Local Optimization of Thermal Storage for Greenhouses: Reduction of Energy Input and Improvement of Inner Climate

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Abstract

In temperate regions, such as the Mediterranean basin, there is a diurnal excess of energy nearly every day of the year, which is usually dissipated through natural ventilation. However, since suboptimal nighttime temperatures limit productivity of unheated greenhouses for several months a year, extracting the daytime excess energy and reusing it to heat the greenhouse during the night would increase productivity, or at least reduce energy consumption of greenhouses that are heated. This heat extraction would have the additional advantage of reducing ventilation requirement thereby increasing the scope for carbon dioxide fertilization. To achieve this, the performance of the greenhouse as a solar collector has to be maximized by an efficient heat exchanging and heat buffering system. The aim of this research was to define the optimum combination of heat exchange rate, maximum water flow rate of a heat storage buffer and buffer capacity in a commercial greenhouse in the Mediterranean region (Sicily, Italy, 37 °N), the cost function being represented by the dose (duration × intensity) of low temperature events. The greenhouse temperature was calculated through a previously validated greenhouse climate simulation model, applied to one-year of real local data. The effect of the buffer on the cost function was first calculated for a range of heat exchange values followed by a cost function evaluation for nodes of a pre-selected grid, each node representing a value-pair for the other two buffer defining parameters. In this paper we analyze the trend of the cost function with respect to each parameter of the buffer and how this is affected by the preset tolerance of low temperatures. Furthermore, we discuss a simple method to find an "optimal" configuration of the buffer. Finally, a combination of 3000 m^3 ha^{-1} buffer capacity, 45 m^3 h^{-1} ha^{-1} maximum water flow rate and an overall heat transfer coefficient of 5 W m^{-2} K^{-1} is selected (heat transfer coefficient is defined per m^2 greenhouse floor area).

INTRODUCTION

High radiation and moderate temperatures in autumn and spring have helped the expansion of protected horticulture in the Mediterranean region during the last decades. In Almería, maximum temperatures inside the greenhouse are well above 30°C most part of the year (Gázquez et al., 2008), in other words a diurnal excess of energy exists. During winter periods with relatively low radiation and lower ambient temperatures, a major fraction of this daytime surplus energy is ventilated away during dehumidification. At night an energy shortage occurs and large differences between the day and night inner climate result.

Variations in temperature and humidity occurring during a 24-hour period in an unheated Mediterranean greenhouse are the main reason for plant stress. Suboptimal nighttime temperatures limit productivity of unheated greenhouses for several months a year. Extracting the daytime excess energy and reusing it to heat the greenhouse during the night can be a valid strategy to increase productivity by handling peak heat and cold demands (Vox et al., 2008; Zaragoza et al., 2008). Moreover, it could reduce energy consumption of heated greenhouses. The additional advantage of such a strategy would be the reduction of ventilation requirement thereby increasing the scope of CO_2 enrichment

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of the greenhouse air and therefore increasing the photosynthetic activity of the plants and hence crop production (Marcelis et al., 1998; Opdam et al., 2005). To achieve this, the performance of the greenhouse as a solar collector has to be maximized by an efficient heat exchanging and heat buffering system. Excess solar energy can be collected and conserved in a heat storage buffer during the day and used to balance the energy demand during the night (Fig. 1). However, since both heat demand and supply of a greenhouse can vary strongly, research on the optimum balance between the two is necessary.

The aim of this research was to define the optimum combination of capacity, heat exchange and maximum water flow rate of a heat storage buffer applied to a passive greenhouse in the Mediterranean region during the winter. However, the benefits in terms of improved production were quite difficult to quantify due to the complexity of the biological processes involved (De Zwart, 2008), especially at suboptimal temperature conditions. Since a fully economic analysis would have been hindered by uncertainty about yield response to extreme temperatures, the cost function had to be represented by the dose (duration × intensity) of extreme temperature events. Based on this cost function, the "optimum" combination of the defining buffer parameters was defined primarily for the cold months of the year. Therefore, as extreme temperature events only the low winter temperatures were considered. Apart from minimizing the cost function, the "optimum" configuration of the buffer depended also on the costs of the static parameters determining the fixed costs of the equipment used. Consequently, configurations where expected marginal gain cannot offset the marginal costs were rejected.

To define the "optimum" combination of the energy storage system defining parameters in minimizing the cost function, several concepts for such a system were developed by using a range of values for each of the three storage buffer defining parameters. Those concepts were subjected to an in depth simulation analysis with GTa-tools (Van 't Ooster, 2010), a greenhouse climate model.

MATERIALS AND METHODS

Model Description

GTa-tools (2010) is a Visual Basic based code of open structure extended with class libraries, calculating the greenhouse climate based on the outside conditions, the greenhouse condition targets set by the user, and the technical installation of the greenhouse. Air temperature, carbon dioxide concentration and humidity are the most important greenhouse climate state variables calculated. Furthermore, it calculates hourly greenhouse requirements in heating, cooling, dehumidification and ventilation and absolute resource use (gas, electricity, CO_2 etc).

A submodel combining a water distribution circuit connected to a heat storage facility with unheated (passive) water was created and integrated to the main code of the simulator. The thermal storage buffer was modeled to have a cold and a warm section. Both sections were modeled as ideally mixed and physically perfectly separated. Thermal energy transfer occurred between the storage buffer and the ambient air. No temperature steps have been modeled in the pipe circuit connecting the storage buffer with the heating pipes, meaning that the water was not passing through heat exchangers to separate buffer water from circuit water. The mode in which the system operated (heating or cooling) was based on the simulated greenhouse air temperature in the passive mode, in order to prevent flip/flop actions between the iterations. The upper threshold above which the system operated in cooling mode and the lower threshold below which the system operated in heating mode were set by the user.

Model Input

Technical construction and crop data from a 14-span multi-tunnel of Azienda Fratelli Dezio, a commercial grower in Italy (loc. Gaspanella, Ragusa, Italy, 36°57'N; 14°26'E; 104 m a.s.l.) were used as initial model settings. Based on those data, the modeled greenhouse, had 14 modules 8 × 120 m each, oriented SE-NW. The eaves and the ridge height were 4 and 5.6 m respectively. Each module had one longitudinal roof ventilator facing NE. There were no side windows. Double inflated plastic polyethylene film was used as cover, with a light transmissivity roughly estimated close to 0.6. The greenhouse was ventilated, in order to control humidity. The leakage ventilation rate was roughly estimated to be 0.4 h^{-1}. The cultivar was cherry tomato ‘Shiren’, planted on September 18th with a density of 3.2 pl. m^{-2}.

The weather data set used had local outdoor data of solar radiation, temperature, humidity, wind speed and direction, recorded over a period of a year. The every ten minutes logged data were converted into hourly averages to make the set compatible to the simulator. Additional data used as a model input are given in Table 1.

The initial fill status of the heat buffer at the start of September was set to 0.8 and the water temperatures in the warm and cold compartment to 25 and 15°C respectively. The heat storage insulation layer depth was 0.15 m and its thermal conductivity was 0.04 W m^{-2} K^{-1}. A range of values of each of the defining buffer parameters was used to define several system concepts to be subjected to simulation.

Development of System Concepts

The initial buffer capacity (V_b) and the maximum water flow rate (φ_w) were kept fixed, set to infinite volume and to 60 m^3 h^{-1} respectively. The heating (T_{heat}) and cooling (T_{cool}) temperature thresholds were set to 12 and 27°C respectively. The first system concepts were developed by using a distribution of values of the heat transfer coefficient defined per unit of greenhouse floor area (α_{grh}). This distribution of α_{grh} values was generated by shifting the values of the pipe area (A_{pipe}) in Equation 1. The heat transfer coefficient of the pipes (α_{pipe}) and the greenhouse floor area (A_f) values of the same equation were kept constant.

$$\alpha_{grh} = \alpha_{pipe} \cdot \frac{A_{pipe}}{Af} \qquad \text{W m}^{-2}\,\text{K}^{-1} \qquad (1)$$

A distribution of values of T_{cool} was used to develop the second set of system concepts with the V_b fixed to 6000 m^{-3} ha^{-1}. The α_{grh} was set to 5 W m^{-2} K^{-1} (result of the first parameter optimization). The last set of system concepts was developed for a range of values for φ_w and V_b. The combination of those parameter values resulted in an optimization grid. Each node of this grid represented a different system concept. The T_{cool} was this time set to 16°C (result of the second parameter optimization).

Some of the parameter values used for system concepts were extreme and not consistent with reality. However, they were accepted for the clarity of the calculation results. Finally, all system concepts were subjected to an in depth simulation analysis and their results were compared in terms of degree hours (duration × intensity) below 12°C. They were also compared to a simulated passive greenhouse of the same area, ventilating during the cold months of the year for dehumidification only.

RESULTS AND DISCUSSION

Increasing value of α_{grh} led to decreased cost function (Fig. 2). However, above 5 W m^{-2} K^{-1} hardly any further reduction of the cost function in degree hours (Dh) was observed. For α_{grh} values above 10 W m^{-2} K^{-1}, the pipe return and greenhouse air temperatures were almost equal (Fig. 3). Based on the Newton’s Law of Cooling, temperature difference between the heating medium (pipe) and the air led to energy transfer (H_{pipe}). Increased H_{pipe} led at constant water flow to a shorter pipe length to adjust pipe water temperature to greenhouse air temperature. At the point where the pipes and the air had the same temperature, no further energy transfer could occur. Therefore, an α_{grh} above 10 W m^{-2} K^{-1} would not improve heat transfer performance, especially when this increase results from a larger A_{pipe}/A_f (Eq. 1). Combined with the fact that at this point the cost function was almost zero (19.1 Dh), no further reduction was expected or required. However, the solution of 5 W m^{-2} K^{-1} was preferred, since it was unlikely that the small improvement of the cost function would yield enough increase in production to offset the difference in investment costs.

The increasing value of T_{cool} led to a higher pipe water temperature (Fig. 4), with a consequent higher thermal energy transfer to the greenhouse air at night in heating mode. The latter resulted in a decreased cost function (Fig. 5). However, switching the operation into cooling mode at low T_{cool} led to prolongation of the heat harvesting period which is crucial for the amount of heat harvested (Fig. 6) and the minimum fill status of the buffer (Fig. 7). The latter is the percentage water volume in the warm section of the buffer. It is important to mention that the earlier switch had hardly any significant effect on decreasing the daily average air temperature (20.5°C for T_{cool} = 22°C and 20.4°C for T_{cool} = 16°C). Although a later switch resulted in minimized cost function, an early switch (at 16°C) was preferred since it minimized the buffer capacity and therefore, the likely investment costs. When T_{cool} was 22°C (Fig. 4), water temperature fell below 15°C for 37 h, due to the fact that at the same period the buffer fill status reached 0% (Fig. 7).

Increasing φ_w and V_b led to a decreased cost function (Fig. 8). The combination of 100 m^3 h^{-1} and 10,000 m^3 resulted in a reduction of 81%, while the combination of 60 m^3 h^{-1} and 4000 m^3 resulted in a reduction of 69% compared to a passive greenhouse. The minimized cost function resulted by the use of V_b and φ_w was followed by a likely increase of the investment and operational costs of the equipment. The investment cost of a pressurized water tank with a capacity of 1700 m^3 was estimated to 21,500 € according to the KWIN report of 2008 (Vermeulen, 2008). Therefore, the intermediary solution of 4000 m^3 (or 3000 m^3 ha^{-1}) in combination with 60 m^3 h^{-1} (or 45 m^3 h^{-1} ha^{-1}) was preferred, since it was unlikely that the further small improvement of the cost function would yield enough increase in production to offset the difference in investment costs. Finally, a smaller capacity buffer was a better solution in terms of economy in space.

For the selected defining parameters of the "optimum" energy storage system and for three consequent nights in March (14-16), the energy transfer from the pipes to the air was relatively low. The reason was the small amount of hours the air temperature of the passive greenhouse fell below the heating threshold (12°C), activating the buffer in the heating mode (Fig. 9). On the other hand, the heat transfer to the pipes during the day was high since the air temperature reached values of even 8°C above the cooling threshold. That could also be derived from Figure 10, were the temperature differences between the supply and the return pipe water are higher during the day. The relatively lower thermal discharging process led to heat accumulation. This heat accumulation led to increased water temperatures, both in the cold and the warm compartment of the buffer (Fig. 11) and to an increased buffer fill status (Fig. 12). The lower day cooling effectiveness was a result of the condensation on the heating pipes (latent heat removal) and indirectly of the crop transpiration. Finally, the total water condensation on the pipes during the cold period which was calculated to 0.4 kg m^{-2}, accounted to a recovery of 7.2% of the crop transpiration water. That led to decreased relative humidity, thus, to decreased ventilation rate (Fig. 13).

CONCLUSIONS

In this research an "optimal" configuration of a thermal storage buffer was defined through a validated greenhouse climate simulation model. Simulations show that there is an obvious limit to the value of increasing the heat transfer coefficient per unit of greenhouse floor area, when the maximum amount of heat transfer from the pipes to the greenhouse air is reached. There is also, a less obvious but still clear limit to the value of the flow rate and the buffer capacity. For all the buffer related parameters, "optimal" configuration strongly depends on the likely increase of the net profit. This was not calculated as a result of uncertainty in crop production.

A relatively low 'heat harvesting' threshold is needed to bridge a period of five successive cold months. Even then, the effect on night heating is several times larger than the undesired daytime cooling. Moreover, the required capacity of the buffer is strongly related to the temperature threshold management and the minimum air temperatures of the area. A decrease of the upper temperature threshold by 6°C leads to a decrease of 30% of the maximum buffer capacity. A future more accurate crop model could be used to realize a cost-benefit analysis based on the net profit of the production. This would allow finding a real optimum for the system parameters.

ACKNOWLEDGMENTS

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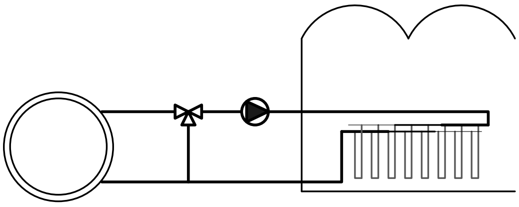
Tables

Table 1. Additional parameters used in GTa-tools (2009).

Parameter	Description	Value	Unit
$RH_{in,d}$	Desirable greenhouse air relative humidity	80	(%)
$RH_{in,max}$	Maximum greenhouse air relative humidity	94	(%)
$U_{d,n}$	Overall heat loss coefficient during day-night	3.5	(W m^{-2} K^{-1})
LAI_{max}	Maximum leaf area index	2.6	(m^{-2} m^{-2})
DW	Dry weight of the mature crop	1250	(g m^{-2})
A_{grh}	Greenhouse floor area	13440	(m^{-2})
A_r	Area of the roof ventilators opening	7383	(m^{-2})

Figures



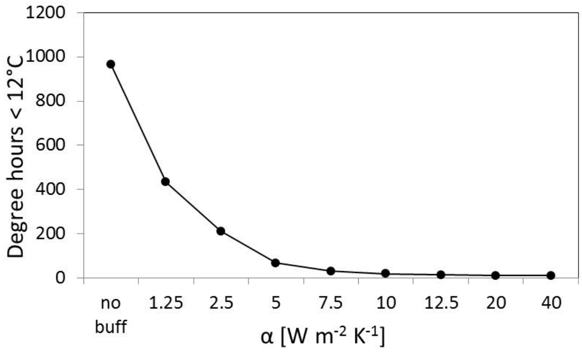
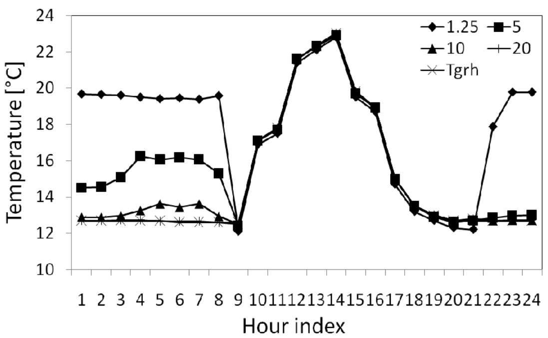


Fig. 1. Design of the thermal storage buffer connected directly to a heat exchanger in the greenhouse. A circulation pump is regulating the water flow rate.

Fig. 2. Dh below 12°C for 8 simulation runs with different α_{grh}.



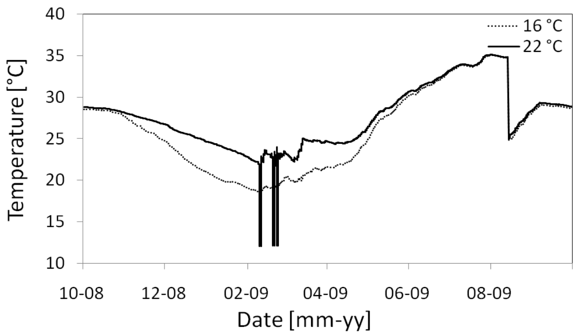
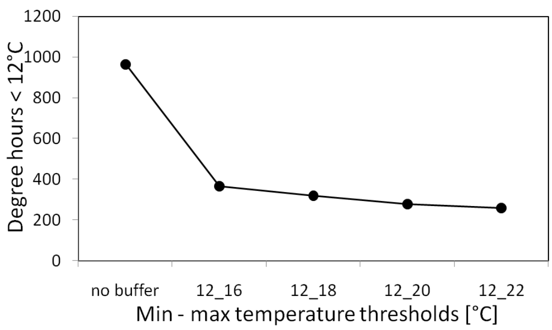


Fig. 3. Return temperature of the water in pipes for four simulation runs with different α_{grh}. Hourly values of 23 Feb. 2009.

Fig. 4. Water temperature of the warm compartment of the storage buffer, resulted for two simulation runs. In the first simulation T_{cool} is 16°C and in the second is 22°C.



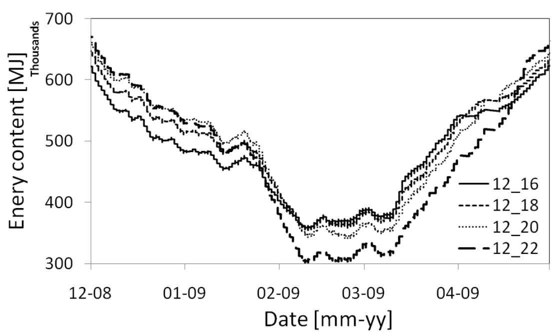
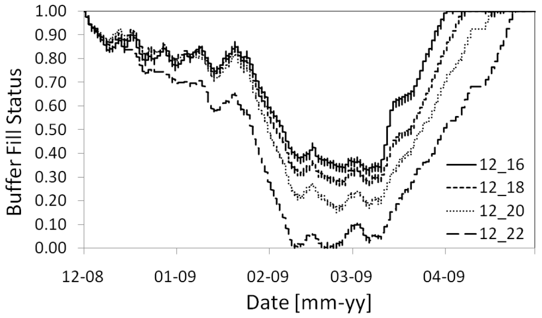


Fig. 5. Dh below 12°C for 4 simulation runs with different upper temperature thresholds.

Fig. 6. Buffer energy content for 4 simulation runs with variable upper temperature threshold (Winter period). The buffer capacity is 6000 m^{-3}. ($E_w = m \cdot c_w \cdot (T_w - T_0)$; $T_0 = 0°C$).



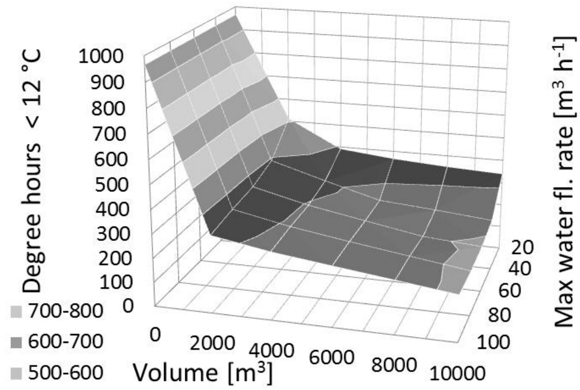
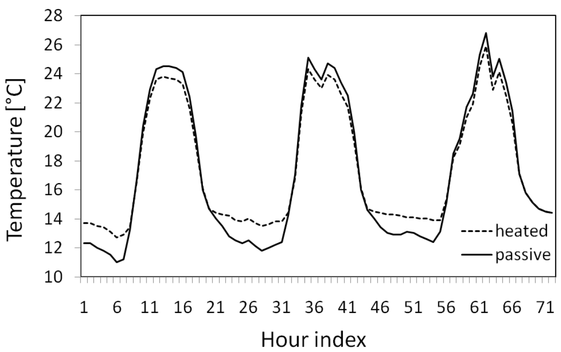


Fig. 7. Buffer fill status for 4 simulation runs with variable upper temperature thresholds. The buffer capacity is 6000 m^{-3}.

Fig. 8. Simulation results in Degree hours below 12°C for each node of the optimization grid.



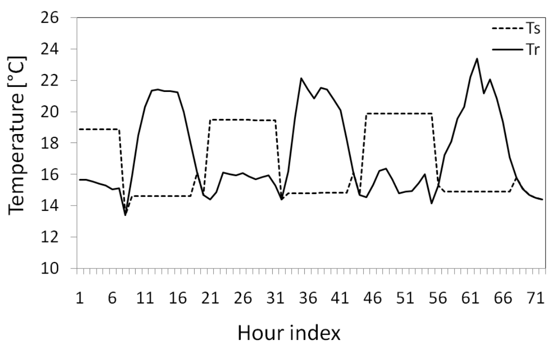
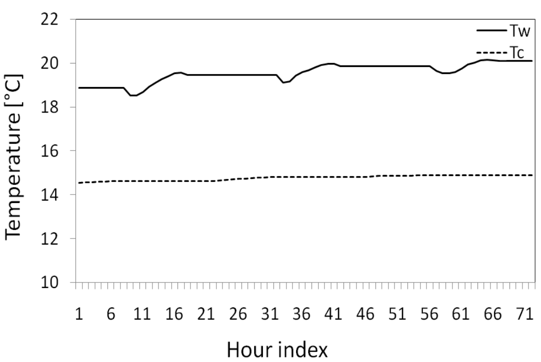


Fig. 9. Hourly air temperature of the passive greenhouse; of the greenhouse equipped with the thermal storage buffer (March, 14-16).

Fig. 10. Hourly temperatures of supply and return water in the pipe system (March, 14-16).



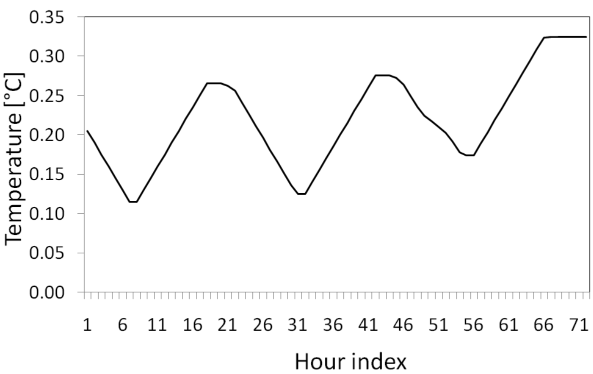


Fig. 11. Hourly temperatures of the water in the two compartments of the buffer (March, 14-16).

Fig. 12. Hourly buffer fill status (March, 14-16).

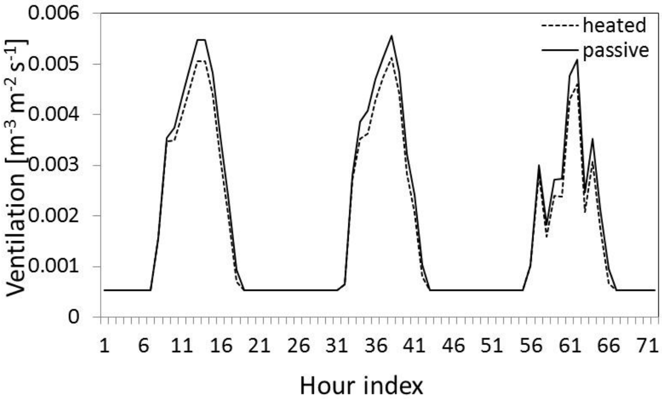


Fig. 13. Hourly ventilation rate of the passive greenhouse; of the greenhouse equipped with the thermal storage buffer (March, 14-16).