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Design and Construction of a Farm Scale Evaporative Cooling System



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Abstract

Indoor temperatures especially in hot climate regions, can reach very high values in closed agricultural areas such as in greenhouses and in poultry houses in summer. This causes heat stress for animals and plants. Higher indoor temperatures can also be fatal. Thus, cooling is indispensable in those closed production structures. A smart and practical solution can be fan-pad evaporative cooling system. In this study, a farm scale fan-pad evaporative cooling system has been designed and constructed and evaporative cooling efficiency has been investigated.

Introduction

The demand for production in closed and environmentally controlled areas are increasing rapidly as the population of the world increases. Greenhouses and poultry houses are good examples of this production type. The two main factors that affect the productivity of poultry and plants are genotype and environment. However, the high quality genotypes are not beneficial unless these are raised in suitable environmental conditions. One of the most important environmental factors which directly effect the productivity is the ambient temperature. For example, the heat stress problem begins when the indoor temperatures exceed the thermo-neutral zone 15 to 25 °C for broilers. Heat stress causes some complex physiological changes in the organism of animals that negatively effect the productivity of poultry (Howlider and Rose, 1989; Kutlu et al., 1996; Leenstra and Cahaner, 1992). High air temperatures in animal houses reduces feed consumption, decreases weight gain, milk and egg yields (Al-Amri, 2000). Evaporative cooling (fan-pad and misting) systems are recommended by researches for high indoor temperature problems in poultry houses (Timmons and Gates, 1988; Wilson et al., 1983).

The use of evaporative cooling can be rather efficient in many cases. These methods of adiabatic cooling based on spraying of water into the incoming air through special nozzles (misting systems) or by fan pad system are being used more in practical reality. This system consists of a number of large high capacity exhaust fans located at one end or on one wall of the building and series of wet fibrous pads in a continuous section located at the opposite end or on opposite wall of the building. The experiments with different pad materials showed the possibilities of lower investment costs (Kic and Sleger, 1992; Bottcher et al., 1993; Chiumenti et al., 1996).

The objective of this study was to determine the evaporative cooling efficiency of a farm scale fan-pad cooling systems which is designed and manufactured at Samsun Ondokuz May at University Turkey. The tests were conducted at different air

Fig. 1 Farm scale fan pad cooling unit

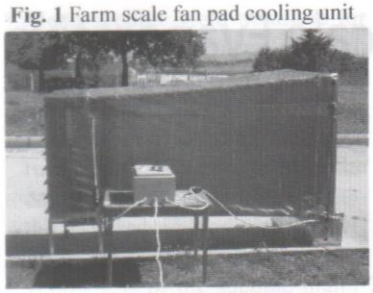


Fig. 2 Cooling fan

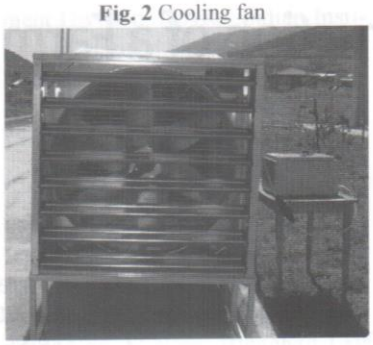
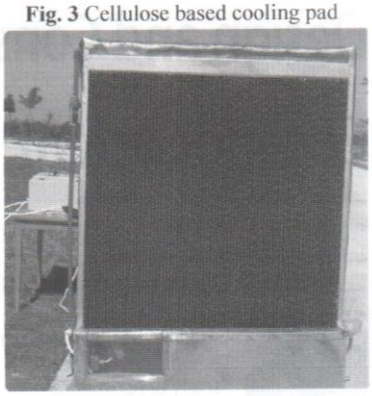


Fig. 3 Cellulose based cooling pad



velocities passing through the pad material. The aim of the study was to provide practical solutions for the farmers suffering from the high indoor temperatures and engineers trying to find simple technical solution for the reduction of heat stress.

Materials and Methods

A farm scale fan pad evaporative cooling unit was designed and constructed (**Fig. 1**) in the workshop of Agricultural Machines and Technologies Engineering Department of Ondokuz Mayis University in Samsun. The tunnel-type cooling unit is 2 m long and constructed from angle iron. The outer surface is environed by a waterproof canvas. A cooling fan having 950 mm diameter (EM-30 Munters) (**Fig. 2**) was mounted at one end and a cellulose based pad (CEL dek 7060-15) to the other end of the unit. The fan had air -flow capacity of 15,000 m3/h and was driven by a 0.5 kW powered electric motor. The fan speed can be changed continuously by an electronic speed changer connected to the fan motor. The dimensions of the decay resistant pad was 1 × 1.2 × 0.1 m with 45° - 15° chamfer angles (**Fig. 3**).

Water flow rate through the pad was adjusted to 4 L/min as recommended by the manufacturer. Three different air stream velocities 1, 1.5 and 2 m/s were tested for the evaporative cooling efficiencies. An electronic data recorder device (Extech 42270) was used for all the air temperature and relative humidity readings for inside and outside conditions (**Fig. 4**). Airflow velocity through the pad material was measured by a special device (Testo AG 309). Evaporative cooling efficiency of the system is calculated by the equation, below (Timmons and Baughman, 1984; Koca et al., 1991).

$\eta = [t_e - t_i] / [t_e - t_w]$ 100

Where:

η = Evaporative cooling efficiency, %,

t_e = Dry bulb temperature of air at pad entrance, °C,

t_i = Dry bulb temperature of air at pad exit, °C,

t_w = Wet bulb temperature of air at pad entrance, °C.

Cooling unit was tested during the hottest period from May to September in that region. The measurements were taken at the hottest period of the day (12:00 am - 15:00 pm) at each selected air velocity.

Results and Discussion

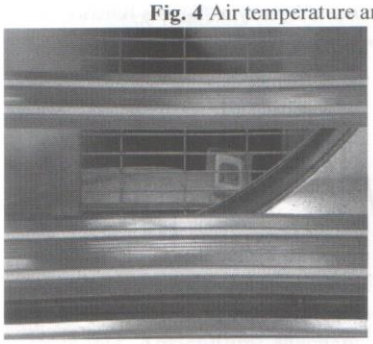
This study analysed the evaporative cooling efficiency of a farm scale fan-pad cooling system at

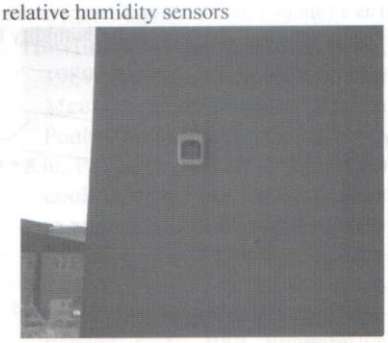
Table 1 Measured and calculated values at 1 m/s air flow velocity

Readings	t_e (°C)	Rh_e (°C)	t_w (°C)	t_i (°C)	Δt_{ei} (°C)	η (%)
May I	28.06	56.88	21.63	23.96	4.10	63.76
May II	29.97	35.66	19.20	22.59	7.38	68.52
June I	30.68	40.24	20.68	26.92	3.76	37.60
June II	29.14	51.72	21.63	24.28	4.86	64.71
June III	24.88	52.16	18.16	21.92	2.96	44.05
June IV	30.12	52.92	22.67	28.91	1.21	16.24
June V	32.79	43.60	23.03	25.94	6.85	70.18
June VI	32.05	47.79	23.29	26.67	5.38	61.42
July I	28.74	55.34	21.94	24.86	3.88	57.06
July II	31.61	43.73	22.12	25.51	6.10	64.28
July III	29.74	36.82	19.26	22.62	7.12	67.94
August I	28.29	57.03	21.85	23.97	4.32	67.08
August II	33.93	44.23	24.07	28.38	5.55	56.29
August III	30.50	39.53	20.39	25.68	4.82	47.68
August IV	29.75	45.53	20.99	24.03	5.72	65.3
August V	28.69	43.86	19.82	24.51	4.18	47.13
Sep I	28.86	55.45	22.06	22.63	6.23	91.62
Sep II	28.41	53.23	21.29	25.44	2.97	41.71
Average	29.79	47.54	21.34	24.93	4.86	57.36

Where: t_e = dry bulb temperature of outside air (°C), Rh_e = relative humidity of outside air (%), t_w = wet bulb temperature of outside air (°C), t_i = dry bulb temperature of air at pad exit (°C), Δt_{ei} = de-crease of air temperature (°C); η = evaporative cooling efficiency (%)

Fig. 4 Air temperature and relative humidity sensors





three different airflow rates. Air temperature, relative humidity and velocity of outside air and temperature of inside air were measured and then temperature decreases, wet bulb temperature of outside air and evaporative cooling efficiency were calculated. The calculated values at 1 m/s airflow velocity through the pad are given in **Table 1**. As seen from the table the average evaporative cooling efficiency was 57.36%, the highest evaporative cooling efficiency was achieved as 91.62% in September, where the lowest was in June period as 16.24%. The decrease in air temperature values provided by this system ranged from 1.21 to 7.38 °C at 1 m/s air flow velocity.

The calculated values at 1.5 m/s airflow rate through the pad are given in **Table 2**. The average evaporative cooling efficiency was 50.32%, the highest evaporative cooling efficiency was achieved as 77.80% in June, where the lowest was again in June period as 22.89%. The cooling unit provided a temperature decrease ranging from 1.83 to 8.69 °C.

The calculated values at 2 m/s airflow velocity through the pad are given in **Table 3**. As seen from the table the average evaporative cooling efficiency was 54.29%, the highest evaporative cooling efficiency was achieved as 82.09% in August, where the lowest was in July period as 25.69%. The provided temperature decrease ranged from 1.77 to 7.29 °C at 2 m/s airflow velocity.

Evaporative cooling efficiency of the system ranged approximately among 16-91% during the total hot May-September period. The unit provided a temperature decrease in the inside air up to 9 °C against hot outside air that is important enough for the animals and plants. These results were in agreement previous literature studies on evaporative cooling pad systems (Dzivama et al., 1999; Helmy et al., 2013; Dayıoglu and Silleli, 2015; Han et al., 2018). All calculated results of evaporative cooling efficiency η (%) are summarized as a function of outside air temperature te (°C) in the **Fig. 5**. There is obvious tendency of the increased evaporative cooling efficiency with growing temperature of external air.

The system best performed in September period because of the lower relative humidity values at outside air. Evaporative cooling efficiency η (%) as a function relative humidity Rhe (%) of outside air presented in the **Fig. 6** is slowly decreasing with growing relative humidity. Previous studies always stressed that the fan-pad evaporative cooling systems performs very well till the outside air relative humidity values exceed 75%. If the region has more than 75% of relative humidity at outside air, then it's recommended to cool down the ambient without wetting the pads just exchanging the inside air rapidly with cooling fans. Another solution can be applying lesser amounts of water flow up to 2 L/min, through the pad.

Table 2 Measured and calculated values at 1.5 m/s air flow velocity

Readings	t_e (°C)	Rh_e (°C)	t_w (°C)	t_i (°C)	Δt_{ei} (°C)	η (%)
May I	23.59	53.97	17.36	21.76	1.83	29.37
May II	24.31	51.44	17.57	20.2	4.11	60.98
June I	24.99	55.73	18.82	22.69	2.3	37.28
June II	29.01	56.42	22.36	26.78	2.23	33.53
June III	30.22	34.09	19.05	21.53	8.69	77.8
June IV	32.23	37.77	21.35	29.74	2.49	22.89
June V	31.31	45.37	22.21	24.46	6.85	75.27
June VI	28.88	48.6	20.85	26.23	2.65	33.00
July I	31.53	43.72	22.05	25.71	5.82	61.39
July II	27.96	54.05	21.06	25.94	2.02	29.28
July III	30.79	48.57	22.41	26.02	4.77	56.92
August I	32.57	38.57	21.78	27.01	5.56	51.53
August II	28.06	46.52	19.81	23.79	4.27	51.76
August III	28.12	53.65	21.12	24.35	3.77	53.86
August IV	25.51	52.35	18.72	22.25	3.26	48.01
August V	29.16	37.02	18.86	23.95	5.21	50.58
Sep I	30.91	45.13	21.84	24.74	6.17	68.03
Sep II	27.28	49.06	19.63	22.36	4.92	64.31
Average	28.69	47.34	20.38	24.42	4.273	50.32

Where: t_e = dry bulb temperature of outside air (°C), Rh_e = relative humidity of outside air (%), t_w = wet bulb temperature of outside air (°C), t_i = dry bulb temperature of air at pad exit (°C), Δt_{ei} = de-crease of air temperature (°C); η = evaporative cooling efficiency (%)

Conclusions

The present study analysed the evaporative cooling efficiency of a farm scale fan-pad cooling system

Fig. 5 Evaporative cooling efficiency η (%) as a function of outside air temperature t_e (°C)

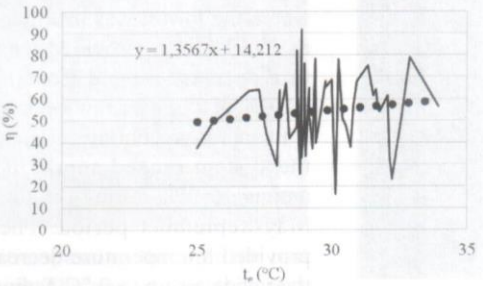
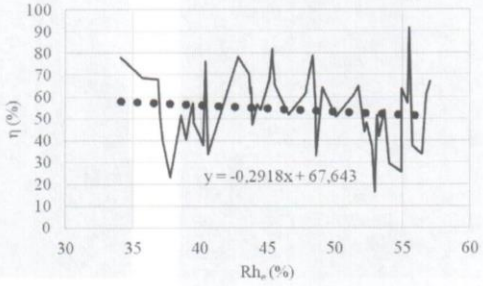


Fig. 6 Evaporative cooling efficiency η (%) as a function relative humidity Rh_e (%) of outside



at varying airflow rates. A mobile and relatively small fan-pad cooling unit was designed and manufactured for farm scale. The system was tested from May to September during the hottest period of the region and all the measurements were taken at the hottest period of the day (12:00 am - 15:00 pm) at each selected air speed. Evaporative cooling efficiency and provided temperature decrease at inside air were calculated from inside and outside air conditions. Best results were taken at 1 m/s of air flow velocity. This is in line with the results of a research done by (Dagtekin et al., 2011). The cooling system gave promising results by achieving 91% of evaporative cooling efficiency. The highest evaporative cooling efficiencies were achieved in the hottest period of the day that is in the 12:30 pm - 14:40 pm time interval. After all, it is recommended to operate this system during periods when the air temperature is high in summer and the air-humidity is low in order to get more efficiency from fan-pad cooling system. Also higher water flow rates than 4 L/min can be applied in dry areas. In conclusion, it is observed that the fan-pad cooling system is effective in removing excessive heat from the interior. Additional researches of this kind are important not only for enhancing the quality and quantity of scientific data, but also as a means of focusing farmers and public attention on this matter.

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Table 3 Measured and calculated values at 2 m/s air flow velocity

Readings	t_e (°C)	Rh_e (°C)	t_w (°C)	t_i (°C)	Δt_{ei} (°C)	η (%)
May I	29.41	37.22	19.09	25.34	4.07	39.44
May II	28.78	38.95	18.95	24.82	3.96	40.28
June I	30.38	50.07	22.36	26.31	4.07	50.75
June II	28.97	40.36	19.38	21.68	7.29	76.02
June III	27.37	39.37	17.95	22.18	5.19	55.10
June IV	32.13	40.57	21.87	28.68	3.45	33.63
June V	29.14	53.09	21.88	25.41	3.73	51.38
June VI	28.82	54.91	21.93	27.05	1.77	25.69
July I	29.27	52.71	21.92	26.54	2.73	37.14
July II	32.87	48.35	24.08	25.93	6.94	78.95
July III	31.74	44.47	22.37	26.7	5.04	53.79
August I	28.96	39.20	19.14	24.03	4.93	50.20
August II	28.72	56.71	22.16	24.74	3.98	60.67
August III	31.68	39.47	21.29	25.76	5.92	56.98
August IV	28.69	45.30	20.09	21.63	7.06	82.09
August V	27.59	39.86	18.21	23.54	4.05	43.18
Sep I	29.38	42.81	20.17	22.17	7.21	78.28
Sep II	26.91	54.92	20.32	22.71	4.20	63.73
Average	29.49	45.46	20.73	24.73	4.755	54.29

Where: t_e = dry bulb temperature of outside air (°C), Rh_e = relative humidity of outside air (%), t_w = wet bulb temperature of outside air (°C), t_i = dry bulb temperature of air at pad exit (°C), Δt_{ei} = de-crease of air temperature (°C); η = evaporative cooling efficiency (%)

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ERRATA

There was a typographical error on the book spine of AMA Vol.51 No.1
Correct as below:

Vol.50, NO.4, AUTUMN 2019 → Vol.51, NO.1, WINTER 2020