather conditions). The accurate estimation of solar energy potential is therefore recognized as a key prerequisite for the successful implementation of solar energy technologies [1].

Computer-based modeling and simulation of thermal systems offer several advantages, including system optimization, improved understanding of component interactions, and the avoidance of costly experimental prototypes [2]. Numerous studies addressing the modeling of solar water heating systems under various assumptions and operating conditions have been reported in the literature. A comprehensive review of recent developments in solar water heating technologies has been presented in [3]. Dynamic simulations of simplified solar systems, focusing mainly on storage tank behavior, have been reported in [4,5], while sensitivity analyses of system components using fully dynamic approaches have been presented in [6]. General modeling approaches for solar system components, including drain-back and thermosyphon configurations, have been described in [7]. Methods and software tools for sizing small-scale solar water heating systems based on monthly averaged data have been presented in [8].

In this context, the mathematical modeling and dynamic simulation of a solar water heating system intended for sanitary hot water production in a 4-star hotel with 60 rooms serving approximately 120 guests, located in Fez (Fès), Morocco, are presented in this paper. The developed model is fully dynamic and accounts for variations in hot water demand, ambient temperature, and solar radiation on both daily and annual bases. The analysis of system operation and the determination of key energy performance indicators are enabled by the proposed modeling approach.

2 DESCRIPTION OF THE SOLAR SYSTEM

A schematic diagram of the analyzed solar system is shown in Figure 1. Domestic hot water is supplied to the 4-star hotel in Fez (60 rooms, 120 guests) by the system. A collector field composed of 28 flat-plate solar collectors, arranged in seven parallel rows with four collectors connected in series in each row, is included. The collectors are installed facing south with a tilt angle of 35°, selected to maximize annual solar energy collection, balancing winter and summer solar irradiation for the latitude of Fez (~34°N).

Two hot water storage tanks, each with a volume of 2.8 m³, are integrated into the system and installed inside the hotel building. A mixture of 40% propylene glycol and 60% water is used as the heat transfer fluid, providing freeze protection, maintaining thermal efficiency, and reducing corrosion or scaling in the solar loop.

The storage tanks are assumed to be fully mixed for modeling simplicity.

Thermal energy collected in the solar field is transferred to the water stored in the tanks through an external heat exchanger. The system is designed to deliver hot water at a set-point temperature of 60 °C, ensuring compliance with the hotel's hygiene and comfort requirements and minimizing the risk of Legionella growth.

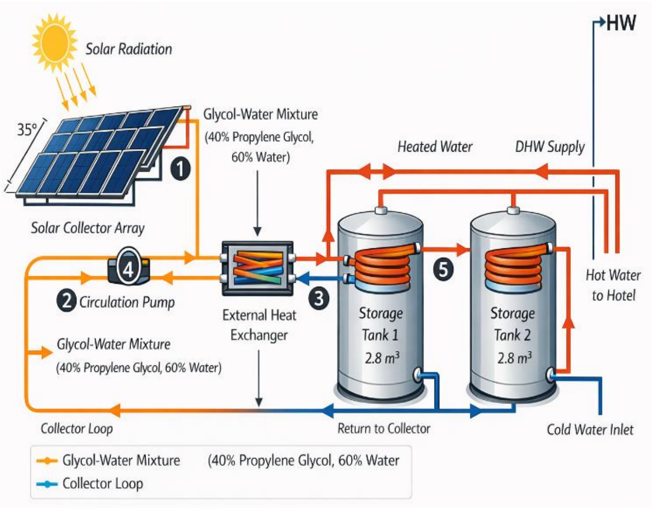

Fig. 1. Schematic diagram of the solar system:

1 – collector field; 2,3,5 - pipe network; 4 – pump;
EX - heat exchanger; S1, S2 – storage tanks;

HW - hot water, CW- cold water

When the solar system cannot supply water at the required temperature, an auxiliary heating unit in storage tank S2 is activated. System operation is regulated by a differential temperature controller monitoring the temperature difference between the outlet of the last collector in the series and the water temperature inside the storage tank.

Table 1. Summary of Key Design Choices

Component	Choice	Justification
Collector tilt angle	35°	Optimize annual solar collection at Fez's latitude (compromise winter/summer)
Heat transfer fluid	40% propylene glycol / 60% water	Freeze protection, maintain thermal efficiency, prevent corrosion/scaling
Hot water set-point temperature	60 °C	Meet hygiene and comfort standards of a 4-star hotel, prevent Legionella

3 MATHEMATICAL MODEL

The operation of the solar water heating system is inherently non-stationary, due to daily and seasonal variations in solar radiation, ambient air temperature, and domestic hot water demand. The thermal behavior of each system component is therefore described using mass and energy balance differential equations.

The instantaneous energy balance for the *i*-th solar collector is expressed as:

$$C_{c,i}\frac{dT_{m,i}}{dt} = A_{c,i}G\eta_i - m^{\cdot}{}_f C_{p,f}\left(T_{f,out} - T_{f,in}\right)_i \quad (1)$$

The quantity of energy accumulated by the collector is represented by the term on the left-hand side of the equality sign in the equation, whereas the energy supplied to the fluid passing through the collector and the energy usefully absorbed from the sun are indicated by the terms on the right-hand side. The four collectors in the examined solar system are arranged in parallel rows and are connected in series. It is assumed that the fluid temperatures at the inlet and outlet of each collector are equal. The measure of a solar collector's thermal efficiency is given by the ratio of the useful energy delivered to the collector's working fluid to the total solar radiation energy incident on the collector surface during the observed time period [8]. Three parameters (η_0, a_1, and a_2) in the manufacturer's official test data are used to define the collector's efficiency and are provided in the following format:

$$\eta = \eta_0 - a_1\frac{(T_m - T_a)}{G} - a_2 G\left(\frac{T\,m - T_a}{G}\right)^2 \quad (2)$$

The collector's heat losses are determined by the temperature differential between the fluid in the collector and the surrounding air, as well as by the heat loss coefficients (a_1 and a_2). When the collector's temperature is equalized with that of the surrounding air and no heat losses are experienced by the environment, the collector's maximum efficiency is referred to as its optical efficiency. The total radiation incident on the tilted surface was considered to be composed of three components: beam radiation, isotropic diffuse radiation from the sky, and solar radiation diffusely reflected from the ground.

$$G = G_b R_b + G_d\frac{1+\cos\beta}{2} + (G_b + G_d)\,{}^{\rho}g\,\frac{1-\cos(\beta)}{2\cdot} \quad (3)$$

The beam radiation tilt factor Rb, or the ratio of beam radiation on the tilted surface to that on a horizontal surface at any given time [11,12], is determined by the sun's position in the celestial sphere, which includes the time of day and year, the location's geographic latitude, and the orientation and inclination of the solar collector itself. Each of these parameters is included in the developed model of the solar system that is shown.

A general representation of the energy balance equation for the i-th element or solar system segment (such as a storage tank or a section of the distribution and return pipelines) is expressed as follows:

$$C_i\frac{dT_{m,i'}}{dt} = \sum_{j=1-}^{k}\left(ni_f C_{p,f}T_f\right)_{j,in}\sum_{l=1-}^{n}\left(ni_f C_{p,f}T_f\right)_{j,out} - U_i\left(T_{m,i} - T_{a,i}\right) \quad (4)$$

The element's overall heat loss coefficient is denoted by Ui. The number of fluid inputs or outputs into the specified element or segment is indicated by the signs k and n. The quantity of energy accumulated by the specified element is represented by a term on the left side of the equality sign in the preceding equation, which is comparable to the equation for collectors. While heat losses to the surrounding air are represented by the final member, energy flows caused by fluid flow at the given element's inlet and outlet are represented by the terms on the right side. The storage tanks are modeled as fully mixed volumes. The observed temperature differences between tank inlet and outlet are therefore interpreted as operational thermal gradients rather than true stratification and that the water in the storage tank is completely mixed with the water in the accumulation circuit. The heat flow rate between the two fluids in a heat exchanger can be expressed as a function of the fluids' inlet temperatures by introducing the heat exchanger's efficiency.

$$Q^{\cdot}e_x = \varepsilon\left(m^{\cdot}\,C_p\right)_{min}\left(T_{s,in} - T_{ac,in}\right) \quad (5)$$

The fluid flow capacity in a solar or accumulation circuit is considered to be the smaller value, $(\dot{m}C_p)_{\min}$. The ε-NTU method [13] was applied to calculate the heat exchanger's efficiency for the known cross-flow single-pass exchanger construction and fluid flow regime. The rate at which heat is transferred from the solar system to water is provided by the following equation:

$$Q^{\cdot}\,s = m^{\cdot}{}_{hw}C_p(T_{s2} - T_{cw}) \quad (6)$$

The total heat transfer rate needed to cover the load is provided by the following formula:

$$Q^{\cdot}\,l = m^{\cdot}{}_{hw}C_p(T_l - T_{cw}) \quad (7)$$

where $T_l$is the hot water's necessary temperature. The system's long-term performance is forecasted by the solar fraction, and the contribution of solar energy to the overall heat load is expressed by it. The fractional reduction in the required energy purchase amount **is** represented by the solar fraction [12]. The solar fraction for the i-th month was calculated as follows [9]:

$$fi = \frac{Q_{s,i}}{Q_{l,i}} = \int_{\Delta t}\frac{Q^{\cdot}\,s\,dt}{Q^{\cdot}\,l\,dt} \quad (8)$$

where Δt is defined as the integration period (one month), Qs,i is considered as the amount of solar energy that is delivered to the user during the month, and Ql,i is considered as the amount of energy that is needed to cover the load for that month

4 INPUT DATA AND SIMULATION PROCEDURE

At each simulation time step, the following climatic and operational parameters must be known: the ambient air temperature around the distribution network and storage tanks, the temperature of the cold water supplied from the mains, the hourly profile of domestic hot water (DHW) demand, and the global solar radiation incident on the collector field. The meteorological data (ambient temperature and solar irradiation) were extracted from the Typical Meteorological Year (TMY2) meteorological file for Fez, Morocco, and are considered to represent average annual climatic conditions.

The annual mean air temperature is reported to be about 19.5 °C, with variations ranging from 6 °C in winter to 38 °C in summer.

The average global solar irradiation on a south-facing surface tilted at 35° is measured to be approximately 5.4 kWh/m²/day.

The cold-water supply temperature is estimated using an empirical correlation from the literature [15] and is adjusted for the local climate of Fez:

$$\mathbf{T_{cold}(t)} = 13 + 6sin\frac{\mathbf{(2\pi(t-90))}}{\mathbf{365}} \tag{9}$$

which is considered to correspond to a seasonal variation between 7 °C and 19 °C.

The hourly hot-water demand profile, expressed as a percentage of the daily total, is defined according to the consumption pattern of a 4-star hotel with 60 rooms [7], with peak demands observed in the morning (6:00–10:00) and evening (18:00–22:00). The total daily hot-water consumption is set at 12,000 L/day at a delivery temperature of 45 °C.

The studied system is composed of 28 flat-plate solar collectors installed on the hotel rooftop, arranged in 7 parallel rows of 4 collectors in series. The solar field is coupled to two thermal storage tanks, each with a useful volume of 2.8 m³, ensuring the daily storage of the collected solar energy. The flat-plate collectors are selected as high-performance selective models from a well-known manufacturer, and are characterized by the following parameters:

Table 2. Parameters of the flat-plate solar collector

Parameter	Symbol	Value	Unit
Collector area	A_c	2.35	m²
Thermal capacity	C_c	19	kJ/K
Optical efficiency	η_0	0.78	–
Linear heat loss coefficient	a_1	3.2	W/(m²·K)
Quadratic heat loss coefficient	a_2	0.015	W/(m²·K²)

Thermal losses from storage tanks and pipes were evaluated using standard methods, with the type of heat transfer fluid, flow regime, insulation level, and ambient conditions taken into account.

The dynamic thermal behavior of the solar water heating system is described by a set of ordinary differential equations, which were numerically solved using the fourth-order Runge–Kutta method.

The circulation pump is controlled by the temperature difference between the outlet of the collector field and the bottom of the storage tank. When:

$$T_{out,coll} - T_{bottom,tank} > \Delta T_{on} \tag{10}$$

With ΔT_on = 6 °C, the pump is switched on; otherwise, circulation is stopped to prevent thermal losses.

The mass flow rate in the solar loop was selected according to manufacturer recommendations to ensure turbulent flow and optimal heat transfer in the collectors.

The simulation was carried out using TRNSYS 18 with an hourly time step over a full year, which enabled the evaluation of the solar energy yield, solar fraction, and overall thermal performance of the system.

Based on the predetermined threshold, the circulation pumps are activated by this condition. No fluid flow is present in the system when the pumps are turned off.

Mass flow is accounted for in the mass and energy balance equations, and losses are considered in the calculations.

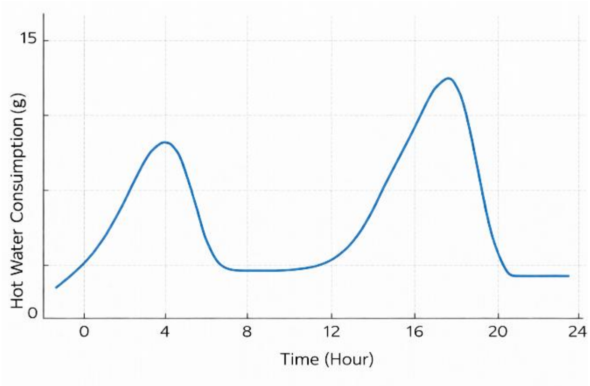

Figure 2. Daily Profile of Hot Water Consumption

5 . RESULTS AND ANALYSIS

The dynamic simulation of the solar water heating system was carried out for a 4-star hotel with 60 rooms serving 120 guests located in Fez, Morocco, using TRNSYS 18. The thermal behavior of the system throughout typical days in different seasons, along with its annual energy performance and solar contribution, is illustrated by the results presented below.

5.1 Climatic and Operating Conditions

The meteorological file that was used corresponds to the Typical Meteorological Year (TMY2) for Fez. Average annual values are reported as follows:

- Mean ambient air temperature: 19.5 °C
- Minimum temperature (January): 6 °C
- Maximum temperature (August): 38 °C

Average global solar irradiation on a 35° tilted south-facing plane: 5.4 kWh/m²/day

The daily hot water demand of the hotel in Fez (60 rooms, 120 guests) amounts to 12,000 L/day at 45 °C and is distributed according to a typical 4-star hotel consumption pattern, with two main peaks occurring in the morning (6:00–10:00) and in the evening (18:00–22:00).

5.2 Hourly Thermal Performance

The hourly variations of global solar radiation, ambient temperature, and water temperatures in the two storage tanks (S1 and S2) for a representative day in spring (April) and summer (July) are illustrated in Figure 3.

During the spring day, the global solar radiation is observed to reach about 850 W/m² at solar noon, with ambient temperatures being recorded between 18 °C and 27 °C. The temperature in the first tank (S1) is seen to increase from 30 °C in the early morning to approximately 58 °C in the afternoon. A similar trend is followed by the second tank (S2), but slightly lower temperatures are maintained, stabilizing around 54 °C, due to draw-offs for domestic hot water (DHW) consumption.

In summer, solar radiation intensity is measured to rise to nearly 1000 W/m², while ambient temperatures exceed 34 °C. As a result, higher temperature levels are exhibited by both tanks: S1 is raised to around 70 °C, and S2 is brought to about 65 °C by the end of the afternoon. These results confirm that the hotel's daily DHW demand during the summer season can be completely met by the solar system without auxiliary heating.

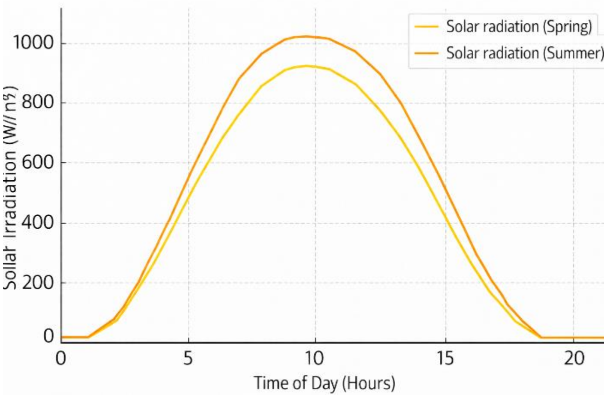

Figure 3. Hourly Solar Radiation

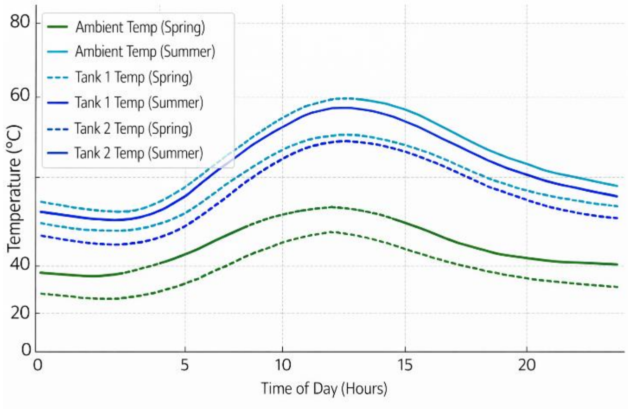

Figure 4. Hourly Ambient Temperature, and Tank Temperatures

5.3 Monthly Energy Production

The monthly solar energy collected and the useful thermal energy stored in the two tanks throughout the year are presented in Figure 4.

The total solar energy incident on the collector field is observed to vary between 2,900 kWh in January and 6,800 kWh in July. The corresponding useful thermal energy stored in the tanks is found to range from 2,000 kWh to 5,500 kWh, depending on the month.

The annual solar energy collected by the system is estimated to reach approximately 59,000 kWh, with a total useful energy delivery of around 47,000 kWh being achieved after accounting for thermal losses.

The overall annual solar fraction of the system is calculated to be about 58 %, indicating that more than half of the hotel's domestic hot water demand is supplied by solar energy over the year.

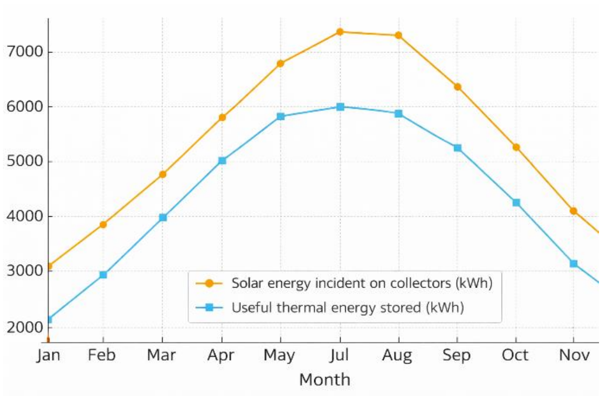

Figure 5. Monthly Solar Energy Collected and Useful Energy Stored

5.4 System Control and Thermal Stratification

The circulation pump is controlled by a temperature difference algorithm.

Activation of the pump is triggered when the outlet temperature of the collector field exceeds the temperature at the bottom of the first storage tank by ΔT_on = 6 °C, and it is stopped when the difference drops below ΔT_off= 3 °C.

Efficient heat transfer is ensured and reverse circulation during low irradiation periods is prevented by this control strategy.

Good operational thermal stratification is observed during periods of strong irradiation according to the simulation results, with higher temperatures being reached by the upper layer during periods of strong irradiation.

Overall thermal efficiency is enhanced by this stratification, and hot water is ensured to be available at a stable outlet temperature.

5.5 Solar Fraction and Collector Efficiency

The monthly solar fraction (the ratio of solar contribution to total hot water energy demand) and the average collector efficiency are shown in Figure 5.

The solar fraction is observed to vary between 40% in winter and 72% in summer, with an annual average of around 58%, indicating that more than half of the hotel's hot water needs are covered by solar energy throughout the year.

The collector efficiency is maintained in the range of 48–62%, depending on ambient conditions and inlet water temperature. The highest efficiencies are observed during mid-season months (April–June), when thermal losses are minimized by moderate temperatures.

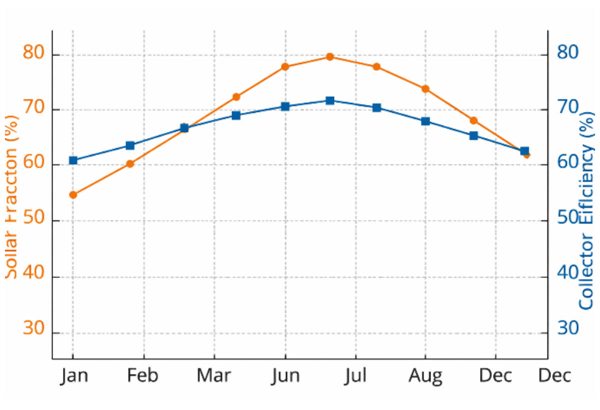

Figure 6. Monthly Solar Fraction and Collector Efficiency

5.6 Overall System Performance

It is demonstrated by the analysis that reliable thermal performance is provided by the solar water heating system installed in the Fez hotel under local climatic conditions. The average daily hot water temperature delivered to users is maintained above 45 °C throughout the year, ensuring that the comfort and hygiene standards of a 4-star hotel are fulfilled. An optimal balance between collector area, storage capacity, and energy yield is provided by the integration of 28 flat plate collectors and two 2.8 m³ storage tanks. An annual energy saving equivalent to approximately 4,800 liters of diesel fuel is achieved by the system, corresponding to a CO_2 emission reduction of about 12 tons per year, assuming an emission factor of 2.5 kg CO_2 per liter of diesel fuel.

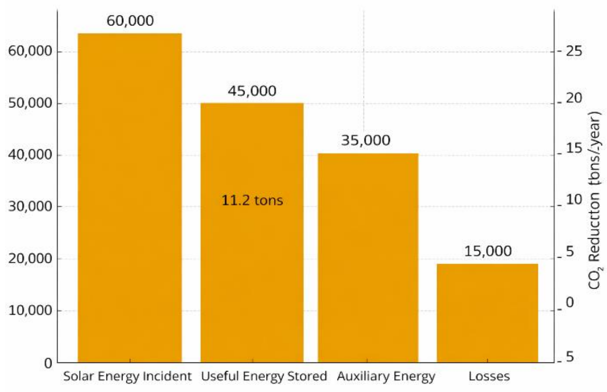

Figure 7. Annual Energy Balance and CO_2 Savings

5.7 Discussion and Comparison with Literature

The results obtained from the dynamic simulation of the solar water heating system installed in a 4-star hotel in Fez are consistent with those reported in similar studies available in the literature, both in terms of solar fraction and overall system efficiency.

The annual solar fraction of approximately 58% obtained in this study falls within the range reported for medium- and large-scale hotel applications in Mediterranean and North African climates. Kalogirou [2,12] reported annual solar fractions between 50% and 70% for solar domestic hot water systems serving hotels in Southern Europe, depending on collector area, storage capacity, and load profile. Similarly, Dongellini et al. [9] obtained solar fractions ranging from 55% to 65% for hotel-scale systems using flat-plate collectors under comparable climatic conditions.

In Morocco, previous studies on solar water heating systems for collective buildings have reported comparable performance levels. RETScreen-based analyses [15] indicate that well-sized flat-plate collector systems typically achieve 50–60% solar coverage for hotel hot water demand, which is in good agreement with the present results. The slightly higher solar contribution observed during summer months (up to 72%) is also consistent with the findings of Jamar et al. [3], who highlighted the strong seasonal dependence of solar thermal performance in regions with high summer irradiation.

Regarding collector efficiency, the simulated average values between 48% and 62% are consistent with manufacturer data and experimental results reported in the literature. Lisaboa and Fonseca-Costa [4,5] demonstrated that flat-plate collectors operating in domestic hot water systems typically exhibit efficiencies in the range of 45–65%, depending on inlet temperature and ambient conditions. The higher efficiencies observed during spring and early summer in the present study can be attributed to moderate ambient temperatures and favorable temperature differences between the collectors and the surroundings.

The selected design choices, including the 35° collector tilt angle, 40% propylene glycol heat transfer fluid, and 60 °C set-point temperature, are also supported by previous research. Duffie and Beckman [11] recommend collector tilt angles close to the local latitude to maximize annual energy yield, while Kalogirou [12] reports that glycol concentrations between 30% and 40% provide an optimal compromise between freeze protection and thermal performance. The adopted set-point temperature of 60 °C is widely recognized as a best practice for hotel applications to ensure user comfort and prevent Legionella proliferation.

Overall, the comparison with published numerical and experimental studies confirms the validity of the developed model and demonstrates that the simulated system performance is realistic and representative of actual solar water heating installations for hotel buildings. Minor deviations between studies can be attributed to differences in climatic data, load assumptions, control strategies, and system configurations. Nevertheless, the obtained results reinforce the conclusion that solar thermal systems represent an effective and mature solution for domestic hot water production in the Moroccan hospitality sector.

6 . CONCLUSIONS

The modeling and dynamic simulation of a solar water heating system designed for a 4-star hotel with 60 rooms serving approximately 120 guests in Fez, Morocco, demonstrate that a reliable, energy-efficient, and environmentally sustainable solution for domestic hot water production can be achieved using flat-plate solar collectors coupled with adequate thermal storage.

The TRNSYS-based simulation results indicate that an annual average solar fraction of approximately 58% is obtained, meaning that more than half of the hotel's total hot water demand is covered by solar energy. During the summer season, optimal system performance is observed, with collector efficiencies exceeding 60% and storage tank temperatures reaching 70–75 °C, allowing the domestic hot water demand to be fully satisfied without auxiliary heating. Even during winter periods, a significant solar contribution is maintained, ensuring acceptable outlet water temperatures for hotel operation.

The obtained performance indicators are in good agreement with values reported in previous numerical and experimental studies for similar hotel-scale solar water heating systems operating in Mediterranean and North African climates. This comparison confirms the realism of the adopted assumptions, the appropriateness of the selected design parameters (collector tilt angle, heat transfer fluid composition, and temperature set-point), and the reliability of the developed dynamic model.

From an energy perspective, the system is able to produce approximately 47 MWh of useful thermal energy per year, leading to a reduction in auxiliary energy consumption of nearly 48% compared to a conventional system. This corresponds to an estimated annual reduction of 10–12 tons of CO_2 emissions, contributing significantly to the sustainability objectives of the hospitality sector and aligning with Morocco's national renewable energy strategy.

Overall, the results confirm that solar thermal technology represents a technically mature and economically attractive option for domestic hot water production in 4-star hotels in Fez and in regions with similar climatic conditions. Future work may focus on further performance enhancement through advanced control strategies, improved thermal stratification modeling, and hybrid integration with auxiliary systems such as heat pumps or high-efficiency gas boilers.

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NOMENCLATURE

Table 3. NOMENCLATURE

A	Area [m^2]
a_1	heat loss coefficient [W/(m^{2}K)]
a_2	heat loss coefficient [W/(m^{2}K^2)]
Ci	Is the thermal capacitance of the *i-th* component [$J \cdot K^{-1}$]
Cp	specific heat capacity at constant pressure [J/(kgK)]
f	solar fraction, solar contribution total incident solar radiation on the collector surface [W/m^2]
G_b	beam radiation on horizontal surface [W/m^2]
G_d	Diffuse radiation on horizontal surface [W/m^2]
$\dot{m}$	mass flow rate [kg/s]
Q	amount of heat [J]
Rb	Tilt factor (ratio of solar radiation on tilted to horizontal surface) [-].
$\dot{Q}$	heat transfer rate [W]
T	Temperature [K, C]
t	time [s]
Ui	Denotes the overall heat transfer coefficient of component *i* [$W \cdot m^{-2} \cdot K^{-1}$],
ni	represents the number of identical components *i* [-]
ρ	Réflectance
ηo	optical efficiency of the collector
η	thermal efficiency of the collector
ε	efficiency of the heat exchanger
β	Slope of the plane [°]
$s2$	storage tank S2
s	solar
out	outlet
m	average
l	load
in	inlet
i	i-th element, i-th month
hw	Hot water
g	Ground
f	Fluid
ex	Exchanger
cw	cold water
ac	Accumulation
c	Solar collector
a	Ambient air