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# PREDICTION OF LONGITUDINAL VARIATIONS IN TEMPERATURE AND RELATIVE HUMIDITY FOR EVAPORATIVE COOLING GREENHOUSES

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## ABSTRACT

Evaporative cooling is an effective method to relieve the heat accumulation in a greenhouse in the summer, and to reduce the temperature of the greenhouse to a level lower than that of the outside air. A mathematical model was presented to describe the gradient of air temperature and relative humidity as a function of distance along an evaporative cooled greenhouse. The model was validated with the measured data in a 25.6 m x 56 m greenhouse-planted tomato crop. Good agreement was found between the measured and predicted values of air temperature and relative humidity when HORTITRANS transpiration model was incorporated. The predictive performance of temperatures was within 2.5 ℃ and that of relative humidity was within 8%. As the solar radiation and outside air temperature changed rapidly and the ventilation rate was lower, the predictive values were delayed as compared to actual data.

***Keywords**: Greenhouse, evaporative cooling, modeling, thermal gradient, moisture gradient*

## 1. INTRODUCTION

Taiwan is located in the subtropical region. The existence of high solar energy leads to heat accumulation in greenhouses in summer, causing malformation of flowers and increased risk of disease, and reducing the quality and quantity of greenhouse crops. The average maximum day temperature goes up to nearly 36 °C. The growing of many horticultural products in summer becomes impossible in open fields because of the high temperature and strong solar radiation.

There are four ways to deal with the heat accumulation problem in greenhouses: 1) shading, 2) ventilation, 3) evaporative cooling, and 4) mechanical refrigeration. The high cost and heavy energy requirement make the refrigeration technique impractical. Shading could decrease solar energy; however, the minimum requirement of light intensity for plants limit the extent of shading. The air exchange between inside and outside of a greenhouse by natural ventilation depends on the difference in temperature between the greenhouse and the ambient air or the difference in pressure between the greenhouse and the outside air. The inside temperature of greenhouses are usually 5–15

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°C higher than the temperature of outside air because of insufficient air exchange rates. Mechanical ventilation rates could be controlled by the power rating and number of fans. The lowest temperature inside a greenhouse with temperatures controlled only by fans could at most be the same as that of the outside air. Many crops would still not survive in a mechanically ventilated greenhouse in the summer. Considering the efficiency and cost of environmental control techniques, evaporative cooling is an attractive option for reducing the greenhouse temperature below that of ambient air.

The major concerns of climate control in a temperate country are heating and energy savings. Few studies from temperate regions, therefore, mentioned the problem of cooling greenhouses. Montero et al. (1981) discussed three evaporative cooling systems and compared their performances by measurement of greenhouse air and plant leaf temperatures. They found that the fan and pad system was the best, with a stable cooling ability. Walker and Cotter (1968) compared three different evaporative cooling systems. They proposed that the fine water-droplet mist system provided the maximum cooling ability, followed by the pad-cooling and the oarse-mist systems. Giacomelli et al. (1985) tested wetted overhead energy saving blanket and fog nozzles on a moveable boom to cool greenhouses. The former could reduce the temperature by up to 4 °C; the latter by up to 10 °C. The temperature uniformity was improved by higher airflow rates. Garzoli (1989) reviewed four methods — shading, natural ventilation, mechanical ventilation, and evaporative cooling to cool the air temperature in greenhouses and proposed design criteria for evaporative cooling systems. Montero and Anton (1994) found four factors that cause the temperature reduction. These were ventilation, shading, evaporative cooling and the crop evapotranspiration. However, measured data was not available to quantify the effect of these techniques.

In Taiwan, the most popular evaporative cooling system adopted by growers is the fan and pad system. As the air passes through the pad by the suction of fans, the evaporation of water decreases the air temperature and relative humidity increases. The temperature gradient throughout the greenhouse leads to uneven growth rates and maturation rates for plants.

A microclimate model for greenhouses is very useful to describe the relationship between microclimatic conditions and the affecting factors. Kano and Saddle (1988) reviewed various greenhouse models. Most greenhouse models were developed taking into consideration the heating requirement. Greenhouse models can be classified as steady state, steady-periodic (Froehlich et al., 1979), and time-dependent (Chandra et al., 1981). Albright (1991) considered the influence of greenhouse parameters and found that the temperatures of cover materials, crops and inside air could be assumed as time-independent. However, ground temperature was time-dependent. Maher and O'Flaherty (1973) established a thermal model by considering the heat balance of crops, inside air and cover materials. Levit and Gaspar (1988) considered the temperature change of these factors as time dependent. Avissar and Mahrer (1982) incorporated the performance of heater and ventilation fans into the greenhouse model. Bakker (1991) proposed a model that included the effect of relative humidity of ambient air on the greenhouse microclimate. Jolliet (1994) developed a model to predict and optimize humidity and transpiration in greenhouses. Boulard and Baille (1993) proposed a greenhouse model that incorporated the effects of ventilation and evaporating cooling.

All models assumed the uniformity of temperature and humidity inside greenhouses. Spatial thermal variations in the air mass were not considered. The experimental data for validating the model were usually measured in the center of the greenhouse (ASAE, 1996). However, non-uniform distribution of temperature and humidity could be found in greenhouses with the operation of ventilation devices. As solar radiation increased or the ventilation capacity lowered, the microclimatic gradient became more significant. A greenhouse model that considers the gradient of air temperature and humidity needs to be developed and validated. This model could serve as a tool to evaluate the performance of controlling equipments.

The objective of this study was to develop a microclimate gradient model to describe the temperature and humidity distribution throughout the greenhouse. Experimental data were collected and used to validate the model.

## 2. MODEL DEVELOPMENT

The model development for the microclimatic gradient was to divide the greenhouse into many sections along the longitudinal direction. The schematic illustration of the thermal and mass fluxes occurring in a typical section of a greenhouse is presented in Fig. 1. The outlet conditions of the previous section were the inlet conditions of the next contiguous section. The first section was located at the exit of the cooling pad. Assumptions for gradient model development were as follows.

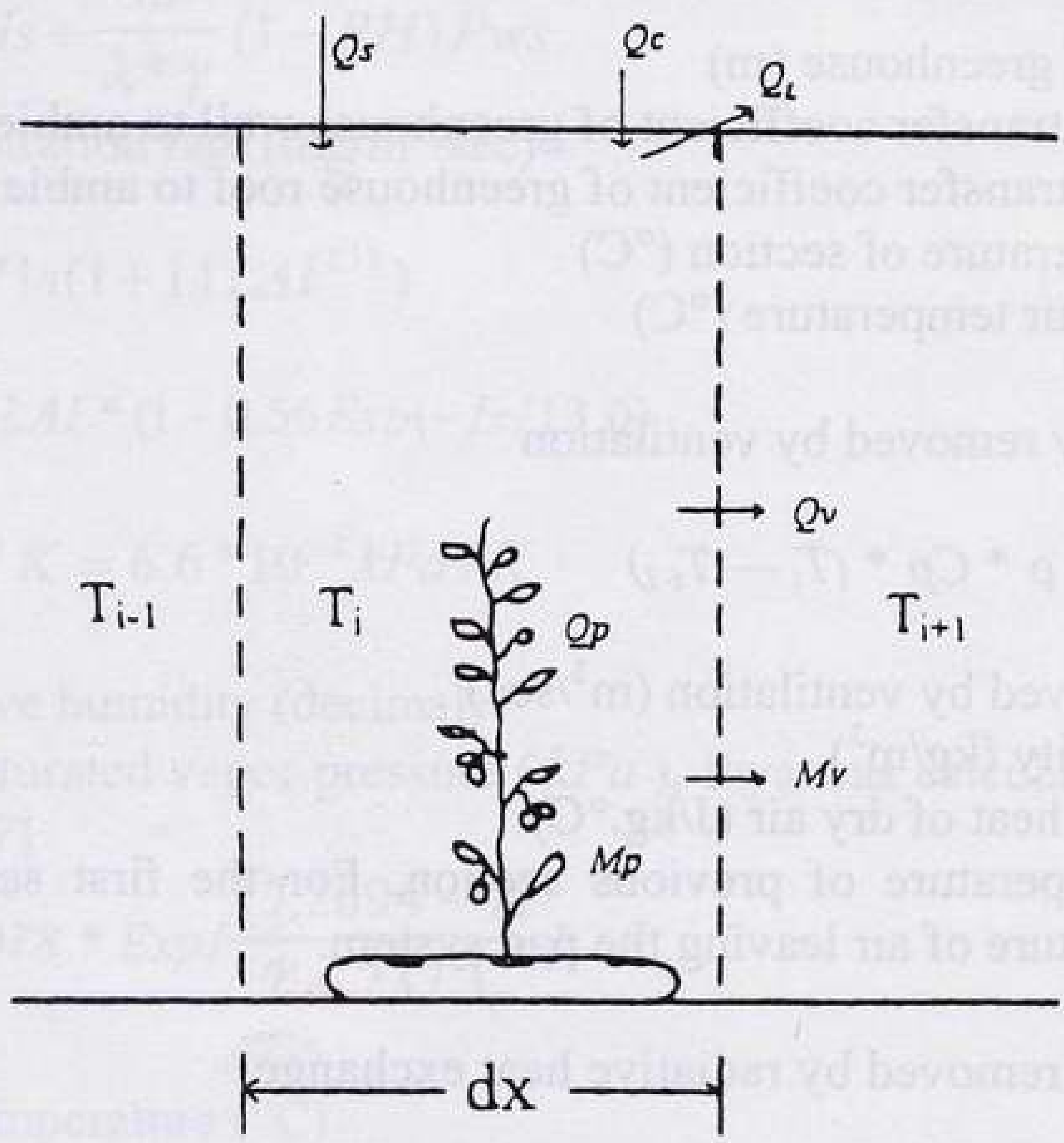


Fig. 1: Schematic of the thermal and mass fluxes occurring in a special section of greenhouse (The notation are described in the list of notation used)

1. The heat transfer coefficients of cover materials were constant. The conductive and convective transfer coefficients were combined as the heat transfer coefficient.
2. The temperatures of the inside air, crops and cover materials were in a steady state.
3. Crops were planted in medium bags. The ground was paved by concrete. The soil thermal energy was not considered because crops covered most of the floor area.
4. The sources of water vapor were from the outside air by ventilation and crop transpiration. Irrigation water and condensation of water on cover materials were not considered.

## 2.1 Thermal Transfer Model

(a) Qs : solar radiation

$$Qs = Is * dx * W \qquad \ldots (1)$$

where, Is = entrance energy of short wave radiation from sun ($W/m^2$)
dx = length of section (m)
W = width of greenhouse (m)

(b) Qc: Energy flow in/out by heat transfer

$$Qc = (2h*dx*Uw+W*dx*Ur)(T_i - T_{air}) \qquad \ldots (2)$$

where, h = height of greenhouse (m)
Uw = the heat transfer coefficient of greenhouse wall to ambient air ($W/m^2$-K)
Ur = the heat transfer coefficient of greenhouse roof to ambient air ($W/m^2$-K)
$T_i$ = air temperature of section (°C)
$T_{air}$ = outside air temperature (°C)

(c) Qv : energy removed by ventilation

$$Qv = Ven * \rho * Cp * (T_i - T_{i-1}) \qquad \ldots (3)$$

where, Ven = air removed by ventilation ($m^3$/sec)
ρ = air density ($kg/m^3$)
Cp = specific heat of dry air (J/kg.°C)
$T_{i-1}$ = air temperature of previous section. For the first section, $T_{i-1}$ was the temperature of air leaving the pad system.

(d) $Q_\lambda$: energy removed by radiative heat exchange

$$Q_\lambda = \tau * Fs * \sigma * dx * W * [T^4_{ik} - T^4_{sky}] \qquad \ldots (4)$$

where, τ = long wave thermal transmittance of cover materials

Fs = Shape factors for the sky as seen from the internal section of the greenhouse
$\sigma$ = Stephan-Boltzman constant ($5.67*10^{-8} W/m^2 .K^4$)
$T_{ik}$ = absolute temperature of $T_I$ (°K); $T_{ik} = T_i + 273.16$
$T_{sky}$ = absolute temperature of sky ( K), $T_{sky}$ was calculated as follows (Swinbank, 1963)

$$T_{sky} = 0.0552*(T_{ik})^{1.5} \quad \text{… (5)}$$

(e). Qp: transpiration energy by crops

$$Qp = \lambda * Tr * dx * W * Pf \quad \text{… (6)}$$

where, Tr = transpiration rate of crops in section ($kg/m^2$)
Pf = ratio of crop-occupied area to greenhouse ground area
$\lambda$ = latent heat of water (KJ/Kg). $\lambda$ was the function of temperature (Albright, 1991)

$$\lambda = 2501 - 2.42T \quad \text{…(7)}$$

In this study, tomatoes were planted in the greenhouse. Two transpiration models of tomatoes were adopted from the literature.

(a) HORTITRANS model (Jolliet, 1994)

$$Tr1 = \frac{a}{\lambda} Is + \frac{ht}{\lambda * \gamma}(1 - RH) Pws \quad \text{… (8)}$$

where, Tr1 = transpiration rate ($mg/m^2$-sec)

$$a = 0.154 * \ln(1 + 1.1 LAI^{1.13}) \quad \text{… (9)}$$

$$ht = 1.65 * LAI * (1 - 0.56 Exp(-Is/13.0)) \quad \text{… (10)}$$

$$\gamma = 66 \rho a / K = 6.6 * 10^{-2} kPa / K \quad \text{… (11)}$$

where, RH = relative humidity (decimal)
Pws = air saturated vapor pressure ( $kPa$ ), Pws was calculated as follows (Weiss, 1977)

$$Pws = 0.61078 * Exp[\frac{17.2694 - Ti}{T + 237.3_i}] \quad \text{… (12)}$$

$T_i$ = air temperature (°C)
LAI = leaf area index

(b) Jolliet & Bailey model (Jolliet and Bailey, 1992)

$$Tr_2 = 0.32 \cdot Is + 5.5Pws(1 - RH) + 5.3u \quad \dots (13)$$

where, $Tr_2$ = transpiration rate ($g/m^2/h$)
u = wind speed (m/sec)

The energy balance equation for this section was:

$$Qs = Qc + Qv + Q_l + Qp \quad \dots (14)$$

## 2.2 Moisture Transfer Model

The moisture transfer system of this section was described as follows.
(a) Mv: the entrance moisture due to ventilation

$$Mv = Ven \cdot \rho \cdot (H_i - H_{i-1}) \quad \dots (15)$$

where,
$H_i$ = absolute humidity of air in the section (kg $H_2O$/kg dry air)
$H_{i-1}$ = absolute humidity of air in the previous section (kg $H_2O$/kg dry air)

(b) Mp: the moisture increase due to transpiration of crops

$$Mp = Tr \cdot dx \cdot W \cdot Pf \quad \dots (16)$$

The moisture balance equation for this section was:

$$Mv + Mp = 0 \quad \dots (17)$$

## 2.3 Relative Humidity

Absolute humidity was selected as a parameter to describe the heat and moisture balance in this study. However, air relative humidity was measured. The transformation between relative humidity ($RH_i$) and absolute humidity ($H_i$) can be described as follows.

The saturated vapor pressure (Pws) can be calculated as in Equation (12). The partial pressure of the water vapor was,

$$Pw = Pws * RH \quad \dots (18)$$

Absolute humidity (Hi) was computed as follows.

$$Hi = \frac{0.62198 * Pw}{P_{atm} - Pw} \quad \dots (19)$$

$P_{atm}$ was the air pressure,

$P_{atm}$ = 101.KPa ... (20)

As the Hi of new section was calculated by equation (17), new Pwi can be found by the following:

$$Pwi = \frac{H_i \cdot P_{atm}}{H_i + 0.622} \quad \text{... (21)}$$

The new Pwsi then would be computed by the new $T_i$ value, and the new RH value was found as follows.

$$RH = \frac{Pwi}{Pwsi} \quad \text{... (22)}$$

## 3. MATERIALS AND METHODS

### 3.1 Experimental Greenhouse

A 25.6 m x 56.0 m x 3.5 m high, 5800 $m^3$ volume, acrylic glazed steel framed structure, closed greenhouse located at Taichung was selected to validate the mathematical model. The transmittance was 70% for external shading nets and 50% for internal shading nets. The ventilation system consisted of 16 units of 135 cm fans with a capacity of 550 $m^3$/min. The maximum ventilation rate was 1.5 air exchange per minutes. The pad system was installed on the wall of the short side. Fans were placed on the opposite side. The thickness of the pad was 15 cm and its area was 80 $m^2$. The flow rate of the pump was 330 liter/min.

Tomatoes were planted in rot medium bags arranged on the floor. Crop spacing was 1.5 m. A drip irrigation system was applied. At the beginning of the experiment, the height of the tomato plants was about 1.5 m. The ratio for the tomatoes to the floor area (Pf) was about 50%.

### 3.2 Measuring Devices

T-type thermocouples (Omega Engineering, USA) were selected to measure the air temperatures. The thermocouple junction was covered with aluminum paper to shield against the solar radiation. These thermocouples were connected to a Delta-T2e data logger. All thermocouples were calibrated using a calibrator (TC-2000, Instrutek. AS Norway). The accuracy of these temperature sensors were within ±0.15 ºC.

The relative humidity sensor was a Shinyei THP-B7T Transmitter (Shinyei Kaisha Co., Tokyo Japan). The sensing element was made of Macro-molecule materials. The measuring range was from 20-90% RH. The accuracy was ±1.0% RH after calibrating with saturated salt solutions.

Solar radiation was measured by two E8-48 paranometers (Eppley Cor., USA). The sensing element was a thermopile. These paranometers were calibrated with the Kipp

& Zonen Model CMll thermopile paranometer (Kipp and Zonen Ltd, The Netherlands). The accuracy of this meter was ±2%.

The wind speed past the crop was measured with a hot-wire anemometer (Sweta-30, Sweta, Sweden). The accuracy of this meter was ±0.15 m/sec. The output signal was 0-2 Volts. The leaf area index was measured by LA1-200 Plant Canopy Analyzer (Li-cor, USA).

The sensors of temperature, relative humidity, and solar radiation were connected to a Delta-T2e data logger (Delta device LTP UK).

## 3.3 Experimental Procedure

The positions of the sensors are shown in Fig. 2. There were two paranometers to measure the outside and side solar radiation of the greenhouse. One thermocouple was used to measure the outside temperature. The other nine thermocouples, denoted as $T_0$ - $T_8$, were installed in a mid-position along the greenhouse's length. $T_0$ was used to measure the temperature of the air just exiting from the cooling pad. $T_8$ was the air temperature near the exhaust fan.

Four relative humidity transducers were arranged as follows: near the pad ($RH_0$), 16 m ($RH_2$), 28 m ($RH_4$), and 52 m ($RH_7$).

In order to compare the effect of the fans' operation on the greenhouse microclimate, the ventilation rate was defined as:

$$\text{Ventilation rate} = TVC / Vol \ (\text{min}^{-1}) \qquad \dots (23)$$

where, TVC = Total ventilation capacity ($m^3$/min)
Vol = Volume of greenhouse ($m^3$)

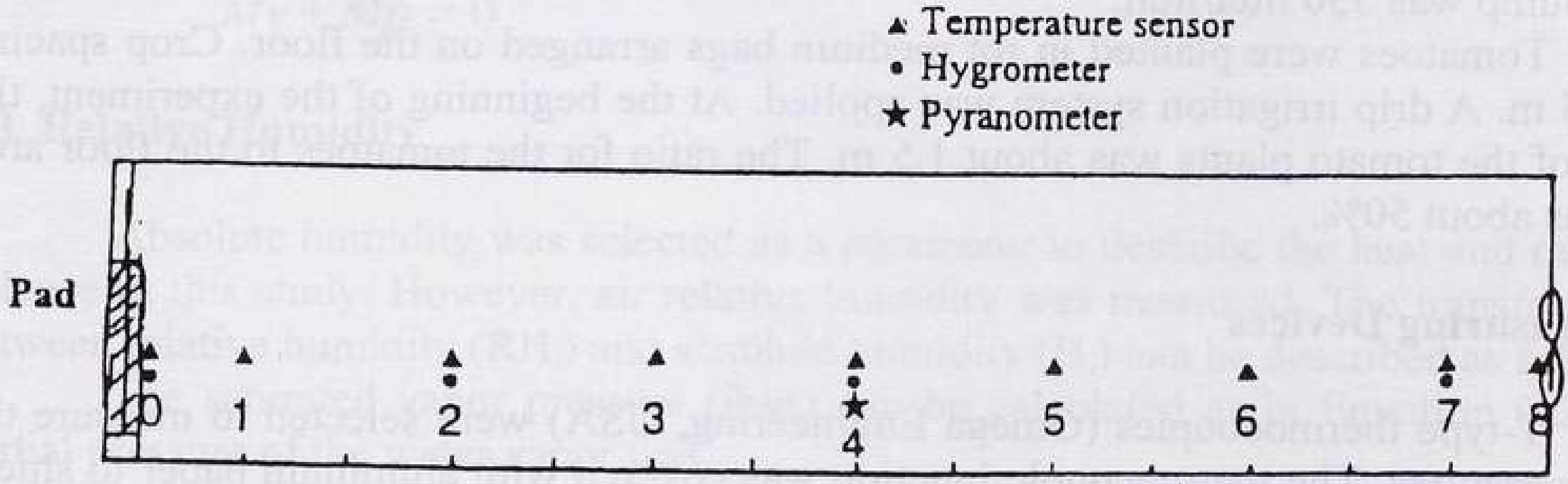


Fig. 2: Sensor layout in the greenhouse

## 3.4 Parameters for the Mathematical Model

The relevant physical parameters measured for this study are listed in Table 1. The thermal conductive coefficients of the wall and roof were adapted from the manufacturer. LAI values were measured by the LAI meter directly for the each experiment.

Table 1: Parameter values for the model

| Symbols | Description | Numeric values | Units |
|---|---|---|---|
| dx | Length of section | 1 | m |
| W | Width of greenhouse | 25.6 | m |
| h | Height of greenhouse | 3.5 | m |
| Vol | Volume of greenhouse | 5,800 | $m^3$ |
| Uw | Thermal conductive coefficient of the wall | 6.4 | $W/m^2$. ºC |
| Ur | Thermal conductive coefficient of the roof | 6.8 | $W/m^2$.ºC |
| τ | Thermal transmittance | 0.12 | |
| Fs | Shape factor | 0.85 | |
| Cp | Specific heat of air | 1006 | J/kg-ºC |
| Pf | Plant ratio in greenhouse | 0.3-0.5 | |
| LAI | Leaf area index | 3.1-4.2 | |

### 3.5 Simulations

The $T_i$ and $RH_i$ values at each section for different operating conditions were computed step by step. Because the energy balance involved the four-power temperature terms ($T_{1k}^4$), $T_i$ values were obtained by a Q-BASIC program (COOLING.BAS). The computing procedures were as follows:

a) Input the outside conditions including air temperature, relative humidity and solar radiation.
b) Input the operation conditions of the greenhouse (action of shading nets, number of fans, and pad system).
c) The air temperature and relative humidity just exiting to cooling pad were the initial conditions ($T_0$, $RH_0$) for the first section, then the $T_1$ and $RH_1$ were computed using Eqns. (14) and (17).
d) $T_1$ and $RH_1$ were the initial conditions for section 2 to compute $T_2$ and $RH_2$.
e) Continue the calculating procedure until all $T_i$ and $RH_i$ are computed.

### 3.6 Evaluation of Model Performance

The quantitative criteria for the comparison of predictive performance were defined as follows.

a) Predictive errors (*Ei*)

$$Ei = Yi - Xi \quad \ldots (24)$$

where, Yi = actual measured values
Xi = predicted values by model

b) Predictive performance criteria (PPC)

$$PPC = \sum |Ei| / n \quad \text{... (25)}$$

where,

$|Ei|$ = absolute values of error

n = number of data

# 4. RESULTS AND DISCUSSION

## 4.1 Distribution Characteristics of Temperature and Relative Humidity

A typical microclimate of the greenhouse for June 24, 1999 is presented in Fig. 3. The maximum solar radiation was nearly 700 W/m$^2$ without using external and internal shading nets. Internal shading nets were used after 14:00 h to reduce the solar energy on the greenhouse crop. Only six fans were operated with the pad. The air temperature ($T_0$) exiting the pad was nearly stable within the range of 26-28 ºC. The inside temperatures increased along the longitudinal direction of the greenhouse because of the absorption of solar energy inside the greenhouse. The temperature $T_7$ at the end section of the greenhouse had the highest value of more than 40 ºC at noon. The difference between $T_0$ and $T_7$ was more than 14 ºC. The temperature gradient at the high solar radiation and low ventilation rate from one greenhouse end to the other is evidently excessive.

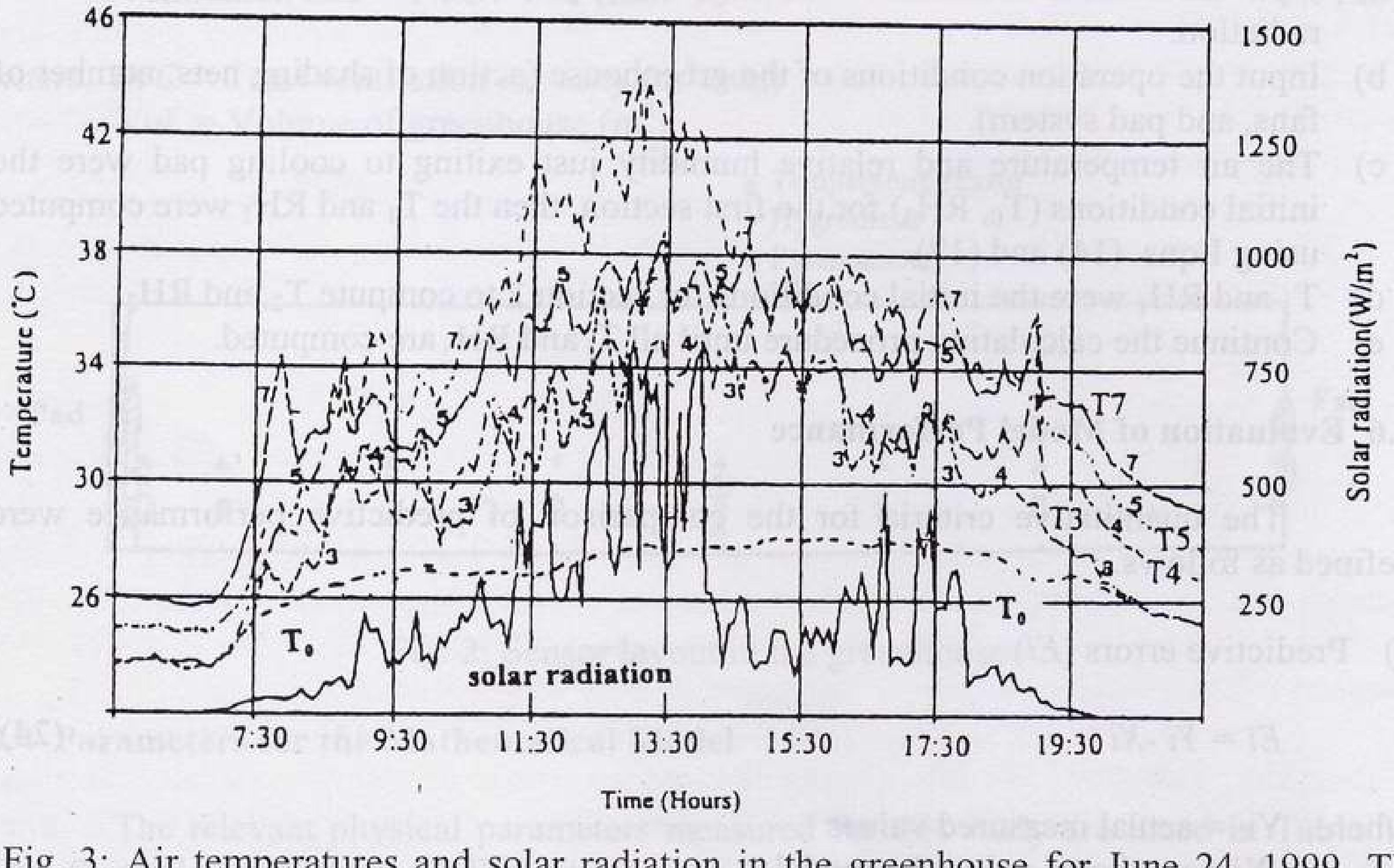


Fig. 3: Air temperatures and solar radiation in the greenhouse for June 24, 1999. The numbers in the figure denote the air temperature of $T_i$, for example, Number 7 represents $T_7$

The gradient of air humidity on the same day is shown in Fig. 4. The distribution of relative humidities was found to be uneven. The relative humidity of $RH_2$ (16 m from the pad) could be kept at 70%; $RH_4$ (32 m from pad) ranged from 50 to 60%. $RH_7$ (54 m from the pad) was the same as the outside air humidity.

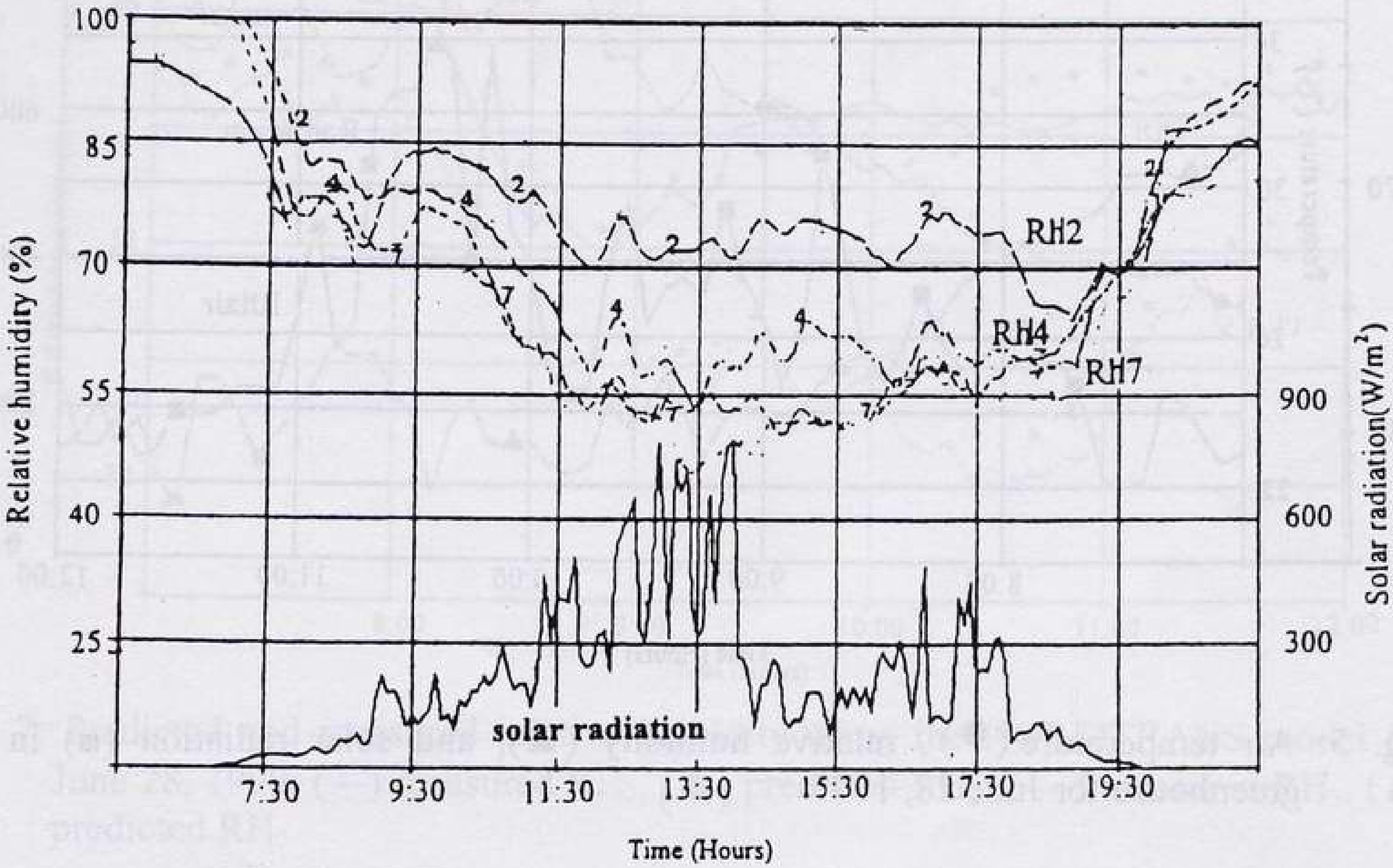


Fig. 4: Air relative humidities and solar radiation in the greenhouse for June 24, 1999. The numbers in the figure denote the air temperature of $RH_i$, for example, Number 2 represents $RH_2$

## 4.2 Effect of Ventilation Rate on Thermal and Moisture Variations

To validate the gradient model, a ventilation rate of 0.75 $min^{-1}$ was used for five days (June 26 - June 30, 1999). For June 28, the time-variations of the outside air temperature ($T_{air}$), relative humidity ($RH_{air}$) and inside solar radiation are presented in Fig. 5. From 7:30 a.m., four fans began operating. The fan system of 16 units begun to be applied at 9:00 a.m. and the internal shading net was drawn over the crop at 10:40 a.m. Exit temperatures from the cooling pad remained within the range 27-28 °C.

The two transpiration models were compared for their desirability. The simulated results with the HORTITRANS model are shown in Figs. 6 and 7. The measured and the simulated values indicated close agreement. The maximum predictive errors were 2.3 °C for $T_1$, 1.5 °C for $T_5$, and 2.2 °C for $T_7$ (Fig. 6.).

Fig. 7 indicates the predicted and measured values of relative humidity for two sections. The maximum predicted errors for $RH_2$ were 7% and PPC was 5.1%. The maximum error in $RH_7$ was 10% and the corresponding PPC was 6.6%. The $RH_7$ values were underestimated between 9:00-10:30 a.m.

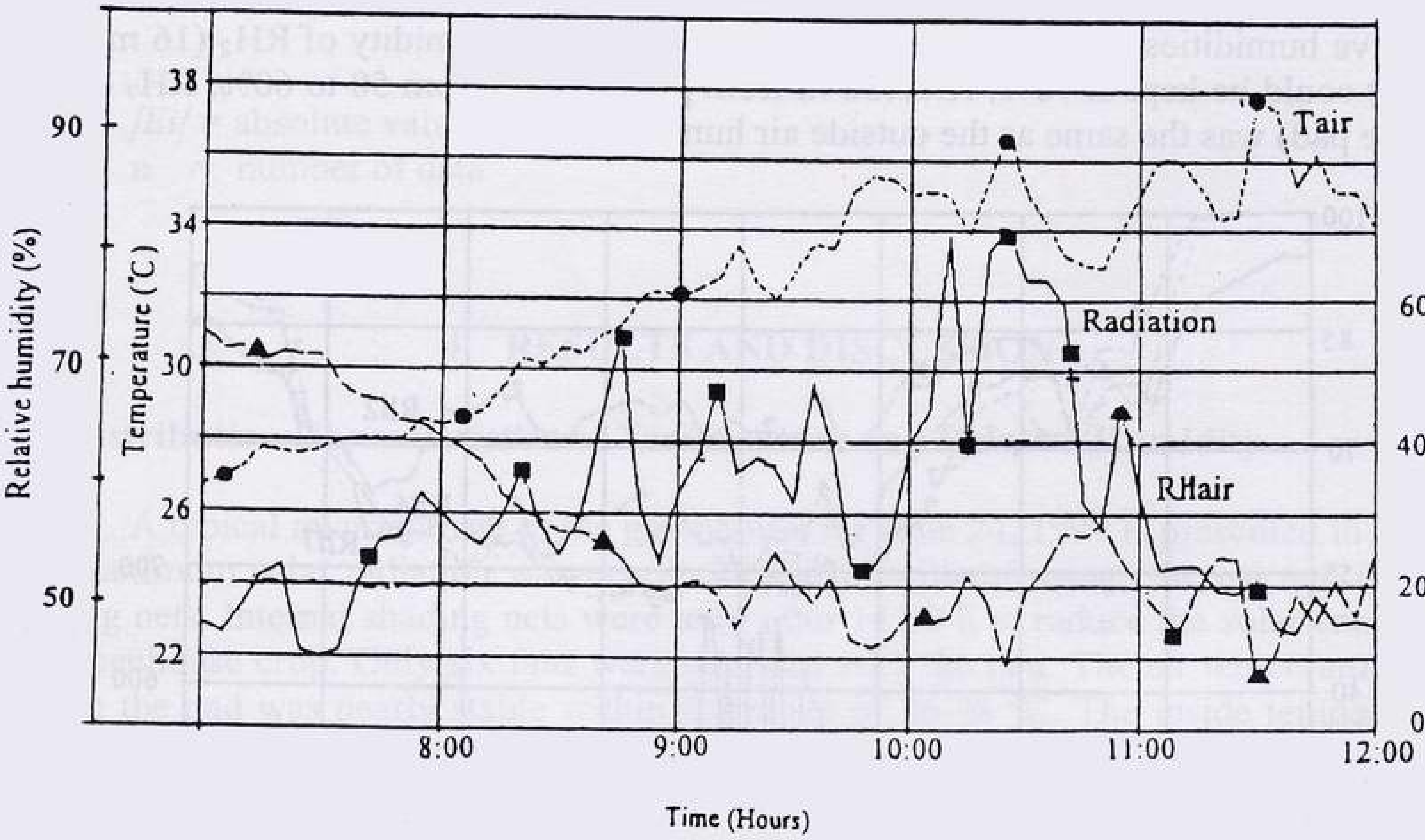


Fig. 5: Air temperature (●), relative humidity (▲), and solar radiation (■) in the greenhouse for June 28, 1999

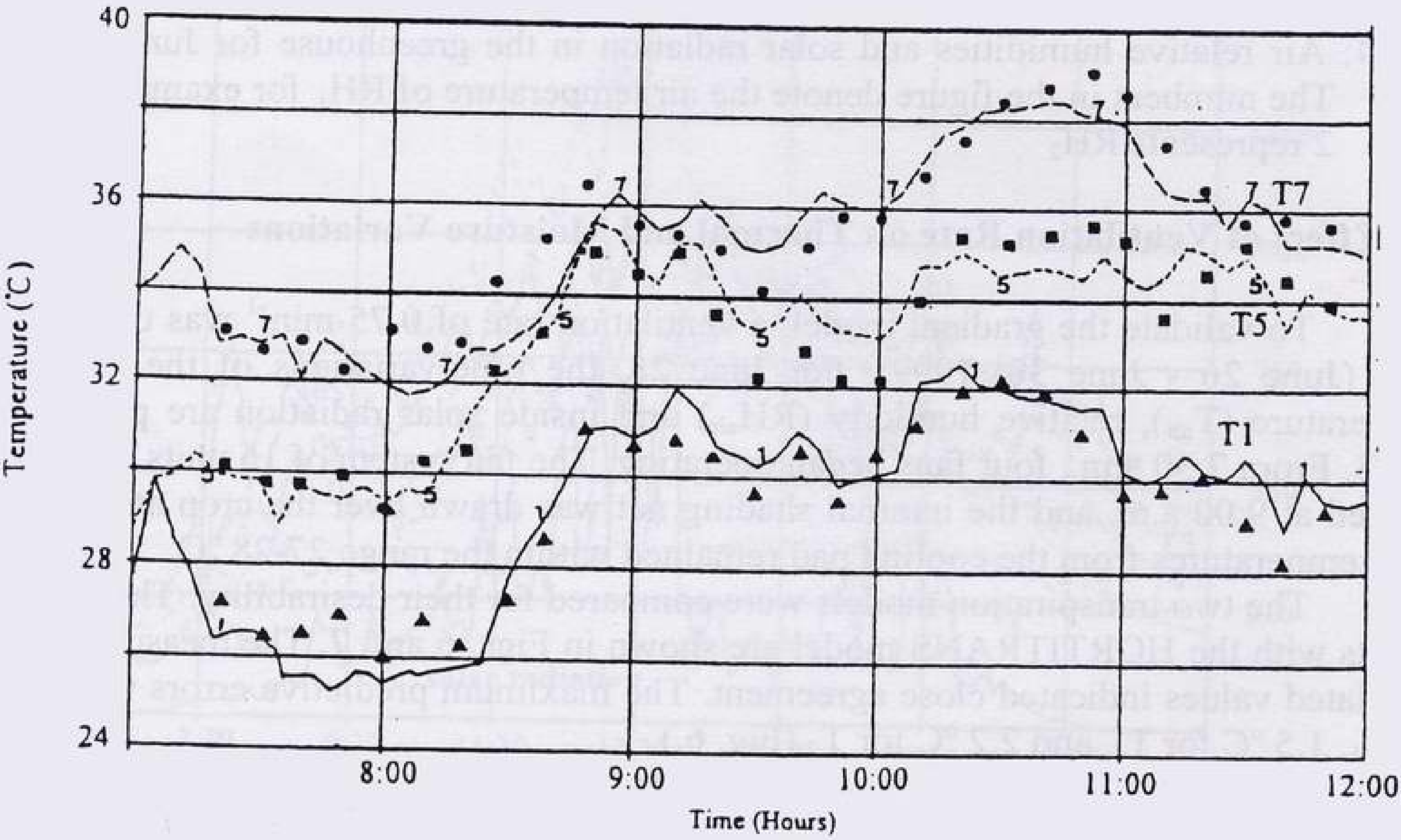


Fig. 6: Predicted and measured temperatures using the HORTITRANS model for June 28, 1999: (—) measured $T_1$, (▲) predicted $T_1$; (----) measured $T_5$, (■) predicted $T_5$; (− −) measured $T_7$, (●) predicted $T_7$

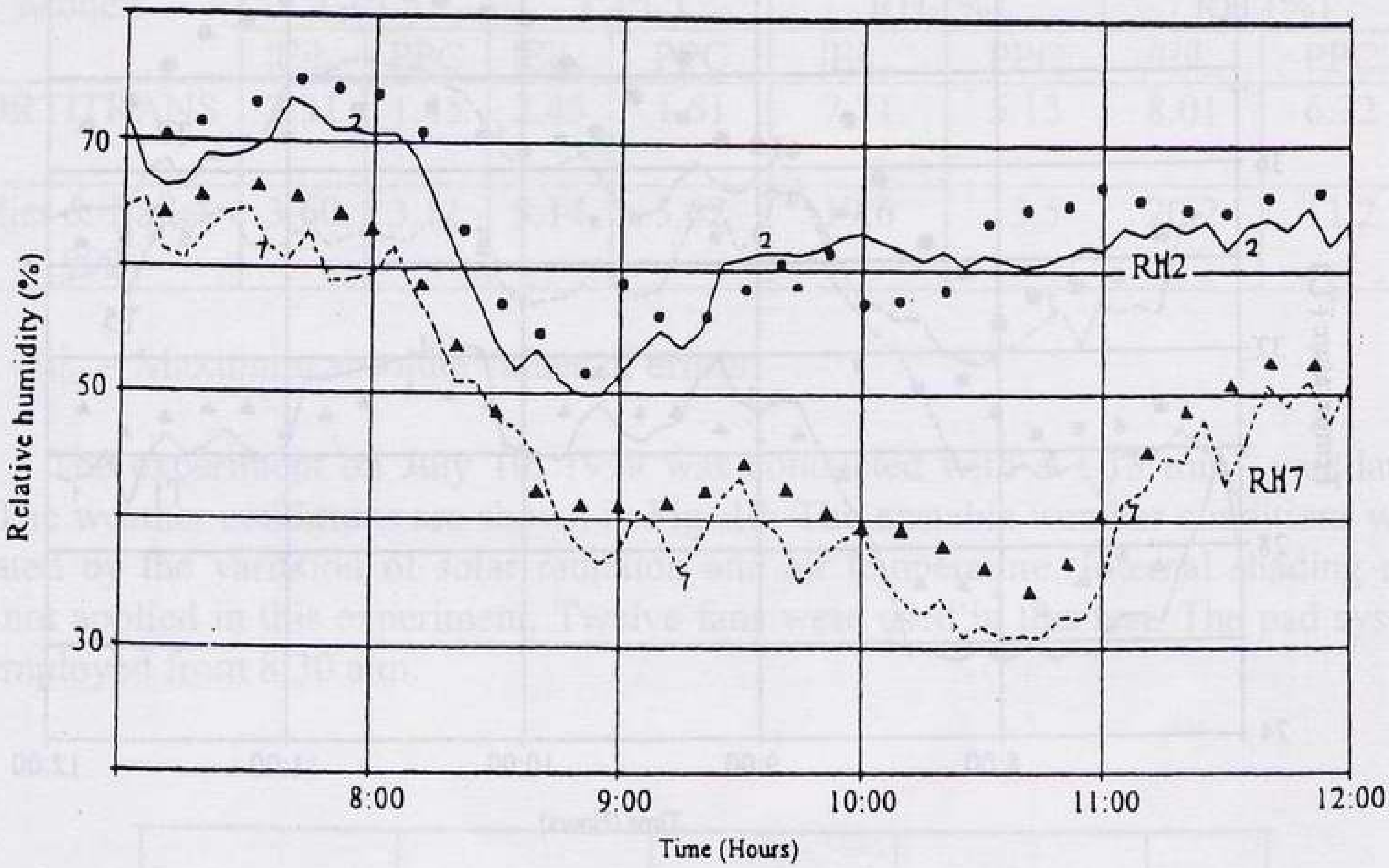


Fig. 7: Predicted and measured relative humidity using the HORTITRANS model for June 28, 1999: (—) measured $RH_2$, (●) predicted $RH_2$; (----) measured $RH_7$, (▲) predicted $RH_7$

The simulated results with the Jolliet & Bailey (1992) transpiration model are presented in Figs. 8 and 9. Only $T_1$ had a better predictive performance. $T_5$ and $T_7$ were overestimated with maximum errors of more than 3.5 ºC. The values of $RH_2$ and $RH_7$ were over-estimated with maximum errors of more than 20%. Similar results were obtained for the others sets of predicted data with the Jolliet & Bailey (1992) model. Therefore, this transpiration model was not considered for further studies.

The simulation interval for the two models was 5 minutes. Nine hour's data (9:00 a.m. – 5:00 p.m.) for five days were used to compare the validity of the two models. The results are listed in Table 2. The higher PPC values and maximum absolute values of errors indicated the inadequacy of the Jolliet & Bailey (1992) model, which could be explained by the parameters of this model. Only the solar radiation, relative humidity, saturated vapor pressure, and wind speed were considered. However, the LAI values were not incorporated into this model. The model was derived only by empirical function. The growth condition of tomatoes was not considered. However, HORTITRANS model was developed on the basis of plant physiology. The growth of tomatoes was considered with the LAI parameter, and so it had a better predictive performance than the Jolliet & Bailey (1992) model.

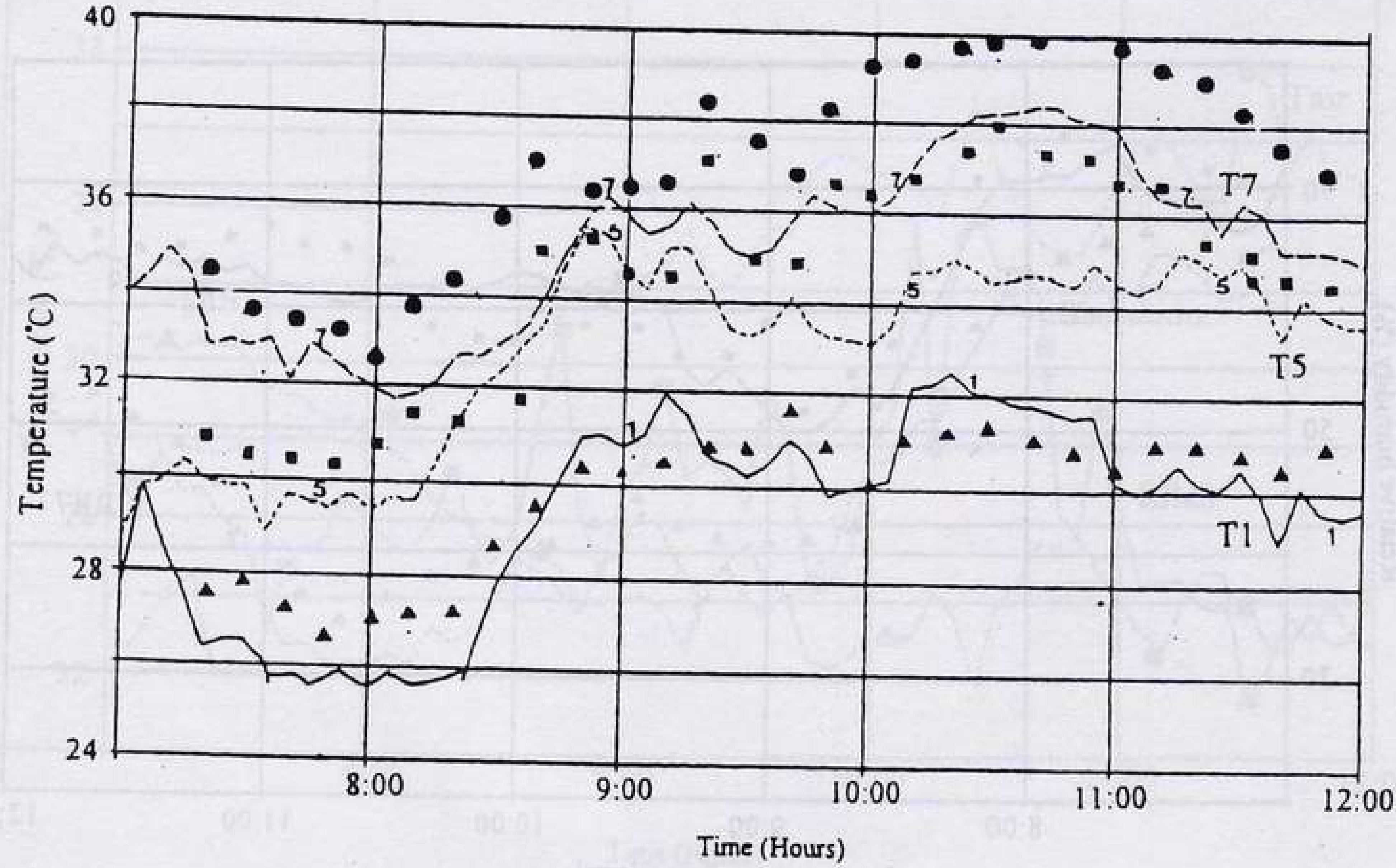


Fig. 8: Predicted and measured temperatures using the Jolliet & Bailey (1992) model for June 28, 1999: (—) measured $T_1$, (●) predicted $T_1$; (----) measured $T_5$, (■) predicted $T_5$; (− −) measured $T_7$, (n ) predicted $T_7$

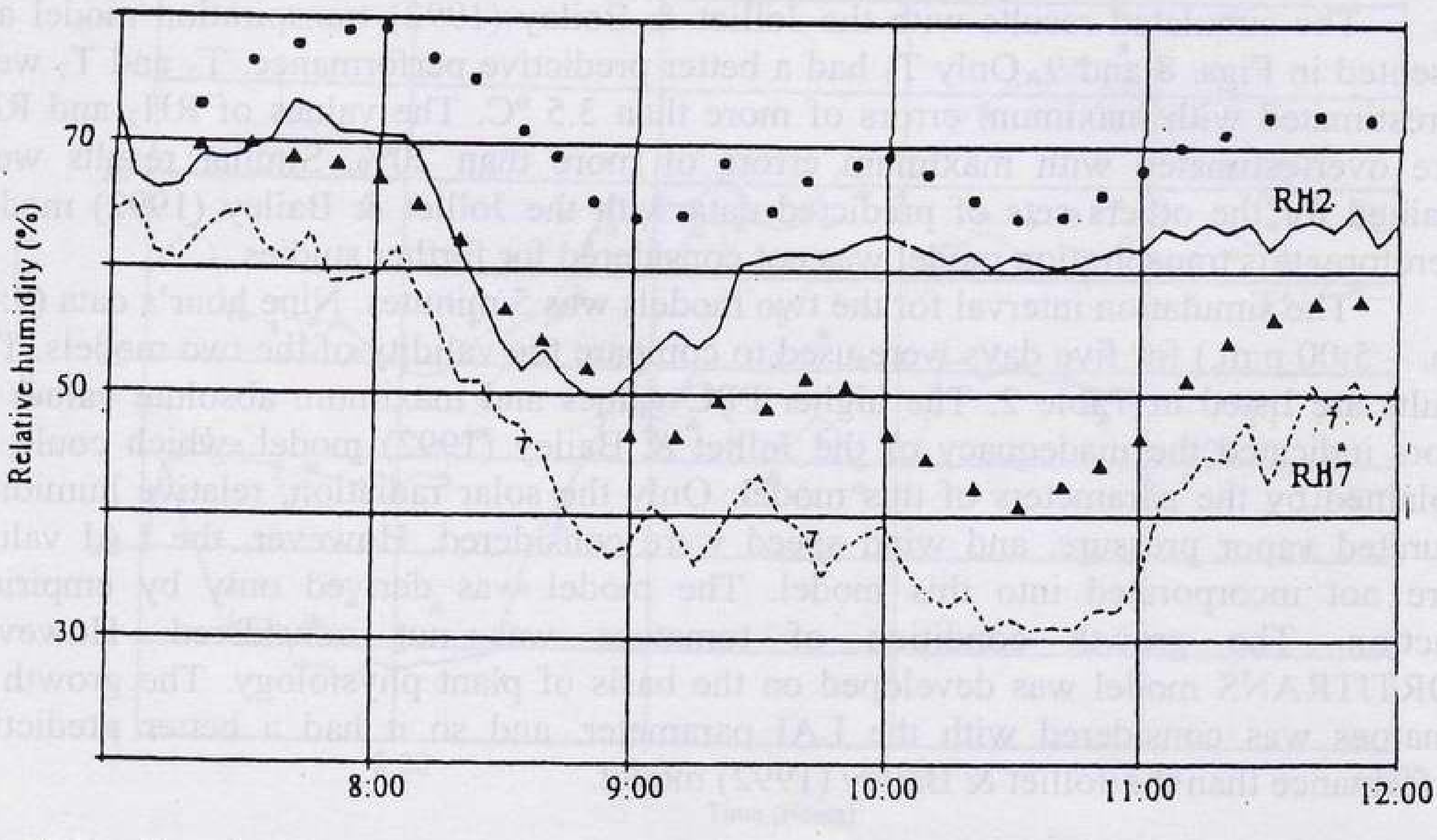


Fig. 9: Predicted and measured relative humidity using the Jolliet & Bailey (1992) model for June 28, 1999: (—) measured $RH_2$, (●) predicted $RH_2$; (----) measured $RH_7$, (▲) predicted $RH_7$

*Table 2: Comparison of the predictive ability of two transpiration models*

| Models | $T_1$(°C) | | $T_5$(°C) | | $RH_2$(%) | | $RH_7$(%) | |
|---|---|---|---|---|---|---|---|---|
| | $\|Ei\|_m$ | PPC | $\|Ei\|_m$ | PPC | $\|Ei\|_m$ | PPC | $/Ei\|_m$ | PPC |
| HORTITRANS | 2.31 | 1.45 | 2.45 | 1.61 | 7.71 | 5.13 | 8.01 | 6.32 |
| Jolliet & Bailey (1992) | 3.60 | 3.11 | 5.14 | 3.62 | 19.6 | 13.5 | 20.2 | 11.2 |

Note: $|Ei|_m$ = Maximum absolute values of errors

The experiment on July 10, 1999 was conducted with a 1.13 $min^{-1}$ ventilation rate. The weather conditions are shown in Fig. 10. The unstable weather conditions were indicated by the variation of solar radiation and air temperature. Internal shading nets were not applied in this experiment. Twelve fans were used in this test. The pad system was employed from 8:30 a.m.

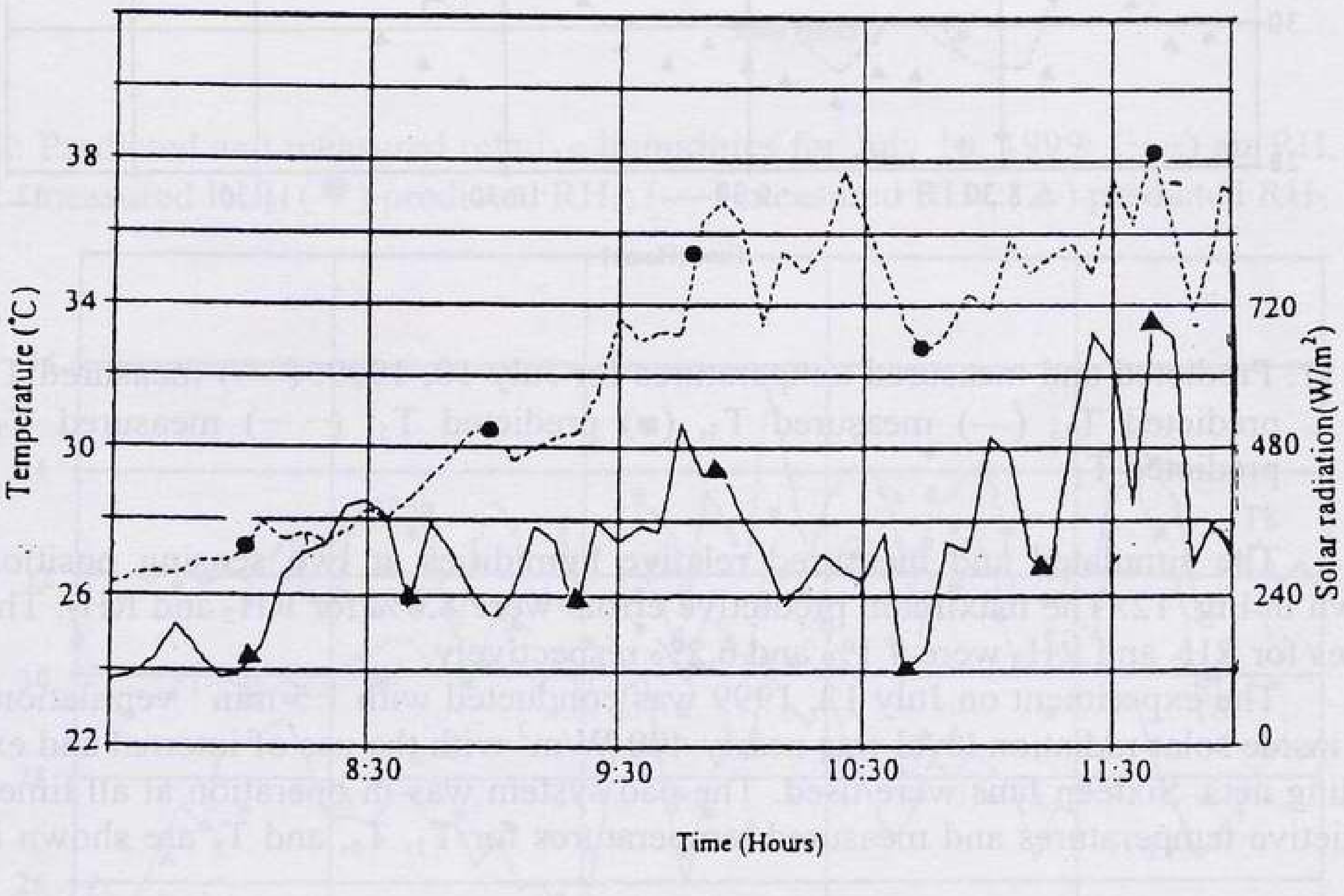


Fig. 10: Air temperature (●) and solar radiation (▲) in the greenhouse for July 10, 1999

The predictive results of temperature are presented in Fig. 11. The maximum difference between predicted and measured values of $T_1$ was lower than 2.0 °C. The largest differences between the observed and the simulated air temperatures occurred between 10:30 a.m. and 12:00 noon, as the solar radiation varied rapidly. The predictive errors for $T_5$ and $T_7$ were below 1.5 ºC except for the period 11:30 a.m. to 12:00 noon for $T_7$. The results could be explained by the rapid change of solar radiation during this period. The assumption of the steady-state model could not respond to the transient conditions of rapid variations of solar radiation and air temperatures.

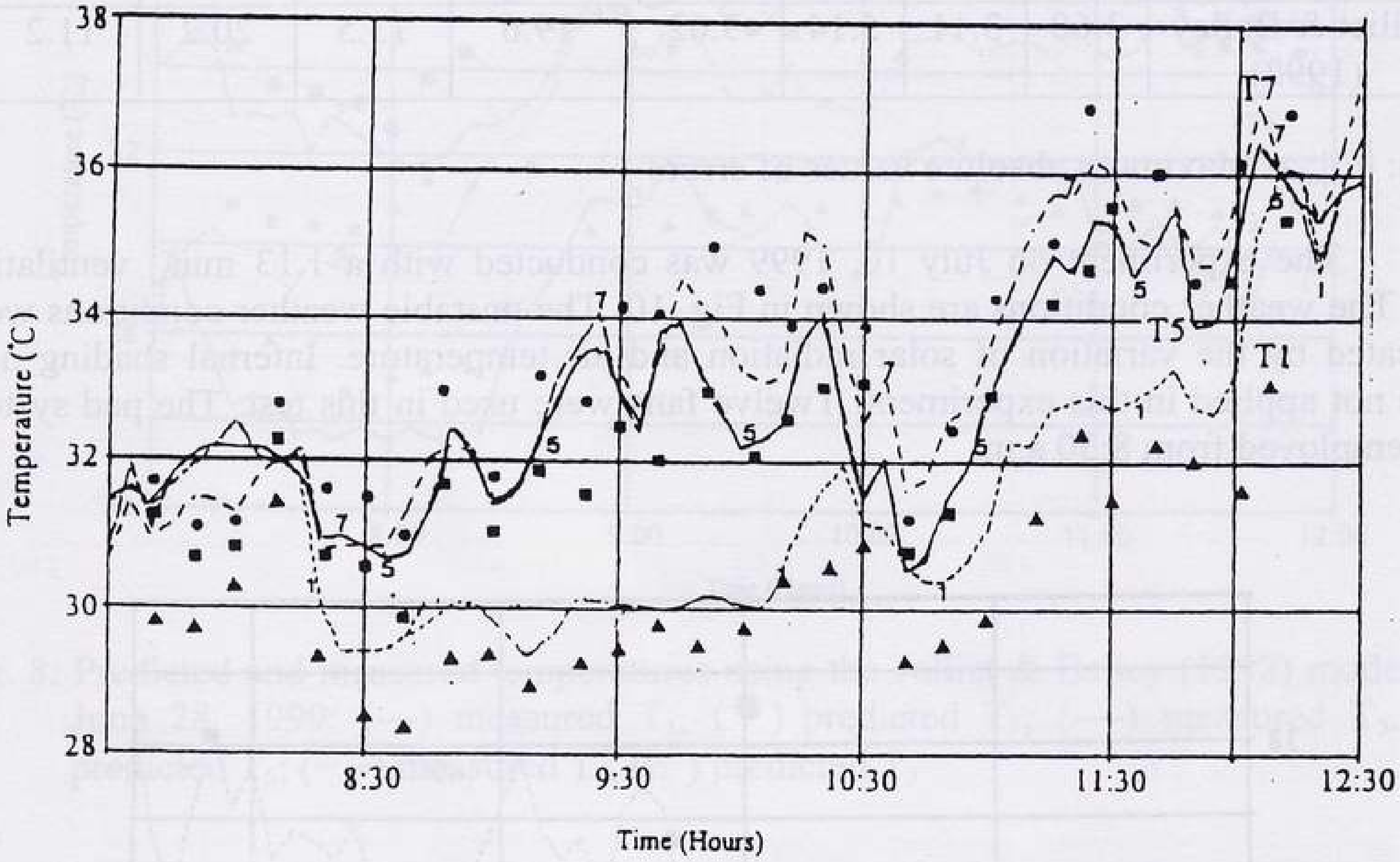


Fig. 11: Predicted and measured temperatures for July 10, 1999: (----) measured $T_1$, (▲) predicted $T_1$; (—) measured $T_5$, (■) predicted $T_5$; (– –) measured $T_7$, (●) predicted $T_7$

The simulated and measured relative humidities at two sensing positions are shown in Fig. 12. The maximum predictive errors were 8.0% for $RH_2$ and $RH_7$. The PPC values for $RH_2$ and $RH_7$ were 4.1% and 6.2% respectively.

The experiment on July 12, 1999 was conducted with 1.5 $min^{-1}$ ventilation rates. The inside solar radiation level was nearly 400 $W/m^2$ with the use of internal and external shading nets. Sixteen fans were used. The pad system was in operation at all times. The predictive temperatures and measured temperatures for $T_1$, $T_6$, and $T_8$ are shown in Fig. 13.

The difference between predictive and measured values of $T_1$ was lower than 2.0 ˚C. The predictive errors of $T_6$ and $T_8$ were less than 1.5 ˚C. Comparing the results with other conditions, the model had the best predictive performance at the higher ventilation rates.

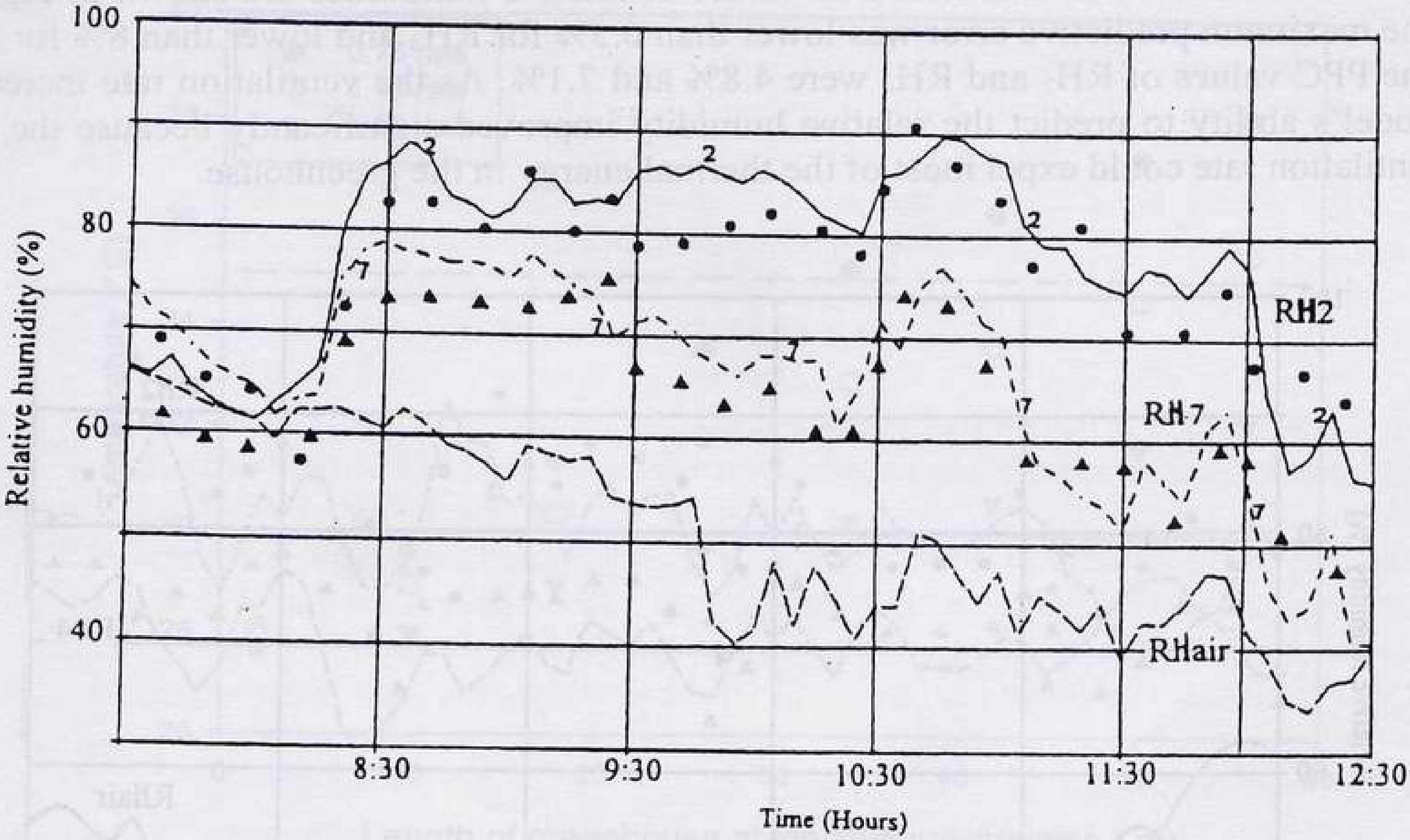


Fig. 12: Predicted and measured relative humidities for July 10, 1999: (– –) air RH, (—) measured $RH_2$, (●) predicted $RH_2$; (----) measured $RH_7$, (▲) predicted $RH_7$

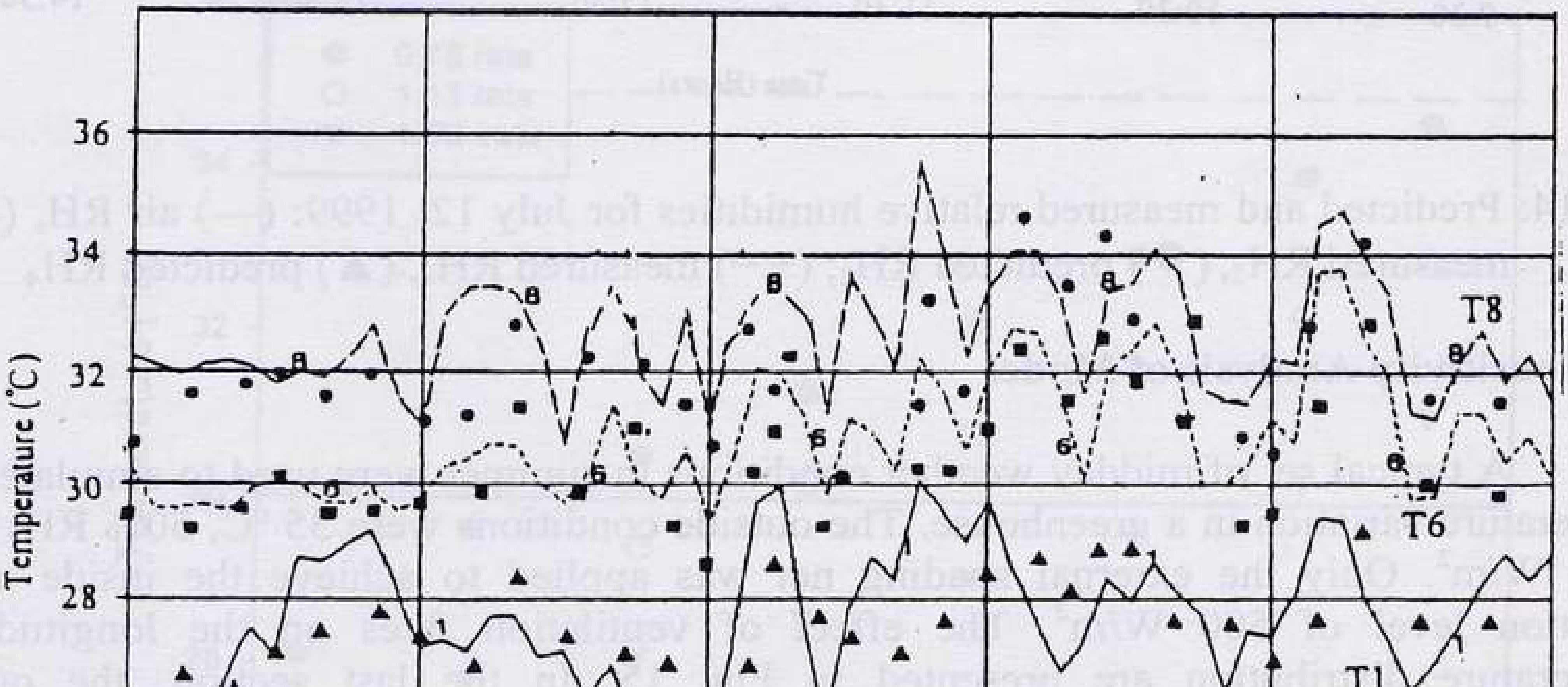


Fig. 13: Predicted and measured temperatures for July 12, 1999: (—) measured $T_1$, (▲) predicted $T_1$; (----) measured $T_6$, (■) predicted $T_6$; (– –) measured $T_8$, (●) predicted $T_8$

The simulated and measured values of relative humidities are shown in Fig. 14. The maximum predictive error was lower than 6.5% for $RH_4$ and lower than 8% for $RH_2$. The PPC values of $RH_2$ and $RH_4$ were 4.8% and 7.1%. As the ventilation rate increased, model's ability to predict the relative humidity improved significantly because the high ventilation rate could expel most of the thermal energy in the greenhouse.

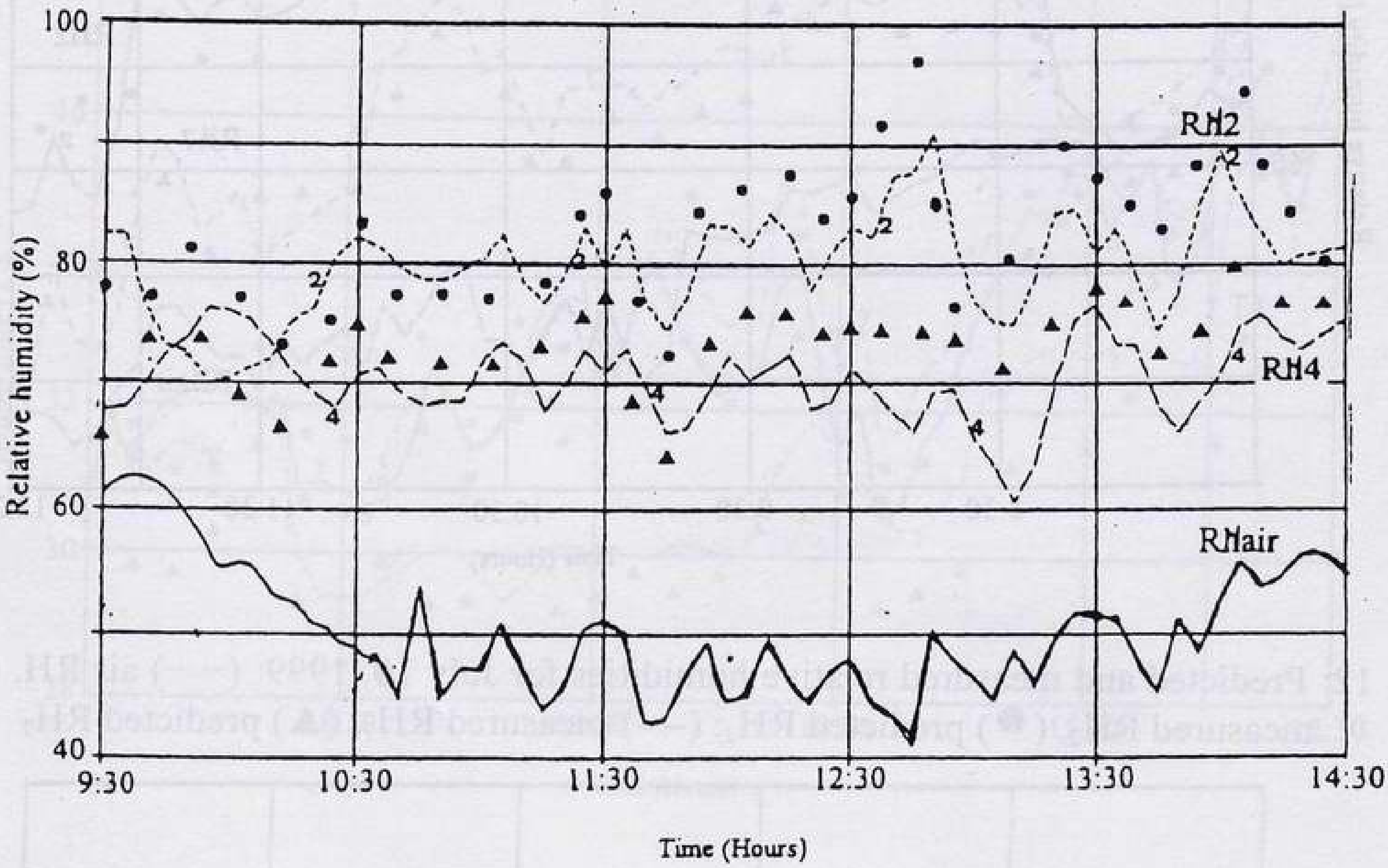


Fig. 14: Predicted and measured relative humidities for July 12, 1999: (—) air RH, (----) measured $RH_2$, (●) predicted $RH_2$; (– –) measured $RH_4$, (▲) predicted $RH_4$

### 4.3 Sensitivity Analysis of Model

A typical set of midday weather conditions in summer were used to simulate the temperature variation in a greenhouse. The outside conditions were 35 °C, 50% RH, and 1100 W/m$^2$. Only the external shading net was applied to achieve the inside solar radiation level of 500 W/m$^2$. The effect of ventilation rates on the longitudinal temperature distribution are presented in Fig. 15. In the last section, the outlet temperature with 0.75 ventilation rate was 38 °C. The highest rate of 1.5 min$^{-1}$ ventilation kept the outlet temperature at 31.5 °C.

As both internal and external shading nets were applied, the inside solar radiation was reduced to 200 W/m$^2$. The effect of the ventilation rate on the temperature along the length of the greenhouse is presented in Fig. 16. In the last section, 0.75 ventilation rate resulted in air temperature of 34.6 °C. However, the outlet air temperature could be kept at 30.6 °C for the 1.5 ventilation rate.

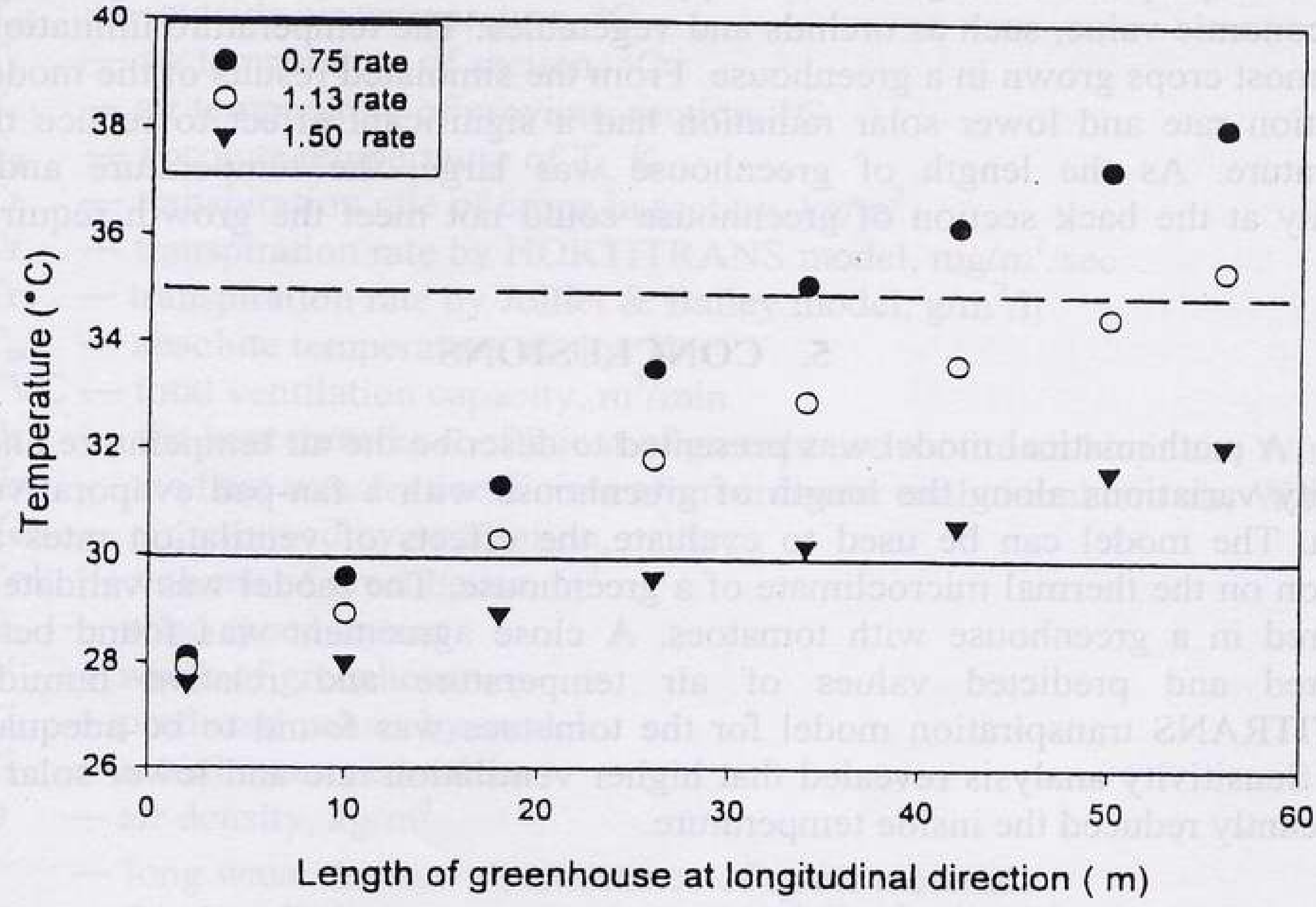


Fig. 15: The temperature distribution in the longitudinal direction at defferent ventilation rates under 500 W/m$^2$ solar solar radiation ($T_{air}$ =35 ºC, RH = 50%)

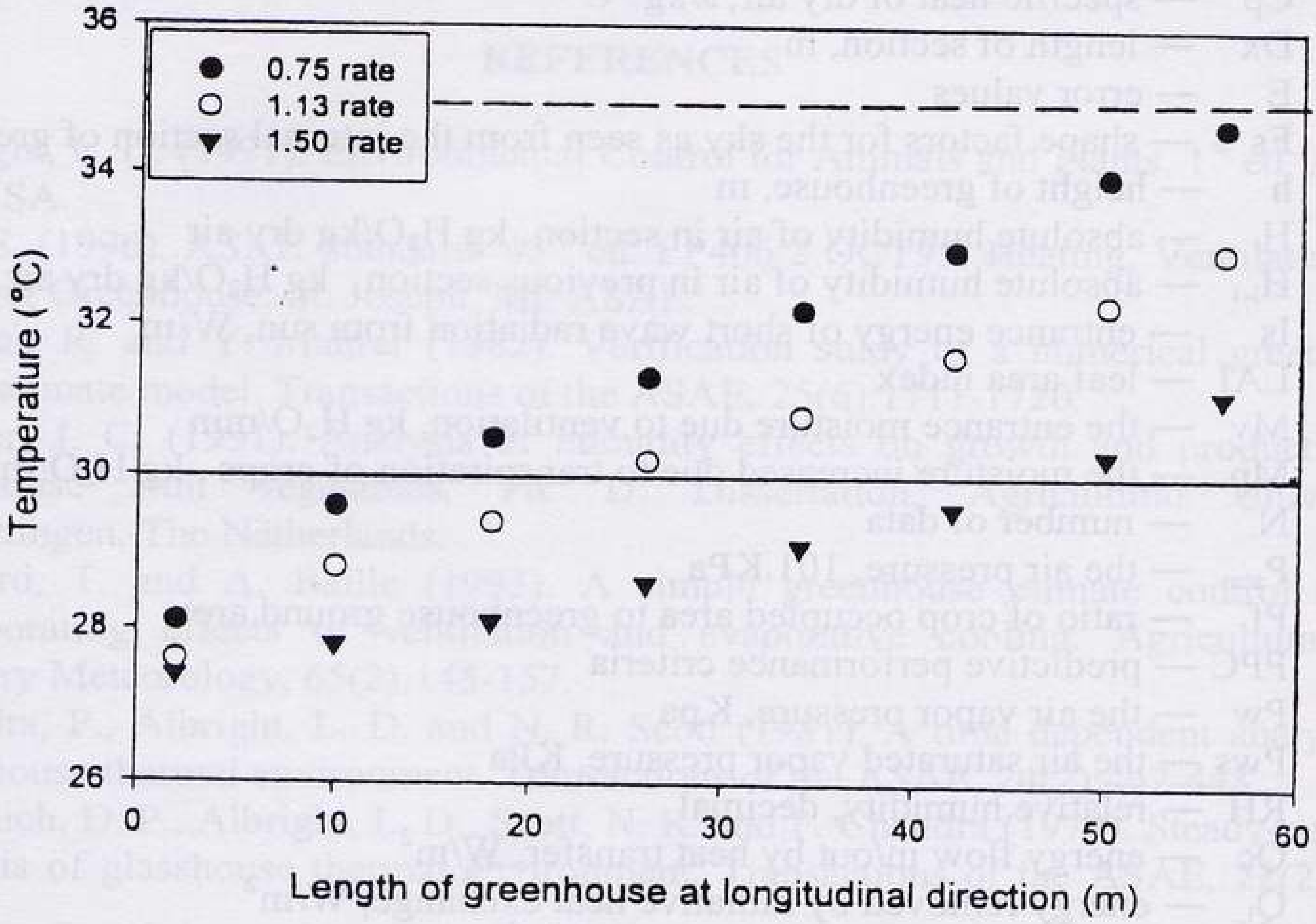


Fig. 16: The temperature distribution in the longitudinal direction at different ventilation rates under 200 W/m$^2$ solar solar radiation ($T_{air}$ =35 ºC, RH = 50%)

The purpose of the greenhouse application in the summer was to grow crops with high economic value, such as orchids and vegetables. The temperature limitation was 30 °C for most crops grown in a greenhouse. From the simulated results of the model, higher ventilation rate and lower solar radiation had a significant effect to reduce the inside temperature. As the length of greenhouse was large, the temperature and relative humidity at the back section of greenhouse could not meet the growth requirement of crops.

## 5. CONCLUSIONS

A mathematical model was presented to describe the air temperature and relative humidity variations along the length of greenhouse with a fan-pad evaporative cooling system. The model can be used to evaluate the effects of ventilation rates and solar radiation on the thermal microclimate of a greenhouse. The model was validated by data measured in a greenhouse with tomatoes. A close agreement was found between the measured and predicted values of air temperature and relative humidity. The HORTITRANS transpiration model for the tomatoes was found to be adequate in this study. Sensitivity analysis revealed that higher ventilation rate and lower solar radiation significantly reduced the inside temperature.

## NOTATIONS USED

Cp — specific heat of dry air, J/kg °C
Dx — length of section, m
E — error values
Fs — shape factors for the sky as seen from the internal section of greenhouse
h — height of greenhouse, m
$H_i$ — absolute humidity of air in section, kg $H_2O$/kg dry air
$H_{i-1}$ — absolute humidity of air in previous section, kg $H_2O$/kg dry air
Is — entrance energy of short wave radiation from sun, $W/m^2$
LAI — leaf area index
Mv — the entrance moisture due to ventilation, kg $H_2O$/min
Mp — the moisture increased due to transpiration of crops , kg $H_2O$/min
N — *number of data*
$P_{atm}$ — the air pressure, 101.KPa
Pf — ratio of crop occupied area to greenhouse ground area
PPC — predictive performance criteria
Pw — the air vapor pressure, Kpa
Pws — the air saturated vapor pressure, KPa
RH — relative humidity, decimal
Qc — energy flow in/out by heat transfer, $W/m^2$
$Q_l$ — energy removed by radiative heat exchange, $W/m^2$
Qp — transpiration energy by crops, $W/m^2$
Qs — solar radiation from sun, $W/m^2$

Qv — energy removed by ventilation, $W/m^2$
$T_{air}$ — outside air temperature, °C
$T_I$ — air temperature of section, °C
$T_{i-1}$ — air temperature of previous section, °C
$T_{ik}$ — absolute tempertaure of $T_i$, K
Tr — transpiration rate of crops in section, $kg/m^2$
$Tr_1$ — transpiration rate by HORTITRANS model, $mg/m^2/sec$
$Tr_2$ — transpiration rate by Jolliet & Bailey model, $g/m^2/h$
$T_{sky}$ — absolute temperature of sky, K
TVC — total ventilation capacity, $m^3/min$
Ur — the heat transfer coefficient of greenhouse roof to ambient air, $W/m^2.K$
Uw — the heat transfer coefficient of greenhouse wall to ambient air, $W/m^2.K$
Ven — air removed by ventilation, $m^3/sec$
Vol — volume of greenhouse, $m^3$
U — wind speed, m/sec
W — width of greenhouse, m
Xi — predicted values by model
Yi — actual measured values
$\rho$ — air density, $kg/m^3$
$\tau$ — long wave thermal transmittance of cover materials
$\sigma$ — Stephan-Boltzman constant, $5.67*10^{-8} W/m^2.K^4$
$\lambda$ — latent heat of water, kJ/Kg

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