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Energy Efficiency of Seasonal Solar Thermal Energy Storage System for Greenhouse Heating

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ABSTRACT

Seasonal thermal energy storage (STES) is widely researched because it utilizes excess energy that would be wasted otherwise. The purpose of this study is to analyze the energy efficiency of seasonal solar thermal energy systems as heating systems for greenhouses and to compare it with conventional variable air volume (VAV) heating systems. A greenhouse was chosen as a simulation model, because it requires constant and stable heating through the winter to extend the growing season and also because one can easily find adequate area to install solar collectors and heat storage tanks in the district for greenhouses. When STES is used in greenhouse buildings to control the temperature, it is expected to perform at its full capacity, because greenhouses only need heating, and a large amount of heating is needed. The proposed seasonal solar thermal energy storage system consists of a solar thermal collector, fully mixed heat storage tank, and VAV heating system. Energy simulation was conducted in two steps: heat storing throughout the year and heating in the winter. 125 greenhouses with area of 32 m^2 each, 125 solar thermal collectors of 10 m^2 each, and heat storage tank of 2000 m^3 was designed. TRNSYS 18 and an engineering equation solver were implemented for simulation and calculation of the system's thermal data. Simulation results showed STES heating contributing to 29% of the total heating load.

1. INTRODUCTION

Thermal energy collected during the summer is dissipated, because the summer has a higher solar thermal collecting rate and lower heating load than the winter. However, the winter lacks thermal energy to supply heating load. In this situation, seasonal thermal energy storage (STES) has its benefits in utilizing excess energy that is wasted in the summer. By simply storing heat in water tanks, collected heat can be used in the winter.

Storing solar thermal energy has been widely researched, especially for storing midday's ample thermal energy. By storing midday's solar thermal energy to support nighttime's heating load, otherwise-wasted midday's excess heat can be put to use. The diurnal offset is known to be relatively easy to compensate with water tanks (Khalifa *et al*., 2009). However, installing small-sized tanks to residential buildings shows lower energy efficiencies because of the storage tank's characteristics of thermal loss. The storage tank has its thermal loss coefficient according to the surface area to volume ratio, and it should be at least 2000 m^3 to be used effectively as a thermal storage tank (Braun *et al*., 1980).

Therefore, a thermal storage system must be integrated with large-scale heating, ventilating, and air conditioning (HVAC) systems. A STES system can be a more effective way to use excess energy than day-to-day energy storage, because STES has bigger and, thus, less-thermal-loss-inducing storage tanks. Sillman *et al*. (1981) concluded that the performance of STES systems becomes more beneficial as the storage size increases to the point of unconstrained operation; further, STES systems may cost the same or less per unit heat delivered than overnight storage systems that contribute to half of STES's heating load.

There are many methods to utilize stored heat. Using heat collected during the summer to charge sorption or desiccant material to cool and dehumidify buildings is a way to use excess summer production and is gaining more attention

(Pinel *et al*., 2011). This study mainly focuses on reliably storing and using the heat of a STES and on whether STES performance varies over years of operation. A greenhouse was chosen as the simulation model to verify the STES's reliability, because a greenhouse only has heating load, and it needs to be heated steadily throughout the year. Validating the reliability and economic benefits of a STES can lead to a more complex system with such components as sorption or desiccant materials. A venlo-type greenhouse that has a gable roof and is made mainly of 6-mm-thick glass was chosen because of its high thermal efficiency

2. SYSTEM SCHEMATICS

As shown in Figure 1, heat collected by a solar thermal collector is transferred to a thermal storage tank. When it is heating season, heat is transferred from a storage tank to a variable air volume (VAV) system via a water-to-air heat exchanger. The storage tank's temperature was evaluated for a 1-h time step. Stored heat in the tank is expected to be depleted during the winter. Therefore, an auxiliary boiler raises the air temperature after it receives heat from the heat exchanger. Supply air (SA) mainly supplies required minimum ventilation, but if more heat needs to be supplied, the volume of SA increases to meet the need.

Ethylene glycol (EG) is chosen as the brine for the solar collector, because the storage tank's temperature is expected to rise to 90–95˚C. To make the temperature more than 90˚C after going through the heat exchanger, fluid that provides heat needs to be at least 95-100˚C, and water temperature should not be more than 100˚C. Therefore, usable EG's highest concentration rate of 60% is chosen, and its boiling point is 111˚C. Furthermore, the specific heat of EG is evaluated according to its function of temperature (Melinder, 2010). As a result, by using EG as the brine for the solar collector, a solar thermal collector can produce enough heat to make a 20˚C tank 90˚C before the heating season.
The sizing of components is related to the weather condition, design load, thermal loss, etc. In this study, sizing was done through a trial and error method to meet the largest solar fraction of total load. According to Braun *et al*. (1980), the optimal ratio of storage volume to collector area is approximately 1.5. In this study, the solar collector was designed to be 10 m^2 and the storage tank 16 m^3. However, because total tank volume must be at least 2000 m^3 to minimize the tank loss (Guadalfajara *et al*., 2014), tank loss was calculated using a model of collective tank size of 125 small tanks (16 m^3 *125 each).

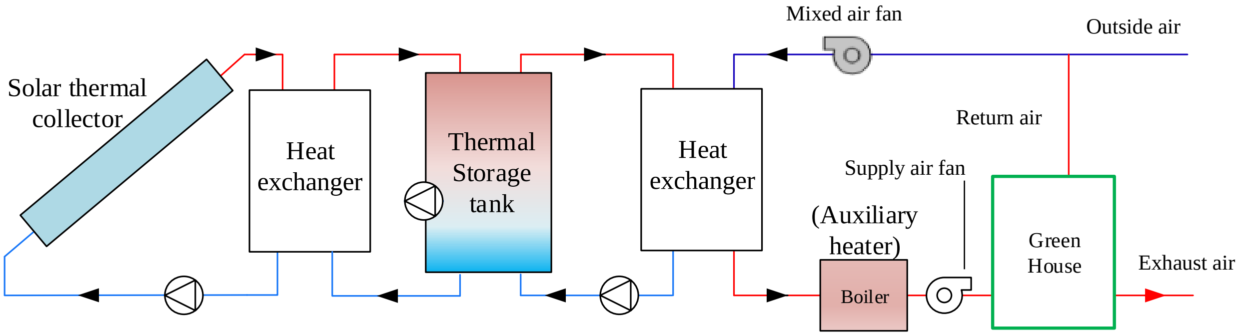

Figure 1: System schematics

3. SIMULATION MODEL OVERVIEW

3.1 Model Building

The sensible and latent loads of the designed greenhouse building were taken from building energy simulation software (i.e., TRNSYS 18). The designed greenhouse is located at Taean, South Korea. Taean has a district of greenhouses, and the simulated design load was validated by comparison with an existing study about greenhouse heating and ventilation load in Korea. For 125 greenhouses, each greenhouse has 4-m width, 8-m length, and 5.5-m height. All walls and roof are 6-mm-thick glass. Korea's greenhouse-grown vegetables' living temperature ranges from 15–25˚C; thus, the room temperature was set at 20˚C throughout the year.

Table 1: Specifications of designed greenhouse

Description		Value
Dimension		4 m (W) × 8 m (L) × 5.5 m (H)
Volume		176 m^3
Capacitance		211.2 kJ/K
Glass	Thickness	6 mm
	Conductivity	5.69 W/m^2 K
	g-value	0.855

3.2 Solar Collector Model

A compound parabolic collector (CPC) was chosen to provide better solar collector performance but less dissipating collected heat to the ambient air. The solar collector was controlled according to whether it can generate heat and whether the collector fluid had a higher temperature than the tank fluid. Typical CPC characteristics from the research of Kalogirou (2004) were used in the simulation. The total useful energy produced by the solar thermal collector were calculated using Equation (1). Each calculation was done in a 1-h time step with hourly weather conditions. If there was useful energy gained and if the collector fluid's temperature was higher than the storage tank fluid's temperature, then the collector fluid gave heat to the tank fluid via a heat exchanger. The solar collector fluid's temperature change is depicted in Equation (2). The heat exchanger had a typical efficiency of 0.7. Both fluids' mass flow rate was 0.005 kg/s. This mass flow rate was the optimum mass flow rate for evacuated tube collectors (Eldighidy *et al.*, 1983). After exchanging heat, the new fluid temperature that flows again into the collector was calculated using Equation (3). Likewise, the tank fluid's temperature was derived using Equation (4). In these cases, because the mass flow rates of both fluids were the same and EG always has a lower specific heat than water, $(\dot{m}Cp)_{EG}$ is the minimum heat capacity.

$$Q_u = A_c F_R \left[G_T(\tau\alpha)_{av} - U_L(T_i - T_a)\right]^+ \tag{1}$$

$$T_{after} = \frac{Q_u}{\dot{m}Cp} + T_{before} \tag{2}$$

$$\varepsilon = \frac{(\dot{m}Cp)_{EG}\left(T_{collector,out} - T_{collector,in}\right)}{(\dot{m}Cp)_{min}\left(T_{collector,out} - T_{tank,out}\right)} \tag{3}$$

$$\varepsilon = \frac{(\dot{m}Cp)_{water}\left(T_{tank,out} - T_{tank,in}\right)}{(\dot{m}Cp)_{min}\left(T_{collector,out} - T_{tank,out}\right)} \tag{4}$$

3.2 Storage Tank Model

A fully mixed tank was designed to serve as the STES tank. There are many solutions that can be used as a storage medium, but, in this study, the storage solution was water, because water has the most reasonable cost, is easy to implement, and has a relatively high specific heat (Socaciu *et al.*, 2011).

The tank was designed to be buried underground. Thermal loss to the ground was considered throughout the year according to the research of Florides *et al.* (2004). Equation (6) was used to get the ground temperature of the depth to which the STES tank was set. For 2000 m^3 of STES tank, it was buried below 10 m from ground level. The U-value was estimated to be 11.1 W/m^2 K

Tank performance was mainly evaluated through Equation (7). The tank's energy loss to the ground was considered, and energy loss to the load was considered by exchanging heat to the water-to-air heat exchanger. The water mass flow rate was 0.005 kg/s and stayed the same. The air flow rate changed according to the VAV's mode of operation. The temperature of the tank fluid after heating air is given by Equation (8). T_{MA} is the temperature before receiving heat from the tank. After the heat exchange between air and tank water, the air is termed, T_{PA}.

$$T_S = T_{\text{mean}} - T_{amp} \exp\left(-z\sqrt{\frac{\pi}{365a}}\right) \cos\left(\frac{2\pi}{365}\left[t_{year} - t_{shift} - \frac{Z}{2}\sqrt{\frac{365}{\pi a}}\right]\right) \tag{6}$$

$$T_s^+ = T_s + \frac{\Delta t}{\left(mC_p\right)_S}\{L - (UA)_S(T_S - T_a)\} \tag{7}$$

$$\varepsilon = \frac{(\dot{m}Cp)_{water}\left(T_{tank,out} - T_{tank,in}\right)}{(\dot{m}Cp)_{air}\left(T_{tank,out} - T_{PA}\right)} \tag{8}$$

After receiving/giving heat from and to the components, the tank's temperature changed. Mainly, the tank was considered full at the start, and tank's temperature was 20°C at the start of the simulation. Equation (9) describes the mass balance of the tank's temperature after heat exchange.

$$\dot{m}_{full}T_{tank} = \dot{m}_{hourly\ mass} \times T_{tank,after\ exchange} + (\dot{m}_{full} - \dot{m}_{hourly\ mass}) \times T_{tank,before\ exchange} \tag{9}$$

3.3 VAV Model

The greenhouse's minimum required ventilation rate was used from an existing study about greenhouses' ventilation system design parameters (Gates *et al.*, 1999). The minimum ventilation rate was zone volume per 1 h. Equation (10) was used to calculate minimum ventilation rate. If the greenhouse needs more heat than what $\dot{m}_{\mathrm{oz}}$ can deliver, SA increases. Otherwise, $\dot{m}_{\mathrm{oz}}$ is always supplied. The increment of air mass is given from return air (RA) from the greenhouse. Therefore, mixed air (MA) was formed, and the MA exchanged heat with the tank fluid and became processed air (PA). This procedure is shown in Equation (11).

The room temperature of the greenhouse was 20°C, which most common in Korea's greenhouse plants. To maintain 20°C, the SA is set to 50°C. When PA's temperature was not high enough, an auxiliary heater operated and set the temperature to 50°C — see Equation (12).

$$\dot{m}_{\mathrm{oz}} = \dot{m}_{zone} \,/\, 3600 \tag{10}$$

$$\varepsilon = \frac{(\dot{m}Cp)_{air}(T_{PA} - T_{MA})}{(\dot{m}Cp)_{air}\left(T_{tank,out} - T_{MA}\right)} \tag{11}$$

$$\dot{Q}_{\mathrm{aux}} = (\dot{m}Cp)_{air}(T_{SA} - T_{PA}) \tag{12}$$

4. SIMULATION RESULTS AND DISCUSSION

Figure 2 shows tank temperature change for a year. The tank temperature was initially 20˚C, but over the time of summer season, because of the high intensity of solar radiation of summer, the tank temperature increased to 82.7˚C. The velocity of the temperature increase decreased as thermal loss to the ambient air increased. Although the tank's storage medium, water, can be heated up to 100˚C, simulation without load to VAV showed that tank temperature setting reached an equilibrium (unable to produce more useful energy) at 87.2˚C. Methods to heighten the equilibrium temperature are adjusting solar collector size and tank solution medium. However, differentiating the size of the components may result in storing less heat, and change of the solution to other brines can cause less heat to be stored and a higher cost for the solution.

In the winter, solar thermal energy was continuously produced, but at a lower rate. Korea's winter solar radiation is higher than that of most northern-latitude countries, but, because the solar zenith angle in the winter is larger than in the summer, the solar thermal energy production rate is lower. Beginning to transfer heat to the VAV system to meet the heat load made the tank's temperature drop drastically. By the end of the winter, when the heating load reached 0, the tank temperature stabilized at 20˚C, which was the starting temperature of the simulation. Furthermore, 20˚C was close to the underground temperature where tank was buried, so the tank no longer lost heat to the ground, causing the tank to maintain its temperature.

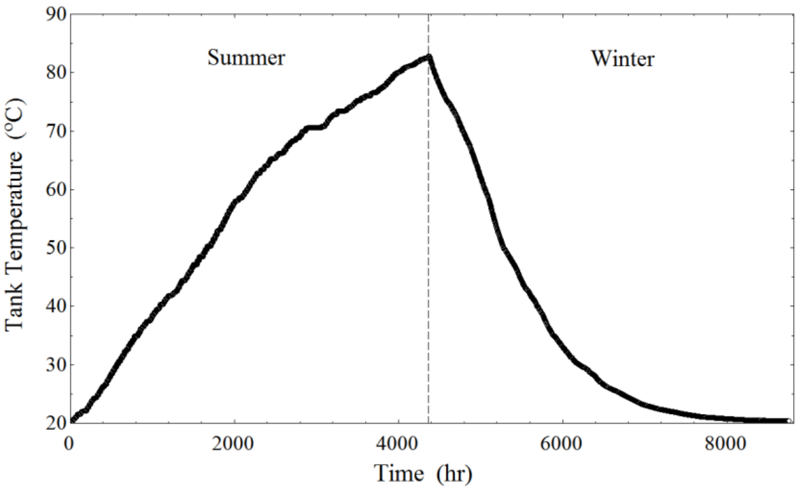

Figure 2: Tank temperature

Figure 3 depicts total sensible heating load of the greenhouse and amount of heat supplied by the auxiliary boiler. Therefore, the black lines that are not covered with orange lines are the amount of heat that the STES provided to heat the SA of the VAV system. The fraction of the STES for total energy consumption was 29%. Total energy needed to meet the heating load for the greenhouses was reduced to 71% by implementing the STES system.

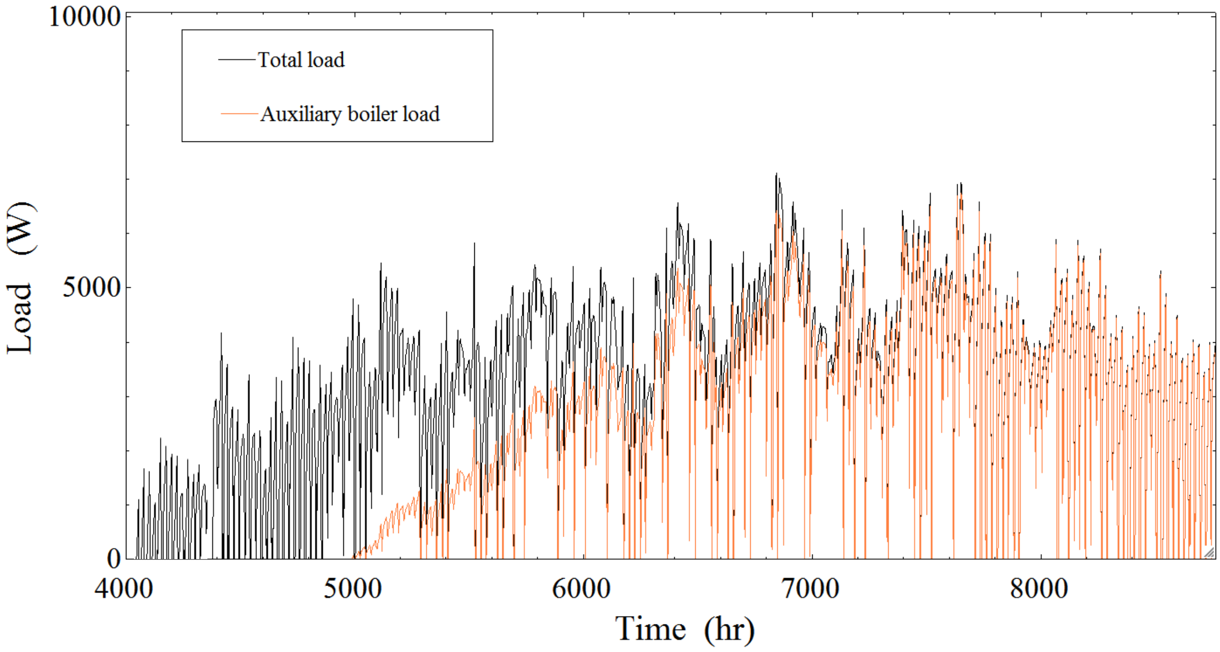

Figure 3: Design load and load of auxiliary boiler

To compare the amount of energy saved by the STES system, the additional energy used by the STES when using three more pumps was calculated. The pump power was estimated from the water flow rate (Q), density (ρ), head (H), gravitational acceleration (g), and pump efficiency (η) using Equation (19). For variable volume pumps, the affinity law of pumps was applied to calculate the actual pump power, as in Equation (13). The reference values for the head (20 m) and pump efficiency (60%) were obtained from EnergyPlus (EnergyPlus, 2013).

$$P_{pump} = \rho g V H / 1000\,\eta \tag{13}$$

The overall energy consumption converted into primary energy is shown in Table 2. Used local primary energy factors were 2.75 for electricity and 1.1 for the natural gas boiler.

Table 2: Overall energy savings

Description	Value
Saved boiler energy consumption [kW]	5,016
STES pump energy consumption [kW]	962
Total saved energy consumption [kW]	4,054

Therefore, by installing the STES system, 4054 kW was saved yearly. Compared with the VAV system, which has a total primary energy consumption of 23,500 kW, approximately 17% was saved.

5. CONCLUSIONS

By installing a storage tank with a solar thermal collector, thus forming an STES system, a heating system was able to utilize excess thermal energy from the summer. A 2000-m^3 storage tank was filled with water heated from 20˚C to 82˚C during the summer. Stored heat depleted by the middle of the winter and tank temperature decreased to 20˚C at the start of next summer. Of the total heating load, 29% was supplied by the STES system. This resulted in savings in primary energy consumption of approximately 17%.

Compared with a conventional VAV heating system for greenhouses, implementing the STES system resulted in energy savings. In addition, adding a storage tank to existing solar collectors can lead to better energy efficiency. The storage tank makes utilizing excess energy from the summer possible to increase the energy usage and overall efficiency of the heating system.

NOMENCLATURE

A_c	Area of solar collector	(m^2)
a	Thermal diffusivity of ground	(–)
C_p	Specific heat capacity	(kJ/kg·°C)
ε	Heat exchanger efficiency	(–)
F_R	Collector heat removal factor	(–)
L	Greenhouse load	(W)
$\dot{m}_{oz}$	Minimum ventilation rate	(kg/s)
Q_{aux}	Auxiliary boiler load	(W)
Q_u	Useful energy	(W)
t_{year}	Current time (day)	(–)
t_{shift}	Day of minimum surface temperature	(–)
T_a	Ambient air temperature	(K)
T_{amp}	Amplitude of surface temperature	(K)
T_{mean}	Mean surface temperature	(K)
T_S	Storage tank temperature	(K)
$(UA)_S$	Tank heat transfer coefficient	(W/m^2·K)
U_L	Collector heat transfer coefficient	(W/m^2·K)
$(\tau\alpha)_{av}$	Effective transmittance-absorptance	(–)
Z	Depth below the surface	(m)

Subscript

collector	Solar thermal collector
EG	Ethylene glycol
PA	Processed air
MA	Mixed air
SA	Supply air
STES	Seasonal thermal energy storage
tank	Storage tank
VAV	Variable air volume system

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