PAD-FAN SYSTEMS IN MEDITERRANEAN GREENHOUSES: DETERMINING OPTIMAL SETUP BY SONIC ANEMOMETRY

A. López, D. L. Valera, F. D. Molina-Aiz, A. Peña

ABSTRACT. *The present work studies the microclimate and airflow inside a greenhouse equipped with a pad-fan cooling system, analyzing several operational alternatives: three ventilation flow rates (18.1 $m^3 s^{-1}$, 20.2 to 23.5 $m^3 s^{-1}$, and 26.3 $m^3 s^{-1}$), and combining the medium flow rate with two interior fans or with a shading screen. The different airflow levels were obtained using a variable frequency drive (VFD) system to control ventilation fans (working at frequencies of 30, 40, and 50 Hz). The use of interior fans increased the velocity and intensity of the turbulent airflow, thus enhancing the mixing of air inside the greenhouse. The lowest fan frequency (30 Hz) reduced the system's cooling capacity, increasing both the horizontal and vertical temperature gradients compared to the results obtained for the frequencies of 40 and 50 Hz. The system's cooling capacity increased using the high-level ventilation flow rate or combining the pad-fan with a shading screen. In both situations, we obtained maximum temperature reductions of 3°C compared to the outside air. Different operational alternatives tested on sunny days show greater temperature reductions (4.4°C to 8.1°C) with respect to a similar naturally ventilated greenhouse (the most widespread type in the Mediterranean region). However, on cloudy days, when the outside relative humidity is high, the cooling capacity is more limited. Lack of system maintenance may lead to a considerable loss of efficiency (η), as this value fell from η = 0.82 when the system was installed to η = 0.65 one year later, at the time of this study. Estimated water consumption of the pad-fan system increases with the capacity to increase the water vapor content of the incoming air and with the volumetric flow rate.*

Keywords. *Cooling systems, Fans, Greenhouses, Pad, Temperature.*

The area of greenhouse cultivation is increasing worldwide. Approximately 20% of this area is concentrated in the Mediterranean basin, consisting for the most part of only rudimentary greenhouses (Baille, 2001). The Spanish Mediterranean coast has 45,000 ha of greenhouses (Castilla and Hernandez, 2005), and the southeast of the country, particularly the province of Almería, has the highest concentration of greenhouses in the world, with 30,000 ha, representing 4% of the total area (Molina-Aiz, 2010).

The climate of the province of Almería allows suitable conditions for crop development to be maintained in greenhouses during most of the year simply by employing a good system of natural ventilation. Evaporative cooling systems are only recommended during the hottest summer months, but the protected horticulture sector in southern Europe is currently facing strong international competition, mainly from areas such as the north of Africa, where production costs, including labor costs, are substantially lower. In order to improve the sustainability of greenhouse crops, it is necessary to improve the final quality of production, increase crop yield, and modify the periods of maximum production. These requirements have led to constant advances in greenhouse technology. As a result, evaporative cooling systems are being used in areas with high spring-summer temperatures, such as the Mediterranean basin. These systems reduce the temperature inside the greenhouse by increasing the humidity. Thus, by maintaining suitable hygrometric levels during the development stages of crops with little foliage and high evapotranspiration requirements, the transplant date of some autumn-winter crops can be brought forward to mid-August. As a result, the crop can be harvested at a time when the produce fetches higher prices (Franco et al., 2011).

Although technological innovations have been increasingly incorporated into Mediterranean greenhouses in recent years, few greenhouses are equipped with active systems that influence the microclimate. Rather, passive systems of natural ventilation tend to be employed by growers. It is therefore considered essential to study how evaporative cooling systems can be adapted to the climate of the Mediterranean region and to the characteristics of these greenhouses. The two most widely use systems are those that combine evaporative pads and exhaust fans (pad-fan systems), and those based on water mist (fog systems).

According to Sethi and Sharma (2007), pad-fan cooling systems are a highly effective means of reducing the air temperature inside greenhouses. They are capable of achieving temperature drops of 4°C to 6°C when used alone and of 4°C to 12°C when used together with shade screens. The main advantage of these systems is that they entail no risk of wetting the crop, whereas their main draw-

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back is the lack of uniformity in the climatic conditions inside the greenhouse (Arbel et al., 2003). Franco et al. (2010) studied the influence of water and air flows on the performance of corrugated cellulose pads, recommending airspeed intervals of 1 to 1.5 m s^{-1}, obtaining a pressure drop between 3.9 and 11.25 Pa, depending on the pad thickness and the water applied. Air saturation efficiency was between 64% and 70%, and the water evaporation rate was between 1.8 and 2.62 kg h^{-1} m^{-2} °C^{-1}.

As far as the above-mentioned lack of uniformity is concerned, pad-fan cooling systems tend to generate thermal gradients along the main direction of airflow, estimated at 0.13°C m^{-1} by Kittas et al. (2003) and at 0.27°C m^{-1} by López et al. (2010). Considerable vertical temperature gradients have also been observed, increasing with the distance from the evaporative pad (López et al., 2010). When these systems are used in combination with shade screens, the horizontal temperature gradient can be reduced by 18% (Willits and Peet, 2000). Compared to fog systems, pad-fan systems allow greater temperature reductions (up to 3.4°C) while at the same time improving the quality and yield of tomato crops (Willits and Li, 2005).

Sethi and Sharma (2007) concluded than none of the currently available technologies can be considered perfect, as none meet all the cooling requirements of greenhouses and inside crops. The selection and operation of a cooling system is based on various parameters, such as type of climate, crop to be grown, cost, maintenance, ease of operation, reliability, life of the system, dependence on electricity, etc. The most suitable technology for greenhouse cooling can therefore be said to be that which meets most of the desired conditions of the farmer to grow off-season crops in order to fetch maximum returns.

Arbel et al. (2003) suggested that future research should be aimed at characterizing the airflow generated inside the greenhouse. To do so, Li and Willits (2008) used a hot-wire anemometer, which only measures the absolute air velocity. In order to characterize the three orthogonal components of air velocity, three-dimensional sonic anemometry is required (López et al., 2010).

The present work follows from two previous studies of airflow inside a greenhouse equipped with a pad-fan cooling system (López et al., 2010) and inside a naturally ventilated multispan greenhouse (López et al., 2011). In the first study, we studied the effect of a well-developed tomato crop on the microclimate inside a greenhouse, comparing it with an empty greenhouse, both equipped with a cooling system working at a constant flow rate and without interior fans or shading screen. The evaporative pads were less effective for the empty greenhouse. The crop inside the greenhouse reduced the temperature difference between the coldest area (close to the pads) and the warmest area (at the exhaust fans) from 4.0°C for the empty greenhouse to 2.3°C with plants (López et al., 2010). The second study demonstrated that opening the roof vent to the windward side caused a combination of contrary wind and thermal effects, reducing the cooling capacity of the natural ventilation system (López et al., 2011).

Considering all of the above, the present work studies the microclimate and airflow in Mediterranean greenhouses with a view to improving their competitiveness. The greenhouses contained a well-developed tomato crop and were equipped with a pad-fan cooling system, and different operational factors were compared: three ventilation flow rates (18.1 m^3 s^{-1}, 20.2 to 23.5 m^3 s^{-1}, and 26.3 m^3 s^{-1}), and a combination of the medium flow rate with two interior fans and with a shading screen. The different ventilation levels were obtained using a variable frequency drive (VFD) system to control ventilation fans (working at frequencies of 30, 40, and 50 Hz, respectively). VFD control can reduce electricity consumption by about 64% compared to on-off operation (Teitel et al., 2008b).

Material and Methods

Experimental Setup

The experimental work took place in two multispan greenhouses (24 m × 45 m) located at the agricultural research farm belonging to the University of Almería (fig. 1) in southeastern Spain (36° 51′ N, 2° 16′ W, and 87 m a.s.l.). The roof ridge orientation is east-west, at an angle of 118° east of north. One greenhouse is equipped with a pad-fan cooling system (greenhouse 1-PF), while the other is naturally ventilated (greenhouse 2-NV). This second greenhouse was used only as a reference to analyze the microclimate improvement that a pad-fan system can produce. Natural ventilation is the main means of climate control in greenhouses in the Spanish province of Almería (Molina-Aiz, 2010) and in the Mediterranean region as a whole. The experimental greenhouses were divided into two similar sections by a polyethylene sheet fixed to a stainless steel structure, which allows us to study the interior microclimate of the two halves separately for other research work. The measurement tests were carried out in the entire length of greenhouse 1-PF (24 m × 45 m) with the cooling system, and in the eastern half of greenhouse 2-NV (24 m × 25 m) with the natural ventilation system. The greenhouses contained a tomato crop (*Solanum lycopersicum* L. cv. Salomee) with average heights of approximately 1.90 and 2.10 m for the first and last measurement tests, respectively, while the leaf area index (m^2 leaf m^{-2} ground) was about 3.00 and 3.15 for the same tests.

To prevent insects from entering the greenhouses, insect-proof screens were placed on all the vents. Greenhouse 1-PF was fitted with a screen of 10 × 16 threads cm^{-2} (0.47 porosity, 383.3 μm pore width, 792.8 μm pore height, and 242.5 μm thread diameter), while the screen in greenhouse 2-NV had 13 × 30 threads cm^{-2} (0.39 porosity, 164.6 μm pore width, 593.3 μm pore height, and 165.5 μm thread diameter). For the same purpose, four antechambers were installed at the four entrances of the greenhouses. The antechambers are equipped with fans that operate automatically every time someone enters the antechamber. Pests and other biotic contaminants are driven back by the blast of air.

Evaporative pads (Celdek, Munters AB, Kista, Sweden; 1.9 m × 40 m for the eastern half; 1.9 m × 22.5 m for the western half) are installed on the southern side of greenhouse 1-PF, and eight exhaust fans (EM50n-d-1-wp-wm,

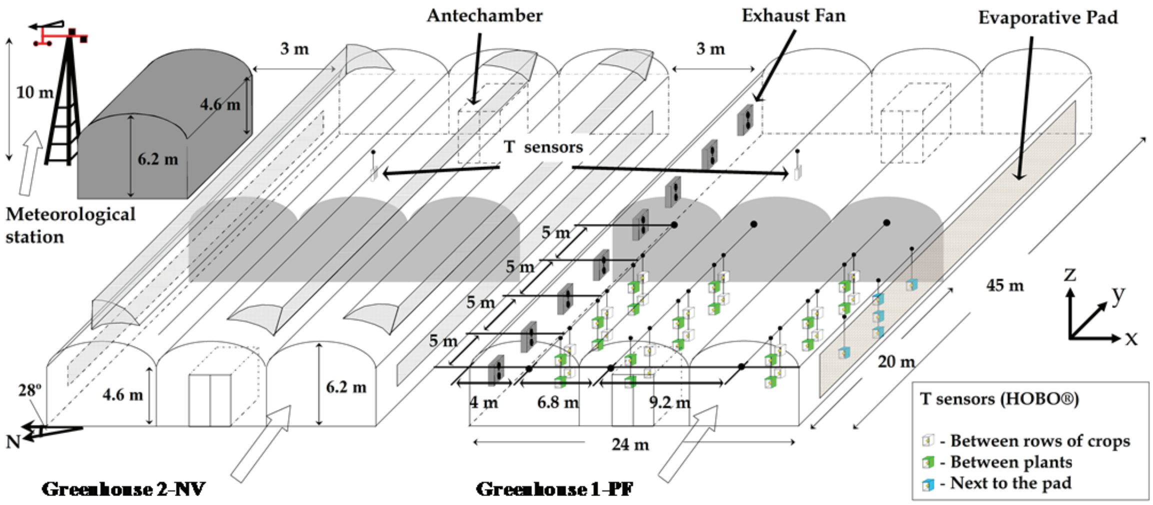

Figure 1. Sketch of the experimental greenhouses at the farm.

Munters Europe AB, Sollentuna, Sweden) are installed on the northern side. The exhaust fans have a nominal power of 735 W and a nominal fan speed of 368 rpm at the maximum electrical frequency of 50 Hz. The fan motors are connected to a variable frequency drive (VFD) unit (NXS 0022, Vacon Drives Ibérica S.A., Terrassa, Spain) with an output frequency of 0 to 320 Hz. Water is supplied by a pump (PRISMA25 4M, ESPA Innovative Solutions, Spain; 1.4 kW) from a warehouse located next to greenhouse 2-NV to two tanks located next to the evaporative pads; from there, the water is pumped to the pads by two submergible pumps (model DOC 76(T), ITT Corp., Lowara Srl., Italy; 0.55 kW).

Both greenhouses are equipped with shade screens (Aluminet 50-I, Polysack Plastic Industries, Ltd., Nir Yitzhak, Israel; 50% to 54% shading) and two interior fans (Euroemme EDC18, Munters AB, Kista, Sweden; 0.37 kW) located at a height of 2.5 m, i.e., the fan axis is at a height of 2.81 m. These fans were placed 4 m away from the side walls and were oriented across the width of the greenhouse (fig. 2). For the natural ventilation of greenhouse 2-NV, two side vents (1.05 m × 40 m) and three roof vents (1 m × 40 m) were opened. This represents 18.9% of the greenhouse ventilation surface (S_V/S_A, vent surface area/soil surface area), and 19.1% for the eastern half of the greenhouse.

Four types of measurement tests were carried out with two replications each, all between 11:30 h and 14:10 h (table 1). In greenhouse 1-PF, the system of evaporative pads was evaluated for three ventilation flow rates (18.1 m^3 s^{-1}, 20.2 to 23.5 m^3 s^{-1}, and 26.3 m^3 s^{-1}) and combining the medium flow rate with two interior fans and with shade screen. The different ventilation levels were obtained using the VFD system to control the exhaust fans at frequencies of 30 Hz (measurement test type D1), 40 Hz (two replications, A1 and A2), and 50 Hz (test type D2).

The medium level of ventilation (obtained with 40 Hz) was also tested in combination with the interior fans (two replications, B1 and B2) and in combination with the shading screen (two replications, C1 and C2). In greenhouse 2-NV (with natural ventilation), the interior fans were acti-

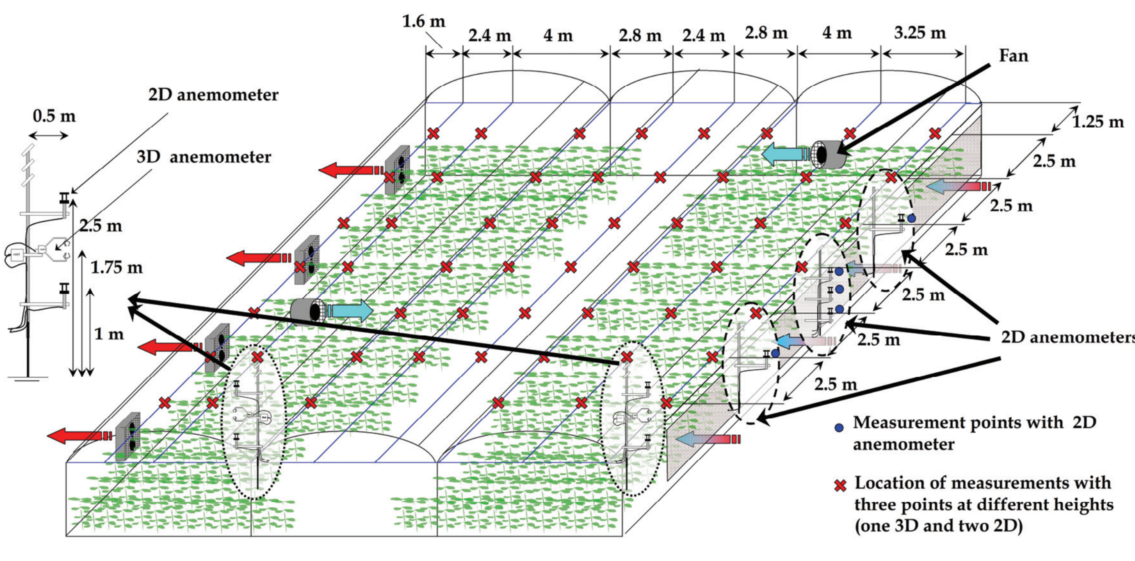

Figure 2. Measurement points with sonic anemometers inside the greenhouse.

Table 1. Greenhouse characteristics during the measurement tests.

		Test Types				
		A1 and A2	B1 and B2	C1 and C2	D1	D2
Greenhouse 1-PF	Exhaust fan frequency	40 Hz	40 Hz	40 Hz	30 Hz	50 Hz
	Water pressure head	5×10^5 Pa	5×10^5 Pa	5×10^5 Pa	5×10^5 Pa	5×10^5 Pa
	Shade screen	No	No	Yes	No	No
	Interior fans	No	Yes	No	No	No
Greenhouse 2-NV	Ventilation surface (S_V/S_A)	18.9%	18.9%	18.9%	18.9%	18.9%
	Shade screen	No	No	Yes	No	No
	Interior fans	No	Yes	No	No	No

vated during type B tests, and the shading screen was in place during type C tests.

Equipment and Instrumentation

Air velocity and temperature inside greenhouse 1-PF were measured with two 3D sonic anemometers (CSAT3, Campbell Scientific Spain S.L., Barcelona, Spain; resolution = 0.001 m s^{-1} and 0.002°C; accuracy = ±0.04 m s^{-1} and ±0.026°C). Air velocity was also measured with nine 2D sonic anemometers (Windsonic, Gill Instruments, Ltd., Lymington, U.K.; resolution = 0.01 m s^{-1}; accuracy = ±2%). Data were recorded by two microloggers (CR3000, Campbell Scientific Spain S.L.) with a data registration frequency of 10 Hz for the 3D sonic anemometers (Shilo et al., 2004) and 1 Hz for the 2D sonic anemometers (López et al., 2011).

Figure 2 shows the location of the airflow measurements recorded in the western half of greenhouse 1-PF. Five 2D anemometers were placed next to the evaporative pad. The pad was divided into three equal vertical spaces, and air velocity was measured at the center of each space. The central space was also divided into three equal horizontal spaces, and air velocity was measured at the center of the upper and lower spaces. To measure the air velocity inside the greenhouse, two devices were used, each with three sonic anemometers (fig. 2). Fifty-three measurement locations were established (fig. 2), and as the three anemometers were placed at different heights (one 3D anemometer at 1.75 m above the ground and two 2D anemometers at 1 m and 2.5 m), the total number of measurement points was 159. At each of the 159 measurement points, air velocity was measured for 3 min (López et al., 2011); this time period is a compromise between a shorter period that may reduce accuracy and a longer period that may increase the overall difference with regard to outside microclimate parameters (Molina-Aiz et al., 2009).

Air temperature and humidity inside the eastern halves of the two greenhouses were measured with an EKTRON II-c sensor unit (Hortimax S.L., Almería, Spain). Outside climatic conditions were recorded by a meteorological station at a height of 10 m located to the north of the greenhouse (fig. 1). The meteorological station included a BUTRON II sensor unit (Hortimax S.L.). The inside and outside sensor units were equipped with temperature sensors (Pt1000 IEC 751 class B, Vaisala Oyj, Helsinki, Finland) with a measurement range of -10°C to 60°C and an accuracy of ±0.6°C. Both sensor units were also equipped with capacitive humidity sensors (HUMICAP 180R, Vaisala Oyj) with a measurement range of 0% to 100% and an accuracy of ±3%.

Temperature and humidity inside the western half of greenhouse 1-PF were measured using 36 autonomous dataloggers (HOBO Pro Temp-HR U23-001, Onset Computer Corp., Bourne, Mass.) placed at heights of 1 and 2 m (only the EKTRON II-c sensor unit was used in greenhouse 2-NV). Another five sensors were placed next to the evaporative pad (fig. 1). These fixed devices allowed temperature measurement in a range of -40°C to 70°C with an accuracy of ±0.18°C and measurement of relative humidity of 0% to 100% with an accuracy of ±2.5%. They were all programmed to register data at 0.5 Hz and were protected against direct solar radiation with an open, passive solar shield.

Outside wind speed was measured with a Meteostation II (Hortimax S.L.) incorporating a cup anemometer with a measurement range of 0 to 40 m s^{-1}, accuracy of ±5%, and resolution of 0.01 m s^{-1}. Wind direction was measured with a wind vane (accuracy of ±5° and resolution of 1°). Solar radiation was measured using a Kipp Solari (Hortimax S.L.) sensor with a measurement range of 0 to 2000 W m^{-2}, accuracy of ±20 W m^{-2}, and resolution of 1 W m^{-2}.

To estimate air temperature (T_S) from the speed of sound measured by the 3D sonic anemometer, we must consider that the speed of sound in moist air depends on both temperature and humidity. From the inside humidity data recorded by the fixed sensors, we can obtain the specific humidity (q) and correct the sonic anemometer temperature (T_{SC}, °C) using the following expression (Tanny et al., 2008):

$$T_{SC} = \frac{T_S}{1 + 0.51q} \tag{1}$$

In May 2009, we made a field calibration comparing all inside temperature probes with an accurate instrument reading taken near the sensor being checked (Stum, 2006). The instrument used as a reference was a fine-wire type-E thermocouple (FW05, Campbell Scientific Spain S.L.) connected to the CR3000 micrologger. For air temperatures in the range of 12°C to 26°C, the accuracy of this sensor is ±0.42°C. Maximum differences in temperatures measured with the different probes were provided by the HOBO autonomous dataloggers, and in all cases they were less than 0.7°C. Differences between the air temperatures estimated by the anemometer and by the fine-wire thermocouple were less than 0.5°C.

Analyses

For air velocity u (m s^{-1}) and its components (u_x, u_y, and u_z; fig. 1), the mean air velocity measured over a period Δt

is (Cebeci, 2004):

$$\bar{u} = \frac{1}{\Delta t}\int_{t}^{t+\Delta t} u\, dt \quad (2)$$

We also calculated the average value of the two-dimensional resultant of air velocity in the *XY* plane (*l*) and in the *XZ* plane (*v*). In equation 3, $u(t)$ is the instantaneous air velocity, which can be expressed as the sum of the time-mean value $\bar{u}$ and a fluctuating component $u'(t)$ (Cebeci, 2004):

$$u(t) = \bar{u} + u'(t) \quad (3)$$

The variance of an air velocity over a period of time Δt is defined as (Cebeci, 2004):

$$\sigma^2 = \overline{u'^2} = \frac{1}{\Delta t}\int_{t}^{t+\Delta t} \left(u - \bar{u}\right)^2 dt \quad (4)$$

Turbulence intensity (*i*) is the standard deviation (σ) divided by mean local velocity ($\bar{u}$), so (Cebeci, 2004):

$$i = \frac{\sqrt{\overline{u'^2}}}{\bar{u}} = \frac{\sigma}{\bar{u}} \quad (5)$$

To analyze the effect of control of the exhaust fans and outside humidity on water consumption, we carried out a multiple regression analysis with Statgraphics Plus ver. 4.1 (Manugistics, Inc., Rockville, Md.).

Table 2. Outside climatic conditions.

Date (2009)	Test Type[a]	Climatic Condition[b]				
		u_o	θ	RH_o	T_o	R_g
8 July	A1	2.47 ±0.42	247 ±14	70 ±2	26.2 ±0.5	838 ±57
13 July	A2	2.95 ±0.40	252 ±9	56 ±1	29.0 ±0.2	791 ±60
10 July	B1	2.89 ±0.32	234 ±10	63 ±3	27.1 ±0.3	834 ±55
14 July	B2	2.94 ±1.64	181 ±111	86 ±2	23.6 ±1.2	82 ±35
11 July	C1	3.48 ±0.83	241 ±13	62 ±2	28.2 ±0.5	826 ±52
15 July	C2	2.13 ±0.39	237 ±14	59 ±2	28.8 ±0.5	829 ±51
16 July	D1	3.17 ±0.32	249 ±11	73 ±2	27.8 ±0.2	813 ±44
17 July	D2	2.08 ±0.33	214 ±9	58 ±4	27.9 ±0.6	870 ±52

[a] A = pad-fan and 40 Hz exhaust fan frequency, B = pad-fan and 40 Hz exhaust fan frequency plus interior fans, C = shaded pad-fan, and D = pad-fan with different exhaust fan frequencies (30 Hz for D1 and 50 Hz for D2).

[b] Means ± standard deviations: u_o = wind speed (m s^{-1}), θ = wind direction (degrees; direction perpendicular to the windows is 208° for a *Poniente* wind from southwest), RH_o = relative outside humidity (%), T_o = outside temperature (°C), and R_g = solar radiation (W m^{-2}).

Results and Discussion

The measurement tests were carried out under prevailing *Poniente* (southwest) winds, the most usual winds in the province of Almería. The outside climatic conditions remained relatively stable over the eight measurement tests (table 2). Outdoor temperatures and solar radiation were similar for all tests, with the exception of test type B2, when the sky was overcast and the temperature lower.

Air Velocity and Airflow Direction

The characteristics of airflow and levels of turbulence inside the eastern half of greenhouse 1-PF with the pad-fan system, under the same experimental conditions as those for the type A tests, are detailed by López et al. (2010). The present study assesses the system in the western half of the greenhouse. As the airflow was produced mechanically, its characteristics were very similar to those in the above-mentioned study.

Figure 3 shows the two-dimensional resultants of air velocity on the *XY* plane (*l*) and frequency histograms of velocity directions (depicted as polar plots). In greenhouse 1-PF, the air has to pass through the evaporative pads, cross the greenhouse, and exit through the exhaust fans. The presence of the antechamber on the western end of the greenhouse has a major impact on air velocity and direction. The air direction and the fact that there is little mixing of inside air are due to the temperature gradients, which are characteristic of pad-fan cooling systems (López et al., 2010). Mean air velocity inside the greenhouse for all tests was within the optimum range of 0.5 to 0.7 m s^{-1}. At velocities greater than 1 m s^{-1}, crop growth is negatively affected (ASHRAE, 2009), and this velocity was only recorded in the zones closest to the interior fans in the type B tests (figs. 3c and 3d). Mean air velocity on leaving the pad was below the recommended value of 1.27 m s^{-1} (*ASABE Standards*, 2008a): 0.61 to 0.71 m s^{-1} at a fan frequency of 40 Hz, falling to 0.54 at 30 Hz, and reaching 0.74 m s^{-1} at 50 Hz.

The use of a shading screen in greenhouse 1-PF does not seem to have a notable effect on the airflow (fig. 3b), at least at the height at which air velocity was measured inside the greenhouse. On the other hand, the use of fans inside greenhouse 1-PF clearly affects the airflow. By installing two fans oriented in opposing directions, a circular air current is generated, increasing the mixing of the inside air. This effect increases with height and is more notable at 2.5 m (fig. 3d), near the plane of the fan axis (2.81 m), than at 1.75 m (fig. 3c). However, the use of these interior fans does not affect the volume of air entering through the evaporative pad (measurements taken at heights of 1 m and 1.75 m).

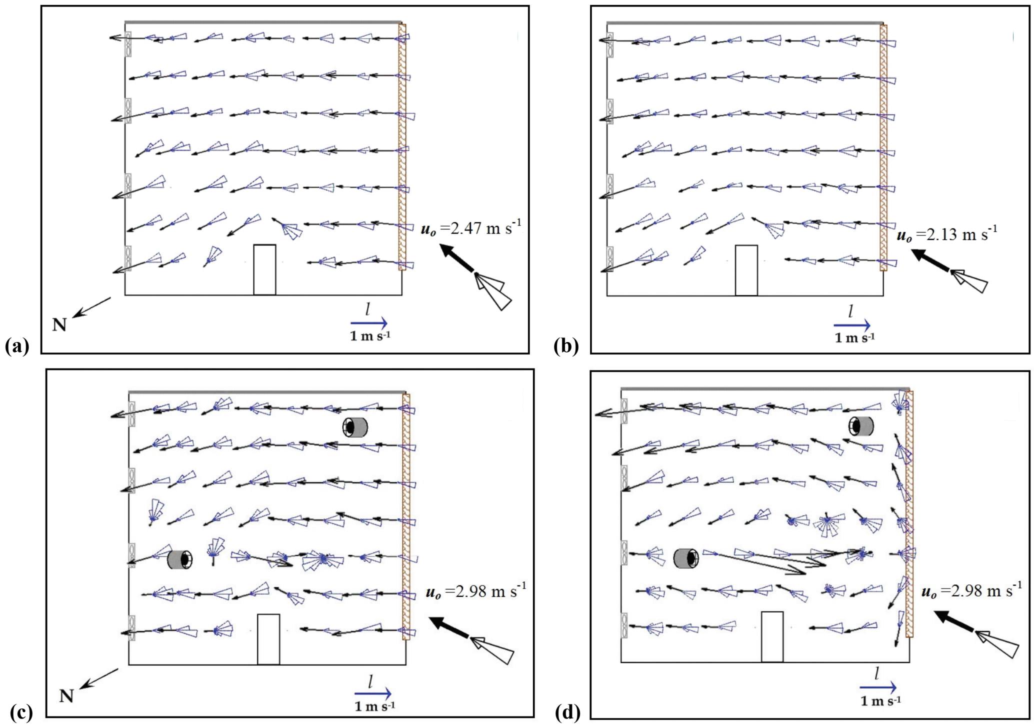

Figure 3. Two-dimensional resultants of air velocity in the *XY* plane (l) and polar plots of airflow direction in (a) test type A1 at 1.75 m height, (b) test type C2 at 1.75 m height, and test type B1 at (c) 1.75 m and (d) 2.5 m heights (test type A = pad-fan and 40 Hz exhaust fan frequency, test type B = pad-fan and 40 Hz exhaust fan frequency plus interior fans, test type C = shaded pad-fan and, test type D = pad-fan with different exhaust fan frequencies, 30 Hz for D1 and 50 Hz for D2).

From the values of air velocity obtained at different points inside the greenhouse (excluding those closest to the vents, pad, and exhaust fans), we calculated in table 3 the heterogeneity of the horizontal (l) and vertical (v) projections of air velocity, dividing the standard deviations of these values (σ_l and σ_v) by their mean values (Kittas et al., 2008).

Under normal conditions of use of the pad-fan system at the medium level of ventilation (tests A1 and A2), airflow was more homogeneous at greater heights (lower values of σ_l/l in table 3). When the two interior fans were operating inside the greenhouse (tests B1 and B2), we can observe that the air mixed better with height, particularly at 2.5 m (higher values of σ_l/l) close to the fan axis at 2.81 m, where the influence of the fan is greatest. However, the use of shading screens (tests C1 and C2) or reduction (test D1) and increase (D2) of ventilation level did not seem to affect the heterogeneity of l (type C tests).

Another indication of the increased mixing of air inside the greenhouse when the interior fans were functioning is provided by the increase in the mean value of turbulence intensity for the three orthogonal components of airflow (table 3). The values of turbulence intensity recorded in the present work (table 3) are lower than those observed by other authors in naturally ventilated greenhouses. In a greenhouse with a mesh cover (83% porosity), Tanny et al. (2006) obtained values of i = 0.2 to 0.8 (u_o < 0.5 m s^{-1}). Teitel et al. (2008a) obtained values of turbulence intensity ranging between 0.53 and 7.69 close to the vents of a monospan greenhouse, whereas values measured by Molina-Aiz et al. (2009) in an Almería-type greenhouse ranged between 0.34 and 2.74.

The fact that the levels of turbulence observed in the present study are lower in comparison to those observed in conditions of natural ventilation may be explained in two ways: (1) incoming air is obliged to pass through two porous media (the evaporative pad and the insect-proof screen installed across the greenhouse opening), which have a sta-

Table 3. Heterogeneity of the horizontal (σ_l/l) and vertical (σ_v/v) projections of air velocity, and mean values of turbulence intensity inside greenhouse 1-PF: i_x = longitudinal component, i_y = transversal component, and i_z = vertical component (means ± standard deviations).

Variable	Measurement Height	Test Type[a]							
		A1	A2	B1	B2	C1	C2	D1	D2
σ_l/l	1.00 m	0.41	0.45	0.54	0.61	0.46	0.46	0.44	0.44
	1.75 m	0.37	0.38	0.48	0.52	0.38	0.34	0.35	0.37
	2.50 m	0.28	0.26	0.77	0.89	0.25	0.25	0.35	0.24
σ_v/v	1.75 m	0.38	0.39	0.49	0.53	0.39	0.36	0.37	0.38
i_x	1.75 m	0.19 ±0.06	0.19 ±0.06	0.31 ±0.15	0.35 ±0.19	0.20 ±0.06	0.18 ±0.06	0.20 ±0.05	0.20 ±0.05
i_y	1.75 m	0.16 ±0.04	0.17 ±0.04	0.24 ±0.11	0.27 ±0.14	0.17 ±0.03	0.16 ±0.03	0.17 ±0.03	0.17 ±0.03
i_z	1.75 m	0.15 ±0.05	0.15 ±0.40	0.23 ±0.11	0.26 ±0.15	0.15 ±0.05	0.14 ±0.04	0.17 ±0.04	0.17 ±0.40

[a] A = pad-fan and 40 Hz exhaust fan frequency, B = pad-fan and 40 Hz exhaust fan frequency plus interior fans, C = shaded pad-fan, and D = pad-fan with different exhaust fan frequencies (30 Hz for D1 and 50 Hz for D2).

Table 4. Volumetric flow rate (*G*) and air exchange rate (*R*) calculated for the different tests.

Variable	Test Type[a]							
	A1	A2	B1	B2	C1	C2	D1	D2
G ($m^3\ s^{-1}$)	21.9	22.4	20.8	21.2	23.5	20.2	18.1	26.3
R ($m^3\ s^{-1}\ m^{-2}$)	0.046	0.047	0.043	0.044	0.049	0.042	0.038	0.055
R (h^{-1})	29.4	30.1	27.9	28.4	31.5	27.1	24.3	35.3

[a] A = pad-fan and 40 Hz exhaust fan frequency, B = pad-fan and 40 Hz exhaust fan frequency plus interior fans, C = shaded pad-fan, and D = pad-fan with different exhaust fan frequencies (30 Hz for D1 and 50 Hz for D2).

bilizing effect on the flow (Fang, 1997); (2) currents of air generated mechanically at constant velocity are characterized by lower levels of turbulence than natural currents (Ouyang et al., 2006). Low levels of turbulence inside the experimental greenhouse indicate that the airflow is less likely to mix or to transport heat and water vapor (Tan-Atichat et al., 1982) than in naturally ventilated greenhouses. The characteristics of mechanically generated airflow using this type of device have a direct impact on temperature distribution inside the greenhouse and give rise to considerable horizontal temperature gradients (López et al., 2010).

As turbulence intensity was less for the vertical component (i_z) than for the horizontal (i_x) and transversal (i_y) components, air did not mix well on the vertical plane, which led to greater vertical temperature gradients (see the Interior Microclimate section).

Estimation of Air Exchange Rates

Ventilation is usually characterized by the air renewal rate, which is defined as the ratio of the total volume of fresh air supplied in 1 h to the greenhouse volume (Mashonjowa et al., 2010). The air exchange rates (R, h^{-1}) for greenhouse 1-PF (table 4) were calculated by dividing the volumetric flow rate (G_p, $m^3\ s^{-1}$) through the pad by the greenhouse volume (2682 m^3 for the western half of greenhouse 1-PF) and multiplying by a conversion factor of time units (3600 s h^{-1}). Volumetric flow rates through the evaporative pad were calculated by multiplying the average of the air velocity component perpendicular to the plane of the pad by the pad's surface area (33.25 m^2).

For test types A, B, and C, with an exhaust fan frequency of 40 Hz, these values ranged between 27.1 h^{-1} (test C2) and 31.5 h^{-1} (test C1). At a fan frequency of 30 Hz, they fell to 24.3 h^{-1} (test D1); at 50 Hz, they rose to 35.3 h^{-1} (test D2).

Sethi and Sharma (2007) observed that an air exchange rate equivalent to 20 h^{-1} through the evaporative pads is sufficient to reach tolerable conditions inside the greenhouse under dry weather conditions. The air exchange rates obtained for greenhouse 1-PF are similar to the results obtained by López et al. (2010), i.e., above 20 h^{-1} but below the optimum value of 45 to 60 h^{-1} (Hellickson and Walker, 1983). The air exchange rates obtained are higher than those observed with natural ventilation in the province of Almería (5 to 18 h^{-1}) in Almería-type greenhouses (Molina-Aiz et al., 2009) and in Mediterranean greenhouses (Valera et al., 2009; López et al., 2011). Of the different variables studied, only the exhaust fan frequency had a direct bearing on the air exchange rate. As the two interior fans were located quite far from the pad (4 m and 20 m away), they did not increase the entrance of air through the pad.

Interior Microclimate

The combination of the shading screen and increasing the exhaust fan frequency to 50 Hz achieved the greatest temperature drop with respect to the outside temperature and the values recorded inside greenhouse 2-NV (table 5). The shading screen limits the entrance of solar radiation (Willits and Peet, 2000; Willits, 2001; Kittas et al., 2001), while the higher exhaust fan frequency increases the air exchange rate (Al-Helal, 2007; Willits and Li, 2005). Table 5 outlines the microclimate conditions inside greenhouse 1-PF for the different tests.

The use of the interior fans (type B tests) did not increase the capacity of the cooling system (table 5), but it did have a direct bearing on the temperature distribution inside the greenhouse by generating greater mixing of the inside air (figs. 3c and 3d). The placing of the interior fans was not the most suitable to reduce the horizontal temperature gradients that characterize this type of cooling system.

With respect to the outside temperature value, the temperature drop (ΔT_{io}) is well below the 5.3°C observed by Willits and Peet (2000) in a greenhouse with an externally mounted shade cloth and below the 10°C indicated by Kittas et al. (2001) with a half-shaded plastic roof. The discrepancy between the temperature differences of 5.3°C and 10°C (observed by Willits and Peet, 2000, in the U.S. and by Kittas et al., 2001, in Greece) and those measured in our experimental greenhouses (0.2°C to 3.0°C) in Almería,

Table 5. Microclimate conditions inside greenhouse 1-PF for the different tests.

Variable[a]	Test Type[b]							
	A1	A2	B1	B2	C1	C2	D1	D2
$T_{1\text{-PF}}$ (at 1.75 m above the ground)	24.7	26.1	25.4	23.8	25.8	25.8	27.1	24.9
$\Delta T_{io} = T_{1\text{-PF}} - T_o$	-1.5	-2.9	-1.7	0.2	-2.5	-3.0	-0.7	-3.0
$\Delta T_{h(1)}$ (at 1.75 m above the ground)	0.12	0.11	0.16	0.07	0.09	0.10	0.15	0.12
$\Delta T_{h(2)}$ (average of 1 and 2 m above the ground)	0.07	0.05	0.10	0.03	0.04	0.04	0.07	0.06
ΔT_v (between 1 and 2 m)	0.9	1.0	1.0	0.0	0.7	0.8	1.2	0.9
$T_{1\text{-PF}} - T_{2\text{-NV}}$	-5.3	-5.4	-4.5	-1.7	-4.4	-5.3	-5.6	-8.1
$T_{1\text{-PF}}$[c] $- T_{2\text{-NV}}$	-4.9	-5.1	-4.5	-2.1	-4.5	-5.5	-5.1	-7.5

[a] $T_{1\text{-PF}}$ = mean inside temperature measured using 3D anemometers, disregarding the measurement points closest to the pad and exhaust fans (eq. 7), T_o = outside temperature, $\Delta T_{h(1)}$ = horizontal temperature gradient (°C m^{-1}) measured using 3D anemometers and including all measurement points, $\Delta T_{h(2)}$ and ΔT_v = horizontal and vertical temperature gradients obtained using fixed sensors (°C m^{-1}), $T_{2\text{-NV}}$ = temperature inside greenhouse 2-NV. Measurement tests included type.

[b] A = pad-fan and 40 Hz exhaust fan frequency, B = pad-fan and 40 Hz exhaust fan frequency plus interior fans, C = shaded pad-fan, and D = pad-fan with different exhaust fan frequencies (30 Hz for D1 and 50 Hz for D2).

[c] $T_{1\text{-PF}}$ = mean inside temperature measured using autonomous dataloggers (HOBO Pro Temp-HR U23-001).

Spain, may be due to the lower outside relative humidity observed in the U.S. (53%) and Greece (31.1% to 44.6%). The outside relative humidity during our measurement tests ranged between 56% and 86%, reducing the high efficiency of the cooling pad-fan system reported under dry climatic conditions.

One positive aspect of the use of the shading screen (type C tests) is that it reduces the horizontal and vertical temperature gradient with respect to the type A tests (table 5). Willits and Peet (2000) observed that this combination can lead to a reduction in temperature increase through the greenhouse of 0.09°C m^{-1} with respect to a greenhouse without screens (0.52°C m^{-1}). In the present study, the use of the shading screen in the type C tests reduced the horizontal temperature gradient by about 0.02°C m^{-1} (for measurements taken with HOBO temperature sensors at 1 m and 2 m heights and measured by 3D sonic anemometers at 1.75 m above the ground) in comparison with the type A tests. The difference between our results and those obtained by Willits and Peet (2000) may be due to their smaller greenhouse size (6.7 m × 12.2 m), i.e., the distance between the pad and exhaust fans was half roughly half that of the present work.

Agricultural fans attached to an evaporate pad may operate against static pressures as high as 60 to 120 Pa. Normally, the fan airflow rate at 25 or 30 Pa static pressure is about 80% of the free air capacity when no resistance is offered to the fan (*ASABE Standards*, 2008b). From calibration tests of exhaust fans at 50 Hz (fig. 4a) and flow rates measured in the experimental greenhouse for this frequency (fig. 4b), we can deduce that the static pressure in the greenhouse was 47.5 Pa. In these operating conditions, reducing the fan frequency to 30 Hz (test D1) reduces the air exchange rate (fig. 4b) and therefore diminishes the system's cooling capacity, expressed as the temperature difference between the inside and outside air (fig. 4c). It also has the effect of increasing the horizontal temperature gradients compared to the type A tests when the fan frequency was 40 Hz (table 5). In addition, at the lower fan frequency, the automatic opening of the fans did not function correctly, requiring manual operation.

Al-Helal (2007) found that, when the fan frequency was reduced, the air exchange rate fell from 60 to 30 h^{-1} and the horizontal temperature gradient increased from 0.23°C to 0.36°C m^{-1}.

Increasing the fan frequency to 50 Hz (test D2) produced the greatest temperature difference between inside and outside temperatures (fig. 4c), particularly between the two experimental greenhouses. However, unlike the use of the shading screen, the fan frequency did not prove effective in reducing the horizontal and vertical temperature gradients (table 5).

Given a horizontal gradient of 0.10°C m^{-1}, the temperature difference between the air leaving the evaporative pad and that entering the exhaust fans 24 m away would be 2.4°C in the experimental greenhouse. In commercial greenhouses, the distance covered by the air would be around 60 m, which would mean a temperature difference of 6°C. If these temperature gradients are not taken into account, the cooling system is likely to consume excessive amounts of water (Boulard and Wang, 2002) or the plants farthest from the pads may suffer due to high temperatures.

Different operational alternatives of the cooling system, tested in greenhouse 1-PF on sunny days, show major reductions of the inside temperature (4.4°C to 8.1°C) with respect to the naturally ventilated greenhouse. This is particularly noteworthy bearing in mind that most greenhouses in Almería rely solely on natural ventilation and inside temperatures can reach 40°C in July (Molina-Aiz, 2010). On cloudy days (test type B2), when the outside relative humidity is high (about 86%) and solar radiation is low (82 W m^{-2}), the cooling capacity is reduced, and the inside temperature is greater than the outside temperature (ΔT_{io} = 0.2°C), reaching similar levels to the naturally ventilated greenhouse (only 1.7°C lower). The temperature differences recorded between greenhouses 1-PF and 2-NV are within the range of 4° to 12°C predicted by Sethi and Sharma (2007).

Horizontal Distribution of Inside Temperature

To study the temperature distribution inside greenhouse 1-PF, based on the 3D anemometer measurements, we can use the normalized difference in air temperature ($\Delta T_{io}{}^{n}$) applied by Boulard et al. (2000):

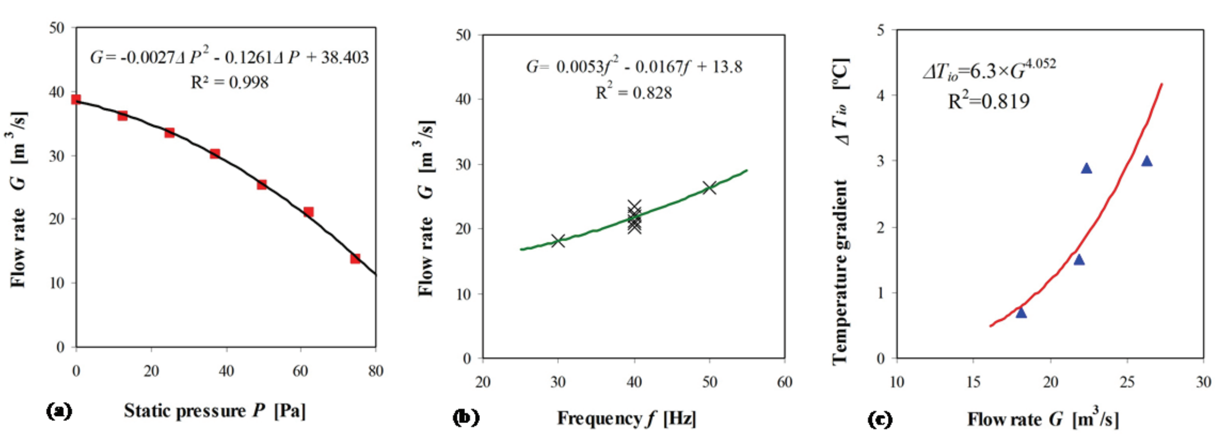

Figure 4. (a) Flow rate (for four exhaust fans) as a function of the static pressure obtained in calibration test at 50 Hz (BESS Lab, 1997), (b) flow rates measured in the experimental greenhouse as a function of the exhaust fans' electrical frequency, and (c) temperature gradient with respect to the outside air as a function of the measured flow rate for measurement tests A1, A2, D1, and D2.

$$\Delta T_{io,j}^{n} = \frac{T_{sc,j} - T_{o,j}}{T_{i,j} - T_{o,j}} \tag{6}$$

where $T_{sc,j}$ is the corrected sonic temperature inside the greenhouse for point j, and $T_{i,j}$ and $T_{o,j}$ are the mean inside and outside air temperatures recorded over the period of measurement at point j. This method proved problematic at times when there was little difference between the two temperatures. Consequently, for greenhouse 1-PF, it was decided to use the air temperature corrected ($T_{1\text{-PF}}{}^{C}$, °C) with the average temperature between inside and outside:

$$T_{1-PF}{}^{C} = T_{sc,j}\,\frac{T_o + T_i}{T_{o,j} + T_{i,j}} \tag{7}$$

where T_o and T_i are the mean inside and outside air temperatures during the test.

Figure 5 shows the temperature distribution in greenhouse 1-PF in five of the eight tests carried out. These temperature maps provide a visual image of how the temperature increases inside the greenhouse at greater distances from the evaporative pad. The antechamber in the western end of the greenhouse gives rise to an area in the northwest corner in which the air velocity diminishes, producing an increase in temperature (figs. 5a, 5b, 5d, and 5e).

The use of the shading screen reduces the temperature increase in that area (fig. 5c). In some of the tests without the shading screen and with a fan frequency of 40 Hz, the temperature in the northern half of the greenhouse is higher than the outside temperature: tests A1, B1 (fig. 5b), B2, and D1. In the two tests that combine the cooling system and the shading screen, and in the test using a fan frequency of 50 Hz, the inside temperature was lower than the outside temperature at all measurement points in the greenhouse (figs. 5c and 5e). When the fan frequency was reduced to 30 Hz, the inside temperature is only below the outside temperature in the first half of the greenhouse (fig. 5d).

During the tests, it was observed that some parts of the evaporative pad were not saturated with water, and this is reflected in the temperature maps by points or zones of higher temperature next to the pad (fig. 5). We should highlight the great difference in temperature between the coldest point (next to the pad) and the hottest point (next to the fans and/or behind the antechamber). This difference was 3.5°C in test A1 when the pad-fan cooling system operated alone, 5.8°C in test B1 with the addition of interior fans, 5.0°C in test C2 with the shading screen, 4.7°C in test D1 with a fan frequency of 30 Hz, and 4.0°C in test D2 with a fan frequency of 50 Hz.

Given that such considerable differences in temperature have been observed in such a small greenhouse, it seems surprising that in much larger commercial greenhouses this cooling system is controlled using only one sensor unit to record temperature and humidity. We therefore recommend carrying out a preliminary study of the temperature distribution inside the greenhouse with the cooling system operating, with a view to designing a suitable distribution of sensors to record the climate variables that will later be used for controlling the system.

Using additional fans inside the greenhouse has a significant effect on the inside temperature distribution (fig. 5b).

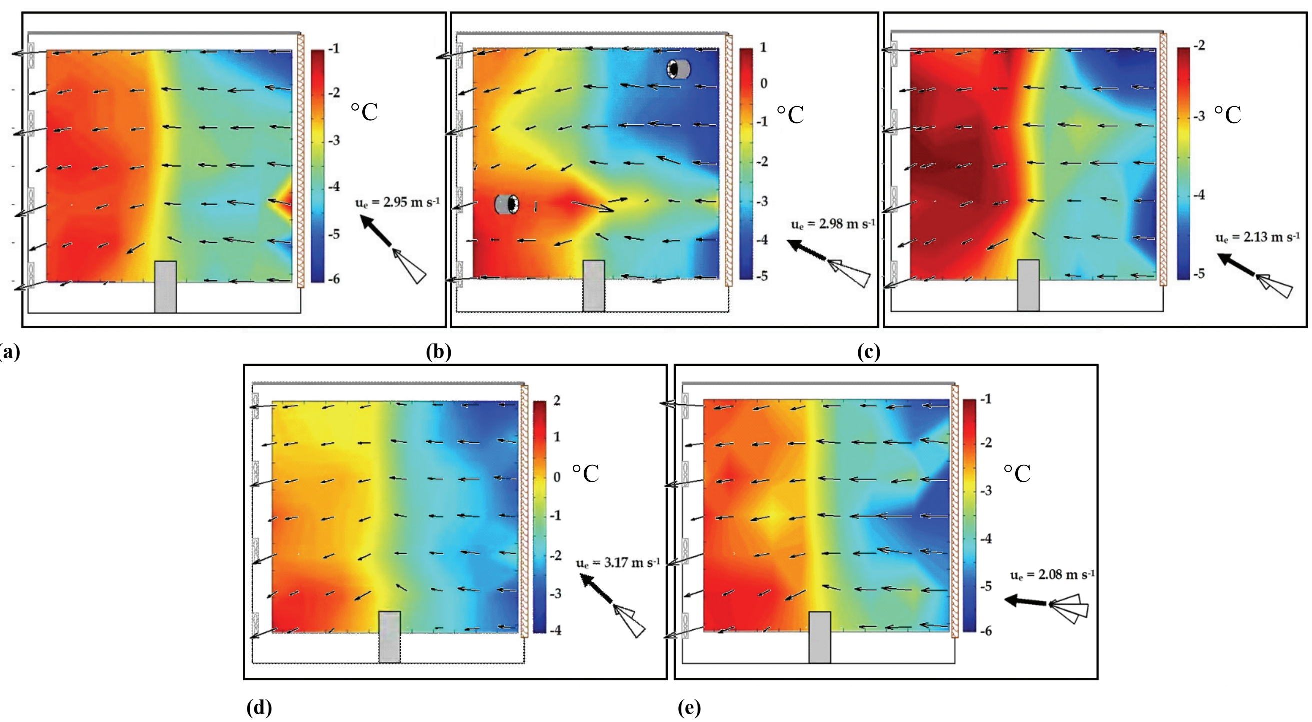

Figure 5. Distribution of the difference between inside and outside temperatures ($T_{1\text{-PF}}{}^{C} - T_o$) in greenhouse 1-PF at a height of 1.75 m: (a) test A2, (b) test B1, (c) test C2, (d) test D1, and (e) test D2 (A = pad-fan and 40 Hz exhaust fan frequency, B = pad-fan and 40 Hz exhaust fan frequency plus interior fans, C = shaded pad-fan, and D = pad-fan with different exhaust fan frequencies, 30 Hz for D1 and 50 Hz for D2). Vectors represent air velocity; the larger the vector, the greater the velocity (scale of the vectors is not shown).

However, this system does not appear to reduce the horizontal and vertical temperature gradients (table 5). Tesi and Ferlicca (1969) used plastic diaphragms and a fan to improve the uniformity of the microclimate in a greenhouse with an evaporative cooling system.

COOLING EFFICIENCY AND WATER CONSUMPTION

The efficiency of a cooling system (η) can be defined as (ASHRAE, 2008):

$$\eta = \frac{T_{bs} - T_{cbs}}{T_{bs} - T_{bh}} \tag{8}$$

where T_{bs} is the dry bulb temperature of the outside air, T_{cbs} is the dry bulb temperature of the cold air leaving the evaporative pads, and T_{bh} is the wet bulb temperature of the outside air.

The mean efficiency of $\eta = 0.65$ for the eight tests (table 6) is much lower than the value of $\eta = 0.82$ recorded in tests carried out in 2008 (López et al., 2010), when the system was newly installed and showed no deficiencies in the saturation of the pad; it is also lower than the value of $\eta = 0.80$ obtained by Kittas et al. (2001). One month before the present study, the cooling system was checked by the company that installed it; however, the results indicate that certain basic criteria should be established regarding system maintenance. At present, there are no such norms on a regional or national level.

The water consumption of the evaporative pad (m_w, kg h^{-1} m^{-2}) can be estimated using the following expression (Hellickson and Walker, 1983):

$$m_w = \frac{G_p \rho_a (x_p - x_o)}{S_A} \tag{9}$$

where G_p is the volume of air passing through the pad (m^3 h^{-1}), ρ_a is the air density (kg m^{-3}), x_p is the absolute humidity of the air leaving the pad (kg kg^{-1}), x_o is the absolute humidity of the outside air (kg kg^{-1}), and S_A is the surface area of the soil (m^{-2}). The water consumption has been estimated for the surface of the pad (S_p), replacing S_A with S_p in equation 9 (table 6). The absolute humidity of the air at saturation can be calculated as follows (*ASABE Standards*, 2010):

$$x_s = \frac{\varepsilon_a e_s(T)}{P - e_s(T)} \tag{10}$$

where ε_a is the ratio between the molecular mass of the water vapor and of dry air ($\varepsilon_a = 0.6219$), P is the atmospheric pressure (Pa), and $e_s(T)$ is the saturation vapor pressure (Pa) as a function of the air temperature, calculated using the expression of Magnus-Tetens (Tetens, 1930):

$$e_s(T) = 6.108 \exp\left(\frac{17.269T}{T + 239.3}\right) \tag{11}$$

The estimated water consumption (m_w, kg h^{-1} m^{-2}) in greenhouse 1-PF increases with the capacity to increase the water vapor content of the incoming air, which is expressed as the difference between the absolute saturation humidity (x_{so}, kg kg^{-1}) and the absolute humidity of the outside air (x_s, kg kg^{-1}), and the volumetric flow rate (G, m^3 s^{-1}). Multiple regression analysis showed the results fitting a multiple linear regression model to describe the relationship between m_w and the two independent variables: $m_w = 0.292G + 717.975(x_{so} - x_o) - 6.583$. The equation showed a statistically significant relationship at a 99% confidence level ($R^2 = 0.96$, $p = 0.0002$).

In the same way, we obtained a significant relationship between water consumption (m_w, kg h^{-1} m^{-2}) in table 6 and the outside-inside temperature difference (ΔT_{io}, °C) obtained with the pad-fan system (table 5) at a 99% confidence level: $\Delta T_{io} = 0.7913 - 0.4732m_w$ ($R^2 = 0.92$, $p = 0.0001$). These types of ratios could be included in the control unit of the cooling system in order to modify the amount of water in the pads according to the capacity to increase the humidity of the incoming air.

The use of interior fans to move air inside the greenhouse (type B tests) or the use of shading screens (type C tests) do not seem to affect water consumption, and consequently they do not affect the pad-fan cooling capacity. Increasing the exhaust fan frequency (type D2 tests) is the only alternative that implies an increase in water consumption, as was observed by Al-Helal (2007), due to an increase in the volumetric flow rate. However, the regression analysis showed that the most influential factor on water consumption was the outside climate (air humidity and temperature). Thus, on a cloudy day, when the outside relative humidity was high (86% for measurement test B2) and the air was near saturation, the capacity of the pad to increase the water content of the air was low. As a result, in these conditions, water consumption diminished drastically (table 6). Therefore, the effect of the cooling system on inside temperature is similar to that obtained with a system of natural ventilation, but with an additional cost derived from the use of electricity and water.

Table 6. Efficiency of the cooling system (η), estimated water consumption (m_w, kg h^{-1} m^{-2} of pad), and difference between absolute saturation humidity and absolute humidity of the outside air (kg kg^{-1}).

	Test Type[a]							
	A1	A2	B1	B2	C1	C2	D1	D2
η	0.65	0.58	0.74	0.46	0.64	0.73	0.64	0.76
m_w	4.71	7.38	5.84	1.11	6.70	6.80	3.87	8.88
x_{so} - x_o	0.0065	0.0113	0.0086	0.0027	0.0095	0.0103	0.0066	0.0100

[a] A = pad-fan and 40 Hz exhaust fan frequency, B = pad-fan and 40 Hz exhaust fan frequency plus interior fans, C = shaded pad-fan, and D = pad-fan with different exhaust fan frequencies (30 Hz for D1 and 50 Hz for D2).

CONCLUSIONS

In the present study in a Mediterranean greenhouse with a well-developed tomato crop, the combination of shading screens with a pad-fan cooling system at medium ventilation rates (0.042 to 0.049 m^3 s^{-1} m^{-2}) achieved the greatest reduction in temperature with respect to the outside temperature (ΔT_{io} = -3°C). At the same time, this option does not imply increased consumption of water or electricity, and once it is installed it can be used for other purposes.

Increasing the exhaust fan frequency from 40 to 50 Hz with a variable frequency drive unit increased the ventilation rate to 0.055 $m^3 s^{-1} m^{-2}$ and therefore led to a variable increase in the system's cooling capacity, expressed as the temperature difference between inside and outside air (ΔT_{io}) ranging from -1.5°C and -2.9°C to -3.0°C. This option implies increased water and electricity consumption by the system. On the other hand, reducing the exhaust fan frequency to 30 Hz implies a ventilation rate of 0.038 $m^3 s^{-1} m^{-2}$ and has negative effects on the system, hampering its capacity to reduce the inside air temperature (ΔT_{io} = -0.7°C) and increasing the horizontal and vertical temperature gradients.

Inside-outside temperature differences (ΔT_{io}) obtained with the different operational alternatives of the cooling system (three ventilation flow rates, and a combination of the medium flow rate with two interior fans and with a shading screen) ranged from 0.2°C to -3°C. These values are lower than those reported in the literature, possibly due to the high outside relative humidity (56% to 86%) during our measurements tests. The proximity of Almería's greenhouses to the Mediterranean Sea implies high outside humidity when the cold damp westerly *Poniente* wind blows in from the sea, thus reducing the efficiency of the cooling pad-fan system.

Different operational alternatives of the cooling system, tested in experimental greenhouse 1-PF on sunny days, show major reductions of the inside temperature (4.4° to 8.1°C) with respect to the naturally ventilated greenhouse. This is particularly important for the province of Almería and the Mediterranean region as a whole, where natural ventilation is the main means of climate control in greenhouses (Molina-Aiz, 2010). Under cloudy conditions, when solar radiation was low (82 W m^{-2}) and the outside relative humidity was high (about 86%), cooling capacity was reduced and the inside temperature was greater than the outside temperature (ΔT_{io} = 0.2°C), similar to that obtained in the naturally ventilated greenhouse (only 1.7°C lower).

Combination of the cooling system with additional fans inside the greenhouse increases the air velocity, favors the mixing of air, and increases the turbulence intensity. However, placing the fans opposite each other at a height of 2.81 m inside the experimental greenhouse (the most common setup in Almería's commercial greenhouses) does not appear to be effective to reduce the horizontal and vertical temperature gradients that characterize the pad-fan cooling system. We recommend that future research analyzes the optimal location and orientation of these fans to improve the temperature distribution inside greenhouses.

Lack of maintenance or unsuitable maintenance can have a marked effect on the system's efficiency (η). In the system under study, this value decreased from η = 0.82 when the system was installed (summer 2008) to η = 0.65 during the present study (summer 2009).

Temperature differences of up to 5.8°C were observed between the coldest point (next to the pad) and the hottest point (next to the exhaust fans and/or behind the antechamber). Surprisingly, even though pad-fan cooling systems are characterized by considerable heterogeneity of the microclimate, a single sensor unit, placed in the center of the greenhouse, is typically used to measure temperature and humidity. We therefore recommend carrying out a preliminary study of the temperature distribution inside the greenhouse with the cooling system operating, with a view to designing a suitable distribution of sensors to record the climate variables that will later be used for controlling the system.

The water consumption of the pad-fan system increases with the capacity to augment the vapor content of the incoming air and the volumetric flow rate: $m_w = 0.292G + 717.975(x_{so} - x_o) - 6.583$ ($R^2 = 0.96$), where m_w is the estimated water consumption (kg $h^{-1} m^{-2}$), G is the volumetric flow rate ($m^3 s^{-1}$), and $(x_{so} - x_o)$ is the difference between absolute saturation humidity and absolute humidity of the outside air (kg kg^{-1}). This type of ratio could be included in the controls of the system to modify the amount of water applied to the pads depending on the capacity to increase the humidity of the outside air.

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NOMENCLATURE

$e_{s(T)}$ = saturation vapor pressure as a function of air temperature (Pa)
G = volumetric airflow rate (m^3 s^{-1})
G_p = volumetric airflow rate through the evaporative pads (m^3 h^{-1})
i = turbulence intensity
l = two-dimensional horizontal resultant of air velocity in XY plane (m s^{-1})
m_w = water consumption per unit of evaporative pad area (kg h^{-1} m^{-2})
P = air pressure (Pa)
q = specific humidity (g g^{-1})
R_g = solar radiation (W m^{-2})
R = greenhouse air exchange rate (h^{-1})
RH = relative humidity (%)
S_A = greenhouse surface area (m^2)

S_p = surface area of the evaporative pads (m^2)
S_V = surface area of the vents (m^2)
T = temperature (°C)
ΔT_h = horizontal temperature gradient (°C m^{-1})
ΔT_{io} = inside to outside temperature difference (°C)
ΔT_v = vertical temperature gradient (°C m^{-1})
t = time (s)
u = air velocity (m s^{-1})
$\bar{u}$ = mean air velocity (m s^{-1})
u' = air velocity fluctuating component (m s^{-1})
v = two-dimensional vertical resultant of air velocity in XZ plane (m s^{-1})
x = absolute humidity (g g^{-1})
x_p = absolute humidity of the air leaving the pad (g g^{-1})
x_s = absolute humidity at saturation (g g^{-1})

GREEK LETTERS

Δ = difference
θ = wind direction (degrees; 0° or 360° represents wind from the north)
ρ_a = air density (kg m^{-3})
σ = standard deviation
η = cooling efficiency (%)
ε_a = ratio of the molecular mass of water vapor and of dry air

SUBSCRIPTS

bh = wet bulb of the outside air
bs = dry bulb of the outside air
cbs = dry bulb of the air leaving the pad
l = horizontal resultant of air velocity
v = vertical resultant of air velocity
x = longitudinal component
y = transversal component
z = vertical component
s = sonic
sc = sonic corrected
j = measurement point
i = inside
o = outside
1-PF = greenhouse 1 with pad-fan system
2-NV = greenhouse 2 with natural ventilation

SUPERSCRIPTS

n = normalized
c = corrected