"ENVIRONMENTAL CONTROL INSIDE AN AGRICULTURAL STRUCTURE"

PERFORMANCE CRITERIA OF TWO EVAPORATIVE COOLING PADS MADE OF AGRICULTURAL RESIDUES TO REDUCE HEAT STRESS INSIDE AGRICULTURAL STRUCTURES

BY

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ABSTRACT

Heat stress inside agricultural structures is one of the problems facing producers especially in hot climate areas such as the situation of summer in Egypt. The need to an environmental control system that can be constructed as a hand made system and easy to be operated to cope with that problem became a virtual requirement. Therefore, the present study was devoted to construct an evaporative cooling system belonging to pad and fan system using agricultural residues as pad materials. Investigating its performance criteria under different technical specifications such as pad material, pad thickness and pad face air velocity were carried out as well.

Rice straw and palm leaf fibers (Kerina) were used as pad materials. Four pad thickness 3, 6, 10, 15 cm and three pad face air velocity 0.3, 0.5, 1.05 m/s were used in the investigation of the performance criteria of the system. It was revealed that both of pad material could be used in the suggested evaporative cooling system. Rice straw pad material proved more efficiency in temperature reduction, i.e. reducing heat stress inside the agricultural structure. On the other hand the disadvantage of increasing relative humidity was obviously noticed. This problem was not a predicable when using palm leaf fibers (Kerina) as had the pad material. The application of temperature- humidity index (THI) concept revealed that the suggested system capability in reducing THI compared with the THI of the ambient air outside the structure in all treatments. However, the suggested evaporative cooling system may not be the efficient one for some biological systems such as broiler housing systems from the view point of THI. The other performance criteria of the suggested system such as saturation efficiency (SE), Unit of evaporative cooler performance (Unit ECP) and the ratio of temperature reduction to airflow rate were introduced as well.

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LIST OF SYMBOLS AND ABBREVIATIONS

BA	is the bird age (day)
BMG	is the bird mass (g)
BMKG	is the bird mass (kg)
Cp_{air}	is the specific heat of air (1.005 kJ/kg. °C)
d	Pad thickness (cm)
EET	is the effective environmental temperature
ECP	is the evaporative cooler performance (kJ/h)
h_{vap}	is the specific heat of vaporization (2430kJ/kg@30°C)

EET	is the effective environmental temperature
LCT	is the lower critical temperature
n_b	is the number of birds
m_b	is the airflow rate (kg_a/h)
PLF	is the palm leaf fibers pad material
Q	is the ventilation rate (m^3/s)
Q_L	is the net latent heat production per unit mass of broilers (kJ/h.kg)
Q_s	is the net sensible heat production per unit mass of broilers (kJ/h.kg)
RS	is the rice straw pad material
SE	is the saturation efficiency (decimal)
t_C	is the dry bulb temperature of the internal cooled air without loading under simulated broiler housing conditions (°C)
T	is the dry bulb temperature (°C)
T_C	is the dry bulb temperature of air exiting the cooling system (°C)
T_{db}	is the dry bulb temperature of air entering the cooling system (°C)
$(T_{db})_{BS}$	is the dry bulb temperature of the internal air under simulated broiler housing conditions (°C)
T_{dp}	is the dew point temperature (°C)
THI	is the temperature humidity index (°C)
$(THI)_{B,}$ $(THIo)_{B}$	is the temperature humidity index for broilers (°C), out side temperature humidity index for broiler (°C)
$(THI)_{BS}$, $(THI)_{BS}$	is the temperature humidity index under simulated broiler housing conditions (°C), out

	side temperature humidity index under simulated.(°C)
$(THI)_{dc}$, $(THIo)_{dc}$	is the temperature humidity index for dairy cows (°C), out side temperature humidity index for dairy cows(°C)
$(THI)_{L,}$ $(THIo)_{L}$	is the temperature humidity index for Layers (°C) , out side temperature humidity index for layers (°C)
$(THI)_{dcf}$	is the temperature humidity index for dairy cows (°F)
TNZ	is the thermo-neutral zone
T_{wb}	is the wet bulb temperature of the outside air(°C)
Unit ECP	Is the unit evaporative cooler performance (kJ/°C)
V_s	is the specific volume of air (m^3/kg)
W_{BS}	is the humidity ratio of the internal air under simulated broiler conditions (kg/kg)
W_C	is the humidity ratio of the internal cooled air without loading (kg/kg)
W_{ev}	is the water evaporated per hour (kg/h)
W_C	is the humidity ratio of the cooled air ($kg/kg_{dry\ ait}$)
W_O	is the humidity ratio of outside air ($kg/kg_{dry\ ait}$)
W_S	is the humidity ratio of the outside air entering the cooling system if saturated to 100% relative humidity in a constant bulb process.
ΔT	is the temperature difference (°C)
$(\Delta T/Q)$	is the ratio of temperature reduction to air floe rate (°C.s/m^3)

1. INTRODUCTION

Environmental control inside the agricultural structure is considered one of the most important factors affecting growth and productivity of organism. Agricultural structures such as livestock and poultry housing and greenhouses …etc, should have the capability to offer the optimum environment for the confined biological system. One of the most important environmental factors that affect a confined biological system is temperature. As it is well known that there is a thermo-neutral (comfort) zone for each biological system, so that changing of temperature beyond the maximum or minimum limits of that zone depresses productivity. However, an environmental control system may be needed to provide and maintain the required comfort zone within a structure. From the viewpoint of thermal environmental factors, an environmental control system may be used as an auxiliary system to control the temperature within the comfort zone via heating or cooling process. Egypt has a climate tend to be warm or hot throughout the most of the year months. Therefore, it is not comfortable for lactating dairy cows and poultry production resulting in great losses of milk production and live weight of broilers caused by the heat stress. This is the main problem facing the development in this field. Such problem could be noticed clearly in summer months and during hot weaves throughout the year. The side effects of heat stress are associated with: increasing mortality in poultry herd, reduced meat or egg production, reduced milk lactating for dairy cows especially foreign herd, reduced yield crops inside greenhouses, and deterioration of grain storages, (Buffington et al. 1983; Bottcher et al., 1990; HeeChul et al., 1998; Deling et al., 2000 and Aggarwal and Singh, 2004). As well, it obviously appears in broiler houses with intensive production specifically before marketing causing severs losses. Temperatures inside the greenhouse are frequently 13°C higher than those outside in spite of open ventilators. Detrimental effects of high temperatures are typified by loss of stem strength, delay flowering, decrease the vitality and die of insemination seeds, loss of fruit set, and increasing the possibilities infection by the pathogenic organisms.

Evaporative cooling is one of the common ways that can be used to reduce heat stress inside the agricultural structures. The evaporative cooling system is based upon the process of heat absorption during the evaporation of water. Evaporative cooling systems, also known as fan and pad cooling have become popular during the past three decades as a remedy for this problem (heat stress). The major components of such system are: pad media (material), water supply system and fan as well as water recirculating system. The commercial pads are usually complicated to manufacture and they are costly and not readily available. More recently, new cellulose paper designs have been developed to make evaporative cooling more efficient. Therefore, there is a paramount need to evaluate locally available materials (field residues) use as a pad media that can be constructed as a hand-made system. Agricultural residues such as rice straw and palm leaf fibers may considered one of the local materials suitable for this purpose. Exploitation of these agricultural residues not only use as an evaporative cooling pad material, but also play an important role in environment conservation.

So the main objective of the present study was to construct an evaporative system belonging to pad and fan system using rice straw and palm leaf fibers as pad materials and investigate their performance criteria. The specific objectives can be summarized as follows:

1- Investigate the feasibility of using rice straw and palm leaf fibers (Kerina) as pad materials of an evaporative cooling system and determination of their performance criteria.
2- Study the effect of pad material, pad thickness and pad face air velocity on the performance of the evaporative cooling system.
3- Investigate the effect of the suggested evaporative cooling system on reducing heat stress in terms of psychometric properties of the cold air comes into agricultural structures (dairy cows, laying hens and broilers housing).
4- Predict the performance of the system in reducing heat stress under simulated broiler housing conditions.

2. REVIEW OF LITERATURE

In the following sections, the term of the biological system will be used to refer to any agricultural species that could be grown inside agricultural structures. The biological system interacts with both the ambient environment as an environmental system and the structure as a thermodynamic one resulting in a combined and complex system.

The interior environment inside the agricultural structure may be needed to be controlled. This can be done by an environmental control system (cooling system, heating system, ventilation system…etc) to control the various factors involved in biological system livability and performance. Severe thermal environment, especially heat stress, which the biological system expressed to, is the corner stone of the present study. Therefore, investigating and searching for an evaporative cooling system as an environmental control is the main scope the present work.

2.1 Thermal environment affecting a biological system.

EFSA Journal (2004) reported that there are numerous literatures defining various thermal zones, thermoregulation and adaptation processes for animals. Because of the intensive and controversial discussions on these terms, definitions of the most frequently used terms are given to facilitate the understanding of the complex animal/environment relationship. As terrestrial farm animals are homoeothermic, they are, within certain limits, able to maintain a relatively constant deep body temperature, different from the environmental temperature. A relatively constant deep body temperature means that heat production and heat loss are equal. An increased difference between deep body temperature and environmental temperature leads to higher heat losses to be compensated by a higher heat production. Body temperature will increase when heat loss is not sufficient and resulting heat stress. Heat may be dissipated through conduction, convection, radiation and evaporation. Figure (2-1) presents a basic scheme of body core temperature control (Yousef, 1985). The temperature is kept constant in the zone of homoeothermic, but for temperatures lower or decreases. When the loss of heat is higher than the heat production, it causes hypothermia and evenly death. On the opposite, when the loss of heat is lower than heat production, e.g. by too high ambient temperature and humidity, it causes hyperthermia heat stress. The animal will die when body temperature continues to stay too low or too high.

EFSA Journal cited from several literatures the basic knowledge describing environmental temperature limits affecting an animal as a biological system. This can be indicated in the fallowing sections:

2.1.1 Micro – climate.

Hilliger (1990) reported that the term micro-climate is used in a specific way to mean the sum of all physical, chemical and biological factors of the air inside buildings or vehicles. It is influenced by ventilation and insulation

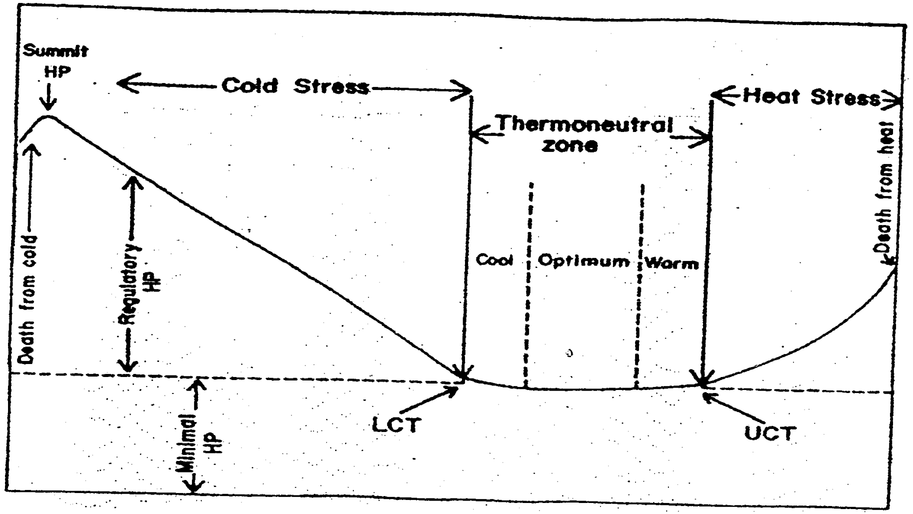


Figure (2-1): Scheme of the thermo-neutral zone of animals (Yousef, 1985). LCT: Lower critical temperature, UCT: Upper critical temperature.

of the surrounding walls, floor and roof. The most important components of the micro-climate are temperature and relative humidity of the air, air velocity and air quality.

2.1.2 Effective environmental temperature (EET).

Curits (1983) expressed the EET theoretically as the total effect of a particular environment on an animal's heat balance. It is the temperature experienced by an animal, being a combined effect of dry air temperature, air humidity measured as wet bulb or expressed as relative humidity, air velocity, radiative and may be conductive heat loss. For example, the effect of a higher ambient temperature or a higher relative humidity of the ambient air on the animal can be compensated a higher air velocity that maintains the same or equivalent effective temperature.

2.1.3 Lower critical temperature (LCT) and upper critical temperature (UCT).

Webster (1981) and McArthur (1987) stated that there is no absolute definition of UCT. However, they described the UCT as the point above which an animal must engage physiological mechanisms to stop the rise body temperatures above normal level.

Ewing et al. (1999) reported that LCT is the point in effective ambient temperature below which an animal must increase its rate of metabolic heat production to maintain homeothermy. Processes related to conservation of heat, including vasoconstriction in the periphery, piloerection and behavioural adjustments to reduce heat loss from body surfaces, are at their maximum at this temperature and below. They added that these processes are related to cooling by evaporation (through increased perspiration and respiration) and behavioural activities (such as wetting the skin and vasodilatation in the periphery) to enhance heat loss from body surfaces through convection, radiation and conduction.

2.1.4 Thermo-neutral zone (TNZ).

Richards (1973) and Charles (1994) reported that the thermo-neutral zone is limited by the lower critical temperature (LCT) and the upper critical temperature (UCT).Within that zone, the regulation of body temperature is physical or behavioural by changing exposed body surface, tissue insulation (sensible heat loss) and latent (evaporative) heat loss without panting. Below LCT and above UCT, there are energy costs thermoregulations.

Ewing et al. (1999) stated that within the thermo-neutral zone, metabolic heat production and energy expenditure are minimal, most productive processes are at their most efficient level and an animal is thermally comfortable without the need to change heat production.

2.1.5 Homoeothermic zone and survival zone.

Homoeothermic zone is the range of EET where an animal is able to keep deep body temperature by all available means at the normal level, which includes normal variability depending on species, age, physiological state, etc. Survival zones that range of EET where an animal is able to survive despite being hypo-or hyperthermic (EFSA Journal, 2004).

2.1.6 Thermoregulation.

Actions undertaken by an animal to meet its thermal needs, i.e. keeping body temperature constant (hypo-and hyperthermia also includes thermoregulation), forced by the physical conditions in the thermal environment (i.e. the EET), and influenced by the emotional perception of the surroundings. Consequently, animal temperature regulation starts with change of sensible and latent heat loss (e.g. through the skin) and –if necessary- due to higher environmental temperature more of the total heat must be lost as latent (Mount, 1974).

2.2 Heat stress in agricultural structures.

Problems of hot weather environment for a biological housing system have not received attention as the cold weather one. Hot weather adversely affects the performance of the biological system production. Heat stress may lead to sever losses in production, reproduction, feed conversion, health and welfare of a biological system. Otherwise, penalties should be paid to overcome this problem.

2.2.1 Heat stress in livestock systems.

Mitra et al. (1972) reported that hormonal changes that occur in response to heat stress might play an integral role in the decline in productivity. Plasma growth hormone concentration and growth hormone secretion rate declined with hot temperatures (35°C).

Alam (1986) reported that the ideal indoor temperature is 25- 36°C according age of birds. In Egypt air temperature in summer, ranges between 18-22 °C at night and 32 – 38 °C during daylight, but some days have a higher temperature from 42-43 °C; also he mentioned that the optimum inside air relative humidity ranged between 60 % to 70 % for broilers. He summarized and listed the suitable temperature for broilers as follows:

Age	Temperature, °C
First 3 days	34 to 36
After 3 days to the end	
1st week	32
2nd week	30
3rd week	28
4th week	25

From the previous temperatures the heats stress appearing beyond these limits. Therefore, the interior housing environment should be controlled according to these limits. When the broilers are reached at the live weight cooling system should be used to provide best control.

Donald (1993) stated that the effect of hot temperature on broiler there are: panting becomes more intense, and many, most or all birds are panting; Normally pink skin areas turn a dark red as more of the blood circulation is shifted to extremities and the surface of the body to dissipate heat; and feed consumption drops even more, or stops entirely. There are effects caused intense panting and the darkened skin are signs of heat stress, which means birds are unable to get rid of internal heat build-up and their internal core temperatures are rising too high. If high temperature conditions continue, performance is seriously hurt and moralities increase.

El-Hadidi (1989) studied and determined the ambient air temperature inside the broiler house under Egyptian conditions as follows:

Age	Temperature, °C
1st day	35
1st week	32
2nd week	28
3rd week	26
4th week	21

He also found that the air relative humidity inside the broiler house must be ranged between 50 to 70%. One concluded that heat stress appeared beyond these limits for air temperatures and air relative humidity. In the market size needed 21°C to growth and production meat without any stress. Any value of temperature and relative humidity cause heat stress, which reduced production.

Brown-Brandl et al. (1997) reported that as an ambient air temperature increases above thermo-neutral, feed consumption and growth rate decrease. Abrupt extreme thermal stress produces significant losses in all production settings.

Bucklin et al. (2000) stated that cows respond to heat stress by eating less which in turn results in a drop in milk production. Florida (USA) dairy cow's milk production decreases up to 25% in summer if housing and ventilation are inadequate.

If the air temperature reached 47°C, it will be a mortal temperature for poultry, and the temperature must range between 16 - 27°C for big birds in hot days, and increasing temperature leads to increase the mortality rate. (El-Deak and Ahmed, 1985 and El-Soaly, 1997).

Frazzi et al. (2000) reported that deep body temperature is usually maintained at a stable level in homoeothermic animals. When the ambient temperature rises or falls from optimum, the animal reacts with different physiological and behavioral means to prevent body temperature from diverging more than a small amount from the optimal set point. The physiological responses of the cow to heat stress are to (1) increase heat loss through evaporation (sweating and panting), and (2) reduce the heat generated by maintenance (15-20% under heat stressing conditions) and due to production (specifically heat generation linked to digestive activity).

Negative effects of heat stress on the performance of dairy cows are not limited to reduced production, but also lead to a decrease in the quality of milk composition, a reduction in fertility, and greater susceptibility of the animals to disease due to the lowering of their immune system that results from heat stress. (Johnson, 1987; Berman, 1991; Bernabucci et al., 1997 and Bertoni, 1998)

Rensis and Scaramuzzi (2003) stated that there is a decrease in fertility of dairy cows inseminated during the hot months of the year. The utilization of cooling systems may have a beneficial effect on fertility but dairy cows cooled in this way are still unable to match the fertility achieved in winter.

Tao and Xin (2003) mentioned that when the effective environmental temperature (EET) is within the thermoneutral zone (TNZ), core body temperature of adult chickens is maintained between 41.2°C and 42.2°C by thermoregulatory mechanisms with minimal effort. When EET rises above thermo-neutral zone, biophysical defense mechanisms against heat challenge, such as reduced energy intake, come into play. If the thermoregulation mechanism is insufficient to maintain homeothermy, core body temperature begins to rise and eventually leads to death from heat exhaustion. Therefore acute thermal stress can cause significant economic losses.

A system of preventing stress from overheating in dairy cows caused by temperatures higher than 25-26 °C. The system does not require great investments and involves installation of fans in combination with showers in the feeding parlour and atomizers in the sleeping zone. A scheme is presented of a parlour equipped with an air conditioning system based on cooling by surface. Mechanisms of natural thermoregulation allowing control of the body temperature in cows were considered. (Frazzi, 2004)

2.2.2 Heat stress in greenhouses.

Aldrich et al. (1983) reported that unlike homothermic animals, plants are not able to maintain their cells and tissues at a constant temperature, as ambient conditions change. Rather, leaves, stems and roots are normally within a few degrees of the temperature of the surrounding air or soil. As a result, temperature changes in the environment exert a marked effect on plant growth and metabolism. Because of the extreme variability of soil and air temperature, it is difficult to establish precise relationships between plant processes and temperature conditions in the environment. Temperatures depends upon many factors such as time of day, month of year, height above soil surface , leaf dimensions…..etc. One can say that the previous sentences according with heat stress on plants. Therefore, reduce heat stress is important way to protect plants when temperature increased. They added that excessively high temperatures destroy proteins, inactivate enzymes and disintegrate cell members. As well, they added that high relative humidities tend to result in taller, more succulent plants, whereas low relative humidities, combined with high temperature may cause growing tips to burn. A secondary effect of air relative humidity is the response of fungal pathogenic organisms which will not germinate until the air relative humidity is the order of 95%.

2.2.3 Heat stress in storages building.

Buffington et al. (1983) mentioned that environmental conditions inside ventilated produce storages are influenced by: outside weather conditions, building construction, environmental modification systems within the building and heat, moisture and volatiles liberated by the stored produce. Major data required for the design of commodity storages include storage temperature, humidity, and air composition and ventilation rates. They added that pests and various micro-organisms also contribute to uncontrolled degradation of the stored product, if not held in check. With grain, very low moisture contents and cold temperature can generally control insects and micro-organisms. However, with many horticultural crops the humidity must remain high to prevent dehydration.

2.3. Environmental control in agricultural structures.

Oliveira and Esmay (1982) found that the thermal environment is a critical part of the external conditions that directly affects production and growth of livestock and poultry.

Alchalabi (1997) determined some important categories when designing poultry housing as follows:

- -Specification of the building, including dimensions, type and orientation of the roof and building
- -Component of the wall and roof selected for the building and the insulation value.
- -Summer and winter environmental conditions used in designing the building and expected environmental conditions.
- -Specification of the ventilation system and the ventilation rates.
- -Specification of the evaporative cooling system and the information needed to assemble this system as well as the number of the evaporative air coolers if the previous system is not desired.
- -Specification of lighting system and its component.
- -Management information for the poultry house designed; and expected quantity of fuel (energy) used in heating.

2.3.1 Methods to reduce heat stress in agricultural structures.

There are many ways to be used to reduce heat stress inside the agricultural structures. Some of theses ways could be classified into three groups. First by using natural ventilation. Second by using mechanical ventilation and evaporative cooling systems. Third, by using a general procedure such as: insulating the agricultural building, painting the building, orientating the building, configuration of the building, shading greenhouses. A combination of these methods from all three groups could be used.

2.3.1.1 Reducing heat stress by general procuders.

A conservation in energy used for cooling or heating the buildings by choosing the appropriate orientation of the building and using the insulation for ceiling and walls, as well as using attic ventilation. (Buffington, 1978).

El-Soaly (1990) recommended that poultry buildings should be wise East–West where the longitudinal axis is facing toward the south. He added that buildings which differ from recommended orientation need correspondingly greater ventilation rate through powered ventilation or larger ventilation openings.

Stark et al. (1994) found that the application of a reflective roof coating to reduce heat load and energy use corresponding to ventilation fan operation in a broiler housing does not appear to be cost effective based on energy savings alone. The treated building with the reflective roof coating realized only a slight energy savings compared with the controlled building. The design and operation of curtain–type broiler facilities in combination with roof insulation and ventilation from the open sidewalls was sufficient to retard or remove the solar radiation from the roof. While the temperature in the building underneath the roof insulation was lower in the treated house with the reflective roof coating, this lower temperature was not translated to bird height which is the important temperature.

Tiwari and Gyyol (1998) reported that the higher level in the greenhouse, as important specifications, should be adequate and uniform for crop growth and the prevailing winds should not adversely affect either the structure or the operation of the facility. In case of multispan greenhouses, however, shadow of structural components persists in some areas of the greenhouse if the gutters are oriented east-west. One can concluded that greenhouse specification has an effect on the interior environment.

EL – Soaly (2001) used a design of shading overhangs for poultry building in northeast Egypt on the southern and eastern windows to reduce solar radiation entering the building. These overhangs are used as a sun breaker in order to decrease the heat diffuse in the building from the direct solar radiation during the period of 10 A.M to 2 P.M. Farmers can use a moving plastic sheet or canvas to protect the windows from direct solar radiation.

Waligóra and Sobczak (2001) determined some ways to reduce heat stress in poultry housing such as: the ventilation, better roof insulation, painting, and reducing the number of birds per building. They employed some different cooling methods such as wet filters and fogging installations to reduce heat stress. They also stated that choice of a method depends on its costs.

Al-Helal and Al-Musalam (2003) studied the effect of shading materials on greenhouse environment including inside air temperature and air relative humidity, and water and electricity consumption for cooling under Riyadh (Saudi Arabia) conditions. The study was executed in five greenhouses cooled by the fan and pad evaporative cooling system. The greenhouses were covered with single layer polyethylene. Four greenhouses were shaded with green plastic nets on the outside surfaces of covers. Shading levels were 55, 65, 70 and 80% according to the factory specifications. The fifth greenhouse was without shade for the purpose of comparison. Experiments were performed in two periods: the first period was from 1 to 30 October 2000 (moderate summer conditions), and the second period was from 16 to 24 July 2001 (extreme summer conditions). The results indicated that shade had no significant effect on inside air temperature during moderate conditions (P = 0.05). But, inside air relative humidity with shading levels of 55, 70 and 80% was significantly higher than that without shade. During extreme summer conditions, shade significantly reduced air temperatures and increased air relative humidity. The 80% shade significantly reduced the consumption of water and electricity during moderate conditions by 16 and 19.3%, respectively. The 70% shade significantly reduced the consumption of electricity by 12.7%. Shade had no significant effect on the consumption of water and electricity during extreme conditions (P=0.05). It was shown that water and electricity consumption during extreme conditions were higher than those of moderate conditions. The increase rate of water consumption ranged from 79 to 112%, while consumption of electricity increase ranged from 12.7 to 36%.

Wei (2003) investigated some ways in which the environment of dairy cattle can be modified and managed in a hot climate. The structure of the dairy barn, including its orientation was also studied. The features such as side curtains or the use of white paint for the roof are effective ways of shielding the cattle from solar radiation, while a well-designed roof vent enables heat to escape. It also includes detailed advice on the size and design of roof vents for maximum effectiveness. Various kinds of cooling systems such as fans, fans combined with moistened pads and intermittent spraying, are also examined.

2.3.1.2 Reducing heat stress by ventilation.

Ventilation in an agricultural structure supplies fresh air, which is essential to sustain life. It also reduces the extremes of temperature, humidity and air contamination to tolerable limits for confined species. Improved ventilation systems also gave the possibility of the high-density populations of biological systems (livestock and poultry). Such a procedure leads to reducing the building cost per the unit housed and in turn reduces production and labor costs.

2.3.1.2.1 Definitions and characteristics for types of ventilation systems.

Mechanical ventilation can be defined as ventilation with fans, equipment automatic controls (Hellickson et al., 1983). Barrington et al. (1994) suggested that the term natural ventilation refers to those systems not using mechanical power to produce air movement.

Shrestha et al. (1993) reported that the importance of natural ventilation has increased because of the rising energy costs associated with mechanical ventilation and the operation simplicity of natural ventilation systems.

Abd El-Bary (1995) reported that natural ventilation is the oldest form of ventilation and has been used for as long as housing has been made it the most common type of ventilation. In other words natural ventilation can be defined as the movement of air through specified building openings by the use of the natural forces produced by wind and temperature differences (ASAE, 1995).

2.3.1.2.2 Types of natural ventilation.

Hellickson et al. (1983) summarized the fundamentals of natural ventilation as follows:

1- Natural forces involved.

2- Wind characteristics.
3- Wind forces.
4- Flow due to wind forces.
5- Temperature differences forces.
6- Flow due to temperature effects.
7- Combined wind pressure and temperature effects.

They also showed that natural forces for moving air into, through, and out of buildings are wind forces and the difference in temperature between the inside and outside of the buildings. The air movement may be caused by either of these forces acting alone or by a combination of the two, depending upon atmospheric conditions, building design and location.

Natural ventilation differs from forced ventilation in that the latter requires a mechanical energy input to produce the pressure differential necessary to cause air flow. Natural ventilation is the oldest form of ventilation and has been used for as long as housing has been made, it the most common type of ventilation. However, ventilation that is dependent on natural forces is inherently variable and consequently has numerous limitations. These include such factors as the nature of climate geographic area, terrain obstructions to the wind, environmental requirements and others that must be considered in the design of a natural ventilation system and its subsequent management. (Abd El-Bary, 1995)

Examined a new integrated tool for naturally ventilated buildings. They showed that a ventilation system that employs air temperature difference as a means of inducing airflow is called flue or stack (thermally induced) ventilation. The pressure due to thermal forces which arise from difference in density between indoor and outdoor air, because of the difference in their temperatures, is commonly referred to as the stack effect. The stack effect will theoretically only result in flow it, there are at least two openings at different heights connected by a flow path inside the building. In practice a very tall single opening will however also allow flow. The top and bottom parts will in effect act as two different openings. (Barre et al., 1988 and Mathews and Rousseau, 1994)

2.3.1.2.3 Types of mechanical ventilation systems.

Hellickson et al. (1983) reported that neutral pressure ventilation systems employ dual-acting fans that simultaneously draw air out of and force into the building creating a pressure difference in the ventilation system but not in the livestock building. They also stated that the commercially available zero pressure ventilation system uses a specially designed fans for supplying and exhausting equal quantities of air. These ventilation units which contain both of the inlet and outlet are normally located near the center of the ceiling or roof of the building. They also mentioned that neutral ventilation systems established by using equally sized exhaust and pressure fans are becoming increasingly popular. Such systems are used where precise control is desired of both inlet and outlet air. With ventilation system involving paired fans, a pressure fan affect air distribution and the exhaust one which is installed in a convenient location removes, from the building, an amount of the air equal to the air introduced by the pressure system.

ASAE (1995) revealed that under pressure (positive pressure) systems, the pressure in the building forces the humid air out through planned outlets, if any, and through leaks in the walls and ceiling. In winter, if the air humid moisture may condense within the walls and ceiling. The positive pressure systems are suitable for the applications where ventilation air must be filtered to prevent contaminants and pathogens from interring the building. The 20 to 30% of the energy used by fan motors that is rejected as heat is added to the building, an advantage in winter but a disadvantage in summer.

Combined mechanical and natural ventilation systems were executed by Harmon (1996). He showed that in some production systems such as broiler production, birds are placed in one house to brood and will be kept there to market weight. Birds that are in the brooding period require a small amount of ventilation but a large amount of supplemental heat in comparison to adult birds. Therefore, the ventilation system to accommodate this must be flexible and meet the changing needs of the birds; otherwise the birds will be stressed at some point during production. He also stated that in some cases no one ventilation system seems efficient. At certain times fine control of the minimum ventilation rate is needed, that can be achieved by mechanical ventilation. At other times the temperature control and therefore ventilation rate is not as critical and can be dealt more efficiently by using natural ventilation. Hot weather combined ventilation systems are used to incorporate cooling strategies into the ventilation system to reduce heat stress effects. This system uses natural ventilation until the temperature reaches a setting at which time the curtains are closed and mechanical ventilation takes over. The most common types of hot weather combined system use mechanical ventilation to provide tunnel ventilation or to draw through an evaporative pad cooling. The simplest cross-flow negative pressure ventilation system consists of exhaust fans on one sidewall of the building and a continuous slot inlet along the opposite wall. (Heber, 1996).

2.3.1.3 Evaporative cooling systems

The evaporative cooling systems are based on the process of heat absorption during the evaporation of water. Evaporative cooling systems for greenhouses or livestock housing are generally of two types, misting or fogging systems and fan and pad cooling systems. These systems are normally evaluated in terms of an evaporative cooling or saturation efficiency, which is defined as the ratio of the temperature drop provided by the system to the difference between dry and wet-bulb temperature. Hahn and Osborn (1969) defined evaporative cooling as an adiabatic process which lowers the dry-bulb temperature while increasing the dew point temperature of an air and water vapor mixture. The reduction in dry-bulb temperature is limited to the wet-bulb depression of the air entering the cooler. Wang (1993) described the evaporative cooling as an air conditioning process that uses the evaporation of liquid water to cool an air stream directly or indirectly so that the final dry-bulb or dry and wet-bulb temperatures of the air stream were being cooled.

2.3.1.3.1 Types of evaporative cooling

Wiresma and Short (1983) indicated that passing air through a wet porous material, or pad, is the more popular method of providing the air moisture content. The porous pad can be wetted by dripping water onto the upper edge of vertically mounted pads, by throwing or spraying water on to the surface face or by rotating a horizontal cylindrical pad with the lower portion submerged on water. Theses methods are often referred to as the drip type, slinger type and rotary type, respectively.

Warm outside air is drawn in through the pads. Water in the pads, through the process of evaporation, absorbs heat from the surrounding pad and frame as well as from the air passing through the pad. The air entering the house can be as much as 14°C cooler than the outside temperature if the humidity is low. There are two main considerations in this system: (1) the rate at which warm air is to be removed, allowing cool air to be drawn in, and (2) the area of the pads. (Nelson, 1990).

Bucklin et al. (1991) mentioned that if the droplets are very small it is called fog or mist system. This system is not recommended in humid climates. The fog can create a "steam bath" effect instead of cooling the cow.

Turner et al. (1993) stated that one of the most commonly used systems to reduce heat stress is sprinkler and fan cooling, the sprinklers create droplets that wet the skin of the cow and with the fans they create an evaporative cooling.

Wang (1993) classified the types of evaporative cooling systems as: (1) direct evaporative, (2) indirect evaporative, and (3) indirect–direct evaporative as shown in Figure (2-2). In a direct evaporative cooling system, the air stream to be cooled comes directly into contact with the water spray or wetted medium, in an indirect evaporative cooling system, the primary air stream to be cooled is separated from a wetted surface by a flat plate or a tube wall. He added that when cooled air leaves an indirect evaporative cooler during a hot summer, its dry-bulb may still be raised. Also, when cooled air by direct evaporative cooling found that higher relative humidity. Therefore, using in a two–stage indirect-direct cooler, a direct evaporative cooler is always connected in series after an indirect evaporative cooler. (or pad cooling system). On the other hand, evaporative air cooling can be divided into two groups, (a) direct cooling, where moisture is added to the air stream and (b) indirect cooling, where the moisture content of the air stays constant.

Fogging nozzles are usually installed laterally across the house in successive rows, but many. Variations are seen, including nozzles mounted inside air inlets. Whatever the configuration, the object is to put a fine mist of water into the tunnel air stream to be evaporated and bring about cooling. The exact number of nozzles depends on the pressure used and the nozzle ratings. The biggest

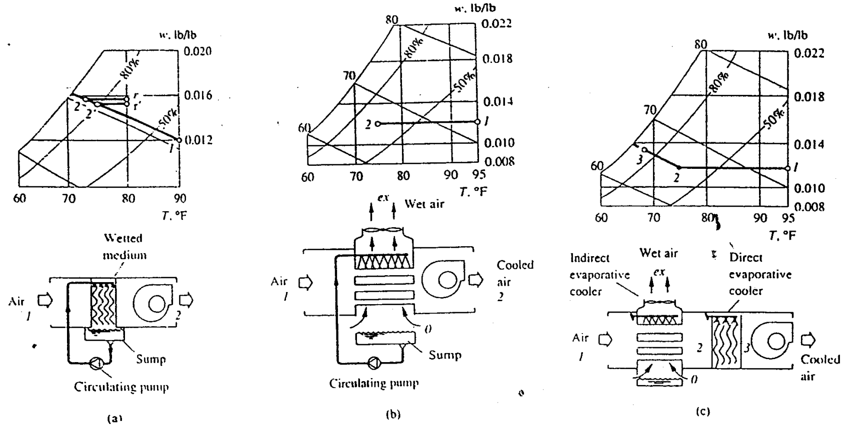


Figure (2-2): Types of evaporative cooling systems: (a) Direct evaporative cooling; (b) Indirect evaporative cooling; (c) Indirect-direct evaporative cooling (Wang, 1993).

systems require close monitoring to make sure the desired cooling effect. (Donald, 2000)

2.3.1.3.2 Utility of evaporative cooling

Esmay (1969) stated that evaporative cooling pad system is efficient where there are low wet–bulb and high dry–bulb temperatures. An efficient pad and fan evaporative cooling system can decrease the dry–bulb temperature of the incoming air adiabatically to within 2°C or 3°C of the wet– bulb temperature.

Reece and Deaton (1971) reported that the performance of broiler males improved when the ambient air temperature was reduced, using evaporative cooling, to 29.4°C from initial temperature of 35 and 37 .8°C . Better female performance was also reported.

Canton et al. (1983) investigated the combined effects of ambient dry-bulb temperature and humidity on deep body temperatures and respiration rates in unrestrained, mature male broiler breeders. They reported that a reduction in thermal stress resulted when environmental conditions were changed from a higher dry-bulb temperature and lower humidity to a lower dry -bulb temperature and higher humidity. They concluded according to the reductions of physiological responses a potential for evaporative cooling to reduce thermal stress of birds in hot humid climates.

A plenum concept applied to evaporative pad cooling for broiler housing has been given by Timmons and Baughman (1984). They mentioned that, a disadvantage of pad system is the tendency to place exhaust fans together at one end of the house to save wiring costs and to install pads at the far end, this can result in a temperature rise 5°C between the pad and exhaust fan due to sensible heating by the birds.

Evaporative cooling using a pneumatic misting system was studied and examined by Bottcher et al. (1989). They showed that non–uniformity of interior temperature and reduced productivity of birds far from the pads would be expected. Under such conditions pad systems also require fans to provide air movement through the wetted media and are therefore not suitable for curtain sidewall houses employing natural ventilation by wind. Misting or fogging can be used in houses with curtain sidewall.

Albright (1990) mentioned that mechanical refrigeration is normally used to cool air within buildings for human habitation. But this type of control has not been found necessary for animal and plants because mechanical refrigeration would be expensive to install and energy intensive to operate to meet the large cooling loads encountered in modern barns and greenhouse range. Hence, evaporative cooling system is most effective in dry and hot climate.

Kassem (1992) revealed that evaporative cooling systems are the choice of many producers in the less humid and dry locations of the world because of relative ease of installation in both new and remodeled farm structures. Evaporative cooling for farm structures (plants and animals structures) vary according to the way in which air is brought in contact with free water surface. It is typically provided by pad or misting system.

Chung and ReuySong (1998) developed and tested a mist cooling system for greenhouses using the principle of evaporative cooling during the summer in Taiwan. The water for evaporation was provided through nozzles via a high pressure pump. An especially long air flow path was designed to increase the cooling efficiency. Results showed that the cooling system operated at 80-90% efficiency, reducing air temperature within the greenhouse to 5-7°C below external air. The structure of the system is both simple and inexpensive. The mist system has the advantage over standard pad systems in that it is not liable to mould growth.

Deling et al. (2000) reported that the cooling of greenhouses in the humid summer can be achieved using a cold fog cooling system controlled by a high-pressure sprayer. The droplet size distribution is 10-15 µm volume mean diameter. During the filling of the greenhouse with the cold fog and the subsequent droplet to evaporation, the air in the greenhouse was cooled and the ground was not wetted. The preliminary experiment showed that the temperature could be decreased effectively.

2.4. Pad–fan evaporative cooling system.

Pad cooling has become much more common than in-house fogging, primarily because pad systems are easier to manage and do not risk wetting the house down. Another reason why pad cooling is now more capability, which can only be achieved by pad cooling (Donald, 2000).

Wiresma and Short (1983) reported that in the pad and fan system, the fans are mounted on one side or end to draw air through a pad or pads located on the opposite wall. Either vertical or horizontal pads may be used. The pads should be located on the wall exposed to the prevailing winds. They added that the pad height is generally between 0.5 to 2.5 m when mounted vertically in order to achieve uniform water flow.

2.4.1 Factors affecting the performance of a pad –fan evaporative cooling system.

The factors affecting evaporative cooling systems using pad-fan system can be determined by several researchers (Benham and Wiresma, 1974; Wiresma and Short, 1983; ASHRAE, 1992; Bottcher et al., 1992; Sharaf, 1994; Abdel-Rahman, 2000; Teitel and Baiely, 2000; Uğurlu and Kara; 2000; Liao and Chiu 2002; St- Pierre et al., 2003 and Flores et al., 2004) as follows:

1-Weather conditions.

2-Pad material.

3- Pad thickness and density.

4- Pad face air velocity.

Cooling potential expressed as temperature reduction (ΔT), saturation efficiency (SE) and unit evaporative cooling performance (Unit ECP) and temperature humidity index (THI) are important parameters that can be used to judge an evaporative cooling system.

2.4.1.1 Weather conditions.

ASHRAE (1992) reported that the effectiveness of evaporative cooling depends on weather conditions. System design is affected by the prevailing outdoor dry and wet–bulb temperatures as well as the specification of the system. For example, a simple residential direct evaporative cooling system, with an effectiveness of 80 %, will provide satisfactory room conditions (given an adequate quantity of outdoor supply air and a well–designed exhaust system) throughout the cooling season in areas such as Reno, Nevada (USA).In the same location, additional cooling effect can be gained by the addition of an indirect evaporative precooling stage, which lowers the temperature (both dry – and wet – bulb) entering the directed evaporative cooling stage and, consequently, lowers the supply air temperature.

Campos et al. (2002) studied the air temperature reduction potential through the use of evaporative cooling systems for the Maringá-Parana region (Brazil) using 13 years data from Maringá Meteorological Station. Temperature reducing potential through evaporative systems was related to the environmental temperature and relative humidity. The obtained results showed that the months with the best potential of temperature reduction were August, September, October and November, due to the low values of air relative humidity. The months of December, January, February and March presented lower potential of temperature reduction due to the incidence of larger values of air relative humidity despite taking place during summer. It is concluded that evaporative cooling is promising to the most critical hours (13-18 h), which presents the highest air temperature.

2.4.1.2 Pad material.

Wiresma and Short (1983) showed that manufacturers have tried pad materials of wood, metal, mineral, glass and more recently, plastic and cement In drip type coolers where water flow is somewhat limited, porous materials generally provide the most efficient cooling. Although the more inert materials resist decay and have longer life than the wood, most lack the “wick–like “ absorbency that ensures good distribution of water. On the less porous material, the water runs down in a solid trickle, providing much less surface area. Aspen wood excelsior has been one of the best materials for blanket type pads in coolers. Although somewhat more resistant to decay than many other woods, deterioration is still a problem such that pads lose much of their efficiency after the first year. Mid–season washing with fresh water to remove filtered dust and debris helps retain efficiency and prolong life, but they should still be replaced at least every season.

Sharaf (1994) used cooler pads that made manually from two different materials. First pad material was from leaf fibers of ornamental palm and the second one was from common reed plant. He found that the first pad material was better in resultant cooling for the whole operating period for all treatments and conditions than the second pad material.

Dzivama et al. (1999) compared and tested four pad materials ground sponge, stem sponge, jute fiber and charcoal as pads for an active evaporative cooler at an ambient air temperature of 32°C and air relative humidity of 25%. They found that the stem sponge gave the best performance, with a temperature reduction of up to 18°C and air relative humidity increase up to 84%. The structure evaluated for storing mangoes (*Mangifera indica*), bananas (*Musa sapientum*) and tomatoes (*Lycopersicon esculentum*). Results also indicated that the produce kept in good physical conditions for 18 days in the cooler, compared with 9 days in ambient conditions.

Abdel-Rahman (2000) evaluate and examined two evaporative cooling materials in greenhouse cooling systems. These materials are aspen fibers and long wheat straw. He found that both evaporative materials had almost the same cooling performance with an average value of 22%. The maximum and minimum cooling performance values for both evaporative materials were 35% and 15%, respectively. He stated that long straw is usable and accessible substance to the greenhouse growers. The proposed long wheat straw will be extremely important when considering the greenhouse growers who are located in arid region areas. The necessity of long straw is become vital when regular aspen fiber wooden fiber or even corrugated vertical pads (cross fluted cellulose) are difficult to acquire as well as the advantage of the less-expense of long straw as a material.

Liao and Chiu (2002) developed a compact wind tunnel to simulate pad-fan evaporative cooling systems and to provide direct measurement of system performance. Two alternative materials including one made of coarse fabric PVC sponge mesh 0.25 cm diameter in pinhole and fine fabric PVC sponge mesh in 0.75 cm diameter pinhole were tested as pads in wind tunnel experiment. This experimental examined the effects of air velocity, water flow rate, static pressure drop across pad and pad thickness on evaporative cooling efficiency. They found that fine fabric PVC sponge was effectiveness.

2.4.1.3 Pad thickness and density:

When using pad-fan evaporative cooling system, vertical pad, that effective cooling can be accomplished by moving air at about 1.27m/s through a 2.5 to 5 cm thickness of aspen excelsior pad of 32 kg/m^3 density. Water is applied to the pads at a rate of 7.5 times as fast as water is evaporated. This will provide a saturation efficiency of about 80%, but efficiency falls off rapidly if the system is not properly maintained (Wiersma, 1969).

Welchert and Wiersma (1972) reported that in vertical pad design, pad deterioration and dust accumulation must be considered when selecting thickness and density. Increasing thickness and density, increases air– water contact time and decreases dry air leakage; increasing both improves saturation efficiency but at the expense of a higher air flow resistance. Pad airflow resistance also increases as dust and salts accumulate. A pad thickness of 2.5 to 5 cm containing 0.15 to 0.22 kg of excelsior per 0.1 m^2 is recommended.

Benham and Wiersma (1974) employed five different thickness (2.5, 5, 7.5, 10 and 12.5 cm) of aspen excelsior as pad material mounted vertically. They found that an effective and economical pad design would be 7.5 cm thickness, 3.66 kg/m^2 with a pad face air velocity 1.02 m/s. Although these recommended values can be considered optimum, moderate deviations do not seriously effect saturation efficiency. However, essential that a pad of any thickness be evenly distributed and a water flow of any rate be designed and maintained to wet the entire pad area.

Wiersma and Short (1983) found that an increase in pad thickness directly increases the resistance to airflow while increasing the contact time of air traversing the pad. However, as air passes through additional thickness of pad, the vapor pressure differential decreases. This results in a decrease in the rate of evaporation into a given element as it continues its path through the pad .The precise relationship is not well known. Increased pad density enhances overall porosity or capillarity providing more uniform distribution of water. It also requires higher water flows and increase resistance to air flow.

Bottcher et al. (1992) used 15 cm thick vertical pad of cellulose materials as pad-fan evaporative cooling system. They found that the evaporative cooling efficiency of the pads was 80 to 89%. An average temperature reduction at bird level of about 6.7 °C was obtained when outside temperature was 32 °C or greater.

Sharaf (1994) studied the effect of pad thickness on evaporative pad cooling system effectiveness, this pad mounted vertically. He used four-pad thickness namely 2, 4, 6, 8 cm. He found that saturation efficiency increased by increasing pad thickness. But there was a limiting pad thickness for a specific pad-face air velocity beyond which the saturation efficiency decreased. He added that the thickness of 2 cm of leaf fibers of ornamental palm was the limiting thickness at 0.5 m/s pad face air velocity. The thickness of 6 cm of common reed plant was the limiting one at the same pad face air velocity. No significant differences in the mean values of saturation efficiency between 2 and 4 cm pad thickness.

Alchalabi (1996) Compared between 10 and 20 cm thick of two types of vertical pad material in an evaporative cooling system. He mentioned that the best selection was when using pad with depth of 20 cm with 1.5 m/s pad face air velocity. Cooling efficiency was 91% and 65% at 20 and 10 cm pad thick, respectively.

Mekonnen and Dodd (1993) studied the effectiveness of different microclimate modifiers for hot weather livestock housing in a model of livestock housing. Fan ventilation and evaporative cooling systems were considered. Temperature and velocity profiles together with air flow patterns were taken as a measure of air distribution in the model. Saturation efficiency and static pressure drop across the pads were also taken as criteria for evaluating evaporating cooling pad characteristics. He found that kool-cel vertical pads 10cm thick appear to have higher saturation efficiency at specific air velocity, with an acceptable level of relative humidity and temperatures.

Papa and El-Galabi (1997) studied the effect of pad thickness on evaporative cooling system, mounted horizontally. They used 9, 13, 22 cm pad thickness palm fibers. They constructed the pad as a ceiling of a dairy cow shelter. They found that cooling efficiency was 70.1, 74.2, and 87.5% for 9, 13, and 22 cm thickness of pad, respectively.

Dzivama et al. (1999) pointed out stem sponge, as vertical pad cooler was effective at 6 cm pad thick at 2.7 air suction velocity and 90 ml/s water flow rate. The reduction temperature at the previous conditions was 12 °C, but air relative humidity increased.

Liao and chiu (2002) studied three different vertical pad thickness namely 5, 10 and 15 cm. These pads were made of industrial materials. They found that the thicker the pad is the higher cooling efficiency. Increasing pad thickness from 5 cm and 10 cm to 15 cm increase cooling efficiency by about 24% and 16%, respectively.

2.4.1.4 Pad face air velocity

Air velocity through the pads varies at different points within the pad and is difficult to measure. The velocity entering or exiting from the pad, referred to as pad face velocity. It is a basic design parameter used for calculating pad area (Wiresma and Short, 1983)

Pieper and Wiresma (1971) reported that increasing air flow in such a way that leads to a higher air velocity through the wetted pad reduces air-water contact time. Consequently, unit weight of air absorbs less water and provides less cooling, but the heat exchange required to maintain the desired temperature level is accomplished by the larger quantity of air. The smaller increase in air relative humidity permits a more comfortable environment. Thus, the extra capacity of the cooler maintains a desired dry–bulb temperature without the associated high relative humidity in conventional systems.

Benham and Wiersma (1974) studied the effect of pad face air velocity on the pad-fan cooling system efficiency, mounted vertically. They used four different pad face air velocities namely 0.51, 0.76, 1.02 and 1.27 m/s. They found that an effective pad face air velocity would be 1.02 m/s. As pad–face air velocity increases, the thickness of the water film decreases resulting in increased heat transfer and evaporation rates and corresponding increases in saturation efficiency .There is, of course, an adequate time for heat and mass transfer to occur. Increasing the air velocity reduces the contact time, which can be restored by increasing pad thickness. They also stated that velocities above 1.52 and 1.25m/s at horizontal and vertical pads, respectively have not been recommended because of increased static pressure loss and a tendency to pull free water into the air stream. Meanwhile Liao and Chiu (2002) found that pad face air velocities greater than 1.75 m/s tended to pull free water into the air stream because air pressure increases always at high velocity. Welchert and Wiresma (1972) pointed out that pad size must be sufficient to provide recommended pad face air velocity of 0.7 m/s (± 0.07 m/s).

Greenhouses, animal shelter, automotive paint booths and similar applications use a system of exhaust fans in the wall or roof of a structure, with wetted pads placed so that outdoor air is drawn through the enclosed space. The pads are wetted from above by a perforated trough or similar device, with the excess water draining to a gutter from which it may be wasted or collected for recirculation. The pad should be sized for an air velocity of approximately 0.5 m/s for standard fiber pads, 1.36 m/s for 10 cm rigid pads and 2.2 m/s for 15 cm (ASHRAE, 1992).

Mekonnen and Dodd (1993) stated that use vertical pads under 1.5 m/s pad face air velocity gave reduction of 10 °C. This can be expected when relative humidity of the supplying air does not exceed 60%. Higher saturation efficiency appeared at, kool-cel pads 10cm thickness with air velocity 1.5m/s and water flow rate of 3 L/min.

Sharaf (1994) studied the effect of pad face air velocity on the performance of evaporative cooling pad system, mounted vertically .He operated the system at five velocities namely 0.5, 1.0, 1.75, 2.5 and 3.5 m/s comparing among means of saturation efficiency at each pad face air velocity L.S.D test revealed that no significant differences between 1.00, 1.75 and 2.5 as well as 3.5 m/s pad–face air velocity.

Wang (1993) reported that the air velocity of the cooled air stream flowing through the passage is usually from 0.5 to 2.03 m/s. It is important to limit the air velocity of the wet air stream in order to prevent carryover of the water droplets.

Alchalabi (1996) used vertical pad fan evaporative cooling system in broiler housing (70*11*3m). Two different levels of pad face air velocity namely 1 and 1.5 m/s were used when outside temperature was 45°C and outside air relative humidity was 10%. He found that better inside conditions occurred at air velocity of about 1.5 m/s. The number of pads with pad face air velocity of 1.5m/s was 35 pads, which is economically better and without side effects on the efficiency. The reduction temperature was 18 °C and 19 °C at 1.5 m/s and 1m/s respectively at 20 cm pad thick. Relative humidity increased by about of 48% and 53% at the same conditions.

Abdel-Rahman (2000) used aspen wooden fibers and long wheat straw as horizontal pad materials. The air velocities through the pads were 1 m/s and 3.5 m/s. He found that there was a strong correlation between incoming air temperature versus air velocity through the pad for aspen wooden fibers rather than that for long wheat straw with R values 0.62 and 0.54, respectively.

2.4.1.5 Saturation efficiency and unit evaporative cooling performance.

Evaporative cooler efficiency is a common term can be used to indicate saturation efficiency, the effectiveness with which water is transferred to the air, which is an important index to assess the performance of an evaporative cooler (Wiresma, 1969 and Wang, 1993). AbouZaher (1998) defined saturation efficiency as the ratio of change in saturation achieved to potential change in saturation. In other words, saturation efficiency is the ratio of temperature drop provided by the system to the difference between dry and wet temperatures or wet bulb depression.

Welchert and Wiresma (1972) stated that cooling efficiency depends upon the uniform distribution of air and exposure to uniform pad moisture. Large holes in the pad must be corrected so all incoming air will pass through a uniform wet pad medium. The building must be reasonably air tight to insure control of the conditioned air. Cleaning the water distribution nozzles and replacing the sump water should be done at least weekly. Daily inspection to assure efficient operation should be a routine procedure.

Benham and wiresma (1974) mentioned that cooling capacity is dependent upon the amount of airflow and the saturation efficiency. These in turn dependent upon such factors as characteristics of the pad, air velocity through the pad and water flow rate. They found that saturation efficiency increases with pad thickness and generally with pad–face air velocity. They added that saturation efficiency appears to increase with pad face air velocity even beyond 1.27 m/s in the 5, 7.5, 10 and 12.5 cm pad thickness.

Wiresma and Short (1983) reported that saturation efficiency (SE) can be calculated from psychometric charts. It helps us to describe efficiency of coolers but it can not determine cooling capacity. Therefore a measurement identified as evaporative cooler performance unit (ECP) is some times used to rate the performance of coolers. The ECP value is numerically equal to the number of heat units involved in the exchange, and can be calculated as either sensible heat or as latent heat. The Unit ECP, however, does not measure absolute performance under base conditions that permit comparison. Since total quality of the exchange depends upon the wet bulb depression, an improved measuring unit which is the Unit ECP. The unit describes cooling effect delivered per hour per degree of cooling potential and eliminates variable associated with local conditions.

Timmons and Baughman (1983) reported that by using a plenum pad concept, evaporative efficiency of 80 % can be obtained with reasonable ventilation rates while simultaneously maintaining average inlet velocities above 3 m/s and average floor velocities of 0.9 m/s. Also, pad location effects can cause differences in evaporative cooling efficiencies of up to 15 %, apparently due to direct solar radiation effects.

Bottcher and Baughman (1990) found that the evaporative efficiency of pad systems typically changes only slightly over a wide range of air velocities through the pad, so that water evaporation rate for given outside psychometric conditions is essentially proportional to the ventilation rate.

ASHRAE (1991) reported that in a hot environment, where ambient heat control is difficult or impractical, cooling is accomplished by passing below skin–temperature air over the body. Evaporative coolers are used for this purpose as well. The performance of evaporative cooling is directly related to climatic conditions. The entering wet–bulb temperature governs the final dry–bulb temperature of the air discharged from a direct evaporative cooler. The capability of the direct evaporative cooler is determined by how much the dry–bulb temperature exceeds the wet–bulb temperature. The performance of indirect evaporative coolers is also limited by the wet –bulb temperature of the secondary air steam.

Abd EL – Bary (1995) reported that when evaporative cooling system was actuate, the air relative humidity of inside conditions was increased, because the maximum relative humidity occur with the minimum dry–bulb temperature and vice versa. The air relative humidity was changed by the time of day and inside relative humidity was ranged between (60 % - 97.5 %), (60 % - 82.5%), and (62% - 81 %) for the three days of test, respectively. He added that by using the mist system, high efficiency cooling that can be achieved maximum evaporative potentials in any environment. The average values of evaporative cooling efficiency were ranged between 23.8% to 66.57%.

Liao and Chiu (2002) used a wind tunnel ventilation to test performance industrial pad materials. They found that higher efficiencies are obtained with thicker pads and slower air velocities. Cooling efficiency expressed in terms of SE was 95.5%, 88.65%, 72.03% at 15, 10, 5 cm thick of pad, respectively. It was 86%, 82%, 80% at 0.75, 1.25, 2.00 m/s pad face air velocity, respectively.

2.4.1.6 Temperature humidity index (THI).

Several indices have been developed and used to predict comfort, or discomfort of environmental conditions by Oliverira and Esmay (1982). One of these indices was dry–bulb temperature and humidity. The most common comfort index is temperature– Humidity Index. Hupp and Rathwell (1998) stated that the temperature humidity index (THI) is used as a guide to measure heat stress. THI combines the effects of temperature and humidity into one value.

Hahn and Osburn (1969) used temperature–humidity index value of 21.1°C (70°F) for dairy cattel. They added that in the practical situation, this threshold could be approximated by a thermostat set for 23.8 °C (75°F) with little or no detrimental effect on milk production.

Gates et al. (1991) modified a method to analyze the operation of misting systems to cool livestock housing by minimizing (THI). This procedure is similar to the analysis for evaporative pad cooling, except that interior maximum relative humidity is specified instead of pad efficiency. The minimization of (THI) is shown to be equivalent to the minimization of interior dry–bulb temperature. They concluded that any environment index whose slope on a psychometric chart is greater than that of the wet– bulb temperature line will be minimized during misting operation, if the interior state point is at the intersection of maximum desired inside relative humidity and outside wet – bulb temperature.

Abdel – Bary (1995) mentioned that temperature–humidity index (THI) was affected by actuation of evaporative cooling system for laying hen housing and it was minimized using the mist system. The minimizing of THI provided a rational objective for environmental control. He found that through three days of this experimental work, the misting system was able to minimize inside THI, which was less than outside THI. He added that the depression in inside THI was due to decrease in inside dry-bulb temperature which resulted from the misting actuation.

Brown-Brandl et al. (1997) studied the effects of temperature and humidity on physiological responses of turkeys. Eight temperature–humidity combinations were evaluated from 23 to 40°C and 40 to 90% RH. For older birds, (15-and-20wk-old), the THI that cannot be exceeded without death is defined by THI= 36.7°C (15 wk) and 36.4°C (20 wk) .In contrast, THI = 37.5 °C are not the maximum for 6-wk-old poults; even though this THI was not life threatening, the poults were clearly heat–stressed.

Bottcher and Hoff (1997) studied the dynamic responses of growing, *ad-libitum*-fed cattle under heat waves conditions in USA. These conditions were expressed by temperature humidity index (THI). They presented an environmental profile supports that the use of THI more than or equal to 26.1°C (79°F) as a threshold for feedlot cattle placed at risk, and suggest that several hours with THI more than or equal to 28.8 °C (84 °F) and limited or no night-time recovery periods with THI less than or equal to 23.3 °C (74 °F) can result in death of vulnerable animals unless immediate action is taken to limit excessive heat loads.

Mayer et al. (1999) recommended some of temperature-humidity index (THI) values which are critical (or break point) for Holstein-friesian cows by using types of cooling strategy. These values can be listed as follows:

A comparison study between evaporative cooling pads system and natural ventilation in broiler housing using temperature-humidity index (THI) was carried out by El-Soaly (2002). He found that the mean reduction of THI were 16.5% and 4% for the evaporative pad cooling and natural ventilation buildings, respectively.

St-Pierre et al. (2003) reported the following THI threshold values for some species:

Dairy cows	broiler chickens and turkeys	laying hens
21.1°C (70 °F)	25.5°C (78° F)	21.1°C (70 °F)

Flores et al. (2004) found that milk yield which produced by dairy cows affected by the heat stress level (temperature and relative humidity). Therefore, they classified the heat stress levels by temperature-humidity index (THI) for dairy cows as fallows:

THI range	heat stress level
THI less than 22.2 °C (72 °F)	no heat stress
THI ranged between 22.2 °C (72 °F) to 24.4 °C (76 °F)	mild heat stress
THI higher than 24.4 °C (76 °F)	moderate heat stress

Martello et al. (2004) studied the influence of some cooling systems inside Holstein cows housing on milk lactating. Three housing systems were used: (i) control housing; (ii) housing with mist and fans, and (iii) shade cloth. The microclimate data of each housing were registered to calculate the temperature humidity index (THI), the black globe humidity index (BGHI) and the enthalpy. Twenty six days with high enthalpy were selected and analyzed. THI and BGHI in the hottest registration times were able to differentiate the climatic housing, which shows better performance than the others. The lower enthalpy was observed in the shade cloth treatment He also showed that temperature humidity index from 23.8 °C (75 °F) to 24.4 °C (76 °F) was not associated with stress conditions for the animals, although many studies propose this situation as stressing.

Somparn et al. (2004) found that a THI more than or equal to 28.8 °C (84 °F) was assumed to represent conditions where production losses of cattle and buffalo in Thailand would be likely to occur.

2.5 Application of evaporative cooling in some agricultural structures.

Timmons et al. (1981) used pad-fan evaporative cooling system with cooling efficiency 84% and pad face air velocity 0.69 m/s in two summer broiler trials. First trials on May where outside conditions are very moderate (outside temperature ranged from 26 to 29°C. Second trials on June where outside conditions are very extremely (outside temperature higher than 32°C). They found that mass gains of the cooled birds were 0.30 and 0.14 kg for the two trails, respectively.

Simmons and Deaton (1989) used an evaporative cooling system for increasing production of larger broiler chickens during hot summer conditions. They showed that an evaporative cooling system improved and can increase body weight and feed conversion ratio as compared with rearing under typical summer conditions.

The chief purpose of evaporative cooling as it is applied to fruits and vegetables is to provide an effective, yet inexpensive, means of improving common storages. However, it also serves a special function in the case of oranges, grapefruit, and lemons. Color change (degreening) is achieved through a sweating process in special rooms equipped with evaporative cooling. The evaporative cooler may be floor-mounted or located near the ceiling in a fan room. The system should be designed to discharge air horizontally at the ceiling level (ASHRAE, 1991).

ASHRAE (1992) revealed that the production efficiency of egg layers and breeders can be enhanced in some areas by using evaporative coolers. Broiler operations are also being considered for evaporative cooling. Experiments have shown that while the feed conversion ratio is not necessarily improved, the average bird has a greater mass at the conclusion of the growth cycle. Most broiler houses today depend on lowering side curtains and natural draft to reduce the temperature within the house. However, when the ambient out side temperature exceeds 38°C, evaporative cooling is often the only way to keep the flock alive.

Al-Amri (2000) conducted a study on two experimental even-span gable greenhouses to grow and produce a cucumber crop during summer. One of these greenhouses was covered by 0.8 mm thick fiberglass reinforced plastic. The other was covered by 0.1 mm thick double-layered polyethylene sheets. Two identical evaporative cooling systems (cooling pads and extracting fans) were designed, built and used to reduce the ambient air temperature inside the two greenhouses. These systems of cooling were automatically operated according to the ambient air temperature inside the greenhouse using a differential thermostat. He found that the evaporative cooling system inside the fiberglass greenhouse was more efficient than that inside the polyethylene greenhouse due to the difference in intensity of solar radiation inside the greenhouses. Consequently, the fiberglass greenhouse enhanced the rate of growth and increased the fresh yield of cucumbers by 32.87% as compared with the polyethylene greenhouse.

Teitel and Bailey (2000) measured and recorded the temperature and humidity gradients during summer in a rose production greenhouse equipped with a ventilated cooling-pad system and a half shaded plastic roof. In a steady regime, the cooling process reached a good efficiency (60%, in agreement with the theoretical value) and succeeded in maintaining the greenhouse temperature at 10 °C lower than outside. The temperature of the fresh air rose from the pads to the middle of the greenhouse, while the humidity content did not increase. The physical data were compared with those predicted by an analytic model describing the greenhouse as a heat exchanger. The model helps to understand the particular temperature and humidity profiles of the airflow along the greenhouse. It also suggests that greenhouse roof shading could be avoided under dry climates, because the evaporative cooling process is sufficient to prevent overheating.

Singh and Mandhar (2002) examined the performance of some exotic cultivars of gerbera (*G. jamesonii*) under fan and pad cooled greenhouse environments at the Indian Institute of Horticulture Research, Bangalore, Karnataka, India from July 1998 to June 1999. They found that the temperature inside the greenhouse could be controlled from 24.7 to 30.5°C when the ambient temperature varied from 27.4 to 35.5°C. The lowest temperatures of 8.0 and

6.7°C were recorded during April and March, respectively. The air relative humidity in the greenhouse varied from 44 to 77%, while the outside RH values ranged from 20 to 67%.

Uğurlu and Kara (2000) used pad-fan evaporative cooling system in battery cage house of a commercial egg producing company in the Konya Province of Turkey. Cooling pads served as air inlets, and evaporative cooling occurred when dry hot air flowed into the pads. The pads reduced air temperature by 4.2 to 16.2°C relative to the outside daily maximum temperatures. The average reduction in air temperature was 10.6 °C. The average evaporative cooling efficiency of the pads was 87.5%. Interior temperatures in the cages were decreased by 5.4 to 6.4 °C when external temperatures were 30 °C or higher.

Kittas et al. (2003) developed a simple climate model which incorporates the effect of ventilation rate, roof shading and crop transpiration to predict the temperature gradients along a greenhouse. In order to calibrate the proposed model, measurements were performed in a commercial greenhouse equipped with fans and pads and shaded in the second half. Experimental data showed that the cooling system was able to keep the greenhouse air temperature at rather low levels. However, due to the significant length of the greenhouse (60 m), large temperature gradients, (up to 8 °C) were observed from pads to fans. The model was calibrated by fitting temperatures in the middle and at the end of the greenhouse. The model was validated on experimental data different from those used for the calibration and then it was used to study: (i) the influence of different ventilation rates combined with shading on air temperature profiles along the greenhouse length; and (ii) the influence of the outside air temperature and humidity on the performance of the cooling system. High ventilation rates and shading contribute to reduce thermal gradients. Despite its simplicity, the model is sufficiently accurate to improve the design and the management of the cooling pad systems.

Correa-Calderon et al. (2004) conducted a study on Holstein and Brown Swiss dairy cows to evaluate the effect of two different cooling systems on physiological and hormonal responses during the summer. A control group of cows had access only to shade (C). A second group was cooled with spray and fans (S/F) and the third group was under an evaporative cooling system called Korral Kool (KK). The maximum temperature humidity index during the trial was ranged from 22.7 °C (73 °F) to 29.4 °C (85 °F). Rectal temperatures and respiration rates of the C group were higher ($P<0.05$) than those of the S/F and KK groups in both Holstein and Brown Swiss cows. There was no significant difference in the hormonal response of the two breeds. These results demonstrate that both cooling systems may be used to increase the comfort of Holstein and Brown Swisss cows during summer in hot and dry climates.

2.6 Potential of evaporative cooling systems as an aid to reduce heat stress.

Wilson et al. (1983) employed an evaporative cooling with fogging nozzles system in broiler houses. The obtained data showed that 2-3 % lower mortality, 0.03 kg heavier broiler birds , and 0.03 – 0.06 kg /kg lower feed conversion ratio in buildings having fogging systems , with exhaust fans and air inlets arranged to provide ventilation across the building width , compared with buildings ventilated similarly without fogging. Timmons et al. (1984) found that pad systems have also been credited by growers and equipment dealers with providing substantially more cooling effect than misting systems in intensive broiler housing.

Bottcher and et al. (1991) showed that the misting systems are often termed fogging systems when relatively high water pressure over 700 kPa are used to provide a fine mist . Evaporative pad systems require mechanical ventilation to force airflow through the pads while misting systems can be employed in houses ventilated naturally or mechanically. Evaporation rates with pad systems are determined by the pad evaporative cooling efficiency and ventilation rate (since ventilation air flows through the pads). However , misting rates are typically not directly related to ventilation rates since the misting and ventilation systems operate independently . Also, the temperature of cooled air exiting pads can be readily measured and used in computing efficiency, but the interior temperature in a poultry building equipped with a misting system is affected by sensible and latent heating from birds and other surfaces. Therefore, the difference between outside and interior air temperature cannot, in general, be used directly as the dry bulb temperature reduction when computing efficiency. This necessitates some modification in the method for determining efficiency. They added that one advantage of pad system is their superior efficiency in evaporating water, compared to conventional misting or fogging systems.

Bottcher et al. (1992) investigated three separate systems for mechanical ventilation in commercial broiler houses during the spring and summer of 1988 in USA. The first system utilized buried earth tubes , plastic ventilation ducts , and hollow wall cavities as air inlets ; the second system used pressure- controlled slot inlets ; and the third system used evaporative cooling pads . They found that the evaporative cooling system provided greater reduction in temperature at bird level than tempering air using earth tubes. They found higher air velocities occurred at bird level in the houses using tunnel ventilation (0.8 – 1.8 m/s) than in the house with plastic ventilation ducts and hollow wall inlets (less than 0.5 m/s) , due to differences in ventilation rate and inlet design. Fan power requirements of the ventilation system for house utilized buried earth tubes resulted in consumption of approximately twice the electrical energy of houses used pressure controlled slot inlets and evaporative cooling pads. Also, the water use for house with evaporative cooling pad during the last two weeks of the summer 1988 flock was approximately three times that of house utilized buried earth tubes. House used evaporative cooling pad had a peak usage of 15,000 L/day, which is typical of houses with efficient evaporative cooling system pads operating continuously during hot weather.

Lei (1994) compared between forced ventilation and an evaporative cooling system in poultry housing. Effects of the system on measured environmental data, theoretical evaluation of system performance for the climate data of the Taiwan area and suitable environmental control methods for laying houses in this area are all considered. The effects of the forced ventilation and evaporative cooling system were evaluated on the basis of operational ranges of 32, 35 and 38 °C dry-bulb temperature and 40, 60 and 80% air relative humidity. Neither dry bulb temperature nor relative humidity reached the comfort zone for forced ventilation. The cooling effect of the evaporative cooling system was 10.2 °C, which is in the comfort zone, but relative humidity was too high. The suitable environmental control method for laying houses in Taiwan area is to decrease both dry bulb temperature and air relative humidity by removing moisture through regenerative drying material and reducing dry bulb temperature using the evaporative cooling system.

HeeChul et al. (1998) compared between the cooling effects of cooling pads and fogging systems in windowless laying hen houses during hot summer weather in the Korea Republic. Air velocity in the house with cooling pad was variable (0.96-3.27 m/s), being slowest at the inlet and fastest at the outlet. They found that the cooling pad decreased the inside air temperature by 5.4 ° C at the inlet site, 5.0 °C in the middle, and 2.8 °C at the outlet. Cooling effects of the fogging system were very low. Relative humidity was increased by 3.5% in the fogging system and 16.4% in the cooling pad system.

Arbel et al. (1999) used a fog system for cooling greenhouses based on spraying water (in the fog-size range) in very fine droplets without wetting the foliage. A mathematical model was developed to characterize the system, followed by the installation of an experimental system in a semi-commercial greenhouse. Ventilation and evaporation flow rates obtained from the experimental system were in accordance with those from the mathematical model. The fog system provided more uniform temperature and humidity conditions than a pad-and-fan system under similar conditions.

Colak (2002) conducted a study to determine the efficiency of three different cooling systems in greenhouses. He found that the lowest temperature was obtained using a fan and pad cooling system and the lowest relative humidity was obtained using a cool air ventilation system.

El-Soaly (2002) compared between two different buildings, the first building have an evaporative cooling system using palm fibers pad, while the second one depends on the natural ventilation. He found that the mean reduction of temperature were 7 °C and 2.6 °C at first and fifth of July for the first and the second buildings, respectively. Relative humidity in the cooled house was higher than that for the natural house by 13.4% as a mean value. He added that relative humidity is still in a suitable range.

Temperature and humidity gradients in fan ventilated greenhouses under two cooling modes were executed by Teitel et al. (2003) Using cooling system by fan ventilation only (mode I) and comparing it with the performance of a fan and pad system (mode II). These two modes may improve climate control, increase crop quality/production, reduce energy costs and improve the economics of greenhouse production. Experiments were carried out a greenhouse equipped

with a fan and pad systemn which pepper [*Capsicum* sp.]. Temperature and humidity gradients along the greenhouse and as functions of height were measured in each of the above-mentioned cooling modes, applying two ventilation rates in each mode. It is shown that in mode I the air temperature within the greenhouse was most of the days lower than the ambient temperature. In mode II the air temperature inside the greenhouse was lower than that obtained with mode I and was all the days lower than ambient temperature. In both modes of operation (I and II), the specific humidity within the greenhouse was always higher than ambient specific humidity, with higher values observed in mode II, as it was expected.

3. METHODOLOGY

Experimental work was conducted to investigate the potential of using some agricultural residues as pad materials in an evaporative cooling system. Technical specifications such as pad thickness and pad–face air velocity were considered in the study. Performance criteria were determined and used to judge the evaporative cooling system in reducing heat stress that can be occurred inside some agricultural structures.

3.1 Theoretical approach.

As described by numerous researchers (Hahn and Obsurn, 1969; Wiersma, 1969 and Hahn and Wiersma, 1972) evaporative cooling is an adiabatic process. Such a process lowers the dry-bulb temperature while increasing dew-point temperature of an air and water vapor mixture (Figure 3-1).

Accordingly, in a pad and fan evaporative cooling system, when non-saturated air (state point A in Figure 3-1) comes in contact with free moisture (assuming isolation from outside heat sources) mass and heat transfer take place. Because the vapor pressure of the free surface is higher than that of unsaturated air, water transfers owing to the differential. This transfer a change of state from liquid to vapor requiring heat of vaporization. This necessary heat comes from the sensible heat content of both air and water, resulting in temperature drop of both. As the temperature in the immediate vicinity of the drops and creating a temperature differential within the air vapor mixture, a transfer of heat occurs as the whole system seeks a thermodynamic balance.

As shown in Figure (3-1), the external air at state point A passing through the wetted pad and following wet-bulb line approaching the state point B if it is assumed that the air became completely saturated. Complete saturation does not occur and the resultant of air leaving the pad would only reach a state point like C. Then wet-bulb temperature remains constant, but dry-bulb temperature is reduced and dew point temperature, i.e. relative humidity, is increased. If is clear that the dry bulb temperature reduction, i.e. cooling potential, is limited to wet bulb depression. The effectiveness with which water is transferred to air is indication to saturation efficiency (SE) and the last is an important index to assess the performance of an evaporative cooling system. Referring to the SE definition mentioned before in section 2.4.1.5, so that on the psychometric chart in Figure (3-1), saturation efficiency (SE) can be computed as follows (Hahn and Obsurn, 1969; Wiersma, 1969 and Hahn and Wiersma, 1972):

$$SE = AC/AB \quad \ldots\ldots\ldots (1)$$

It can be calculated as a temperature difference ratio using the following formula (ASHRAE, 1992 and Wang, 1993):

$$SE = (T_{db} - T_C) / (T_{db} - T_{wb}) \quad \ldots\ldots\ldots (2)$$

Where:

SE = saturation efficiency (decimal)

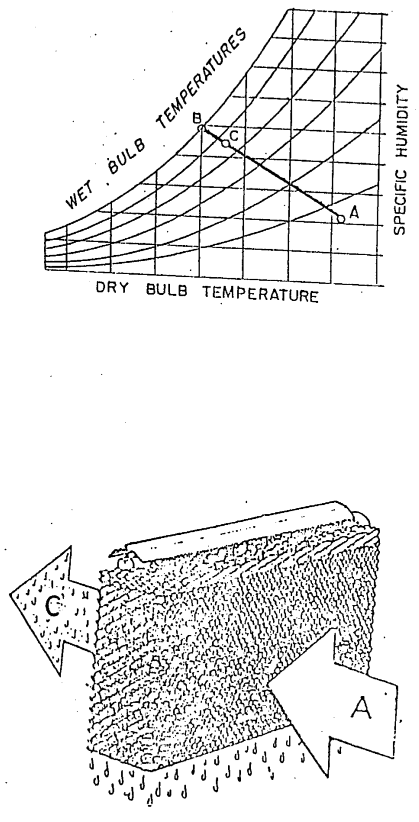


Figure (3-1): Explanation of the evaporative cooling process in pad and fan system (Abo-Zaher, 1992).

T_{db}= dry bulb temperature of air entering the cooling system (°C)
T_C = dry bulb temperature of air exiting the cooling system (°C)
T_{Wb}= wet bulb temperature of the outside air (°C)
$T_{db} - T_{Wb}$= wet bulb depression (°C)

Saturation efficiency(SE) can be estimated using humidity ratios as follows (Timmons and Baughman, 1983):

$SE = (W_C - W_O) / (W_S - W_O)$ .... (3)

Where:
W_C = humidity ratio of the cooled air ($kg/kg_{dry\ air}$)
W_O = humidity ratio of outside air ($kg/kg_{dry\ air}$)
W_S = humidity ratio of the outside air entering the cooling system if saturated to 100% relative humidity in a constant bulb process.

Evaporative cooling performance (ECP) is a common term for measuring cooling capacity and it can be calculated as a latent or sensible heat as follows (Wiresma, 1969):

As a latent heat:
$ECP = W_{ev} * h_{vap}$ .... (4)

Where:
ECP = evaporative cooler performance (kJ/h)
W_{ev} = water evaporated per hour (kg/h)
h_{vap} = specific heat of vaporization water (kJ/kg)

As sensible heat:
$ECP = (Q/ V_s) * \Delta T * Cp_{air}$ .... (5)

Where:
Q = ventilation rate (m^3/s)
V_s = specific volume of air (m^3/Kg)
ΔT= temperature reduction (i.e. cooling potential)
$= T_{db} - T_C$ (°C)
Cp_{air}= specific heat of air (1.005 kJ/kg.°C)

For measuring the absolute performance under base conditions that permit comparison, an improved measuring unit namely Unit ECP can be used and it expressed as (Wiresma, 1969):

$\text{Unit ECP} = ECP/ WBD_C$ .... (6)

Where:
Unit ECP = unit evaporative cooler performance (kJ/°C)
WBD_C = wet bulb depression of cooled air (°C)
$= Tdb - T_{wb}$ (°C)

Substituting from equation (5) in equation (6). Then
$\text{Unit ECP} = (Q/V_s) * (Tdb - T_C) * Cp_{air} / (T_{db} - T_{Wb})$.............. (7)

And by substituting from equation (2) in equation (7). Then
$\text{Unit ECP} = (Q/V_s) * SE * Cp_{air}$.......................... ...... (8)

3.2 Materials

All experimental work of the present study was conducted at Denosher village – El Mehalla EL Kobra- Gharbia Governorate. Data collected in two stages. First was conducted at the end of summer 2003 as a preliminary study to stand on some parameters and problems. Second stage which considered the main experiments was executed during summer 2004, specifically the operating of 23 June to 16 July in which all studied treatments were considered.

3.2.1 Experimental units

Two identical experimental rooms (experimental units) were constructed and built on the roof (third floor) of a domestic house. Each one having gross dimensions of 5.70 m long, 4.20 m wide, and 2.75 m high. The vertical walls were built using red bricks and the ceiling was covered by reinforced concrete of 12 cm thick. Each of the two rooms was prepared in such a way to represent an agricultural structure to be cooled (Figure 3-2). Longitudinal axis of the structure was oriented East-West direction. An open window of about 2.92 m long and 1.12 m high was made in the Northern wall at a height of 1.25 m from the floor surface to locate the pad of the cooling system. Two suction fans (single speed, direct driven, 50 cm diameter, and 2 $m^3/$ s calibrated discharge) were studied on the leeward side of the two rooms and the pads on the opposite sides toward the prevailing winds. On the Southern wall a wooden door (2.08 x1.12 m) was installed. The dry and wet-bulb temperatures inside the structure were measured at eighteen different points. Nine of them at a height of 0.25 m from the floor (lower level) and the reminder nine at a height of 1.92 m from the floor (Upper level) as illustrated in (Figure 3-3). Each experimental unit or was equipped with one type of the evaporative cooling pads, so that the two different pads can be operated at the same time.

3.2.2 Pad construction

Two agricultural residues namely rice straw (RS) and palm leaf fibers (PLF), commercially known as Kerina, were investigated as pad materials. Two vertical pads (2.92 x 1.12 m) were fabricated manually by filling the pad material (RS or PLF) in a wire net at a specific pad thickness to provide a constant pad density of about 32 kg/m^3. For instance, for (d) cm pad thickness, total pad volume is calculated. Then the required mass of the pad material density remain at the constant value of density 32 kg/m^3 was determined as follows:

$$\text{Required pad mass} = \text{Total pad volume} * \text{pad density}$$
$$= 2.92 * 1.12 * (d/100) * 32 \quad (kg) \ldots\ldots\ldots\ldots\ldots\ldots (9)$$

Where:

d = pad thickness (cm)

The resultant pad mass was uniformly distributed within the wire net to construct the pad having the specific thickness (d) cm. The prepared pad was constructed and fixed in a steel frame (steel angle 3 * 3 * 2 mm) equipped with steel screen and has the capability of matching with the pad thickness by means of screw bolts (Figures 3- 4 and 3-5). Water distribution system consists of a water tank, water control valve and perforated water pipe was used to uniformly distribute water over the pad. The water tank is filled by a connection with a domestic tap. Water passed from the tank through a water control valve to the

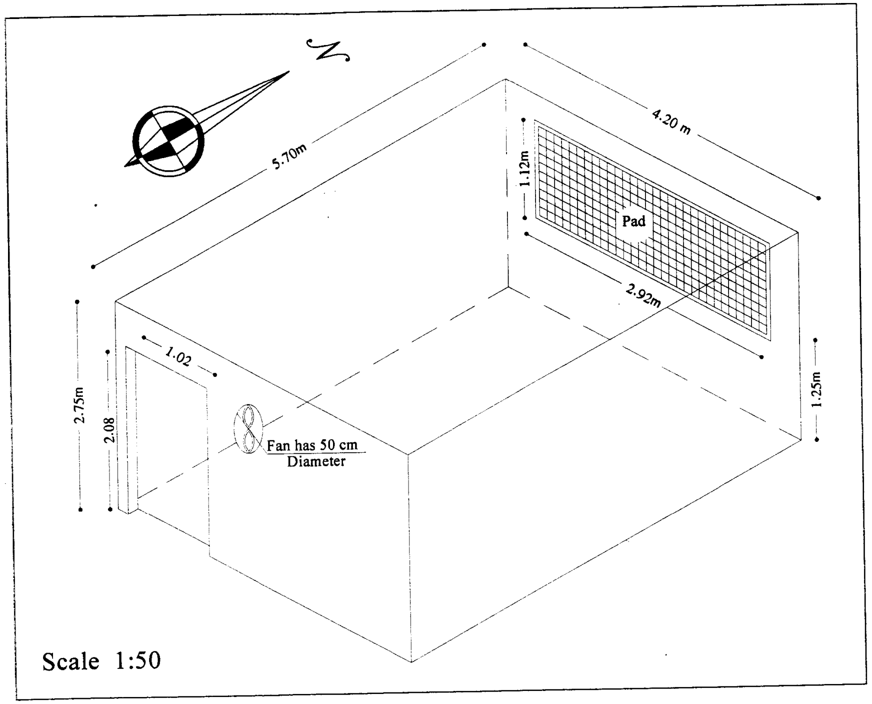


Figure 3-2: A schematic diagram the experimental of rooms representing an agricultural structure to be cooled

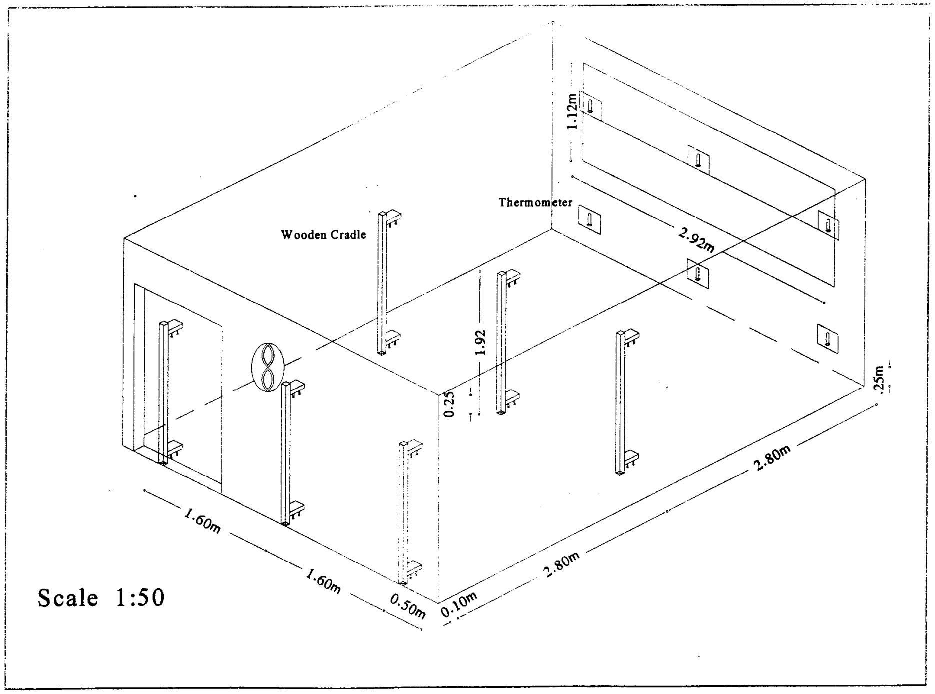


Figure 3-3: Diagram of thermometers arrangement inside the room

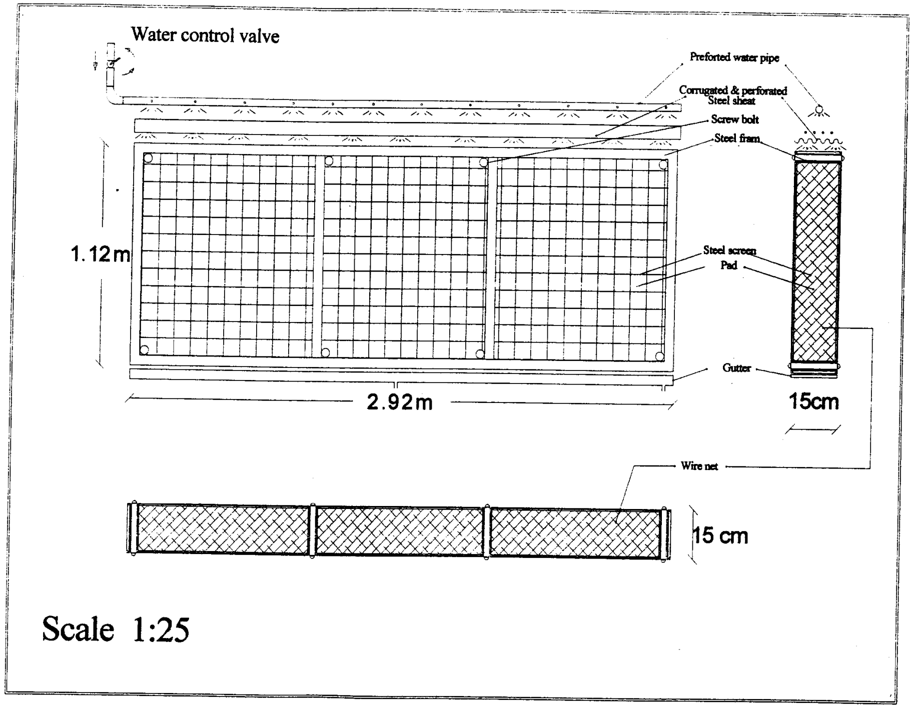


Figure 3-4: A schematic diagram of pad and water distribution system

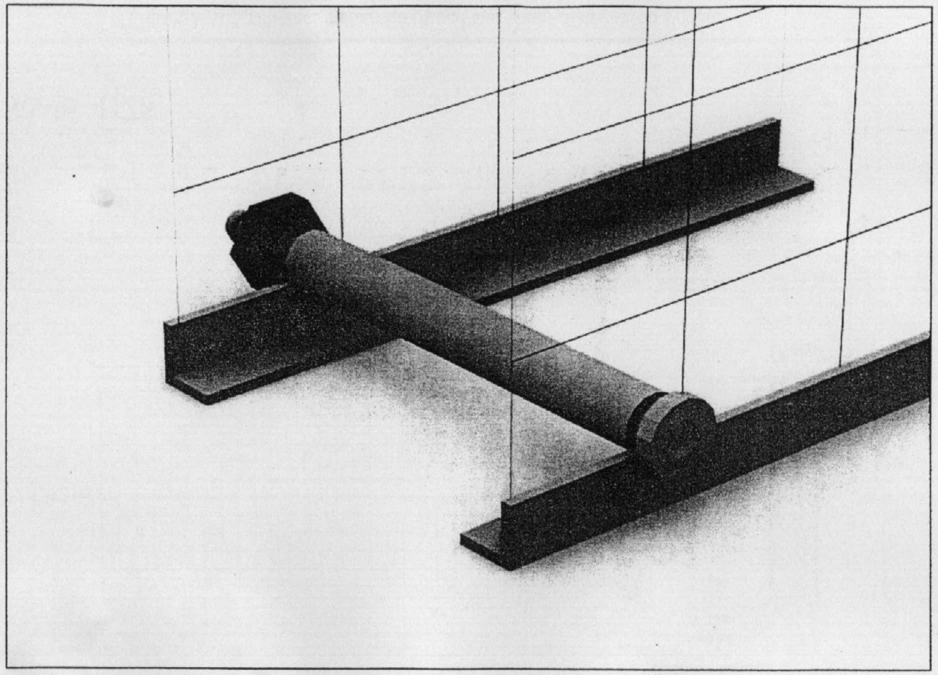


Figure 3-5: Screw bolts used to fix the steel frame to determined and fixed pad thickness

perforated pipe and falls upon the pad via the corrugated and perforated steel sheet. The last assures that the water can be reached over the whole pad thickness. Pad was continuously kept wetted by adjusting the water flow rate at a fixed value of about 0.0984 m^3/h by means of the water control valve. Excessive water was received in a gutter located below the pad and it was drained out. Main components of the water distribution system are shown in Figure (3- 4). Figure (3-6) indicates the water control valve. Figures (3-7) and (3-8) represent an internal view for both rice straw (RS) pad and palm leaf fibers (PLF) pad, respectively after installation inside the experimental structure. Some thermometers were used to measure dry and wet-bulb temperatures as shown in Figure (3-3). External view of both pads is illustrated in Figure (3-9).

3.2.3 Measurements

3.2.3.1 Temperature and relative humidity

Ordinary thermometers (have range of -20 to 50° C) were used to manually provide a number of 36 different points for determining environmental parameters, namely air temperatures and air relative humidities. Eighteen thermometers were used to measure dry and wet-bulb temperatures inside each experimental structure at different locations as illustrated in Figure (3-3). Two thermometers were placed outside the structures for measuring the outside dry and wet-bulb temperatures which were used to determine the air relative humidity. A digital hygrometer (model TFA) (Figure 3-10) was also placed outdoor for determining outside air relative humidity directly.

3.2.3.2 Pad face air velocity

A digital fan anemometer (model TFA) was employed to measure pad face air velocity and to calibrate the axial fan installed on the Southern wall as a part of the evaporative cooling system. It has the range of 0.20 to 30 m/s with the precision of 0.02 m/s, as shown in Figure (3-11).

3.3 Methods

3.3.1 Providing different pad face air velocities

Pad face air velocity were varied by changing the fan air flow rate by using an electric switch having a certain ability for varying the r.p.m of the fan. For each pad material and at a specific pad thickness, the fan was operated and the position of the electric switch was changed gradually while measuring the pad face air velocity at a predetermined nine points on the pad face area. Then the mean value of the measured nine values was calculated. The same procedure was reported at each pad thickness for both pad materials. Consequently, 12 different positions correspond 12 airflow rates at the fan outlet for each pad material could be determined. Finally, three pad face air velocities namely 0.3, 0.5 and 1.05 m/s have been determined and investigated at each of the four studied pad thickness for both pad materials. On the other hand, the axial fan was calibrated by measuring the air speed exiting the fan outlet at four different points within the circular cross sectional area of the fan using a horizontal cardboard cylinder. Air speed exiting the fan outlet was determined as the mean value of the four measured values. Then airflow rate at the fan outlet was calculated by

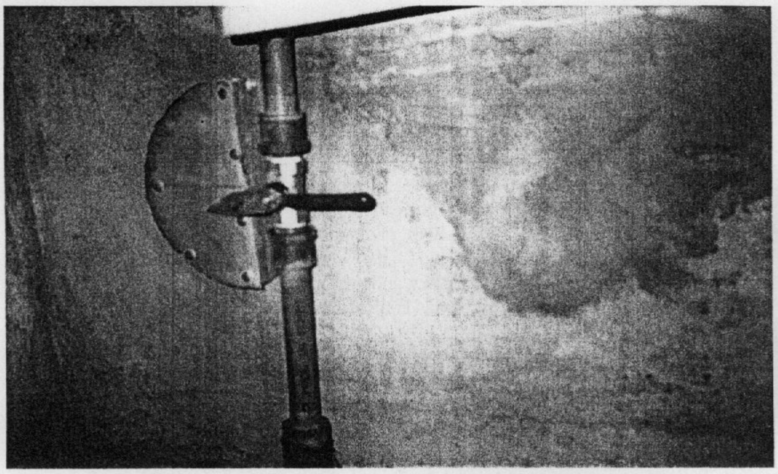


Figure 3-6 : Water control valve

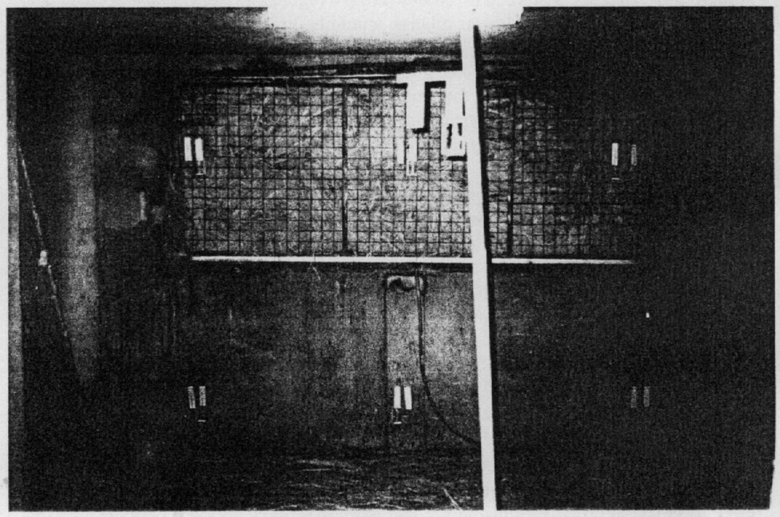


Figure 3 - 7 :Internal view for rice straw (RS) pad

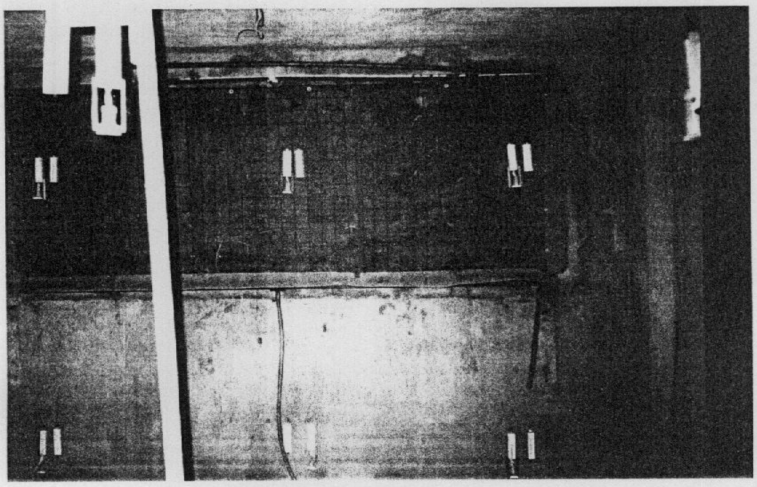


Figure 3-8 :Internal view for palm leaf fibers (PLF) pad

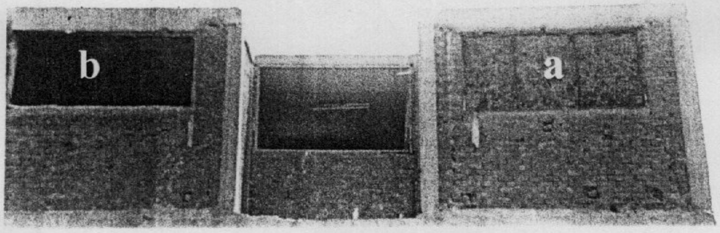


Figure 3-9 : External view for both rice straw (RS) pad (a) and palm leaf fibers (PLF) pad (b)

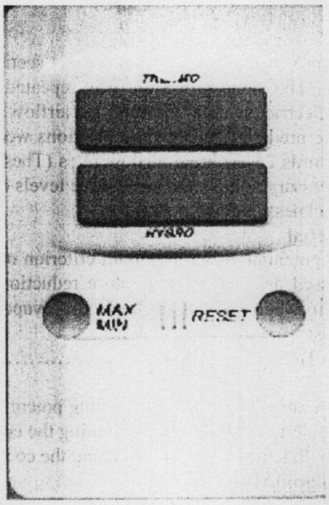


Figure 3-10: A digital hygrometer to measure temperature and relative humidity

Figure 3-11: A digital anemometer to measure air velocity

multiplying the mean value of air speed and cross sectional area of the fan (i.e. cardboard cylinder). The same procedure was repeated for each one of the 12 positions for the electric switch. Finally, 12 airflow rates at the fan outlet corresponding to the predetermined switch positions were provided for each pad material. In other words 24 different airflow rates (These values are indicated in Table 3-1) were necessary to have the same three levels of pad face air velocity at the different pad thickness and pad materials.

3.3.2 Cooling potential

Cooling potential, as an important criterion of the evaporative cooling system, was expressed as the air temperature reduction (ΔT). The temperature reduction of the air when passing through the evaporative cooling pad was calculated as follows:

$$\Delta T = T_{db} - T_C \qquad (10)$$

Where:

ΔT = temperature reduction, i.e. cooling potential (°C)

T_{db} = dry bulb temperature of air entering the cooling system (°C)

T_C = dry bulb temperature of air exiting the cooling system (°C)

3.3.3 Air relative humidity (RH)

Dry and wet bulb temperatures were measured and used as inputs to PLUS computer program (Albright, 1990) to obtain the air relative humidity data depending on psychometric chart relations. Regarding the outside air relative humidity it was recorded directly as the value that has read directly by the digital hygrometer.

3.3.4 Ratio of temperature reduction to airflow rate ($\Delta T/Q$)

To judge the evaporative cooling system in reducing dry bulb temperature under different air flow rates and to permit comparisons on the base of air flow rate unity. Ratio of temperature reduction to air flow rate was the new suggested criterion or parameter to cope with the variation in airflow rate among all treatments. It was computed using the following formula:

$$\Delta T/Q = (T_{db} - T_C) / Q \qquad (11)$$

Where:

$\Delta T/Q$ = ratio of temperature reduction to airflow rate (°C.s/m^3)

$\Delta T = T_{db} - T_C$ (°C)

Q = airflow rate (m^3/s)

3.3.5 Temperature humidity index (THI)

Temperature humidity index (THI) is considered as one of the most important criterions to describe the comfort level from the view point of heat stress. The importance of such parameter takes more attention in the systems contributes in raising air relative humidity or moisture content. Its determination depends on the species confined within the structure or in other words on the comfort (thermo-neutral) zone of the biological system. Temperature humidity index (THI) for dairy cows, laying hens and broilers were calculated to relative the potential of the evaporative cooling system in reducing heat stress for these selected species (eg. biological systems) that can be housed in an evaporatively

Table 3-1: Applied airflow rates for both pad materials at different pad thickness and pad face air velocities.

Pad thickness (cm)	Pad face air velocity (m/s)	(Q) m^3/s	
		RS	PLF
3	0.3	0.46	1.18
	0.5	1.08	1.32
	1.05	1.47	1.43
6	0.3	0.73	0.95
	0.5	1.61	1.47
	1.05	1.68	1.56
10	0.3	0.92	1.23
	0.5	1.55	1.44
	1.05	1.23	1.47
15	0.3	1.52	1.11
	0.5	1.70	1.53
	1.05	1.76	1.61

cooled structure. THI was calculated first assuming no biological system is confined inside the structure (i.e. under no load). Predicted THI under loading for broilers was estimated as well. The whole procedure can be illustrated in the following sections.

3.3.5.1 Dairy cows

Temperature humidity index (THI) for dairy cows was calculated using the following equation (Mayer et al., 1999)

$(THI)_{dcf} = T_{db} + 0.36Tdp + 41.2$..(12)

$(THI)_{dcf}$ = temperature humidity index for dairy cows (°F)

T_{db} = dry bulb air entering the cooling system (°C)

T_{dp} = dew point temperature (°C)

In the present study $(THI)_{dc}$ was computed in °C depending on equation (12) as follows:

$(THI)_{dc} = [(T_{db} + 0.36Tdp + 41.2) - 32] * (5/9)$.................... (13)

Where:

$(THI)_{dc}$= temperature humidity index for dairy cows (°C)

Two values, namely 25.6°C (78°F) and 21.1°C (70°F) were assumed to be the threshold range (Mayer et al 1999 and St-Pierre et al 2003) beyond which, the calculated $(THI)_{dc}$ could be judged under evaporative cooling conditions.

3.3.5.2 Laying hens

Temperature humidity index for laying hens $(THI)_L$ was estimated as follows (Gates et al, 1995):

$(THI)_L = 0.6\ T_{db} + T_{wb}$.. (14)

Where:

$(THI)_L$ = temperature humidity index for layers (°C)

T_{db} = dry bulb temperature (°C)

T_{wb} = wet bulb temperature (°C)

Maximum value for the range of $(THI)_L$ was calculated using equation (14) in which T_{db} is 28°C and air relative humidity is 70% that represent the maxing value for comfort zone for laying hens (Alam, 1986). Corresponding (THI) was 26°C. However, Gates et al (1995) determined the value of 28°C as a threshold temperature humidity index for laying hens. The minimum value for the range of $(THI)_L$ was assumed to be 21.1°C (70°F) (St-Pierre et al 2003).

3.3.5.3 Broilers

Temperature humidity index for broilers $(THI)_B$ was computed using the following equation (Tao and Xin, 2003) :

$(THI)_B = 0.85\ Tdb + 0.15\ T_{wb}$... (15)

Where:

$(THI)_B$ = temperature humidity index for broilers (°C)

Threshold range of $(THI)_B$ was calculated depending on the comfort (thermo-neutral) zone for broilers in which dry bulb temperature ranged from 21°C to 25°C and air relative humidity ranged from 60% to 70% (Alam, 1986 and El-Hadidi 1989). This is the comfort zone for broilers at marketing size, (i.e. The age of 4 weeks and thereafter). Accordingly, the corresponding wet-bulb temperatures (T_{wb}) was firstly determined from psychometrics, then the threshold values of $(THI)_B$ was found to be ranged from 20.10°C to 24.50°C using equation (15).

3.3.5.4 Temperature humidity index under simulated broiler housing conditions $(THI)_{BS}$

Considering broilers as the biological system that is to be housed inside the experimental structure, temperature humidity index was calculated under optimal conditions for broilers. This was accomplished through a simulation procedure in which internal air dry bulb temperature was predicted under the assumption with loading of 10 birds/ m^2 at marketing size. In addition, internal air moisture content in terms of humidity ratio predicted internal air wet-bulb temperature would be expected via psychometrics. Therefore, predicted temperature humidity index under simulated broiler housing conditions $(THI)_{BS}$ can be estimated using equation (15) in which T_{db} and T_{wb} were replaced with $(T_{db})_{BS}$ and $(T_{wb})_{BS}$, respectively as follows (Gates et al, 1995):

$(T_{db})_{BS} = t_C + [(Q_s * n_b * m_b)/ (m_a * Cp)]$ .... (16)

Where:

$(T_{db})_{BS}$ = dry bulb temperature of the internal air under simulated broiler housing conditions (°C)
t_C = dry bulb temperature of the internal cooled air without loading (°C)
Qs = net sensible heat production per unit mass of broilers (kJ/h .kg)
n_b = number of broilers (240 birds)
m_b = airflow rate (kg_a/h)
Cp_{air} = Specific heat of air (1.005 kJ/ kg_a °C)

Sensible heat production per unit mass of broiler was calculated as follows (AbouZaher, 1998):

$Q_s = 21.86332 + 1.65082 * BA - 51.90657 * BMKG - 123.15205 * (BMKG/BA) + 0.4019 * (BMKG * BA)$.. (17)

Where:

BA = bird age (d)
BMKG = bird mass (kg)

Bird mass was calculated using the following equation (Flood et al., 1992): $BMG = -533.1 + 61.69\ BA$ ....$29 \leq BA \leq 49$ $_{day}$........... (18)

Where:

BMG = the bird mass (g).

In the present study, broilers age of 42 days were considered as the age corresponding the marketing size. However, this age represent an age urgent need especially in hot weather conditions.

Predicted humidity ratio of the internal air inside the structure under the same simulated conditions for broilers was computed as follows (Gates et al 1995):

$$W_{Bs} = w_C + [(Q_L * n_b * m_b) / (m_a * h_{vap})] \text{ .... } (19)$$

Where:

W_{BS} = humidity ratio of the internal air under simulated broiler housing conditions (kg/kg)

w_C = humidity ratio of the internal cooled air without loading (kg/kg).

Q_L = net latent heat production per unit mass of broilers (kJ/h.kg)

h_{vap} = latent heat of vaporization (2430 kJ/kg) at 30°C.

Latent heat production per unit mass of broiler was calculated as follows (AbouZaher, 1998):

$$Q_L = 62.7799 - 2.7603 * BA + 39.00587 * BMKG - 216.05812 * (BMKG/BA) - 0.04135 * (BMKG*BA) \text{ .... } (20)$$

Both $(T_{db})_{BS}$ and W_{BS} were used as two psychometric parameters input to the computer program PLUS to get wet bulb temperature of the internal air under simulated broiler housing conditions $(T_{wb})_{BS}$.

Again, using $(T_{db})_{BS}$ and $(T_w)_{BS}$ as inputs in equation (15) results in calculating temperature humidity index under simulated broiler housing conditions $(THI)_{Bs}$. The last was used as an another criterion to judge the investigated evaporative cooling system.

3.3.6 Saturation efficiency (SE)

Saturation efficiency (SE) as an important criterion to judge the evaporative cooling system was determined for each treatment throughout the operating day time period. It was defined before in section 2.4.1.5 and it was calculated using equation (2).

3.3.7 Unit evaporative cooler performance (Unit ECP)

Unit ECP is another criterion to measure the absolute performance of evaporative cooling system. It was calculated using equation (5).

4. RESULTS AND DISCUSSION

The present study was carried out to investigate the performance criteria of rice straw (RS) and palm leaf fibers (PLF) as pad materials of a pad and fan evaporative cooling system. Pad thickness and pad face air velocity were taken into consideration. Reducing heat stress in terms of temperature reduction (T and ΔT) and temperature humidity index (THI), saturation efficiency (SE) of the pad and unit of the evaporative cooler performance (Unit ECP) were the main criteria to judge both pad materials. An attention was given to explore the performance criteria throughout summer daytime operating periods to illustrate the performance under different weather conditions. The effectiveness of evaporative cooling system in providing and maintaining the microclimate at desired level of an experimental structure was demonstrated as well.

4.1 Reducing heat stress

Cooled air temperature and relative humidity, absolute temperature reduction, ratio of temperature reduction to airflow rate and temperature-humidity index (THI) were determined. These parameters were involved the performance criteria to judge the two pad materials in reducing heat stress under different investigated technical specifications.

4.1.1 Temperature (T) and relative humidity of air (RH)

Figures from (4-1) to (4-12) illustrated the recorded air temperatures (T) and relative humidities (RH) throughout the operating period for both rice straw (RS) and palm leaf fibers(PLF) pad materials at different pad thickness and pad face air velocity. Generally inside air temperature was lower than the outside one and as it was expected the inside air relative humidity was higher than that outside. The reduction in air temperature which observed inside the structure was due to the use of evaporative cooling system in which the air stream to be cooled comes directly in contact with the wetted medium (pad material), and evaporates some water by exploiting heat for evaporation from surroundings and carrying some moisture through an adiabatic process.

Cooled air temperature and relative humidity have an almost same behavior of outside conditions but associated with very small amplitude comparing with that of the outside air. Such a result reveals better environmental control level that can be achieved using the sudied system. Actually the main cause to have this advantage is the fact that the cooling potential of a direct evaporative cooling system increases with the decrease in air relative humidity of the air entering the cooler, such a fact leads to the above mentioned narrow amplitude for both upper and lower levels of air temperatures and air relative humidity.

Thermal lag has been found in some treatments such as at 6 cm pad thickness and for all pad face air velocities. Since the maximum outside temperature occurred around hour 14 during daytime, whereas the maximum temperature of the cooled air for both upper and lower levels at the mentioned pad thickness occurred around hour 17 during daytime. It seems that the outside

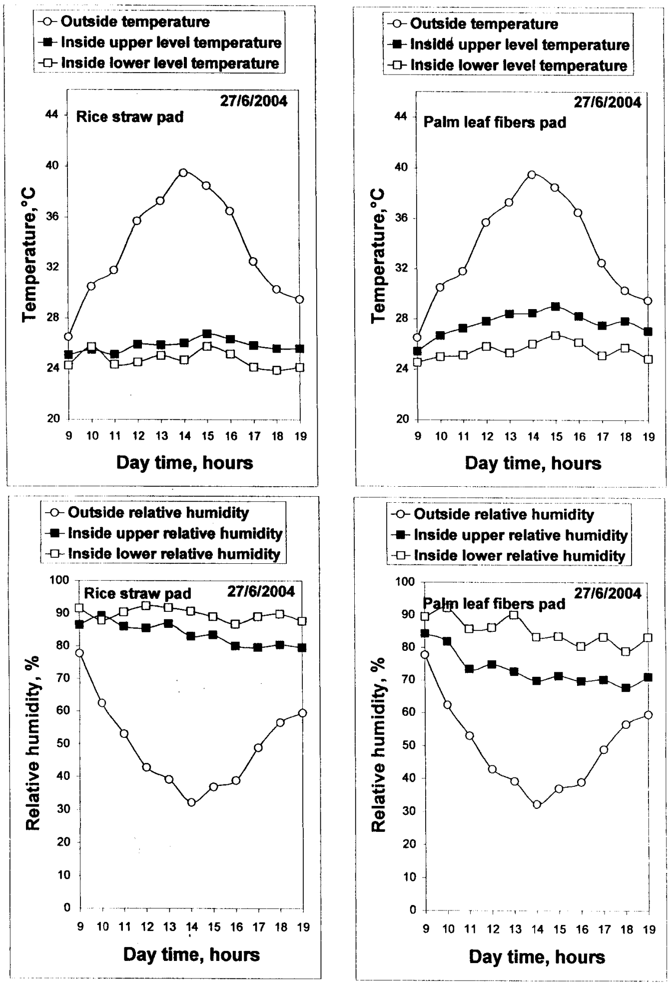


Figure (4-1): Air temperatures (T) and air relative humidities (RH) throughout the operating period for both pad materials at 3 cm pad thickness and 0.3 m/s pad face air velocity.

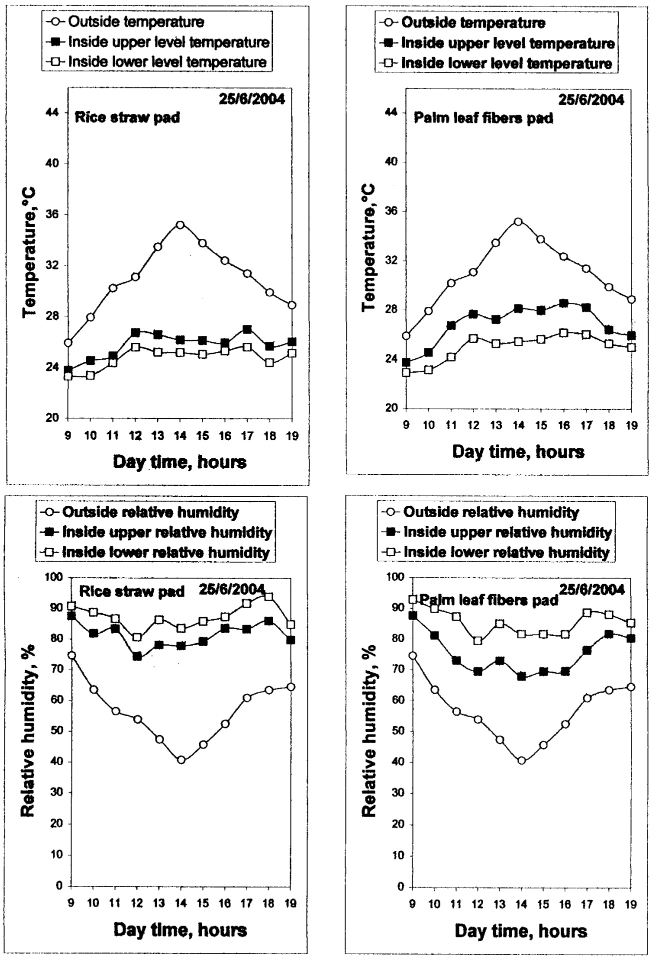


Figure (4-2): Air temperatures (T) and air relative humidities (RH) throughout the operating period for both pad materials at 3 cm pad thickness and 0.5 m/s pad face air velocity.

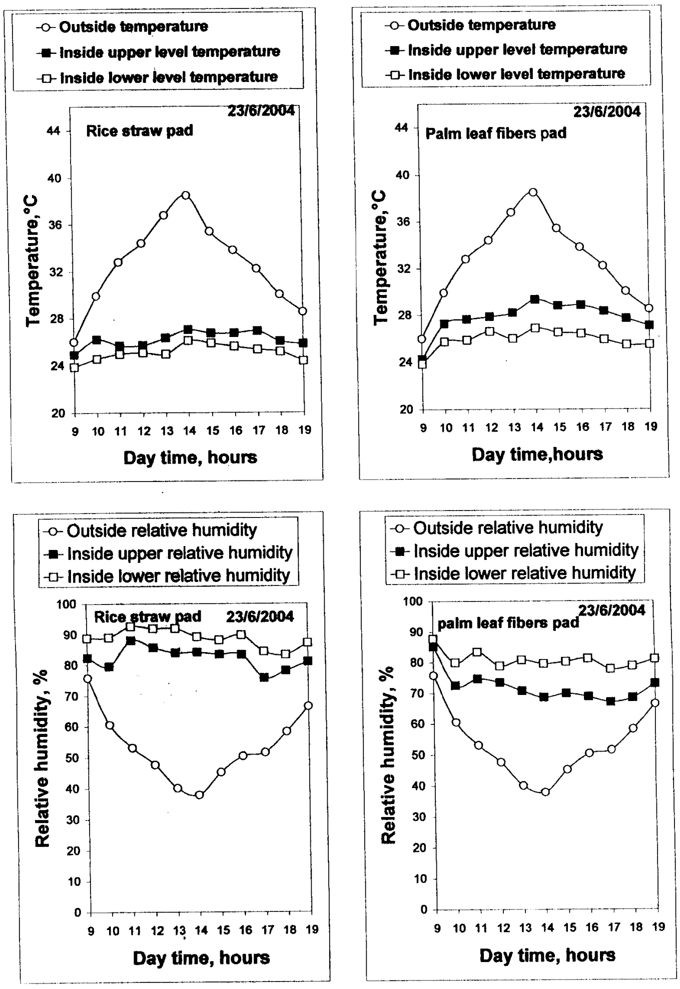


Figure (4-3): Air temperatures (T) and air relative humidities (RH) throughout the operating period for both pad materials at 3 cm pad thickness and 1.05 m/s pad face air velocity.

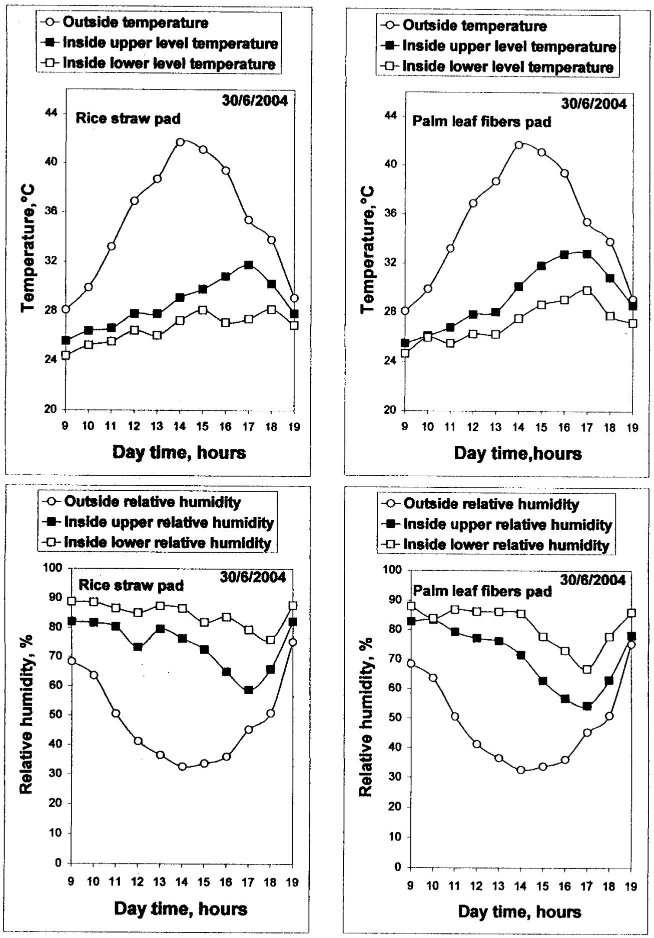


Figure (4-4): Air temperatures (T) and air relative humidities (RH) throughout the operating period for both pad materials at 6 cm pad thickness and 0.3 m/s pad face air velocity.

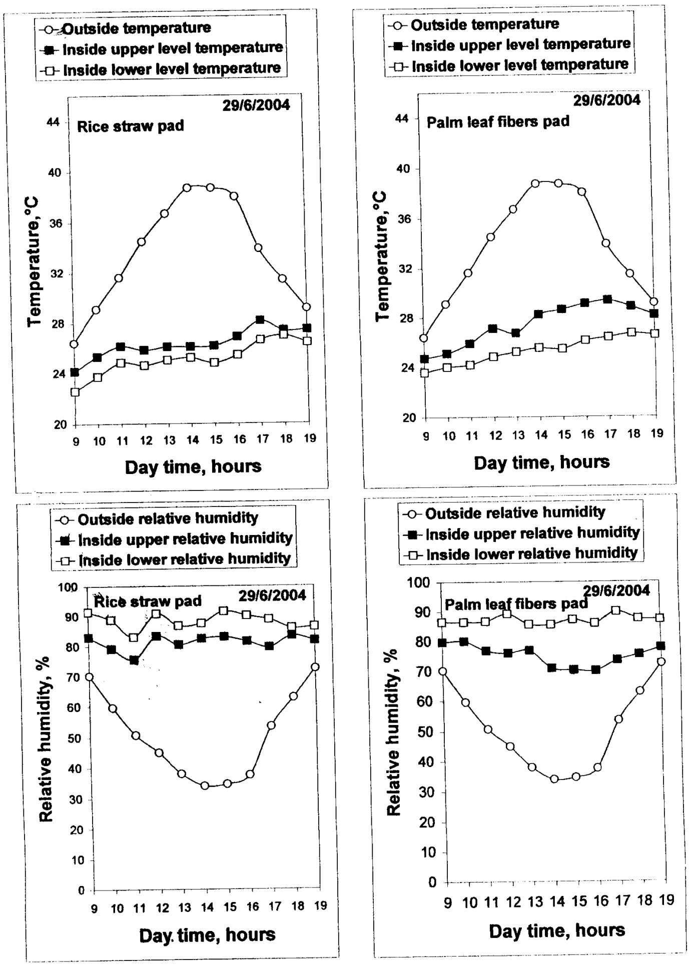


Figure (4-5): Air temperatures (T) and air relative humidities (RH) throughout the operating period for both pad materials at 6 cm pad thickness and 0.5 m/s pad face air velocity.

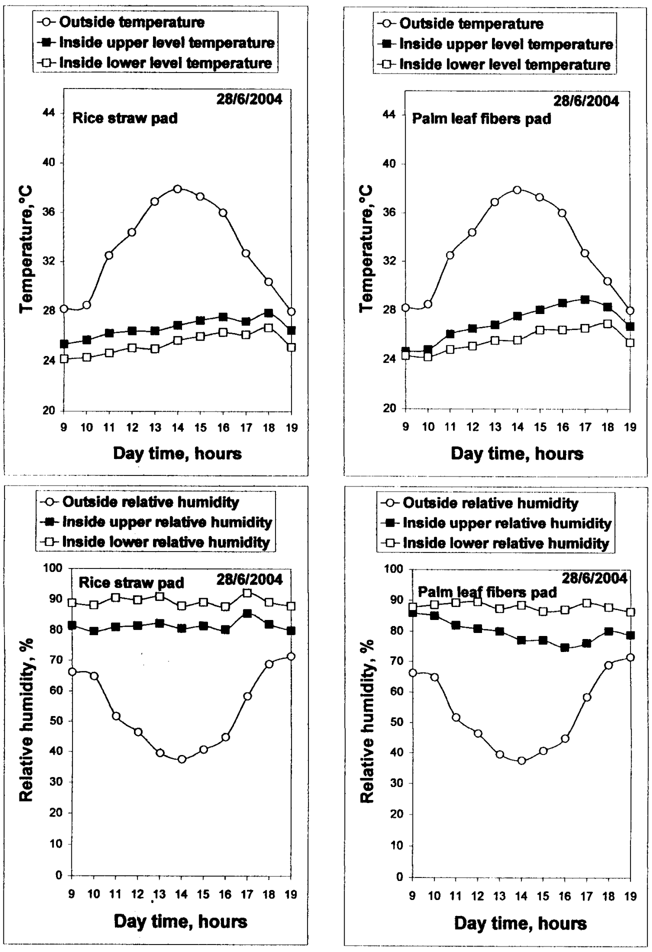


Figure (4-6): Air temperatures (T) and air relative humidities (RH) throughout the operating period for both pad materials at 6 cm pad thickness and 1.05 m/s pad face air velocity.

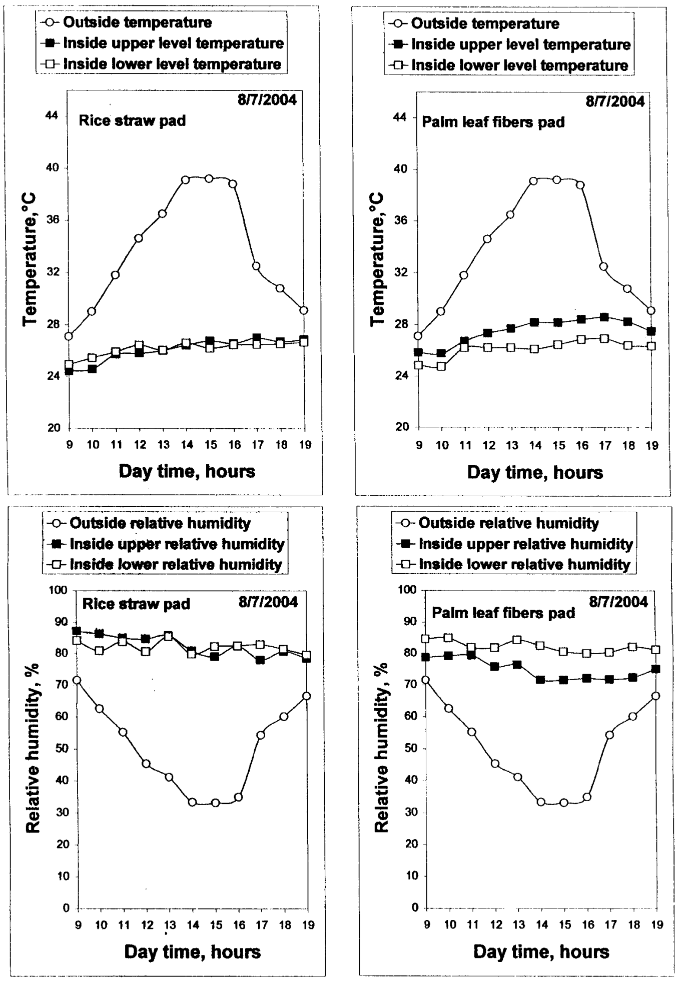


Figure (4-7): Air temperatures (T) and air relative humidities (RH) throughout the operating period for both pad materials at 10 cm pad thickness and 0.3 m/s pad face air velocity.

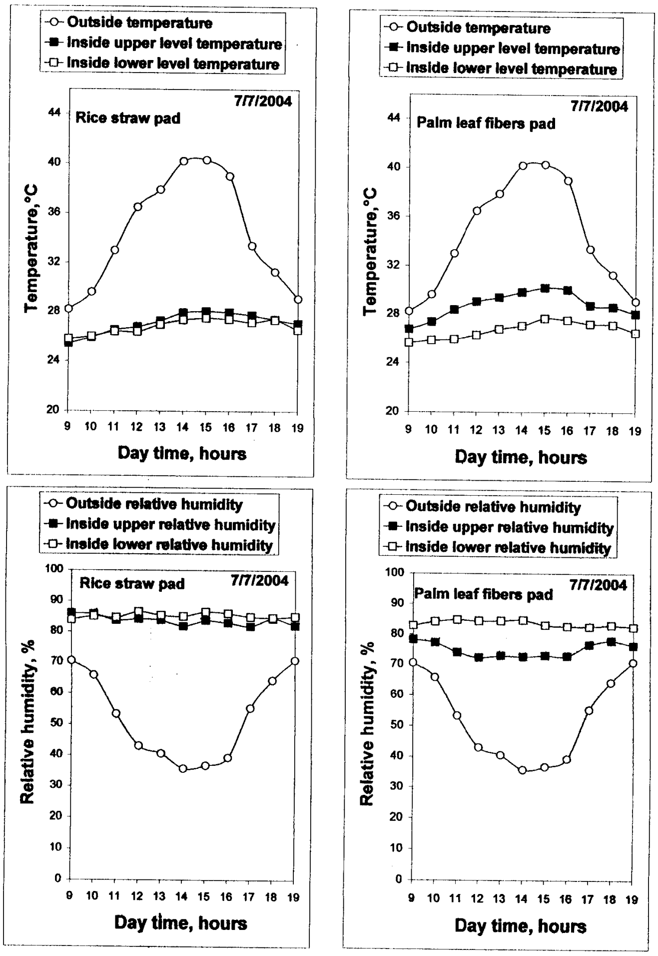


Figure (4-8): Air temperatures (T) and air relative humidities (RH) throughout the operating period for both pad materials at 10 cm pad thickness and 0.5 m/s pad face air velocity.

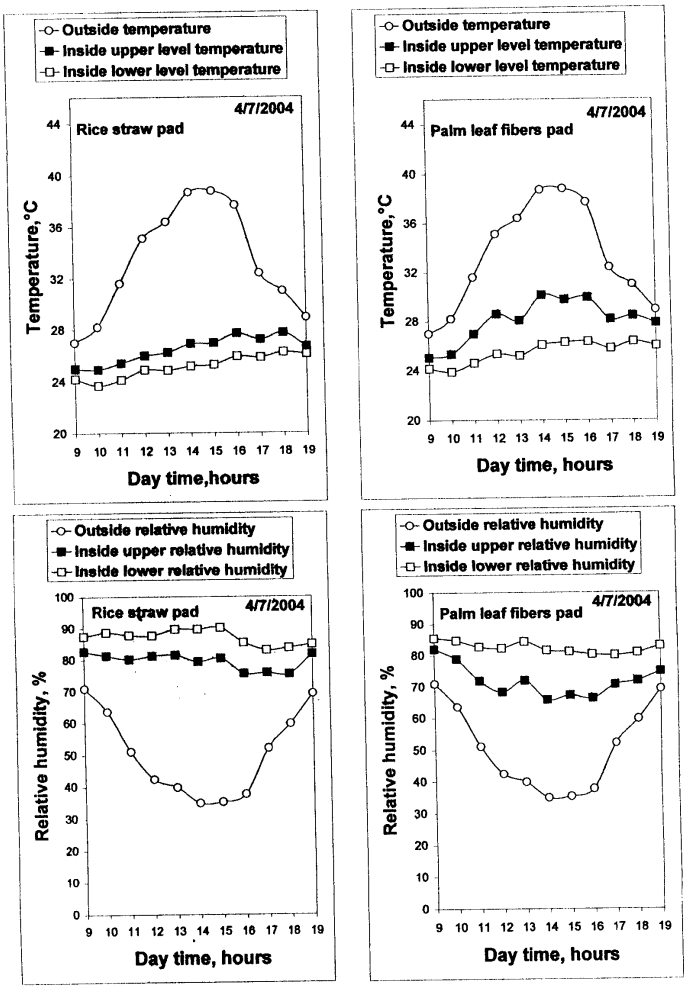


Figure (4-9): Air temperatures (T) and air relative humidities (RH) throughout the operating period for both pad materials at 10 cm pad thickness and 1.05 m/s pad face air velocity.

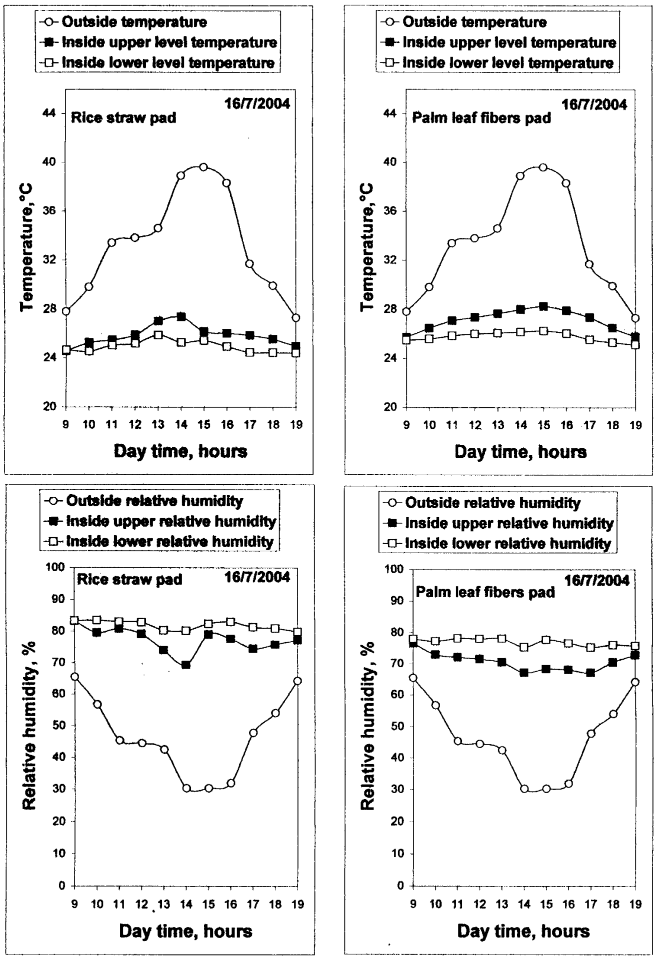


Figure (4-10): Air temperatures (T) and air relative humidities (RH) throughout the operating period for both pad materials at 15 cm pad thickness and 0.3 m/s pad face air velocity.

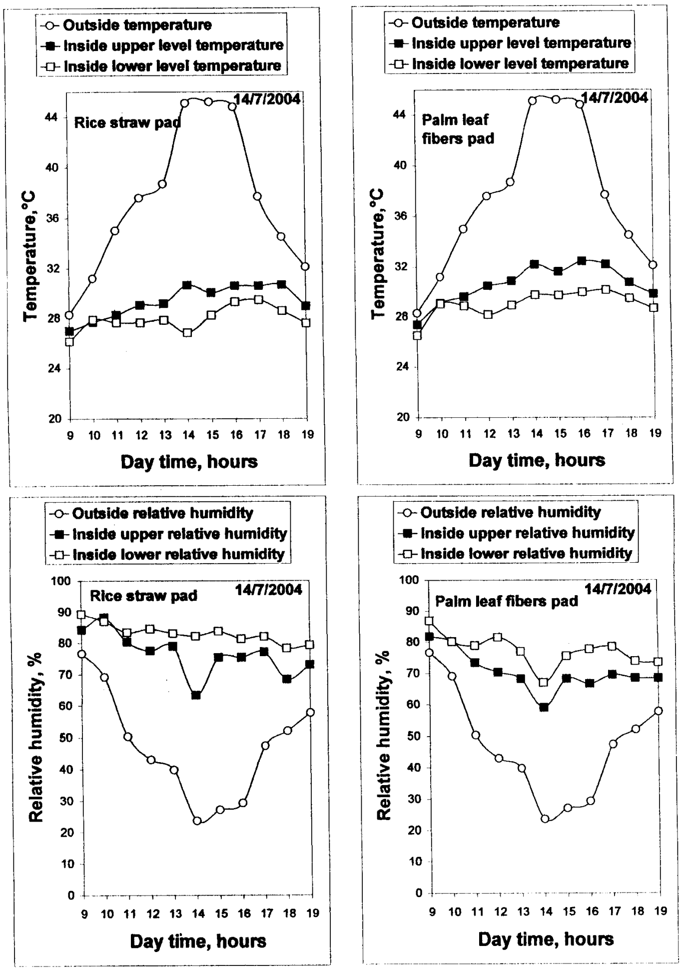


Figure (4-11): Air temperatures (T) and air relative humidities (RH) throughout the operating period for both pad materials at 15 cm pad thickness and 0.5 m/s pad face air velocity.

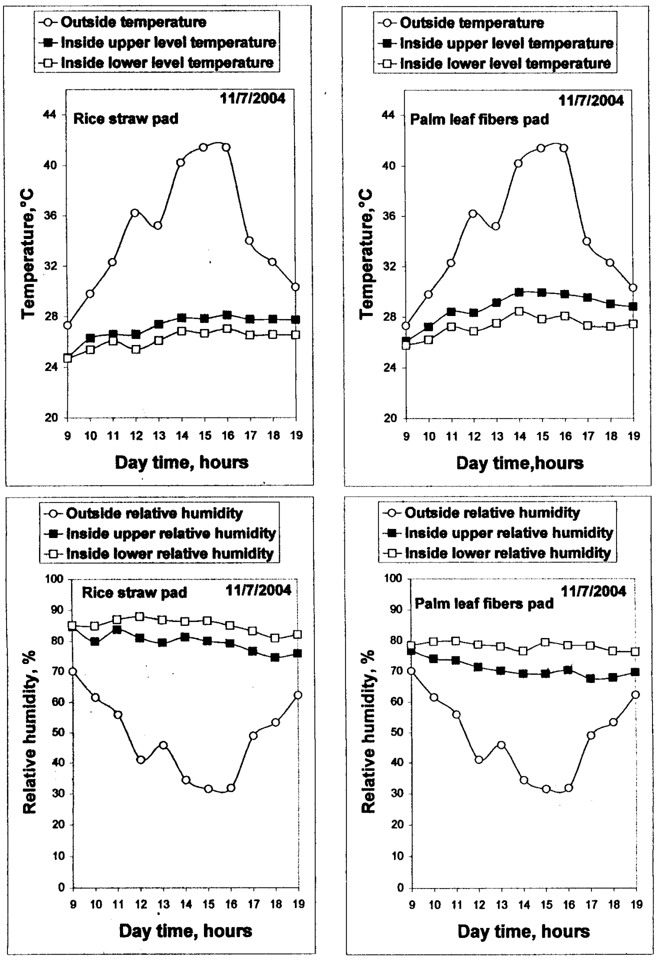


Figure (4-12): Air temperatures (T) and air relative humidities (RH) throughout the operating period for both pad materials at 15 cm pad thickness and 1.05 m/s pad face air velocity.

air relative humidity at these treatments at 6 cm pad thickness tend to rise suddenly around hour 17 of daytime, as illustrated in Figures (4-4) to (4-6).

Generally, as shown in Figures (4-1) to (4-12), rice straw pad material was found to be better in cooling process than that for palm leaf fibers. This may refer to the physical properties of the fibers for each pad materials and the variation in airflow resistance that may affect on the capability of the air to evaporate and carry water. Fluctuation in air temperature and air relative humidity was lower for rice straw than that for palm leaf fibers; such a result indicates better environmental control that can be achieved using rice straw pad material.

As it was expected, air temperatures and air relative humidities at the upper level were higher and lower, respectively, than that at lower level. This is due to the fact that, the lower the air temperature is the higher air density, so that the warm air moves upward and the cool air moves downward. A unique exception was found at 10 cm rice straw pad material and 0.3 m/s pad face air velocity where the air temperature and air relative humidity were almost the same for both upper and lower levels (Figure 4-7).

Table (4-1) illustrates the temperature average and standard deviation (SD) during the operating period for both materials at various pad thickness and pad face air velocity. The minimum mean (mean of upper and lower level values) value of about 25.24°C (SD=0.57°C) was found when using RS pad material at 3 cm thick and 0.3 m/s pad face air velocity. For PLF pad material the minimum mean value of about 25.94°C (SD=1.33°C) was found at 3 cm pad thick and 0.5 m/s pad face air velocity. Corresponding outside temperatures were 33.51 °C (SD= 4.21°C) and 30.93°C (SD= 2.75°C) for RS and PLF pad materials, respectively.

Lower standard deviation values indicated in Table (4-1) and belonging to RS pad material confirms the previously mentioned result of the better environmental control level in cooled air temperature using RS pad material.

Table (4-2) shows the air relative humidity average and standard deviation (SD) during the operating period for both pad materials at various pad thickness and pad face air velocity. The minimum mean(mean upper and lower levels values) value of about 79.47% (SD= 6.10%) was found when using RS pad material at 6 cm thick and 0.3 m/s pad face air velocity. For PLF pad material the minimum value of about 73.79% (SD=2.02%) was found when using 15 cm pad thick and 0.3 m/s pad face air velocity. Corresponding outside air relative humidities were 48.48 %(SD=14.77%) and 46.65% (SD=12.63) for RS and PLF pad materials, respectively.

Lower standard deviation values indicated in Table (4-2) and belonging to RS pad material confirms the previous result of the better environmental control level in cooled air temperature using RS pad material. Only one exception was found at 15 cm pad thick at all pad face air velocities, since SD

for RS was higher than that for PLF. However, the acceptable range of air relative humidity for most of biological systems is a wider one compared with

Table 4-1: Average air temperature and standard deviation (SD) during the operating period for both pad materials at various pad thickness and pad face air velocity.

Pad thickness (cm)	Pad face air velocity (m/s)	Outside temp, °C	SD, °C	R S						P L F					
				Upper Level	S D	Lower Level	S D	Mean		Upper Level	S D	Lower Level	S D	Mean	
								temp	SD					temp	SD
3	0.3	33.51	4.21	25.78	0.49	24.71	0.64	25.24	0.57	27.61	1.00	25.48	0.65	26.54	0.83
	0.5	30.93	2.75	25.75	0.99	24.77	0.82	25.26	0.90	26.83	1.56	24.99	1.10	25.91	1.33
	1.05	32.57	3.73	26.22	0.62	25.07	0.66	25.64	0.64	27.75	1.35	25.88	0.82	26.81	1.09
6	0.3	35.21	4.81	28.52	1.97	26.61	1.17	27.57	1.57	29.18	2.63	27.16	1.60	28.17	2.12
	0.5	33.46	4.27	26.31	1.10	25.15	1.25	25.73	1.18	27.43	1.49	25.32	0.98	26.37	1.24
	1.05	32.98	3.79	26.67	0.78	25.37	0.84	26.02	0.81	26.99	1.45	25.56	0.92	26.27	1.19
10	0.3	33.50	4.40	26.08	0.89	26.16	0.55	26.12	0.72	27.49	0.97	26.13	0.71	26.81	0.84
	0.5	34.41	4.56	27.10	0.87	26.82	0.61	26.96	0.74	28.76	1.09	27.07	0.74	27.91	0.92
	1.05	33.25	4.30	26.42	1.01	25.11	0.86	25.76	0.94	28.01	1.70	25.86	0.93	26.94	1.31
15	0.3	33.19	4.36	25.83	0.82	24.94	0.48	25.39	0.65	27.09	0.87	25.76	0.38	26.43	0.63
	0.5	37.29	5.83	29.33	1.30	27.94	0.97	28.63	1.13	30.58	1.54	29.01	1.54	29.80	1.26
	1.05	34.58	4.82	27.13	1.01	26.14	0.72	26.63	0.86	28.74	1.20	27.26	0.78	28	0.99

Table 4-2: Average air relative humidity (RH) and standard deviation (SD) during the operating for both pad materials at various pad thickness and pad face air velocity.

Pad thickness (cm)	Pad face air velocity (m/s)	Outside RH (%)	S D (%)	Pad material											
				RS						PLF					
				Upper Level	S D	Lower Level	S D	Mean		Upper Level	S D	Lower Level	S D	Mean	
								RH	SD					RH	SD
3	0.3	49.77	13.57	83.57	3.24	89.69	1.68	86.63	2.46	73.28	5.23	85.08	4.09	79.18	4.66
	0.5	56.70	8.76	81.37	3.90	87.31	3.80	84.34	3.85	75.40	6.50	85.62	4.16	80.51	5.33
	1.05	53.35	10.34	82.26	3.49	88.72	2.98	85.49	3.23	72.04	4.96	80.84	2.73	76.44	3.85
6	0.3	48.48	14.77	74.26	8.08	84.68	4.12	79.47	6.10	71.31	10.49	81.61	6.89	76.46	8.69
	0.5	50.75	14.09	81.17	2.37	88.27	2.67	84.72	2.52	75.19	3.58	87.05	1.40	81.12	2.49
	1.05	53.65	12.71	81.38	1.58	89.26	1.43	85.32	1.51	79.73	3.54	87.97	1.13	83.85	2.33
10	0.3	50.77	13.85	82.56	3.36	82.13	1.85	82.34	2.60	74.92	3.20	82.23	1.73	78.57	2.46
	0.5	52.25	13.89	83.65	1.43	85.24	0.80	84.45	1.12	74.88	2.36	83.54	0.97	79.21	1.67
	1.05	50.62	13.59	79.58	2.68	87.14	2.52	83.36	2.60	71.76	5.12	82.35	1.87	89.72	3.49
15	0.3	46.65	12.63	77.25	3.81	81.87	1.39	79.56	2.60	70.67	2.87	76.91	1.16	73.79	2.02
	0.5	46.48	16.73	76.50	6.86	83.09	3.15	79.79	5.01	70.36	5.77	77.39	4.49	73.87	5.13
	1.05	48.76	13.15	79.65	3.12	85.09	2.20	82.37	2.66	70.87	2.86	78.24	1.27	74.55	2.06

that for temperature. So that, a proper controlling of temperature and relative humidity of air at an acceptable level can be achieved using RS as a pad material of an evaporative cooling system.

Figures (4-13) and (4-14) show the relationship between outside temperature and cooled air temperature as affected by outside air relative humidity for all investigated treatments of RS and PLF pad materials, respectively. The same point symbol represents the cooled air temperature recorded at a specific outside temperature at the same specific range of outside air relative humidity. Each line is the regression line for points recorded at the same specific outside air relative humidity as well. In general, since all recorded cooled air temperatures lye below no cooling line, cooling effectiveness achieved by both pad materials is very clear. In addition, increasing outside air relative humidity at the outside same temperature increases cooled air temperature (i.e. depresses the desired cooling process) cooling depression, in terms of increasing cooled air temperature, rate increases when outside air relative humidity tends to the higher range as well. This can be noticed in increasing the slope of the regression lines by increasing the range of outside air relative humidity. It should be mentioned that extrapolations beyond the range of outside temperature that corresponding a range of outside air relative humidity is not recommended.

4.1.2 Cooling potential (Temperature reduction)

Temperature reduction (ΔT) was determined to describe the cooling potential of the investigated system. Figures (4-15) to (4-17) show temperature reduction (ΔT) throughout the operating period for RS and PLF pad materials at various technical pad specifications. Values of ΔT at upper and lower level within the experiment structure were indicated. Recorded outside temperature (T_{db}) was indicated for comparison as well. Comparing between RS and PLF pad materials revealed that ΔT values for RS were higher than that for PLF. Such a result reflects more cooling potential when using RS pad material. This may be due to the variation in fibers physical properties (absorption rate of water)of both materials in such a way that permit air to carry more water when passing through RS media. Airflow resistance of both pad materials may play another rule in this phenomenon.

Maximum temperature reduction of about 18.24 °C, 15.48°C was found at the lower level when using 15 cm RS and PLF pad thickness and 0.5 m/s pad face air velocity (Figure 4-16). This maximum ΔT occurred with outside temperature of about 45.1 °C, 45.2 °C at hours 14 and 15 afternoon, for RS and PLF pad materials, respectively.

In general, as it is illustrated in the three Figures the higher the outside temperature corresponds lower relative humidity, is the higher is the temperature reduction. As a conclusion it is very clear that the pad material has a great effect on ΔT, (i.e. on cooling potential) this in agreement with the data published by Sharaf (1994), Dzivama et al (1999) and Liao and Chiu (2002). Therefore and based on the cooling potential the rice straw as a pad material in a pad-fan evaporative cooling system can satisfactory be used. A similar recommendation

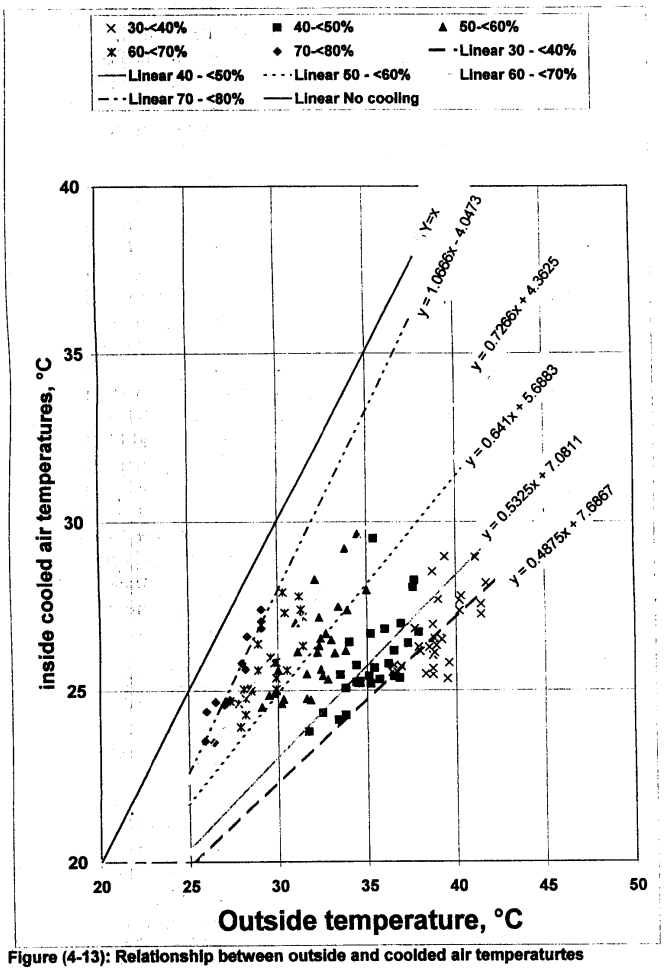


Figure (4-13): Relationship between outside and coolded air temperaturtes as affected by outside air realative humidity for all investigated treatments of rice straw pad materials

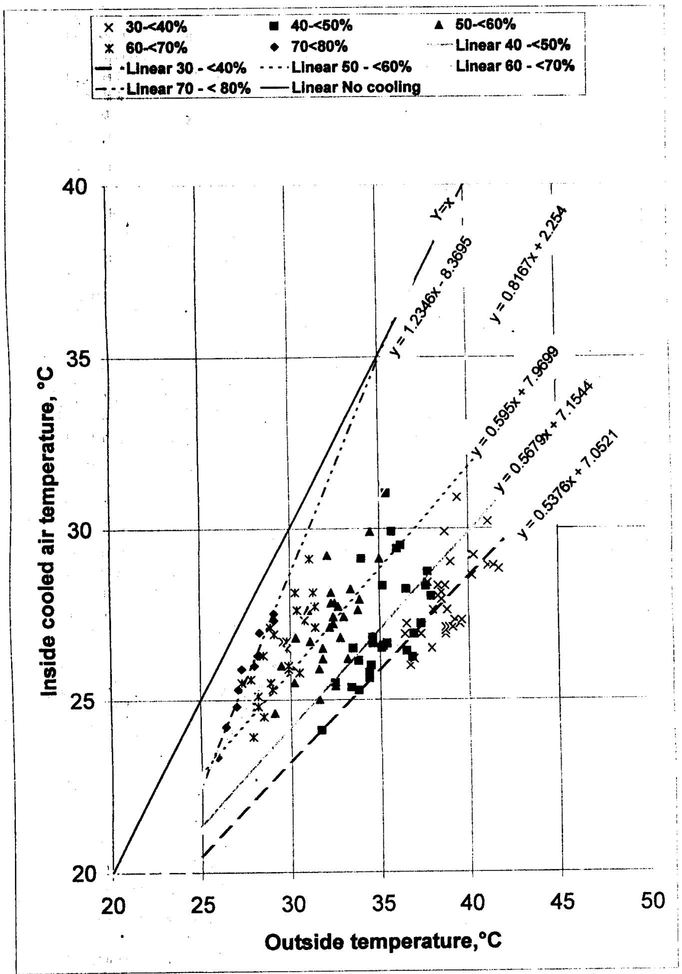


Figure (4-14): Relationship between outside and coolded air temperaturtes as affected by outside air realative humidity for all investigated treatments of palm leaf fibers pad material

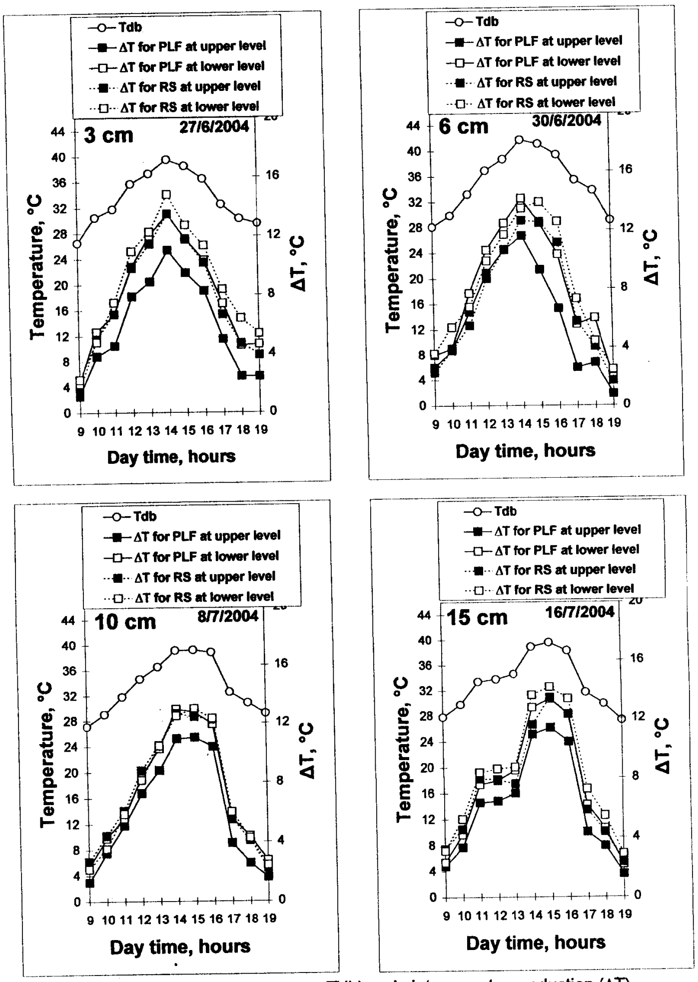


Figure (4-15): Outside air temperature (Tdb) and air temperature reduction (ΔT) throughout the operating period for rice straw (RS) and palm leaf fibers (PLF) at different pad thickness and 0.3m/s pad face air

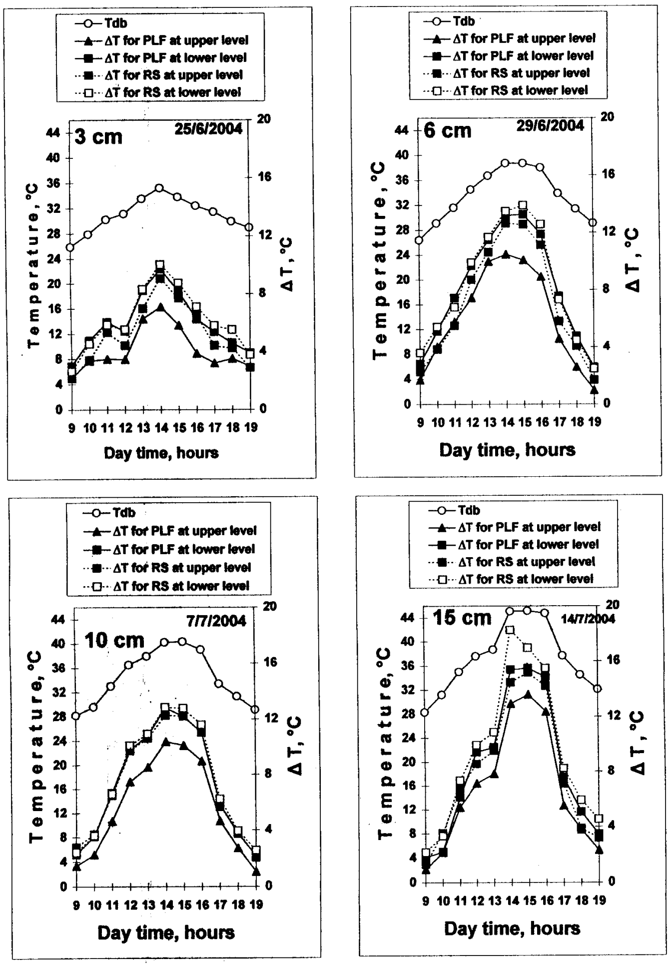


Figure (4-16): Outside air temperature (Tdb) and air temperature reduction (ΔT) throughout the operating period for rice straw (RS) and palm leaf fibers (PLF) at different pad thickness and 0.5 m/s pad face air velocity.

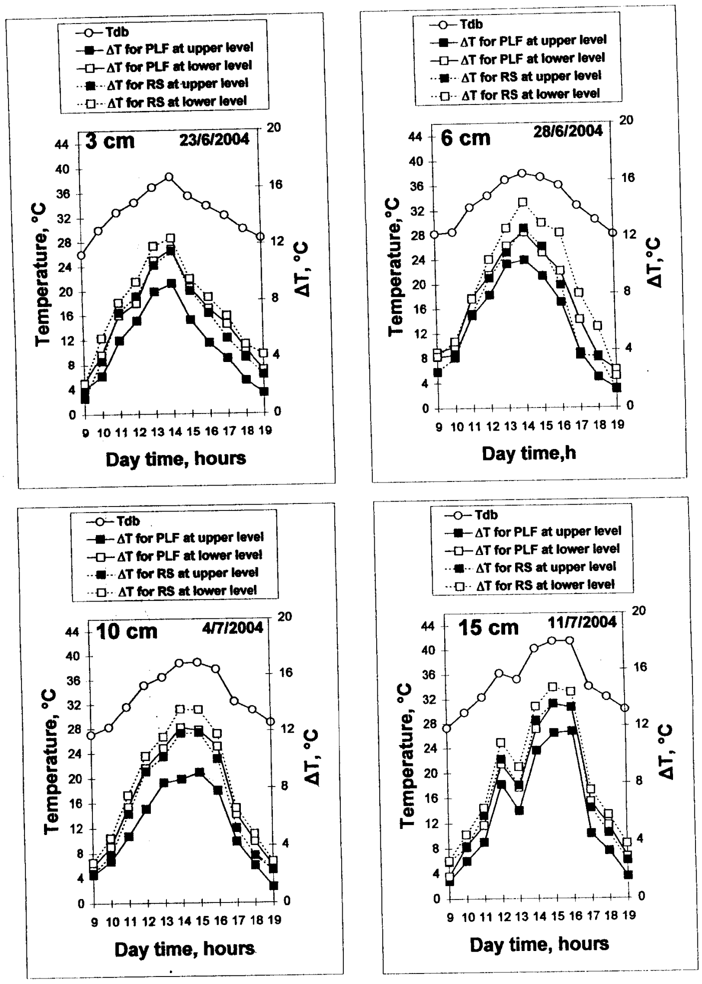


Figure (4-17): Outside air temperature (Tdb) and air temperature reduction (ΔT) throughout the operating period for rice straw (RS) and palm leaf fibers (PLF) at different pad thickness and 1.05 m/s pad face air has been advised by Abdel-Rahman (2000) when indicated the importance of using wheat long straw as a pad material. However, both rice and wheat straw are belonging to Family Graminae which have a stem nature close to each others.

Table (4-3) indicates temperature reduction (ΔT) average and standard deviation (SD) during the operating period for both pad materials at various pad thickness and pad face air velocities. Mean values of upper and lower levels prove that RS is better than PLF in reducing air temperature. Mean values of about 8.66 °C and 7.50 °C were found as a maximum air temperature reduction for RS and PLF, respectively. RS pad material, with 3 cm pad thick the maximum mean of ΔT was found to be 8.27 °C and occurred at 0.3 m/s pad face air velocity. Increasing pad face air velocity to 0.5 and 1.05 m/s results in mean values of upper and lower levels of about 5.76 and 6.93 °C, respectively. At 6 cm and 10 cm pad thickness changing pad face air velocity from 0.3 to 1.05 m/s seems to have a very small effect on ΔT values of upper and lower levels. The same trend was found for PLF pad material. For RS pad material and based on the mean values of ΔT for upper and lower levels, it can be observed that the higher the thickness is the higher pad face air velocity is required to get more temperature reduction. It seems that there was not fixed trend between pad thickness and pad face air velocity. This is due to the contradiction effect of the higher kinetic energy corresponds to the highest air velocity and the contact time between air and water films on the pad fibers. Increasing pad face air velocity passing through the pad media leads to break the films between pad fibers and consequently increasing the water vapour transfer to the air stream which in turn increases temperature reduction, (Pieper and Wiresma, 1971). On the other hand, increasing pad face air velocity decreases the contact time between air and water films. This provided a good chance for air to carry on more water depending on its thermal specification. Another cause makes difficulty of distinguishing between pad thicknesses and pad face air velocity to point out unique recommended values is the variation in outside temperature and relative humidity of air for each treatment as well as the variation in airflow rate.

4.1.3 Ratio of temperature reduction to airflow rate (ΔT/Q)

Because of the variation in airflow rate applied in each treatment as previously mentioned in section 3.3.1, it was considered to determine the temperature reduction per unity of airflow rate. This procedure may provide a better criterion for comparison between the studied factors. The ratio of temperature reduction to airflow rate (ΔT/Q) throughout the operating period for both pad materials at different pad thickness and pad face air velocities are illustrated in Figures (4-18), (4-19) and (4-20). Each value of (ΔT/Q) in the Figures represents an average of 18 values corresponding to the 18 measuring points inside the experimental structure. In general, the unity of airflow rate has more capability to reduce temperature in case of RS pad material. The only exception was found at 15 cm pad thick and 0.3 m/s pad face air velocity since (ΔT/Q) was higher for PLF than that for RS as revealed Figure (4-18). Again in all treatments, the highest (ΔT/Q) occurred at hours 14 and 15 during the

Table 4-3: Average air temperature reduction (ΔT) average and standard deviation (SD) during the operating for both pad materials at various pad thickness and pad face air velocities.

Pad thickness (cm)	Pad face air velocity (m/s)	Outside temp, °C	SD, °C	R S						P L F					
				Upper Level (ΔT, °C)	S D	Lower Level (ΔT, °C)	S D	Mean		Upper Level (ΔT, °C)	S D	Lower Level (ΔT, °C)	S D	Mean	
								ΔT	SD					ΔT	SD
3	0.3	33.51	4.21	7.73	3.82	8.80	3.91	8.27	3.87	5.90	3.34	8.03	3.70	6.96	3.52
	0.5	30.93	2.75	5.18	1.84	6.16	1.89	5.67	1.86	4.09	1.56	5.93	2.04	5.01	1.80
	1.05	32.57	3.73	6.36	3.29	7.50	3.21	6.93	3.25	4.83	2.69	6.69	3.05	5.76	2.87
6	0.3	35.21	4.81	6.69	4.00	8.59	4.23	7.64	4.11	6.03	3.72	8.05	4.18	7.04	3.95
	0.5	33.46	4.27	7.15	4.11	8.32	4.17	7.73	4.14	6.04	3.54	8.15	4.00	7.09	3.77
	1.05	32.98	3.79	6.32	3.51	7.61	3.52	6.96	3.51	6.00	3.25	7.42	3.50	5.67	3.37
10	0.3	33.50	4.40	7.42	4.03	7.34	4.15	7.38	4.09	6.01	3.81	7.37	3.93	6.69	3.87
	0.5	34.41	4.56	7.31	3.95	7.59	4.01	7.45	3.98	5.65	3.55	7.34	4.10	6.50	3.82
	1.05	33.25	4.30	6.84	3.85	8.15	4.14	7.49	4.00	5.24	2.68	7.39	3.93	6.32	3.07
15	0.3	33.19	4.36	7.36	3.36	8.25	3.63	7.80	3.50	6.10	3.53	7.43	4.02	6.54	3.78
	0.5	37.29	5.83	7.96	4.92	9.36	5.53	8.66	5.23	6.71	4.53	8.28	5.23	7.50	4.88
	1.05	34.58	4.82	7.45	4.20	8.45	4.36	7.95	4.28	5.84	3.89	7.32	4.20	6.58	4.05

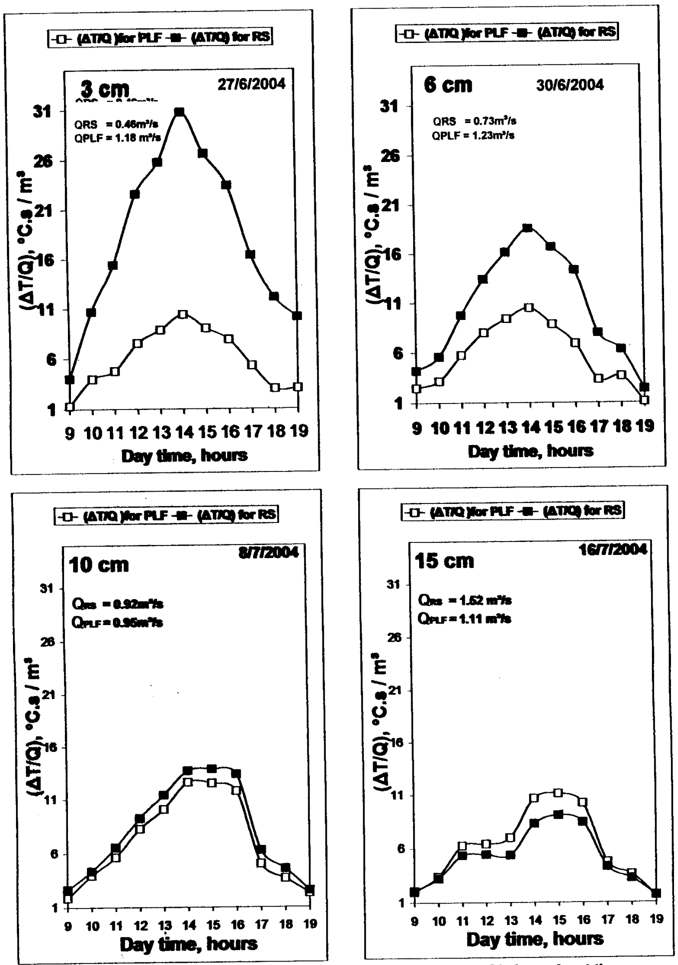


Figure (4-18): Ratio of temperature reduction to airflowrate (ΔT/Q) throughout the operating period for rice straw (RS) and palm leaf fibers (PLF) at different pad thickness and 0.3 m/s pad face air velocity.

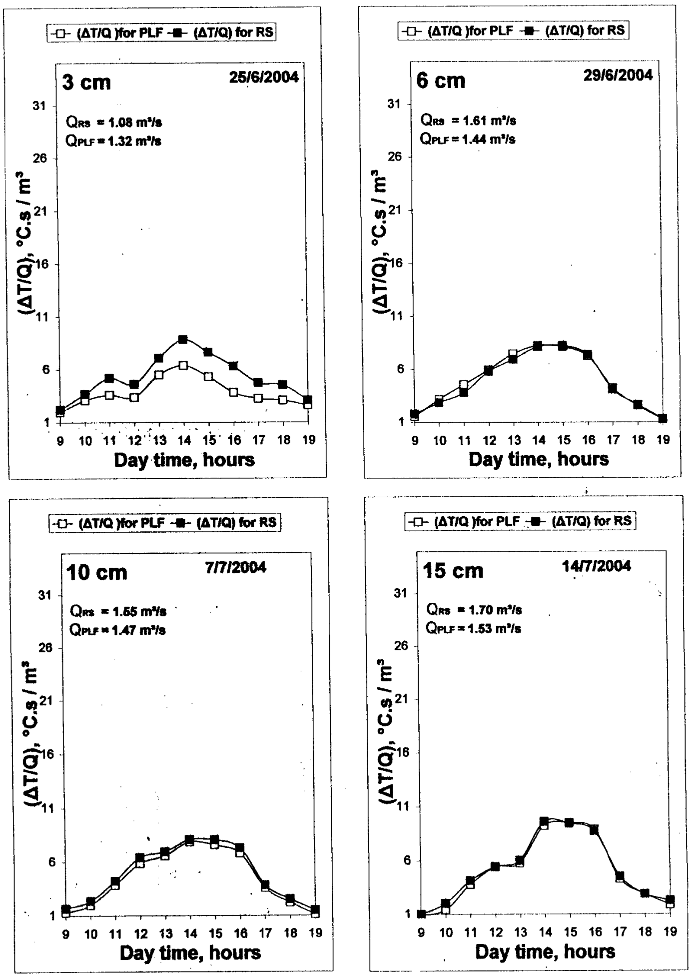


Figure (4-19): Ratio of temperature reduction to airflowrate (ΔT/Q) throughout the operating period for rice straw (RS) and palm leaf fibers (PLF) at different pad thickness and 0.5 m/s pad face air velocity.

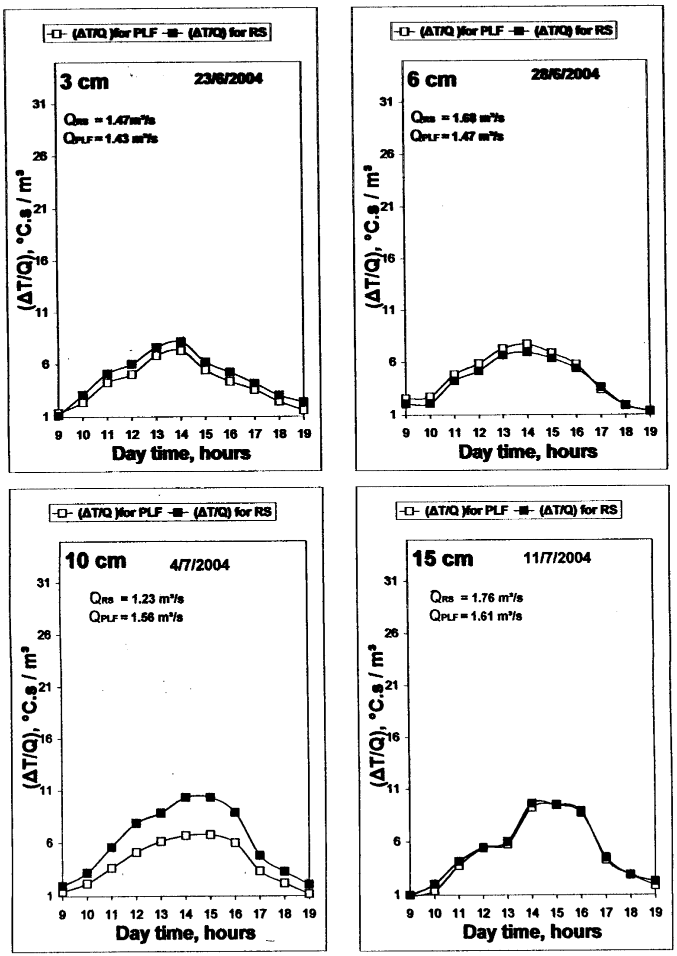


Figure (4-20): Ratio of temperature reduction to airflowrate (ΔT/Q) throughout the operating period for rice straw (RS) and palm leaf fibers (PLF) at different pad thickness and 1.05 m/s pad face air velocity.

operating period when the highest absolute temperature reduction occurred. At 0.3 m/s pad face air velocity the ($\Delta T/Q$) ratio decreases by increasing pad thickness as shown in Figure (4-18),. As mentioned before the variation in airflow resistance due to the variation in pad thickness plays an important rule in this phenomenon. However, at both 0.5 and 1.05 m/s pad face air velocity the ($\Delta T/Q$) ratio failed to have the same specific relationship with the pad thickness as illustrated in Figures (4-19) and (4-20). The same result can be observed in Table (4-4).

4.1.4 Temperature–humidity index (THI)

Temperature humidity index (THI) as an important criterion to judge such evaporative cooling system in reducing heat stress was determined for both pad materials and different pad specifications during the operating period. THI values of non-cooled (outside) air and cooled (inside) air for dairy cows, laying hens and broiler, as biological systems differ in their sensibility to heat stress were also calculated. THI of cooled air was computed under the assumption of no biological systems are loaded within the experimental structure. This procedure permits to judge the effect of the investigated system itself on the cooling process from the view point of heat stress since such direct evaporative cooling system evolved in increasing air relative humidity while decreasing dry-bulb temperature.

$(THI)_{Bs}$ of cooled air under simulated broiler housing conditions (under loading) was determined as well. This procedure permits to investigate the effect of the studied evaporative cooling systems on the simulation of the internal conditions in such a biological system not only on the specifications of the cooled air itself. However, threshold THI range belongs to each one of the three biological systems according to the specific comfort. This THI range was used as another datum in THI comparisons although the upper limit of the range is the important one. Each value of THI indicated in the following sections represents an average of 18 values corresponding to the 18 measuring points inside the experimental structure.

THI belongs to dairy cows:

Figures (4-21), (4-22) and (4-23) show the temperature humidity index $(THI)_{dc}$ for dairy cows when using both pad materials at different pad specifications. THI values of cooled air during the operating period for all treatments are lower than that for outside $(THIo)_{dc}$ or non-cooled air. Moderate fluctuations in $(THI)_{dc}$ of the cooled air were observed at all treatments. Generally, both RS and PLF pad material have an opportunity to keep $(THI)_{dc}$ of cooled air under the maximum threshold one. However, they are failed to keep this advantage in case of 15 cm thick of pad material and 0.5 m/s pad face air velocity as shown in Figure (4-22). This may refer to the high value of $(THIo)_{dc}$ at non-cooled air. Few hours during which, from hour 14 to hour 18, for both pad materials at 6 cm pad thick and 0.3 m/s pad face air velocity and from hour 14 to hour 16 for PLF at 10 cm pad thick and 0.5 m/s pad face air velocity revealed the some disadvantage as illustrated in Figures (4-21) and (4-22).

Table (4-4): Average ratio of temperature reduction to air flow rate (ΔT/Q) during the operating period for both materials of pad at different pad thickness and pad face air velocities.

Pad thickness (cm)	Pad face air velocity (m/s)	($\Delta T/Q$) °C.s/m^3	
		RS	PLF
3	0.3	17.97	5.90
	0.5	5.25	3.80
	1.05	4.71	4.03
6	0.3	10.47	5.72
	0.5	4.80	4.93
	1.05	4.14	4.56
10	0.3	8.02	7.04
	0.5	4.81	4.42
	1.05	6.09	4.05
15	0.3	5.13	6.19
	0.5	5.09	4.20
	1.05	4.52	4.09

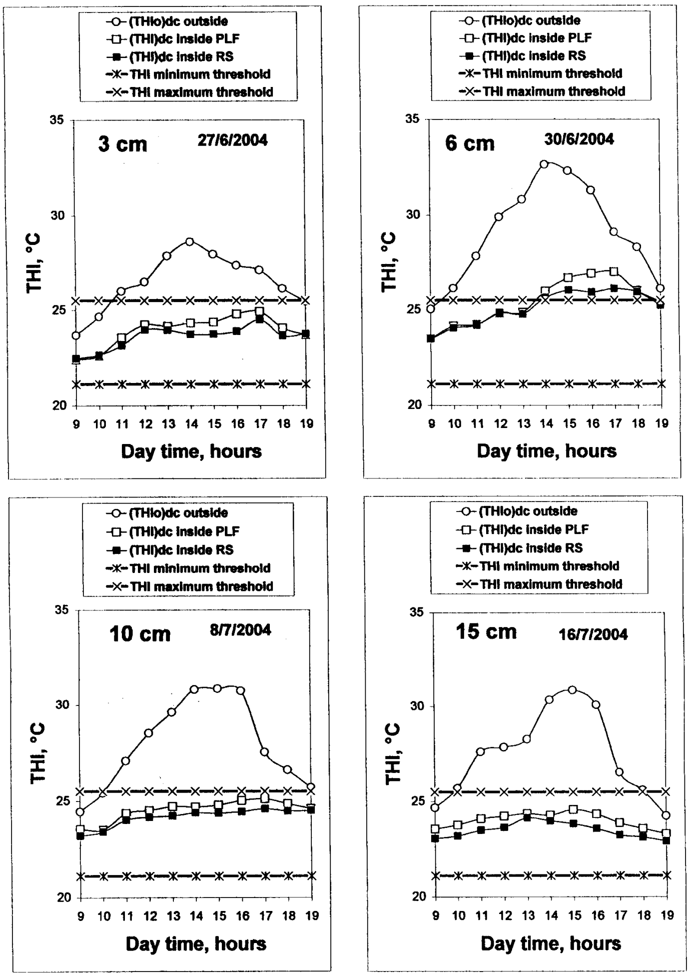


Figure (4-21): Temperature humidity index (THI) for dairy cows throughout the operating period for rice straw (RS) and palm leaf fibers (PLF) at different pad thickness and 0.3 m/s pad face air velocity.

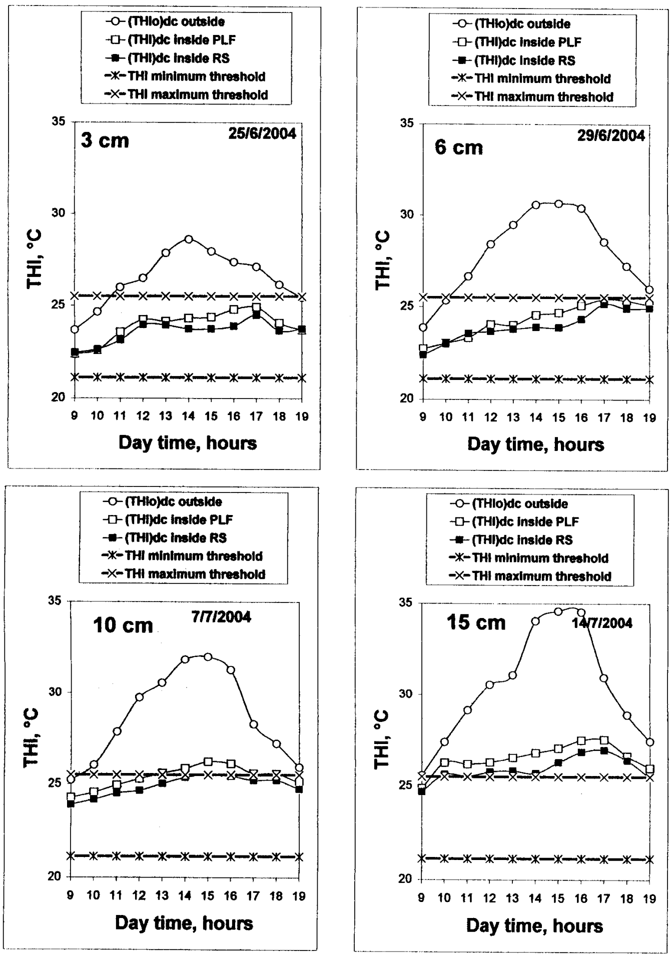


Figure (4-22): Temperature humidity index (THI) for dairy cows throughout the operating period for rice straw (RS) and palm leaf fibers (PLF) at different pad thickness and 0.5 m/s pad face air velocity.

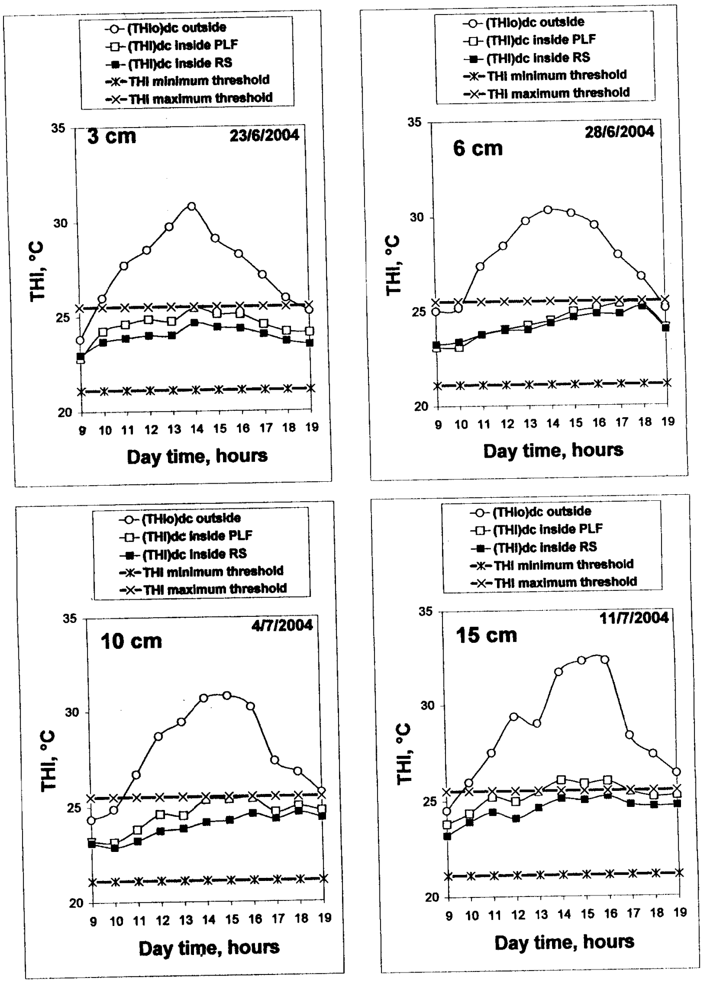


Figure (4-23): Temperature humidity index (THI) for dairy cows throughout the operating period for rice straw (RS) and palm leaf fibers (PLF) at different pad thickness and 1.05 m/s pad face air velocity.

respectively. However, the average values of $(THI)_{dc}$ of cooled air during the operating period for these two last cases were 25.11 °C for RS, 25.41 °C for PLF at 6 cm thick and 0.3 m/s pad face air velocity and 25.37 °C for PLF at 10 cm thick and 0.5 m/s pad face air velocity as indicated in Table (4-5). Comparing with the maximum threshold $(THI)_{dc}$ which is 25.5 °C concludes that both pad materials at these mentioned cases are still having the potential for reducing heat stress throughout the operating period. Regarding case of 15 cm pad thickness of both pad materials at 0.5 m/s pad face air velocity, the average values of $(THI)_{dc}$ during the operating period were 25.91 °C and 26.52 °C for RS and PLF, respectively as shown in Table(4-5). These values are very close to maximum threshold $(THI)_{dc}$, (25.5 °C)that means an acceptable performance. The whole values of $(THI)_{dc}$ and reduction temperature humidity index for dairy cows $(\Delta THI)_{dc}$ during the operating period for both pad materials at various pad thickness and pad face air velocities are indicated in Table (4-5). The lowest values of $(THI)_{dc}$ were 23.08 °C and 23.93°C for RS and PLF at 3 cm pad thick and 0.5 m/s pad face air velocity, respectively. Also, Table (4-5) indicates the maximum values reduction temperature humidity index $(\Delta THI)_{dc}$ of about 4.48°C and 3.87 °C for RS and PLF, respectively, at 15 cm pad thick and 0.5 m/s pad face air velocity. The previous result happened when $(THIo)_{dc}$ was 30.39°C , this value was higher than the values of about all treatments. Thus the thicker pad has an ability to reduce heat stress at 0.5 m/s pad face air velocity.

THI belongs to laying hens:

Figures (4-24), (4-25) and (4-26) show the temperature humidity index $(THI)_L$ for laying hens when using both pad materials at different pad specifications. THI values of cooled air during the operating period for all treatments are lower than that for outside $(THIo)_L$ or without cooling. Moderate fluctions in $(THI)_L$ of the cooled air were observed at all treatments. Generally, both RS and PLF pad materials have an opportunity to keep $(THI)_L$ of cooled air under the maximum threshold one. However, they are failed to keep this advantage in case of 15 cm thick of pad material and 0.5 m/s pad face air velocity as shown in Figure (4-25). This may refer to the high value of $(THIo)_L$ of the non- cooled air. Few hours during operating period from hour 14 to hour 19 for both pad materials at 6 cm pad thick and 0.3 m/s pad face air velocity Figure (4-21).The $(THI)_L$ for both pad materials the maximum threshold was higher than that from hour 12 to hour 18 at 10 cm pad thick and 0.5m/s as illustrated in Figure (4-25). $(THI)_L$ for PLF material was higher than the maximum threshold from hour 13 to hour 17 at 15 cm pad thickness and 1.05 m/s pad face air velocity as revealed in Figure (4-26). This previous result means that these treatments are failed to keep under maximum $(THI)_L$ threshold. However, maximum threshold THI for laying hens may have a higher value of about 28°C as stated by Gates et al. (1995). The whole average values of $(THI)_L$ and reduction temperature $(\Delta THI)_L$ during the operating period for the two pad materials at various pad thickness and pad face air velocities are indicated in Table (4-6). The lowest values of $(THI)_L$ (i.e. the reduction in heat stress) of about 24.07 °C for RS pad

Table (4-5): The average temperature humidity index $(THI)_{dc}$ belongs to dairy cows during the operating period for both pad materials at various pad thickness and pad face air velocities.

Pad thickness (cm)	Pad face air velocity (m/s)	Outside $(THIo)_{dc}$ (°C)	Pad material			
			RS		PLF	
			Inside $(THIo)_{dc}$ (°C)	$(\Delta THI)_{dc}$ (°C)	Inside $(THIo)_{dc}$ (°C)	$(\Delta THI)_{dc}$ (°C)
3	0.3	27.90	23.66	4.24	24.33	3.57
	0.5	26.48	23.08	3.40	23.93	2.55
	1.05	27.48	23.92	3.56	24.54	2.94
6	0.3	29.04	25.11	3.93	25.41	3.63
	0.5	27.92	23.94	3.98	24.29	3.63
	1.05	27.78	24.20	3.58	24.33	3.45
10	0.3	27.95	24.16	3.79	24.52	3.43
	0.5	28.73	24.88	3.85	25.37	3.36
	1.05	27.77	23.92	3.85	24.53	3.24
15	0.3	27.44	23.47	3.97	23.99	3.45
	0.5	30.39	25.91	4.48	26.52	3.87
	1.05	28.61	24.52	4.09	25.24	3.37

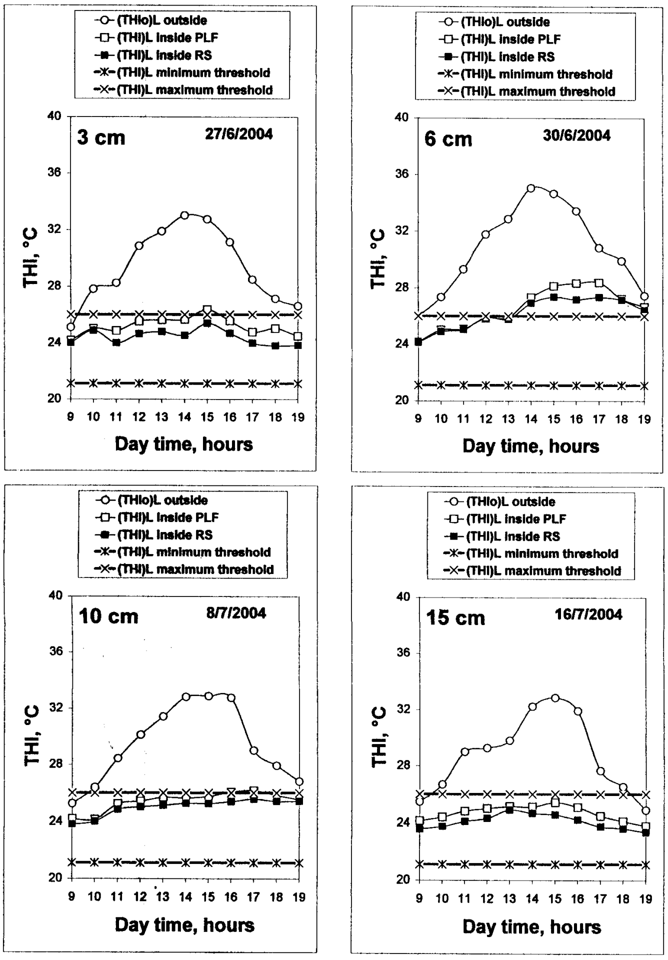


Figure (4-24): Temperature humidity index (THI) for laying hens throughout the operating period for rice straw (RS) and palm leaf fibers (PLF) at different pad thickness and 0.3 m/s pad face air velocity.

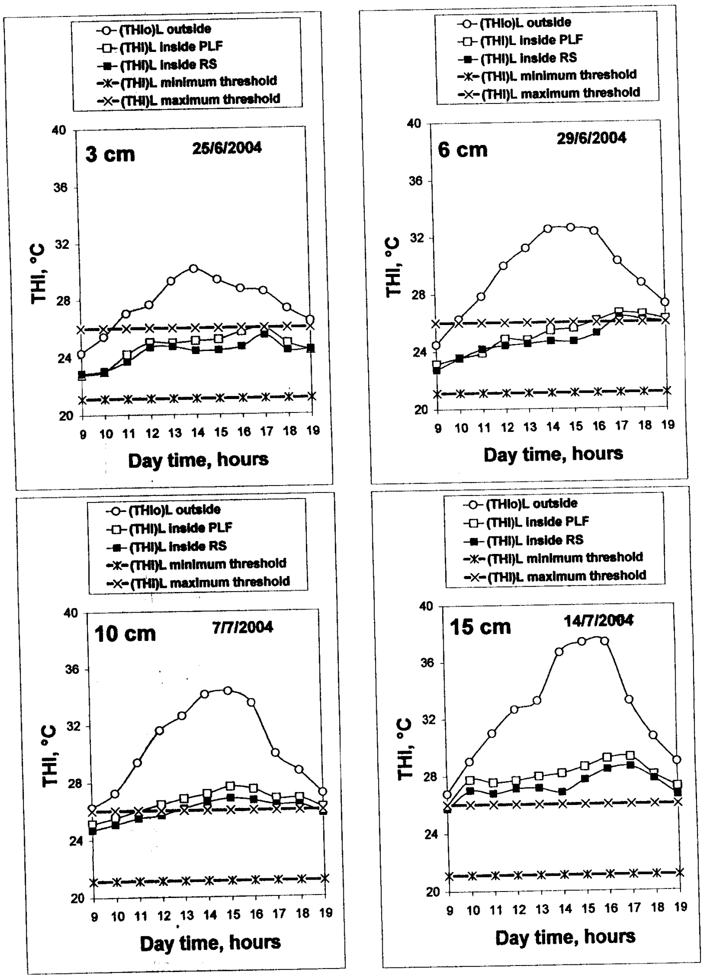


Figure (4-25): Temperature humidity index (THI) for laying hens throughout the operating period for rice straw (RS) and palm leaf fibers (PLF) at different thickness and 0.5 m/s pad face air velocity.

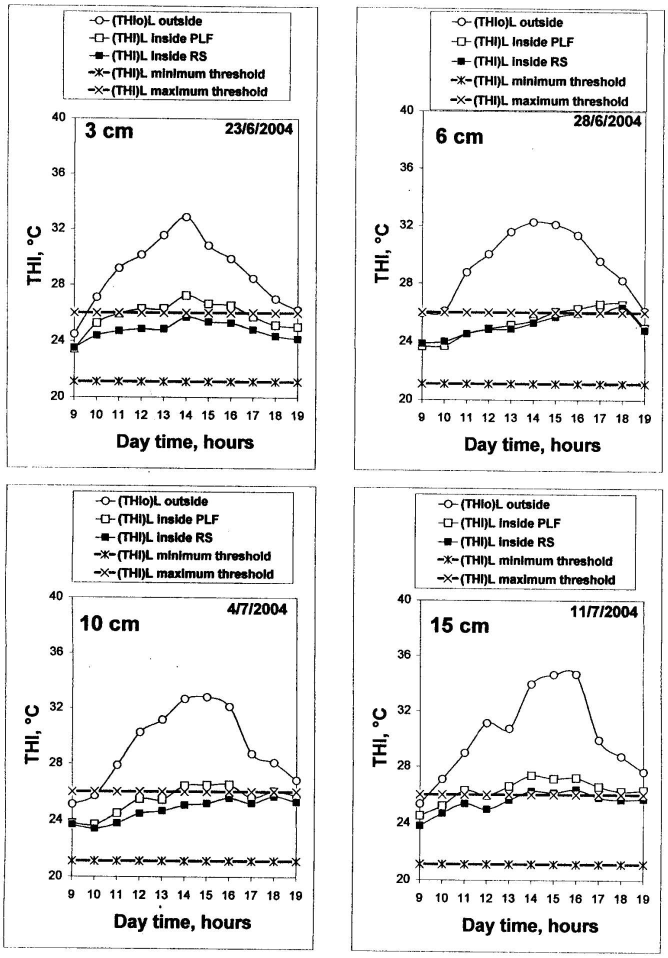


Figure (4-26): Temperature humidity index (THI) for laying hens throughout the operatingperiod for rice straw (RS) and palm leaf fibers (PLF) at different pad thickness and 1.05 m/s pad face air velocity.

Table 4-6: The average temperature humidity index $(THI)_L$ belongs to laying hens during period for both pad materials at various pad thickness and pad face air velocities.

Pad thickness (cm)	Pad face air velocity (m/s)	Outside $(THIo)_L$ (°C)	Pad material			
			RS		PLF	
			Inside $(THIo)_L$ (°C)	$(\Delta THI)_L$ (°C)	Inside $(THIo)_L$ (°C)	$(\Delta THI)_L$ (°C)
3	0.3	29.92	24.41	5.51	25.19	4.73
	0.5	27.48	24.27	3.21	24.66	2.82
	1.05	29.40	24.73	4.67	25.78	3.62
6	0.3	31.25	26.20	5.05	26.56	4.69
	0.5	29.73	24.77	4.96	25.16	4.57
	1.05	29.75	25.10	4.65	25.25	4.50
10	0.3	29.90	25.12	4.78	25.54	4.36
	0.5	31.00	25.98	5.02	26.55	4.45
	1.05	29.60	24.72	4.88	25.42	4.18
15	0.3	29.43	24.07	5.36	24.70	4.73
	0.5	33.04	27.26	5.78	27.96	5.35
	1.05	30.72	25.64	5.08	26.47	4.25

material at 15 cm pad thick and 0.3 m/s pad face air velocity. The lowest value of $(THI)_L$ was 24.66 °C for PLF pad material at 3 cm pad thick and 0.5 m/s pad face air velocity. The maximum values of $(\Delta THI)_L$ were 5.78 and 5.35 °C, respectively were found for RS and PLF at 15 cm thick and 0.5 m/s pad face air velocity. This were found at outside $(THI)_L$ of about 33.04 °C which considered as the highest value.

THI belongs to broilers:

Temperature humidity index of non-cooled air (outside) belongs to broilers $(THIo)_B$ cooled air for broilers $(THI)_B$ and under simulated broiler housing conditions $(THI)_{BS}$ for both pad materials at different pad specifications throughout the operating period are illustrated in Figures (4-27), (4-28) and (4-29). Generally, in all treatments, it can be said that the $(THI)_B$ and the $(THI)_{BS}$ were remain lower than the outside air temperature humidity index $(THIo)_B$ during the operating period. As it was expected temperature humidity index under simulation broiler housing conditions $(THI)_B$ was higher than that for the cooled air belongs to broilers (without loading) $(THI)_B$. As well, during few hours from the beginning and end of starting and the operating the outside air relative humidity was high and cooling potential was low, the $(THI)_{BS}$ when using RS pad material at 3 cm, 6 cm pad thickness and at 0.3 m/s pad face air velocity revealed higher values than the outside non- cooled air temperature humidity index $(THIo)_B$ as shown in Figure (4-27). This is due to the prevailing effect of increasing inside air relative humidity compared with the effect of decreasing dry-bulb temperature for the cooled air during these specific hours. In general values of the $(THI)_B$ and the $(THI)_{BS}$ for RS pad material were lower than that for PLF. Only one exception was found when using 3 cm thick of RS pad material at 0.3 m/s pad face air velocity as illustrated in Figure (4-27) since the $(THI)_{BS}$ for RS were higher than that for PLF throughout the operating period. This is due to the lowest air flow rate (0.46 m^3 /s) that has been used for RS pad material as compared with 1.18 m^3/s used for PLF pad material to have the same pad face air velocity of 0.3 m/s.

Comparing temperature humidity index of cooled air for broilers $(THI)_B$ with the maximum threshold value reveals that the $(THI)_B$ is out of the range in all treatments which means undesirable effect. But, on the other hand, the evaporative cooling still has virtual advantage in reducing the THI of the non-cooled (outside) air when passing through the investigated pad materials. Such an advantage is considered as a respectable effect specifically if one is compared with this valuable reduction in the THI and the small deviation above the maximum threshold level. The same behavior was found for temperature humidity index under simulation broiler housing conditions $(THI)_{BS}$. Since it has a more deviation above the maximum threshold level than that happened just for the cooled air (without loading) for broilers $(THI)_B$. But, again, it is still lower than the THI of non-cooled air by too much units compared with less units in raising $(THI)_{BS}$ above the maximum threshold level. As well as, it should be

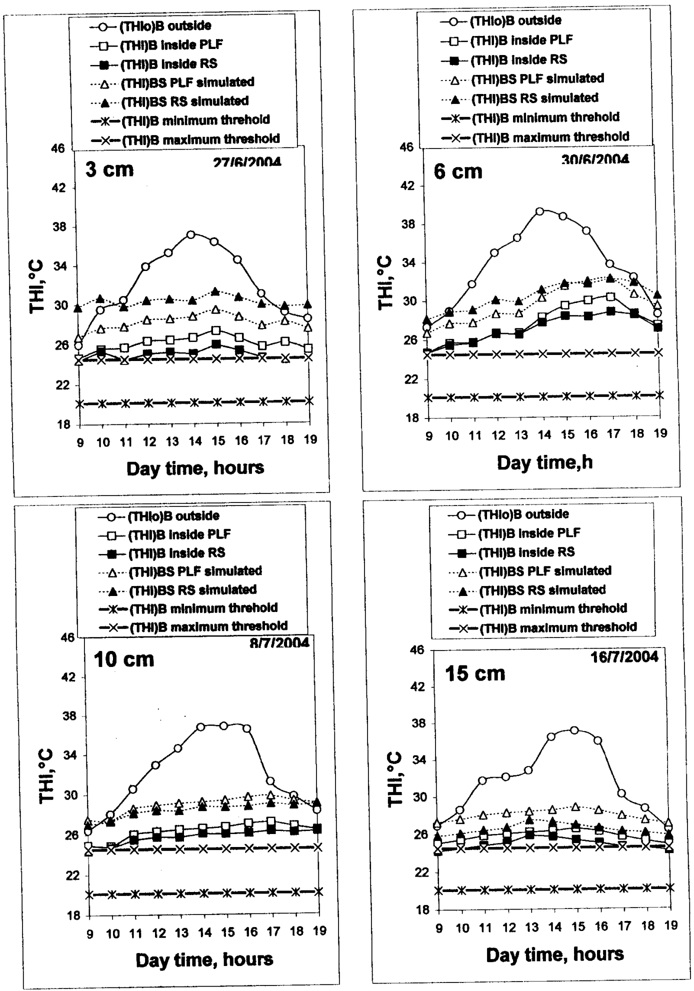


Figure (4-27): Temperature humidity index (THI) for broiler throughout the operating period for rice straw (RS) and palm leaf fibers (PLF) at different pad thickness and 0.3 m/s pad face air velocity.

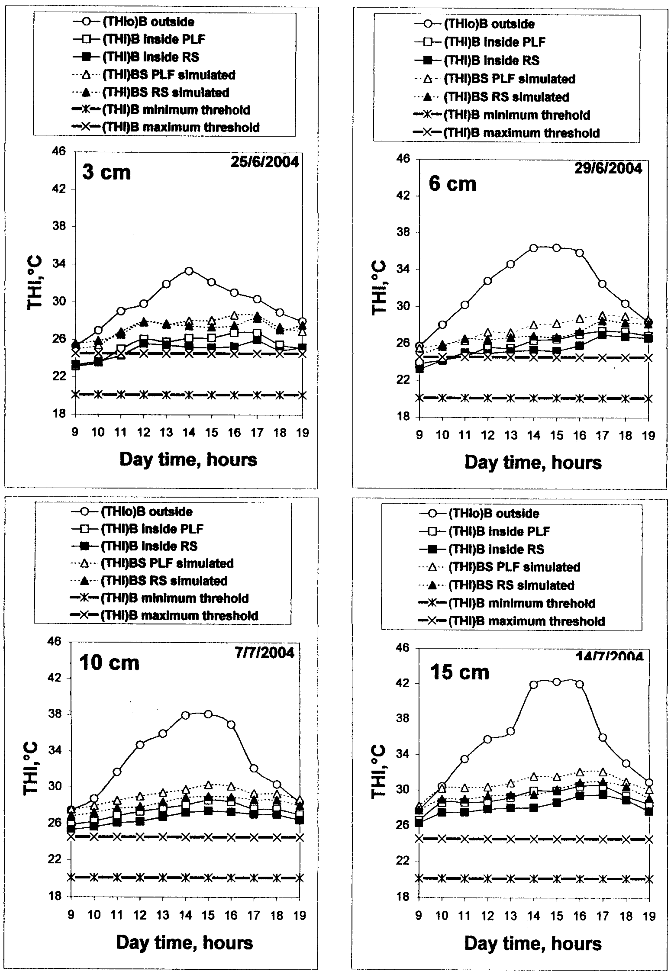


Figure (4-28): Temperature humidity index (THI) for broiler throughout the operating period for rice straw (RS) and palm leaf fibers (PLF) at different pad thickness and 0.5 m/s pad face air velocity.

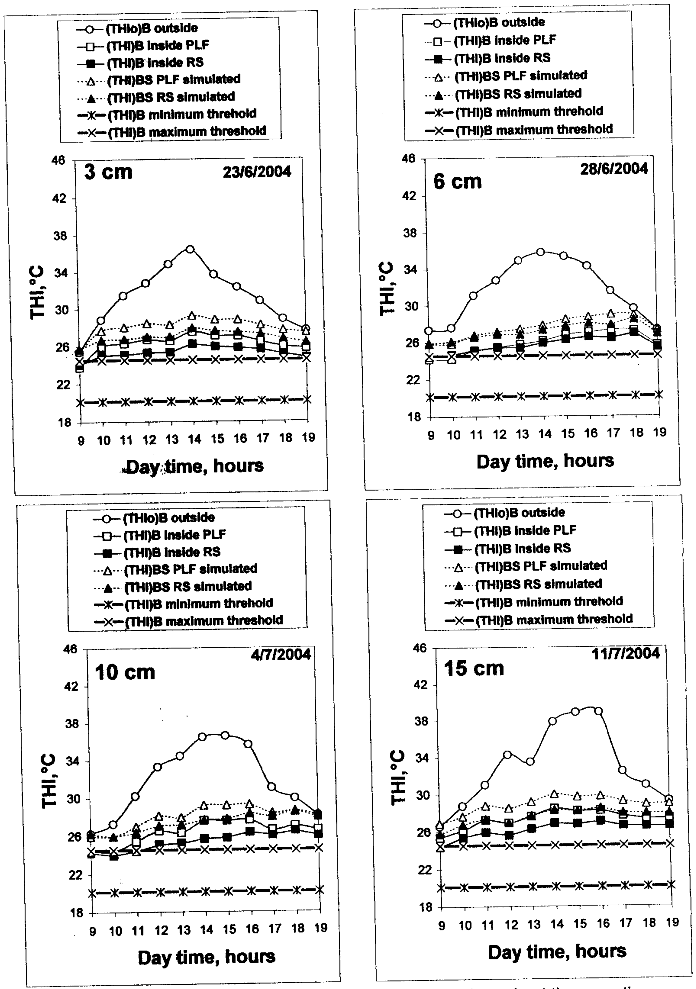


Figure (4-29): Temperature humidity index (THI) for broiler throughout the operating period for rice straw (RS) and palm leaf fibers (PLF) at different pad thickness and 1.05 m/s pad face air velocity.

mentioned here that there is a part of heat generated and added to the cooled air may be consumed in evaporating some water from litter within the actual broiler housing system. Such a quantity of eliminated heat results in some reduction in the $(THI)_{BS}$ was neglected in the present study. However, in severe circumstances reducing the stocking rate may be a valuable solution. This suggestion may be already applied since many procedures begin to sell a percentage of their flock before the sixth weak of age. Otherwise an indirect evaporative cooling system could be used. Table (4-7) indicates the average temperature humidity index $(THI)_{B}$, $(THI)_{BS}$ and reduction temperature humidity index $(THI)_{B}$, $(THI)_{BS}$ belongs to broiler and simulation broiler. The lowest values of $(THI)_{B}$ was 24.89 °C for RS pad material with 3 cm pad thick and 0.5 m/s pad face air velocity. The same previous value was found for RS pad material with 15 cm pad thick and 0.3 m/s pad face air velocity. The lowest value for PLF pad material was 25.45 °C at 3 cm pad thick and 0.5 m/s pad face air velocity. The greater $(\Delta THI)_{B}$ was 7.36 °C, 4 when using RS pad material at 15 cm pad thickness and 0.5 m/s pad face air velocity when $(THIo)_{B}$ was 34.48 °C. The greater $(\Delta THI)_{B}$ for PLF was 5.60 °C with 15 cm pad thick and 1.05 m/s pad face air velocity when (THIo) was 32.96 °C. The (THIo) values in the last two cases were higher than that the first. From previous results we can concluded that under thicker pad, and middle and upper pad face air velocities there are have an ability to reduce heat stress. The above results are summarized and listed in Table (4-7). The greater values for $(THI)_{BS}$ were 4.90 °C and 4.73 °C for RS and PLF with 15 cm pad thick and 0.5 m/s pad face air velocity as shown in Table (4-7).

4.2 Saturation efficiency (SE)

The saturation efficiency (SE) which indicates evaporative cooling efficiency is illustrated in Figures (4-30) to (4-32) for both rice straw (RS) and palm leaf fibers (PLF) pad materials throughout operating period with pad specifications. In all Figures, it seems that RS pad material has higher values of SE than that for PLF. In few hours, values of SE for PLF pad material were higher than that for RS pad material. Values of SE for PLF pad material was higher than SE values for RS pad material from hour 13 to hour 14 with 15 cm pad thick and 0.3 m/s pad face air velocity as revealed in Figure (4-30). As shown in Figure (4-31) SE values for PLF were higher than RS from hour 9 to hour 10 with 3 cm pad thick and 0.5 m/s pad face air velocity, from hour 10 to hour 11 with 6 cm pad thick and 0.5 m/s pad face air velocity. SE for PLF was higher than SE for RS from hour 9 to hour 10 with 6 cm pad thick and 1.05 m/s pad face air velocity.

In all treatments the higher values of SE for both pad materials observed in middle operating period as expected when the higher reduction in temperature was found. The values of SE decreased at the beginning and end of operating period for both pad materials. These values which gave higher values in middle operating period were ranged from 82% to 87%, and 70% to 80% for both RS and PLF pad materials, respectively. These previous results occurred when the air temperature was reduced. The values of SE decreased at the

Table (4-7): The average temperature humidity index $(THI)_B$, $(THI)_{Bs}$ and reduction temperature humidity index $(THI)_B$, $(THI)_{Bs}$ belongs to broiler and broiler simulated.

Pad thickness (cm)	Pad face air velocity (m/s)	Outside $(THI)_B$ (°C)	Pad material							
			RS				PLF			
			Inside $(THI)_B$	$(\Delta THI)_B$	$(THI)_{BS}$ simulated	$(\Delta THI)_{Bs}$ simulated	Inside $(THI)_B$	$(\Delta THI)_B$	$(THI)_{BS}$ simulated	$(\Delta THI)_{Bs}$ simulated
3	0.3	31.96	24.93	7.03	30.27	1.69	26.04	5.92	28.14	3.82
	0.5	29.71	24.89	4.82	27.18	2.53	25.45	4.26	27.32	2.39
	1.05	31.19	25.30	5.89	26.79	4.40	26.30	4.89	28.04	3.15
6	0.3	33.50	27.05	6.45	30.46	3.04	27.57	5.93	29.59	3.91
	0.5	31.94	25.37	6.57	26.90	5.04	25.92	6.02	27.64	4.30
	1.05	31.59	25.67	5.92	27.14	4.45	25.89	5.70	27.57	4.02
10	0.3	31.98	25.71	6.27	28.39	3.59	26.29	5.69	28.91	3.07
	0.5	32.92	26.59	6.33	28.20	4.72	27.40	5.52	29.10	3.82
	1.05	31.74	25.30	6.44	27.37	4.37	26.37	5.37	27.98	3.76
15	0.3	31.35	24.89	6.63	26.51	4.84	25.78	5.57	28.01	3.34
	0.5	34.48	28.12	7.36	29.58	4.90	29.11	5.37	30.74	4.73
	1.05	32.96	26.20	6.76	27.61	5.35	27.36	5.60	28.91	4.05

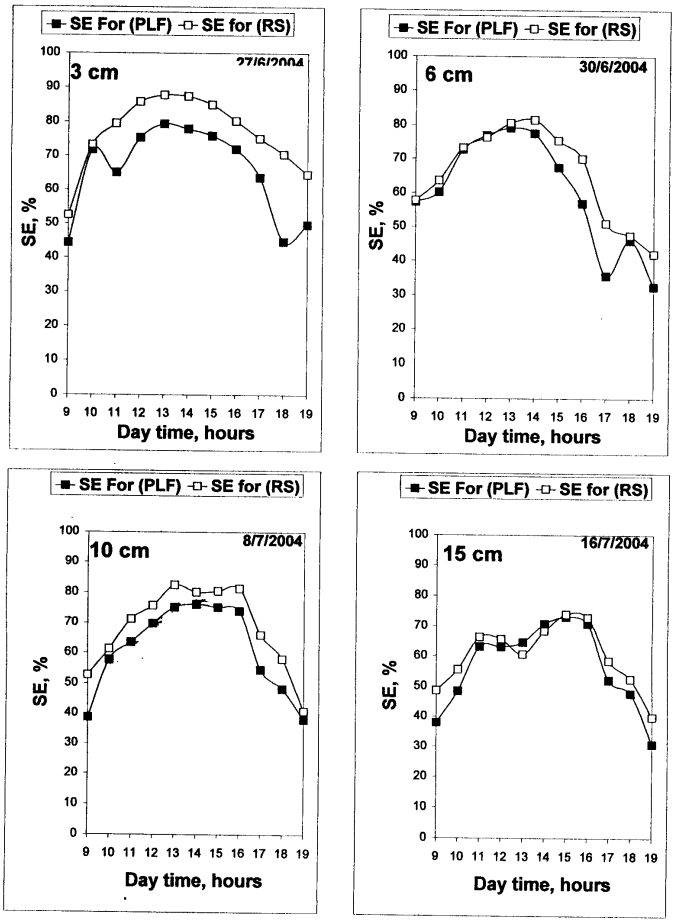


Figure (4-30): Saturation efficiency (SE) throughout the operating period for rice straw (RS) and palm leaf fibers (PLF) at different pad thickness and 0.3 m/s pad face air velocity.

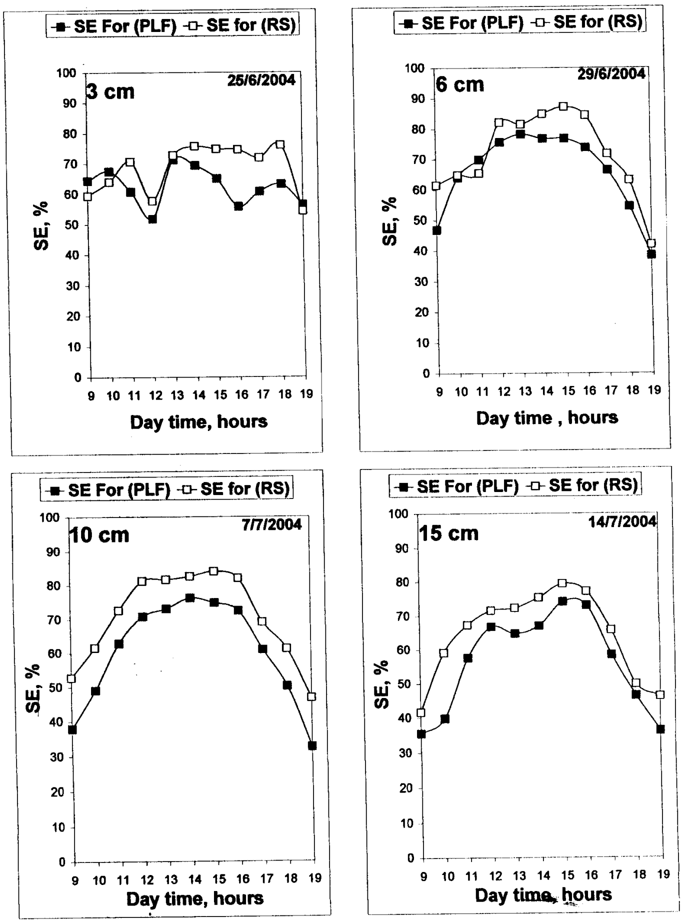


Figure (4-31): Saturation efficiency (SE) throughout the operating period for rice straw (RS) and palm leaf fibers (PLF) at different pad thickness and 0.5 m/s pad face air velocity.

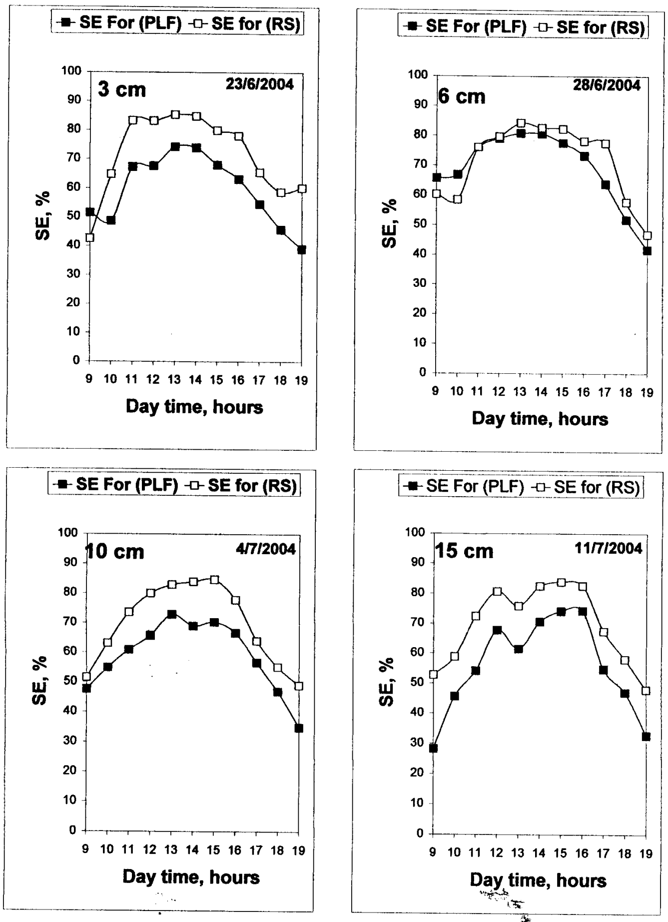


Figure (4-32): Saturation efficiency (SE) throughout the operating period for rice straw (RS) and palm leaf fibers (PLF) at different pad thickness and , 1.05 m/s pad face air velocity.

beginning and end of operating period. The obtained data are in agreement with Wiresma 1969, Bottcher and Baughman 1983, Bottcher at al 1992, Uğurlu and Kara (2000) and Liao and Chiu (2002).

Table (4-8) indicates saturation efficiency average during the operating period for both pad materials with various pad thickness and pad face air velocities. The higher values of SE for RS pad material was 76.51% with 3 cm pad thick and 0.3 m/s pad face air velocity when outside air temperature was 33.51 °C and outside air relative humidity was 49.77% as shown in Table (4-8). The higher value of SE for PLF pad materials was 68.70% with 6 cm pad thick and 1.05 m/s pad air velocity. When outside air temperature was 32.98 °C and outside air relative humidity was 53.65% as revealed in Table (4-8). Also, change of airflow rate effect on this phenomenon.

Saturation efficiency is a function of both air-water contact time and pad-face air velocity up to a maximum level. Increasing air velocity through a given pad thickness decreases the contact time. The improved efficiency with increased air velocity was the reverse of what one might except because with higher velocity, the opportunity for evaporation or contact time is reduced.

Increasing the velocity through a given pad thickness decreases the contact time. The reasons for improvement in efficiency with contact time are readily apparent, and in practice it can be controlled by varying the velocity. The influence of variations in velocity with a given air-water contact time has been given a little since, other than to caution against excessive velocities. The close association between velocity and contact time and the complex reasoning behind the improvement in efficiency with increased velocity has masked the recognition of independent importance of velocity. The improvement in efficiency with velocity can be explained by considering the heat and mass transfer activities in the vicinity closely surrounding the pad material fibers. A pad is characterized by numerous saturated pad material fibers surrounded by a viscous film of very nearly saturated air. The adiabatic saturation process of evaporative cooling involves heat and mass transfer, which must be accomplished during a very brief period of time. As air passes through the pad, water must vaporize and pass out ward through the film. Heat necessary to evaporate the water is provided by sensible heat that is transferred by conduction into the film from the surrounding air, achieving the desired effect of decreasing air temperature. The heat transfer rate through the film, which in turn influences the rate of heat transfer and evaporation. As pad-face air velocity increases, the thickness of the film decreases resulting in increased heat transfer and evaporation rates and corresponding increase in saturation efficiency. Increasing the air velocity reduces the contact time, which can be restored by increasing pad thickness. This comment about the relationship between SE and pad thickness and pad face air velocity according to Benham and Wiersma (1974).

From all previous results for SE, relationship between SE and pad thickness and pad face air velocity. We can concluded that for each thickness increasing pad face air velocity may be decreased cooling potential.

Table 4-8: The average saturation efficiency (SE) for both materials at various pad thickness and pad face air velocities under different conditions

Pad thickness (cm)	Pad face air velocity (m/s)	Outside conditions		Saturation efficiency (SE), (%)	
		Temperature (° C)	Relative humidity (%)	RS	PLF
3	0.3	33.51	49.77	76.51	65.40
	0.5	30.93	56.70	63.34	63.40
	1.05	32.57	53.35	71.42	59.38
6	0.3	35.21	48.48	65.40	60.25
	0.5	33.46	50.75	71.73	65.60
	1.05	32.98	53.65	71.79	68.77
10	0.3	33.50	50.77	68.21	60.98
	0.5	34.41	52.25	70.47	60.06
	1.05	33.25	50.62	69.57	58.72
15	0.3	33.19	46.65	60.24	56.58
	0.5	37.29	46.68	64.14	56.32
	1.05	34.58	48.76	69.30	55.49

Figures (4-33) and (4-34) illustrate saturation efficiency as affected by pad thickness at different ranges of relative humidity of the outside air for rice straw and palm leaf pad materials, respectively. Saturation efficiency decreased by increasing pad thickness for all treatments of rice straw pad material. However, some exceptions were found at 0.3 m/s pad face air velocity when the outside air relative humidity ranged from 70 to 80%, at 0.5 m/s pad face air velocity when the outside air relative humidity ranged from 40 to 50 %, and at 1.05 m/s pad face air velocity when the outside air relative humidity ranged from 60 to 70% and 70 to 80%. It should be mentioned here that the corresponding outside dry-bulb temperature for each pot eel value of SE was not the same at the different pad thickness and the different ranges of outside air relative humidity. Such an experimental conditions may have an effect that leads to the above mentioned exceptions. However, regression analysis showed that outside dry-bulb temperature had no significant effect on SE as shown in Figure (4-33).

Saturation efficiency decreased by increasing pad thickness for all treatments of palm leaf fibers pad material as shown in Figure (4-34). The following two multiple regression equations were developed to describe the relationship between SE and pad face air velocity (V), pad thickness (d), outside air relative humidity (RHo) and outside air dry bulb-temperature:

$$SE_{RS} = 100.63559028 + 5.8455345681*V + (-0.9463783908*d) + (-0.7308798366*RHo) + 0.2889852776\ T_{db}.\ [R^2= 0.75]\ldots (21)$$

$$SE_{PLF} = 116.09462508 + 1.972018229*V + (-0.992382025*d) + (-0.8811109953*\ RHo) + (-\ 0.998542207*\ T_{db}).[R^2= 0.69].\quad (22)$$

Where:

SE_{RS} = saturation efficiency for rice straw pad material

SE_{PLF} = saturation efficiency for palm leaf fibers pad material

Regression analysis showed that the outside air relative humidity had the highest effect on SE followed by pad thickness, pad face air velocity and outside air dry-bulb temperature as illustrated in Tables (7-1 and 7-2).

4.3 Unit evaporative cooler performance (Unit ECP)

Figures from (4-35) to (4-37) illustrate unit evaporative cooler performance for both rice straw (RS) and palm leaf fibers (PLF) pad materials throughout operating period at pad specifications. Unit ECP for PLF pad material was higher than Unit ECP for RS pad material when using 10 cm pad thickat 0.3 and 0.5 m/s pad face air velocities as shown in Figures (4-35) and (4-37). As well, Unit ECP for PLF pad material was higher than RS pad material as revealed in Figure (4-35) when using 3 cm pad thick and 0.3 m/s pad face air velocity. Despite of the higher values of SE for RS apparent when using 3 cm pad thick and 0.3 m/s pad face air velocity but this values of SE corresponded the lowest Unit ECP at the same specifications as shown in Tables (4-8) and (4-9). This due to airflow rate which was at (0.46 m^3 /s) for all treatments. Table (4-9) indicates Unit ECP average during the operating period for both pad material at various pad thickness and pad face air velocities. This table shows that Unit ECP is increased by increasing pad thickness at 0.3 m/s pad face air velocity when using

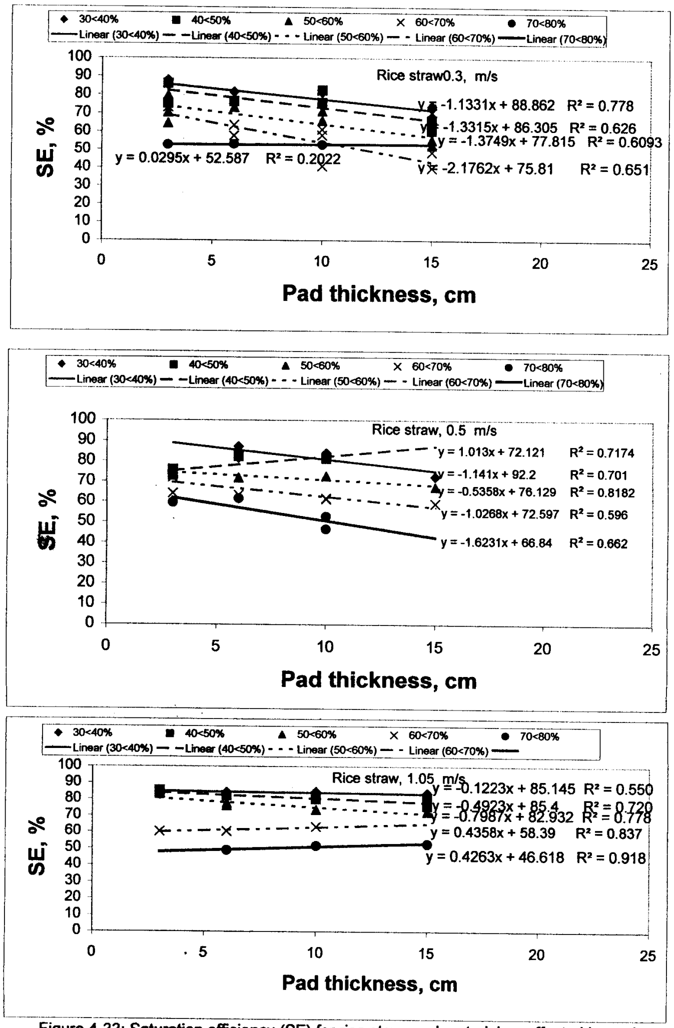
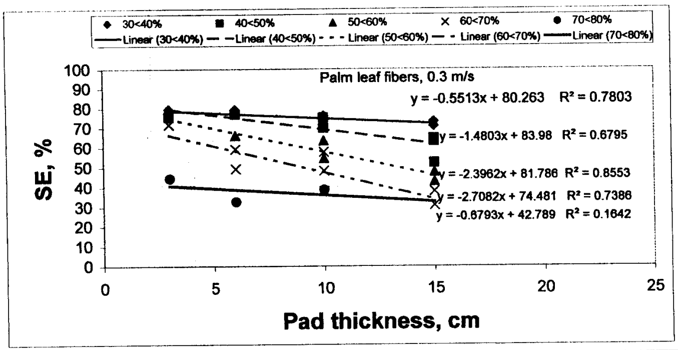
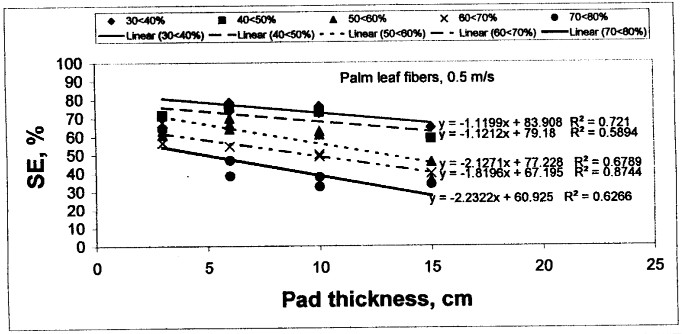


Figure 4-33: Saturation efficiency (SE) for rice straw pad material as affected by pad thickness at different ranges of relative humidity of the outside air.





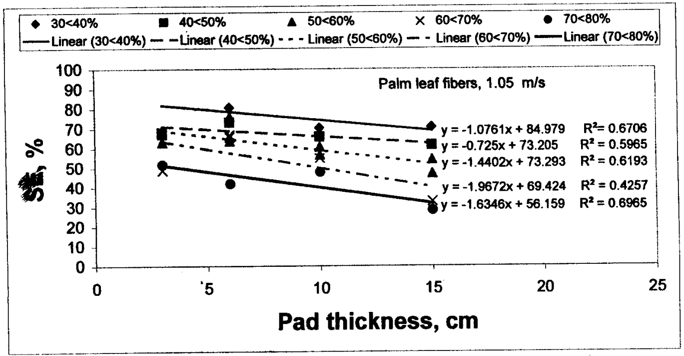


Figure 4-34: Saturation efficiency (SE) for palm leaf fibers pad material as affected by pad thickness at different ranges of relative humidity of the outside air.

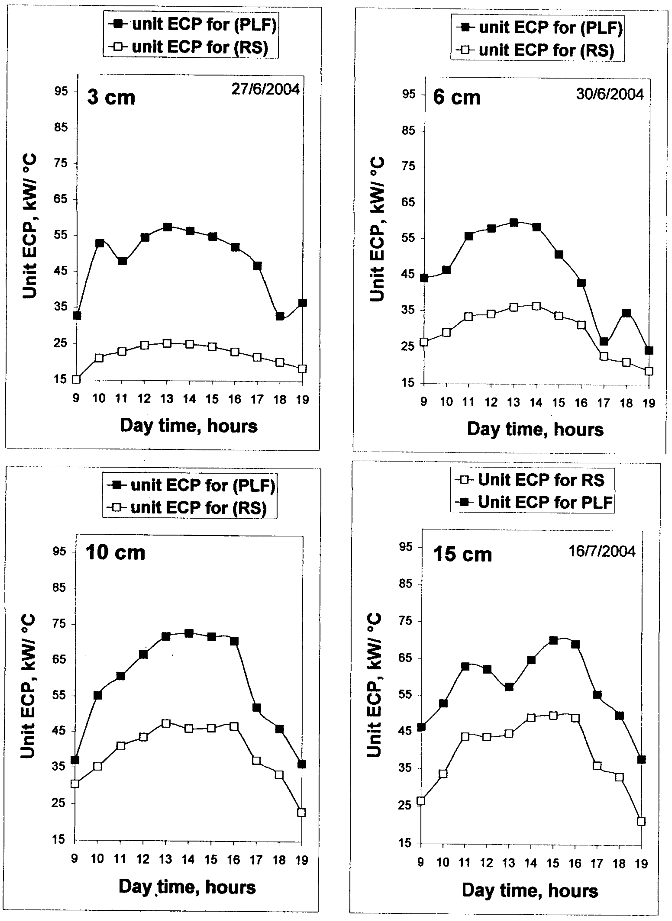


Figure (4-35): Unit evaporative cooler performance (Unit ECP) throughout the operating period for rice straw (RS) and palm leaf fibers (PLF) at different pad thickness and 0.3 m/s pad face air velocity.

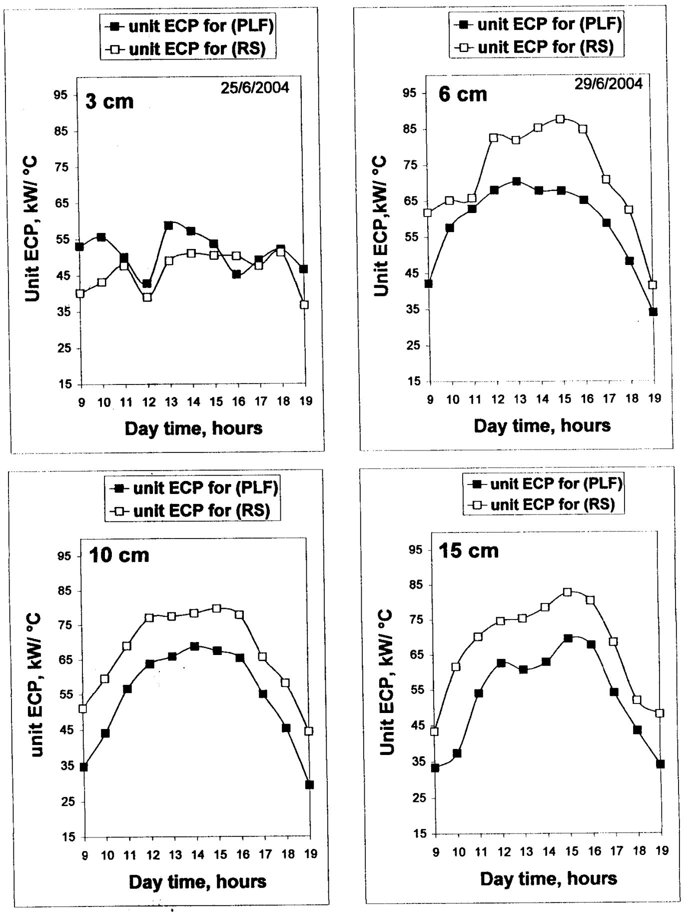


Figure (4-36): Unit evaporative cooler performance (Unit ECP) throughout the operating period for rice straw (RS) and palm leaf fibers (PLF) at different pad thickness and 0.5 m/s pad face air velocity.

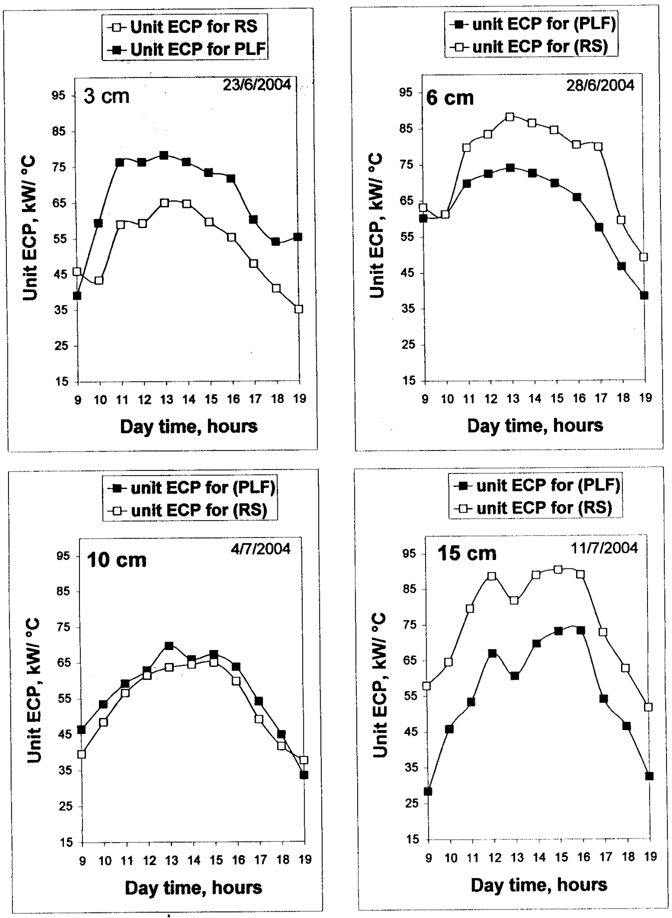


Figure (4-37): Unit evaporative cooler performance (Unit ECP) throughout the operating period for rice straw (RS) and palm leaf fibers (PLF) at different pad thickness and 1.05 m/s pad face air velocity.

Table 4-9: The average unit evaporative cooler performance (Unit ECP) for both materials at various pad thickness and pad face air velocities under different conditions.

Pad thickness (cm)	Pad face air velocity (m/s)	Outside conditions		Pad material	
		Temperature (° C)	Relative humidity (%)	Unit ECP for RS (kW/°C)	Unit ECP for PLF (kW/°C0
3	0.3	33.51	49.77	21.97	47.71
	0.5	30.93	56.70	45.99	51.25
	1.05	32.57	53.35	65.41	52.31
6	0.3	35.21	48.48	29.41	45.65
	0.5	33.46	50.75	71.79	58.39
	1.05	32.98	53.65	74.14	62.57
10	0.3	33.50	50.77	39.07	58.22
	0.5	34.41	52.25	67.11	54.16
	1.05	33.25	50.62	53.35	56.40
15	0.3	33.19	46.65	57.15	39.12
	0.5	37.29	46.68	66.82	52.67
	1.05	34.58	48.76	75.23	54.87

RS pad material. Unit ECP increased by increasing thickness from 3 cm to 6 cm and decreased again at 0.5 and 1.05 m/s pad face air velocities This result mean that 6 cm is the preferred depending on value of Unit ECP. For PLF increasing thickness lead to decrease the value of Unit ECP at 0.3 m/s pad face air velocity. However, increasing pad thickness from 6 cm to 10 cm pad thickness cause increasing of Unit ECP at 0.3 m/s pad farce air velocity. Increasing pad thickness from 3 cm to 6 cm as a case in RS pad material Unit ECP increased by increasing pad thickness and decreased again. This previous result may leads to a concluded that 6 cm pad thickness was the preferred depending on value of Unit ECP. We should be considered that eliminates variable associated with local conditions according with Wiresma and Short (1983).

Ultimatly evaporative pad cooling system using rice straw or palm leaf fibers as a pad materials is a suitable method to reduce heat stress in agricultural structures. Also, the present study indicates the rice straw pad material have higher performance than palm leaf pad material. Therefore, we can utility rice straw as pad materials depending on two facts. The first fact is conservation environment from population which producing by kindle rice straw by farmers. The second fact is rice straw as pad material considering available and not costly pad material when comparing it with other natural or industrial pad materials. We recommended that when operating pad evaporative cooling system continuously throughout the day time with control system provided and maintained a desired level of environment l in agricultural structures. Recommendations for further experimental work are to study deterioration of rice straw and palm leaf as pad materials to increase their performance, as well study reused these pad materials after used it in another application in agricultural field.

5 SUMMARY AND CONCLUSION

Environmental control inside the agricultural structure is considered as one of the most important factors affecting productivity of protected cropping, animal, and poultry. Agricultural structures such as livestock and poultry housing and greenhouses,...etc, should have the capability to offer the optimum environment for the confined biological system. One of the most important environmental factors that affect a confined biological system is temperature. As well known, there is a thermo-neutral (comfort) zone for each biological system, so that chagrining of temperature beyond the maximum or minimum limits of that zone depresses productivity. However, an environmental control system may be needed to keep the required comfort zone inside the structure. From the viewpoint of thermal environmental factors, an environmental control system may be used as an auxiliary system to control the temperature within the comfort zone via heating or cooling process. Egypt has a climate tend to be warm or hot throughout the most of the year months. Therefore, heat stress is considered as an accompanied problem that the biological system such as poultry or dairy animals can be exposed to. Such problem could be noticed clearly in summer months and during hot waves throughout the year. As well, it obviously appears in broiler houses with intensive production specifically before marketing increasing percentage of mortality. Accordingly, searching for a simple or not sophisticated system to cope with the heat stress problem inside the agricultural structures is needed.

Evaporative cooling is one of the most important method that can be used to reduce heat stress inside the agricultural structures. To make this method available and easy to be used, it was suggested to make its material depends on local materials that can be found at any farmer and in such a way that can be constructed as a hand-made system.

Objectives:

The main objective of the present study was to construct an evaporative cooling system based on pad and fan system using agricultural residues as pad materials and investigate its performance criteria.

Specific objectives can be summarized as follows:

1- Investigate the feasibility of using rice straw and palm leaf fibers (Kerina) as pad materials of an evaporative cooling system and determination of their performance criteria.

2- Study the effect of pad material, pad thickness and pad face air velocity on the performance of the evaporative cooling system.

3- Investigate the effect of the suggested evaporative cooling system on reducing heat stress in terms psychometric properties of the cooled air comes into agricultural structures (dairy cows, laying hens, and broilers housing).

4- Predict the performance of the system in reducing heat stress under simulated broiler housing conditions.

Procedure:

All experimental work of the present study was conducted at Denosher village-El Mehalla El Kobra-Gharbia governorate. Two similar rooms were (5.70*4.2*2.75 m) build of red brick and reinforced concrete and provided to be used as an agricultural structure. Two pads (2.92*1.12 m) the first one made of rice straw and the second made of palm leaf fibers (Kerina) were constructed in a steel frame capable of changing the pad thickness. Each pad was fixed on a wall of a room and a suction fan (0.50 m diameter and discharge of 2 m^3/s) was constructed in the other facing wall. Dry and wet-bulb temperatures were measured and recorded at 18 locations within the structure. Dry and wet-bulb temperatures of the ambient air outside the structure were also measured and recorded as well. These data were recorded hourly from hour 9 to hour 19 for each treatment.

Factors under study:

1- Pad material: Rice straw and palm leaf fibers (Kerina).
2- Pad thickness: 3, 6, 10 and 15 cm.
3- Pad face air velocity: 0.3, 0.5 and 1.05 m/s.

Parameters of the study:

1- Dry-bulb temperature (T_{db}).
2- Relative humidity (RH)
3- Ratio of temperature reduction to airflow rate ($\Delta T/Q$)
4- Temperature humidity index (THI).
5- Saturation efficiency (SE).
6- Unit evaporative cooler performance (Unit ECP).

Some data were collected in summer 2003. Some problems occurred during this season, such as suitable pad density, adjusting pad face air velocity and the poor of distribution of water over the pad at the higher pad thickness. Therefore, pad density was modified to suit the suggested density of materials (32kg/m^3) and pad face air velocity was adjusted. Better distribution of water over the pad was achieved by adding a corrugated and perforated sheet metal over the pad. Main Data were recorded in summer 2004. Specifically twelve days within the period of 23 June to 16 July representing hot condition were selected to run the experiments of all treatments.

Important results:

The important results of the present study can be summarized as follows:

1- Utilization of rice straw and palm leaf fibers (Kerina) as pad materials in an evaporative cooling system may be feasible. This is can contribute in solving the accumulation of some agricultural residues such as rice straw. In other words it can play a role in the environmental dimension.
2- Temperature reduction for all rice straw treatments was higher than that for palm leaf fibers (Kerina). It ranges from 5.67 °C (278.67 K) to 8.66 °C (281.66 K) for all rice straw treatments, while in palm

leaf fibers (Kerina) treatments it ranged from 5.01(273.01 K) to 7.50 °C (280.50 K).

3- For all treatments of the suggested evaporative cooling system, it was found, as it is expected, an increase in air relative humidity, which considered the main disadvantage of the system. The minimum mean value of inside air relative humidity was 73.79% with 15 cm palm leaf fibers (Kerina) pad thick and 0.3 m/s pad face air velocity. The maximum mean value was 86.63% with 3 cm rice straw pad thick and 0.3 m/s pad face air velocity.

4- For both pad materials, the highest value of temperature reduction was achieved with 15 cm pad thick and 0.5 m/s pad face air velocity. While the lowest value occurred with 3 cm pad thick and 0.5 m/s pad face air velocity.

5- The highest mean value of the ratio of temperature reduction to airflow rate was 17.97 °C / (m^3 / s) and occurred with 3 cm pad thick of rice straw and 0.3 m/s pad face air velocity. The lowest mean of 3.8 °C / (m^3 / s) was found with 3 cm pad thick of palm leaf fibers (Kerina) and 0.5 m/s pad face air velocity.

6- Applying the temperature-humidity index (THI) concept for some biological systems such as broilers, laying hens and dairy cows revealed that the suggested system was considered to be efficient. For all treatments of the present study, inside THI values were less than that for the outside ambient air temperature.

7- Rice straw pad material reduced THI more than that for palm leaf fibers (Kerina) in most cases.

8- Predicted values THI inside a laying hens housing or inside a dairy cows housing were lower than the threshold THI at the upper limit of the optimum zone for the interior conditions. However, higher values of inside THI than the threshold value at the upper limit of the optimum zone for the interior conditions were achieved in a broiler housing system. It was concluded that in such agricultural structures, an indirect evaporative cooling system may be needed to improve the inside THI.

9- Despite that $(THI)_B$ and $(THI)_{BS}$ were higher than maximum threshold all time at all treatments, but it were all most time less than outside $(THI_O)_{BS}$.

10- Mean of saturation efficiency for rice straw treatments was higher than that for palm leaf fibers (Kerina). It ranged from 60.24 % to 75.51% and from 55.49% to 68.77% for rice straw and palm leaf fibers, respectively.

11- The highest average value of saturation efficiency of about 76.51% was achieved using 3 cm pad thick of rice straw with 0.3 m/s pad face air velocity. While the lowest value of 55.49% was recorded when using 15 cm pad thick of palm leaf fibers (Kerina) with 1.05 m/s pad face air velocity.

12-The highest and lowest mean values of unit evaporative cooler performance (Unit ECP) were found for rice straw pad material with 15 cm thick, 1.05 m/s pad face air velocity and 3 cm pad thick, 0.3 m/s pad face air velocity, respectively.

Recommendations:

1- Using rice straw and palm leaf fibers as in a pad and fan evaporative cooling system is feasible for dairy cows (or any large animals need to be cooled) and laying hens housing systems.

2- In general, it can be concluded that rice straw pad material can be used as an evaporative cooling pad material for agricultural structures that give the first priority to reduce temperature regardless relative humidity level such as dairy cows housing systems. This can offer a high temperature reduction when using 15 cm pad thick and 0.5 m/s pad face air velocity. On the other hand, when relative humidity has the first priority, palm leaf fibers (Kerina) can be used as a pad material at the same specifications as rice straw.

3- Possibility using the suggested system in broiler housing systems is permitted when outside relative humidity is low although the THI of cooled air is higher the threshold THI, since the cooled air THI is still lower than that of the outside air. In case of the necessity of keeping the cooled air THI lower than that of THI threshold at a limit value, the suggested system can be used as a part of an indirect evaporative cooling system. In which the direct cooled air can be used to cool the entering air in directing via a heat exchanging system.

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7 APPENDIX

Table 7-1: ANOVA of multiple regression for rice straw as pad material.

Source	SS	df	MS	F	P
Total	21435.840299	131			
Regression	16081.990809	4	4020.4977023	95.371229455	.0000***
V	169.8417549	1	169.8417549	4.0288586581	.0469*
d	874.72185523	1	874.72185523	20.749495446	.0000***
RHo	15017.749018	1	15017.749018	356.23977268	.0000***
T_{db}	19.67818069	1	19.67818069	0.4667910355	.4957 ns
Error	5353.8494901	127	42.156295198		

Table 7-2: ANOVA of multiple regression for palm leaf fibers as pad material.

Source	SS	df	MS	F	P
Total	24448.606166	131			
Regression	17053.891236	4	4263.4728089	73.222707277	.0000***
V	2.2123727524	1	2.2123727524	0.0379962368	.8458 ns
d	1053.3860008	1	1053.3860008	18.091302147	.0000***
RHo	15995.943411	1	15995.943411	274.72118025	.0000***
T_{db}	2.3494508318	1	2.3494508318	0.0403504744	.8411ns
Error	7394.7149302	127	58.226101813		

بسم الله الرحمن الرحيم

"تهيئة البيئة داخل منشأ زراعي"
معايير الأداء لوسادتين تبريد تبخيري مصنعتين من مخلفات زراعية لخفض الإجهاد الحراري داخل المنشآت الزراعية

الملخص العربي

يعتبر التحكم البيئي داخل المنشآت الزراعية واحداً من أهم العوامل التي تؤدي إلي زيادة الإنتاجية. و المنشآت الزراعية مثل مزارع الإنتاج الحيواني، و كذلك الصوب الزراعية.... الخ. يجب أن يكون لها المقدرة علي توفير البيئة المثلي للمنظومة البيولوجية داخل المنشأ. و تعتبر درجة الحرارة من أهم العوامل التي تؤثر علي الكائن الحي داخل أي منشأ زراعي، و كما هو معروف أن هناك منطقة أو حدود لدرجة الحرارة يطلق عليها منطقة التعادل (الراحة) لكل منظومة بيولوجية معينة. و تغيير درجة الحرارة خارج الحد الأعلى و الحد الأدنى لهذه المنطقة يعمل علي إضعاف و خفض الإنتاجية و من ثم قد يكون من الضروري وجود نظام التحكم البيئي للحفاظ علي أن تكون البيئة الحرارية داخل المنشأ بين حدود تلك المنطقة و من وجهة نظر عوامل البيئة الحرارية قد يستعمل نظام التحكم البيئي كوسيلة مساعدة للتحكم في درجة الحرارة داخل المنشأ للحفاظ عليها دائما في حدود منطقة الراحة و ذلك عن طريق إما عملية تدفئة أو عملية تبريد.

تعتبر جمهورية مصر العربية من الأقطار ذات المناخ المائل للدفء أو للحرارة في معظم شهور العام، و من ثم فإن مشكلة الإجهاد الحراري هي دائما ما تكون مشكلة مصاحبة للمنظومة البيولوجية داخل المنشأ الزراعي كما هو الحال في مجال الإنتاج الحيواني و الداجني حيث قد تتعرض الحيوانات و الدجاج لمثل ذلك، تلك المشكلة يمكن ملاحظتها بوضوح في شهور فصل الصيف و كذلك أثناء أي موجات حارة خلال العام،بالإضافة إلي ذلك فتظهر هذه المشكلة بوضوح في بعض المنشآت الزراعية مثل منشآت إيواء دجاج اللحم خاصة التي تخضع لنظام التربية المكثفة و تزداد المشكلة تفاقما عندما تكون الطيور في عمر قرب عمر التسويق مما قد يتسبب عنه خسارة كبيرة. و من ثم كان البحث عن نظام بسيط و غير معقد في تركيبه و تشغيله للتغلب علي هذه المشكلة مطلباً ملحاً.

يعتبر التبريد التبخيري أحد الطرق المعروفة و التي يمكن استخدامها لخفض الإجهاد الحراري في المنشآت الزراعية، و لجعل ذلك متاحاً و سهلاً للاستخدام فقد افترضت الدراسة الحالية نظام للتبريد التبخيري يعتمد في إنشاءه علي مواد محلية و كذلك متوافرة لدي المزارع أو المنتج علاوة علي أن يكون هذا النظام سهل التركيب و الإنشاء يدوياً.

الأهداف:

الهدف الرئيسي من الدراسة الحالية هو إنشاء و تركيب نظام تبريد تبخيري ينتمي إلي النوع ذو الوسادة و المروحة وذلك باستخدام بعض المخلفات الزراعية كمواد للوسادة بالإضافة إلي دراسة معايير أداء هذا النظام المقترح . **ويمكن تلخيص الأهداف الفرعية للدراسة الحالية كما يلي :**

١- دراسة إمكانية استخدام قش الأرز و ألياف أوراق النخيل (الكرينه) كمواد لوسادة نظام تبريد تبخيري و تحديد معايير الأداء الخاصة بهما.

٢- دراسة تأثير نوعية مادة الوسادة و سمك الوسادة و سرعة الهواء عند وجه الوسادة علي أداء نظام التبريد التبخيري المقترح.

٣- التوقع بأداء النظام المقترح لبعض المنشآت الزراعية وذلك اعتمادا علي السلوك السيكرومتري للهواءعندما يتم تبريده فور دخوله المنشأ وذلك في مساكن أبقار اللبن والدجاج البياض ودجاج اللحم (التسمين) .

٥- التوقع بأداء النظام كوسيلة لخفض الإجهاد الحراري وذلك بمحاكاة ظروف مساكن دجاج التسمين .

الإجراء:

تم العمل التجريبي بالكامل في الدراسة الحالية في قرية دنوشر مركز المحلة الكبرى بمحافظة الغربية . تم بناء غرفتين متماثلتين (٥,٧×٤,٢×٢,٧٥م) من الطوب الأحمر و الخرسانة المسلحة و إعدادهما ليمثلا منشأ زراعي ما. تم عمل وسادتين (٢,٩٢×١,١٢م) الأولي تم عملها من قش الأرز و الثانية من ألياف أوراق النخيل (الكرينه) و تم تثبيتهما في إطار من الحديد له القابلية لتغيير سمك الوسادة بداخله، ثم تم تثبيت كل واحدة من الوسادتين علي الحائط لتحل محل النافذة في دخول الهواء من خلالها. و تم تركيب مروحة ذات قطر ٠,٥ م في الحائط المقابل لتقوم بسحب الهواء من المنشأ و الذي يمر بالطبع من خلال الوسادة. تم تسجيل درجات الحرارة الجافة و الرطبة في ثماني عشر نقطة داخل المنشأ كما تم تسجيل درجات الحرارة الجافة و الرطبة للهواء الخارجي المحيط بالمنشأ، و تم تسجيل هذه البيانات كل ساعة ابتداء من الساعة التاسعة صباحاً و حتى السابعة مساءً لكل معامله. **وكانت المتغيرات قيد الدراسة كما يلي:**

١ـ مادة الوسادة:

قش الأرز و ألياف أوراق النخيل (الكرينه).

٢ـ سمك الوسادة:

٣، ٦، ١٠، ١٥ سم.

٣ـ سرعة الهواء عند وجه الوسادة:

٠,٣، ٠,٥، ١,٠٥ م/ث.

وكانت مؤشرات الدراسة كما يلي:

١ـ درجة الحرارة الجافة.
٢ـ الرطوبة النسبية.
٣ـ دليل درجة الحرارة- الرطوبة.
٤ـ كفاءة التشبع.
٥ـ وحدة أداء المبرد التبخيري.
٦ـ النسبة بين خفض درجة الحرارة و معدل سريان الهواء.

تم تجميع البيانات أولاً في صيف ٢٠٠٣ وقد ظهرت بعض المشكلات في تشغيل النظام المقترح مثل الكثافة المناسبة لمادة الوسادة، التحكم في و ضبط سرعة الهواء عند وجه الوسادة بالإضافة إلي سوء توزيع المياه فوق الوسادة و بصفة خاصة عند السمك الأكبر ومن ثم تم تعديل كثافة الوسائد لتلائم المواد المقترحة (٣٢كجم/م^{٣}). أيضاً تم ضبط سرعة الهواء عند وجه الوسادة بالإضافة إلي تزويد النظام المقترح بشريحة معدنية مجعدة و مثقبة توضع فوق الوسادة مباشرة مما أدي إلي انتظامية توزيع الماء فوقها. ومن ثم تم تجميع البيانات الأساسية في صيف عام ٢٠٠٤ و بصفة خاصة تم اختيار اثني عشر يوماً داخل

الفترة من ٢٣ يونيو إلي ١٦ يوليو حيث تمثل ظروف حارة قاسية إلي حد كبير و ذلك لإجراء جميع التجارب لجميع المعاملات.

أهم النتائج:

يمكن تلخيص أهم النتائج للدراسة الحالية فيما يلي:

إمكانية استخدام قش الأرز و ألياف أوراق النخيل (الكرينه) كمواد لوسادة نظام التبريد التبخيري يمكن أن نستنتج من ذلك أيضاً أن مثل هذا الاستخدام لتلك المخلفات الزراعية قد يساهم بشكل ما في حل مشكلة تراكم تلك المخلفات و بصفة خاصة قش الأرز أو بمعني آخر يمكن أن يلعب ذلك دوراً فيما يخص البعد البيئي.

١- امكانية استخدام قش الأرز و ألياف أوراق النخيل كمادة و سادة في نظام التبريد التبخيري هذا من ناحية ، و من ناحية أخري يتم حل مشكلة المخلفات الزراعية مما يلعب دورا هاما في البعد البيئي.

٢- خفض درجة الحرارة لجميع معاملات قش الأرز كان أعلى منه لجميع معاملات ألياف أوراق النخيل(الكرينه) و قد تروواح هذا الخفض في درجة الحرارة من ٥,٦٧ إلي ٨,٦٦°م لجميع معاملات قش الأرز، بينما في جميع معاملات ألياف أوراق النخيل (الكرينه) قد ترواح هذا الخفض في درجة الحرارة من ٥,٠١ إلي ٧,٥٠°م.

٣- كانت أعلي قيمة لخفض درجة الحرارة لكلا المادتين عند استخدام وسادة ذات سمك ١٥سم وسرعة هواء عند وجه الوسادة مقدارها ٠,٥م/ث. بينما كانت أقل قيمة لخفض درجة الحرارة عند استخدام وسادة ذات سمك ٣ سم و سرعة هواء عند وجه الوسادة مقدارها ٠,٥م/ث.

٤- كان متوسط كفاءة التشبع لمعاملات قش الأرز أعلي منه لمعاملات ألياف أوراق النخيل (الكرينه) وقد تروواحت من ٦٠,٢٤٪ إلي ٧٦,٥١٪ ومن ٥٥,٤٩٪إلي ٦٨,٧٧ ٪ و ذلك لكل من قش الأرز و ألياف أوراق النخيل (الكرينه) علي الترتيب.

٥- بلغت اعلي قيمة متوسطة لكفاءة التشبع حوالي ٧٦,٥١٪ وذلك عند استخدام وسادة من قش الأرز ذات سمك ٣ سم و عند سرعة هواء عند وجه الوسادة مقدارها ٠,٣ م/ث. بينما كانت أقل قيمة لكفاءة التشبع هي ٥٥,٤٩ ٪ وذلك عند استخدام وسادة من الياف أوراق النخيل (الكرينه) بسمك ١٥ سم وسرعة هواء عند وجه الوسادة ١,٠٥ م/ث.

٦- لجميع المعاملات لنظام التبريد التبخيري المقترح وجد كما هو متوقع زيادة في الرطوبة النسبية و التي تعتبر هي التأثير السلبي لمثل هذا النظام المقترح. وقد كانت أقل قيمة متوسطة للرطوبة النسبية داخل المنشأ ٧٣,٧٩٪ وذلك عند استخدام وسادة من ألياف أوراق النخيل (الكرينه) ذات سمك ١٥ سم و سرعة هواء عند وجه الوسادة مقدارها ٠,٣ م/ث. و بلغت أقصى قيمة متوسطة ٨٦,٦٣٪ عند استخدام وسادة من قش الأرز ذات سمك ٣ سم و سرعة هواء عند وجه الوسادة مقدارها ٠,٣ م/ث.

٧- عند تطبيق مفهوم دليل درجة الحرارة- الرطوبة علي بعض النظم البيولوجية مثل دجاج التسمين و الدجاج البياض و أبقار الحليب فقد تم التوقع بكفاءة النظام المقترح. فلجميع معاملات الدراسة الحالية كانت قيم دليل درجة الحرارة-الرطوبة داخل المنشأ أقل منها للهواء الخارجي المحيط بالمنشأ مما يعني بصفة عامة نجاح النظام في خفض الإجهاد الحراري داخل المنشأ.

٨- في معظم حالات أو معاملات الدراسة كانت قيم دليل درجة الحرارة- الرطوبة عند استخدام قش الأرز أقل منها عند استخدام ألياف أوراق النخيل (الكرينه) .

٩- القيم المتوقعة لدليل درجة الحرارة-الرطوبة داخل منشأ لإيواء الدجاج البياض و كذلك داخل منشأ لإيواء أبقار الحليب كانت أقل من القيمة الحدية لدليل درجة الحرارة-الرطوبة عند الحد الأعلى لمنطقة الراحة المثلي للظروف الداخلية، و لكن بالنسبة للقيم المتوقعة ذاتها في حالة منشأ لإيواء دجاج التسمين كانت أعلي من القيم الحدية لدليل درجة الحرارة-الرطوبة عند نفس الظروف و لكنها ظلت أقل من دليل درجة الحرارة-الرطوبة للهواء الخارجي المحيط بالمنشأ كما سبق ذكره. و طبقا لذلك فقد تم استنتاج أنه في مثل هذه المنشآت قد يكون التبريد التبخيري غير المباشر هو الوسيلة الأفضل و ذلك لتحسين قيم درجة الحرارة-الرطوبة داخل المنشأ

١٠- وجد أن أعلي و أقل قيم متوسطة لوحدة أداء المبرد التبخيري كانت عند استخدام قش الأرز كمادة للوسادة و ذلك عند ١٥ سم كسمك للوسادة، ١,٠٥ م/ث كسرعة هواء عند وجه الوسادة و عند ٣ سم كسمك للوسادة ، ٠,٣ م/ث كسرعة هواء عند وجه الوسادة علي الترتيب.

١١- أعلي قيمة متوسطة للنسبة بين خفض درجة الحرارة إلي معدل سريان الهواء بلغت ١٧,٩٧ °م/(م^{٣}/ث) وذلك عند استخدام وسادة من قش الأرز بسمك ٣ سم و سرعة هواء عند وجه الوسادة مقدارها ٠,٣ م/ث ، وبلغت أقل قيمة لنفس النسبة ٣,٨ °م/(م^{٣}/ث) وذلك عند استخدام وسادة من ألياف أوراق النخيل (الكرينه) ذات سمك ٣ سم و سرعة هواء عند وجه الوسادة مقدارها ٠,٥ م/ث.

١٢- أعلي قيمة لوحدة أداء مبرد تبخيري كانت عند سمك وسادة قدره ١٥ سم وسرعة وجه وسادة ١,٠٥ م/ث وذلك لقش الأرز ، أما أوراق ألياف النخيل فكانت أعلي قيمة بها عند استخدام سمك وسادة قدره ٣ سم و سرعة وجه وسادة ٠,٣ م/ث.

توصيات تطبيقية:

١- يمكن استخدام كا من قش الأرز و ألياف أوراق النخيل (الكرينه) كوسائد في نظم التبريد التبخيري ذات الوسادة المروحة في منشآت إيواء كا من الأبقار الحلابة (أو أي حيوانات كبيرة بصفة عامة تحتاج لتبريد) و كذلك الدجاج البياض.

٢- يفضل استخدام قش الأرز بصفة خاصة كوسادة للتبيرد التبخيري عندما تكون الأولوية لعملية تخفيض درجة الحرارة بغض النظر عن مستوي الرطوبة النسبية. بينما يفضل استخدام ألياف أوراق النخيل كوسادة للتبريد التبخيري عندما تكون الأولوية لرطوبة مناسبة بغض النظر عن تخفيض درجة الحرارة.

٣- في حالة عدم وجود أي وسيلة أخري للتبريد في مساكن دجاج اللحم يمكن استخدام النظام المقترح خاصة عند إنخفاض الرطوبة النسبية للهواء الخارجي، حيث أنه رغم زيادة دليل الحرارة-الرطوبة للهواء المبرد عن الحد الأقصي المسموح به لدجاج اللحم فإنه لا يزال يتميز بإنخفاضه عما يناظره للهواء الخارجي ، في مثل نظم الإيواء تلك و في حالة ضرورة الحفاظ علي دليل الحرارة-الرطوبة للهواء المبرد أقل منه للهواء الخارجي فإنه يمكن إستخدام النظام المقترح كجزء من نظام تبريد تبخيري غير مباشر ، والذي يمكن فيه إستخدام الهواء المبرد مباشرة لتبريد الهواء الداخل للمنشأ بطريقة غير مباشرة عن طريق منظومة للتبادل الحراري.

المستخلص العربي

يعتبر الإجهاد الحراري داخل المنشآت الزراعية أحد المشاكل التي تواجه المنتجين خاصة في المناطق ذات الطقس الحار كما هو الحال في مصر أثناء شهور الصيف. و من ثم يصبح هناك احتياج لنظام تحكم بيئي للتغلب علي هذه المشكلة، وبصفة خاصة إذا كان هذا النظام يمكن إنشاؤه و تركيبه يدوياً و من خامات متوافرة لدي المزارع أو المنتج بالإضافة إلي أنه يكون سهل التشغيل. ومن ثم فقد أجريت هذه الدراسة لاقتراح و إنشاء و تركيب نظام للتبريد التبخيري و الذي ينتمي إلي النوع ذو الوسادة و المروحة و ذلك باستخدام بعض المخلفات الزراعية كمواد لوسادة المبرد. بالإضافة إلي ذلك فقد تم دراسة معايير الأداء لنظام التبريد المقترح تحت مواصفات فنيه مختلفة مثل نوعية مادة الوسادة و سمكها و سرعة الهواء عند وجه الوسادة. وقد استخدم كل من قش الأرز و ألياف أوراق النخيل (الكرينه) كمواد للوسادة، كما تم استخدام أربع قيم لسمك الوسادة وهم ٣، ٦، ١٠، ١٥ سم. و ثلاث قيم لسرعة الهواء عند وجه الوسادة وهم ٠,٣، ٠,٥، ١,٠٥ م/ث.

وقد وُجد أن كلاً من المادتين يمكن استخدامهما كمواد لوسادة نظام التبريد المقترح. كما أثبت قش الأرز كوسادة كفاءة أعلي في خفض درجات الحرارة و من ثم خفض الإجهاد الحراري في المنشآت الزراعية. و من جهة أخري فقد ظهرت بوضوح مشكلة زيادة الرطوبة النسبية تلك المشكلة التي لم تكن علي نفس القدر عند استخدام ألياف أوراق النخيل (الكرينه).

و عند تطبيق مفهوم دليل درجة الحرارة-الرطوبة علي بعض المنشآت الزراعية اثبت النظام المقترح قدرته علي خفض دليل درجة الحرارة-الرطوبة مقارنة بذاته للهواء الخارجي المحيط بالمنشأ. و من وجهة نظر دليل درجة الحرارة- الرطوبة توقعت الدراسة بأن نظام التبريد التيخيري المقترح قد لا يكون هو النظام الأكفأ لبعض النظم البيولوجية مثل نظم إيواء دجاج التسمين. كما قدمت الدراسة أيضاً معايير الأداء الأخرى للنظام المقترح و هي كفاءة التشبع، وحدة أداء المبرد التبخيري نسبة خفض درجة الحرارة إلي معدل سريان الهواء.

لجنة الإشراف

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كلية الزراعة بكفر الشيخ-جامعة طنطا

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كلية الزراعة بكفر الشيخ – قسم الميكنة الزراعية

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كلية الزراعة بطنطا- جامعة طنطا

موضوع البحث:

تهيئة البيئة داخل منشأ زراعى

عنوان الرسالة:

معايير الأداء لوسادتين تبريد تبخيرى مصنعتين من مخلفات زراعية لخفض الإجهاد الحرارى داخل المنشآت الزراعية

رسالة مقدمة من

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للحصول على درجة الماجستير فى العلوم الزراعية (الميكنة الزراعية)

لجنة المناقشة والحكم على الرسالة:

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"تهيئة البيئة داخل منشأ زراعي "

معايير الأداء لوسادتين تبريد تبخيري مصنعتين من مخلفات زراعية لخفض الاجهاد الحراري داخل المنشآت الزراعية

رسالة مقدمة من

محمد رمضان أحمد درويش
بكالوريوس في العلوم الزراعية (شعبة الإنتاج الزراعي)
كلية الزراعة بطنطا – جامعة طنطا ١٩٩٩

كجزء من المتطلبات للحصول علي درجة
الماجستير
في
العلوم الزراعية (الميكنة الزراعية)
قسم الميكنة الزراعية
كلية الزراعة بكفر الشيخ
جامعة طنطا
٢٠٠٦