Dynamic Modeling of Microclimate and Environmental Control Strategies in a Greenhouse Coupled with a Heat Pump System

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Abstract

The purpose of this study was to develop and validate a dynamic simulation model to be employed in accurate prediction of microclimate in a greenhouse as a function of dynamic environmental factors. The model has options to evaluate the effects of location, time of the year, orientation, single and double polyethylene glazings, conventional and heat pump heating and cooling systems, open and confined greenhouse systems, CO_2 enrichment, variable shading, and the use of night curtains. Conventional gas furnace and evaporative cooling, respectively, provided heating and cooling in the conventional system. In the heat pump systems, gas-fired heat pump units provided both heating and cooling. The heat pump systems were operated both as an open and a completely confined system. Outputs of the simulation model included both temporal and vertical distribution of air, leaf, floor and cover temperatures, CO_2, relative humidity, solar radiation, and photosynthetically active radiation in addition to the dynamics of photosynthesis, respiration, transpiration, energy and CO_2 use and fixation. Comparison of experimental and predicted results showed that the compared microclimatological parameters were in fairly good agreement. The greenhouse model developed in this study is useful for ecologists, plant scientists, and engineers to evaluate individual or combined effects of various forcing functions on the enclosed environment and plant responses; and to develop control strategies for different parameters.

INTRODUCTION

Accurate prediction models for greenhouse and plant growth performance can be used as a design tool and in economic feasibility analyses as well. A dynamic analysis is required for more accurate prediction and control of greenhouse thermal environments. In addition to experimental tests, efforts have been made to predict the greenhouse environment under both steady state and transient conditions. Some reported work on greenhouse models and thermal performance tests include the work of Chandra et al. (1981), Glaub and Trezek (1981), Kindelan (1980), Navas et al. (1998), Pita and Vargues (1998), and Rijsdijk and Hauter (1993). The purpose of this study was to develop and validate a dynamic simulation model to be employed in accurate prediction of greenhouse energy and moisture exchanges as a function of dynamic environmental factors such as solar energy, outside temperatures and moisture levels, plant moisture and energy exchanges and heat removal or storage. This article deals with the model development, validation and preliminary simulation results.

PROCEDURES

Physical Model, Weather File, and Greenhouse Characteristics

Using Fortran 77, a modular computer simulation model was developed, optimized and run on the CRAY supercomputer using dynamic analysis tools. A description of cucumber plant canopy as a series of parallel rows with rectangular cross sections and variable architectural features was extended to an overall greenhouse model having single or double plastic glazings and three greenhouse floor layers. In the energy and mass exchanges of the greenhouse system, the outside weather conditions and the deep ground temperature served as boundary conditions. Finite difference methods were used to solve the set of differential equations. Five vertical nodes were used for the canopy stand to reduce the computational time without disrupting the solution accuracy. Integration over space was performed when dealing with the radiative heat transfer. The integral equation was approximated using the Composite Trapezoidal Rule for the solar radiation. However, the more complicated closed Newton-Cotes formula with Simpson's Rule was employed when the spatial integration was performed for other selected basic variables such as cumulative leaf area index in describing heat and mass exchanges. This resulted in better accuracy. The ordinary differential equations with an initial value were solved using Euler's Method, and no stability problems were observed. A time interval of 10 sec. was used. A small time interval was required due to the rapid response of plants to dynamic environmental parameters. This small time interval caused a considerable increase in computational time, but it represented plant responses in a reasonably accurate manner. Small oscillations were observed in greenhouse climatic quantities, when a time interval of 100 s was used.

January, April, and July weather files for Delaware (latitude 40°17' N, longitude 83°05' W), Ohio, USA were used to represent winter, spring and summer climates in the simulations. The heat pumps evaluated for open (OHP) and closed loop (CHP) greenhouse systems were 3-ton (based on system heat removal capacity) gas fired units, and to provide multiposition proportional control it was assumed that 3 units were used in each greenhouse. The heat pump consists of a Rankine power cycle and a vapor compression cycle which uses a novel hydraulically connected rolling diaphragm piston cylinder device as motor, compressor and pump (Yildiz, 1993; Yildiz et al., 1993). R123 (dichlorotrifluoroethane) and R22 (chlorodifluoromethane) refrigerants were used for the power and refrigeration cycles, respectively. For the conventional greenhouse (CON) simulations, it was assumed that the heating units provided 24,612 W of heat input each. The greenhouse characteristics are presented in Table 1.

Operational and Control Strategies

The day or nighttime greenhouse temperature set points were based on the solar position. Based on the indoor air temperature, the control system operated in either the heating or cooling mode. If the system was in heating mode and if heating was required, the ventilation rate was first set to the minimum rate. The control system turned on other heating units based on the difference between the indoor and set point temperatures, providing a multi-position proportional control. If no heating was required in this mode no heating unit operated; but the system remained in the heating mode until it was switched to the cooling mode.

The cooling mode operated in two steps. The first step was to reduce the cooling load using a variable shading system and to cool the inside air by increasing ventilation rates. Two shading cloths provided the variable shading with transmissivities of 0.75 and 0.50 used individually or together. The minimum and maximum ventilation rates were 0.01 $m^3/s.m^2$ and 0.08 $m^3/s.m^2$, respectively. If the first step in cooling could not handle the cooling load, then the second step was activated, in which the heat pump units or evaporative cooling provided the cooling. In the conventional (CON) system, introducing an outside airflow rate of 0.08 $m^3/s.m^2$ when the second step was activated in the cooling mode provided evaporative cooling. Relative humidity levels in the conventional system were controlled indirectly by the temperature control. In the open loop heat pump (OHP) system, however, additional relative humidity control was provided. When the inside relative humidity levels exceeded 80%, additional ventilation was introduced to decrease inside relative humidity. In the closed loop heat pump (CHP) system, the same criterion was used to prompt the heating mode. However, the cooling mode was activated at lower inside temperatures than those used in the other two systems. The operation of the heating system was the same as in the other two systems. However, the minimum ventilation rate was used in the open loop system while no ventilation was used in this confined system. In the cooling mode of the closed loop system, there was only one step unlike the conventional and open loop systems, which had two-step cooling systems. Here, no cooling was provided by ventilation; instead, the cooling was provided by the three heat pump units providing a multiposition proportional control, after reducing the cooling load using the variable shading system. The operation of the shading system was the same as in the other two systems. Either the cooling units or the dehumidifier (the first heating unit) controlled inside relative humidity. When the inside relative humidity levels exceeded 80%, this heating unit operated as a dehumidifier to prevent excess moisture within the closed loop heat pump system.

Energy and Mass Balances

The details of energy and moisture balances of the plant leaves were previously reported by Yang (1990). For instance, energy balance of the internal air for a single layer greenhouse was expressed as

$$\rho_a * V_a * C_{pa} * dT_a / dt = Q_{sr} - Q_{ac} - Q_{vio} - Q_{flux} \qquad (1)$$

where ρ_a was the density of the bulk air (kg/m^3), V_a was the volume of the bulk air above the canopy stand (m^3), C_{pa} was the specific heat of the air (J/kg.C), T_a was the bulk air temperature (°C), and t was the time (s). Q_{sr} was the heat input by the heating source into the internal air (W), Q_{ac} was the heat transferred from the internal air to the structural cover (W/m^2), Q_{vio} was the amount of heat transferred from the inside air to the outside air due to ventilation (W), and Q_{flux} was the amount of heat transferred from the bulk air to the top layer of the canopy stand (W). Moisture balance of the internal air was

$$(1 / \nu_a) * V_a * dw / dt = - M_{conc} - M_{vio} - M_{coil} - M_{flux} \qquad (2)$$

where ν_a was the specific volume of the air (m^3/kg dry air), and w was the humidity ratio (kg H_2O/kg dry air), M_{conc} was the amount of moisture condensed on or evaporated from the inside surface of the cover, M_{vio} was the amount of moisture transferred from the inside air to the outside air via ventilation (kg H_2O/s), M_{coil} was the amount of moisture condensed on the cooling coil (kg H_2O/s), and M_{flux} was the amount of moisture transferred from the bulk air to the top layer of the canopy stand (kg H_2O/kg dry air).

Thornley and Johnson (1990) indicated that, in describing CO_2 concentrations, the unit of parts per million (ppm) has two deficiencies. First it is not clear whether the definition is kilograms per million kilograms or molecules per million molecules. The second problem is that photosynthesis depends on the absolute number of CO_2 molecules per unit volume, and not just the proportion of CO_2 molecules in the air. Therefore, they suggest that the use of parts per million should be avoided. Instead, they recommend the following definition of concentration to be used in any model. From the gas laws, the concentration of any gas at an arbitrary temperature and pressure is given by

$$\text{Concentration} = (273.15 / T) * (P / 101325.0) * (0.044618) \qquad (3)$$
$$\text{Density} = (\text{Concentration}) * (\text{Relative Molecular Mass}) \qquad (4)$$

where the concentration is in kmole CO_2/m^3, T is the temperature (K) and P is the pressure (Pa), and 0.044618 is the concentration of pure CO_2 (or any other gas) in kmole/m^3 at normal temperature (273.15 K) and pressure (101325.0 Pa). The relative molecular mass of CO_2 is 44.0098 kg CO_2/kmole CO_2. Using these relations, the following equation was derived for the CO_2 balance of the inside air

$$dCO_2 / dt \text{ (kg } CO_2\text{/s)} = C_{inj} - (C_{vio} + C_{flux}) \qquad (5)$$

where C_{inj} was the CO_2 injection rate (kg CO_2/s), C_{vio} was the CO_2 exchange rate due to ventilation (kg CO_2/s), and C_{flux} was the CO_2 exchange rate between the bulk air above the canopy stand and the top layer of the canopy (kg CO_2/s). CO_2 concentrations for inside and outside air were in ppm. The instantaneous gross rate of canopy photosynthesis was defined according to Thornley and Johnson (1990), whereas dark respiration was expressed using the Q_{10} factor.

In dealing with the energy and mass exchanges of the structural cover, it was assumed that the exchanges occurred homogeneously on the cover, the heat storage capacity of the cover material was small compared to the existing fluxes, and no condensation or evaporation occurred on or from the cover. The steady-state energy balance equation for a single cover greenhouse was

$$0 = Q_{ac} + Q_{oc} + \alpha_{s,c} * A_c * IG_s + LW_c \quad (6)$$

where Q_{ac} was the convective heat transfer between the internal air and the cover (W), Q_{oc} was the corresponding term between the outside air and the cover (W), $\alpha_{s,c}$ was the short-wave absorptivity of the cover, A_c was the cover area (m^2), IG_s was the amount of solar radiation on the cover (W/m^2), and LW_c was the net long-wave radiation on the cover (W).

It was assumed that the floor was covered with a polyethylene film; however, an option was provided so that bare soil could also be used. A one-dimensional heat conduction equation was used in dealing with the energy balance of the greenhouse floor, by dividing the floor into three layers (0.01, 0.10 and 0.50 m) with the assumption of homogeneous thermal and hydraulic properties within each layer (Arinze, 1984; Avissar and Mahrer, 1982; Kindelan, 1980). It was also assumed that the deep ground temperature was constant at 15°C (Takakura et al., 1971), and no condensation or evaporation occurred on or from the floor surface.

The solar radiation was treated by splitting it into direct, diffuse, and scattered components and assuming that all the radiation reflected by and/or transmitted through foliage elements contributed only to the diffuse component. The expression widely used in microclimatological studies for the penetration function of direct solar radiation for uniformly distributed plant canopies was expanded to a row plant stand whose foliage area distribution varied both vertically and horizontally. It was assumed that the scattering distributions (both upward and downward) were uniform horizontally.

A resistance concept was used in dealing with the thermal radiation as outlined by Incropera and DeWitt (1985). A parallel plane analysis was employed whenever it was applicable. For the other cases, a multiple surface radiation exchange analysis using the same approach (resistance concept) was employed.

RESULTS AND DISCUSSION

The comparisons of simulation results with experimental findings were made using the following inputs for a conventional greenhouse system (the shading and evaporative cooling systems were not operational for these comparison simulations): a plant height of 2.00 m, a distance of 0.86 m between rows, a north-south row direction, and a greenhouse floor of uncovered soil. Fig. 1 shows the diurnal changes in predicted and measured air and leaf temperatures for two successive days. The predicted and the measured temperatures were very close to each other. Generally, the predicted air temperatures were slightly lower than the measured temperatures. The predicted inside relative humidity levels were also compared to the measured levels, and plotted together with the predicted and the measured transpiration rates (Fig. 1). The model consistently overestimated (~7%) the daytime inside relative humidities while underestimating (~10%) at night. Fig. 2 shows the comparison between predicted and measured transpiration rates per unit ground area for the two successive days, plotted together with measured solar radiation. The highest transpiration rate (365 g/hr.m^2) was observed on the second day when high solar radiation existed. The lowest transpiration rate (20 g/hr.m^2)

was observed at night. Transpiration rates closely followed the changes in the solar radiation, and the values were in fairly good agreement during the day. The transpiration rates at night, however, were overestimated. This was due the stomatal resistance expression, which was derived from daytime data only, not counting for the effects of climatic variables other than solar radiation. Since the nighttime transpiration rates account for a very small portion of the total transpiration, the absolute magnitude of the error due to overestimation at night was not significant. Predicted and measured relative humidity, and air and leaf temperatures were also in fairly good agreement. The predicted air temperatures were slightly lower than the measured temperatures. Also, the model consistently overestimated the daytime relative humidity levels inside (approx. 7%) while underestimating them at night (approx. 10%).

Fig. 3a shows the diurnal solar radiation, temperature and humidity regimes in response to the outside climatic conditions within an open loop heat pump (OHP) greenhouse system in winter (January). Only small temperature fluctuations were observed in this greenhouse during the day, because the two heating units were operating continuously. Temperature fluctuations were observed again in the late afternoon. This time the third heating unit was turned on due to the reduced solar radiation and the decrease in the outside air temperature. Leaf temperatures were affected by the variations in the air temperature. The fluctuations in the inside relative humidity caused fluctuations in the transpiration rates due to the resulting changes in vapor pressure deficit. Temperature fluctuation of 3°C resulted in fluctuations of 1°C, 5% and 3 mg $H_2O/s.m^2$ in the leaf temperature, inside relative humidity, and transpiration rates, respectively. Fig. 3b shows the diurnal changes in solar radiation, temperature and moisture regimes in the same greenhouse system in spring (April). Additional control (increased ventilation rates) was employed to control the relative humidity at about 80%. Fig. 3c shows the diurnal changes in solar radiation, temperature and moisture regimes in the same greenhouse system in summer (July). Inside air temperatures were very close to the outside air temperatures because of the increased ventilation rates due to the increased inside relative humidity levels. No heating or cooling was required at night. The increase in the air temperature resulted in the use of the shading system between noon and 16:00. However, it was still required to do some mechanical cooling between the hours of 13:00 and 15:00. The difference between the inside air and leaf temperatures was about 1.6°C at night, and it was 5.0°C at noon. This was because of the low leaf temperatures due to increased transpiration rates (44 mg $H_2O/ s.m^2$) as a result of the increased solar radiation and inside air temperature. The inside relative humidity followed the variations in the outside relative humidity levels due to the increased ventilation rates.

Generally speaking, regardless of the individual greenhouse system and the season, when high relative humidity levels prevailed when there was no or low solar radiation, then the leaf temperatures were lower than, but very close to the air temperatures. However, high solar radiation along with high relative humidity levels resulted in leaf temperatures which were higher than the air temperature. At this point, the plants could not transpire much, and they did not remove the excess solar energy on their surfaces. This resulted in increased leaf temperatures.

CONCLUSIONS

In the conventional and the open loop heat pump systems, inside air temperatures fluctuated within a temperature range of 3°C due to the operation and control of the heating units. Leaf temperatures closely followed the inside air temperature fluctuations because of their low thermal storage capacities. The floor temperatures, however, showed no short-term fluctuations due to the high storage capacity of the floor. Leaf temperatures were about 1.3°C (night time) and 4°C (day time) lower than the inside air temperatures in winter. During periods of high solar radiation, the temperature difference was about 5°C due to increased transpiration rates. When very high relative humidity conditions prevailed (99%) in the conventional system, then the temperature differences between the two were negligible due to reduced transpiration rates. Inside relative humidity followed the variations in indoor air temperature very closely. The 3°C variations in the air temperature resulted in 5% relative humidity fluctuations. In the closed-loop heat pump system, inside relative humidity was maintained at 80% plus or minus 10%. Leaf temperatures were about 0.5°C lower than the air temperatures. This was a result of reduced transpiration rates due to increased relative humidity levels and low inside solar radiation. Leaf temperatures were about 0.8°C higher than the air temperatures when high solar radiation and high inside relative humidity levels existed.

This study showed that the developed model with all its components performed very well, and the greenhouse with the heat pump system could be operated as a confined system, because the dehumidifier handled the excess moisture. The study also showed that proportional control was an essential part of the control system. A multiposition proportional control, as employed in this study, would provide a reasonable control for the greenhouse environment. The greenhouse model developed in this study is useful for ecologists, plant scientists, and engineers to evaluate individual or combined effects of various forcing functions on the enclosed environment and plant responses; and to develop control strategies for different parameters such as energy and water conservation.

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Literature Cited

Arinze, E.A., Schoenau, G.J. and Besant, R.W. 1984. A dynamic thermal performance simulation model of an energy conserving greenhouse with thermal storage. Trans. ASAE 27:508-519.

Avissar, R. and Mahrer, Y. 1982. Verification study of a numerical greenhouse microclimate model. Trans. ASAE 1711-1720.

Chandra, P. and Willits, D.H. 1981. An analysis to predict thermal behavior of a greenhouse collection/storage system. ASAE Paper No: 80-4025, ASAE, St. Joseph, MI 49085, USA.

Glaub, J.C. and Trezek, G.J. 1981. Heat and mass transfer analysis of greenhouse. ASAE Paper No: 81-4031, ASAE, St. Joseph, MI 49085, USA.

Incropera, F.P. and DeWitt, D.P. 1985. Introduction to heat transfer. John Wiley & Sons, Inc., New York, USA.

Kindelan, M. 1980. A dynamic modeling of greenhouse environment. Trans. ASAE 23(5): 1232-1239.

Navas, L.M., de la Plaza, S., Garcia, J.L., Duran, J.M. and Retamal, N. 1998. Formulation and sensitivity analysis of a dynamic model of the greenhouse climate validation for a mild Mediterranean climate. Acta Hort. 456:305-312.

Pita, G.P.H., Pontes, M. and Vargues, A. 1998. Mediterranean greenhouse energy balance. Acta Hort. 456:375-382.

Rijsdijk, A.A. and Houter, G. 1993. Validation of a model for energy consumption, CO_2 consumption and crop production (ECP-Model). Acta Hort. 328:125-131.

Takakura, T.K., Jordan, A. and Boyd, L.L. 1971. Dynamic simulation of plant growth and environment in the greenhouse. Trans. ASAE 14(5):964-971.

Thornley, J.H.M. and Johnson, I.R. 1990. Plant and crop modelling. A mathematical approach to plant and crop physiology. Clarendon Press, Oxford, UK. p.48-49, 128.

Yang, X., Short, T.H., Fox, R.D. and Bauerle, W.L. 1990. Dynamic modeling of the microclimate of a greenhouse cucumber row-crop. Part I. Theoretical model. Trans. ASAE 33(5):1701-1709.

Yildiz, I. 1993. A performance study of a Rankine cycle driven heat pump system. M.S. Thesis, The Ohio State University, Columbus, Ohio, USA.

Yildiz, I., Lin, C. and Christensen, R.N. 1993. A performance study of a Rankine cycle driven heat pump system. AES-Vol. 29, Heat Pump and Refrigeration Systems Design, Analysis, and Applications, ASME 1993, USA. p.9-18.

Tables

Table 1. Greenhouse characteristics used in the simulation model.

Greenhouse length	7.5 m (Conventional and OHP) and 25.0 m (CHP)
Greenhouse width	7.50 m
Greenhouse height at eaves	2.50 m
Greenhouse height at ridges	4.50 m
Crop row orientation	North - South
Distance between plant rows	0.75 m
Floor surface material	Reflective mulch
Glazing	Single and double polyethylene

OHP: Open loop heat pump system; CHP: Closed loop heat pump system

Figures

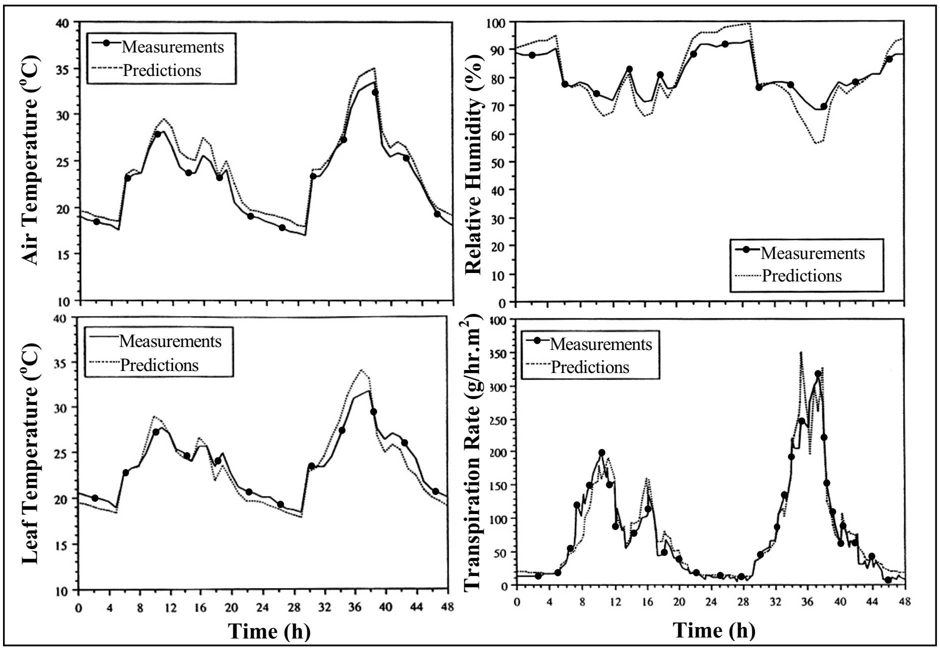

Fig. 1. Comparisons between predicted and measured diurnal changes of selected parameters (a) on June 1st and 2nd.

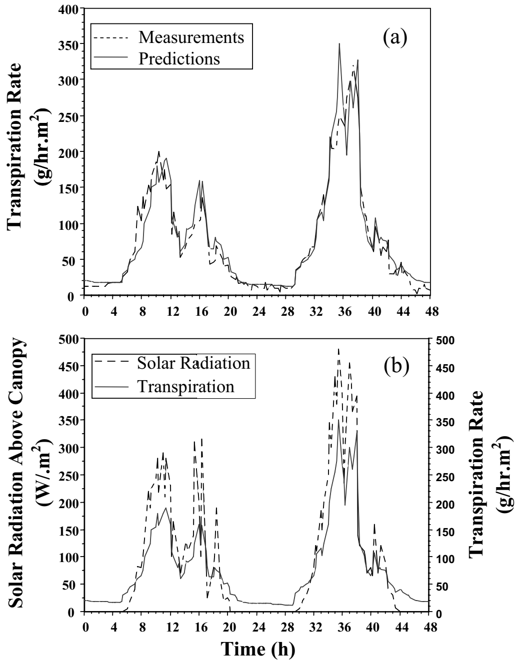

Fig. 2. Comparisons between predicted and measured diurnal changes of transpiration rates (a) on June 1st and 2nd, plotted together with measured solar radiation (b).

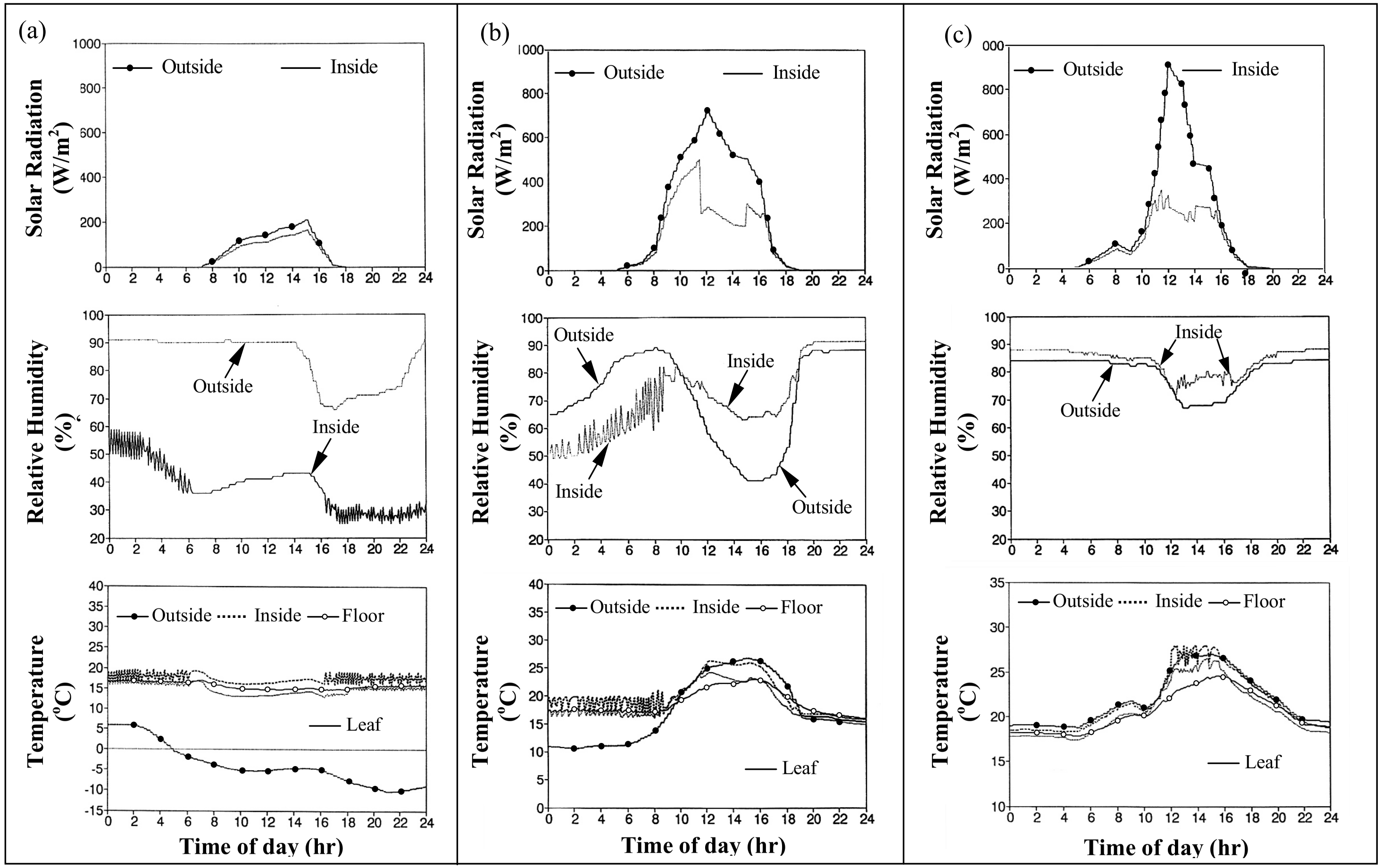

Fig. 3. Diurnal changes in predicted greenhouse climatological quantities in the open loop heat pump greenhouse system in winter (a), spring (b), and summer (c).