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Analytical Analysis of Thermal Energy Storage Performance of Room Heating System by Solar Energy

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ABSTRACT
A mathematical model is presented to study the performance of thermal storage for excess solar energy at day to heat a living room using a tank of water as thermal storage. The study studies the relationship between the room temperature and the appropriate tank geometries through the day hours. Two storage geometries have been studied; the tank surface area which was varied from 6-12 m^2 and the storage water mass per unit area of the collectors (50-70 kg m^{-2}) on the storage temperature. The study is based on several years collected data including solar intensity, ambient temperature and the wind speed which were measured in Tikrit city-North West of Iraq-in January. The solar collector was constructed from multiple standard units; each unit is (1 m) width and (2 m) length. Each unit consists of six copper pipes with 10 mm diameter and 2 m long covered by galvanized steel sheet 1 mm thickness and a glass cover of 3 mm thickness. The mathematical model was converted to a simulation computer program accounting for all effective parameters on heat transfer in the heated room space, the collector and the storage tank. The results show that thermal storage has an important effect on raising the room temperature for many hours in the evening and nearly reaching the comfortable conditions requested at night.

Key words: Thermal energy storage, integrated solar collector, solar energy, solar heating, solar water heater

INTRODUCTION
Solar heating has good potential in the Middle East region particularly the Arab Gulf countries since the solar intensity is sufficient all around the year to provide the required thermal energy. By the solar heating, the consumption of electrical energy can be reduced. For space heating at night, the collected solar energy is required to be stored during the day and released during the night. Since, water has high specific heat, it is recommended for Thermal Energy Storage (TES). One of the earlier studies about using solar energy for heating purpose was carried out experimentally by Hutchinson (1956). He utilized the negative effect for heating by designing and constructing two houses made of wood and insulations, one of them its south wall had a large double glass windows and the other without them. The effect of solar heating was clearly recorded in heating at day. To achieve the thermal storage, water tanks was added behind the windows to study the effect of thermal storage on heating during hours when sun's ray is disappeared. The researcher noticed the effect of this type of thermal storage on solar heating and made it

appropriate for most day hours. Since, the beginning of the last century studies expanded regarding the positive impact of heating included a myriad of applications ranging from sizes of solar heater collectors form no more than a few square meters to industrial applications and productivity which is used by pool solar size up to thousands of square meters. All these applications depend on the nature and intensity of solar radiation and environmental and climatic conditions of the region. Therefore, the heating process, as one of these applications, depends on the design nature of the solar collector, the area and the size. Also, nature of the thermal reservoir and mass of storing water per square meter of the area of the solar collector is of concern for the house heating applications.

An indirect forced circulation solar water heating systems using a flat-plate collector was modeled by Hobbi and Siddiqui (2009) for domestic hot water of a single-family residential unit in Montreal, Canada using TRNSYS simulation program. The solar fraction of the entire system is used as the optimization parameter. In their simulation, the design parameters of both the system and the collector were optimized including the collector area, the fluid type, the mass flow rate, the volume of storage tank, the heat exchanger effectiveness, the sizes and lengths of the connecting pipes and all the absorber related parameters. Their simulation results show that the designed system could provide 83-97 and 30-62% of the hot water demands in summer and winter, respectively.

Waluyo and Majid (2011) presented a practical method to determine the performance parameters of stratified TES based on thermocline profile. They represented the temperature distribution by non linear regression fitting to identify the function which predicts the temperature distribution profile. The function was used to define performance parameters namely limit points of thermocline, thermocline thickness, lost capacity, integrated capacity and the theoretical capacity. Results identified a function which could represent S-curve of temperature distribution, namely Sigmoid Dose Response (SDR) equation. The function was observed to fit the temperature distributions with high accuracy. They claimed that the method was capable to be utilized for evaluation of the performance of the stratified TES.

Haillot *et al.* (2011) evaluated the performance of a Solar Domestic Hot Water (SDHW) system including a latent storage material. The approach consists of a composite made of Compressed Expanded Natural Graphite (CENG) and Phase Change Material (PCM) directly inside a flat plate solar collector in order to replace the traditional copper-based solar absorber. The focus was on the selection of the most promising composite to implement in the solar collector. Several composites based on CENG and various storage materials (paraffin, stearic acid, sodium acetate trihydrate and pentaglycerin) have been elaborated and characterized. The synthesis of all these measurements allowed them to select three composites whose characteristics match the integration into a solar thermal collector.

A system for capturing and storing solar energy during the summer for use during the following winter has been simulated by Sweet and McLeskey Jr. (2012). Flat plate solar thermal collectors attached to the roof of a single family dwelling were used to collect solar thermal energy year round. The thermal energy was then stored in an underground fabricated Seasonal Solar Thermal Energy Storage (SSTES) bed. The SSTES bed allowed for the collected energy to supplement or replace fossil fuel supplied space heat in typical single family homes in Richmond, Virginia, USA. TRNSYS software was used to model and simulate the winter thermal load of a typical Richmond home. The optimization of the SSTES scheme showed that a 15 m^3 bed volume, 90% of the south facing roof and a flow rate of 11.356 L m^{-1} through the solar collectors were optimal parameters. The overall efficiency of the system ranged from 50-70% when compared to the total useful energy gain of the solar collectors.

The objective of the recent study was planned for space heating in Iraq winter season through a theoretical modeling of the collector/storage system and documental measurements for the most environmental and climatic conditions to get optimal solar heating. The solar collector selected for the study is a flat plate type with six copper pipes with diameter of 10 mm and a pitch step of 0.12 m. The thermal storage system is a 100 L water tank. The storage tank surface area was one of the evaluated parameters. Also, the water storage size required to meet the comfortable temperature in the room was estimated.

ANALYTICAL MODEL

The proposed arrangement of the system is shown schematically in Fig. 1. The early Hottel-Whillier equation is used in the evaluation of useful energy of the solar collector (Duflie and Beckman, 1991) as:

$$Q_u = A_C F_R\,[I\alpha_S\tau - U_C(T_C - T_a)] \tag{1}$$

where, F_R is the solar collector heats reject which is given by:

$$F_R = \frac{G_w C_p}{U_C}\left[1 - \exp\left(-\frac{U_C F^-}{G_w C_p}\right)\right] \tag{2}$$

where, F^- is the coefficient of solar collector efficiency. For tubes under absorber surface, it is given by:

$$F^- = \frac{1/U_c}{D_o + 2W\left[\frac{1}{U_c\,(D_o + 2W\eta_f)} + \frac{1}{h_f \pi D_i}\right]} \tag{3}$$

As commonly practiced, the arrangement of the pipe-plate of the collector is considered as a fin with standard straight rectangular section, where, it's efficiency can be calculated as:

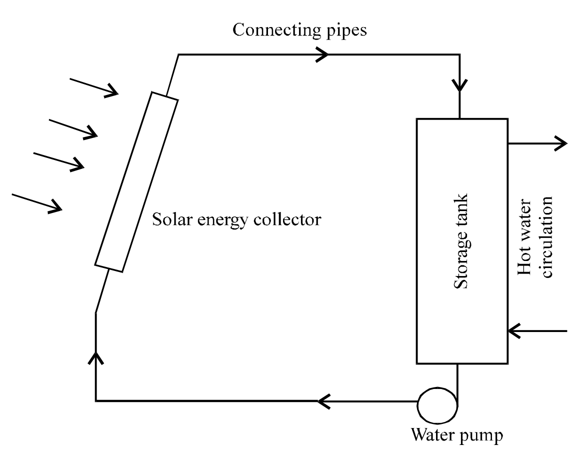

Fig. 1: Schematics of the solar heating system

$$\eta_f = \frac{\tanh ZW}{ZW} \tag{4}$$

And the dimension less parameter of the fin, Z can be defined as:

$$Z = \left(\frac{U_c}{K_c t}\right)^{\frac{1}{2}} \tag{5}$$

It is possible to calculate the heat transfer coefficient of water flow inside the solar collector tubes, h_i from the following correlation, Gary and Prakash (2005):

$$Nu = \frac{h_i D_i}{K_w} = 3.66 + \frac{0.668\left(\frac{D_i}{L_i}\right) Re\ Pr}{1 + 0.04\left[\left(\frac{D_i}{L_c}\right) Re\ Pr\right]^{\frac{2}{3}}} \tag{6}$$

The temperature of the solar collector can be determined from the energy balance between the heat energy for the water in pipes and the heat energy of the collector surface which after arranging becomes as:

$$T_c = T_m + \frac{m_w C_p (T_{w_o} - T_{w_i})}{h_i \pi D_i L_c} \tag{7}$$

Overall heat transfer coefficient for solar collector: To calculate the overall heat transfer coefficient for solar collector, three heat transfer sides must be done, as following.

Heat transfer coefficient from upper surface of the collector as follow:

$$U_{tc} = \left[\frac{N_g}{\left(\frac{344}{T_c}\right)} + \frac{1}{hw} + \frac{\sigma\ (T_c + T_a)\ (T_c^2 + T_a^2)}{[\varepsilon_o + 0.0425\ N_g\ (1-\varepsilon_g)]^{-1} + \left[\frac{2N_g + f - 1}{\varepsilon_g}\right] - N_g}\right] \tag{8}$$

where, h_o and f can be determined as:

$$h_w = 5.7 + 3.8 V_{wi} \tag{9}$$

$$f = (1+0.089h_w - 0.116h_w\varepsilon_c)\ (1+0.07866N_g) \tag{10}$$

Heat transfer coefficient from lower surface of the collector as follow:

$$U_{bc} = \frac{K_{bc}}{X_{bc}} \tag{11}$$

Heat transfer coefficient from sides surfaces of the collector:

$$U_{ec} = \left(\frac{K_{ec}}{X_{ec}}\right)\left(\frac{A_{ec}}{A_c}\right) \tag{12}$$

Then the overall heat transfer coefficient for the solar collector will be the sum of the three above equations:

$$U_c = U_{tc}+U_{bc}+U_{ec} \tag{13}$$

Heat transfer coefficient of the reservoir: Heat transfer coefficient of the reservoir can be determined as follow (Judi, 1986):

$$U_S = \frac{1}{2\left(\frac{X_{gs}}{K_{gs}}\right)+\frac{X_{fg}}{K_{fg}}} \tag{14}$$

Storage temperature: Storage temperature within Δt period can be calculated by:

$$T_{sn} = T_s + \frac{\Delta t}{(mC_p)_s} Q_u - L - (UA)_s\,(T_s - T_a) \tag{15}$$

where, T_{sn} is the new storage temperature and L is the heating load in above equation represents the summation of energy losses of each part of the room.

Solar collector configuration: The solar collector configuration which was used in the prediction of the thermal analysis in this paper is of the type of the plate collector with (1 m) width and (2 m) in length. The collector consists of six copper pipes with diameter of (1 cm) and the pitch step (the distance between the centers of two narrow tubes is about (0.12 m). The tubes covered by galvanized steel sheet (0.001 m) thickness and a glass cover (0.003 m) thickness. A thickness (0.1 m) glass wool insulated the collector from bottom and the edges with thickness about (0.05 m).

RESULTS AND DISCUSSION

The main focus of this study is to find the mass of water to be used as a thermal storage medium that provide the required heating load for maintaining the comfortable room temperature in the night. Figure 2 shows the room heating load changing during day as shaded horizontally and the amounts of the useful solar energy which are collected by the solar collectors at different proposed absorber areas. It is clear that as the collector area increases, the useful solar energy increases, too. The gain reaches its maximum value after midday because of the high intensity of the solar radiation. It then returns to reduce towards the evening hours due to the decreasing of the intensity of solar radiation and vanishes during the night hours. The amount of the collected useful energy at a collector of 8 m^2 area which is shaded by the vertical rectangles, is equivalent to the calculated heat load to maintain the room at 18°C during the night hours.

Figure 3 and 4 shows changing of the thermal reservoir temperature using solar collector with area from 6-12 m^2 at constant room temperature when the mass of storage water is 50 kg m^{-2} of

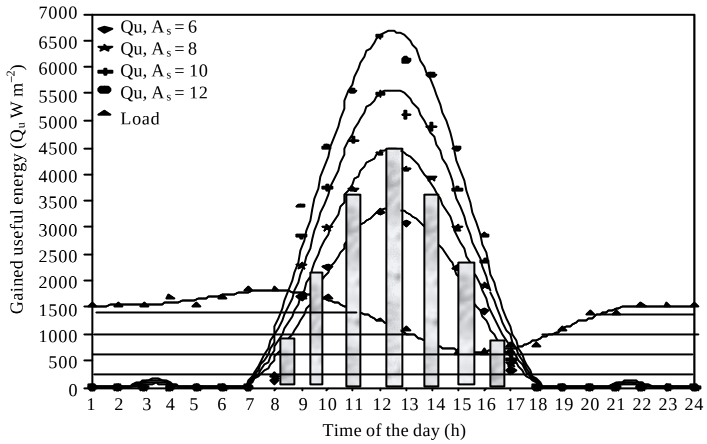

Fig. 2: Variation in the useful energy for the solar collector via heat load during day light

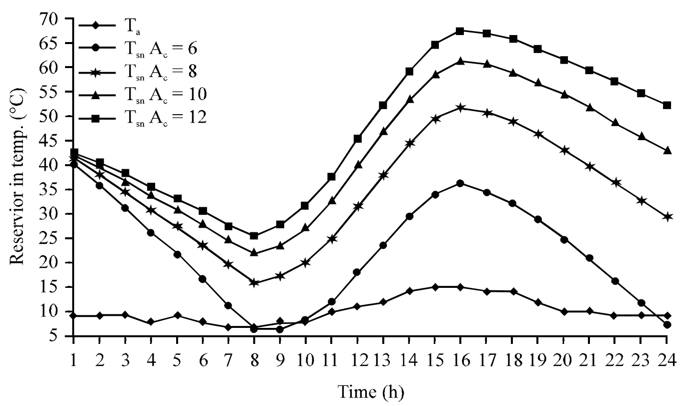

Fig. 3: Variation in the storage temperature for the reservoir via ambient temperature during daylight

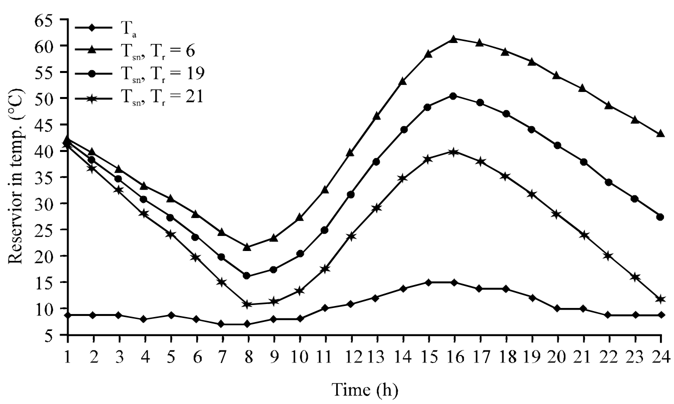

Fig. 4: Variation in the storage temperature for the reservoir during daylight at 50 kg m^{-2}

the solar collector area. The effect of the thermal storage is clear in increasing the temperature of storage water and moving the maximum point towards the evening hours. Also, the

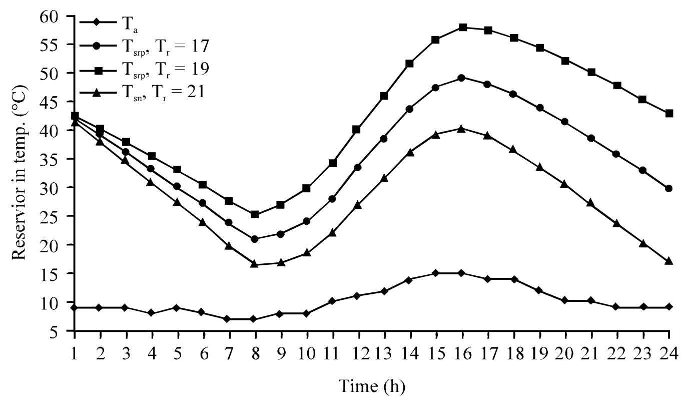

Fig. 5: Variation in the storage temperature for the reservoir of 60 kg m^{-2} during daylight

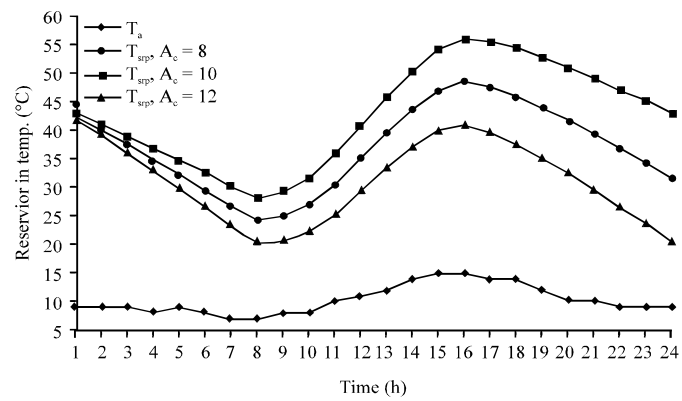

Fig. 6: Variation in storage temperature for the reservoir of 70 kg m^{-2} during daylight

value crowd in the same direction by increasing the solar collector area which means there is ability to have an acceptable heating all day hours when the room temperature is 17-19°C.

The obtained results are agreed with the TRNSYS simulation results reported by Hobbi and Siddiqui (2009). They mentioned that the optimum values of the design parameters at 6 m^2 collector area is providing 60°C storage temperature and 17°C room temperature with 55-60 L m^{-2} volume of storage water per unit collector area.

Another simulation has been carried out by Sweet and McLeskey Jr. (2012). Their results showed that the temperature of storage water between 51-58°C while the room temperature was between 19-20°C. Comparison summary of the three works is presented in Table 1. The three simulations are consistence with small margin of difference due to the difference in the volume of the water storage, ambient conditions and the absorbers areas.

Similar behavior could be noticed if the storage capacity changed to 60 and 70 kg m^{-2} as shown in Fig. 5 and 6, respectively. These Fig. 5 and 6 show that increasing the mass of storage water increases the ability of heating for longer day hours to maintain the heating at acceptable conditions.

Figure 7-9 show the variation of the reservoir temperature at 8, 10 and 12 m^2 collector area, respectively, where the room temperature to be 18°C and the mass of the thermal reservoir is 70 kg m^{-2} which is the closest predicted design parameters to the conditions of the study area for

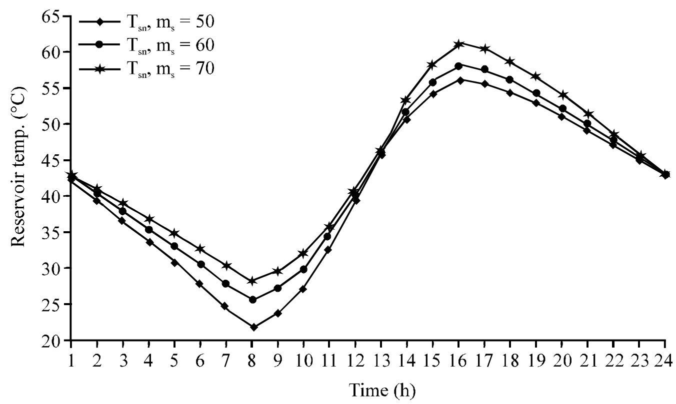

Fig. 7: Variation in storage temperature for the reservoir during daylight at A_c = 8 m^{-2}

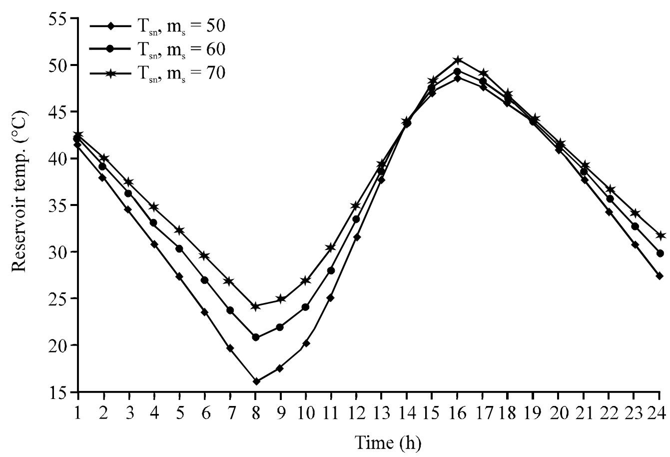

Fig. 8: Variation in storage temperature for the reservoir during daylight at A_c = 10 m^{-2}

Table 1: Comparison of results with previous researches

Procedure	Collector area	Storage temp. (°C)	Storage media	Room temp. (°C)	Remarks	References
Simulation by TRNSYS	6 m^2	60	Water	17	55-60 L m^{-2} volume of storage per unit collector area	Hobbi and Siddiqui (2009)
Simulation by TRNSYS	38.871 m^2 for floor area of 800 ft^2 99.17 m^2 for floor area of 2400 ft^2	51-58	Sand bed	19-20	11.356 L min^{-1}. The collector area increases as the house area increases	Sweet and McLeskey Jr. (2012)
Mathematical modeling	6 m^2 8 m^2 10 m^2 12 m^2	37 52 61 66.5	Water	17-21	At 50 kg m^{-2} storage capacity	Present work

increasing the intensity of the solar radiation during collecting period. Maximum heating can be achieved and maximum temperature for highest storage capacity to get longest heating period can be covered during day hours.

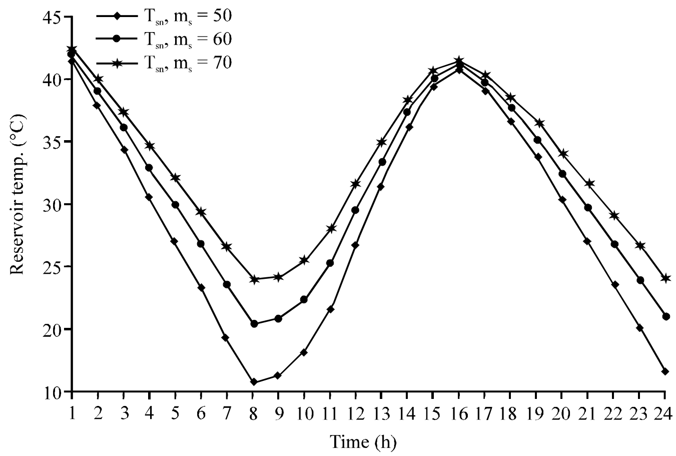

Fig. 9: Variation in the storage temperature for the reservoir during daylight at $A_c = 12$ m^{-2}

The results of the present study are in good agreement with the experimental and modeling analysis conducted and reported by Qin (1998). The result obtained is 50°C water storage temperature and 20°C room temperature at area of collector 4 m^2 to cover the heating load 1 kW m^{-2}.

CONCLUSION

An analytical model has been established and converted to a computer program to investigate the solar heating of a room space in Iraqi winter weather using solar collector integrated with water thermal storage tank. The study reveals that the optimum mass of the water in the thermal storage tank per meter square of solar collector is 70 kg m^{-2}. Water storage temperature is between 50-60°C and the area of collector is 8 m^2 to cover approximately 1.8 kW m^{-2} heating load. These parameters are able to sustain the room temperature at around 18°C over the night hours. The recent work can be applied to accomplish optimum design for house heating using water thermal storage integrated to a solar collector.

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NOMENCLATURE

A	=	Area (m^2)
N_g	=	Number of glasses covers for solar collector
C_p	=	Specific heat (kJ kg^{-1} °C^{-1})
Nu	=	Nusselt number
D	=	Diameter (m)
Pr	=	Prandtl number
F	=	Film heat transfer coefficient (W m^{-2} °C^{-1})
Q_u	=	Useful energy from solar Collector (W)
F_R	=	Heat rejection coefficient for solar collector
Re	=	Reynolds number

F^{-}	=	Solar collector efficiency coefficient
T	=	Temperature (°C)
G	=	Mass velocity (kg sec^{-1} m^{-2})
t	=	Absorber thickness (m)
h	=	Heat transfer coefficient $(<\text{W m}^{-2}\,{}^{\circ}\text{C}^{-1})$
Δt	=	Time difference (sec)
I	=	Solar radiation intense (W m^{-2})
U	=	Overall heat transfer coefficient (W m^{-2} °C^{-1})
k	=	Conduction heat transfer coefficient (W m^{-2} °C^{-1})
V	=	Velocity (m sec^{-1})
L	=	Heat load (W)
W	=	Half distance between solar collector tubes (m)
L_c	=	Solar collector length (m)
X	=	Thickness (m)
M	=	Mass (kg)
Z	=	Dimensionless factor for fin efficiency
$\dot{m}$	=	Mass flow rate (kg sec^{-1})
a	=	Ambient
m	=	Mean
bc	=	Lower surface of solar collector
o	=	Outer
c	=	Collector
s	=	Storage
ec	=	Solar collector edge
tc	=	Upper surface of solar collector
fg	=	Fiber glass
w	=	Water
gs	=	Aluminum
W	=	Wind
I	=	Inside
ε_c	=	Emission for solar collector absorber
ε_g	=	Penetration of glass cover
σ	=	Emission for solar collector cover
τ	=	Absorption for absorptive surface
α_g	=	Stefan-Boltzmann constant
η_f	=	Fin efficiency

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