Sonic anemometry to evaluate airflow characteristics and temperature distribution in empty Mediterranean greenhouses equipped with pad-fan and fog systems

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

Universidad de Almeria. Campus de Excelencia Internacional Agroalimentario ceiA3. Ctra. de Sacramento s/n. 04120 Almería, Spain.

* Corresponding author: Tel.: +34 950015546; fax: +34 950015491.

E-mail address: dvalera@ual.es (D.L.Valera).

Abstract

Sonic anemometry has been used to analyse two greenhouse evaporative cooling systems: a pad-fan system and a low pressure water/air fog system. These systems were used in empty greenhouses to simulate the microclimatic conditions produced inside Mediterranean greenhouses when crops are seeded in nurseries or transplanted in commercial greenhouses. Evaporative cooling systems could be necessary in the future for all Mediterranean greenhouses to reduce excess heat and to maintain certain levels of relative humidity on hot days from spring to autumn. The pad-fan system proved capable of maintaining more favourable conditions than the fog system. The best results were obtained by combining the evaporative pads with shading screens (differences of 1.4ºC to 1.8ºC between inside and outside temperature). The main drawbacks of the pad-fan system were the horizontal and vertical temperature gradients, with a maximum temperature difference between pads and fans of up to 11.4ºC, and a maximum difference of 6.7ºC between heights of 2 m and 1 m. However, inside temperature and relative humidity were more stable over time in the greenhouse using the pad-fan system. The fog system required higher energy consumption (7.2-8.9 kWh) than the pad-fan system (5.1 kWh) for continuous operations over one hour. Nevertheless, the average water consumption of the pads (122.3 l h^{-1}) is greater than that of the fog system (9.4 l h^{-1}).

Keywords: Greenhouse; Cooling; Evaporative pads; Fog system.

1. INTRODUCTION

In Mediterranean climates, excess heat from spring to autumn may affect the yield of greenhouse crops (Kittas, Draoui, Boulard, 1995). Under such circumstances, special care must be taken on dry days, when a balance must be attained between reducing excess heat and maintaining certain levels of relative humidity (70-90%) inside the greehouse (Von Elsner *et al*., 2000), since levels of below 60% may produce hydric stress in the crop (Bailey, 2006) and require additional irrigation.

The greenhouse effect brought about by the increase in the concentration of greenhouse gases in the atmosphere is giving rise to global warming. Boulard and Fatnassi (2010) simulated the climatic conditions in the region of Avignon (France) for 2070-2099, predicting an average yearly air temperature increase of only 1°C inside the greenhouse for an augment of 2.2°C outside. These predictions are an indication that evaporative cooling systems may be required in the future in Mediterranean greenhouses.

In the province of Almería very few greenhouses are equipped with systems that act on the microclimate. The vast majority merely rely on natural (passive) ventilation. Natural ventilation is usually the simplest and preferred greenhouse cooling method due to its low cost and simplicity (Villarreal-Guerrero *et al*., 2012a). The main advantage of sonic anemometry is that it allows natural ventilation flow patterns to be established, as well as determining whether air enters or exits through the greenhouse vents, and differentiating between the mean and turbulent contributions to air renewal and the removal of heat (Molina-Aiz, 2010). When roof vents are opened to windward, the wind and buoyancy effects can produce air movements in different senses, counteracting and reducing the cooling capacity of the natural ventilation system (López, 2011). However, natural ventilation alone is generally not adequate to remove efficiently the surplus energy during very hot periods (Villarreal-Guerrero *et al*., 2012a). In addition to ventilation, daily cooling must be provided for greenhouses located in semiarid climates to maintain the desired conditions for year-round crop production (Villarreal-Guerrero *et al*., 2012b).

The high air temperature and humidity in the greenhouse can severely limit greenhouse production, causing reduced yield and low produce quality during a significant part of the year. However, greenhouse crop production can be maintained during warm periods by proper use of greenhouse climate control and various cooling methods (Villarreal-Guerrero *et al*., 2012a).

Two evaporative cooling systems can be used to reduce the temperature inside the greenhouse: those employing evaporative pads and extractor fans and those based on fog systems, which are associated with natural ventilation. Pad-fan evaporative cooling systems can significantly reduce the inside temperature by 4-6ºC in comparison to the outside air temperature (Sethi and Sharma, 2007). Pad-fan systems allow lower inside temperatures (28ºC) than natural ventilation with fogging nozzles (31.4ºC), improving the quality and yield of tomato crops (Willits and Li, 2005). Another advantage of the pad-fan system compared to misting is that it does not imply any risk of wetting the plants (Arbel, Barak, Shklyar, 2003).

However, the use of the pad-fan system to cool greenhouses normally produces non-uniform climatic conditions (Arbel, Barak, Shklyar, 2003), and significant temperature gradients of around 0.13-0.27ºC m^{-1} are observed in the direction of the airflow (Kittas, Boulard, Papadakis, 1997; López, Valera, Molina-Aiz, Peña, 2010). The vertical temperature gradient, which increases at greater distances from the pad, must also be considered (López, Valera, Molina-Aiz, Peña, 2010). The use of shade screens can reduce the horizontal temperature gradient produced by this evaporative cooling system by 18% (Willits & Peet, 2000) and can increase the inside-outside temperature difference to 12ºC (Sethi and Sharma, 2007).

Fog systems may be recommended for hot and even desert regions, as they can achieve greater uniformity of temperature and higher levels of relative humidity than evaporative pads (Luchow & von Zabeltitz, 1992). Çolak (2002) compared the two systems, finding that the temperature drop achieved in the greenhouse by pads was 1.4ºC greater. By combining a fog system with mechanical ventilation Arbel, Barak and Shklyar (2003) were able to maintain the greenhouse temperature at 28ºC and the relative humidity at 80% at midday during the summer months (for outside temperatures of 40-42ºC). Fog systems can also be combined with shade screens to achieve greater reductions in temperature (Perdigones *et al*., 2008).

However, no system can satisfy all the cooling requirements for all the greenhouses types and crops grown (Sethi and Sharma, 2007). Arbel, Barak and Shklyar (2003) recommend future research centering on anlaysis of the characteristics of the airflow generated inside the greenhouses. Li and Willits (2008) carried out this type of analysis with a hot wire anemometer, measuring only the magnitude of the mean air velocity vector, whereas a more interesting characterisation of the airflow can be obtained using three-dimensional sonic anemometry, which allows the three orthogonal components of air velocity to be measured (López, Valera, Molina-Aiz, Peña, 2010; López, Valera, Molina-Aiz, 2011).

Temperatures in excess of 32ºC are considered excessive for greenhouse crops (Baudoin *et al*., 1999). When mean daily temperatures above 22ºC are common, artificial cooling may be necessary (Baudoin *et al*., 1999; Von Elsner *et al*., 2000) or cultivation in greenhouses has to be stopped, and with mean temperatures between 12 and 22ºC natural ventilation is sufficient; mean daily temperatures above 27ºC could be considerate excesive (Von Elsner *et* *al*., 2000). In Almería, these limiting temperatures are reached in late July and throughout August (Fig. 1), restricting crop transplant to the greenhouses during this period. Seedlings are grown in soil cubes and are transplanted in late summer, normally in late August and early September (Orgaz, Fernández, Bonachela, Gallardo, Fereres, 2005; Magán, Gallardo, Thompson, Lorenzo, 2008).

Fog has long been used to increase cooling when the ability of the plants to transpire is expected to be insufficient (e.g., in rooting and transplant greenhouses) (ASABE, 2008). In Almería, the main utility of evaporative cooling systems can be to reduce high temperatures inside greenhouses in August, in order to bring forward the transplanting date. In nursery greenhouses, the use of evaporative cooling systems can allow seedlings to be available for early transplanting. Thus, the nurseries could be guaranteed a better price (Gupta, Samuel, Sirohi, 2010). In crop production greenhouses, the use of evaporative cooling systems can allow young transplants to acclimatize to changing environmental conditions, allowing them to withstand the environmental stress caused by transplant to the greenhouse (Gupta, Samuel, Sirohi, 2010). These systems are also recommended when plant evapotranspiration is negligible. By bringing forward the transplanting date farmers can have an early yield when production prices are at a maximum, and thus increase their income.

The present work was carried out in summer 2010 with the aim of analysing the capacity to reduce the temperature inside empty greenhouses (when crops are seeded in nursery or transplanted in greenhouses) using different evaporative cooling systems (pad-fan and fog), either working alone or combined with aluminised shading mesh and interior fans. This work is a continuation of two studies carried out in the same experiemental greenhouses over the two previous summers. In 2008 we studied the airflow and distribution of temperature and humidity in a multi-span greenhouse equipped with a pad-fan cooling system operating with a well-developed tomato crop and without crop (López, Valera, Molina-Aiz, Peña, 2010). The system of evaporative pads was less effective for the empty greenhouse. The crop inside the greenhouse allowed the temperature in the area occupied by the plants (1 m above the ground) to be reduced by about 4°C with respect to the empty greenhouse (López, Valera, Molina-Aiz, Peña, 2010). In summer 2009 we studied the microclimate and airflow inside the same multi-span greenhouse (with a well-developed tomato crop) equipped with a pad-fan cooling system, analysing several operational alternatives: three ventilation flow rates, and combining the medium flow rate with two interior fans or with a shading screen (López, Valera, Molina-Aiz, Peña, 2012). The combination of shading screens with the pad-fan working at medium level ventilation rates (0.042-0.049 $m^3 s^{-1} m^{-2}$) achieved the greatest reduction in temperature with respect to the outside value (inside-outside temperature difference of –3ºC).

The main aim of the present work is therefore to compare the microclimate in three Mediterranean multi-span greenhouses (arch-shaped roof) with different cooling systems (pad-fan system, fog system and natural ventilation as reference) without the contribution of cooling and humidification from the crop by evapotranspiration. Together with the fog system, inside fans were used to homogenize the microclimate inside the greenhouse. To compare these cooling systems we analyse the inside air velocity vector, which is important to understand the airflow pattern and its uniformity, since this explains the temperature distribution inside the greenhouse. This knowledge can help to improve these cooling systems. In addition, the methodology allows us to study the characteristics of the air flow turbulence, providing useful information for future validations of Computational Fluid Dynamics (CFD) simulations.

2. MATERIAL AND METHODS

2.1. Experimental Setup

The experimental work took place in three multi-span greenhouses located at the agricultural research farm belonging to the University of Almería (Fig. 2), in southeastern Spain (36° 51′ N, 2° 16′ W and 87 MASL). Two greenhouses of 24×45 m^2 [greenhouses 1 (Pad-Fan, PF) and 2 (Fog System combined with natural ventilation, FG)] and one of 18×45 m^2 [greenhouse 3 (Natural Ventilation, NV)] were each divided into two halves by a polyethylene sheet, which allowed us to study the inside microclimate of the two halves separately for other research projects. The measurements were carried out in the western half of the experimental greenhouses 1-PF (24×20 m^2) and 2-FS (24×20 m^2), but with the evaporative pad (PF) and fog systems (FS) operating in the whole greenhouse. The polyethylene sheet that divided the greenhouses into two halves may have a slight bearing on the airflow patterns inside the naturally ventilated greenhouses 2-FG and 3-NV. However, the airflows produced for several wind conditions (intensity and direction) inside both halves of the experimental greenhouse 2-FS were analysed simultaneusly in 2009, observing similar airflow patterns (López, 2011). We have compared the three cooling systems activated manually. Thus the pad-fan and the fog system worked continuously over the whole measurement period and the vent opening was opened at a fixed position (Table 1).

During the measurement tests the greenhouses contained no crop to simulate conditions with seedlings. In order to prevent insects entering the greenhouses, insect-proof screens were placed on all the vents. The screens' geometric characteristics were obtained by the methodology developed by Valera, Álvarez and Molina-Aiz (2006): 10×16 threads cm^{-2} (47.0% porosity) in greenhouse 1-PF; 13×30 threads cm^{-2} (39.0% porosity) in greenhouse 2-FS; and 14×27 threads cm^{-2} (38.5% porosity) in greenhouse 3-NV.

In greenhouse 1-PF, Celdek evaporative pads (Munters AB, Kista, Sweden) (1.9×40 m^2; 1.9×17.5 m^2 in western sector) were installed on the southern side, and eight EM50n-d-1-wp-wm extractor fans (Munters Europe AB, Sollentuna, Sweden) were placed on the northern side. The extractor fans have a nominal power of 735 W, a nominal propeller (1.27 m diameter) speed of 368 rpm for a maximum electrical frequency of 50 Hz. The motors of the fans were 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. The water was supplied by a PRISMA25 4M pump (ESPA, Innovative Solutions, Spain; 1.4 kW) from the store located next to greenhouse 2-FS to two tanks located close to the evaporative pads; from there it was pumped to the pads using two submergible LOWARA pumps of 0.55 kW - model DOC 76(T) (ITT Corporation, Lowara Srl., Italy). The water consumption at the pads was measured by a multi-jet dry-rotor water meter MTK-HWVB (Wehrle Gmbh, Furtwangen, Germany), with a nominal flow rate of 2.5 $m^3\ h^{-1}$ and a minimum scale value of $10^{-5}\ m^3$.

Greenhouse 2-FS was equipped with a low pressure water/air fogging system model CLIMA – FUM (Mondragón Soluciones, Albuixech, Spain) with a volumetric water flow of 1.2 $l\ h^{-1}$ for water pressure of 0.3 MPa and air pressure of 0.4 MPa (droplet size <10 μm). The outlet pipes were placed at a height of 4.6 m, and nozzles were placed at a density of 1 nozzle per 12 m^2. Water was pumped to the system by a PRISMA25 5M (ESPA, Innovative Solutions, Spain; 1.7 kW), and one air compressor was used: PUSKA model RTA 10/8 (Puska Pneumatic, S.A., Spain; 7.5 kW). The water consumption of these cooling systems was obtained from readings of the water meter M150-20 (Elster Metering Limited, Luton, UK), with a permanent flow rate of 2.5 $m^3\ h^{-1}$ and a minimum scale value of $2{\times}10^{-5}\ m^3$.

Greenhouse 3-NV was naturally ventilated. All three greenhouses were fitted with the same aluminised shading mesh Aluminet 50-I (Polysack Plastic Indutries Ltd., Israel; 50-54% shading) and with two Munters Euroemme® EDC18 ventilators (Munters AB, Kista, Sweden;

0.37 kW) in each sector, located at a height of 2.5 m, i.e. the axis of the propeller was 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. The cost of the pad-fan cooling system was 11 € per m^{-2} of the greenhouse surface area, as compared to 3.2 € m^{-2} for the fog system. The lower price of the fog system is the main reason why this system is used more frequently in Almería's commercial greenhouse than the pad-fan (only used in some nurseries).

Natural ventilation in greenhouse 2-FS consisted of opening the two continuous side vents and one continuous roof vent, while greenhouse 3-NV had an additional continuous roof vent (Fig. 2). In the western sector of greenhouse 2-FS the area of the side vents was 17.50×1.05 m^2 and that of the roof vent was 17.50×1.00 m^2. In the western sector of greenhouse 3-NV (18×20 m^2) the northern side vent had an area of 15.00×1.05 m^2, the southern one had an area of 17.50×1.05 m^2 and the roof vent 17.50×1.00 m^2. The ventilation surface S_V/S_A (vent surface opened / ground surface) was 11.3% for greenhouse 2-FS (except for experiment A with 7.5%) and 19.2% for greenhouse 3-NV (Table 1), where all vents were fully opened for the eight tests.

Four different experiments were carried out (between 11:30 and 14:30) each with two replications (Table 1). In greenhouse 1-PF the system of evaporative pads was evaluated working in isolation (experiments A, B and D) and in combination with shade screen (experiment C). The experiments were carried out in the middle of the day and care was taken to ensure that external climatic conditions were as stable as possible for the duration of the experiments.

In greenhouse 2-FS the fog system was tested using different set-ups. In Almería greenhouses it is common practice to use active and inactive cycles (On/Off) in order to avoid wetting the crop, and the same practice was adopted for all experiments, although in this case there was no crop to simulate conditions shortly after transplanting. In all the experiments the On/Off cycles were programmed at intervals of 360/60 seconds. In experiment A the side vents were opened 50% and the roof vent 100%, while in experiment B all vents were fully opened (Table 1). In experiment C the system was combined with a shade screen. Finally, in experiment D it was combined with a shade screen and two interior fans.

2.2. Equipment and instrumetation

Air velocity and temperature inside greenhouses 1-PF and 2-FS were measured with two 3D sonic anemometers (model CSAT3, Campbell Scientific Spain S.L., Spain; resolution 0.001 m s^{-1} and 0.002ºC; accuracy ±0.04 m s^{-1} and ±0.026ºC). Air velocity was also measured with ten 2D sonic anemometers (mod. Windsonic, Gill Instrument LTD, United Kingdom; resolution 0.01 m s^{-1}; accuracy ±2%). Data from the 12 anemometers were recorded by two microloggers (model CR3000, Campbell Scientific Spain S.L.), with a data scan and storage rate of 10 Hz for 3D sonic anemometers (Shilo, Teitel, Mahrer, Boulard, 2004) and 1 Hz for 2D sonic anemometers (López, Valera, Molina-Aiz, 2011).

Figure 3 shows the 30 locations of the airflow measurements taken in the western sector of the experimental greenhouses. The 2D anemometers remained fixed in the positions indicated in Figure 3: five were placed at the roof vent of greenhouse 2-FS (this vent could only be accessed by the resistent horizontal beams of the greenhouse structure, which are 5 m apart, and so air velocity was measured in the middle of the vertical axis of the vent); the other five anemometers were located next to the evaporative pad in greenhouse 1-PF (3.5 m apart, measuring air velocity in the middle of the pad's vertical axis).

The two 3D anemometers, one in each greenhouse, measured air velocity in each of the 30 locations 1.75 m above the ground over 3 minutes. This time period is a compromise between a shorter one that may reduce accuracy and a longer one that may increase the distorsion produced by the change in microclimate parameters such as wind speed or inside-outside temperature difference (Molina-Aiz, Valera, Peña, Gil, López, 2009). For the analysis of the airflow characteristics we have calculated the following parameters from the 3D anemometers measurements over 3 minutes: mean air velocity $\bar{u}$, turbulence intensity i, the macroscale or integral length scale (the average size of the largest eddies) L_i, the discrete power spectrum density function $E(f)$, the average negative slope (β value) of the logarithmic power spectrum curves, the total turbulence kinetic energy k and the turbulence energy dissipation rate ε. An exhaustive description of all these parameters was given in a previous work (López, Valera and Molina-Aiz, 2011).

The wire frame intended to support and guide the crop was used to position the sonic anemometers inside the greenhouse. Each anemometer was mounted on a horizontal arm, which was fixed to a 3 m long aluminium pipe (Fig. 4a). At the upper end of the vertical pipe a U-shaped clamp was attached in order to fix it to the wire frame. At the lower end of the vertical pipe, a rod of smaller diameter was inserted to anchor the device to the ground once the anemometer had been placed at the correct point. A similar system was used to place the anemometers at the roof vent (Fig. 4b). A steel cable was extended under the greenhouse roof parallel to the roof vent, from which the vertical aluminium pipe with the anemometers was suspended. Once the anemometers were placed at the correct level, the device was secured to the greenhouse structure.

Outside climatic conditions were recorded (frequency 0.5 Hz) by a meteorological station at a height of 10 m located to the north of the greenhouse (Fig. 2). The meteorological station included a BUTRON II (Hortimax S.L., Almería, Spain) measurement box with a Pt1000 IEC 751 class B temperature sensor (Vaisala Oyj, Helsinki, Finland) with a measurement range of –10 to 60ºC and an accuracy of ±0.6ºC, and a capacitive humidity sensor HUMICAP 180R (Vaisala Oyj, Helsinki, Finland) with a measurement range of 0% to 100% and an accuracy of ±3%. Outside wind speed was measured with a Meteostation II (Hortimax S.L., Almería, Spain), 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 vane (accuracy ±5° and resolution 1°). Incoming shortwave radiation was measured using a Kipp Solari (Hortimax S.L., Almería, Spain) 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}.

Temperature and humidity inside the western halves of the three greenhouses were measured using 36 autonomous dataloggers (HOBO Pro Temp-HR U23-001, Onset Computer Corp., Bourne, Massachusetts, USA) placed at heights of 1 and 2 m. Another four sensors were placed next to the evaporative pad, two inside the greenhouse and two outside (Fig. 2). 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 passive solar radiation open boxes, allowing natural air movement around the sensors. The sensors were not mechanically ventilated to avoid the air mixing from different heights (Molina-Aiz, Valera, Álvarez, 2004). These sensors were used to measure the difference of temperature at 1 and 2 m, and for correcting (using humidity data) and scaling (using temperature data) the air temperatures measured inside the greenhouses with the sonic anemometers.

The speed of sound measured by the sonic anemometers depends both on temperature and on humidity. Therefore, in humid air it is necessary to correct the temperature of air T_S [ºC] obtained by the 3D sonic anemometer from the speed of sound. From the data of inside humidity recorded by the fixed sensors, we can obtain the specific humidity q [kg kg^{-1}] and calculate the corrected sonic temperature T_{SC} [ºC] using the following expression (Tanny, Haslavsky, Teitel, 2008):

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

To analyse the differences of temperature and humidity between the three greenhouses analysis of variance (ANOVA) with Statgraphics Plus v4.1 (Manugistics Inc., Rockville, Maryland, USA) was carried out. Furthermore, multiple regression analysis evaluated the effect of the volumetric flow rate generated by the extractor fans and moisture content in outside air on the water consumption of greenhouse 1-PF.

3. RESULTS AND DISCUSSION

In order to avoid discrepancies in the results, all eight measurement tests (Table 2) were carried out at around noon, when the outside climatic conditions remained relatively stable. Four of the measurement tests were carried out under prevailing *Levante* wind (from the northeast), and the other four with *Poniente* wind (from the southwest). For both northeast and southwest winds the polyethylene sheet that divided the experimental greenhouse may have a slight bearing on the natural airflow inside the naturally ventilated greenhouses 2-FG and 3-NV.

Let us first analyse the airflow inside greenhouses 1-PF and 2-FS with the two cooling systems; we shall then go on to estimate the air renovation rate per hour in these greenhouses; thirdly we shall compare the different levels of turbulence observed inside these greenhouses; next the temperature distribution will be analysed in the same greenhouses in comparison with the pattern observed in greenhouse 3-NV; finally, we shall analyse the differences in electricity and water consumption between the two cooling systems.

3.1. Air Velocity and Airflow Direction

For the analysis of the effect of the different cooling treatements on the airflow generated inside the greenhouses we have plotted in Figure 5 the two-dimensional resultants of air velocity on the XY plane (*l*) and the XZ plane (*v*) with the frequency histograms of velocity directions (depicted as polar plots). Vectors in blue correspond to a height of 1.75 m, while those in red represent the airflow at the roof vent.

The extractor fans (greenhouse 1-PF) produced a suction of air from the evaporative pad, where air entering the greenhouse was humidified, to the extractor fan where heated air exited the greenhouse. For all the measurement tests, air passed through the greenhouse, following the direction perpendicular to the greenhouse ridge and parallel to the ground (Fig. 5). However, we can observe that in the centre of the greenhouse airflow was distorted (Fig. 5) by the presence of the entrance chamber in the western side of the greenhouse (Fig. 3b). The direction in which the air advances and the fact that there is little mixing of air inside the greenhouse are responsible for the temperature gradients which are characteristic of these cooling systems (López, Valera, Molina-Aiz, Peña, 2010). The airflow produced by the extractor fans (1-PF) generated a more uniform air direction than the natural ventilation in the greenhouse with fog system (2-FS). Besides, the magnitude of air velocities was lower in greenhouse 2-FS, with an average value for the 8 measurement tests of 0.24 m s^{-1} (standard deviation of 0.22 m s^{-1}) than in greenhouse 1-PF, with an average value of 0.40 m s^{-1} (standard deviation of 0.23 m s^{-1}). The maximum value of air velocities inside the two experimental greenhouses was 1.04 m s^{-1}, the critical value above which excessive flow can be detrimental to the crop (ASHRAE, 2009). However, the mean air velocity entering the greenhouse through the pad ranged between 0.5 and 0.7 m s^{-1}, below the recommended value of 1.27 m s^{-1} for corrugated cellulose pads (ASABE, 2008).

On the other hand, in greenhouse 2-FS, the frequency histograms show greater fluctuation in the direction of the air (greater mixing of air) than in greenhouse 1-PF, although the air velocity was much lower (Fig. 5). The proximity of the fans in greenhouse 1-PF to the southern side of greenhouse 2-FS has been observed to foment the entrance of air into the latter greenhouse (Fig. 5), thereby possibly improving its ventilation capacity. However, the entrance of warm air from greenhouse 1-PF into greenhouse 2-FS limited their cooling capacity. A distance of 4 to 5 fan diameters (5.1 to 6.4 m for our fan extractors) should be maintained between the fan discharge and any nearby obstructions (ASABE, 2008). However, the experimental greenhouses 1-PF and 2-FS were only 3 m apart (Fig. 2). On the whole, the separation between commercial greenhouses in the area of Almería is less than 3 m. In this particular case, the extractor fans should have been installed on the south side of greenhouse 1-PF to avoid the negative influence on ventilation and cooling of greenhouse 2-FS (Fig. 2).

The airflows generated in both greenhouses with evaporative cooling systems (1-PF and 2-FS) with (Fig. 5c) and without shade screens (Fig. 5b) were very similar, although their use reduces the natural ventilation capacity (in greenhouse 2-FS). The use of the two horizontal airflow fans (working in opposing directions) produced a circular current of air inside the naturally ventilated greenhouse 2-FS (Fig. 5d).

These interior ventilators (Fig. 5d) allowed similar values of air velocity inside greenhouses 2-FS (between 0.10 and 0.98 m s^{-1}) and 1-PF (between 0.10 and 1.04 m s^{-1}), although no change was observed in air velocity though the greenhouse windows. Without the fans and with low to moderate wind speeds, the air velocity in greenhouse 2-FS was considerably lower (between 0.06 and 0.62 m s^{-1}) than in greenhouse 1-PF (between 0.09 and 0.98 m s^{-1}) as Figs. 5a and 5c illustrate.

Under conditions of natural ventilation the airflow does not follow such a clear, uniform pattern as with mechanical ventilation. The airflow pattern and the exhange rate inside naturally ventilated greenhouses depend on the interaction between the thermal or buoyancy effect, proportional to the inside-outside air temperature difference ΔT_{io}, and the wind effects (depending on wind speed). Papadakis, Mermier, Meneses and Boulard (1996) observed that the thermal buoyancy effect could not be neglected in a plastic greenhouse (with continuous roof and side openings) when the wind speed was lower than 1.8 m s^{-1}, although it had less bearing than the wind effect. However, to determine when one of the two effects is predominant it is preferable to use the ratio between wind velocity and the root of the inside-outside temperature difference. Kittas, Boulard and Papadakis (1997) observed, in a Mediterranean greenhouse with ridge and side openings, that the wind effect predominated over the thermal buoyancy effect when this ratio $u_o/\Delta T_{io}^{0.5}$ became greater than 1, while Bot (1983) set this limit at 0.3 for a Venlo type greenhouse with only continuous roof windows.

When the *Levante* wind prevails the air has been observed to enter through the northern windward vent, which is free of obstacles, and through the southern side vent. Once inside, due to buoyancy the air rises and leaves through the roof vent, which is on the leeward side (Figs. 5a, 5b and 5c). In previous works carried out in the experimental greenhouses, at low wind speed the thermal effect was predominant, with outside air entering through the side vents and leaving through the roof vent (López, 2011). However, for strong *Poniente* wind, as the roof vent is windward, air entered through the roof vent and exited the greenhouse through the side vents. Thus, for *Poniente* wind (Fig. 5d), depending on the wind intensity (wind effect) and the difference between inside and outside temperatures (thermal effect), ventilation occurs differently. In these tests (A1, C1, D1 and D2) the $u_o/\Delta T_{io}^{0.5}$ ratio was less than 1 but more than 0.3 (0.55, 0.72, 0.86 and 0.63, respectively), making it difficult to establish a ventilation pattern. During the same test air has been observed to both enter and leave through the roof vent. Moreover, the southern side was affected by the fans of greenhouse 1-PF.

To analyse the airflow uniformity, the heterogeneity of l and v has been calculated at the measurement points inside the greenhouse (excluding the points closest to greenhouse sides), dividing their standard deviation (σ_l and σ_v) by the mean value (Kittas, Katsoulas, Bartzanas, Mermier, Boulard, 2008). On the whole, the heterogeneity of both parameters was greater in greenhouse 2-FS (Table 3), which indicates that the air mixed to a greater degree, which leads to greater homogeneity of the microclimate (see section 3.4).

3.2. Air Exchange Rates

For greenhouse 2-FS, the mean ventilation flow rate has been calculated using the air velocities $\bar{u}_{x,j}(t)$ measured by the 3D sonic anemometer at each position j at vents through a process of scaling with the wind speed (Molina-Aiz, Valera, Peña, Gil, López, 2009):

$$G = \sum_{j=1}^{m} S_j \bar{u}_{x,j}(t) \frac{\bar{u}_o}{u_o(t)} \tag{2}$$

where S_j is the unit surface of vent corresponding to each measurement point j, $\bar{u}_{x,j}(t)$ is the perpendicular component to the greenhouse openings of mean air velocity at minute t at each point j in the greenhouse openings, $\bar{u}_o$ is the average wind speed for the overall test period (several hours) and $u_o(t)$ is the instantaneous value (average for each minute t) .

The air exchange rates R (in h^{-1}) for greenhouses 1-PF and 2-FS were calculated by dividing the volumetric flow rate G (in m^3 s^{-1}) obtained from equation 2 by the greenhouse's volume (2682 m^3 for the western sector of both greenhouses) and multiplying by a conversion factor of time units (3600 s h^{-1}). For greenhouse 1-PF, the air exchange rates were obtained from the mean values of air velocity entering the greenhouse through the pad (with a surface area of 33.25 m^2) measured at ten locations indicated in Figure 3 (five locations with one 3D anemometer and five 2D anemometers that remained fixed). For greenhouse 2-FS, air velocity was measured at the roof vent with 5 fixed 2D anemometers and at the two side vents with the 3D anemometers at 5 different locations (Fig. 3).

The air exchange rates obtained for greenhouse 1-PF were in the range of 27-31 h^{-1} (Fig. 6). These values fall between the value of 20 h^{-1} suggested by Sethi and Sharma (2007) and the optimum value of 45-60 h^{-1} recommended by Hellickson and Walker (1983). However, in the greenhouse with fog system (2-FS) the air exchange rates were similar to these observed in the province of Almería (5 to 15 h^{-1}) in naturally ventilated Almería-type and multispan greenhouses (Molina-Aiz, 2010; López, 2011). The use of fans inside the greenhouse does not lead to an increase in the air exchange rate, but these fans contribute to homogenizing inside air. We recommend placing them next to the lateral vents in order to benefit from both effects (increase in the air exchange rate and homogenizing inside air).

3.3. Turbulence Flow Characteristics

3.3.1. Turbulence Intensity and Energy Levels

Average turbulence intensity *i* was greater in the naturally ventilated greenhouse 2-FS (0.42-0.82) than in greenhouse 1-PF (0.33-0.38), where the extractor fans normally generate a less turbulent airflow (Ouyang, Dai, Li and Zhu, 2006), as Table 4 illustrates. Besides, this mechanically generated airflow can be stabilised when passing through two porous media (Fang, 1997), the insect-proof screen and the humidified pad installed at the opening. In general, the average turbulence intensities measured within both greenhouses (Table 4) are similar to those observed inside a banana screenhouse (0.49±0.12) by Tanny, Haijun and Cohen (2006), and to those measured in a naturally ventilated multi-span glasshouse (0.16 to 0.47 for the *x* direction) by Boulard, Wang and Haxaire (2000). In greenhouse 2-FS, the three components of turbulence intensity were similar. In greenhouse 1-PF, however, turbulence intensity was less for the vertical component (i_z) than for the others (i_x and i_y), which may be the reason behind the vertical temperature gradient observed in this greenhouse.

As regards turbulence kinetic energy *k* and turbulence energy dissipation rate ε, we can observe that the energy levels inside greenhouse 2-FS were only greater than in greenhouse 1-PF for test B2, which was carried out in conditions of strong *Levante* wind, and for tests D, when the internal fans were activated. The use of these fans inside the greenhouse does not lead to an increase in the air exchange rate, but these fans could contribute to homogenizing inside air, increasing turbulence kinetic energy. We recommend placing them next to the side vents to augment the ventilation airflow as well as the value of *k*.

The *k* values obtained (Table 4) inside the experimental greenhouses equipped with insect-proof screens were much lower than those observed by Boulard, Wang and Haxaire (2000) in the centre of an empty greenhouse tunnel without screens (0.28 $m^2\ s^{-2}$) and close to the windward opening (reaching 1.44 $m^2\ s^{-2}$). However, the values of the turbulent kinetic energy measured in our 8 measurement tests (Table 4) were similar to these observed in a multi-span glasshouse, ranging from 0.003 to 0.068 $m^2\ s^{-2}$ (Wang and Deltour, 1999).

The values of turbulence kinetic energy measured in both experimental greenhouses indicate a low capacity of the inside airflow to mix and to transport heat and water vapour (Tan-Atichat, Nagib, Loehrke, 1982). This may constitute a major drawback for the low capacity of the airflow to remove the solar energy absorbed by the soil when the seedlings are transplanted into the greenhouse at the end of summer and low capacity to remove water vapor generated by a mature crop at the end of growing season in late spring or early summer.

3.3.2. Measures of Turbulence Macroscale

The macroscale represents the dimension of the most energetic eddies, which have the most significant effect on the mixing of air and therefore on ventilation (Tanny, Haslavsky, Teitel, 2008). In greenhouse 1-PF the airflow is characterised by similar levels of macroscale (L_{ix}, L_{iy}, L_{iz} and L_i) over all eight tests. This type of forced ventilation generates a predominant airflow perpendicular to the pads, meaning that the dimension of the turbulence scales is greater for the longitudinal component L_{ix} than for the transversal L_{iy} and vertical L_{iz} ones (Table 5). Consequently, the air mixes less and horizontal temperature gradients are generated.

In greenhouse 2-FS, the turbulence scales increase with the air velocity, reaching maximum values in tests B1, B2 and C1. The use of fans in the greenhouse (tests D) leads to an increase in the turbulence scales compared to the other tests. Under conditions of natural ventilation (greenhouse 2-FS), with moderate wind speeds (tests A2, B1, B2 and C2), L_{iy} is higher than in greenhouse 1-PF, which suggests greater transversal mixture of air. In greenhouse 2-FS the macroscale is highest for L_{ix} and lowest for L_{iz}, indicating more mixture of air on longitudinal and transversal planes than in the vertical one. The presence of the insect-proof screens installed in the openings of both experiamental greenhouses may reduce the average values of the macroscale L_{ix}, which is much lower (ranging from 0.13 to 0.61 m) than those measured in an unscreened tunnel greenhouse (1.19-2.11 m) (Boulard, Wang and Haxaire, 2000).

3.3.3. Discrete Energy Spectrum

Breaking down the time series in frequency components allows us to see how the eddies of the different scales contribute to the overall turbulence (Fig. 7). In greenhouse 1-PF the energy levels were highest when the fans were activated (Fig. 7c) and in conditions of strong *Levante* wind (Fig. 7a). In both cases the energy levels were greater than in greenhouse 1-PF. In conditions of weak to moderate wind, the energy levels in greenhouse 2-FS without internal fans were lower than in greenhouse 1-PF (Fig. 7b). In greenhouse 1-PF, unlike greenhouse 2-FS, the energy levels did not depend on the wind characteristics.

Inside the greenhouse, the slope of the β spectrum was less for greenhouse 1-PF (Table 5) than for greenhouse 2-FS (with values close to 5/3, characteristic of natural airflows). On activating the fans inside greenhouse 2-FS the slope of the spectrum decreases, approaching the values recorded in greenhouse 1-PF. These low values of β measured when using the extractor and inside fans are characteristic of mechanical airflows, and they indicate that turbulent energy is distributed uniformly over the range of frequencies considered (Ouyang, Dai, Li, Zhu, 2006). Mechanical airflows usually produce a higher transport of energy at high frequencies than natural airflows, in which energy is transported at low frequency (Ouyang, Dai, Li, Zhu, 2006).

In short, the levels of turbulence intensity measured inside greenhouse 1-PF were lower than the values observed in greenhouse 2-FS. This lower intensity of turbulence of the cooling airflow reduces the mixing of the inside air with the outside air entering the greenhouse through the pad (López, Valera, Molina-Aiz, Peña, 2010). The levels of turbulence kinetic energy and turbulence energy dissipation rate measured inside greenhouse 2-FS were higher than in greenhouse 1-PF on days with strong winds.

3.4. Interior Microclimate

The higher air exchange rate (which implies greater evacuation of the heat accumulated inside the greenhouse) and the higher water consumption in greenhouse 1-PF with respect to greenhouse 2-FS (the water consumed is used to reduce temperature and increase the moisture content of air entering the greenhouse through the evaporator panels) implies that greater reductions in temperature were obtained in the former. The lower inside temperatures obtained using the pad-fan system for the empty greenhouses studied in this work suggest that this evaporative system is the most effective for cooling greenhouse when young plants are transplanted at the end of the summer.

However, the differences observed in airflow characteristics between these greenhouses have repercussions on the interior temperature distribution. In greenhouse 2-FS there was greater mixing of air than in greenhouse 1-PF, where the airflow was more uniform, entering through the evaporative pads and heading towards the extractor fans. The results obtained in this study are conditioned by the fact that the greenshouse was empty (without crop).

3.4.1. Mean temperature and relative humidity values inside the greenhouse

On the whole, air temperature inside greenhouse 3-NV was lower than in greenhouse 2-FS due to the lower ventilation rate in the latter. The temperature in greenhouse 2-FS was lower than in greenhouse 3-NV only under conditions of strong wind and with the vents fully opened (test B2; u_o=7.51 m s^{-1}) or when combined with a shade screen (test C2; u_o=4.98 m s^{-1}) (Table 6). Fogging systems have been successfully developed for greenhouse cooling. However the lack of control strategies, in combination with ventilation systems, especially passive ventilation, has limited their capabilities (Villarreal-Guerrero *et al.*, 2012b).

In greenhouse 1-PF the temperature was always below the values recorded in the other two greenhouses (Table 6). The lowest temperature difference recorded was in tests C, when a shade screen was used. Differences of up to 11.6ºC (test B2) were observed between temperatures in greenhouses 1-PF and 3-NV, while the maximum difference between greenhouses 1-PF and 2-FS was 10.4ºC (test A1).

With no crop in the greenhouse and at the hottest time of day, the temperature differences between greenhouses 1-PF and 2-FS (Table 6) were much higher than the 3.4ºC reported by Willits and Li (2005) in their comparison of the same two systems in the presence of a tomato crop (cooling effect of evapotranspiration of the crop) over two years. The mean differences in temperature between both systems can be smaller if the analysis is carried out over long periods of time and with a crop inside the greenhouse However, special attention should be paid to certain situations (hottest times of the year and/or when the crop has recently been transplanted) in which the cooling requirements may be greater, making the pad-fan system preferable to the fog one, as the results obtained in the present work illustrate.

Commonly grown horticultural species in the Mediterranean region are adapted to mean temperatures of 17 to 28ºC, with maximum and minimum limits of 12 and 32ºC, respectively (Amsen & Nielsen, 1988). In greenhouse 1-PF, without crop, the mean temperature ranged between 29.6ºC (test C2 with shade screen) and 35.0ºC (test A2), but if a crop were present it would have a cooling effect due to crop evapotranspiration and the air temperatures would be lower. Thus, the temperatures recorded in the area occupied by a mature tomato crop in greenhouse 1-PF in mid-summer 2008 were 3 to 4ºC below the outside temperature of 29.9ºC (López, Valera, Molina-Aiz, Peña, 2010). In addition, evapotranspiration of the crop augmented the water content of the inside air transported by the forced airflow, and as a result the air humidity increased at greater distances from the evaporative pads, whereas in the assays without crop we observed that the air accumulates greater water content closer to the pads (López, Valera, Molina-Aiz, Peña, 2010).

In greenhouse 2-FS the mean temperature (between 36.4ºC and 43.7ºC) was much higher than the recommended maximum values. This situation could arise in commercial greenhouses with a recently transplanted crop in early August. The pad-fan system combined with a shade screen proves to be the best alternative for bringing forward transplanting in summer, improving the results obtained in 2008 (López, Valera, Molina-Aiz, Peña, 2010) with the pad-fan system without the shade screen and in accordance with the results obtained in 2009 (López, Valera, Molina-Aiz, Peña, 2012).

The relative humidity measured inside greenhouse 1-PF at a height of 1 m above the ground was 25% and 26% greater on average than in greenhouses 2-FS and 3-NV, respectively (Table 6). However, these differences only reached 7% and 9%, respectively, at a height of 2 m. We can observe that the evaporative effect of the pad is greater at 1 m, making this system suitable for use when young seedlings are transplanted to the greenhouses.

The fog system analysed in this work seems to be inadequate to increase the relative humidity inside the greenhouse. Thus, the average increase in humidity with respect to the outside air was 6% inside greenhouse 2-FS for days with outside relative humidity lower than 60%. Moreover, for the three days with higher outside relative humidity, in the region of 70% (tests A1, C1 and D2), the relative humidty was greater in greenhouse 3-NV than in greenhouse 2-FS. The greater ventilation capacity in greenhouse 3-NV allows the inside humidity to be increased with the water vapour supplied by the humid air entering the greenhouse through the vent openings. However, this effect observed in the empty greenhouses (similar to a greenhouse with young plants) could be different when better ventilation can allow elimination of the water vapour supplied by the evapotranspiration of a well-developed crop.

The low pressure water/air fog system is the most widespread in Almería's commercial greenhouses as it is cheaper than the pad-fan system. However, the capacity and density of nozzles installed seem insufficient to increase inside humidity and reduce temperature. Further research could focus on improving this cooling system by increasing the density of nozzles and their capacity.

In general, the temperature and relative humidity measured by the fixed sensors at each location were more stable over the three-hour measurement period for the 8 tests in greenhouse 1-PF (standard deviation of ±0.6ºC and ±1.7%, respectively) than in greenhouses 2-FS (±1.3ºC and ±2.4%) and 3-NV (±1.6ºC and ±2.8%). The use of the pad-fan system seems to allow greater temporal stability of temperature and humidity. However, this advantage is offset by their lower spatial uniformity. Thus, the total variation in space and time (standard deviations in Table 6) was similar for the three greenhouses.

3.4.2. Temperature gradients

The analysis of temperature gradients inside the greenhouse can help growers to optimise fertilisation and irrigation systems (Boulard & Wang, 2002) and avoid the problem of thermal stress of seedlings being transplanted in areas of air stagnation. The average differences between temperatures measured at 2 m and 1 m above the ground by the fixed sensor (vertical temperature gradients ΔT_v) was around 25 times greater in the greenhouse with mechanical ventilation (1-PF) than in the naturally ventilated greenhouses (2-FS and 3-NV), with a maximum value of 6.7ºC m^{-1} for test B1 (Table 6). The cooling effect produced by the pad-fan system was greater at 1 m (where relative humidity was greater) than at 2 m above the ground. Consequently, we can observe that the temperature differences between the three greenhouses were greater at a height of 1 m than at 2 m (Table 6). At the lower height, where in late summer the young crop has a low evapotranspiration cooling capacity, the average temperature for the 8 tests was about 10ºC lower in greenhouse 1-PF than in the naturally ventilated greenhouses (2-FS and 3-NV). However, at 2 m this average difference was only about 5ºC.

On the other hand, the horizontal gradients measured with the 3-D sonic anemometer in the mechanically ventilated greenhouse 1-PF were between 0.16 and 0.26ºC m^{-1} (Table 6). These values are close to the 0.13ºC m^{-1} reported by Kittas, Bartzanas and Jaffrin (2003) and the 0.27ºC m^{-1} observed by López, Valera, Molina-Aiz and Peña (2010) in greenhouses with pad-fan cooling systems. The use of shade screen in combination with pad-fan system can reduce horizontal temperature gradients in greenhouses (Willits & Peet, 2000; López, 2011).

Thus, the lowest horizontal gradient in experimental greenhouse 1-PF was recorded in test C1 with the shade screen (0.16ºC m^{-1}). The values measured without screen ranged from 0.19 to 0.26 ºC m^{-1}. Overall, the horizontal temperature gradients measured in the greenhouse with fog system and natural ventilation (2-FS) were lower than those observed in greenhouse 1-PF (Table 6).

3.4.3. Greenhouse temperature distribution

To study the heterogeneity of temperature distribution (Figs. 8-10) we have also used the air temperatures measured at a height of 1.75 m with the 3-D anemometers at 30 different positions inside the two experimental greenhouses with evaporative cooling systems (1-PF and 2-FS). To compare the temperatures measured at different times in the 30 different positions in each experimental greenhouse (Fig. 3), we need to consider the effect of changes in outside temperature throughout duration of the tests. This problem can be overcome using the difference in air temperature between the centre of the greenhouse and the external air stream as a scaling parameter (Boulard, Wang and Haxaire, 2000). However, this method can be problematic when there is little difference between inside and outside temperatures (Lopez, 2011). Consequently, we have used the average inside and outside temperatures as the parameter to scale the inside temperature measured with the anemometers. The inside-outside temperature difference used in Figs. 8-10 has been calculated as (López, Valera, Molina-Aiz and Peña, 2012):

$$\Delta T^{c}_{io,j}(1) = T_{sc,j}\,\frac{T_o + T_i}{T_{o,j} + T_{i,j}} - T_o \qquad (3)$$

where $T_{sc,j}$ is the corrected sonic temperature [Eq. (1)] for position j inside the greenhouse, T_o and T_i are the mean outside and inside air temperatures during the test and $T_{o,j}$ and $T_{i,j}$ are the mean outside and inside air temperatures recorded by the fixed sensors over the 3 minutes used for measurement at position j.

Figure 8 illustrates how the extractor fans in greenhouse 1-PF favour the entrance of air through the southern side vent in greenhouse 2-FS. In this greenhouse, with light wind and the side vents open 50%, the temperature in almost the whole greenhouse is between 11 and 12ºC higher than outside. In conditions of strong *Levante* wind, and with the windows fully open, the situation is more favourable (Fig. 9). However, we can observe the negative effect produced on ventilation by the warm airflow entering greenhouse 2-FS from the extractor fans located in the side wall of greenhouse 1-PF. At the leeward side vent of greenhouse 2-FS (close to the extractor fan of greenhouse 1-PF) the air enters 4ºC warmer than through the windward side vent (Fig. 9). The cooling capacity of the fog system was limited by the entrance of warm outside air arriving from the other greenhouse. The use of internal fans has the effect of homogenizing the greenhouse air temperature (Fig. 10), but it does not foment the exchange of air with the outside since the fans are located 4 m from the side vents.

Temperature maps allow us to visualise the increase in temperature in greenhouse 1-PF at greater distances from the evaporative pads. The entrance chamber to the greenhouse (Fig. 3) gives rise to an increase in temperature in the northwestern corner (Figs. 8, 9 and 10). In greenhouse 1-PF without the shade screens, the temperature differences between the coldest and hottest points ranged from 8.5 to 11.4ºC (Figs. 8 and 9). The use of the shade screen in this greenhouse (tests C) slightly reduces these temperature differences to 7.3 and 8.0ºC.

3.5. Electricity and water consumption

Functioning continuously for one hour, greenhouse 2-FS consumed more electricity (7.2 kWh - tests A; 8.2 kWh - tests B and C; 8.9 kWh - tests D) than greenhouse 1-PF (5.1 kWh). On the other hand, consumption of water was much greater in greenhouse 1-PF (average value of 115.1 l h^{-1}, minimum of 44.3 l h^{-1} for test A1 and maximum of 209.3 l h^{-1} for test B1) than in greenhouse 2-FS (average value of 9.4 l h^{-1}, minimum of 5.3 l h^{-1} for test C1 and maximum of 16.7 l h^{-1} for test D1), which may be due to the greater temperature drop in the former. We should bear in mind that water is a valuable natural resource which must be managed responsibly.

The results obtained in the empty greenhouses show that the use of the pad-fan system can be more adequate than fog system for cooling greenhouses when seedlings are transplanted at the end of the summer. At this stage the plants' evapotranspiration is very low (Orgaz *et al*., 2005) and their cooling effect is negligible. In these conditions, the greater water consumption of the pad-fan system has an appreciable effect on temperature reduction. However, a well-developed crop can reach average daily values of evapotranspiration in Almería of 4-4.5 mm day^{-1} (Orgaz *et al*., 2005), equivalent to 0.40-0.45 l h^{-1} m^{-2}, i.e. 4 times greater than the pads' consumption (about 0.107 l h^{-1} m^{-2}) and 50 times greater than that of the fog system (about 0.008 l h^{-1} m^{-2}).

Results showed a statistically significant increase in the measured water consumptions in greenhouse 1-PF m_w with the capacity of increasing the water vapour content of the air entering the greenhouse (expressed as the difference between the absolute saturation humidity x_{so} and the absolute humidity of the outside air x_s) and the volumetric flow rate G. The multiple regression analysis carried out showed that the results can be fitted (at 99% confidence level, with R^2 =0.95 and p-value=0.0005) to a multiple linear regression model shown by the equation:

$$m_w\ [\mathrm{kg\ h^{-1}\ m^{-2}}] = 0.150\ G\ [\mathrm{m^3\ s^{-1}}] + 347.13\ (x_{so} - x_o)\ [\mathrm{kg\ kg^{-1}}] - 3.744 \quad (4)$$

4. CONCLUSIONS

The use of internal fans in the greenhouse in combination with the fog system produces an increase in the turbulence kinetic energy that foments mixing of the air and therefore tends to homogenize inside air temperature distribution. However, the location of the internal fans tested in the present work, rather distant from the vents, does not increase the entrance or exit of air. We recommend placing them next to the side vents to increase the airflow through the greenhouse.

The evaporative pad cooling system maintains more favourable conditions in all tests when compared to both the fog system and natural ventilation. The greatest decreases in temperature were achieved by combining the pad-fan system with a shade screen (ΔT_{io} = 1.4ºC and 1.8ºC). The pad-fan system combined with a shade screen proves to be the best alternative for cooling greenhouses when crop evapotranspiration is lower (in nurseries or just after transplant in commercial greenhouses), and it may allow the date for transplant to be brought forward at the end of summer.

Temperatures in the empty greenhouse with the pad-fan system were up to 11.6ºC lower than in the naturally ventilated one, and up to 10.4ºC lower than in the one equipped with the fog system. On average, the differences in temperature at 1 m height (where young plants grow) between the greenhouse with pad-fan system and the naturally ventilated one with and without fog system were 10.1ºC and 9.8ºC, respectively. In a previous work, we observed an averaged difference in temperature between the pad-fan system and the naturally ventilated greenhouse of only 5.0ºC, a well-developed tomato crop (leaf area index of 3 $m^2 m^{-2}$) was growing (López, Valera, Molina-Aiz, Peña, 2012).

The low pressure water/air fog system tested in this work (one of the most commonly used in the commercial greenhouses of Almería due to its low price compared to the pad-fan system) did not prove capable of maintaining temperatures below the minimum recommended values for most horticultural crops (between 36.4 and 43.7ºC). On the other hand, in some tests, the pad-fan cooling system maintained a mean air temperature of between 29.6 and 35.0ºC, which is more suitable for such crops.

The main drawbacks of the pad-fan cooling system were the high water consumption and the horizontal and vertical temperature gradients. We have observed a maximum temperature difference between pads and fans of up to 11.4ºC and a maximum difference in temperature between 1 m and 2 m height of 6.7ºC. The water comsuption was about 0.107 $l\ h^{-1} m^{-2}$ for the pad-fan system and about 0.008 $l\ h^{-1} m^{-2}$ for the fog system in the empty greenhouses. These values were 4 and 50 times lower, respectively, than the water supply (0.40-0.45 $l\ h^{-1} m^{-2}$) produced by a well-developed crop evapotranspirating 4-4.5 mm day^{-1} (Orgaz *et al*., 2005). The water flow supplied by the low pressure water/air fog system tested was insufficient to reduce the inside temperature below 36ºC and to increase relative humidity up to 50%. The design parameters habitually used for this type of fog system need be improved by increasing the nozzle capacity and density.

The methodology presented in this work has allowed us to quantify accurately the greenhouse ventilation rate under varying conditions of outside wind, and to describe accurately the airflow and the temperature distribution inside the greenhouse. It has also allowed us to characterise the turbulence of the airflow inside the greenhouse, providing useful information for future validations of simulations based on CFD.

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Nomenclature

E	spectral density [m^2 s^{-1}]
f	frequency [Hz]
G	volumetric flow rate [m^3 s^{-1}]
G_p	volumetric flow rate through the evaporative pads [m^3 h^{-1}]
HR	relative humidity [%]
i	turbulence intensity
k	turbulence kinetic energy [m^2 s^{-2}]
l	two-dimensional horizontal resultant of air velocity in XY plane [m s^{-1}]
L_i	integral length scale [m]
m	number of measurement points in vents
m_w	water consumption of wed pad surface [kg h^{-1} m^{-2}]
q	specific humidity [g g^{-1}]
R	incoming shortwave radiation [W m^{-2}]
$R(t)$	normalized autocorrelation function [m^2 s^{-2}]
S_A	greenhouse surface area [m^2]
S_j	unit surface area corresponding to each measurement point in the vent [m^2]
S_V	vent surface area opened [m^2]
T	temperature [ºC]
ΔT_h	horizontal temperature gradient [ºC m^{-1}]
ΔT_v	vertical temperature gradient [ºC m^{-1}]
t	time [s]
t_{int}	integral time scale [s]
u	air velocity [m s^{-1}]
u'	fluctuating component of air velocity [m s^{-1}]
v	two-dimensional vertical resultant of air velocity in XZ plane [m s^{-1}]
x	absolute humidity of air [g g^{-1}]
$X(f)$	fast Fourier transfer (FFT) of sample data
$X(t)$	sample data
$X^*(f)$	conjugate complex number of X
x_p	absolute humidity of the air leaving the wed pad [kg kg^{-1}]
x_s	absolute humidity in saturation [kg kg^{-1}]

Greek Letters

Δ	difference
β	power spectrum exponent
δ_t	increment of time [s]
ε	turbulence energy dissipation rate [m^2 s^{-3}]
η	cooling efficiency [%]
θ	wind direction [º]
ρ_a	air density [kg m^{-3}]
σ	standard deviation

Subscripts

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

Supercripts
c corrected
n normalized

Figure captions

Fig. 1. Incoming shortwave radiation versus monthly means of temperature for Almeria region: (●) Average of 1934-2000 (Molina-Aiz, 2010); (■) Average of 2000-2010 (López, 2011) and average of 2008-2011 in the research farm of the University of Almería (▲).

Fig. 2. Location of the experimental greenhouses at the farm.

Fig. 3**.** Measurement points with sonic anemometers inside the western sectors of greenhouse 2-FS (a) and greenhouse 1-PF (b).

Fig. 4. Details of the experimental setup using 3D anemometers placed inside the greenshoues (a) and 2D anemometers at the roof vent (b).

Fig. 5. Two-dimensional resultants of air velocity in the XY *(l)* and XZ *(v)* plane, and polar plots of airflow direction in the measurement tests type A2 (a), B2 (b), C2 (c) and D2.

Fig. 6. Estimated air exchange rate [h^{-1}] against wind speed [$m\ s^{-1}$]. Greenhouse 1-PF (×); Greenhouse 2-FS with *Poniente* winds (red): measurement tests A1 (□), C1(▲) and D1 and D2 (■); Greenhouse 2-FS with *Levante* winds (blue): measurement tests A2 (◊), B1 and B2 (♦), and C2 (Δ).

Fig. 7. Average energy density spectra in the interior of greenhouse1-PF (□) and greenhouse 2-FS (□). Measurement tests: B2 in conditions of strong Levante wind (a), C1 in conditions of weak Poniente wind (b) and D1 with weak Poniente wind and the fans inside greenhouse 2-FS (c).

Fig. 8. Difference in corrected air temperature (ΔT_{io}^{c}) in greenhouses 1-PF and 2-FS (test A1).

Fig. 9. Difference in corrected air temperature (ΔT_{io}^{c}) in greenhouses 1-PF and 2-FS (test B2).

Fig. 10. Difference in corrected air temperature (ΔT_{io}^{c}) in greenhouses 1-PF and 2-FS (test D1).