Full Length Research Paper

The effects of air velocity on the performance of pad evaporative cooling systems

Metin Dağtekin[1], Cengiz Karaca[2]*, Y ilmaz Yildiz[3], Ali Basçetinçelik[3] and Ömer Paydak[4]

[1]Çukurova University, Ceyhan Vocation High School, Ceyhan, Adana, Turkey.
[2]Mustafa Kemal University, Faculty of Agriculture, Department of Biosystems Engineering, Hatay, Turkey.
[3]Çukurova University, Faculty of Agric., Department of Agriculture Machinery, Adana, Turkey.
[4]Gaziantep Univ. Faculty of Architecture, Department of Architecture, Gaziantep, Turkey.

The aim of this study is to determine the relationship that is found among the velocity of the air passing through the pad, the obtained decrease at the temperature of the air passing the pad and the cooling efficiency at the cellulose based evaporative cooling pad, and be of help to the people who designed and operate these systems. In this study, cellulose based evaporative cooling pad (CELdek ^{R} 7060-15, Munters AB, Kista, Sweden) was used and the tests conducted were completed at 2 periods. Air velocities were determined as 0.5, 1.0 and 1.5 ms^{-1} at the first period, and 0.75, 1.25 and 1.75 ms^{-1} at the second period. The water flow rate at the air velocities, chosen for both periods, was selected as 4 Lmin^{-1}m^{-2}. As a result of the research, it was determined that it was not possible to make a mathematical connection among velocities of the air passing through the pad, obtained decrease at the temperature of the air passing the pad and the cooling efficiency. However, it can be said that the most appropriate air velocity for the pad used in the test should be higher than 0.5 ms^{-1} and lower than 1.5 ms^{-1}.

Key words: Pad evaporative cooling, air velocity.

INTRODUCTION

At hot climate conditions, the inner ambient temperature of the closed agricultural production buildings, such as green house, poultry house, stable and barn may reach very high values. This situation causes heat stress at the produced livestock. Pad evaporative cooling systems are used in such closed production buildings to decrease the ambient temperature. The obtained decrease at the temperature of the air passing the pad and the cooling efficiency, of the pad evaporative cooling systems, depends on structure features of the pad as well as velocity of the air passing through the pad. The obtained temperature decrease of the air passing through the pad and the cooling efficiency values decrease at high air velocities, but this situation is only valid if the flow character of the air (laminar, turbulance) is the same while passing the pad. When the flow character changes, this relationship becomes invalid. Due to the

fact that the velocity of the air passing through the pad affects both the trajectory of the air flow in the pad and the duration of contact with the wet surface, the air flow has a laminar character at excessively low velocities. At the laminar character flow, few layers of air in contact with the moist pad surface, in a particular thickness, find the chance to get moistened. Therefore, the obtained temperature decreases of the air passing the pad and the cooling efficiency decreased.

When the air velocity increased, the laminar flow layers were broken and the flow turns into a turbulance. Thus, the chance of the air getting moistened from the surface of the pad rises. In other words, as the velocity of air increases, the character of flow turns into a turbulance and evaporation speed from the pad surface rises. As a result of this, the obtained temperature decrease of the air passing through the pad and the cooling efficiency values increase (Albright, 1989; ASAE, 1983, 1994; Yıldiz et al., 2010). Some studies made on the pad evaporative cooling systems are summarized and thus explained.

Buffington et al. (1978) compared the evaporative

---

*Corresponding author. E-mail: cengiz.karaca@gmail.com.

cooling pads made of four different materials because of cooling efficiency. In the study, the velocity of the air passing through the pad was kept stable at 0.75 ms^{-1}. At the end of the study, they stated that the maximum cooling efficiency was attained at a cement compounded sugar beet pulp pad, which was followed by a cellulose-based pad, while the lowest efficiency was attained at a pad made of rubberized pig bristle. Koca et al. (1991) assessed the functionality of three different cellulose pads in an experiment set in accordance with the standards of AMCA (Air Movement and Control Association) and ASHARE (American Society of Heating, Refrigerating and Air-Conditioning Engineers). The first pad examined at the study had a thickness of 10 cm with 45 to 45° chamfer angle, while the second had a 15 cm thickness with 45 to 45° chamfer angle, and the last pad had a 15 cm thickness with 30 to 30° chamfer angle. They found that the cooling efficiency of the pad with 10 cm thickness and 45 to 45°chamfer angle, varied from 73 to 90% at 1.5 to 2.5 ms^{-1} air velocity. Dağtekin et al. (1998), carried a study in Çukurova Region (Adana, S. Turkey), to determine the most suitable pad material for evaporative cooling system, by comparing pads made of poplar sawdust, nutshell and cellulose. As a result of the study, the authors determined that cellulose based pad was the most suitable material for the mentioned region; however, they also affirmed the use of nutshell pads due to their inexpensive cost and long durability.

Atikol and Haci evkil (2001), based on the Nicosia-Cyprus region 1996 and 1997 climate data, theoretically studied the availability of evaporative cooling systems in closed environments in the region. As a result of the study, they determined that evaporative cooling systems could be used especially between May and June months in Nicosia-Cuprus region. Cruz et al. (2006) evaluated three different pad materials efficiencies at different temperatures and air velocity in a study conducted in Evora - Portuguese. In the study conducted at four different temperature ranges with an air velocity of 1.6, 3.2, 4.8 and 5.6 ms^{-1} respectively, the highest cooling efficiency (80% and more), was achieved at 3.2 ms^{-1} air velocity at temperatures of 32 to 34°C. Kocatürk and Yild iz (2006) examined some performance characteristics of the cellulose-based evaporative cooling pad at different air velocities in Mediterranean climate conditions. In their study, 0.5, 1.0 and 2.0 ms^{-1} values were chosen for the velocity of the air passing through the pad, while the pad wetting water flow rate (4 Lmin^{-1}m^{-2}) was kept stable. As a result of the study covering the period from June to September, it was determined that the cooling efficiency at the selected air velocity was 77 to 84%, provided temperature decrease was between 6.7 and 5.6°C and the amount of vaporized water varied between 0.078 and 0.210 L min^{-1} m^{-2}. Dağtekin et al. (2006) examined some performance characteristics of an evaporative pad cooling system in a broiler poultry house at Mediterranean climate conditions. As a result of the

study that was completed in July to August, it was determined that the cooling efficiency of the system varied between 70 and 80% and nearly 10°C temperature decreases were achieved at the outer environment air extracted into the poultry house by passing through the pads. Dağtekin et al. (2009: 1) examined the temperature distribution at long axis in a broiler poultry house in Mediterranean climate conditions. The study was carried out in a commercial broiler poultry house with 15.000 hen capacity. According to the evaluation of the measurements made in August and September, the researchers determined that the average values for cooling effectivity were 69.35%, provided the temperature decrease of the air passing through the pad was 5.19°C and the increase at the temperature of the air extracted into the poultry house, passing through the pad until it reaches the expel point, was 1.52°C.

Abdulrahim et al. (2010) assessed the parameters that were effective on design and operation of evaporative cooling systems in greenhouses. The study was conducted at two different air velocities and water flow rates. In their study, the temperature of the air taken from the greenhouse, relative humidity and water temperature values were measured. The measured air and water temperature values were compared to the model of Halasz. The result of the study stated that the measured values were in accordance with the values of the model. In this study, the relationship found among the cooling efficiency, obtained decrease at the temperature of the air passing the pad and the velocity of the air passing through the pad at pad evaporative cooling systems could not be found in the literature review. The velocity of the air passing through the pad is the most important feature that needs to be chosen carefully at the design and operation of these systems (pad evaporative cooling systems). When this feature is not chosen correctly, the increases at the construction and operation costs of the system would be too much. However, the remaining installation and operation costs affect the competition chances of the enterprises negatively.

In this study, the aim is to determine the relationship found among velocities of the air passing through the pad, the obtained decrease at the temperature of the air passing the pad and the cooling efficiency at the pad evaporative cooling systems, and help the people who design and operate these systems.

MATER IALS AND METHODS

The evaporative cooling systems pads, made of different materials, are used in this study. The most common among them is the cellulose based pads. In this study, 100 mm thick cellulose based pad (CELdek ^{R} 7060-15, Munters AB, Kista, Sweden), produced by Mounters Company, was used. This material was commonly used in cooling applications in Turkey. The study was carried out in laboratory conditions and three different experimental sets in the same features were made for the study (Figure 1). Every one of the experimental sets consists of nearly 3.25 m long channel. There
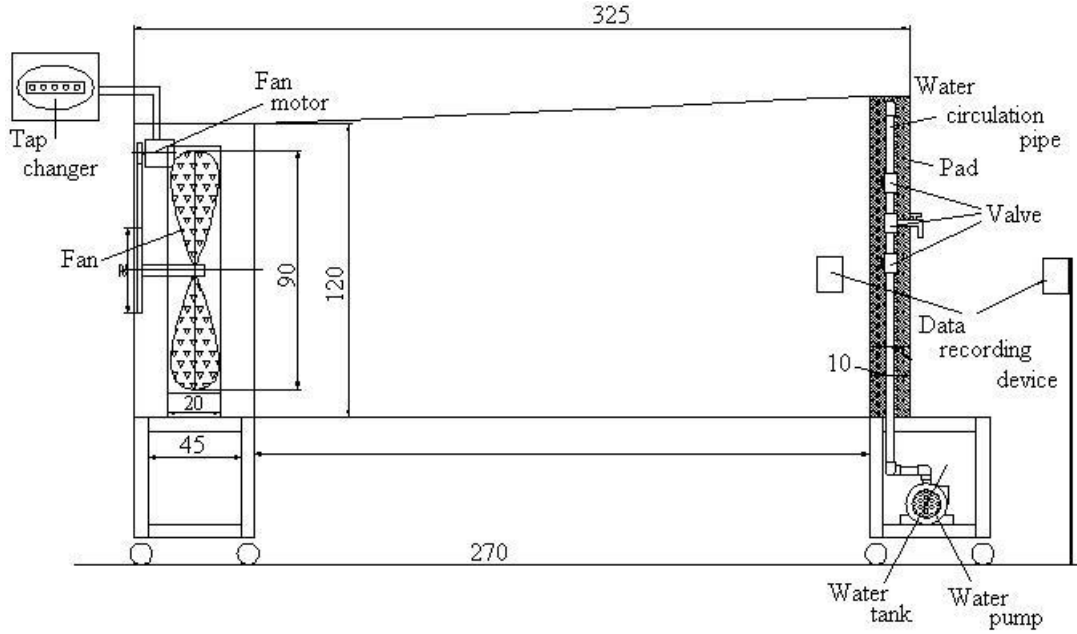
Figure 1. Experimental set.

was a 0.90 m diameter fan (EM-30 Munters) at one end and there was a pad with 1.2 x 1.2 m surface area at the other end of the channel that was made by covering the metal supported roof with canvas. The movement was conveyed to the 15.000 m^3h^{-1} air capacity fans by slip hoop from the electrical motor. An electrical tap changer was attached to every one of the fan motors. With the help of this possibility, the chosen air velocity values had been changed easily at the experiments. Pads were moistened with the water poured from the perforated pipes placed above them.

The water taken from the tank below the pipe with an electrical motor pump was pumped to this pipe. The pad moistening water flow rates were changed with three valves placed at the water pumping line, and the experiments were completed at 2 periods. At the first period, every one of the experiment sets was set to 0.5, 1.0 and 1.5 ms^{-1} air velocities and measurements were taken at 4 $Lmin^{-1}$-m^{-2} water flow rate between 09:00 and 17:00 h. To reduce the possible measure mistakes, measurements were repeated three times. At the second period, every one of experiment sets was set to 0.75, 1.25 and 1.75 ms^{-1} air velocities. Tests were repeated at these velocities and at 4 $Lmin^{-1}m^{-2}$ water flow rate. The measurements were taken between 09:00 and 17:00 h, while 0.5 to 2.0 ms^{-1} values were used as the velocity of air passing the pad at pad cooling systems (Albright, 1989; ASAE, 1983; ASAE, 1994; Yıldız et al., 2010). Therefore, the values of three different air velocities within this interval were used. Equation (1) was utilised to determine the cooling efficiency of the system (ASAE, 1983; Koca et al., 1991; Simmons and Lott, 1996; Bilge and Bilge, 1999; Yıldız et al., 2010):

$$\eta = \frac{t_{kd} - t_{ki}}{t_{kd} - t_{vd}} \; 100 \tag{1}$$

where η is cooling efficiency (%) and tkd, tki and tyd are the dry bulb temperature of the air entering the pad, the dry bulb temperature of the air exiting the pad and the wet bulb temperature of the air entering the pad (°C), respectively.

The dry bulb temperature and relative humidity values measured on the outer surface of pads were analysed as the wet bulb temperature of the air entering the pad and were determined from a psychrometric diagram. By placing values in Equation 1, the cooling efficiency was calculated. The dry bulb temperature of the air and the relative humidity values were measured in the air outlet side of the pads and 0.1 m away from the pads, while the outer environment temperature and the relative humidity values were however, measured at the field where air enters the pads and at a point close to the three experimental sets. For measurement of the dry bulb temperature of the air and relative humidity values, HOBO brand measurement and data recording devices (HOBO U12 Temp/Rh/2 External Data Logger) were used. The sensitivity of the devices, which have a temperature measurement interval of -20 and +50°C and a humidity measurement interval of 5 to 95%, is stated to be ± 0.35°C and ± 2.5%. Consequently, the data recording devices were set to take measurements at 5 min intervals.

For the measurement of the velocity of the air passing the pad, a portable anemometer (OMEGA HHF710, OMEGA, Engineering, Inc, USA) with ± 0.01 ms^{-1} sensitivity was used, while for the measurement of the water flow rate, a measuring cup and a chronometer were used. The calculated and measured values were interpreted with graphical and statistical evaluations. In the statistical analysis, one factor variance analysis and the Duncan multiple comparisons were applied to the calculated values at a randomized factorial experimental design.

RESULTS AND DISCUSSION

The obtained temperature decrease of the air passing through the pad at chosen air velocities (0.5, 1.0 and 1.5 ms^{-1}) and the 4 $Lmin^{-1}m^{-2}$ water flow rate (I. period) on the 1st, 2nd and 3rd day can be seen in Figure 2. The cooling effectivities calculated on the 1st, 2nd and 3rd
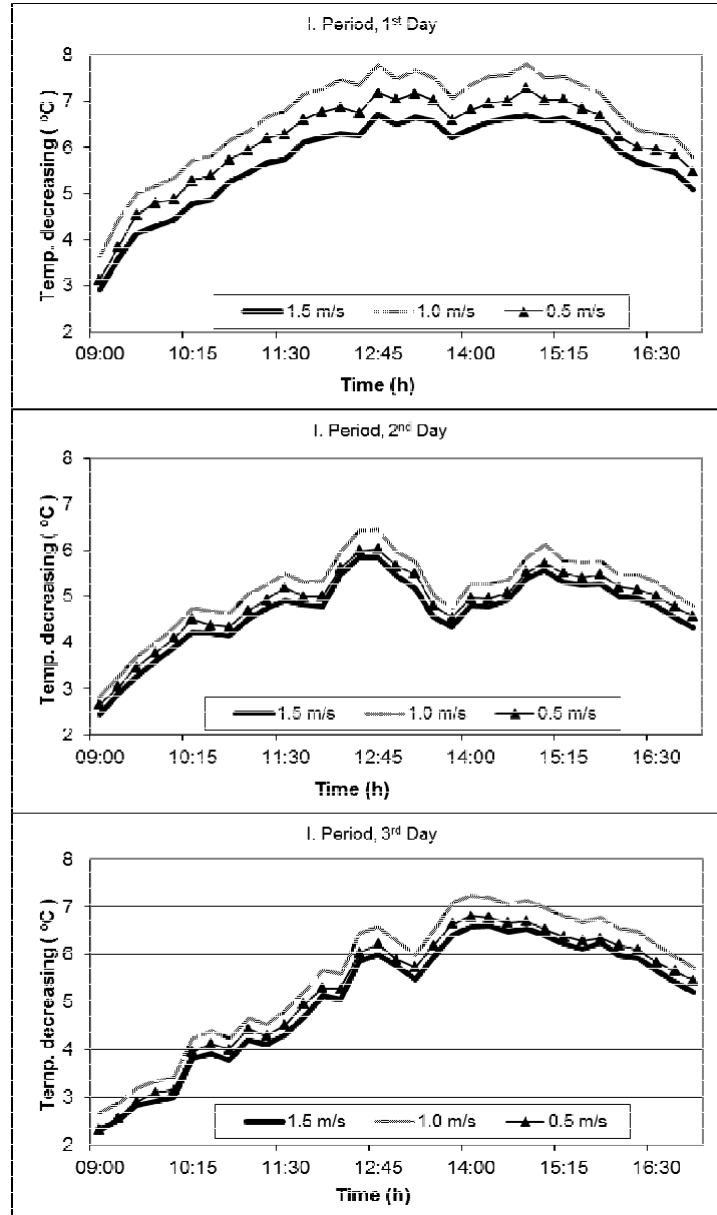
Figure 2. The obtained temperature decreases of the air passing through the pad at chosen air velocities.

day at the same period can be seen in Figure 3. As seen in Figure 2, the decreases made at the temperature of the air passing the pad at chosen air velocities in three repeated tests showed similar changes. In other words, the decrease that occurred at the temperature of the air passing through the pad was the lowest at 1.5 ms^{-1} air velocity, while the highest was at 1.0 ms^{-1} air velocity and the values between both of them was at 0.5 ms^{-1} air velocity. This change was also seen in the calculated cooling efficiency values (Figure 3). The cooling efficiency was the lowest at 1.5 ms^{-1} air velocity, while the highest was at 1.0 ms^{-1} air velocity and the values between both of them was at 0.5 ms^{-1} air velocity. The

result of variance analysis for the effect of the air velocity on temperature decrease and the cooling efficiency can be seen in Table 1.

As seen in Table 1, the effect of air velocity on temperature decrease was unimportant on the 2nd and 3rd day; although, it was important on the 1st day (P<0.01). Nevertheless, the cooling efficiency was important on all three measurement days (P<0.01). DUNCAN test was applied in order to determine the differences of temperature decrease or cooling efficiency averages for each of them at the chosen air velocities belonging to the important days found and the results were presented in Table 2. As seen in Table 2, the
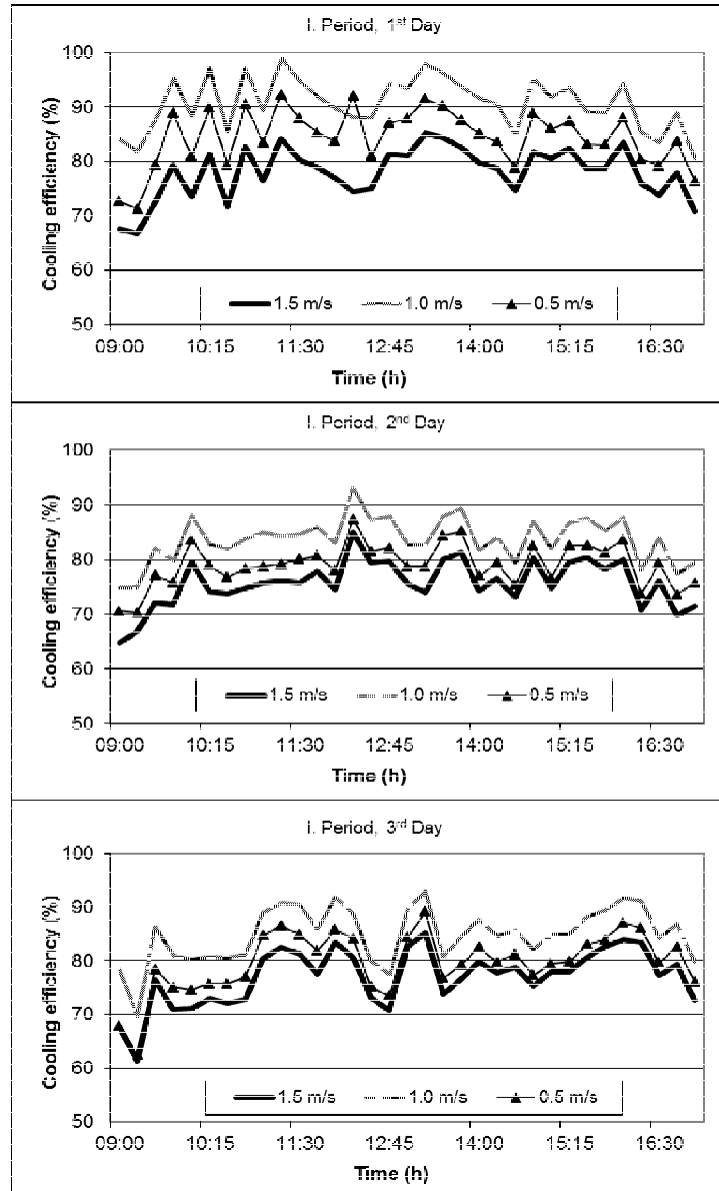
Figure 3. The cooling efficiency calculated at the chosen air velocities.

Table 1. The result of variance analysis for the effect of air velocity on temperature decrease and cooling efficiency.

I. Period	Source	Temperature decrease		Cooling efficiency	
		df	Mean square	df	Mean square
1st Day	Between groups	2	6.912[**]	2	1334.777[**]
	Within groups	96	1.042	96	24.852
2nd Day	Between groups	2	1.964	2	531.534[**]
	Within groups	96	0.646	96	17.074
3rd Day	Between groups	2	2.235	2	548.218[**]
	Within groups	96	1.843	96	29.422

**: $P < 0.01$.

Table 2. Duncan test results for the effect of air velocity on temperature decrease and cooling efficiency.

	Measurements	Air velocity (ms^{-1})		
		1.5	1.0	0.5
Temperature decrease	1st day	5.71±0.98[b]	6.63±1.05[a]	6.15±1.02[ab]
	2nd day	4.66±0.79	5.15±0.83	4.86±0.79
	3rd day	5.07±1.32	5.58±1.40	5.25±1.35
Cooling efficiency	1st day	77.97±4.75[c]	90.69±4.85[a]	84.44±5.33[b]
	2nd day	75.69±4.32[c]	83.68±4.12[a]	79.04±3.96[b]
	3rd day	76.97±5.39[c]	84.99±5.18[a]	79.74±5.69[b]

highest average temperature decrease at chosen air velocities was at 1.0 ms^{-1} air velocity, while the lowest value was at 1.5 ms^{-1} air velocity. The average value belonging to 0.5 ms^{-1} air velocity had common features with the values obtained at two other air velocities. At cooling efficiency, the highest value reached 1.0 ms^{-1} air velocity on three different days, while the values of 0.5 and 1.5 ms^{-1} air velocities followed this respectively.

When the literature was taken into consideration, it was expected that the obtained temperature decrease of the air passing through the pad and cooling efficiency values would be the highest at 0.5 ms^{-1} air speed, but this situation was not observed at the graphical and statistical evaluations. It was considered that this situation was incompatible with the literature data, caused by the flow character of the air passing through the pad at 0.5 ms^{-1} air velocity being different in comparison with the other two velocity values. In other words, it was thought that rather few layers of air in contact with the moist pad surface had a chance of getting moisture because the air passing through the pad pores at 0.5 ms^{-1} air velocity showed a laminar character. As a result of this, the obtained temperature decrease and the cooling efficiency values were lower. The tests were repeated at 0.75, 1.25 and 1.75 ms^{-1} velocities in order for the air to pass the pads at the same flow character in each of the experimental sets (I I. period). As regards the result of the trials made at these air velocities and the 4 Lmin^{-1}m^{-2} water flow rate in 3 repetitions, the obtained temperature decreases of the air passing the pad, depending on the air velocity, can be seen in Figure 4, while the cooling efficiency values can be seen in Figure 5.

As seen in Figures 4 and 5 at the chosen air velocities (0.75, 1.25 and 1.75 ms^{-1}), the obtained temperature decrease of the air passing the pad and the cooling effectivities showed similar changes. In other words, the obtained temperature decrease of the air passing the pad at 1.75 ms^{-1} air velocity and the cooling efficiency values were the lowest at 1.25 and 0.75 ms^{-1} velocities, repectively since their values were close to each other. Although it was expected that the obtained temperature decrease of the air passing the pad and the cooling

efficiency values would be higher at 0.75 ms^{-1} air velocity, this situation did not happen. The result of the variance analysis made for the effect of the air velocity at I I. period on temperature decrease and cooling efficiency can be seen in Table 3. As seen in Table 3 for each of the three days experiment, the relationship between the velocity of the air passing through the pad and the obtained temperature of the air passing the pad was found to be unimportant, while the relationship with the cooling efficiency was found to be important (P<0.01).

Duncan test was applied in order to determine the differences of temperature decrease or cooling efficiency averages for each of them at chosen air velocities belonging to the important found days and the results were presented in Table 4.

As seen in Table 4, the lowest cooling efficiency for all three days occurred at 1.75 ms^{-1} air velocity. This situation was in accordance with the study's literature. At 1.25 and 0.75 ms^{-1} air velocities, the calculated cooling effectivities were close to each other and their effects showed similarities. As the velocity of air passing through the pad decreased, the cooling efficiency increased. Taking this reality into consideration, the highest cooling efficiency was expected at 0.75 ms^{-1} air velocity. Yet, cooling effectivities at 0.75 and 1.25 ms^{-1} air velocities were too close to each other. It was considered that this situation was caused by these two chosen velocities that were between the most appropriate values of the pad used in the experimentations.

When similar studies were examined, it was seen that Timmos and Baughman (1984) determined the average cooling efficiency at 0.9 ms^{-1} air velocity as 80% in a study they conducted in a poultry house with pad evaporative cooling house in the South-east region of USA. Timmos et al. (1981) specified the cooling efficiency of the system as 82% at a study made in a poultry house in North Karoline, which had a pad evaporatif cooling system. Uğurlu and Kara (2000) observed some performance characteristics of a pad evaporative cooling system in a poultry house, which made commercial production, provided the temperature decrease of the air passing through the pads was
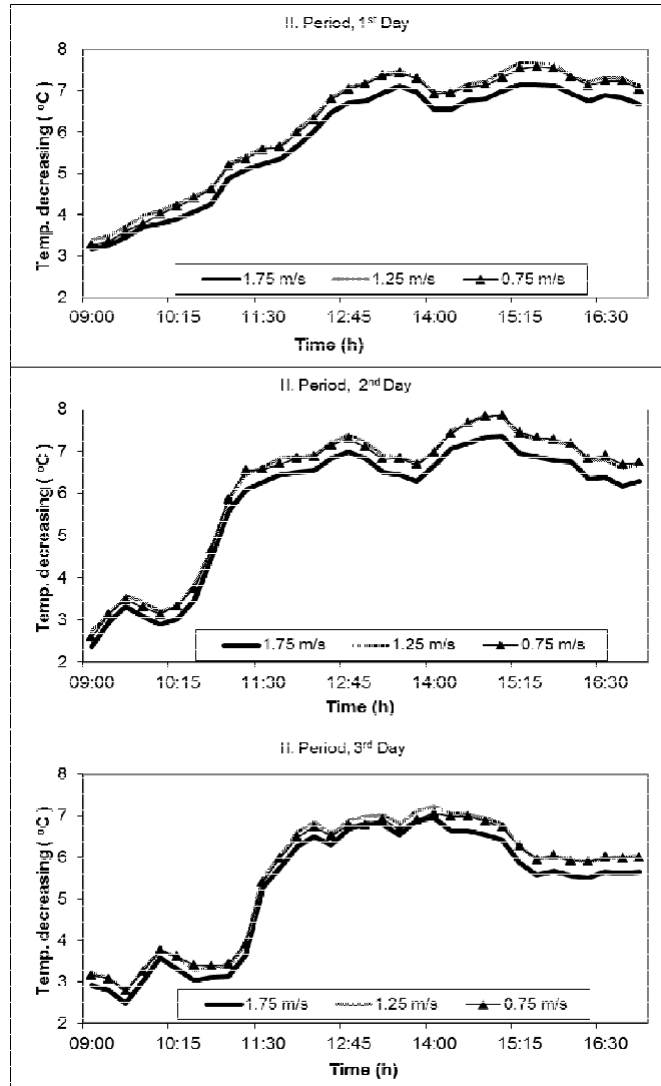
Figure 4. Obtained temperature decreases of the air passing through the pad.

between 4.2 and 16.2°C and the cooling efficiency of the system changed between 77 and 92%. Dağtekin et al. (2009: 2) observed some functional characteristics of a pad evaporative cooling system in a broiler poultry house in Çukurova region (Adana, S. Turkey). As a result of the evaluations calculated, the cooling efficiency was 69.2 to 72.0%, provided the temperature decrease of the air passing through the pads was 4.4 to 7.3°C.

It was seen that the temperature decrease and cooling efficiency values at both periods of the study were in accordance with the aforementioned studies.

Conclusion

At graphical evaluations, the obtained temperature decrease of the air passing through the pad and the

cooling effectivities and cooling efficiency values at the I. Period were lowest at 1.5 ms^{-1} air velocity, in that the values were close to each other at 1.0 and 0.5 ms^{-1} air velocities. When the literature was taken into consideration, it was expected that the obtained temperature decrease of the air passing through the pad and the cooling efficiency values would be the highest at 0.5 ms^{-1} air speed. This situation was not observed at graphical and statistical evaluations. It was considered that this situation, which was incompatible with the literature data, was caused by the flow character of the air passing through the pad at 0.5 ms^{-1} air velocity that was different in comparison with the other two velocity values. In other words, the air passing through the pad pores at 0.5 ms^{-1} air velocity showed a laminar character, where few layers of air in contact with the moist pad surface had a chance of getting moisture. As a result
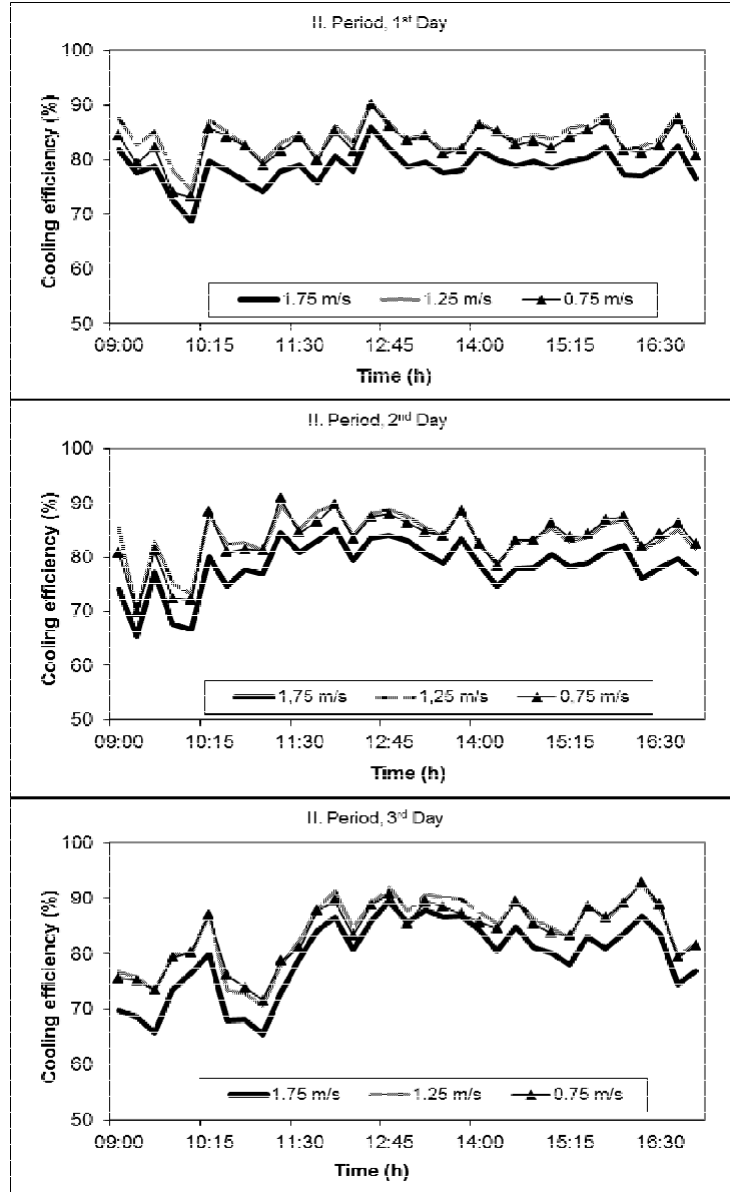
Figure 5. The calculated cooling effectivities at chosen air velocities.

Table 3. The result of variance analysis for the effect of air velocity on temperature decrease and cooling efficiency.

I I. Period	Source	Temperature decrease		Cooling efficiency	
		df	Mean square	df	Mean square
1st Day	Between groups	2	1.403	2	261.655[**]
	Within groups	96	1.998	96	10.574
2nd Day	Between groups	2	1.432	2	295.500[**]
	Within groups	96	2.567	96	22.143
3rd Day	Between groups	2	0.919	2	230.559[**]
	Within groups	96	2.280	96	40.172

**: P<0.01.

Table 4. The results of the DUNCAN test for the effect of the air velocity on temperature decrease and cooling efficiency.

	Measurements	Air velocity (ms^{-1})		
		1.75	1.25	0.75
Temperature decrease	1st Day	5.81±1.36	6.20±1.43	6.12±1.44
	2nd Day	5.79±1.56	6.15±1.61	6.14±1.63
	3rd Day	5.24±1.52	5.54±1.52	5.51±1.49
Cooling dfficiency	1st Day	78.60±3.12[b]	83.92±3.19[a]	82.85±3.43[a]
	2nd Day	78.43±4.80[b]	83.77±4.50[a]	83.44±4.81[a]
	3rd Day	79.37±6.98[b]	84.18±6.24[a]	83.67±5.74[a]

of this, the obtained temperature decrease and the cooling efficiency values became lower.

In the statistical analysis, the effect of air velocity on temperature decrease was unimportant on the 2nd and 3rd day, although it was important on the 1st day (P<0.01). According to DUNCAN test results, on the 1st day, the highest value in the light of average temperature decrease occurred at 1.0 ms^{-1} air velocity, while the lowest value occurred at 1.5 ms^{-1} air velocity. At the same period, on the 1st, 2nd and 3rd day, the effect of air velocity on cooling efficiency was found to be important (P<0.01). In the DUNCAN test made to determine the differences of averages, the effect of air velocity on cooling efficiency at 1.0 and 0.5 ms^{-1} air velocities on the 1st and 3rd day was the highest. On the 2nd day, the calculated cooling efficiency values were different from each other and the highest average value was at 1.0 ms^{-1} air velocity, while the lowest value was at 0.5 ms^{-1} air velocity.

The tests were repeated with 0.75, 1.25 and 1.75 ms^{-1} values chosen as velocities of air passing through the pad in order to provide the air to pass the pads at the same flow character (I I. Period). At the evaluations, the lowest values for the obtained temperature decrease of the air passing through the pad and the cooling efficiency was observed at 1.75 ms^{-1} air velocity. This situation was in accordance with the literature data. The values for the obtained temperature decrease and the cooling efficiency in the other two velocity stages (0.75 and 1.25 ms^{-1}) showed changes that were rather close to each other. For the statistical analyses, the relationship between air velocity and temperature decrease was found to be unimportant, yet the relationship between air velocity and cooling efficiency was found to be important (P<0.01). According to DUNCAN test results, on the 1st, 2nd and 3rd days, the average calculated cooling efficiency values were the highest at 0.7 and 1.25 ms^{-1} air velocities and were the lowest at 1.75 ms^{-1} air velocity. On the basis of the data obtained, it was considered that it was not possible to make a mathematical connection among velocity of the air passing through the pad, the obtained decrease at the temperature of the air passing through the pad and the cooling efficiency; but it can be said that the most appropriate air velocity for the pad used in the experiments should be higher than 0.5 ms^{-1} and lower than 1.5 ms^{-1}.

REFERENCES

Albright (1989). Environment Control for Animal and Plants. St Joseph, Michigan.

Al- Ismaili AM, Weatherhead EK, Badr O (2010). Validation of Halasz's Non-dimensional Model Using Cross-flow Evaporative Coolers Used in Greenhouses. Biosyst. Eng., 107: 86-96.

ASAE (1983). Ventilation of Agricultural Structures (Edited: Mylo A. Hellickson and John N. Walker). ASAE Monograph, Number: 6, St. Joseph, Michigan.

ASAE (1994). ASAE EP270. 5: Design of Ventilation Systems For Poultry and Livestock Shelters. 4 th Edition. Standards Engineering Practies Data, p. 501-518.

Atikol U, Hac i evki H (2001). Evaporative Cooling Feasibility Study for Nicosia Region. V. National Installation Engineering Congress and Exhibition, Notifications, Đzmir, pp. 421-425

Bilge D, Bilge M (1999). Indirect/ Direct Evaporative Cooling Systems Combination. IV. National Installation Engineering Congress and Exhibition, Notifications, Đzmir, pp.197-204.

Buffington DE, Skinner TC, Collette B, Borer D (1978). Evaporative Cooling for Hot, Humid Climates. Paper Presented at the Scutheast-Scuthwest Regional Meeting of ASAE at Hoston, TX. February 5-8.

Cruz VF, Perissinotto M, LucasEM, De Moura DJ (2006). Efficiency evaluation of pad cooling systems design with different pad materials. XV I. C IGR World Congress, AgEng Bonn 2006, Book of Abstracts, p. 403, 404.

Dağtekin M, Gürdil GAK, Yıldız Y (1998). Determination of Suitable Pad Material for Evaporative Cooling of Broiler Houses in Turkey. AgEng Oslo-98, Đnternational Conference on Agricoltural Engineering, p.271-272, Augost 24-27, Oslo.

Dağtekin M, Karaca C, Yıldız Y (2009). Long Axis Heat Distribution in a Tunnel-Ventilated Broiler House Equipped With an Evaporative Pad Cooling System. Anim. Prod. Sci., 49: 1125-1131.

Dağtekin M, Karaca C, Yıldız Y (2009). Performance Characteristics of a Pad Evaporative Cooling System in a Broiler House in a Mediterranean Climate. Biosyst. Eng., 103: 100-104.

Kocatürk Ü, Yıldız Y (2006). Some Performance Characteristics Of Evaporative Cooling Pad At Three Air Velocities Đn Cukurova Region. J. Agric. Fac., 2006, 21(4): 101-108.

Koca RW, Hughes WC, Christianson LL (1991). Evaporative Cooling Pads: Test Procedure and Evaluation. Trans. ASAE, 7(4): 485-490.

Simmons J D; Lott B D (1996). Evaporative Cooling Performance Resulting from Changes in Water Temperature. Appl. Eng. Agric., 12(4): 497-500.

Timmons MB, Baugman GR (1984). A plenum Concept Applied to Evaporative Pad Cooling for Broiler Housing. Trans. ASAE, 27(6): 1877-1881.

Timmos MB, Baughman GR, Parkhurst CR (1981). Use of Evaporative Cooling to Reduce Poultry Heat Stress. ASAE Paper No. 81-4558, ASAE, St. Joseph, M I 49085.

Uğurlu N, Kara M (2000). Performance Data of Cooling with Wet Pad Systems and its Effect on Decreasing Inner Temperature of Cage System Poultry House. Turkish J. Agric. For., 24: 79-86. TÜBĐTAK.

Yıldız Y, Dağtekin M (2006). Some Functional Properties of a Padded Evaporative Cooling System of a Broiler Poultry in Çukurova Region. J. Agric. Fac., 21(3): p.59-764.

Yıldız Y, Karaca C, Dağtekin M (2010). Environment Contro at Animal Housings (Ventilation, Heating, Cooling and Illumination). ISBN: 978-975-8377-75-6. Hasad Publishing Ltd. Co., P.B. 35, Ümraniye-3470-Đstanbul.