Fan and Pad Evaporative Cooling System for Greenhouses: Evaluation of a Numerical and Analytical Model

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Abstract

An experimental greenhouse equipped with fan and pad evaporative cooling is analysed using two different models. The first one consists of a numerical simulation approach applying a commercial CFD code. The main aspects of evaporative cooling systems, in terms of heat and mass transfer and both the external and internal climatic conditions were integrated to set up the numerical model. The crop (tomato) was simulated using the equivalent porous medium approach by the addition of a momentum and energy source term. The temperature and humidity of incoming air, the operational characteristics of exhaust fans and the pressure drop occuring in the pad, were specified to set up the CFD model. The second model considers the greenhouse as a heat exchanger. Based on greenhouse structural characteristics, external climatic conditions, pad efficiency and ventilation rate, the air temperature distribution is predicted. The results, concerning the air temperature, provided both by numerical and analytical model, were validated by experimental measurements obtained at a height level of 1.2 m above the ground in the middle of the crop canopy. The correlation coefficient (R^2) between computational results and experimental data was at the order of 0.96 for the numerical model and 0.77 for the analytical one, with average percentage error of 3.5% and 7.6%, respectively. The analytical model proved to be a useful simple evaluation tool, but the numerical one provides a more accurate overview of the air flow in the greenhouse showing that fan and pad evaporative cooling system could be effectively parameterized in numerical terms, in order to improve system's efficiency.

INTRODUCTION

One of the most important issues in modern greenhouse cultivation is to extend the production season, in order to maximize the use of greenhouse equipment, extend the export season, increase the annual yield per unit area and increase the profitability. Nevertheless, in many Mediterranean greenhouses such a practice is limited because the cooling method used (mainly ventilation and shading) does not provide the desired conditions, especially during the hot summer months.

Natural ventilation and roof shading are the most common techniques. Ventilation reduces greenhouse overheating, but it may even enhance the risk of water stress because it often increases crop transpiration (Seginer, 1994). Kittas et al. (2001) reported that high ventilation rates were not, a priori, the best solution for alleviating crop stress in greenhouses during summer conditions. Shading screens mounted externally or internally, may be used to reduce radiation inside the greenhouse but the effective temperature reduction is not really proportional to the shading rate.

Evaporative systems for cooling greenhouses have been developed to provide the desired growing conditions in the greenhouse during the hot period of the year. The principle underlying direct evaporative cooling is the easy conversion of sensible to latent heat while unsaturated air is cooled by exposure to free and colder water, both thermal isolated from other influences. Various studies on evaporative cooling systems applied to horticulture, mainly fog and pad and fan systems, were already published, and among others, those by Montero et al. (1981, 1990) and Giacomelli et al. (1985). Most of these works analyse the thermodynamic efficiency of the system and its climatic effects. Seginer (1994) found that evaporative cooling systems are mainly effective when crop transpiration is low, and Fuchs (1993) reported that a highly transpiring crop combined with a proper ventilation rate is the most effective mechanism to keep leaf temperatures moderate. A theoretical study was conducted by Arbel et al. (1999) to evaluate an evaporative cooling system for greenhouses by installing uniformly distributed fog generating nozzles in the space over the plants. More recently, Willits (2003) proposed a numerical model to predict air and crop temperatures as a function of ventilation rate and external temperature. Kittas et al. (2003) presented and validated a model to predict temperature gradients in a large evaporative cooled greenhouse; and finally Fuchs et al. (2006) developed a numerical model based on energy balance equation which solved numerically.

The main advantage of fan and pad evaporative cooling system lies in its simplicity of operation and control and also in that it does not entail any risk of wetting the foliage. The main drawback is high cost and lack of uniformity of the climatic conditions which expressed with large temperature and humidity gradients along the greenhouse (from evaporative pads to exhaust fans). The amplitude of such gradients is affected by many factors such as the geometry of the greenhouse, the outside climate conditions, the ventilation rate and the flow rate of the water in the evaporative pad. In order to determine the influence of each parameter experimental investigations could be carried out, but these would be very expensive in time and money. Moreover, it is very difficult to give fairly identical and stable boundary conditions in a field experiment, due to unstable and unpredictable weather conditions. Numerical (Landsberg et al., 1979) or analytical models (Kittas et al., 2003; Willits, 2003; Fuchs et al., 2006) can be used as an alternative for this purpose. Recent progress in flow modelling using computational fluid dynamics is also a good alternative. Computational fluids dynamics is an advanced technique for design in engineering; it is increasingly being used to analyze greenhouse microclimate with respect to structural specifications (Boulard and Wang, 2002; Bartzanas et al., 2004).

Even the temperature gradients from pad to fans are well described by many researchers; we have pure knowledge about how the air flow, resulted by the negative pressure of a combined operation of different fans, affect the cooling efficiency, especially in large greenhouses and how the variable speed fans could be integrated to modern environmental control systems. The aim of the present study is using two different models, a numerical and analytical one, to simulate an experimental greenhouse equipped with fan and pad evaporative cooling system and to validate the results with experimental data obtained during experiments conducted the summer period of 2007 in Farm of Aristotle University of Thessaloniki.

MATERIALS AND METHODS

The experiments were carried out in a single-span, 8 m x 15 m greenhouse with an arched roof (Fig. 1); it's orientation was 30º from North and its position was at: Latitude 40.54 N, Longitude: 22.99 E. The greenhouse had FRP (fiberglass reinforced plastic) sidewalls and a tetrafluoroethylene copolymer 60 microns film roof attached to which a 40% shading net was applied. The gutter height was 2.6 m and the ridge height was 4.2

m. A cooling pad of width 6.0 m and height 1.0 m was positioned at the center of the north-wall, at 1.0 m above the ground. On the south wall, two single speed exhaust fans (propeller diameter of 0.76 m and 0.60 m and nominal propeller speed 590 rpm and 900 rpm, respectively), were placed at 1.32 m above the ground. The period of measurement, from August to September 2007, coincided with the nature stage of tomato crop cultivated using the common one stem technique (162 plants were transplanted at 19 May 2007). A more detailed description of the experimental set up could be found at Sapounas et al. (2008).

The Numerical Model

The experimental greenhouse was designed and meshed with the geometrical processor Gambit® as a 3D full scale model. The main characteristics of the experimental greenhouse, such as pad, fans, frame, covering materials and individual plants, were thoroughly integrated in the geometrical model (Fig. 1). The mesh consists of 1,010,812 hexahedral, pyramidal and wedge elements, result provided after many attempts in order to achieve grid independent results and acceptable time needed for the convergence. The grid quality, according to the EquiAngleSkew criterion (Fluent, 1998), was characterised as very good for the 92% of the cells.

The commercially available CFD code Fluent® (1998) uses a finite volume numerical scheme to solve the equations of conservation for the different transported quantities in the flow (mass, momentum, energy and water vapour concentration). The set up of simulation model mainly consisted of the definition of boundary conditions which based on experimental data obtained the specific time period. The RNG k-ε turbulence model (Launder and Spalding, 1974) was adopted to describe turbulent transport and buoyancy effect (Bartznas et al., 2007). The incoming air consisted of air and water vapour in a mass fraction corresponds to the relative humidity recorder by the humidity sensors. The crop was simulated using the equivalent porous medium approach (Boulard and Wang, 2002; Lee and Short, 2001), as a source term of both latent and sensible heat and as a sink momentum which dominated viscous and inertial resistance factors, calculated according to the procedure described in Fluent's manual (1998). The air density and the latent heat of vaporization of water were calculated according to the equations described by Fuchs et al. (2006). The pressure loss of each exhaust fan was represented by a 4th order polynomial function corresponds to its operational characteristics. The pad was simulated as porous medium with exponential pressure loss. Both functions of pressure losses were calculated according to technical specifications provided by the manufacturer. The main definitions of the simulation model are presented in Table 1.

The Analytical Model

The greenhouse is considered as heat exchanger. We suppose, for simplicity, that the fraction of the incident solar radiation responsible for sensible heat transfer is fixed and equal to $(1-a)$ where a is the fraction of the incident solar energy to the crop which is responsible for evapo-transpiration. In this case a equals to the correspondence latent heat calculated for the numerical model. Taking into account the structural characteristics of the experimental greenhouse, the heat balance, for a differential increment (dx) along the air flow, gives an equation for the internal greenhouse temperature T_{in} (°C), (Kittas, 2001).

$$V\rho C_{pa} dT_{in} = \left[\tau(1-a)Rg - \beta p_{ws,in}\right] l dx - KcL\left[T_{in}(x) - T_{ext}\right]dx \qquad (1)$$

where:

l = greenhouse width (perpendicular to the air flow), (m)
L = roof perimeter corresponding to the greenhouse width, (m)
V = rate of ventilation, (m^3 s^{-1})
R_g = outside global solar radiation, (W m^{-2})

p_{ws} = water vapour saturation partial pressure, (Pa)
K_c = heat loss coefficient of the greenhouse cover, (W m^{-2} K^{-1})
C_{pa} = specific heat of air, (J kg^{-1} K^{-1})
β = characteristic coefficient of the crop

A more detailed description of the analytical model could be found at Kittas et al. (2001).

RESULTS AND DISCUSSION

The numerical and analytical models were used to calculate the temperature distribution in the greenhouse environment under steady-state conditions, using the average values of 15 min time interval samples between the hours 13:00–15:00, as boundary conditions for both temperature and humidity of incoming air. The calculations concern the period 12–17/8/2007. During the experiments the outside air temperature ranged between 33.5–35.0°C and the global radiation between 706–886 W m^{-2}. In order for the results to be comparable, the same values, concerning the air temperature of incoming air and the ratio of radiation which converted to sensible heat β, were used for both models.

The results provided by both models for all the simulations cases were compared with experimental data in average terms for 14 points (direction from pad to fans, in the middle of the greenhouse 1.2 m above the ground), which correspond to the points were the experimental measurements were obtained. In general, the results were showing a qualitatively good agreement. The correlation coefficient (R^2) was 0.96 and 0.77 for numerical and analytical model, respectively, with absolute percentage error of 3.5% and 7.6% (Fig. 2). Both models underestimate the air temperature of the greenhouse, especially inside the crop canopy. The calculated average air temperature was 28.89ºC and 27.65ºC, for numerical and analytical model, respectively, while during the experiments the average temperature recorded was 29.95ºC. Neither models predict the small decrease of the air temperature, appears just before the fans, even though the numerical model performs better than the analytical one which is actually linear.

The results showed that the tested cooling system was able to keep the greenhouse temperature only few degrees below outside air temperature for the specific ventilation rate (0.0376 m^3 s^{-1} m^{-2}). Although the length of the greenhouse is not too long, important thermal gradients were observed in the direction from evaporative pads to exhaust fans. Figure 3 shows the air temperatures along greenhouse length at a cross section surface 1.2 m above the ground and in the middle of it. A thermal gradient in vertical direction was also predicted by the numerical model. The same phenomenon appeared during the experimental process. Part of the cold air passed below the canopy as the tomato crop was cultivated according to the one stem technique resulting, obviously, in a reduction of the crop resistance to the air flow. Even though the analytical model proved to be a useful simple evaluation tool, since it is linear, it is impossible to predict the vertical air temperature gradients. The numerical one provides more accurate overview of the air flow in the greenhouse showing that, fan and pad evaporative cooling system could be effectively parameterized in numerical terms, in order to improve system's efficiency. Furthermore, greater emphasis should be placed on the uniformity of conditions within the crop canopy rather than on the air temperature differences between the pad inlet and fan exhaust locations.

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Tables

Table 1. Main constant input values used in 3D CFD model.

Boundary element	Value	Unit
Viscous resistance	2.532	m^{-2}
Inertial resistance	1.92	m^{-1}
Porosity of the plant canopy	20	%
Sensible heat of plants	315.99 – 382.88	$W\ m^{-3}$
Latent heat of plants	$5.72 \times 10^{-5} – 5.98 \times 10^{-5}$	$kg\ m^{-3}\ s^{-1}$
Pad	porous media, power low model, C_o=-12.367, C_1=1.9385	
Exhaust fan-1	polyn. factors: 3.97,-81.4, 609.6, -1997.9, 2795.5	
Exhaust fan-2	polyn. factors: 0.71,-23.2, 276.4, -1442.7, 2859.4	
Mass fraction of water vapor	$8.54 \times 10^{-3} – 1.32 \times 10^{-2}$	

Figures

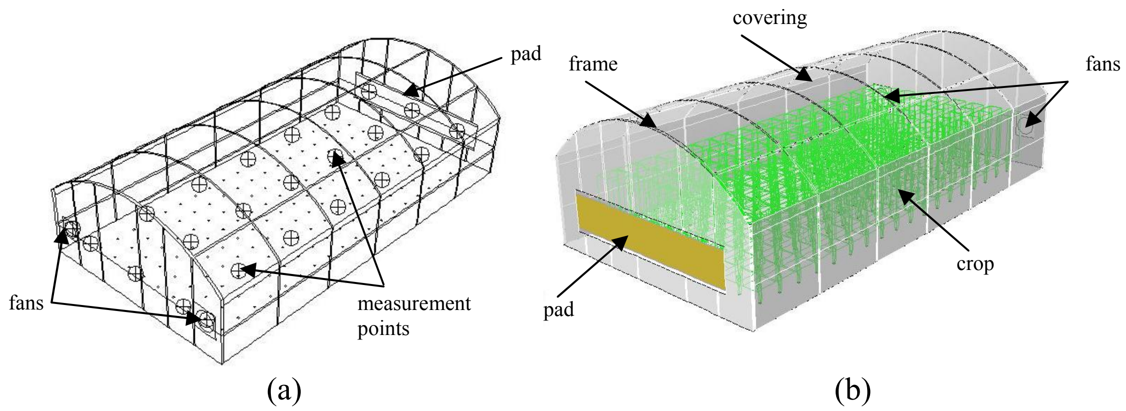

Fig. 1. (a) Wire-frame rendering of the greenhouse CAD model designed by Gambit® and experimental measurement points at level 1.2 m above ground. (b) 3D full scale simulation model of the experimental greenhouse with tomato crop.

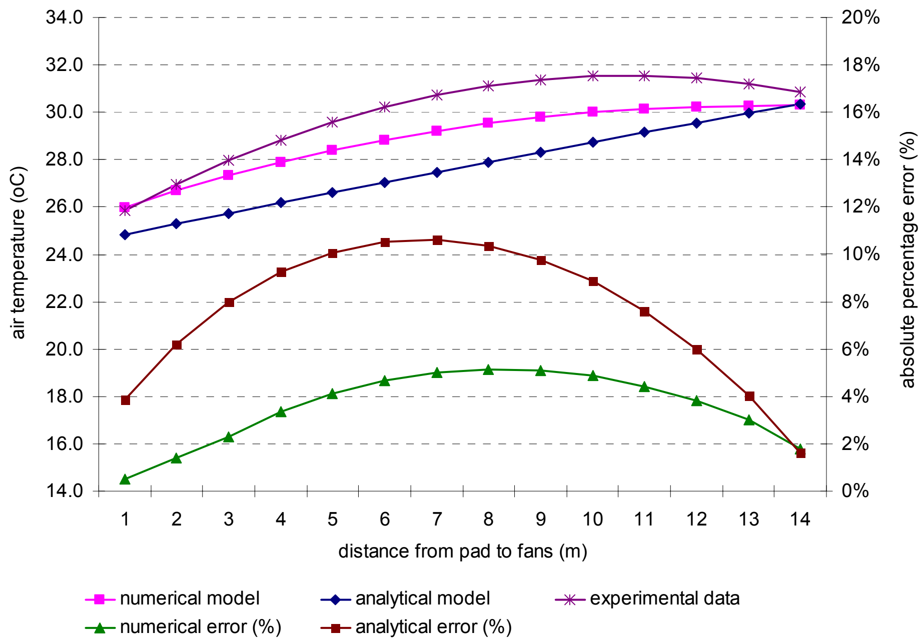

Fig. 2. XY scattering of air temperature inside the greenhouse. Comparison of the average values obtained by the numerical and analytical model with experimental data.

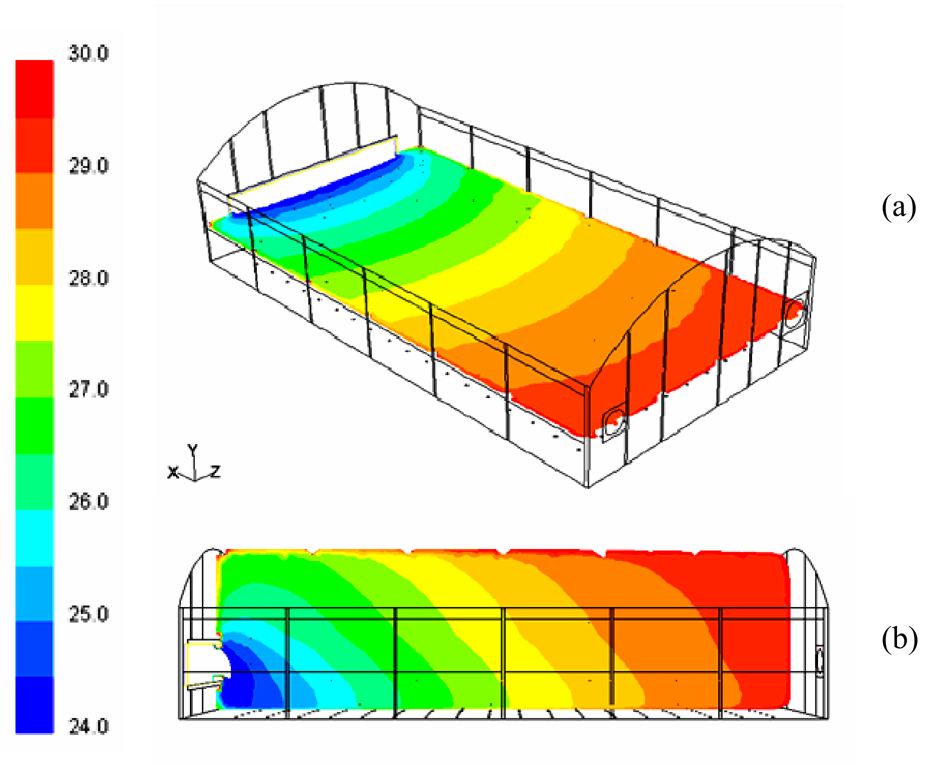

Fig. 3. Contours of air temperature a) plane surface 1.2 m above the ground and b) vertical surface in the middle of the greenhouse z=4.0 m (range 24–30ºC).