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IMPROVING THE PERFORMANCE OF CONDITIONING EQUIPMENT IN POULTRY FACILITY

Settore Scientifico Disciplinare: AGR/10

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DOCTORATE IN

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ABSTRACT

Earth-to-air heat exchangers (EAHEs) can reduce the energy consumption required for the heating and cooling of buildings. Besides soil temperature and composition, soil moisture can affect thermal performance of EAHE. The aim of this study was to compare thermal performance of EAHE in dry and artificially wetted soil. Tests were carried out in the Basra Province (Iraq), in a semi-desert area. Two experimental EAHEs were built and tested from June 2013 to February 2014, plus the entire month of August 2014. Pipe exchangers were buried at 2 m depth. One EAHE operated in dry soil (DE), while the other one in artificially wetted soil (WE). In the WE system, a drip tubing placed 10 cm above the air pipe wetted the soil around the exchanger. Air temperature at the inlet, at 12.5 m and 24.5 m distance and at the outlet of both of the exchangers, as well as soil temperature at 2 m depth, 25 cm, 50 cm and 100 cm far from the pipe were continuously monitored. The experimental results confirmed that wetting the soil around EAHE improves the general heat exchange efficiency. In the hottest day of the hottest period, the WE system recorded an average cooling coefficient performance of 9.39 against 7.69 of the DE. In the coldest day of the coldest period, the WE system recorded an average heating coefficient performance of 11.08 against 9.86 of the DE. On maximum, in the hottest hours of the day, the Δt of the WE was 12.60°C while in the DE it was 10.60°C. Moreover, during the nighttime in summer, the WE system warmed the air more than the DE system.

Keywords

Earth-to-air heat exchangers, thermal performance, artificially wetted soil, poultry, cooling.

1 INTRODUCTION

1.1 Preface

Livestock production is about forty percent of the world's total agricultural production. It fulfills one-third of humanity's protein consumption. According to the Food and Agriculture Organization (FAO) of the United Nations, breeding of animals also contributes at the income, social status and food security of about one billion people in the world. Due to the importance of the animal production in human life, several international organizations have been formed to take care of the animals such as the FAO, the World Organization for Animal Health (WOAH) and the World Society for the Protection of Animals (WSPA). One of their aims is to secure adequate climatic conditions for animals breeding because healthy environment is a fundamental objective for the construction of animal production facilities. Within the livestock shelters, control of climatic conditions is one of the main factors that can affect the quality and quantity of production. Therefore, the conditioning systems of internal environment are considered among the most important parts of the agricultural engineering.

Microclimatic conditions play a very important role on the animal production, especially in closed environment. Particularly during the summer months, high ambient temperatures are a major concern for farmers. In fact, the high ambient temperatures adversely affect the productive and reproductive performances in livestock farms. The presence of animals under high temperature leads to many negative results such as decrease of food intake, increase in drinking water consumption and lower growth rates, thus low animal weight. Animals cannot get a good production without the presence of thermal comfort conditions. The thermal comfort is defined in different ways. American Society of Heating, Refrigerating and Air Conditioning Engineers defines it as “the condition where an individual expresses satisfaction with the surrounding environment”. At the same way, according to the European Passive Solar Handbook the thermal comfort “represents the feeling of physical welfare”.

A delicate equilibrium exists between the metabolisms of the animal and physical environmental parameters, such as the temperature. The heat exchange between the animal and a surface is based on conduction if the surface is a solid or a liquid. If the surface is a gas, the heat exchange occurs by convection. When the surface is colder than the body, the animal loses heat. Conversely if the contact surface is warmer than the body, the animal acquires heat. Heat exchange is related also to radiation, which occurs directly and indirectly by animal body.

Researchers have tried to develop many systems to defend animals from hot and from cold. Adequate fairly resistant structures to climatic fluctuations and fans sufficient to ensure clean air and temperature acceptable by sending a light breeze without nuisance for animals present in the room have been designed. Evaporative cooling techniques to lower the temperature is also an option. In a physical sense, the cooling of air is a result of the fact that the sensible heat of the air is converted into the latent heat. Nevertheless, in this case the problem could be an excessive increase of relative humidity.

Furthermore, using the technique of conditioning the ventilation air by using underground pipes it is possible to provide warm air in the cold situation and cold air in hot situation.

In desert and semi-arid areas, many breeders have abandoned farming. To create a suitable environment, large sums of money can be necessary. In many cases breeders are unable to overcome ambient climatic conditions where the amount of costs and incomes become approximately equal. Thus, the process of livestock farms is not economically feasible.

This research will shed light on the technology of air conditioning by underground pipes and an improving of plant performance. Furthermore, this technology can be used in difficult and exceptional circumstances, such as in warm conditions with high temperatures and high relative humidity. On top, it can be added to other technology without a significant increase of running costs.

1.2 Heat transfer

1.2.1 Basic Concepts

The internal energy of a system is defined as the sum of all microscopic forms of energy related to the molecular structure of a system and the degree of the molecular activity. The international unit of energy is joule (J) or kilojoule (1 kJ = 1000 J). Another famous unit of energy is the calorie (1 cal = 4.1868 J), which is defined as the energy needed to raise the temperature of 1 gram of water at 14.50°C by 1°C.

There are two mechanisms for the transfer of energy which are work and heat. When the driving force of energy interaction is the difference in temperature, the energy transfer is done by heat. Consequently, the energy interaction is heat transfer. Otherwise, it is work. Anyway, many types of energy like kinetic and electrical energy are an interaction between both mechanisms of transfer. Power is the amount of work per unit of time. The measure of the power is Watt (W). Also Horsepower (HP) is commonly used, and 746 W are equal to1 HP.

In thermodynamics science, the amount of heat transfer is considered as a system that undergoes a process from one equilibrium state to another without considering the time required. On the other hand, for engineering the heat transfer is determined by the rates of such energy transfers. The transfer of energy between two objects is occurring because of the difference of temperatures. Energy is transferred from the warm object toward the cold one. It means the transfer of energy is continuous from the higher temperature medium to the lower temperature medium. The process stops when the two temperatures of the mediums are the same. According to thermodynamics, the energy exists in various forms.

In the science of heat transfer, the rates of heat transfer to or from a system, times of cooling or heating, and variation of the temperature can be determined.

The study of heat transfer cannot be based on the principles of thermodynamics alone, because there is a physical difference between heat transfer and thermodynamics. The heat transfer is a non-equilibrium phenomenon that deals with systems. It lacks of thermal equilibrium while thermodynamics deals with equilibrium states and changes from one equilibrium state to another. Nevertheless, the laws of thermodynamics constitute the main base for the science of heat transfer.

The first law requires that the rate of energy transfer into a system is equal to the rate of increase of the energy of that system.

The second law requires that heat is transferred in the direction of decreasing temperature.

The energy required to raise the temperature of a unit mass of a substance by one degree is called *specific heat*. In general, this energy depends on how the process is executed. In thermodynamics, there are two kinds of specific heat.

In the first kind, the energy required to raise the temperature of a unit mass of a substance by one degree with the volume hold constant is called the specific heat at constant volume (C_V).

In the second kind, the specific heat at constant pressure is the energy required to do the same process with the pressure hold constant (C_p).

A common unit for specific heats is kJ/kg·°C or kJ/kg·K in which 1°C change in temperature is equivalent to a change of 1 K; ΔT(°C)=ΔT(K). In general, the temperature and pressure, two independent properties of a substance, affect specific heat. At low pressures all real gases approach ideal gas behavior. However, the specific heat changes in temperature only for an ideal gas (Figure 1.1).

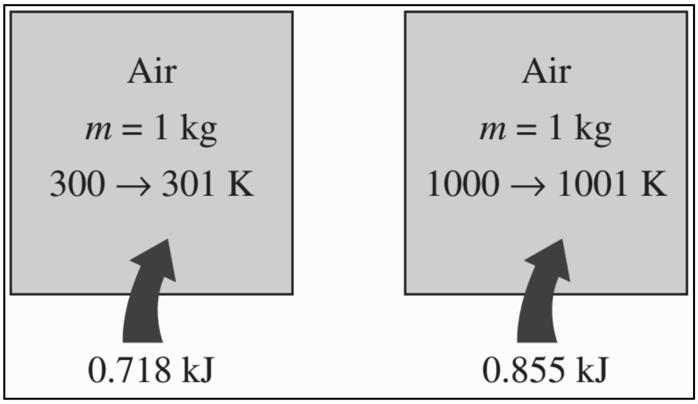

Figure 1-1 Specific heat changes in temperature (Çengel, 2002).

The internal energy of bodies represents heat through sensible and latent forms. In thermodynamics, to prevent any confusion with heat transfer those forms of energy are usually referred as thermal energy. The heat addition means that heat transfer is done via absorption of heat by the mass, while heat rejection means the heat transfer occurs through release of heat from the mass.

Sensible energy or sensible heat is a portion of the internal energy of a system associated with the kinetic energy of the molecules, considering that internal energy is the sum of the kinetic and potential energies of the molecules. In other words, sensible heat is the amount of energy absorbed or released (Q) from the material to determine the change in temperature (Δt). It is expressed in the following equation:

$$Q = m * c * \Delta t$$

Where:

c: Specific heat (J/kg.K),

m: Mass quantity of body (kg).

The methods of sensible heat transfer are divided into three different processes: conduction, convection and radiant heat transfer.

Latent energy or latent heat is the internal energy associated with the intermolecular forces between the molecules that bind the molecules to each other in a system. In other words, latent heat can be described as an amount of energy added or offered to a system necessary to change from one state to another or a phase change. In the thermodynamic system, during the phase change, the added or subtracted energy from the system does not raise or lower the temperature of the system itself. That energy will overcome these molecular forces. Depending on the type of phase transition considered, different types of latent heat can be distinguished: the latent heat of fusion, the latent heat of vaporization and latent heat of sublimation (Fermi, 1972). The amount of energy absorbed or released from the material during its transformation from one state to another (Q) is expressed in Joule and is given in the following equation:

$$Q = L * m$$

Where:

m: Mass quantity (kg).

L: Latent heat of the material [it varies for each substance] (J/ kg).

Methods of latent heat transfer are very complex issues because they suffer the effects of parameters such as air velocity and relative humidity which are not yet specified. Thus, the basic requirement for heat transfer between two systems is the temperature difference that is considered the driving force for heat transfer. So heat transfer does not occur between systems that have the same temperature and the main direction of the heat transfer from the higher temperature towards the lower one. The greater the temperature difference, the higher the rate of heat transfer (Fermi, 1972).

As previously mentioned, the total amount of heat transferred Q during a time interval Δt depends on the heat transfer rate. The heat transfer rate is the amount of heat transferred per unit time $\bar{Q}$ and is expressed in Joules per second (J/s), or Watt (W):

$$Q = \int_{0}^{\Delta t} \bar{Q} * \mathrm{dt}$$

Provided that the variation of $\bar{Q}$ with time is known. For the special case of $\bar{Q}$ = constant, the equation above reduces to

$$Q = \bar{Q} * \Delta t$$

The first law of thermodynamics for a closed system is the following:

$$\bar{Q} = Wk + \frac{dU}{dt}$$

Where:

$\bar{Q}$: heat transfer rate, positive toward the system,

Wk: work transfer rate, positive away from the system,

$\frac{dU}{dt}$: rate of change of internal thermal energy, U, with time, t,

It is positive when the system's energy increases.

If pdv work is the only work occurring, then:

$$\bar{Q} = p\frac{dv}{dt} + \frac{dU}{dt}$$

Where:

p: pressure,

$\frac{dv}{dt}$: rate of change of volume with time,

$\frac{dU}{dt}$: rate of change of internal thermal energy with time.

Special cases:

The heat transfer rate with the constant volume process is:

$$\bar{Q} = \frac{dU}{dt} = m.Cv.\frac{dT}{dt}$$

Where:

Cv: specific heat at constant volume,

m: mass quantity of body (kg),

$\frac{dT}{dt}$: rate of temperature with time.

The heat transfer rate with the constant pressure process is:

$$\bar{Q} = \frac{dH}{dt} = m.Cp.\frac{dT}{dt}$$

Where:

H = U + pv is the enthalpy,

Cp: specific heat at constant pressure,

$\frac{dT}{dt}$: rate of temperature with time.

When the volume and pressure are constant the two specific heats are equal:

$$Cp = Cv = C \leftrightarrow \bar{Q} = m.C.\frac{dT}{dt}$$

Thus the total amount of heat transferred is:

$$Q = m.C.\Delta T$$

1.2.2 Conduction heat transfer

The difference in the energies of adjacent particles of two substances leads to the transfer of energy from the more energetic particles to the adjacent less energetic ones as a result of interactions between them. This transfer is called conduction heat transfer. The heat exchange of conduction is depending on the temperature difference and occurs without sensitive mass displacements.

Heat flows from the high-temperature zone to another at lower temperature within the body that is not thermally homogeneous across the contact points. In solids, energy transport is done by free electrons as a result of the combination of vibrations of the molecules in a lattice. Whereas, in stationary fluids it is a consequence of higher-temperature molecules interacting and exchanging energy with molecules at lower temperatures. The geometry, material, and thickness of the medium, as well as the temperature difference across the medium, are factors affecting the rate of heat conduction through a medium (Çengel, 2002).

Fourier's law is the basic equation for the analysis of heat conduction transfer. Heat flux is created during a heat transfer process by conduction. Its intensity grows with the increasing of temperature difference of molecules in the same body (Fracastoro, 2000). This heat flux $\bar{Q}$ (W/m^2) is determined by knowing the thermal conductivity λ (W/m.K or J/m·s·K) of the material, the surface area A (m^2), the temperature difference and the distance Δx (m) between the ends of the surfaces (thickness). Thus, Fourier equation becomes:

$$Rate\ of\ heat\ conduction\ \alpha\ \frac{(Area)(Temperature\ difference)}{Thickness}$$

$$\bar{Q} = -\lambda \frac{\Delta T}{\Delta x}$$

Where:

$\bar{Q}$: Heat transfer rate in the x direction per unit area perpendicular to the direction of heat flow.

$\frac{\Delta T}{\Delta x}$: Temperature gradient in the direction x.

Consequently, the rate of heat conduction transmitted through a surface is compatible with the following equation:

$$\bar{Q} = -\lambda * A \frac{\Delta T}{\Delta x}$$

Negative signal accords with the second law of thermodynamics in terms of the heat flux toward the molecules with lower temperatures and therefore unlike thermal gradient.

Whereas, the heat flow in a cylinder is associated with the physical characteristics of the tube such as length, internal and external diameter as well as the surface of the tube. Then the heat transfer equation become:

$$\bar{Q} = \lambda * A \frac{\Delta T}{ln(\frac{r2}{r1})}$$

Where:

$$A = 2\,\pi\,L$$

L: Length of the tube, r_2: External radius, r_1: Internal radius.

According to the final equation for a surface, the rate of heat transfer through a unit thickness of the material per unit area per unit temperature difference is called thermal conductivity. Thermal conductivity is a thermo-physical property of the material, which is the same in all directions in isotropic materials. Otherwise, the thermal conductivity λ is considered a measure of a material's ability to conduct heat. Consequently, material that has a high thermal conductivity value conducts heat more than the ones with low conductivity, and it indicates that this material is a good heat conductor. Whereas, a low value indicates that the material is a poor heat conductor (Çengel, 2002).

Table 1.1. Approximate values of the thermal conductivity (λ) of some materials (Çengel, 2002).

Materials	**Thermal conductivity λ (W/m.K)**
Diamond	2300
Copper	400
Aluminum	240
Steel C	40 - 60
Stainless steel	15
PVC pipes	2
Glass	1 - 1.5
Bricks	0.7
Water	0.6
Gas	0.02 - 0.2

In addition, value of thermal conductivity can change from one material to another because it depends on the atomic and molecular structure of the material. In general, thermal conductivity increases with the density (Table 1.1). Therefore, thermal conductivity of gases usually is lower than liquids, as well as this property of liquids is lower than solids (Bejan and Kraus, 2003). As exposed in the Figure 1.2, in the solid phase, the thermal conductivity is the highest, while in the gas phase is the lowest. Only for gas situation the increase of temperature and the decrease of molar mass lead to higher thermal conductivity.

The heat conduction analysis takes into consideration that the thermal conductivity rate is fixed because the changes of thermal conductivity with a variation of temperature are too complicated. Moreover, thermal conductivity is proportional to the amount of heat transmitted. In general, the efficiency of air conditioners depends mainly on the thermal properties and on the ability of the cooling mediator to transfer heat.

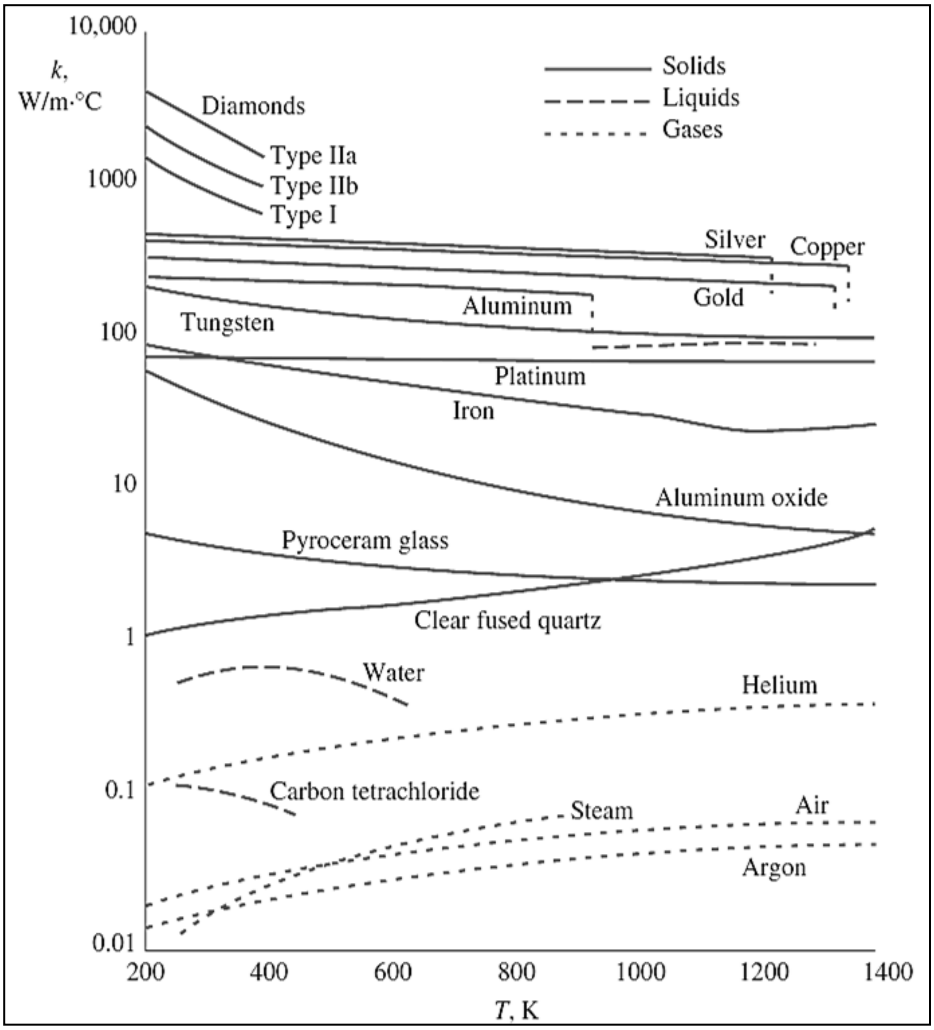

Figure 1-2 The variation of the thermal conductivity with temperature (White, 1988).

1.2.3 Convection heat transfer

The energy transfer process by convection can be observed between two surfaces in two different state of material. In most of the real situation, one of them is solid and another could be liquid or gas that is in motion. Then the mechanisms of convection are more complicated than the mechanisms of heat transfer by conduction, since the transfer of energy from the solid surface to the other adjacent is done through a two-stage. That occurs by means of a combined action of conduction energy and transports by moving particles. Such process of carrying heat away by a moving fluid is called *convection* (Lienhard IV and Lienhard V, 2006). Heat transfer between a solid surface and the adjacent fluid is by pure conduction as soon as the fluid motion is absent, because the liquid becomes a strict material. In the presence of motion of the fluid, the heat transfer between the solid surface and the fluid is enhanced, but it also complicates the determination of heat transfer rates.

When the cold air is beside a solid surface at high temperature, the heat-transfer between a solid surface and the adjacent molecules of air (air surface in contact with the solid surface) occurs by conduction. This energy leads to increase internal energy of the particles. Subsequently, it includes the effect of random motion of molecules and macroscopic motion. This heat is then carried away from the surface solid by convection. Accordingly, the random motion of molecules leads to the transfer by conduction within the air, while the macroscopic motion of the air removes the heated air near the surface and replaces it by the cooler air. Ultimately, the process is not only dependent on the temperature gradient but also on the microstructure of the substance motion.

From the practical viewpoint, the convection can be classified into two types (Figure 1.3). It is called natural convection if the convection occurs as a result of the forces of buoyancy, or due to temperature differences. The density changes near the surface that cause the motion of the fluid. On the other hand, forced convection occurs when the relative motion between the fluid and the surface is induced from the outside by different means. They can be of natural origin such as wind or anthropogenic as fans. Convection can occur in two ways: internal convection when the fluid flows inside a duct (typically a pipe) in such a way that the presence of the wall causes effects on the motion of the entire fluid. Otherwise, there is external convection when the fluid touches the outside of an object (the wing of an airplane, the blade of a turbine), and, at a sufficient distance from the object, it is not influenced by the wall itself.

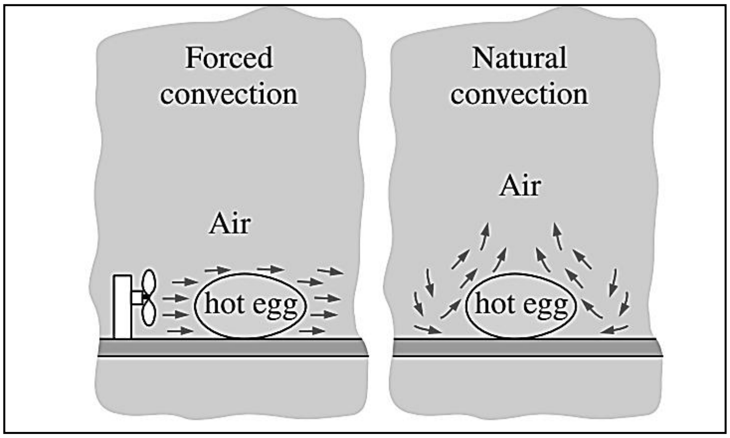

Figure 1-3 The difference between forced and natural convection (Çengel, 2002).

Furthermore, the convection, both forced and natural, can be single-phase or phase change when the fluid changes phase (evaporation or condensation) because of the contact with the surface (Marco and Forgione, 2007). Apart from this difference between forced and normal convection, the rate of heat transmitted by convection is conveniently expressed by Newton’s law:

$$\bar{Q} = h.A.(Ts - Tf)$$

Where:

$\bar{Q}$: Heat transfer rate by convection in Watt (W) during the unit of time (s).

h: Convective transfer coefficient (W/ m^2.K).

A: Exchange surface area (m^2).

$(Ts - Tf)$: Kelvin temperature difference between the temperature of surface lapped by the fluid (Ts) and fluid temperature (Tf).

The convective transfer coefficient depends on several factors such as the surface geometry (flat, curved, spherical, etc.), position of the surface (vertical, horizontal, inclined), type of fluid (gas or liquid), properties of fluid (density, viscosity, temperature specification, velocity) and the type of motion (laminar flow, turbulent flow). Anyway, it is possible to identify typical values of convective transfer coefficient in several conditions as shown in Table 1.2.

Table 1.2. Typical values of convective transfer coefficient in several conditions (Marco and Forgione, 2007).

<table>
<tr><th>h
[W/m^2 K]</th><th>Natural convection</th><th>Forced convection</th><th>Convection with phase change (boiling, condensation)</th></tr>
<tr><td>Liquid</td><td>50 - 2000</td><td>100 - 20 000</td><td rowspan="2">2500 - 100 000</td></tr>
<tr><td>Gas</td><td>2 - 25</td><td>25 - 250</td></tr>
</table>

1.2.4 Radiation heat transfer

Radiation is a phenomenon due to the changes of the electronic configurations of the atoms or molecules of all bodies (gas, liquid or solid) that emit or absorb the energy, transported from the material in the form of electromagnetic waves. Mechanism of heat transfer by radiation is very different from the mechanisms of conduction and convection because there is no contact between the bodies involved and it does not require an intervening medium for the transportation of energy. Thus, thermal radiation is as a result of a temperature difference (Holman, 2010). Moreover, at temperature above absolute zero, all bodies emit thermal radiation. The rate of radiation that can be emitted from a surface (A_s) at absolute temperature (T_s) can be calculated by the *Stefan–Boltzmann law*:

$$\bar{Q}_{emit} = \sigma . A_s . T_s^4$$

Where: σ: *Stefan–Boltzmann constant* [σ = 5.67 ×10^{-8} W/m^2·K^4].

If the surface is ideal, the rate of radiation emitted from a surface is the maximum possible value. This surface is called blackbody. At the same temperature, all real surfaces emit a radiation less than the blackbody and radiation emitted is expressed as:

$$\bar{Q}_{emit} = \sigma . \varepsilon . A_s . T_s^4$$

Where: ε is the emissivity of the surface. (ε = 1 for blackbody).

Whereas, the property emissivity for the rest of surfaces ranges between $0 \leq \varepsilon \leq 1$. In general, a surface has important radiation properties: not only emissivity, but also absorptivity, which is the amount of radiation of energy absorbed by the surface, relative to the total amount of energy incident on the surface. The value of surface absorptivity, like emissivity, is in the range $0 \leq \alpha \leq 1$. In addition, the amount of radiation energy absorbed by a surface as form of radiation energy is given by the equation:

$$\bar{Q}_{abs} = \sigma.\ \alpha.A_s.T_{sur}^4$$

Where: T_{sur} is also known as an equivalent temperature of the blackbody that is mainly a function of relative humidity and air temperature.

Moreover, a blackbody which is considered perfect absorber and emitter absorbs all radiation incidents on it. In addition, the temperature and the wavelength of the radiation govern both emissivity and absorptivity of a surface. According to the *Kirchhoff's law* of radiation states, at a given temperature and wavelength, the α and ε are equal (Çengel, 2002).

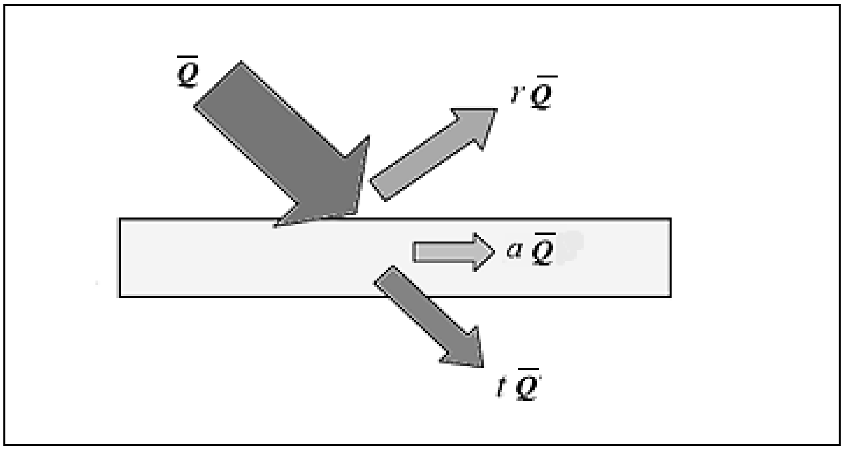

Figure 1-4 The radiation incident on a material (Marco and Forgione, 2007).

When the radiation is incident on a body, a portion of it is reflected, another absorbed and in some cases transmitted as shown in Figure 1.4. The sum of absorbance (α), reflectivity (г) and transparency coefficients (τ) are equal to one.

$$\alpha + г + \tau = 1$$

Otherwise, if the body has an opaque surface, the radiation incident will be reflected and absorbed only (Figure 1.5).

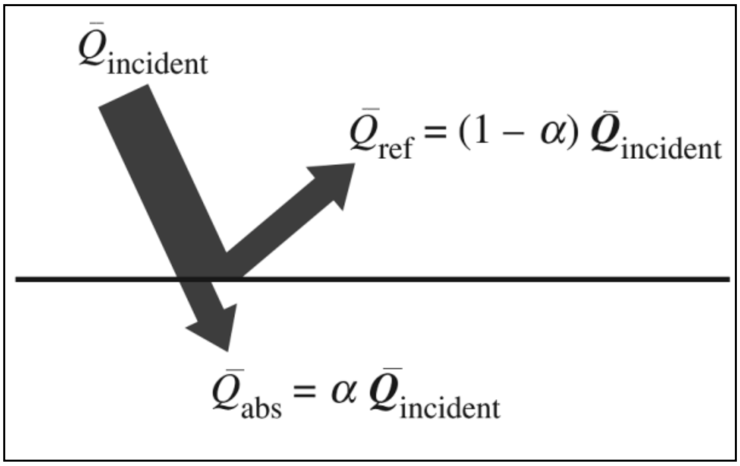

Figure 1-5 The radiation incident on an opaque surface (Çengel, 2002).

In general, the phenomenon of absorption and radiation occur simultaneously in all objects. Therefore, the net radiation heat transfer is the difference between the rates of radiation emitted by the surface and the radiation absorbed.

If the rate of radiation emission is greater than the rate of radiation absorption, the surface is losing energy by radiation. Otherwise, the surface gains energy by radiation.

Then, the net rate of radiation heat-transfer in Watt (W) between the surface of the body and the medium that surrounds it can be calculated as follow:

$$\bar{Q}_{rad} = \bar{Q}_{emit} - \bar{Q}_{abs} = \sigma.\ \varepsilon.A_s.\ (T_s^4 - T_{sur}^4)$$

However, this is a special case because it occurs without any effect on the net radiation heat transfer by the emissivity and the surface area of the surrounding surface. Anyway, in the normal cases, the radiation is conjugated with thermal conduction or convection. Therefore, the total rate of heat transfer is determined by including the effects of both convection and radiation as:

$$\bar{Q}_{total} = h_{combined}.A_s.(T_s - T_{\infty})$$

where: $h_{combined}$ is a combined heat transfer coefficient.

When the surfaces involved have low emissivity and low moderate temperatures, the radiation together with forced convection can be left out. Nonetheless, these surfaces are usually significant relative to conduction or natural convection.

1.2.5 Soil and heat transfer

The heat is generally transferred in the soil in the three aforementioned methods: conduction, convection and radiation. These mechanisms of heat-transfer vary depending on the components of the object. In this context, the various components of the soil lead to different values of thermal conduction. So, thermal conduction of the soil depends on the types of constituent metals, content of organic matter, moisture content and porosity (Farouki, 1981). It also varies according to the proportion of air and water in the soil, as well as it changes with the structure of the soil depth. Thus, the heat-transfer in the soil is a function of depth and time. Consequently, as established in the Table 1.3, the thermal properties of the components of the soil play an essential role in determining the amount of heat transmitted.

Table 1.3. Thermal properties of the components of the soil at 20°C and 1 atm (Hillel, 1982).

Component	Density **ρ (kg / m^3)**	Volumetric thermal capacity ***C h* (MJ/m^3.K)**	Thermal conductivity **λ (J/m.s.K)**	Thermal diffusivity **D_T (m^2/s)**
Quartz	2650	1.94	8.410	43×10^{-4}
Minerals	2650	1.92	2.930	15×10^{-4}
Organic	1300	2.51	0.251	1.0×10^{-4}
Water	1000	4.18	0.594	1.4×10^{-4}
Air	1.25	0.00125	0.026	2.1×10^{-4}

In addition, the thermal properties of the soil are influenced by climatic conditions and several other factors (Heusinkveld *et al.*, 2004). The following thermal properties have to be known to determine the rate or amount of heat transfer in the soil.

Volumetric thermal capacity of soil **($C\,h$).** It is defined as the change in heat content of the unit volume of soil from the virtual unit variation of heat. It is expressed in J/m^3/K or in cal/cm^3/°C. Its value depends on the components of the soil (solid materials, organic materials, virtual density, moist temperature of the soil, etc.). So, the heat capacity can be calculated by collecting the heat capacities of all components in the volume according to the equation (De Vries, 1975):

$$C\,h = f_m\,C_m + f_o\,C_o + f_w\,C_w + f_a\,C_a$$

Where:

f is the volumetric fraction of each one component,

C_m thermal capacity of minerals,

C_w thermal capacity of water,

C_o thermal capacity of organic materials,

C_a thermal capacity of air.

Thermal capacity is an indicator of the ability of a material to store heat per unit volume. For any components, thermal capacity is obtained by taking the product of density and specific heat.

$C_m = \rho_m\,C_{mm}$	C_{mm}	specific heat of minerals
$C_w = \rho_w\,C_{mw}$	C_{mw}	specific heat of water
$C_o = \rho_o\,C_{mo}$	C_{mo}	specific heat of organic materials
$C_a = \rho_a\,C_{ma}$	C_{ma}	specific heat of air

The mineral soil is ideal: $C\,_h$ = 1 MJ/m^3.K if dry, or $C\,_h$ = 3 MJ/m^3.K if saturated. The density related to moisture content is shown in Figure 1.6.

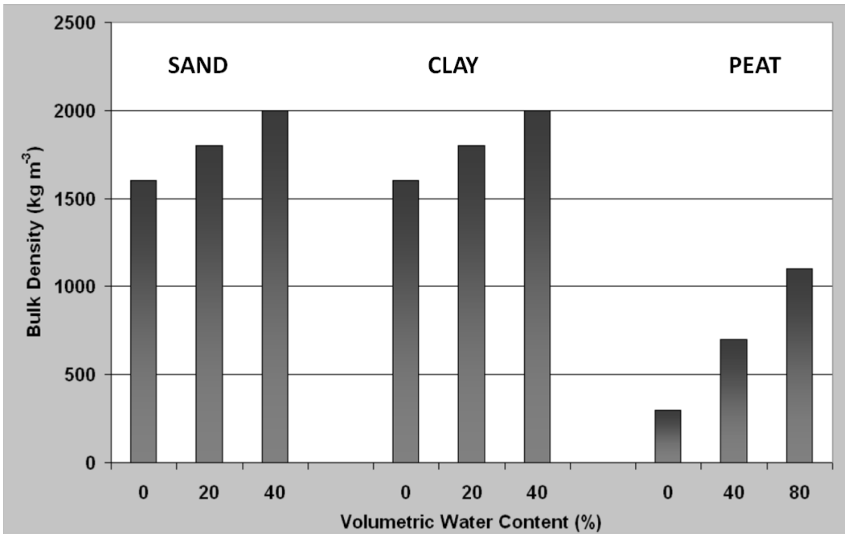

Figure 1-6 Changes in the density of the components of the soil with moisture content (Monteith and Unsworth, 1990).

Thermal conductivity (λ) is defined as the quantity of heat transmitted through a unit area in unit time under gradient temperature equal one. Furthermore, thermal conductivity of the soil components changes so much between one and the other that the average thermal conductivity of the entire soil depends on the content of mineral, organic and volume fraction of water and air.

Moreover, thermal conductivity of air is very low compared to other components as well as the physical characteristics and thermal properties of air and water that always change, so thermal conductivity changes with time (Ghuman and Lal, 1985; Wang *et al.*, 2005). If the soil is wet, the convection heat-transfer takes place with the latent heat of evaporation. Thus, during the motion of water or water vapor in the soil, while in colder areas heat is released during condensation, in some others it is absorbed through evaporation of liquid water.

Since the latent heat transfer cannot practically be separated from the thermal conduction in moist soil, the soil thermal conductivity should be considered the apparent thermal conductivity of the soil and it is calculated as follows (Hillel, 1998):

$$\lambda = \lambda' + DT\, vapor \times L$$

Where:

λ' : Instantaneous thermal conductivity,

DT: Thermal vapor diffusivity,

L: Latent heat of vaporization (2.449 MJ/kg).

As a result of this situation, as exposed in Figure 1.7, the thermal conductivity of the soil is significantly affected by moisture content: dry soil (λ=0.125–0.209 J/m.s.K), wet soil (λ=0.836–1.674 J/m.s.K).

Many factors affect the thermal conductivity compared to the heat capacity in the normal moisture soil. If the heat capacity varies C_h 3-4 times, the thermal conductivity λ varies 100 times or more.

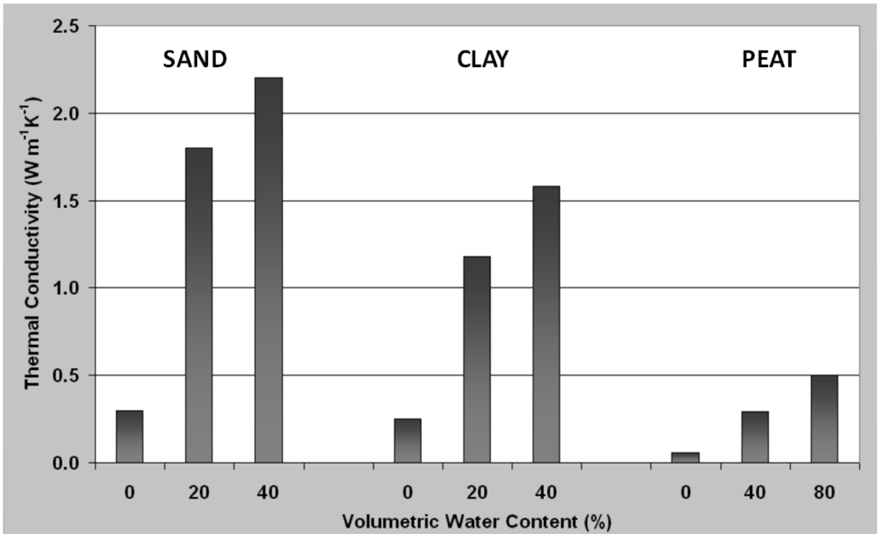

Figure 1-7 Changes in the thermal conductivity of the components of the soil with a moisture content (Monteith and Unsworth, 1990).

Thermal diffusivity (DT). Sometimes the thermal diffusivity is requested in place of thermal conductivity. It can be defined as the change of temperature in a unit of density of quantity of heat that passes through the volume in unit of time under thermal gradient equal one. In other words, it could be defined as the ratio between thermal conductivity and volumetric specific heat.

$$DT = \frac{\lambda}{Ch}$$

It is necessary to say that the specific heat of solid material and water must be taken into account when the volumetric heat capacity is calculated. In addition, the moisture content in the soil has a significant impact on the thermal diffusivity of the soil as shown in Figure 1.8.

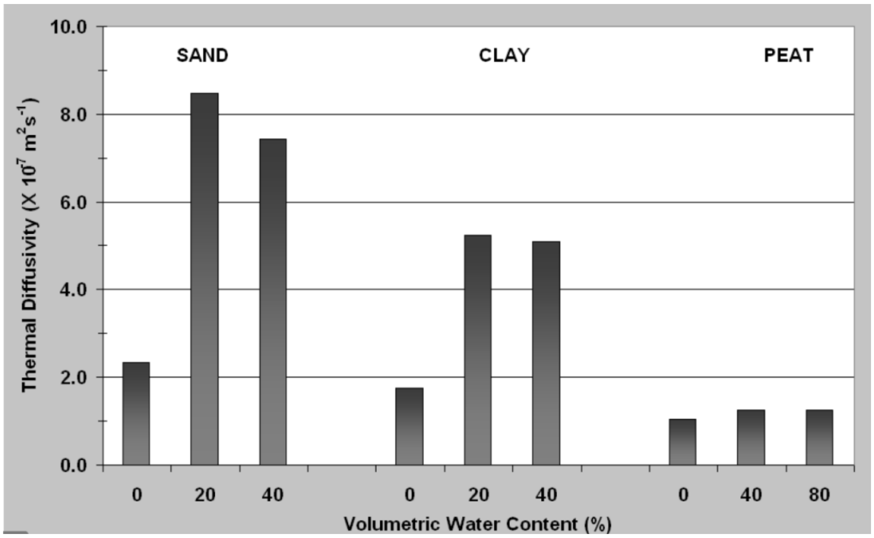

Figure 1-8 Changes in the thermal diffusivity of the components of the soil with a moisture content (Monteith and Unsworth, 1990).

1.2.6 Main assumptions made in this thesis work

This study is focused on heat, which is the form of energy that can be transferred from a system to another one as a result of temperature difference (Çengel, 2002).

A thermodynamic analysis is used to determine the amount of heat transfer for any system undergoing any process, without how long the process will take. In this situation, it is known how much heat must be transferred to get a specified change of state to maintain the principle of energy conservation.

In this study it is important to calculate the amount of convection heat transfer by a gain or a loss of heat of air passing through a tube. The rate of gaining or losing heat by convection by the fluid when it passes within the tube can be calculated by equation:

$$\bar{Q} = m'.Cp.(T2 - T1)$$

Where:

m': fluid mass flow rate,

Cp: specific heat at constant pressure.

$$m' = \rho.\nu.As,$$

Where:

ρ: density of the fluid (kg/m^3),

ν: velocity of the fluid (m/s),

As=π r^2 area of tube section (m^2).

It is also necessary to know the amount of heat transmitted in the soil by heat conduction. The amount of heat lost from the fluid itself is acquired by the wall tube (Figure 1.9). The equation is the following:

$$\bar{Q} = m'.Cp.(T2 - T1) = h.A.(Ts - Tf)$$

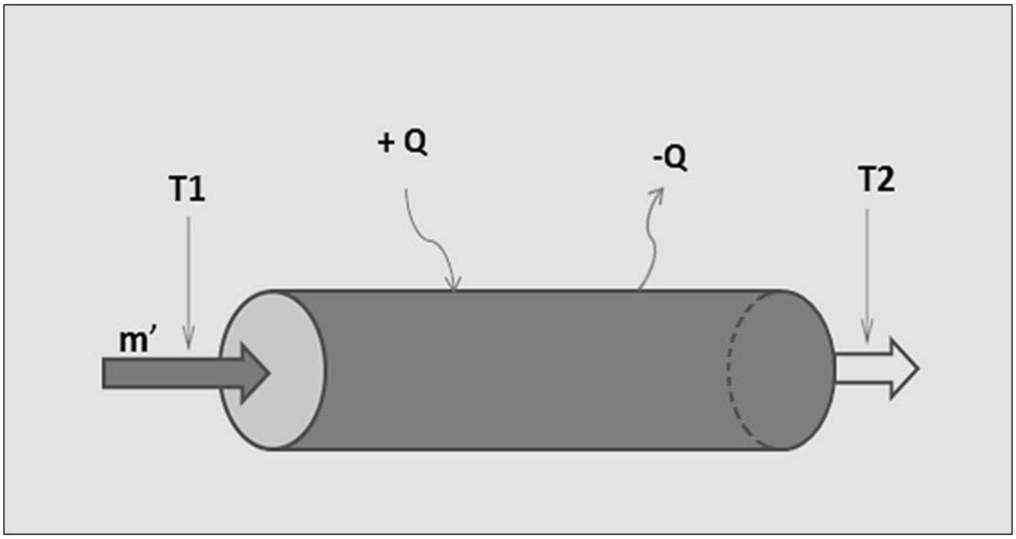

Figure 1-9 Convection heat transfer in a tube.

Based on previous explanations, to increase the thermal efficiency of soil the thermal properties such as conductivity, diffusivity and density should be improved. Therefore, the key idea in this thesis was to increase the thermal conductivity of all the components of the underground pipes system in order to raise much is possible the performance of the air conditioning plant.

1.3 Influence of climate on animals

The climate has an important influence on all animals, especially the farming animals. There are two kinds of animals: cold-blooded and warm-blooded. The cold-blooded animals are very active at high temperatures but they react with hibernation in other conditions. Otherwise, the warm-blooded animals change slightly their body temperature when a noticeable variation of the air temperature occurs. In other words, they have the ability to regulate body temperature within a narrow field versus large change in the surrounding air temperature.

Because most of their heat is due to the thermogenic endogenous processes, it is necessary that the heat produced is ready to maintain constant their body temperature. If the outside temperature is comfortable and there are conditions of thermal neutrality (neither hot nor cold), thermogenesis and physiological thermolysis are equal. Consequently the animal spends little of thermal energy to get the thermoregulation (Sastry and Thomas, 1987).

In cold environment, the animals adopt several systems of protection. The individual in the herd tries to rally side by side in a place protected from the wind. The animals tend to eat more, because physiological thermogenesis may not be enough to compensate the heat losses. Then, they begin to move by body mechanisms (in the short, medium and long term) to increase the production of heat and reduce losses of it.

As opposed, in hot environment, the heat produced by the body is added with a portion coming from the outside. Animals begin to carry out mechanisms to increase heat loss and they tend to eat less so they produce less metabolic heat. Then, this process leads to a loss of production and a decrease in animals' weight.

The constancy of body temperature is due to the complex metabolic mechanisms, which respond to external temperature variations bringing loss or accumulation of heat. Even when animals are kept in thermally neutral environment, some changes in physiological characteristics occur (Giorgetti and Gallai, 2007).

1.3.1 Effect of heat stress on broilers

Poultry meat is considered one of the most important food commodities in the world. In recent years, production and market of poultry meat in the Middle East has evolved as result of an increased demand since the improvement of the standard of living. The retrieval speed of capital invested in advanced techniques determined a decrease of poultry production costs. Since the poultry meat is a substitute for red meat, people have increased the consumption boosting the expansion of the poultry breeding activity. In the past several problems were related to breeding such as lack of veterinary medicines for diseases, erroneous breeding methods and unavailability of adequate environmental conditions. Today instead, poultry facility has a good productive efficiency.

Poultry production has a less detrimental impact on the environment than other types of livestock and it uses less water. Poultry activity is one of the main sectors to create the basic pillars for strategic economic improvement because it contributes effectively to the food security of the population and it participates to achieving greater self-sufficiency of local animal productions. At the same way, it contributes to the achievement of agricultural development and the improvement of animal farming together with important industrial branches such as feed and veterinary medicines, packaging materials and all types of equipment for poultry production.

Heat stress (HS) in hot and semi-arid areas is one of biggest chronic problems that breeders have to face. Figure 1.10 shows distribution of dry lands in the world. The HS due to high ambient temperatures has been a major factor hindering production of broilers in these areas, especially, in summer months. Many studies have been carried out about detrimental effects of high ambient temperatures on feed intake, mortality rate, growth rate, feed efficiency, rectal temperature and respiratory rate of broilers.

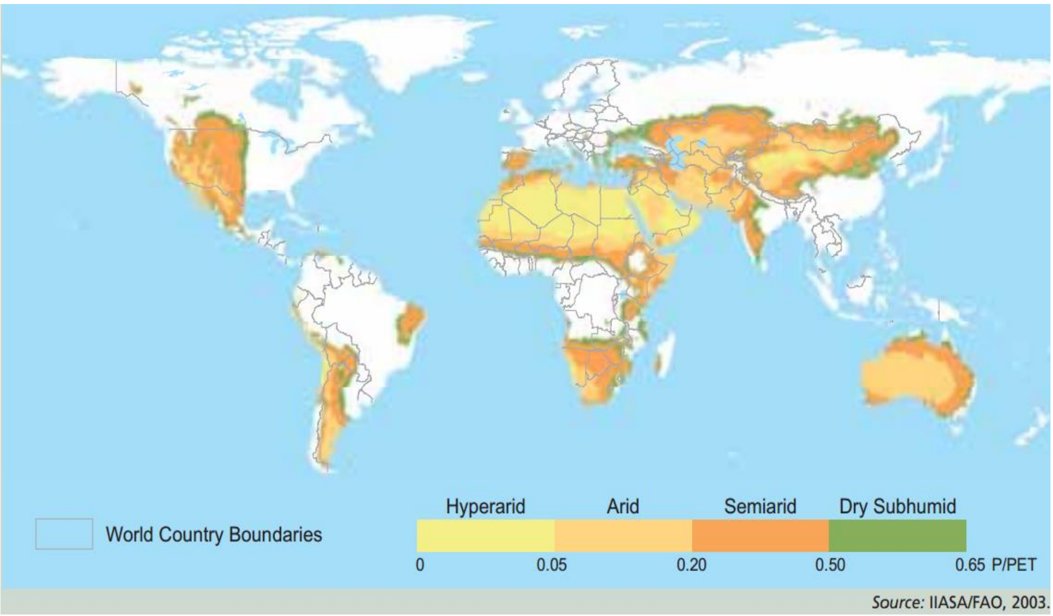

Figure 1-10 Distribution of dry lands in the world (IIASA/FAO, 2003).

Ziad (2006) found that the ambient temperature played an important role in the depressed performance observed. In fact, with outside temperature of 35°C, the rectal temperature and respiratory rate of the birds were not significantly related with the pair-fed and ad libitum fed. The panting increases to dissipate heat because this mechanism requires expenditure of energy resulting in higher feed conversion ratios.

Moreover, the reason for a reduced feed efficiency at high ambient temperature was explained by McDowell (1972): “in warm climates, generally, chemical costs for a unit of production are higher than in cooler climates because a portion is siphoned off for the process required to dissipate body heat”.

Body weight and feed intake were studied at different temperatures (North and Bell, 1990). The results showed that rising of temperature leads to reduced feed intake and body weight (Figure 1.11). At the same time, the water consumption increases with the rise of temperature inside the farm (Figure 1.12).

The loss of bird’s weight at 32°C was of 500g compared to the bird’s weight at 22°C. Furthermore, the nearly doubled water for feed ratio for birds at 32°C compared to birds at 22°C is well documented (Bonnet *et al.*, 1997). Moreover, the difference of means ±SE of live body weight and feed intake of ad libitum fed broilers reared at 20°C and at 35°C is significant. At the same time, the ad libitum birds at 35°C weighed approximately 1350 g less than the same birds at 20°C (Ziad, 2006).

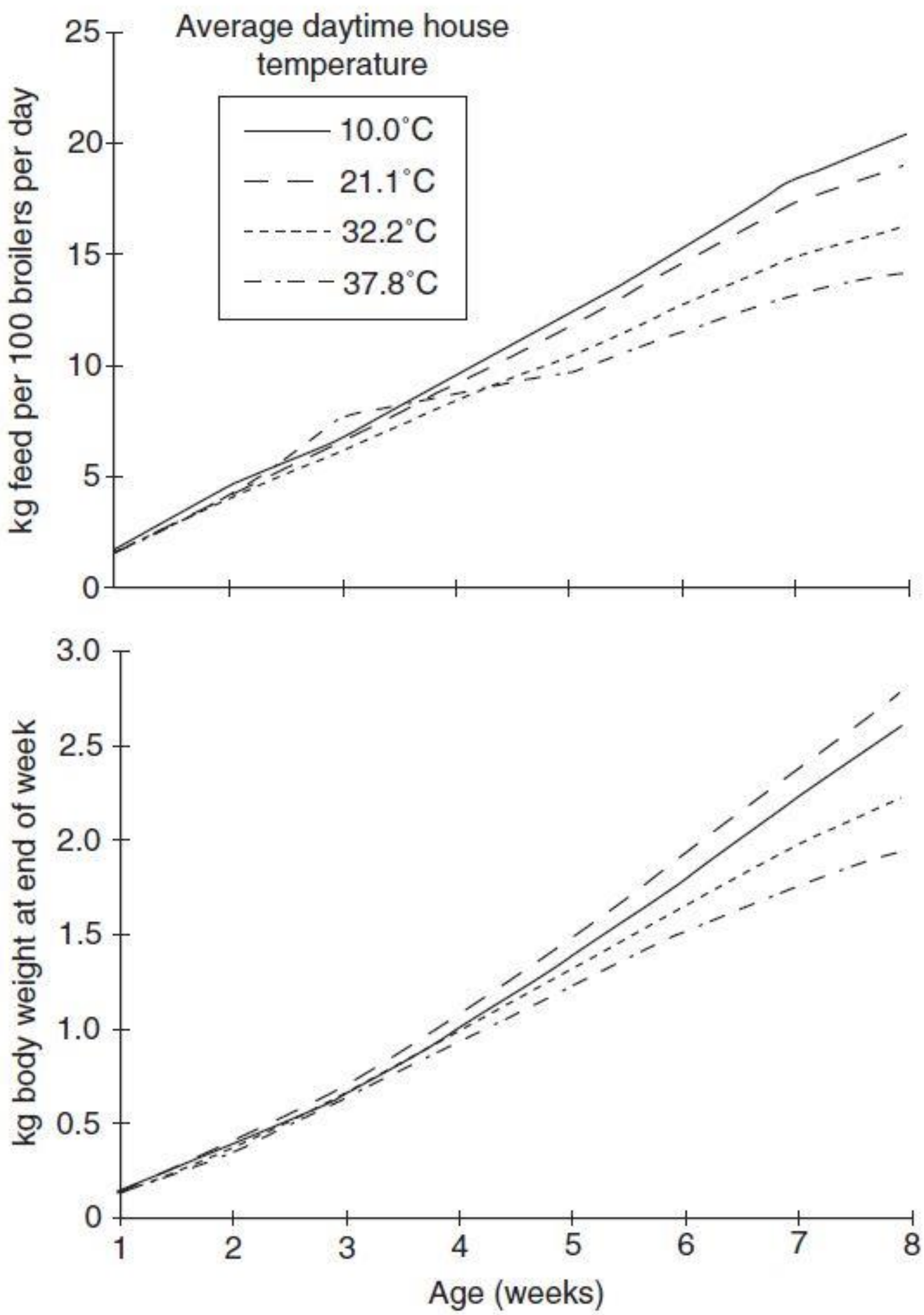

Figure 1-11 Changes of body weight and feed intake of straight-run broilers with different temperatures (Daghir, 2008).

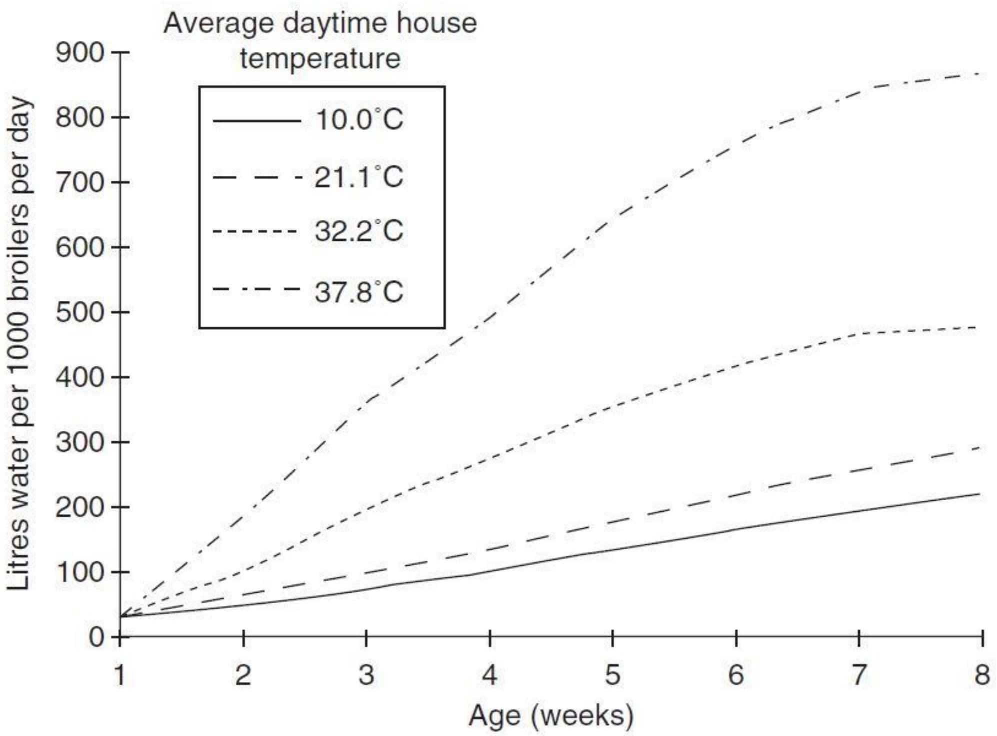

Figure 1-12 Consumption of water with different temperatures (Daghir, 2008).

However, in heat-exposed birds, feed digestibility was reduced, particularly when birds were fed with summer diet. These results indicate a higher direct effect of heat on food digestion. A similar conclusion was observed by Donkoh (1989) and Temim *et al*. (1998), with a significantly reduction in growth and in feed intake, but with a greater reduction in growth compared to the feed intake (Table 1.4).

Table 1.4. Broiler performance from 21-49 days of age when exposed to four different ambient temperatures (Temim *et al*., 1998; Donkoh, 1989).

Response criteria	**Temperature treatment (°C)**			
	20	*25*	*30*	*35*
Initial 3-week-old body weight (g)	640 ±8.60[a]	642 ± 7.88[a]	642 ±7.49 [a]	641 ±8.20 [a]
Body weight gain (g)	1569 ±8.88[a]	1544 ±5.57[a]	1230 ±7.55 [b]	1060 ±8.19 [c]
Food intake (g)	3516 ±15.57[a]	3492 ±7.81[a]	3210 ±4.58 [b]	3063 ±12.49 [c]
Food conversion ratio (food/gain)	2.241 ±0.016[a]	2.262 ±0.004[a]	2.610 ±0.018 [b]	2.889 ±0.013 [c]
Water consumption (ml)	3930 ± 7.55[a]	4010 ±12.77[a]	4680 ± 15.72 [b]	5004 ±9.64 [c]
Mortality	4/48[a]	3/48[a]	3/48[a]	4/48[a]

[a,b,c] *Means (±SD) within a row showing different superscripts are significantly different ($P < 0.0001$).*

Gonzalez-Esquerra and Leeson (2005) explained that the effects of HS on performance of broilers are negative, and the reduced productivity of broilers was closely associated with the intensity of HS and duration of exposure to high temperature. The feed consumption and body weight of broilers subjected to HS (31.4°C) were 370.7 g/day and 1.093 g, much lower than the values for the broiler at thermo-neutrality (20.3°C) which were 526.5 g/day and 1.485 g. These results were obtained between 21 - 42 days of age of broilers.

The HS leads to poor performance, which may be due to lower synthesis and increased degradation of proteins. Furthermore, Zuo *et al.* (2014) are in agreement with the above authors, in the sense that the feed intake and body weight of broilers in the constant heat stress group were significantly lower than in the control group by 15.7% and 24.42% respectively (Table 1.5).

Table 1.5. Growth performance of broilers (Zuo *et al.*, 2014)

Growth parameter	Constant heat stress group	Pair-fed group	Control group
Initial body weight (g)	1294.8 ±10.2	1298.3 ±12.6	1302.1 ±24.7
Final body weight (g)	2540.5 ±29.2^{c}	3029.3 ±32.5^{b}	3361.7 ±45.2^{a}
Average daily gain (g)	59.33 ±1.10^{c}	82.41 ±1.90^{b}	98.03 ±1.24^{a}
Average daily feed intake (g/d)	175.10 ±2.10^{b}	175.52 ±1.80^{b}	207.76 ±2.06^{a}
Feed/ gain ratio	2.95 ±0.04^{b}	2.12 ±0.05^{a}	2.11 ±0.04^{a}

Results are given as mean ± SE
^{a,b,c} *Means within a row with different superscripts differ at P<0.05.*

In broiler chickens, some physiological changes occur due to heat stress such as elevated body temperature and higher panting (Deyhim and Teeter, 1991). At the same time, the change of rectal temperature of broiler chickens in heat stress after a 3h exposure indicates a state of heat shock that showed effect of heat stress on broiler chickens. Lin *et al.* (2006) concluded that the heat stress has fast, direct and severe effects on plasma ascorbic acid, that it is necessary for various biosynthesis. In particular, in the experiment of Al-Ghamdi (2008) in Saudi Arabia, thirty Cobb 500 chickens (22-day-old) were exposed to 40°C, 4 hrs/day for 10 days. The results revealed that HS led to significant (p<0.01) decrease in plasma ascorbic; antibodies levels changed in all stages.

Nevertheless, good overall management of poultry is considered one of the first important steps which leads to optimal poultry production during periods of heat stress. Many solutions based on physical and biological principles have been proposed to reduce the impact of heat stress on broiler. Management considerations include facility design as well as feed and water composition (Teeter and Belay, 1996). Management procedures of food restriction used for broiler chickens during an extended period of heat stress could play an important role in reducing the mortality without affecting body weight while improving food conversion (Yalçin *et al.*, 2001). Moreover, the nutritional strategies minimizing the negative effects of heat stress have been proven advantages (Lin *et al.*, 2007; Gregorio, 1994).

Over time, it has been clear that high ambient temperature is the cause of the heat stress on poultry which is a chronic problem encountered by chicken's breeders in desert and semi-arid areas. In this environment, the effects of heat stress on performance of broilers, such as body weight, weight gain, feed conversion ratio, feed consumption and mortality are negative. These effects increase the discontent of the breeders because of higher costs and lower economic returns.

Furthermore, the high ambient temperature is one of the most important challenges facing the breeders. Thus, to get a comfortable environment that chickens require, it is necessary lowering the temperatures by using well conditioning means together with breeding strategies to reduce the amount of internal heat generated by the animal body.

1.3.2 Effect of heat stress on laying hens

Climatic conditions surrounding the animals have several effects on the well-being and productivity on laying hens (Śottník, 2002). The high temperatures in the summer, as well as high indoor thermal environments, lead to straining and impede laying hens to express their genotypes. In addition, heat stress conditions increase mortality and reduce egg production as well as abate fertility and lack the hatching in main farms.

In Turkey, during the summer seasons of 2003 and 2004 a research about egg poultry housing for the evaluation of heat stress on laying hens was conducted (Karaman *et al.*, 2007). The researchers found that due to thermal stress the daily loss of egg production ranged from 18.48 to 51.32 g egg per hen per day, as well as the daily decrease of dry matter intake ranged from 27.07 to 75.23 g per hen per day for commercial egg production facility. Daghir (2008) achieved similar results by analyzing 12 different references about effects of high temperatures on feed intake in laying hens (Table 1.6).

The daily egg loss and the reduction of dry matter intake is an indication of the negative effects of heat stress on laying hens. The impact of heat stress on the performance of the laying hens is shown in Figure 1.13 where the rising of temperatures increased the weight loss and decreased the dry matter intake.

Table 1.6. Changes in feed intake of laying hens with temperatures (Daghir, 2008).

Temperature (°C)	**Feed intake decrease per 1°C rise (%)**
20	-
25	1.4
30	1.6
35	2.3
40	4.8

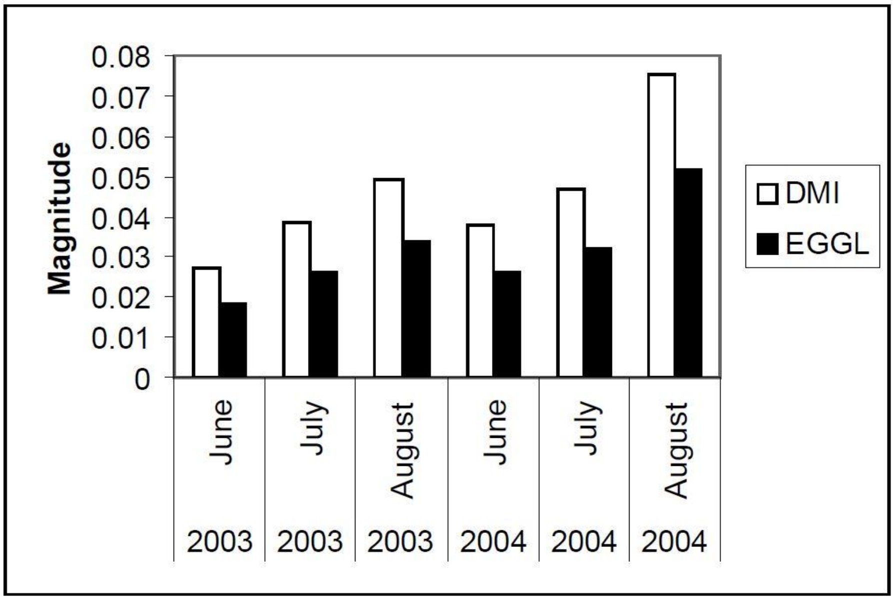

Figure 1-13 Daily loss of egg production and decrease of dry matter intake of a standard laying hen (Karaman et al., 2007).

Mashaly *et al.*, (2004) conducted a study in Egypt about laying hens in three different environmental conditions for 5 weeks to determine the effects of heat stress on them. The three environments were: controlled (average temperature and relative humidity), cyclic (daily cyclic temperature and humidity), and heat stress (constant temperature and humidity). They observed that high temperature and humidity have adverse effects on the live performance and the egg quality in commercial laying hens.

Mortality for the heat-stressed group (31.7%) was much higher than for the cyclic (6.7%) or control (5%) groups. In hens group exposed to constant high temperature, the hen-day egg production was significantly decreased through all 5 weeks compared with those in the cyclic or control groups. This was due to a decreased plasma protein concentration and plasma calcium concentration, conditions both required for egg formation (Mahmoud *et al.*, 1996; Zhou *et al.*, 1998).

The quality of the eggs was significantly affected by low egg weight and shell thickness when the birds were exposed to heat stress. The egg weight and egg shell thickness for heat-stressed group were less than of control group. In average they were 46.9 g and 0.283 mm for the HS hens, 56.4 g and 0.348 mm for the control group (Mashaly *et al.*, 2004). These results agree with those of Kirunda *et al.*, (2001) who found that the high temperatures reduce egg weight.

The decrease quantity of food consumption in laying hens due to heat stress leads to a decrease in body weight, production performance, egg production and egg quality (Emery *et al.*, 1984; Lara and Rostagno, 2013). Corroborating these reports, Lin *et al.* (2004) demonstrated that the effects of heat stress cause decrease of production performance, as well as reduce egg shell thickness. But the decrease in the quality and egg weight and feed intake were negatively affected by heat stress in all strains of laying hens (Jimenez *et al.*, 2007).

Bartlett and Smith (2003) observed a significant reduction in all mentioned parameters during primary and secondary responses during acute heat exposure in broiler chickens. Many studies (Razuki *et al.* 2011; Lin *et al.*, 2006) discussed various strategies to prevent heat stress in broilers such as nutritional, environmental and genetic strategies.

1.4 Ventilation techniques and equipment in the poultry facilities

The conditions applied on the animals, such as surrounding environment and nutrition, have a significant impact on the productivity. It is well known that the temperature is one of the most important environmental conditions in poultry facilities. Therefore, it is necessary to reduce the high temperatures to avoid heat stress of birds. Such thermal stress can be limited effectively using advanced methods of ventilation and air conditioning in poultry facilities.

This section focuses on the economic and environment-friendly technologies for ventilation and air temperature control in farms. However, the amount of air required for ventilation varies based on the type of construction (closed or opened), climatic area, season of the year and system of breeding (cages or floor). In addition, the fluctuations of weather conditions, such as temperature and humidity of the external air, influence not only the type of cooling system, but also the cooling system efficiency.

As a result, the use of cooling systems becomes powerful under proper climatic conditions, but these systems fail under other critical situations, e.g. high humidity and extreme high temperature of the air.

Furthermore, the heat and moisture produced by animals must be taken into account for the design and calculation of ventilation and cooling systems in the facilities.

1.4.1 Natural ventilation systems

Pure and fresh air must be provided regularly to the poultry facility to ensure health conditions and to reduce the amount of heat and humidity caused by animals. Therefore, the use of natural or forced ventilation is essential to achieve the appropriate breeding conditions (Seo *et al.*, 2009). In addition, the selection of the ventilation type depends on the weather conditions surrounding the place of breeding and the efficiency of the ventilation.

There are two important mechanisms of natural ventilation systems used in facilities. The first one is based on the buoyancy strength of the air. Where air density decreases, as a result of the high temperature, it leads the air to rise up. The second one depends on the pressures, velocities and forces caused by the wind. The presence of the upper vents in the buildings allows the exit of hot air, causing rarefaction inside the building that leads the entry of cold air from the side vents. In addition, orientation and dimensions of the building and side vents are crucial to obtain an effective ventilation (Sayegh, 2002).

Several researches have been carried out to determine the optimal dimensions of the building and vents in order to get the best efficiency of natural ventilation (Mutaf *et al.*, 2004; Seo *et al.*, 2009). Mutaf *et al.* (2004) revealed that to increase the efficiency of the natural ventilation system in poultry houses many criteria should be achieved in the design characteristics of natural ventilation and structural dimensions.

These criteria include:

1. the roof slope not less than 20%–30%,
2. building height not less than 4-5 m,
3. building width not exceeding 12 m,
4. continuous roof of ventilation opening,
5. proportion of the outlet area and inlet area at least 1/2 or 1/3.

On the other hand, Sayegh (2002) studied the optimal air exchange. The results show that the area of the side wall openings and upper openings must be at least 15% of the floor area. He also recommended to use openings area up to 30% of the total floor area.

Finally, the proper design of entrance and exit vents of the air in combination with building dimensions and orientation is very important to achieve accepted efficiency of natural ventilation under temperate climatic conditions.

On the contrary, natural ventilation systems are ineffective under a hot climate and also the manual control of ventilation vents becomes difficult. Deficiency of manual control can be compensated by automatic control systems of the vents, but in this case the costs increase.

1.4.2 Mechanical ventilation techniques

In moderate weather conditions, natural ventilation is able to secure a change of air at the required temperature and relative humidity for optimum ventilation within the poultry facility. However, it becomes insufficient with the rise of the air temperature and relative humidity inside and outside the facility.

Therefore, some mechanical tools such as fans, water fogging technique, evaporative pad cooling and underground pipe systems can be used to obtain the proper environmental conditions.

1.4.2.1 Forced ventilation fans

The fans are used in moderate environmental conditions and warm weather to realize two goals. The first goal is to move and to flip the air inside the facility. The second one is to replace the polluted air with clean air in the facility. In the same way, in this method of conditioning, the place of the fan depends on the mode of ventilation. Therefore, three types of mechanical ventilation have been explained by Sayegh (2002):

1. The extraction system (negative pressure) uses the fans to expel the air outside the facility and to favor the air through the side vents.
2. The pressure system (positive pressure) uses the fans to pressure the air within the facility and to expel the indoor air through the vents.

3. The neutral system uses the fans to concurrently enter and exit the air.

Furthermore, air flow and speed are considered as the basic factors to calculate the poultry ventilation and cooling in the facility. Moreover, the optimum ventilation is done by increasing the times of air change (high volume) at low velocity (Bottcher *et al.*, 1998). Although, the flow and speed control of the air inside the facility is easy, the fans alone cannot be enough to ventilate and cool the facility in high temperature climate.

1.4.2.2 Fogging cooling technique

Through the process of evaporation, the water cools the surface where is in contact. The evaporation process removes 2415 kJ of heat in air using just 1 liter of water.

Consequently, evaporative cooling is a very useful tool for the poultry breeding under hot climate. In the fogging system, the water is sprayed through nozzles into the air. Thus very small water droplets form a cloud of fog all over the facility. This cover of small water droplets evaporates by absorbing heat from the air, leading to a decrease in air temperature (Figure 1.14).

The effectiveness of the fogging system depends on good design, which should provide a perfect cover of fog. The distribution and position of the nozzles are the essential key for the formation of a regular cover. So, the effective position of the nozzles should be near the air inlets. However, this type of cooling is used to be integrated in the ventilation tunnel or directly in the ventilation by fans, resulting more efficient and effective. The factual reduction in temperature is produced by evaporative cooling, and the cooling effect of the current of air is generated by the ventilation fans. Together the two systems allow to reach an excellent cooling even in very hot climate. This efficiency is provided when the design and management of the ventilation are correct (Sayegh, 2002).

Figure 1-14 Fogging cooling technique.

The efficacy of the evaporative cooling by water fogging rises with the increasing of the exchangeable surface between the water and the air. As far as the water fogging is well functioned, the efficiency is the best. Bottcher *et al.* (1992) and Singletary *et al.* (1996) studied several factors which affect the efficiency of water fogging technology such as the initial temperature and the relative humidity of the outside air. The possibility of cooling can be better as much as the initial temperature is high. Whereas, the efficiency is high when the relative humidity of the outside air is low.

Thus, the knowledge of the thermal properties of moist air is very important to solve the problems related to air conditioning (Sayegh, 2002). All these properties are determined by the knowledge of only two parameters of the air using the psychometric chart which is an important instrument for air conditioning (Figure 1.15).

The forced movement of the air caused by fans helps in carrying and distributing the mix of water of air. Hence, using the fogging system, the efficiency of the cooling increases over 50% compared to naturally ventilated facility (Anderson and Carter, 2007).

Nevertheless, the evaporative cooling can be useful even in very humid climates, but in this environment evaporative system could be applied mainly during the hottest hours of the day.

Particularly at high humidity, the evaporative cooling efficiency by fogging becomes lower due to less air capacity to evaporate the water, therefore reducing the temperature drop.

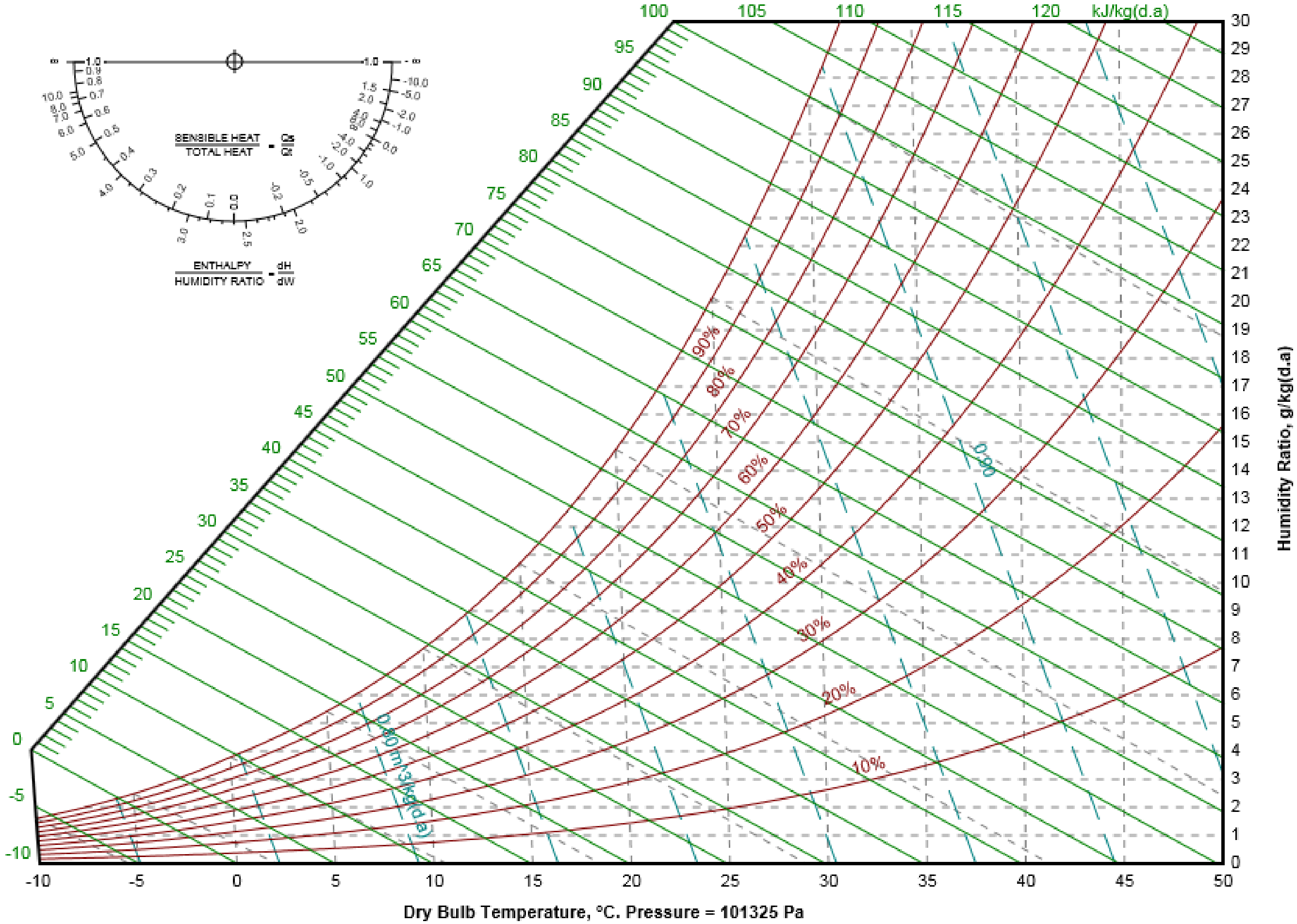

Figure 1-15 Psychometric chart to humid air (Sayegh, 2002).

1.4.2.3 Pad evaporative cooling technique

Evaporative cooling is considered one of the best common economic methods to cool the air in the livestock building and greenhouses. Depending on the physical principle, the transformation process of a material from liquid to vapor requires the absorption of the heat. The evaporative cooling uses the air heat to evaporate the water that is in contact with it. As a result, the air humidity rises and the temperature decreases (Chiappini and Barbari, 1985). In other words, the evaporative cooling decreases the temperature of the air by turning sensible heat into latent heat in the air, through converting the water into vapor. The required energy to evaporate the water is taken from the air in form of sensible heat, which leads to the decreases of air temperature. At the same time, the humidity of the air increases with latent heat gained in the form of water vapor (Albright, 1990).

Fan-pad evaporative cooling technique is called the evaporative cooling system. The process of evaporative cooling is known as an adiabatic process because it occurs at a constant enthalpy value. The transformation of the heat and of the mass occurs when the unsaturated air is in contact with free humidity once they are isolated from any external heat source. In principle, the evaporative cooling system is like the fogging cooling system, but the technique of interaction between the water and the air is different between the two systems. Therefore, in the evaporative cooling system, the air is drawn by the fans through a wetted porous material called pad. This material is installed at the air inlet. The moistening process of the porous material with water is done in several ways; by dotting the water above the upper edge of the porous material, by water spraying or by rotation of the porous material in which the bottom section of the pad is immersed (Sayegh, 2002). Consequently, the hot air flow is forced to pass through the wetted porous material leading to water evaporation, which causes a high humidity, as well as a low temperature of the air as shown in Figure 1.16.

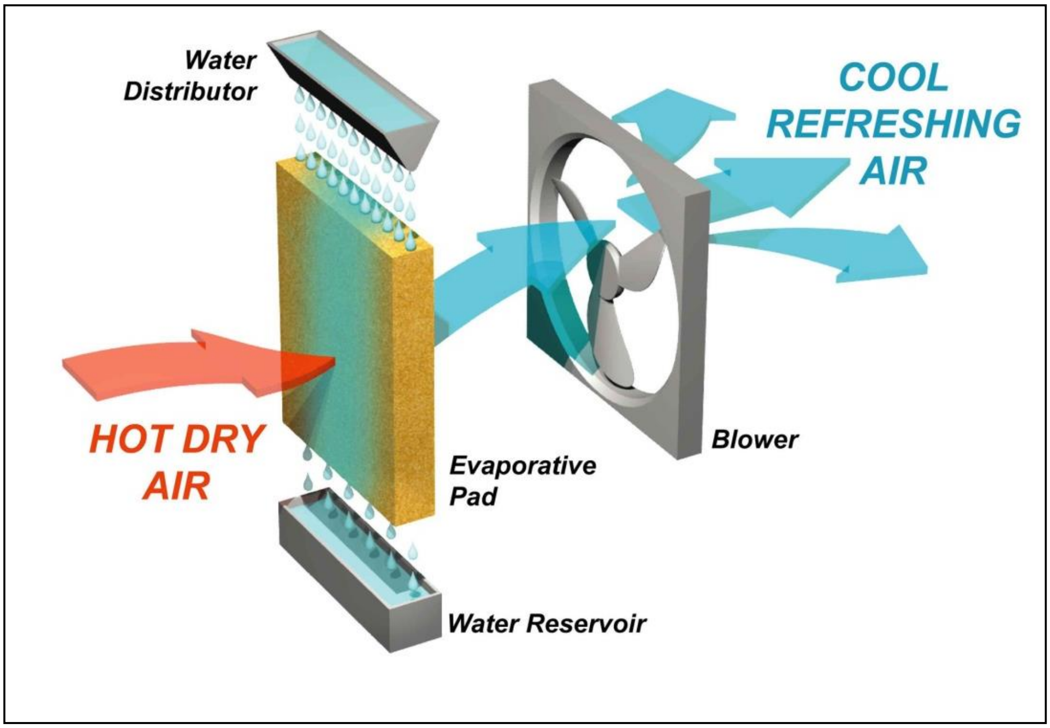

Figure 1-16 Evaporative cooling method by technique with pad and fan.

Evaporative cooling systems are currently considered the most effective and efficient systems for cooling the broiler facility. The cooling efficiency depends heavily on the performance of porous material. Many researches (Gunhan *et al.*, 2007; Alodan and Al-Faraj, 2005) investigated design, type and characteristics of the pad to provide the highest possible efficiency. They found that to get the air to the point near saturation, the hot air must be subjected, with sufficient time, to the largest possible area of wet pad where is passing through it.

In temperate, hot and dry climates like the Middle East, the evaporative cooling technology is the most effective, efficient and economical technique of air conditioning. For these reasons, the evaporative cooling techniques are widespread in poultry facilities, greenhouses, and also residential, industrial, commercial buildings.

Since the high cost of the commercial pad materials (CELdek®) in Turkey, Gunhan *et al.* (2007) studied the suitability of pumice stones and volcanic tuff, both local materials, as alternative materials for cooling pads. The test was conducted taking into account several factors affecting the cooling efficiency such as the speed of air stream passing through the pads, the rate of water flow and the thickness of the pads. Consequently, the authors suggested to take into account the optimal pad size, the highest evaporative saturation efficiency and the lowest airflow resistance possible. The highest evaporative saturation efficiency of fine pumice stones, CELdek®, volcanic tuff and coarse pumice stones were 93.1%, 82.1%, 81.1%, and 76.1%, respectively.

Ideal parameters have been found to be 100 – 150 mm for the thickness of the pad material, 0.6 – 1.6 m/s for air velocity and 1 – 1.75 l/min for water flow rate.

Other studies investigated different types of materials for pad. Dates palm leaves and fibers were tested in Saudi Arabia to be used as an alternative to cellulosic paper pads (Al-Suliman, 2002). In the same environment conditions, testing pads made of galvanized metal sheets, were designed in order to avoid algae and dust to clog the cellulosic pads (Alodan and Al-Faraj, 2005).

In the facility of South Iraq where the experiment of the current research was carried out, traditional pads made of wire mesh and reeds were installed (Figure 1.17).

Figure 1-17 Pads of reeds in poultry facility in South Iraq.

According to statistics obtained from 50 facilities (Kalidar *et al.*, 2010), due to the easy of the use and the low-cost of the evaporative cooling technology, 80% of the poultry breeders in Iraq use this technology for conditioning the ventilation air. The pads of date palm leaves or reeds from marshes are processed manually and they are widely used in South Iraq.

Al-Hilphy and Mishaal (2014) studied the use of the magnetized water for wetting the cooling pad in order to improve the efficiency of evaporative cooling. They found that using magnetized water led to a significant ($p<0.05$) increasing of the cooling efficiency.

The wide use of evaporative cooling in poultry facilities is due to its ability to diminish the heat stress and to enhance gain weight of the animals (Willis *et al.*, 1987). Nevertheless, many researches (Dağtekin *et al.*, 2009; Liang *et al.*, 2012) found that with the use the evaporative cooling in Mediterranean areas it is difficult to reach the optimal temperature necessary in poultry facilities, but the system can help to reduce the negative effects of the heat stress. Also with condition of high humidity in the sunset hours of the morning and in conditions of lack of usable water to moisten the pad, the evaporative cooling pad technique could reduce the temperature about 9°C whatever the temperature is high.

Furthermore, the use of salt water, that is largely available in the region object of study, leads to a short life of the pad as well as speeding up the clogging of the pores in the pad (Figure 1.18).

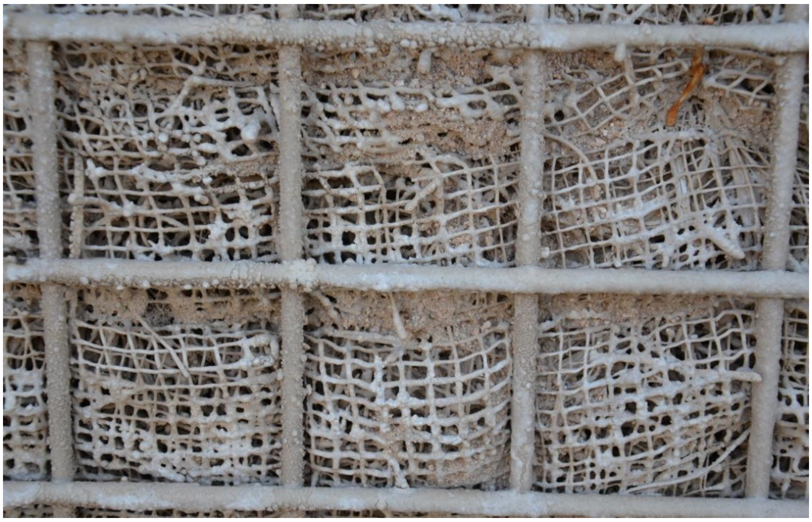

Figure 1-18 Effect of dust and salt water on the pad material (Southern Iraq).

1.5 Earth-to-air heat exchanger technique

Earth-to-air heat exchanger technique is considered one of the oldest techniques of air conditioning in the world. Since about 3000 B.C. it has been used firstly by Iranian architects for cooling buildings.

The basic principle of the technology is related to the use of the ground as a heat sink. Air is pulled by the fan from the surrounding environment to pass through tunnels or pipes under the ground into the facility. During this process there is a heat exchange between air and the walls of the pipe at a certain depth. Hence, this method could be used to reduce the temperature of the air during the summer (Scott *et al.*, 1965; Goswami and Ileslamlou, 1990).

This technique is known by several names such underground air tunnel, earth–air–tunnel, earth-tube fan system and earth-to-air heat exchanger (EAHE).

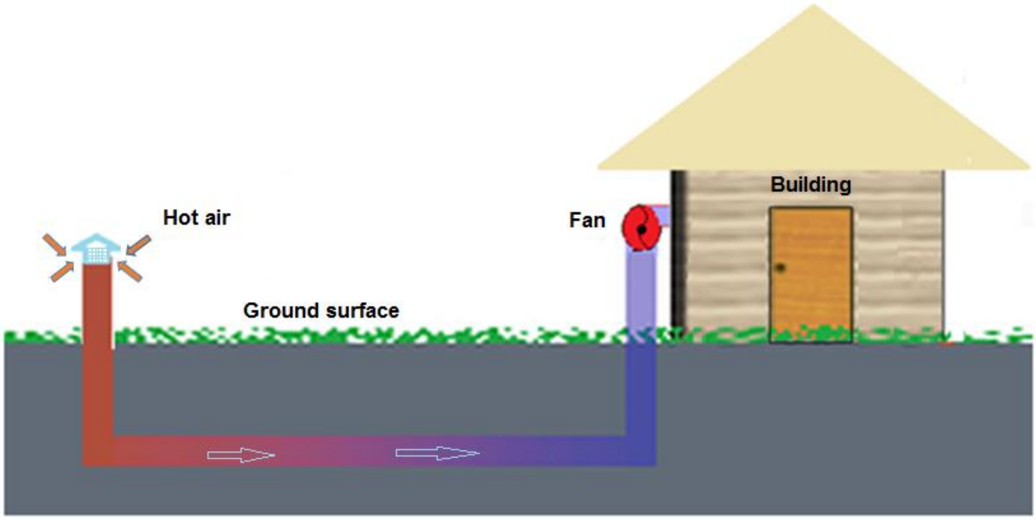

Figure 1-19 Earth-to-air heat exchanger technique.

The interest in the achievement of indoor thermal comfort by mean of the EAHE technique is related to the increased spread of solutions based on renewable energy sources. The diffusion of such technique is attributed to the high costs of energy and to the large availability and cheapness of pipes of several materials and different dimensions.

First studies have been carried out in the sixties of the last century to cool commercial livestock buildings in the USA (Scott *et al.*, 1965) and in the early of eighties in Europe (Mattesson *et al.*, 1981). Afterwards, the system was used to heat and cool greenhouses in Europe and North America (Santamouris *et al.*, 1995). In time, the mathematical model of earth-to-air heat exchanger has been developed by Puri (1986).

The EAHE is able to reduce the energy consumption required for heating and cooling of buildings (Grosso, 2008; Pfafferott, 2003). Shi and Chew (2012) and Hamada *et al.* (2007) have well documented the important advantages of this technology, in particular the low cost and the capacity to decrease the carbon dioxide emissions of farm conditioning.

Traditional conditioning consumes a lot of energy to heat and cool protected cultivation (Ghosal and Tiwari, 2002; Ozgener and Ozgener, 2010; Sharan and Jadhav, 2003). In Asia countries, such India and Turkey, the EAHE technique is used to condition greenhouses because the inside temperatures can exceed 45°C in the summer and drop to 6°C in the winter.

Furthermore, in other areas of the world (e.g. Kuwait, Italy, Germany, Poland, etc.) the system is used for air conditioning of the domestic buildings (Al-Ajmi *et al.*, 2006; Pfafferott, 2003; Maryanczyk *et al.*, 2014).

Nonetheless, for the purpose of the present research the most important application of using the earth-to-air heat exchanger technique is to secure comfortable environmental conditions in livestock buildings, particularly in closed facilities for poultry breeding (Barbari *et al.*, 1996; Petek *et al.*, 2012).

1.5.1 Factors affecting efficiency

Several analytical and experimental studies (Barbari and Chiappini, 1984; Ghosal and Tiwari, 2006) of earth-to-air heat exchanger technique have been conducted to evaluate the technology performance for cooling buildings and poultry facilities. Then, many researches were carried out to determine all factors that impact directly or indirectly the performance of the cooling or heating of EAHE (Chel and Tiwari, 2009; Bansal *et al.*, 2009).

1.5.1.1 Pipe properties

The characteristics of the pipes (length, diameter, spacing, shape, material and depth of the pipe) have been explained by many studies (Baxter, 1994; Deglin *et al.*, 1999; Petek *et al.*, 2012). These researches have revealed that increasing of the pipe length leads to increase the effectiveness of the system. Anyway there are no significant advantages in using pipes over 70 m length because a too large high-powered fan would be requested. In particular, the researchers suggested that the average length has to be between 35-40 m, the pipe diameter 15-30 cm and not less than 10 cm.

In theory, small diameters would be better due to the higher heat exchange. In spite of this, they are not suitable because the airflow becomes insufficient and an increase of the number of pipes to reach sufficient rate of air is necessary, bringing to costs rise.

About the depth of the pipe, it must be not less than 120 cm below ground surface to avoid any influence of solar radiation. Increasing the pipe depth more than 3 m provides a considerable improvement in the system's potential cooling capacity, but of course costs raise drilling deeper. In order to prevent thermal interaction of the pipes, space between them must be more than 100 cm from each other (Barbari and Chiappini, 1984; Sayed, 2015).

Material determines the thermal conductivity of the pipe. Subsequently metal is better than PVC, although it is more expensive. A corrugated surface of the pipe leads to increase the exchanges with the surrounding soil, thereby improving the cooling efficiency of the system.

1.5.1.2 Soil properties

The composition of the soil, the soil surface, the moisture and its thermal characteristics have been discussed in several works (Sharan and Jadhav, 2003; Givoni, 2007; Ghosal and Tiwari, 2006). The results confirmed that the existence of a soil surface layer made of green or gravel relieves the impact of the sun's rays. Thus, the temperature of the ground remains low improving the cooling performance of the system. These findings are consistent with results obtained by Sanusi and Zamri (2014). The surface of ground was covered with layers of three different materials (used tires, recycled timber palette and short grass), useful to isolate the soil for EAHE application. The layers of used tires were the best to achieve a more cooling outlet temperature. In the same way, thermal characteristics analysis of several kind of soil revealed clay soil as the best insulator.

On the other hand, increasing the proportion of metals in the soil leads to improve the heat exchange because of better thermal properties of the soil. In the same context, all studies demonstrated that better heat exchange and greater cooling of the outlet temperature occur with an increase of the soil moisture. Furthermore, all these factors affect the soil temperature, which is a fundamental parameter in the heat transfer from the air to the soil (Barbari and Chiappini, 1984).

In addition, the speed of the air flow inside the pipe (Deglin *et al.*, 1999; Al-Ajmi *et al.*, 2006) is considered one of the fundamental elements that affects the cooling performance of underground pipes technology. It must be a low value in order to provide the required time for the heat exchanger to achieve the best cooling of the air. Consequently, the speed range values suggested are 2- 5 m/s.

In order to present a vision of the performance of the cooling plant, Heidt (2004) created an application (GAEA) for automatic calculation (Figure 1.20) of the information to assist the design of the underground pipe system. The program simulates the thermal performance of a geothermal system to support the decision on the basis of dimension and position of the heat exchangers. Unfortunately, GAEA was designed taking into account records of experiments carried out in Germany and other European countries.

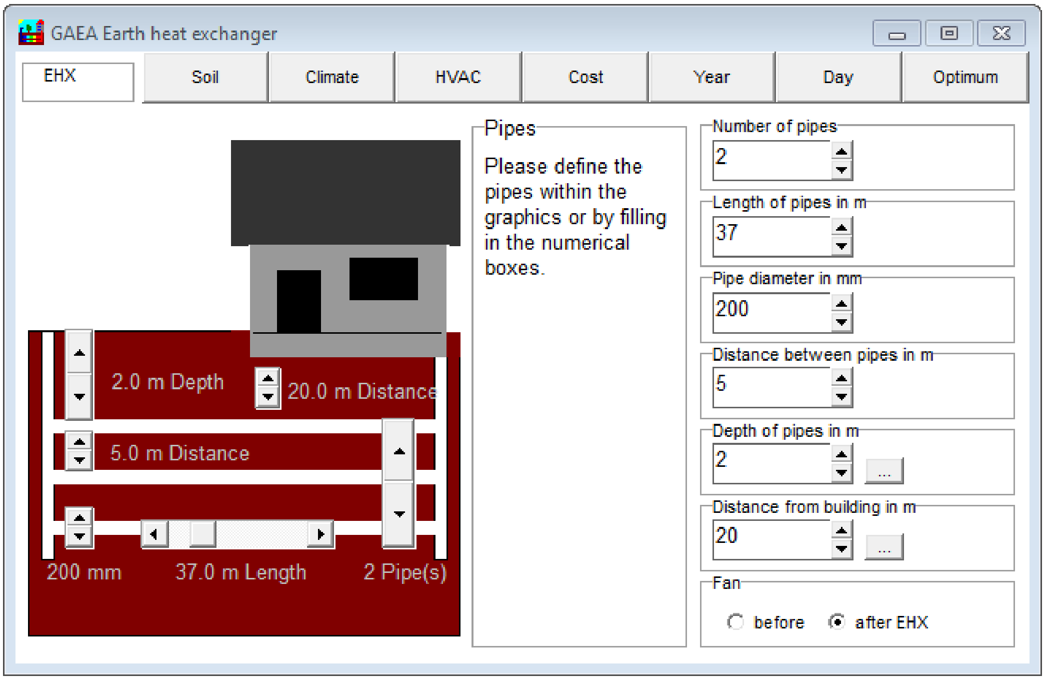

Figure 1-20. Calculation tool for Earth Heat Exchangers (Heidt, 2004).

1.6 Objectives of the research

The aim of this study is to demonstrate the effect of thermal conductivity on the cooling performance of earth-to-air heat exchanger.

In particular, the thermal performances of earth-to-air heat exchanger in dry soil or in artificially wetted soil around the pipe were tested and compared.

In order to achieve this goal, the following challenges were faced:

1. Study the efficiency of the earth-to-air heat exchanger system in the desert and semi-arid area of southern Iraq.
2. Study the effect of the wetting of soil surrounding pipes on the efficiency of the earth- to- air heat exchanger system.
3. Determine the effect of the length of the pipe on system performance with particular regard to the air temperature at the outlet of pipes.
4. Determine the effect of the temperature of air, which passes in the pipes on the soil surrounding them.
5. Study the impact of using EAHE system on the performance and on the efficiency of the system in the long time.

2 MATERIAL AND METHODS

2.1 Conditions of the study area

In agriculture, both plants and animals face the challenges of climatic conditions of desert and semi-desert areas such as South Iraq, and the difficulty of overcoming the temperatures and high humidity. Many researchers in the livestock field of Iraq are still tackling the effects of heat stress caused by high temperatures in order to manage the facilities in difficult climatic conditions (Mohammed *et al.*, 2000; Razuki *et al.*, 2007).

In Iraq, poultry breeding (broilers and laying hens) is an important economic source for many farmers and a source of food necessary for a large segment of the population. As stated above, the high temperatures are considered the biggest challenge facing the Iraqi poultry breeding.

2.1.1 Improving the performance of the EHAE in South Iraq

In Southern Iraq, the evaporative cooling of ventilation air to reduce heat-stress of broilers is widely adopted. However, during the hottest months of summer (June-August), when relative humidity is high, the evaporative cooling system alone is not able to create the minimum acceptable conditions for poultry production, especially in desert or semi-arid areas. In such conditions, other cooling systems have to be coupled with evaporative cooling. Thanks to a great variation between the outside air temperature and the ground temperature the underground cooling technology can be an effective and sustainable system to add. The number of contact points between the soil and the outer surface of the pipe increases if the soil around the pipe is wetted.

2.2 The experimental site

The experiment setups were installed in a semi-arid area in southern Iraq. Figure 2.1 shows the location of the poultry houses, placed in the region of the Basra Province (30°19′0″ N, 47°42′0″ E). This area is characterized by a sandy soil with high permeability and a low content of nutrients.

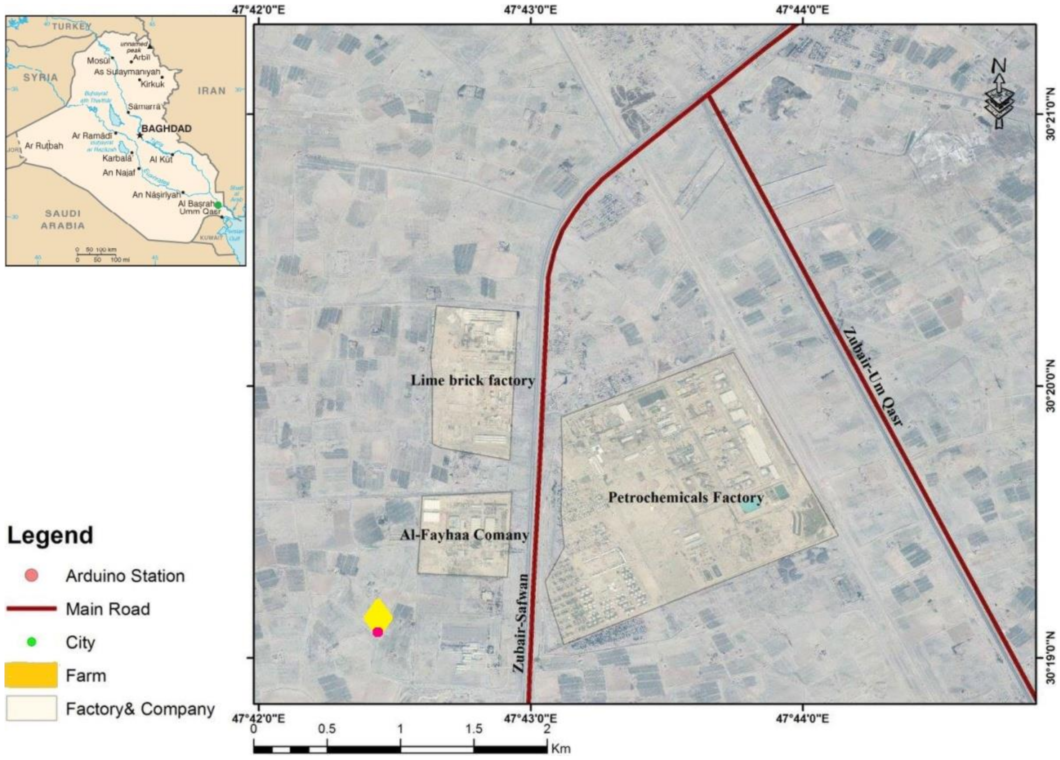

Figure 2-1 Site of poultry facility in Al-Zubayr - South Iraq.

The facility was 64 m length, 6 m width and 3 m height. Pads for evaporative cooling technology were installed in northern side of the building increasing the width of 1 m. A total number of 14 pads (2 m width, 1.30 m height) were installed along all the facility while 20 axial fans (Damandeh - Type: D140/26/4s) were installed on the other side of the barn (the southern one). In the facility, about 5000 broilers were kept in winter and about 3500 broilers in summer.

2.3 Description of the earth-to-air heat exchangers system

The experiment consists of two lines parallel pipes made up of PVC (Figure 2.2). The distance between each pipes is 5 m. The length of the lines is 37 m. The diameter is 20 cm. The pipes were buried at a depth of 2 m (Figure 2.3). Due to the high air humidity at the inlet of the system, a slope of 1% was given to the pipe in order to collect the water formed inside.

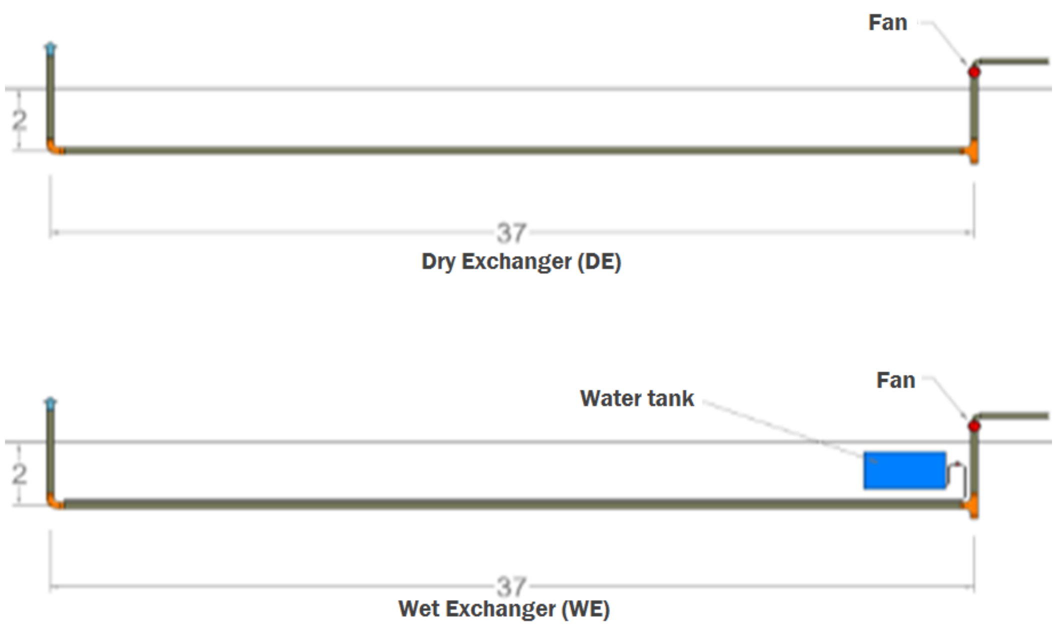

Figure 2-2 Scheme of buried pipes

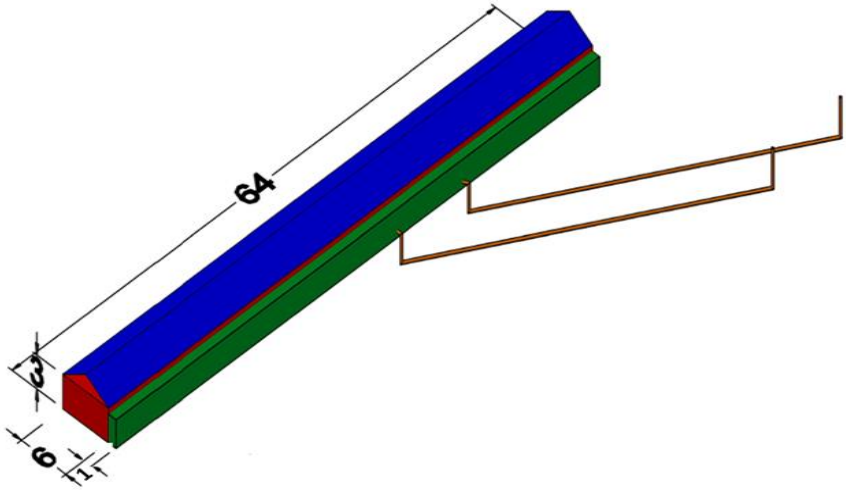

Figure 2-3 Scheme of the poultry barn with the two lines of underground cooling system, one line wetted with water and one line dry.

A T connection was put between the vertical and the horizontal part of the pipes. The bottom of the connection was covered with a plastic net (Figure 2.4b) used to filter the pulling air flow from the sand.

As it is shown in Figure 2.4a, a protective cover (Rain cowl) with a grid wall was installed at the entrances of the pipes to avoid the entry of rainwater and insects. In addition, this entrance was equipped with an adjustable butterfly damper to provide the possibility to control the air flow inside the tubes.

Furthermore, the inlet pipe raised above the ground about 1.5 m to prevent the access of dust and animals.

The outside parts of the system were painted with white color to diminish the impact of solar radiation. In addition, the bent tube at the barn entrance was very well wrapped with insulating fabric and sheltered with a wooden box (Figure 2.4c).

(a) Rain cowl (b) End of the pipe slope (c) Pipe exit from the soil

Figure 2-4 Accessories of the heat exchanger technology.

The outside air was pulled through the pipe by a 125W axial extractor fan, one for each line, pushing the air inside the poultry barn. They were made of self-extinguishing polymer with a degree of protection from splashing of water (IP X4).

Air extractors were selected according to quietness and dimensions (Figure 2.5). The axial fans have two features. The first one is the possibility of intervene on the engine without disconnecting the tubes. The second one is a special tubular shape studied to make it suitable for easy connection to flexible pipes.

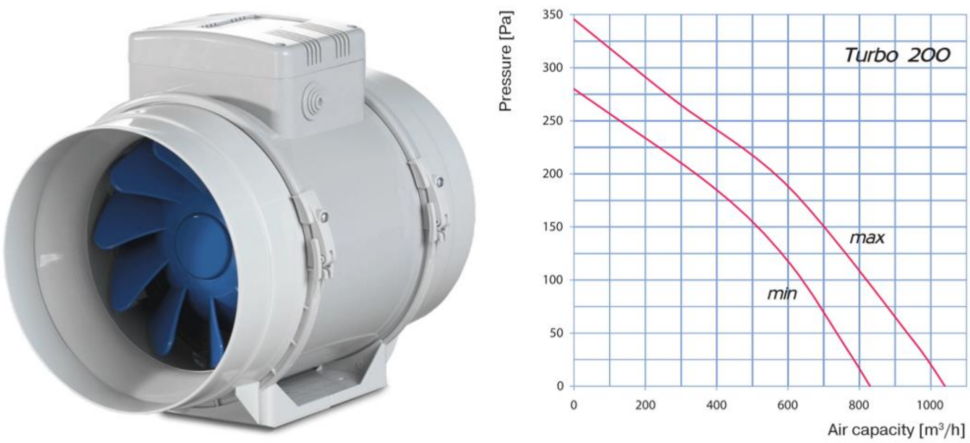

Figure 2-5 Axial extractor fan (125W- 20 cm).

2.4 Description of the wetting system

A soil layer around one exchanger was wetted once every 48 hours for 20 minutes at a dawn time.

The wetting process was done by using a drip tubing placed 10 cm above the air pipe. The distance between the moisturizing slots is 30 cm along the pipe which transport the air.

The wetting system was equipped with a tank of 1 m^3 volume, able to contain the minimum amount of water necessary to wet the ground during a period of 5 days. The tank was placed in a hole under the surface of the ground, fully covered to maintain the temperature of the water and to reduce the impact of the solar radiation (Figure 2.6). The water was pumped by an immersed pump placed in the tank and connected to a timer.

Figure 2-6 Wetting system.

2.5 Monitoring system

The monitoring system for the air temperature and humidity was made up with the following hardware:

1. Two microcontrollers Arduino UNO R.3.
2. A microcontroller Arduino Mega R.3.
3. Nine temperature analog sensors Atlas Scientific mod. ENV-TMP.
4. Three digital sensors of temperature-humidity Sparkfun (Sensor: Honeywell HumidIcon HIH-6130/6131 Series).
5. Two datalogger shields for Arduino Adafruit (mod. Data Logging Shield for Arduino) with RTC (Real Time Clock) and read/write support SD (Secure Digital).
6. Two solar panels of 10 W.
7. Two regulators for solar panels.
8. Two rechargeable lead batteries.

Since South Iraq is affected by frequent blackouts it was necessary to be independent from the main current. Thus, all the monitoring system was equipped with a power unit consisting of two solar panels and two rechargeable lead batteries.

On the other hand, the air fans were powered from the main network. In addition, a system to monitor the power supply itself was installed in order to know the moments of non-activity of the fans.

In this way, when the system was down the records were discarded by the analysis process of data.

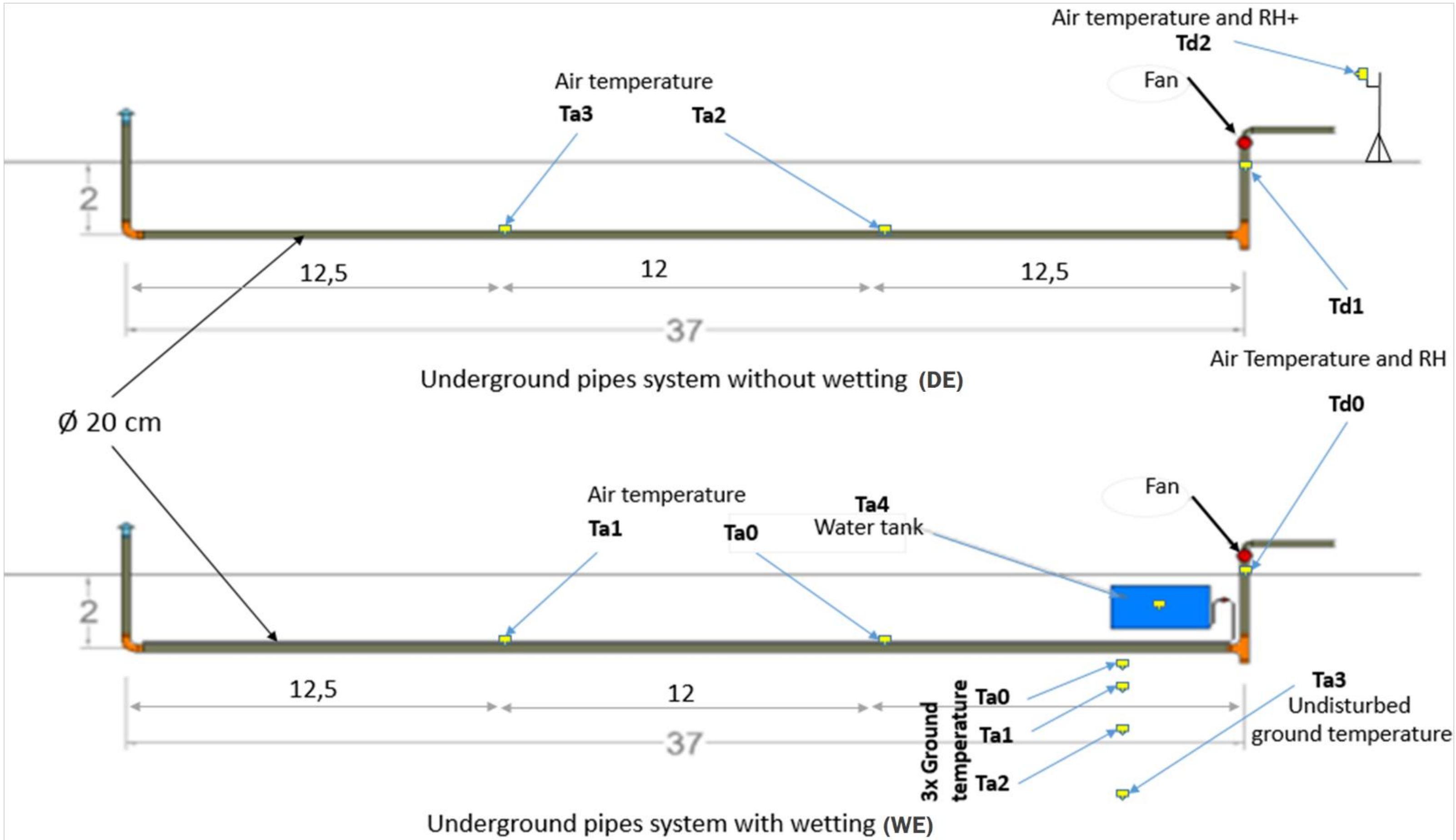

Figure 2-7 Position of the probes on the two lines.

Figure 2.7 shows the scheme of the monitoring system position in the field test.

Four temperature analog sensors linked to First Arduino were put inside the two lines to measure the temperature after 12.5 m (Ta1, Ta3) and 24.5 m (Ta0, Ta2), as shown in the scheme of Figure 2.8.

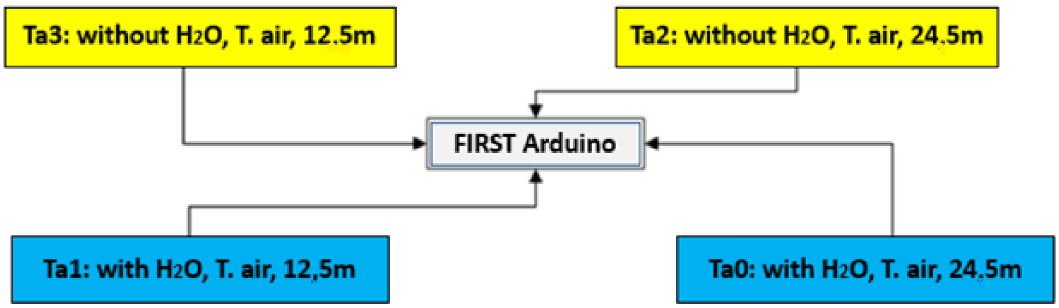

Figure 2-8 Connection diagram of the sensor with Arduino UNO.

In addition, two digital sensors of temperature-humidity were placed in the pipe outlet (wet Td0, dry Td1), and one was placed in external air (Td2).

Three temperature analog sensors (Ta0, Ta1, Ta2 respectively) were installed for measuring the soil temperature at different distance from the pipe: 25 cm, 50 cm and 100 cm. Additional analog sensors measured the temperature of undisturbed ground (Ta3), as well as the temperature of water in the tank (Ta4). All mentioned sensors were linked to Mega Arduino data logger as presented in the scheme of Figure 2.9.

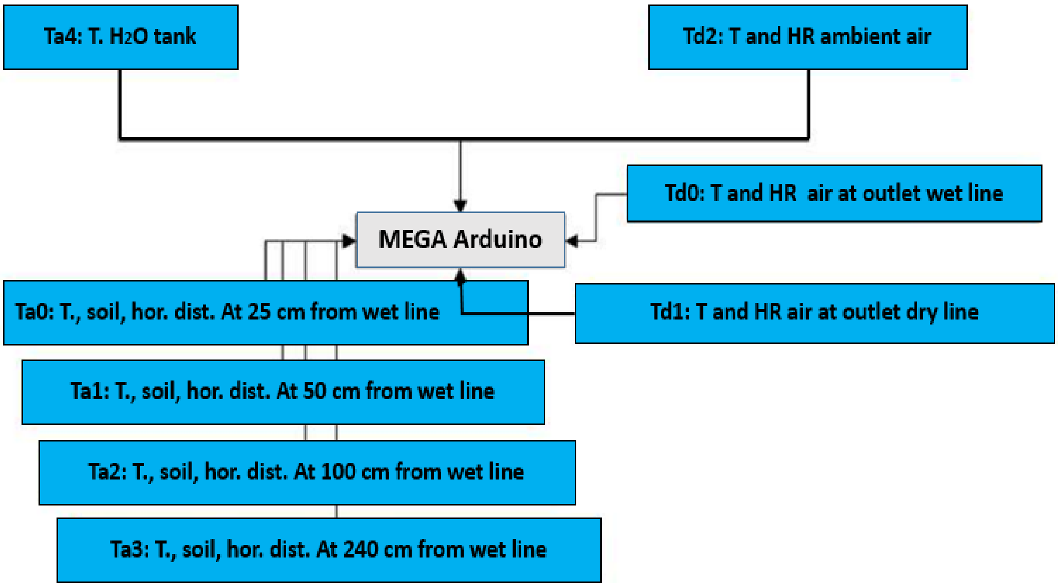

Figure 2-9 Connection diagram of the sensor with Arduino Mega.

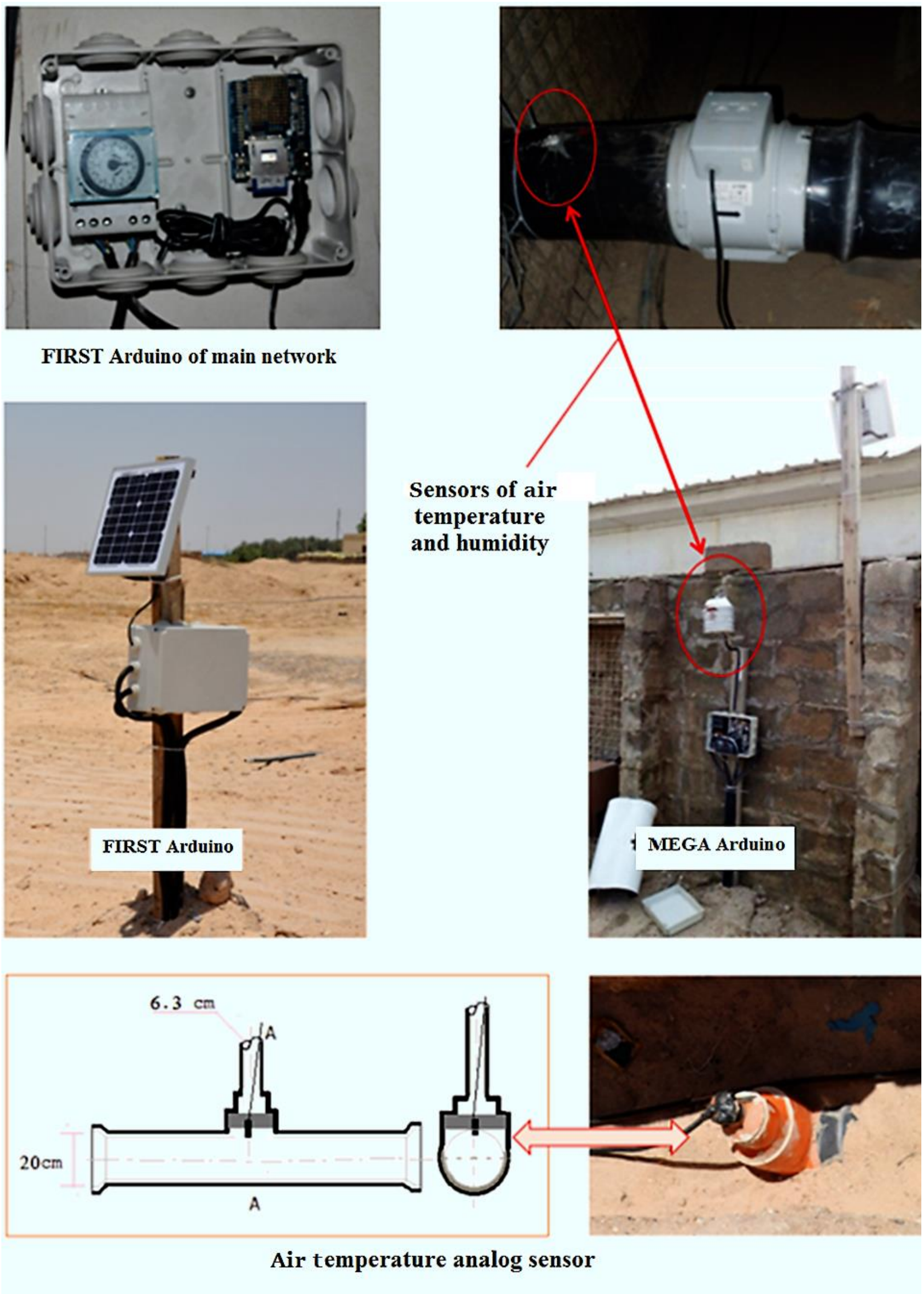

Figure 2-10 The scheme of monitoring system.

Figure 2.10 shows some details of the installation steps and the places of equipment and sensors. In addition, Figure 2.11 captures the final moments of the installation and start of the test. The picture shows the entrances of the air to the earth-to-air heat exchangers system and the box of Arduino Uno with the microcontroller Arduino UNO R.3, one datalogger shields for Arduino Adafruit (mod. Data Logging Shield for Arduino) with RTC (Real Time Clock) and read / write support SD (Secure Digital) and one regulator for solar panel.

Figure 2-11 The beginning of the operation phase of the earth-to-air heat exchangers system.

2.6 Measurements

After the installation of all the pipes, fans, moisturizing plant, recording and monitoring devices, the system's reliability was proved. A test was done before the definitive burying of pipes in order to ensure that earth-to-air heat exchangers system, moisturizing system and recording devices worked well. Then, the EAHE systems worked regularly since 01 / 06 / 2013.

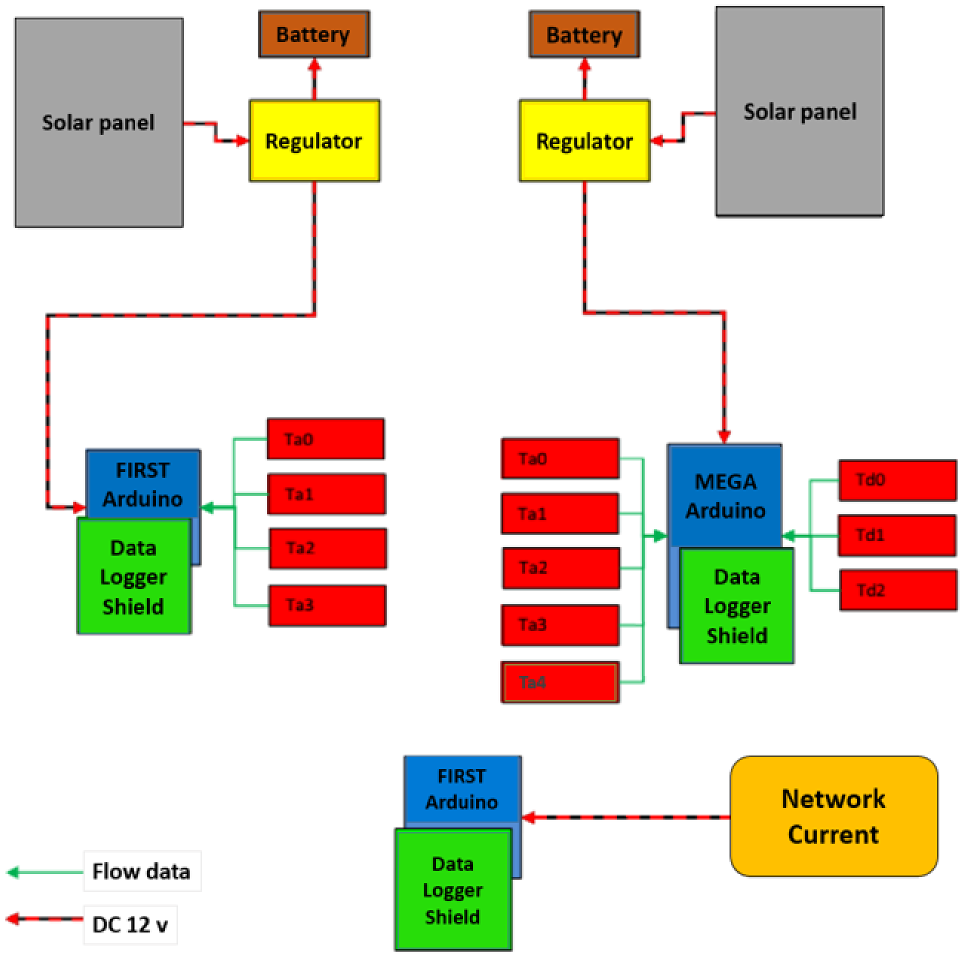

Figure 2-12. Hardware flow of data and of the solar power system.

Thanks to the presence of the continuous current (the solar supply system or battery) that provided the energy for the data recording units (data-logger), the data were collected from all sensors every 15 minutes. These data were recorded and stored by a memory card for each datalogger. Figure 2.12 shows the hardware scheme flow of the data and of the solar power system.

Figure 2.13 shows the chart of data logical flow of the system. The distributed sensors transmitted data to the datalogger which processed and registered them into the SD that had to be replaced periodically.

The substitution of the data recording cards was done every two weeks by a collaborator employed at Agriculture Directorate of the Province of Basra (Iraq). Then the data were sent via e-mail to the Department of Agricultural, Food and Forestry Systems of the University of Florence.

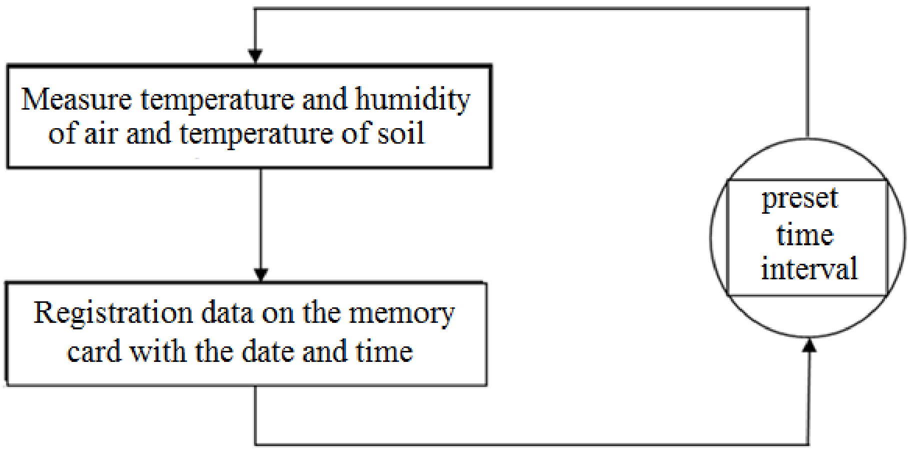

Figure 2-13. Logical flow of the system.

2.7 Mathematical models

2.7.1 Centrifugal fan

The following equation is used to calculate the capacity of centrifugal fan (P):

$$P = \frac{Q * \Delta p}{\tau} \qquad (1)$$

where:

Q is the air flow,

$\tau \approx 0.6\text{-}0.9$ the performance of the fan,

Δp is the pressure difference between both sides of the fan.

The air flow rate is calculated on the basis of volume of air which has to be changed during one hour in order to provide a conform environment for the animals. According to Sabagh (2002), the required rate of air for one chicken should be 3 m^3/h.

Then, the air speed (v) is calculated depending on the pipe section area and on the number of broilers using the following equation:

$$v = \frac{Q}{At} \qquad (2)$$

Where: $A_t = \pi R^2$

Δp is found by the centrifugal fans scheme.

In addition, if the loss of pressure is high it is necessary to calculate the loss of pressure when the fluid passes inside the pipe using the **Darcy-Weisbach** equation:

$$\Delta p' = \frac{1}{2} f \frac{L \rho v^2}{D} \qquad (3)$$

Where:

$\Delta p'$: main loss of pressure.

f: friction coefficient.

L: length of the tube.

ρ: air density.

v: speed of fluid.

D: diameter of the tube.

Friction Coefficient is calculated based on the **Reynolds number (Re)**:

$$\boldsymbol{Re} = \frac{\rho V D}{\mu} \qquad (4)$$

Where: μ = 1,225 kg / m^3 (viscosity of air).

If:

Re<2000 Laminar flow, then $f = \frac{64}{Re}$

Re>2000 Turbulent flow, then f is taken from *Moody Chart* according to Re and Ɛ/D (Ɛ: roughness).

The secondary loss of pressure $\Delta p''$ depends on form of connections and change of directions.

$$\Delta p'' = K \,.\, \frac{\rho\, v^2}{2} \qquad (5)$$

Where: K is the coefficient of connection.

Finally, the total loss pressure is:

$$\Delta p = \Delta p' + \Delta p'' \qquad (6)$$

2.7.2 Soil temperature

The mathematical model to calculate the annual soil temperature at a certain depth is a function depending on the theory of conduction heat transfer in a semi-infinite homogeneous solid. As a result of the annual temperature fluctuations affecting the soil surface, this function takes the sinusoidal pattern. Kusuda and Achenbach (1965) have mathematically modeled the following equation:

$$T(z,t) = T_m - A_{surf} \times e^{[-z(\pi/8760\alpha)^{0.5}]} \times \cos\left[\frac{2\pi}{8760} \times \left(t - t_0 - \frac{z}{2}\left(\frac{8760}{\pi \times \alpha}\right)^{0.5}\right)\right] \qquad (7)$$

Where:

T (z,t) is the temperature of undisturbed soil (°C) at depth z (m) and time t (hours),

z is the depth of a point below the soil surface, the temperature of this point has to be known, t is the time elapsed from the beginning of the year in hours,

T_m is the annual mean temperature of the soil (°C),

A_{surf} is the amplitude of temperature variation on soil surface in the year (°C),

α is the thermal diffusivity of the soil (m^2/h),

t_0 is the time to occurrence of minimum temperature of the surface since start of year, in hours (phase constant, hours).

2.7.3 Air outlet temperature equation

The air temperature at the outlet of buried pipes can be provided taking into account all the factors affecting the system (Demir, 2006). The temperature profile in relation to the pipe length may be expressed as in the following equation:

$$T_{a,o} = T_s - (T_s - T_{a,i})e^{\frac{-k_s L}{m_a C_{p,a}}} \qquad (8)$$

Where:

$T_{a,o}$: the air outlet temperature (°C);

$T_{a,i}$: air inlet temperature (°C);

T_s: soil temperature (°C);

K_s: soil thermal conductivity (W/m K);

L: pipe length (m);

e: Napier's constant;

m_a: mass flow rate of the air (kg/s);

$C_{p,a}$: specific heat of fluid (J/kg K).

2.7.4 Thermal efficiency and coefficient of performance

To evaluate the work of earth-to-air heat exchanger systems the efficiency and coefficient of performance should be calculated. The efficiency of EAHE system (Ɛ) is a proportion which expresses the ability of the system (depending on the outside air temperature) to cool or to heat the air passing through the pipes. Hence, the range of efficiency is between 0-1 and it is expressed as a percentage. It is calculated by the following equation (Baxter, 1994):

$$\varepsilon = \frac{T_i - T_o}{T_i - T_a} * 100 \qquad (9)$$

Where:

Ɛ: efficiency of the EAHE system (%),

T_i: air temperature at the entrance of pipes,

T_o: air temperature at the exit of pipes,

T_a: undisturbed soil temperature.

The coefficient of cooling performance of the earth-to-air heat exchangers system is evaluated by using the following general equation (Ashrae, 1989):

$$COP = \frac{Q_{out}}{W_{in}} \qquad (10)$$

Where:

COP: coefficient of cooling performance of EAHE system,

Q_{out}: heat removed or added to the air; it is expressed in Watt,

W_{in}: energy input, that is the amount of electrical energy consumed by the fan in Watt.

The amount of heat transferred, removed or added, to the air is expressed by the equation of heat exchange (Alghannam, 2012):

$$Q_{out} = m \cdot c_p \cdot \Delta T \qquad (11)$$

Where:

m: mass flow rate of the air, kg/s,

c_p: specific heat at constant pressure (kJ/ kg ºC),

$\Delta T = T_i - T_o$: difference in temperature between the entrance and the exit of the system.

2.8 Statistical analysis

The experiment lasted for 9 months continuously, from June 2013 to February 2014 plus the month of August 2014.

The use of the data collected in the entire experimental period appeared to be the most logical choice. Anyway, due to the large amount of data (28700 records for each sensor), the size of the dataset was reduced to minimize the information loss. One day of each season was identified as representative of the whole season experimental period. These days (for summer 2013, autumn 2013, winter 2013 and summer 2014) are called “typical days”.

To select the typical day, the sum of squared differences between the mean external temperature among all the days at every time of the day and the external temperature measured at the same time was calculated. At last, the typical day had the lowest sum of squared differences.

Therefore, to assess whether artificially wetted soil around the pipe affected significantly outlet pipe temperature, repeated-measures ANOVA were performed on all data within all typical days with time of measurements as the repeated subject. Models were fitted using the ‘nlme' package of R while analyses of variance were carried out using the package ‘stat’ (R Development Core Team, 2011). Least squares means and SEM are reported for all data.

To test pairwise comparisons (temperature, humidity) between wet line and dry line during two typical days (for summer 2013 and summer 2014), posthoc analyses were carried out by Tukey test using the R package ‘multcomp’ (R Development Core).

Descriptive statistics (mean, SD and range) were used to describe the effect of temperature of the air passing through the pipes to the surrounding soil particles at 2 m depth with the distance from the pipe (25, 50, 100 cm and undisturbed ground), and the relationship between the air passing through the pipes with the length of the pipes.

Polynomial regression analyses were performed to identify temperature changes with distance. A t-test was performed to determine whether there was a significant relationship between variables.

In order to better demonstrate the effect of artificially wetted soil around the pipe on the outlet temperature during the extreme outside temperature, the hottest and coolest days of the year were selected. Two distinct datasets were built for the afternoon (11:00 to 16:00) and for the night (01:00 to 05:00) periods for each day.

Descriptive statistics (mean, SD and range) were used to describe air characteristics, temperature and humidity at the outlet of the dry and wet pipe, as well as, the difference between the ambient air temperature and outlet of dry and wet pipe. Results are presented as mean±SD and range.

To illustrate the capacity of the system of cooling and heating, efficiency and coefficient performance of EAHE were calculated in the critical environmental conditions. The hottest and coolest period (4 hours) during the time of the experiment were selected.

Air velocity inside the tubes was calculated as the average value of several readings taken at the outlet pipe at the beginning of the experiment (2.9 m/s).

3 RESULTS

In the experimental area, during the period from June 2013 to February 2014, the average, maximum and minimum external air temperatures (T_ext) were 25.07 ±12.24°C, 52.30°C and 0°C, respectively (Table 3.1). The maximum value of 52.30°C is very close to the national record value.

At the same time, the average, maximum and minimum outlet air temperature were 26.48 ±8.63°C, 40.60°C and 9.40°C for wet pipe (T_wet), and 26.32 ±9.09°C, 42.70°C and 9.10°C for dry pipe (T_dry), respectively.

On the other hand, the average, maximum and minimum undisturbed soil temperature were 26.89 ±5.11°C, 32.47°C and 17.48°C, respectively.

Table 3.1. Temperatures measured in the experimental period (June 2013-February 2014).

	T_ext	**Tgr**	**T_wet**	**T_dry**	**T_ext-T_wet**	**T_ext-T_dry**
Mean	25.07	26.89	26.48	26.32	-1.41	-1.25
SD	*12.24*	*5.11*	*8.63*	*9.09*	*4.88*	*4.13*
Min	0.00	17.48	9.40	9.10	-12.50	-11.20
Max	52.30	32.47	40.60	42.70	12.60	11.00

The average relative humidity of external air (HR_ext), outlet air of wet (HR_wet) and dry pipe (HR_dry) were 22.98 ±18.81%, 38.50 ±23.90%, and 38.72 ±25.25%, respectively. As well as, the maximum and minimum of the inlet and outlet relative humidity are shown in Table 3.2.

Table 3.2. Relative humidity of external air, outlet air of wet and dry pipe during the experimental period (June 2013-February 2014).

	HR_wet	HR_dry	HR_ext
Mean	38.50	38.72	22.98
SD	*6.30*	*5.20*	*1.90*
Min	23.90	25.25	18.81
Max	97.40	99.00	94.30

3.1 Typical day of summer 2013

The typical day was found to be the 4th August 2013. During that day the average, maximum and minimum external temperatures were 37.66°C, 46.10°C and 30.30°C respectively, while the undisturbed ground temperatures were 31.67°C, 31.73°C and 31.48°C, respectively. Figure 3.1 shows external air (T External) and ground (T Ground) temperatures, and also temperatures measured at the outlet of each experimental pipeline during the typical day of the summer. Wetting the soil surrounding the pipe significantly affected outlet air temperature and humidity ($p<0.001$; Table 3.3). During daytime differences were more evident while during the night the wet and dry pipe produced convergent outlet temperatures. During the typical day, the EAHE operating in dry soil (DE) had significantly higher temperatures (T Outlet_DRY = 36.47°C) than those in wet soil (WE) (T Outlet_WET = 36.1°C). As well as, the relative humidity of WE (RH Outlet_WET = 18.51%) and DE (RH Outlet_DRY = 17.2%) had significant differences, while, the relative humidity of external air was 14.55%.

Table 3.3. Least square means, SEM and significance of effects for outlet air temperatures with different types of underground pipes (WET or DRY) during the summer 2013 typical day (4th August 2013).

	External	Ground	Outlet_DRY	Outlet_WET	SEM	Effect, p≤
T, °C	37.66	31.67	36.47[a]	36.1[b]	0.067	0.001
RH, %	14.55	-	17.2[a]	18.51[b]	0.059	0.001

[a, b] *Least square means in the same row with different superscripts differ (p<0.05).*

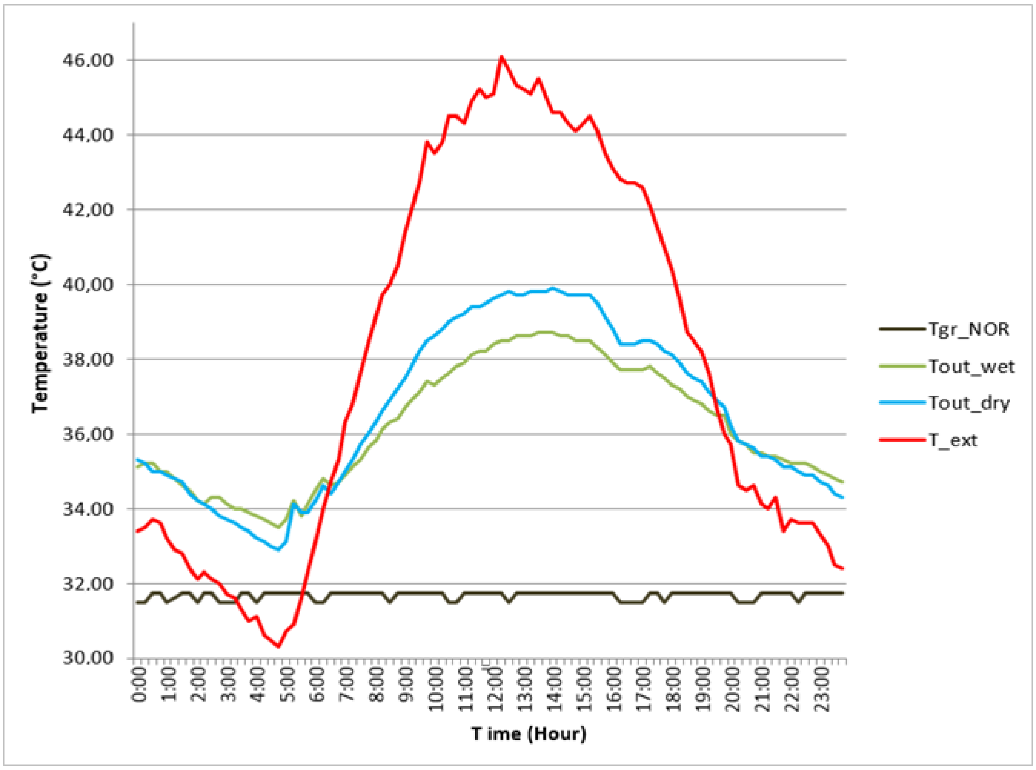

Figure 3-1. Air temperatures at the outlets of the EAHE systems in relation with outside and ground temperature during the typical day (4th August 2013). Tgr_NOR=undisturbed soil temperature at 2m, Tout_wet=outlet air temperature for wet pipe, Tout_dry=outlet air temperature for dry pipe, T_ext=external temperature at the experimental site.

3.2 Typical day of autumn 2013

The typical day of autumn was 9th October 2013. During that day, the average external temperature (T External) was 22.28°C and the maximum and minimum temperatures were 30.40 and 14.20°C. In the same period, the undisturbed ground temperatures (T Ground) decreased slightly from those of summer. The average, maximum and minimum undisturbed ground temperatures were 31.10, 31.24 and 30.99°C, respectively. External, undisturbed ground, wet exchanger outlet and dry exchanger outlet temperatures during the typical day of autumn are shown in Figure 3.2. By wetting the soil around the pipe there was a significant effect on outlet temperature ($p<0.001$; Table 3.4). As found in the typical day of summer, wet exchanger had significantly higher temperatures (T Outlet_WET = 27.29°C) than dry exchanger (T Outlet_DRY = 25.88°C) but, in the typical day of autumn, the differences during the night were higher than during daytime. The external relative humidity (RH External) was 19.22%, although the differences of relative humidity between the wet (RH Outlet_WET) and dry (RH Outlet_DRY) exchanger were significant.

Table 3.4. Least square means, SEM and significance of effects for outlet air temperatures with different types of underground pipes (WET or DRY) during the autumn 2013 typical day (9th October 2013).

	External	Ground	Outlet_DRY	Outlet_WET	SEM	Effect, p≤
T, °C	22.28	31.10	25.88^a	27.29^b	0.07	0.001
RH, %	19.22	-	17.18^a	16.38^b	0.086	0.001

$^{a, b}$ Least square means in the same row with different superscripts differ ($p<0.05$).

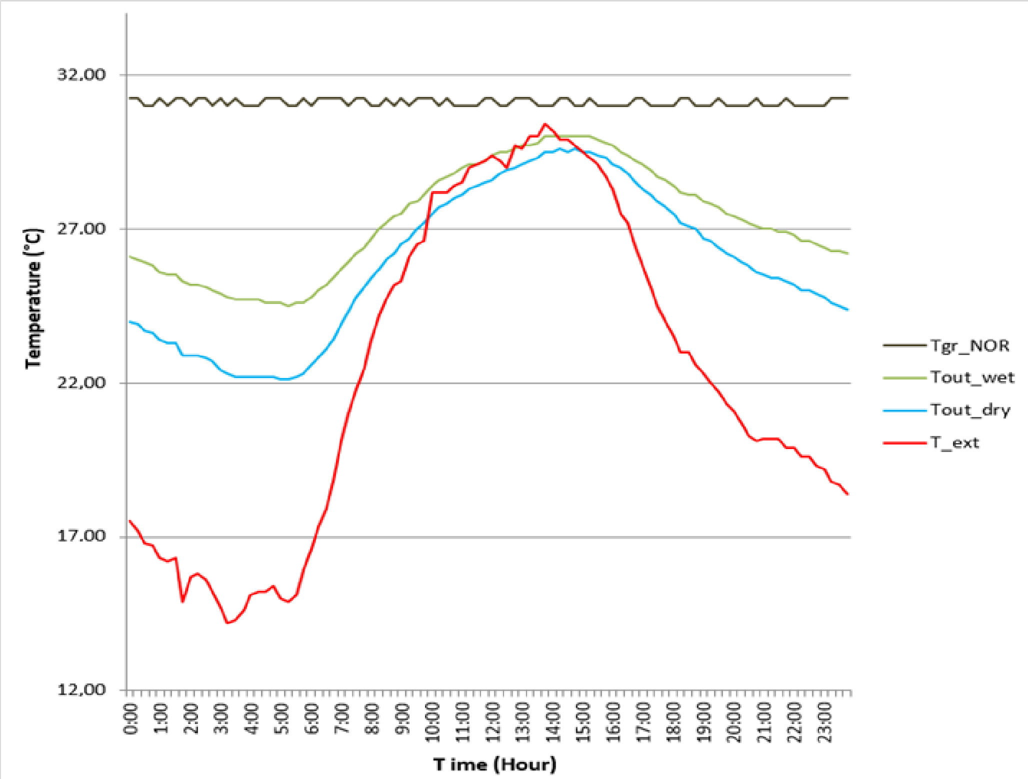

Figure 3-2. Air temperatures at the outlets of the EAHE systems in relation with outside and ground temperature during the typical day (9th October 2013). Tgr_NOR=undisturbed soil temperature at 2m, Tout_wet=outlet air temperature for wet pipe, Tout_dry=outlet air temperature for dry pipe, T_ext=external temperature at the experimental site.

3.3 Typical day of winter 2013

The typical day of the winter was the 16th January 2014. During that day, the average external temperature (T External) was 12.31°C and the maximum and minimum temperatures were 18.20 and 6.20°C while the average, maximum and minimum undisturbed ground temperatures (T Ground) were 19.09, 19.20 and 18.95°C, respectively. Figure 3.3 shows external and ground temperatures as well as the temperatures measured at the outlet of wet and dry line during the typical day of the winter. Wetting effect of the soil around the pipe had a significant effect on outlet temperature ($p<0.001$; Table 3.5), despite, in the daytime, differences were less marked than those observed in the night period. Although the reduction of external and ground temperatures of almost 10°C compared to the autumn, the temperature and humidity at the exits of the pipe maintained their behavior. Analysis showed that temperature of wet line (T Outlet_WET = 14.39°C) and dry line (T Outlet_DRY = 13.80°C) had significantly differences between them. Relative humidity (RH Outlet_WET) of WE (78.62%) and (RH Outlet_DRY) of DE (81.69%) had significant differences, although the values of the relative humidity for two lines were high.

Table 3.5. Least square means, SEM and significance of effects for outlet air temperatures with different types of underground pipes (WET or DRY) during the winter 2013 typical day (16th January 2013).

	External	Ground	Outlet_DRY	Outlet_WET	SEM	Effect, p≤
T, °C	12.31	19.09	13.8^{a}	14.39^{b}	0.626	0.001
RH, %	na	-	81.69^{a}	78.62^{b}	0.221	0.001

^{a,b} *Least square means in the same row with different superscripts differ (p<0.05)*

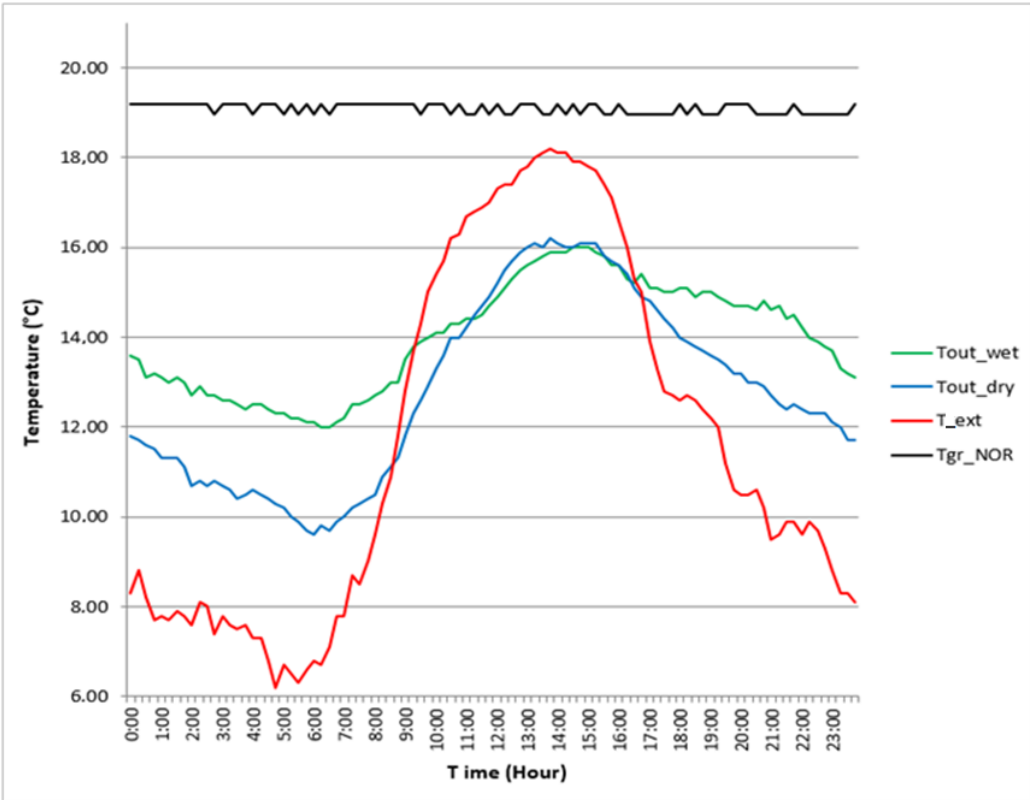

Figure 3-3. Air temperatures at the outlets of the EAHE systems in relation with outside and ground temperature during the typical day of winter (16th January 2014). Tgr_NOR=undisturbed soil temperature at 2m, Tout_wet=outlet air temperature for wet pipe, Tout_dry=outlet air temperature for dry pipe, T_ext=external temperature at the experimental site.

3.4 Typical day of summer 2014

The typical day was found to be the 7th August 2014. During that day the average, maximum and minimum values of the external temperatures (T External) were 37.98°C, 43.97°C and 33.81°C, respectively. While, the undisturbed ground temperatures (T Ground) were 32.22°C, 32.47°C and 31.98°C, respectively. Figure 3.4 shows external and undisturbed ground temperatures during the typical day of the summer 2014 and measured temperatures at the outlet of each experimental pipeline as well. Treatment effect of the surrounding soil around the pipe brought to significant difference of temperatures and humidity ($p<0.01$; Table 3.6).

During the typical day, the EAHE operating in dry soil (T Outlet_DRY = 34.23°C) and in wet soil (T Outlet_WET = 34.12°C) had significantly different temperatures between them. As well as, the relative humidity of WE (RH Outlet_WET = 22.12%) and DE (RH Outlet_DRY = 21.26%) had significant differences.

Table 3.6. Least square means, SEM and significance of effects for outlet air temperatures with different types of underground pipes (WET or DRY) during the summer 2014 typical day (7th August 2014).

	External	Ground	Outlet_DRY	Outlet_WET	SEM	Effect, P≤
T, °C	37.98	32.22	34.23^{a}	34.12^{b}	0.034	0.001
RH, %	na	-	21.26^{a}	22.12^{b}	0.045	0.001

^{a,b} Least square means in the same row with different superscripts differ ($p<0.05$)

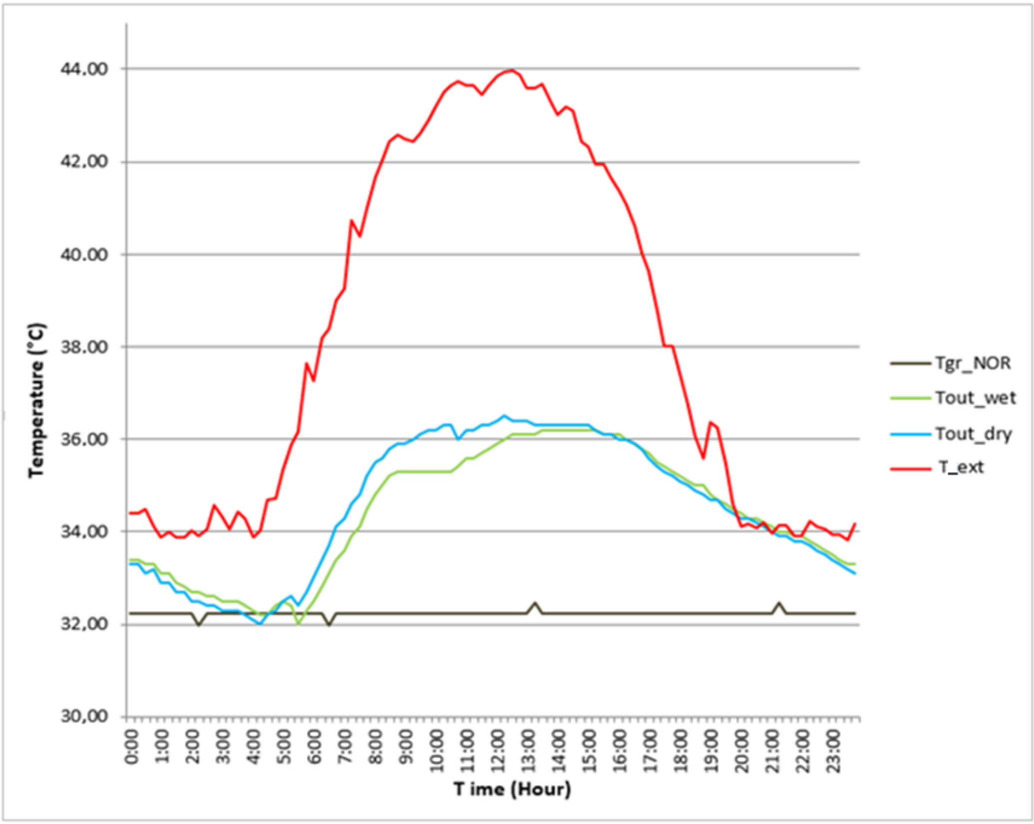

Figure 3-4. Air temperatures at the outlets of the EAHE systems in relation with outside and ground temperature during the typical day (7th August, 2014). Tgr_NOR=undisturbed soil temperature at 2m, Tout_wet=outlet air temperature for wet pipe, Tout_dry=outlet air temperature for dry pipe, T_ext=external temperature at the experimental site.

3.5 Performance of EAHE over time

To evaluate the performance of EAHE system with time, typical days of summer 2013 (4th August) and summer 2014 (7th August) were taken into account. For this reason, the temperature difference of both pipelines (WE and DE) between the outlet air temperature of the pipe and the temperature of the inlet air (ambient temperature) was calculated. The average ΔT for wet and dry lines in 2013 was -1.37°C while -3.81°C in 2014. The use of EAHE systems with the time had a significant effect on the temperature difference between pipes entrance and exit ($p<0.001$; Table 3.7).

Table 3.7. Comparison summers 2013 versus 2014; ΔT means, SEM and significance of effects for ΔT after one year of use.

	Summer 2013	**Summer 2014**	**SEM**	**Effect, p≤**
ΔT(Outlet Air Temp-External Air Temp), °C	-1.37^{a}	-3.81^{b}	0.225	0.001

$^{a, b}$ Least square means in the same row with different superscripts differ ($p<0.05$).

3.6 Hottest day of experimental time

The hottest day was found to be the 11th of July 2013. During that day the average, maximum and minimum external air temperatures (Text) were 40.30°C, 51.10°C and 28.40°C, respectively, while the average, maximum and minimum outlet temperatures were 36.14°C, 38.80°C and 32.90°C for wet line (Twet), 36.90°C, 41.00°C and 31.90°C for dry line (Tdry).

Table 3.8 shows external air temperature and air temperatures measured at the outlet of each experimental pipeline during the hottest day. On this day, the maximum temperature difference between external air temperatures and outlet temperatures were 12.60°C for wet line, 10.60°C for dry line.

Table 3.8. Temperatures measured on the hottest day of summer 11th July 2013.

	Text	**Twet**	**Tdry**	**Text-Twet**	**Text-Tdry**	**Tdry-Twet**
Mean	40.30	36.14	36.90	4.16	3.40	0.76
SD	*8.64*	*2.13*	*3.24*	*6.51*	*5.42*	*1.13*
Max	51.10	38.80	41.00	12.60	10.60	2.20
Min	28.40	32.90	31.90	-4.50	-4.00	-1.00

Figure 3.5 shows external temperature and temperatures measured at the outlet for each experimental line during the hottest hours of the hottest day (from 11:00 to 15:00). Figure 3.6 shows external temperature and temperatures measured at the outlets (wet, dry) during the coldest period. The coldest period of the hottest day was from 1:00 to 5:00.

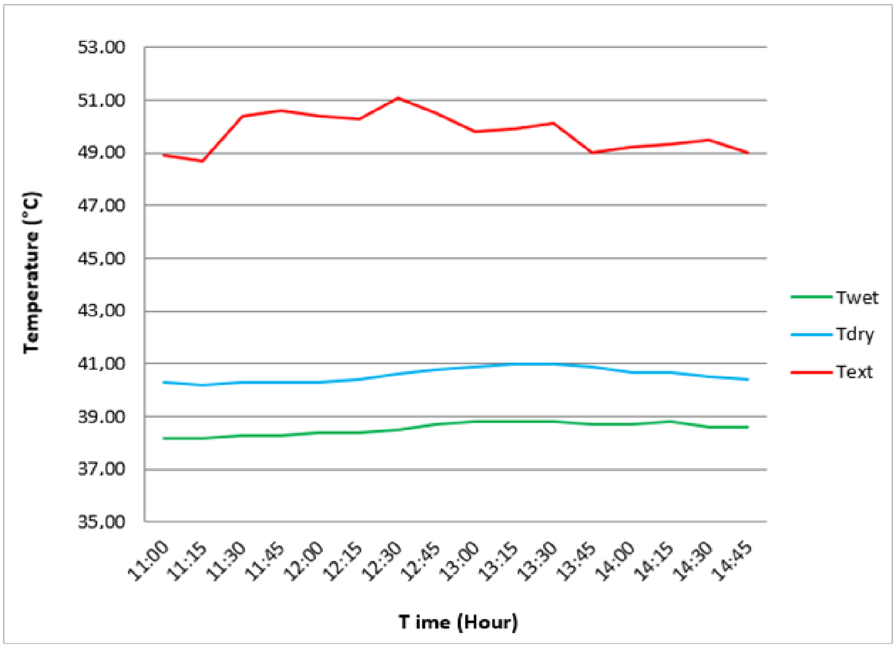

Figure 3-5 Hottest four hours of the hottest day of summer, 11th July 2013. Air temperatures at the outlets of the EAHE in relation with outside temperature.

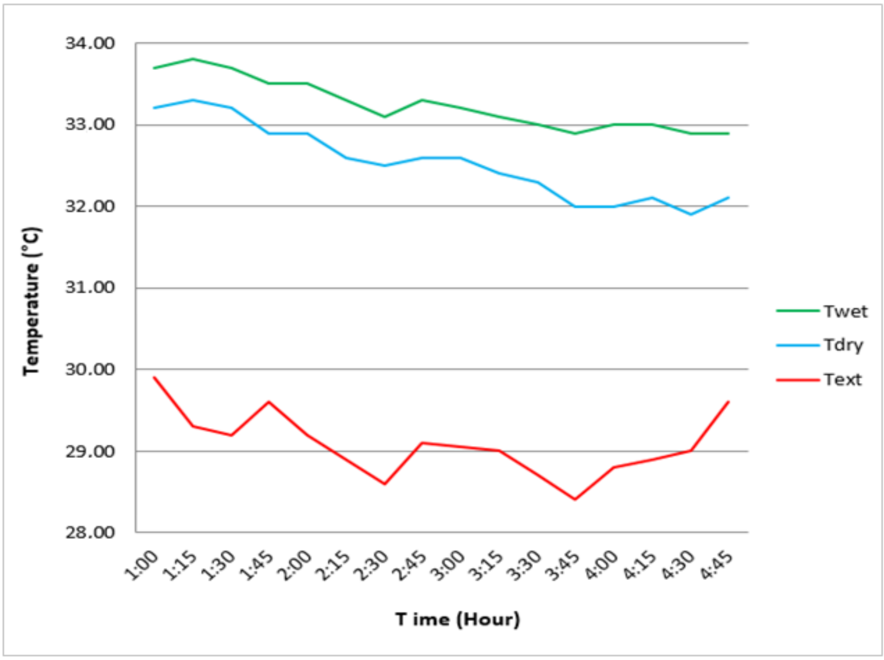

Figure 3-6 Coldest four hours of the hottest day of summer, 11th July 2013. Air temperatures at the outlets of the EAHE in relation to the outside temperature.

3.7 Coldest day of experimental time

The 11th of December 2013 has been found as the coldest day of the experimental period. During that day the average of external air temperatures (Text) were 7.28°C and with a range from 0°C to 13.20°C, while the average, maximum and minimum temperatures at the outlet were 15.07°C, 17.40°C and 12.30°C for wet line (Twet), 14.11°C, 16.70°C and 10.90°C for dry line (Tdry) respectively.

Table 3.9 shows external air temperature and air temperatures measured at the outlet of each experimental pipeline during the coldest day. On that day, the maximum temperature difference between external air temperatures and outlet temperatures were 12.50°C for wet line and 11.20°C for dry line.

Table 3.9. Temperatures measured on the coldest day of winter, 11th December 2013.

	Text	Twet	Tdry	Twet-Text	Tdry-Text	Twet-Tdry
Mean	7.28	15.07	14.11	7.79	6.83	0.96
SD	*5.13*	*1.89*	*2.14*	*3.41*	*3.12*	*0.33*
Max	13.20	17.80	16.70	12.50	11.20	1.20
Min	0.00	12.30	10.90	3.60	3.00	-0.20

Figure 3.7 shows external temperature and temperatures measured at the outlets during the hottest period of the coldest day (from 11:00 to 15:00).

The Figure 3.8 shows external temperature and temperatures measured outlet of each experimental line (wet, dry) during the coldest period. The coldest period of the coldest day was from 3:00 to 7:00.

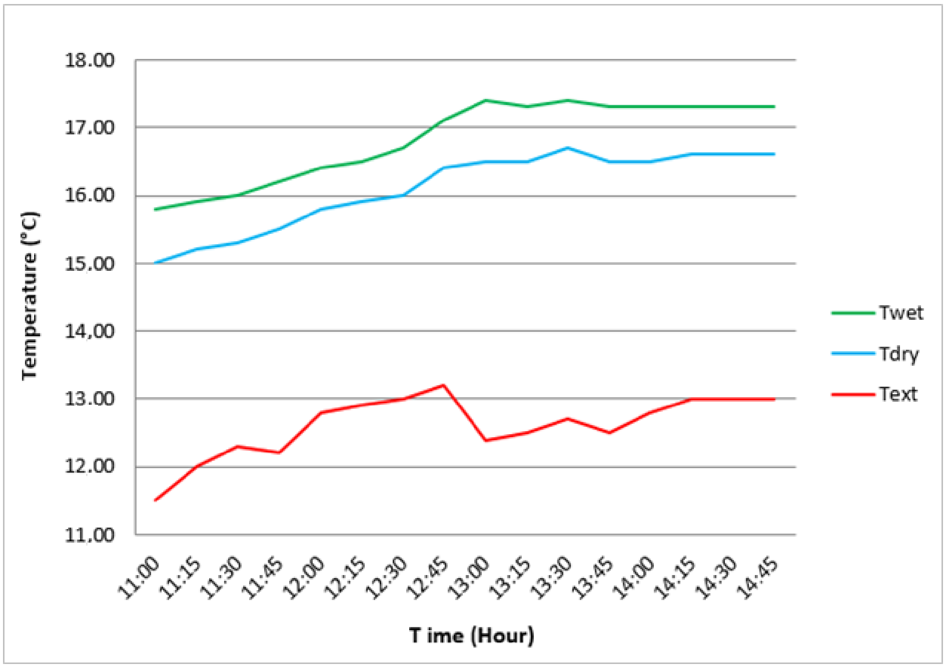

Figure 3-7 Hottest four hours of coldest day of winter, 11th December 2013. Air temperatures at the outlets of the EAHE in relation with outside temperature.

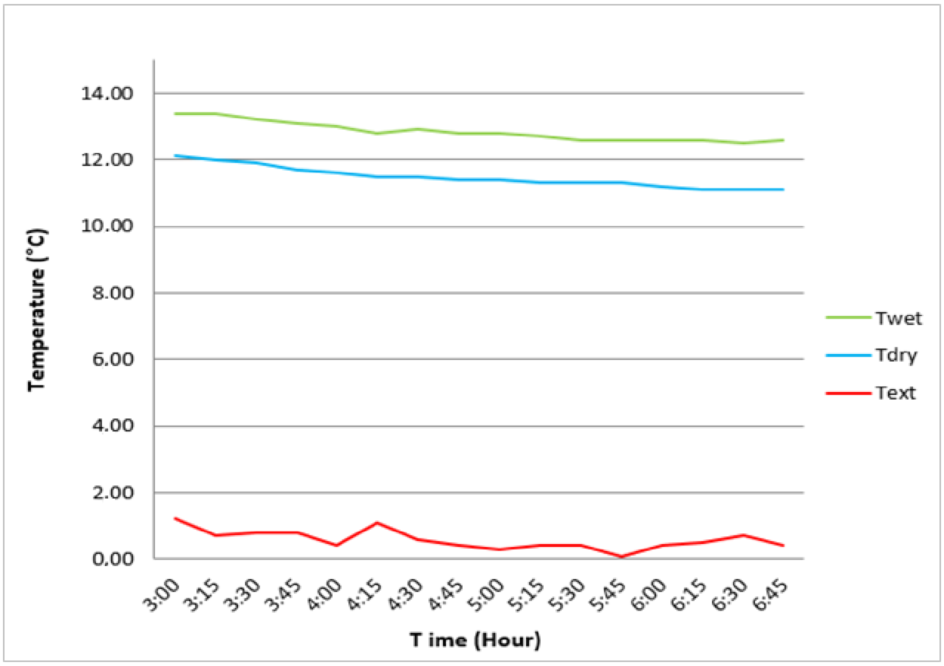

Figure 3-8 Coldest four hours of coldest day of winter, 11th December 2013. Air temperatures at the outlets of the EAHE in relation with outside temperature.

3.8 Effect of pipe length during the hottest period

From June 1st to August 15th 2013, during the 4 hottest hours of each day 12:00 to 16:00, the average air temperature at different lengths of the pipe (12.5 m (Ta_wet 12.5) and 24.5 m (Ta_wet 24.5)) was 41.27°C and 39.44°C for WE while 41.35°C (Ta_dry 12.5) and 40.52°C (Ta_dry 24.5) for DE (Table 3.10). At the same time, the average air temperature at the outlet of pipe was 37.78°C for wet line, while it was 39.18°C for dry line.

Figure 3.9 shows external temperature (Text, distance = 0 m), temperatures inside experimental lines (12.5 m, 24.5 m) and at the outlet of the pipe during this period (12:00 to 16:00; June 1st - August 15th 2013).

Table 3.10. Temperatures measured at the different lengths of the pipeline of the dry and artificially wetted EAHE during the hottest four hours of the day (from 12:00 to 16:00).

	Text	**Ta_wet 12.5**	**Ta_wet 24.5**	**Ta_wet**	**Ta_dry 12.5**	**Ta_dry 24.5**	**Ta_dry**
Mean	46.03	41.27	39.44	37.78	41.35	40.52	39.18
SD	*2.14*	*1.92*	*1.35*	*0.96*	*1.86*	*1.41*	*1.15*
Max	52.30	45.60	42.69	40.60	45.40	43.94	42.70
Min	40.30	36.71	36.02	35.20	36.55	36.69	36.30

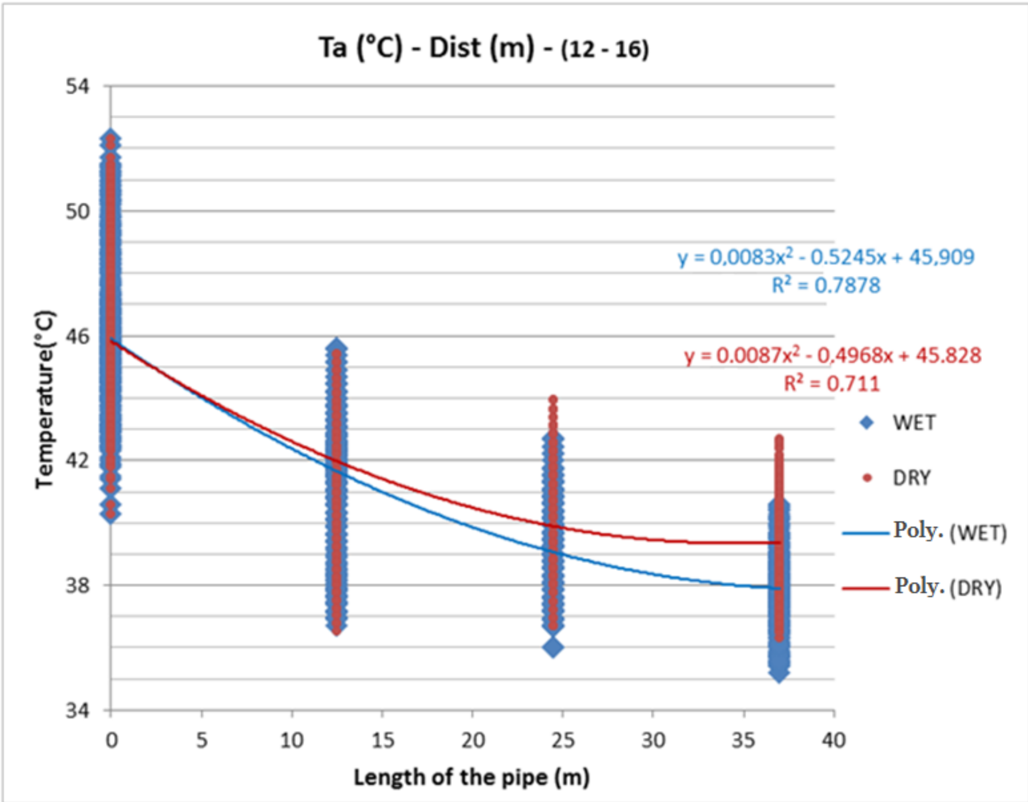

Figure 3-9 Air temperature in the dry and artificially wetted EAHE measured at different lengths of the pipeline during the hottest period of the day (from 12:00 to 16:00; period June-15th August 2013).

3.9 Effect of pipe length during the coldest period

In the other hand, from June 1st to August 15th 2013, during the 4 coldest hours of each night (2:00 to 6:00), the average air temperature at the different lengths of the pipe (12.5 m (Ta_wet 12.5), 24.5 m (Ta_wet 24.5)) and at outlet (Ta_wet) of the pipe were 31.98°C, 32.59°C and 33.26°C for WE while 32.19°C (Ta_dry 12.5), 32.03°C (Ta_dry 24.5) and 32.79°C (Ta_dry) for DE (Table 3.11). Figure 3.10 shows external temperature (distance = 0 m), temperatures inside experimental lines

(12.5 m, 24.5 m) and at the exit of the pipe during that period (2:00 to 6:00; June-15th August 2013).

Table 3.11. Temperatures measured at the different lengths of the pipeline of the dry and artificially wetted EAHE during the coldest four hours of the day (from 2:00 to 6:00).

	Text	Ta_wet 12.5	Ta_wet 24.5	Ta_wet	Ta_dry 12.5	Ta_dry 24.5	Ta_dry
Mean	30.25	31.98	32.59	33.26	32.19	32.03	32.79
SD	*2.58*	*1.33*	*1.11*	*1.22*	*1.37*	*1.12*	*1.48*
Max	37.40	35.74	35.68	36.50	35.91	35.26	36.80
Min	25.10	28.86	29.78	28.90	28.84	29.51	29.00

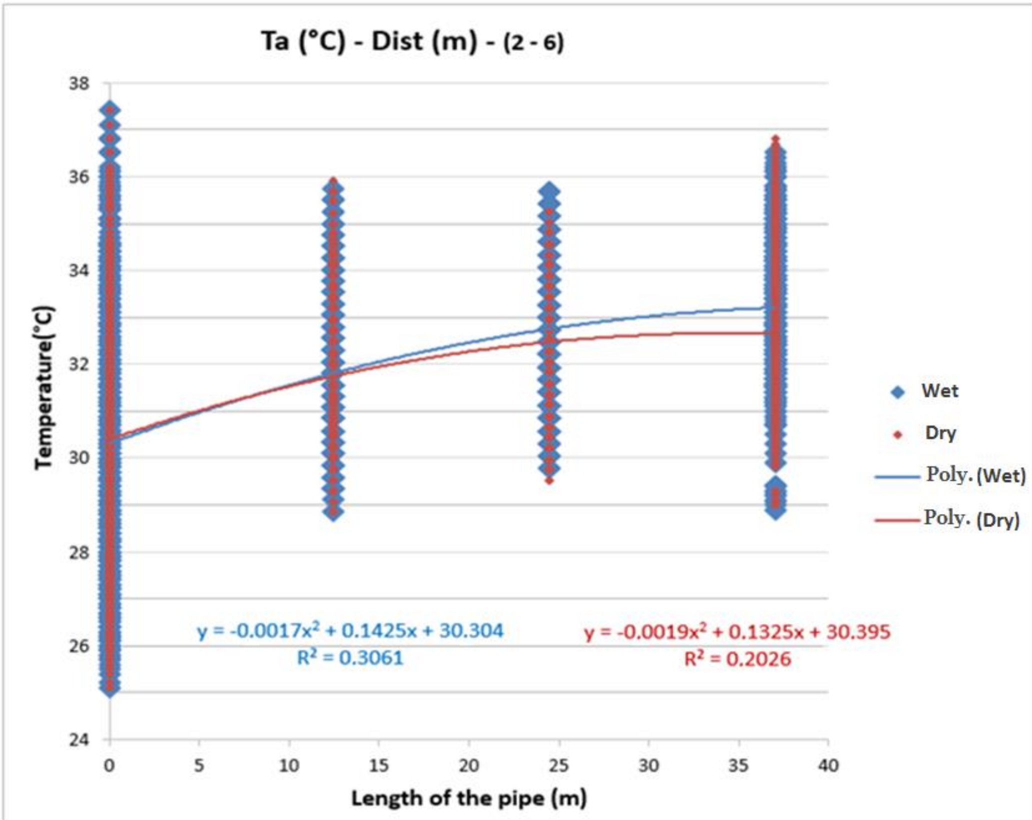

Figure 3-10 Air temperature in the dry and artificially wetted EAHE measured at different lengths of the pipeline during the coldest period of the day (from 2:00 to 6:00; period June-15th August 2013).

Effect of the air running through the pipe on the surrounding soil

From June 2013 to February 2014, during the 24 hours of all days, while temperature of undisturbed ground (Tgr_NOM) was 26.89°C (Table 3.12), the average temperatures of the ground at different distances from the pipe (25 cm (Tgr_25), 50 cm (Tgr_50) and 100 cm (Tgr_100)) were 27.95°C, 27.89°C and 26.94°C, respectively.

Table 3.12. Soil temperatures measured at the different distances from the pipeline of the EAHE from June 2013 – February 2014.

	Tgr_25	**Tgr_50**	**Tgr_100**	**Tgr_NOM**
Mean	27.95	27.89	26.94	26.89
SD	*6.13*	*6.16*	*5.11*	*5.11*
Min	16.86	16.77	17.33	17.48
Max	34.68	34.12	32.24	32.47

To demonstrate the impact of external conditions on the soil temperature at a depth of 2 m, the undisturbed soil temperature at this depth was registered. Figure 3.11 presents the fluctuation of the soil temperatures with the succession night and day during the seasons of the year.

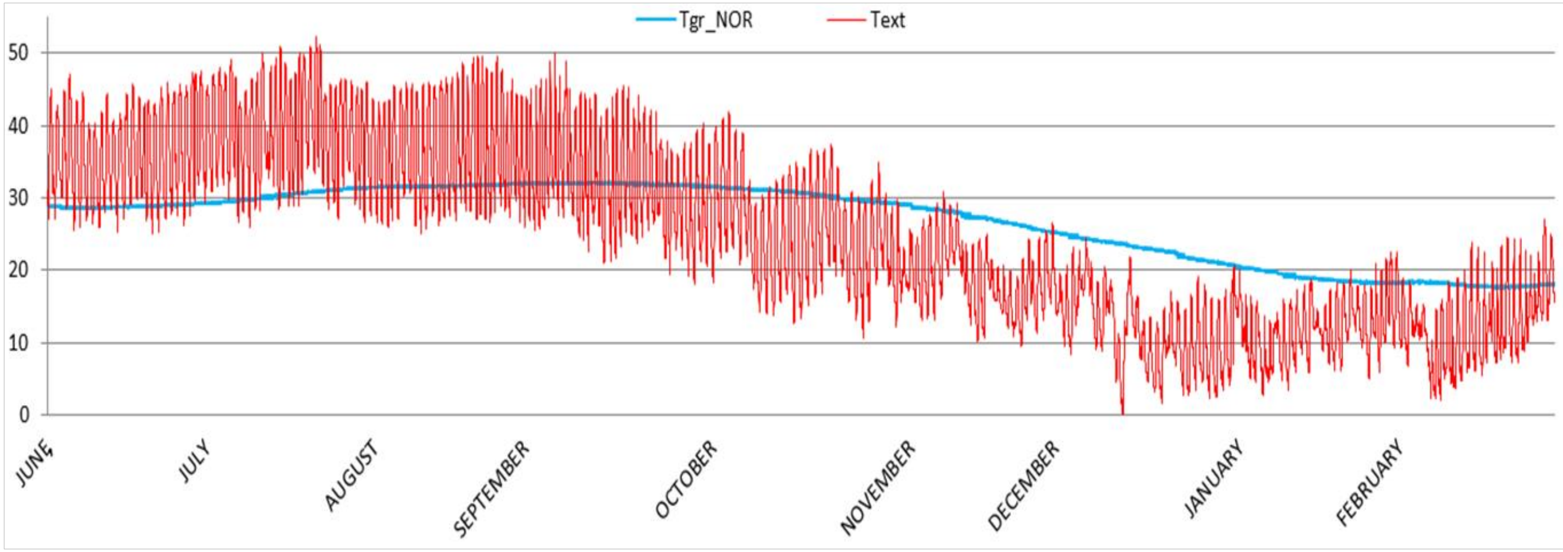

Figure 3-11 Undisturbed ground temperature at 2 m with the ambient temperature from June 2013 to February 2014.

Figure 3.12 shows the relationship between the soil temperatures measured at different distances from the pipeline (25 cm (Tgr_25), 50 cm (Tgr _50) and 100 cm (Tgr_100)) during summer 2013.

A larger effect was observed up to 50 cm distance compared with the 100 cm distance, while the effect of air temperature on the ground at 100 cm was very slight compared with undisturbed soil (Table 3.13). The average ground temperatures at the different distances from the pipe at 25 cm, 50 cm and 100 cm were 33.23°C, 32.99°C and 30.56°C, respectively, while the average temperature was 30.42°C for undisturbed ground (Table 3.13).

Table 3.13. Temperatures measured at the different distances from the pipes during summer 2013.

	Tgr_25	**Tgr _50**	**Tgr_100**	**Tgr_240**
Mean	33.23	32.99	30.56	30.42
SD	*0.85*	*0.90*	*0.88*	*1.25*
Min	31.54	31.41	29.13	28.54
Max	34.68	34.12	31.83	32.22

Tgr_25: soil temperature at a distance of 25 cm, Tgr_50: soil temperature at a distance of 50 cm, Tgr_100: soil temperature at a distance of 100 cm; Tgr_240: undisturbed soil temperature.

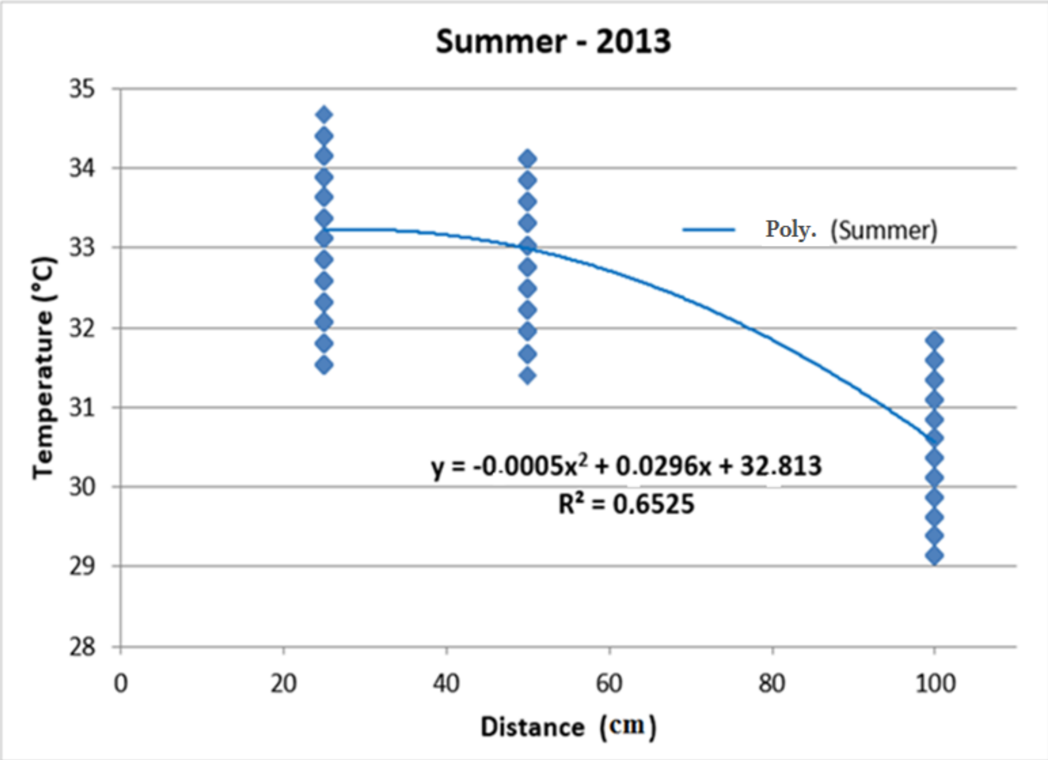

Figure 3-12. Soil temperatures measured at different distances from the pipe of EAHE system during summer 2013.

During autumn 2013, the average soil temperatures measured at different distances from the pipeline (25 cm (Tgr_25), 50 cm (Tgr _50) and 100 cm (Tgr_100)) of the EAHE system were 30.96°C, 30.86°C and 30.10°C, respectively (Table 3.14). The average temperature of undisturbed soil was 30.01°C with a little change respect to summer.

Figure 3.13 shows the relationship between the soil temperatures measured at different distances from the pipeline (25 cm (Tgr_25), 50 cm (Tgr _50) and 100 cm (Tgr_100)) of the EAHE system for autumn 2013.

Table 3.14. Temperatures measured at the different distances from the pipes during autumn 2013.

	Tgr_25	Tgr _50	Tgr_100	Tgr_240
Mean	30.96	30.86	30.10	30.01
SD	*2.54*	*2.68*	*2.01*	*2.05*
Min	25.31	25.18	25.12	25.10
Max	34.41	34.12	32.80	32.47

Tgr_25: soil temperature at a distance of 25 cm, Tgr_50: soil temperature at a distance of 50 cm, Tgr_100: soil temperature at a distance of 100 cm; Tgr_240: undisturbed soil temperature.

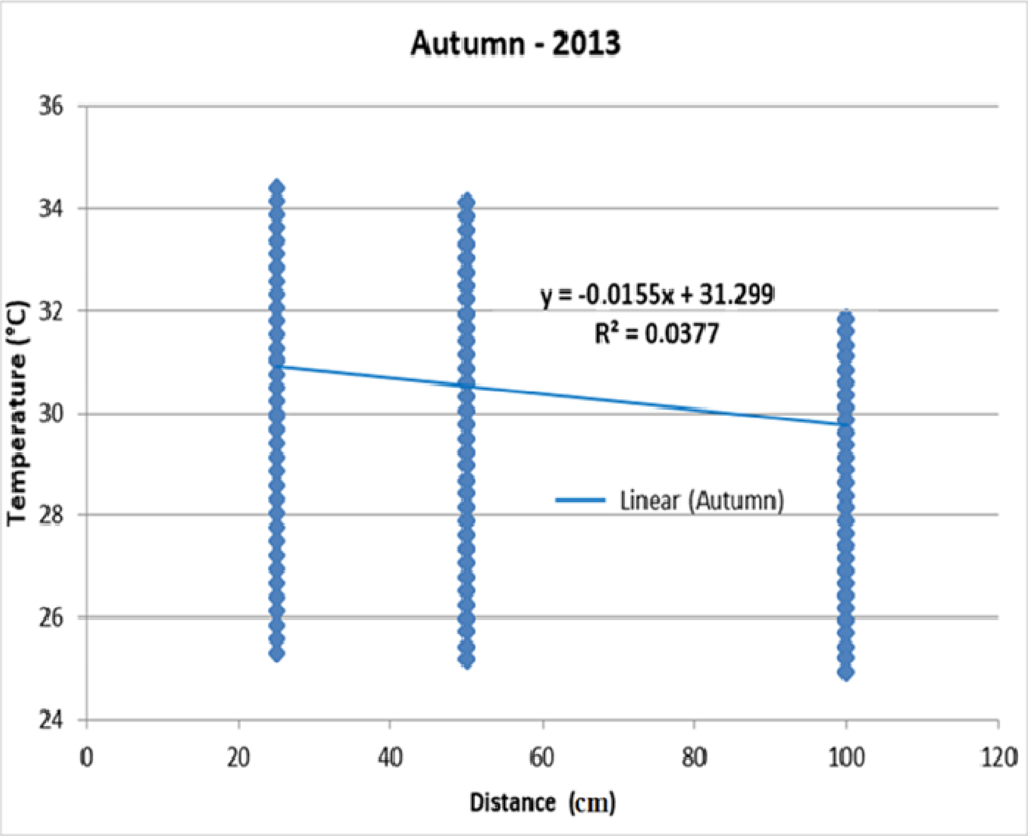

Figure 3-13 Soil temperatures measured at different distances from the pipe of EAHE system during autumn 2013.

During winter 2013, the average soil temperatures measured at 25 cm (Tgr_25), 50 cm (Tgr _50) and 100 cm (Tgr_100) distance from the pipeline of the EAHE system were 19.72°C, 19.76°C and 19.99°C, respectively (Table 3.15). The average temperature of undisturbed soil was 20.18°C with a great change compared to summer and autumn.

Figure 3.14 shows the relationship between the soil temperatures measured at different distance from the pipeline (25 cm (Tgr_25), 50 cm (Tgr _50) and 100 cm (Tgr_100)) of the EAHE system for winter 2013.

Table 3.15. Temperatures measured at the different distances from the pipes during winter 2013.

	Tgr_25	**Tgr _50**	**Tgr_100**	**Tgr_240**
Mean	19.72	19.76	19.99	20.18
SD	*2.34*	*2.33*	*2.30*	*2.34*
Min	16.86	16.77	17.33	17.48
Max	25.31	25.18	24.95	25.34

Tgr_25: soil temperature at a distance of 25 cm, Tgr_50: soil temperature at a distance of 50 cm, Tgr_100: soil temperature at a distance of 100 cm; Tgr_240: undisturbed soil temperature.

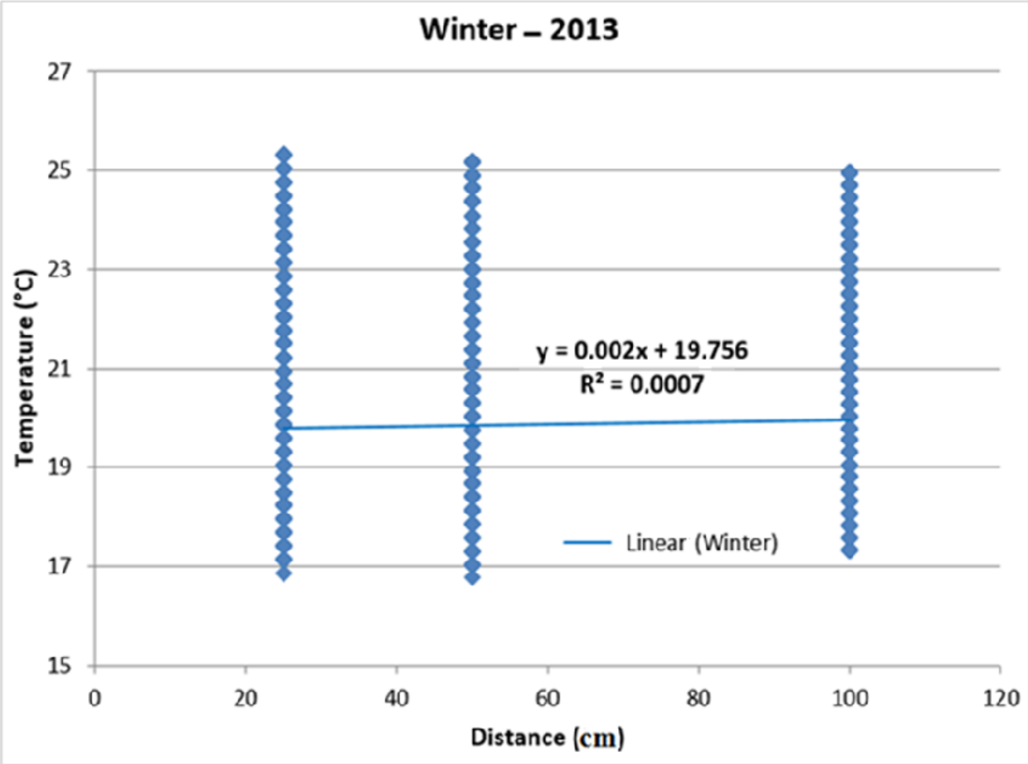

Figure 3-14 Soil temperatures measured at different distances from the pipe of EAHE system during winter 2013.

Figure 3.15 demonstrates the seasonal changes during the experimental time of the temperature of the ground at depth of 2 m and the impact of EAHE system pipe on the surrounding soil. The diagram contains three separate intervals one per each season. The trend lines show the effect of time and distance from the pipeline.

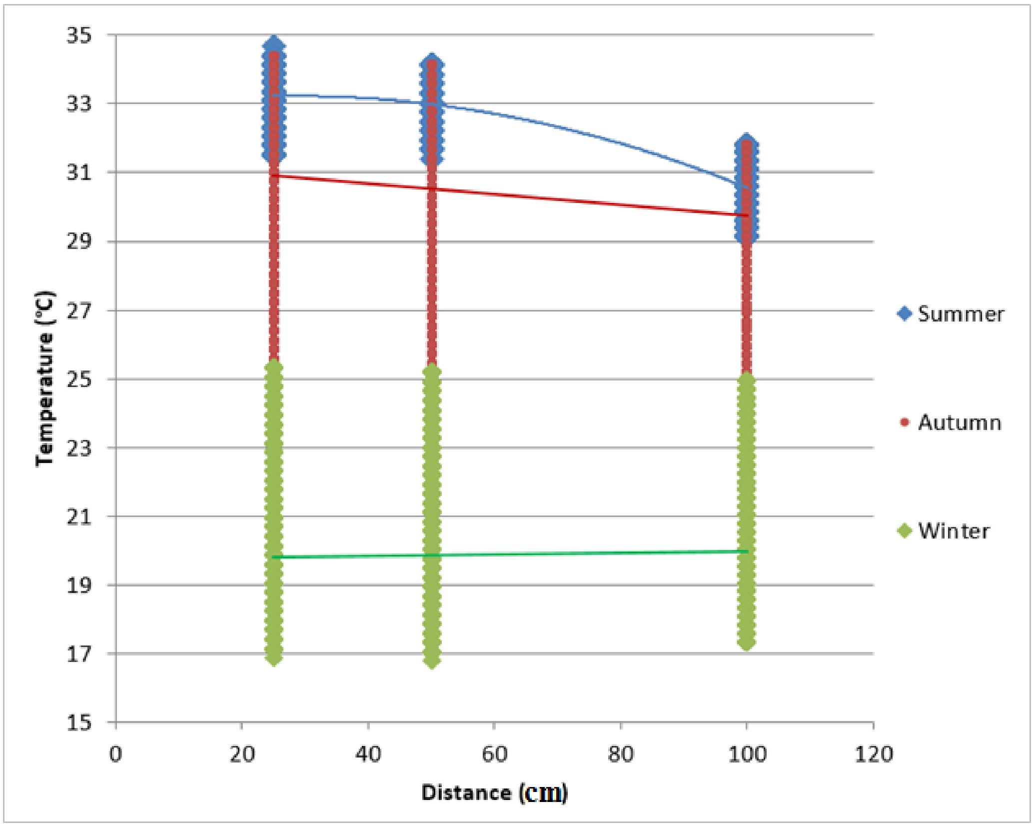

Figure 3-15 Relationship between the soil temperatures measured at the different distances from the pipeline of the EAHE system from June 2013 to February 2014.

Figure 3.16 shows the effect of the temperature of the air passing through the pipe of EAHE system on the surrounding soil. Period of summer was selected among the other to demonstrate this effect. In fact, during this period the influence of the air was the highest. In the study case, considering the equation of the trend line of the temperatures measured at different distances from the pipe, the air effect ends when the distance is up to 1.20 m.

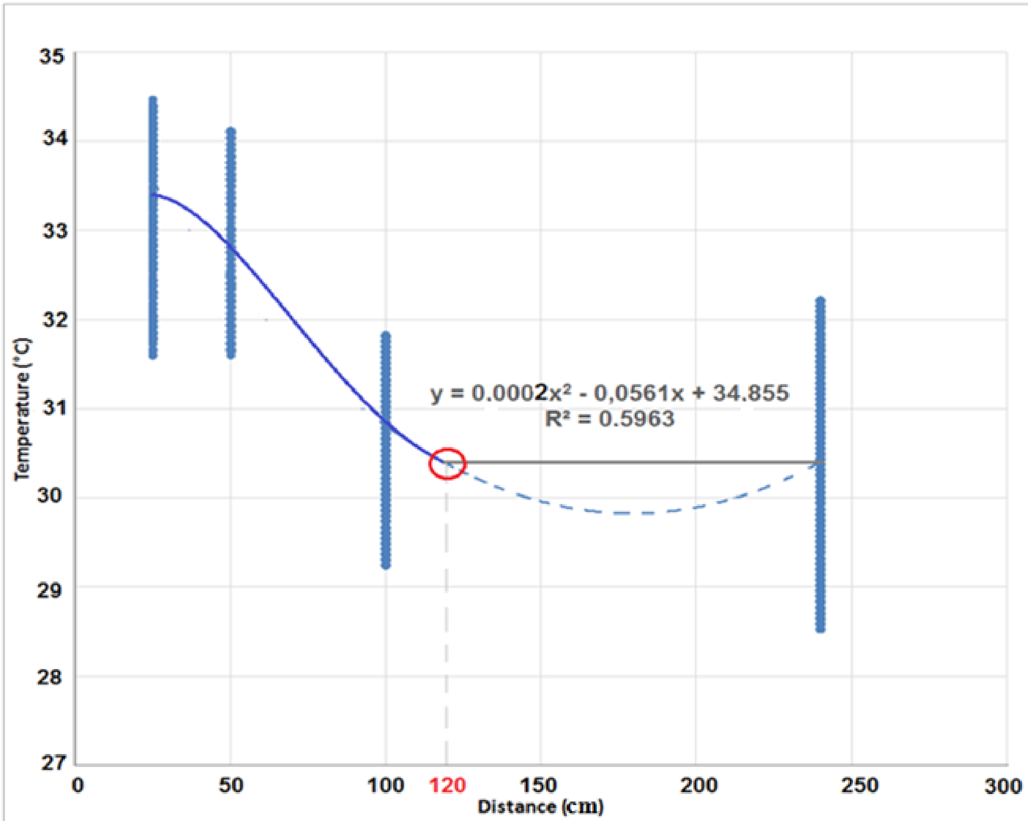

Figure 3-16 Effect of air temperature at different distances from the pipe of the EAHE system.

3.10 Thermal efficiency and performance of EAHE system

During the hottest period of year 2013, the four hottest hours of the hottest day (11 – 15; 11th July 2013) were selected to calculate thermal efficiency and the coefficient of performance of EAHE. The average, maximum and minimum thermal efficiencies were 57.73%, 61.17% and 54.96%, for wet line (EF_wet %), while 47.28%, 50.97% and 43.22% for dry line (EF_dry %) (Table 3.16). The average coefficient of thermal performance (9.39) for wet line (COP_wet) was about two points higher than the one of the dry line (COP_dry = 7.69). At the same way, the range of the coefficient of performance for wet line was 8.60 – 10.52 compared to the range of dry line that was 6.76 – 8.77 (Table 3.16). Figure 3.17 shows the average coefficient of thermal performance during the typical day of summer 2013 for DE and WE.

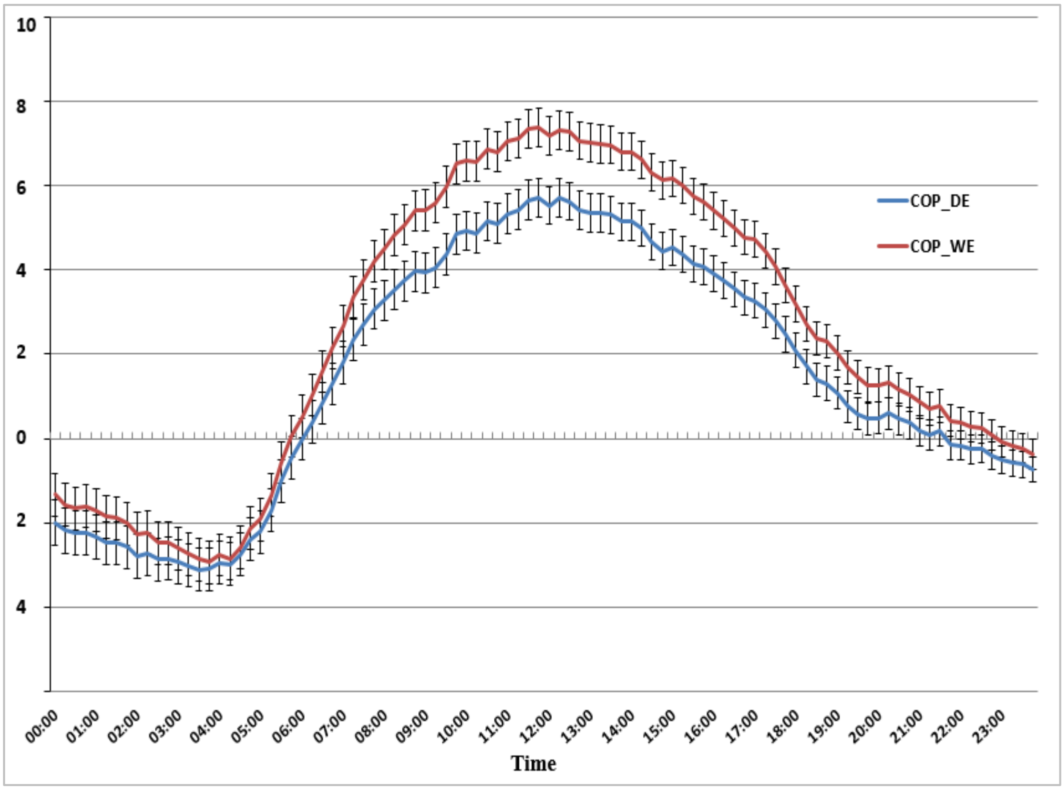

Figure 3-17 Coefficient of performance of EAHE systems during the typical day of summer 2013.

Table 3.16. Thermal efficiency and coefficient of performance of the EAHE system of the hottest period.

	EF_wet %	EF_dry %	COP_wet	COP_dry
Mean	57.73	47.28	9.39	7.69
SD	*2.02*	*2.40*	*0.64*	*0.64*
Min	54.96	43.22	8.60	6.76
Max	61.17	50.97	10.52	8.77

During the coldest four hours of the coldest day (3 – 7; December 11th 2013), the thermal efficiency and coefficient of performance of the EAHE were calculated. The average, maximum and minimum thermal efficiency were 53.45%, 54.66% and 51.82% for wet line (EF_wet %), while 47.57 %, 48.71 % and 45.67 % for dry line (EF_dry %) (Table 3.17).

The average coefficient of thermal performance for wet exchanger (COP_wet = 11.08) was higher than the one of dry line (COP_dry = 9.86). Likewise, the range of the coefficient of performance for wet line was 10.45 –11.56 while the range of dry line was 9.25–10.36 (Table 3.17). In the Figure 3.18, the average of coefficient of thermal performance during the typical day of winter 2013 for DE and WE is shown.

Table 3.17. Thermal efficiency and coefficient of performance of the EAHE system of the coldest period.

	EF_wet %	**EF_dry %**	**COP_wet**	**COP_dry**
Mean	53.45	47.57	11.08	9.86
SD	*0.83*	*0.89*	*0.30*	*0.28*
Min	51.82	45.67	10.45	9.25
Max	54.66	48.71	11.56	10.36

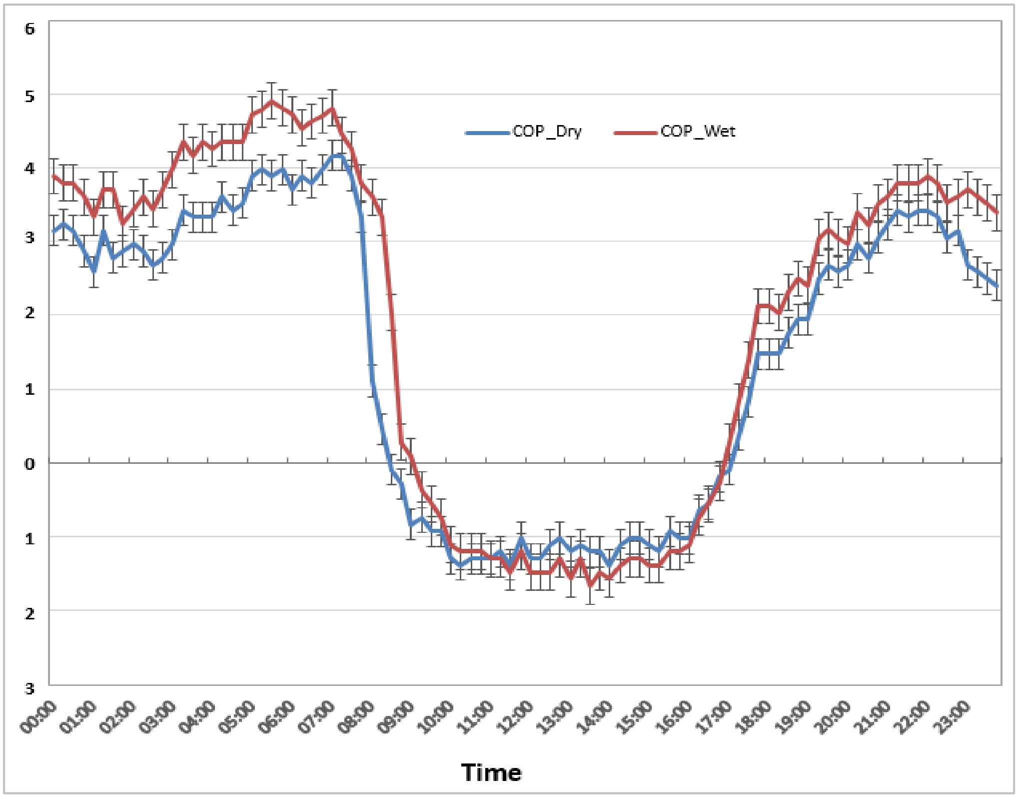

Figure 3-18 Coefficient of performance of EAHE systems during the typical day of winter 2013.

4 DISCUSSION

All tests, which were carried out in the experimental time, show that the WE and DE systems had good capacity to heat or to cool the ambient air temperature, and therefore to improve the environmental conditions for animals in poultry barns. In particular, the WE had the best thermal performance especially during the hottest and the coldest part of the day.

The possibility of cooling and heating was high for both WE and DE systems, because of the wide difference between the external temperature and the undisturbed soil temperature at 2 m depth. These results confirm the effectiveness of earth–air heat exchanger technology in desert or arid land climate (Alghannam, 2012; Al-Ajmi *et al.*, 2006).

The differences for the outlet air temperature and relative humidity of WE and DE systems were significantly different ($p<0.001$, $p<0.001$). The maximum difference between outlet air temperature and external air temperature was 12.60°C for WE system while it was 11.00°C for DE system in cooling mode (Table 3.1). These results are linear with what discovered by Xamán *et al.* (2015). These authors, in researches carried out México City, found that the maximum cooling potential provided by the EAHE was 10°C.

In the present study in the heating mode the maximum difference between inlet and outlet air temperature of WE system was 12.50°C, while in the DE system was 11.20°C. Even in this case, experimental outcomes confirmed the results of heating that were obtained by Bansal *et al.* (2009). They have achieved a heating capacity of 4.10 – 4.88°C, considering a flow velocities of 2 – 5 m/s, 0.15 m inner diameter of the tube with length of 23.42 m and buried at a depth of 2.7 m in a flat land of dry soil.

The maximum difference between outlet air temperatures DE and WE was 3.10°C reached on 3th June 2013 at 14:54. Considerable difference between WE and DE systems was found, included in a range of 0 - 3.10°C. This range depends on the water added to increase the thermal conductivity of the surrounding soil of the pipe.

Another very important achievement of the two EAHE systems in desert conditions was the capacity of switch from cooling in daytime to heating in nighttime. Figures 3.1, 3.2, 3.3, 3.4 show that in typical days when the difference between the external air and outlet temperatures tends to near-zero values the systems changed mode of work.

A second important parameter of climatic conditions influencing the health and productivity of poultry is relative humidity. In the summer times when the value drop under an ideal range of 50% – 70%, an increase of humidity is considered a positive fact (Daghir, 2008; Mashaly *et al.*, 2004). In the time of experiment, both the EAHE systems have demonstrated of being able to increase the relative humidity significantly (Table 3.2).

The soil temperature at a depth of 2 m has been influenced by outside climatic conditions significantly from one month to another, and dramatically between seasons as shown in the Figure 3.11. Therefore, in the month of September the average temperature of the ground was 32.47°C compared to the 17.48°C reached in the month of February.

These outcomes are in accord with the results of several researches about the effect of climatic conditions on soil (Florides and Kalogirou, 2007; Congedo *et al.*, 2012; Peretti *et al.*, 2013), which recommended that the better depth to achieve a constant temperature should be more than 4 m to prevent the solar radiation and climatic conditions effect (Figure 4.1).

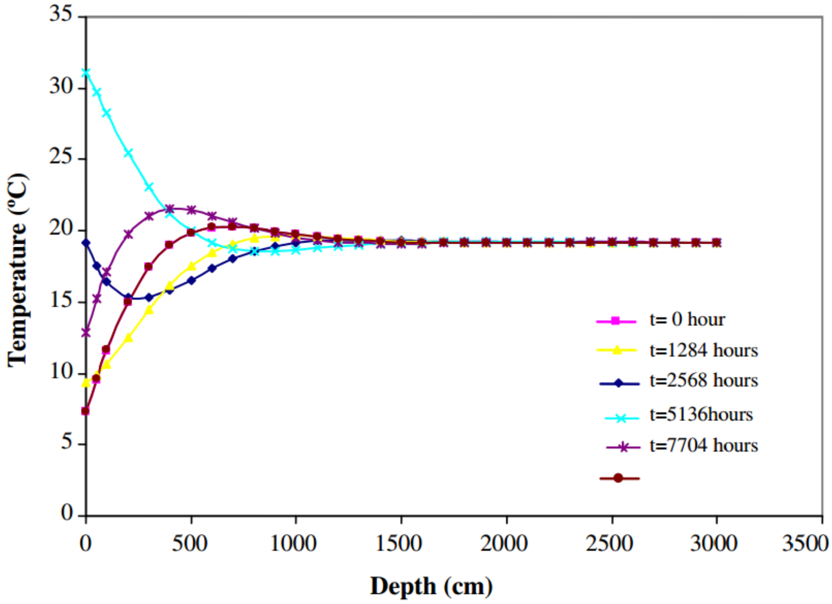

Figure 4-1. Effect of solar fluctuations on soil temperature as a function of depth and time (Ozgener *et al.*, 2013).

Nevertheless, the effect of air in the pipe on the temperature of surrounding ground follows a gradual curve. In particular, the maximum effect of air temperature on soil is near the pipe as shown in the Figure 3.16. The distance from the pipe and the effect of temperature of air is an inverse relationship. Consequently, increasing the distance from the pipe leads to a less effect. Then this effect ends after a specified distance from the pipe (Barbari and Chiappini, 1984; Deglin *et al.*, 1999). The effect of distance depends on physical and mechanical properties of the soil surrounding the pipe.

In this study, the effect of air on the ground temperature reaches 120 cm of distance. Therefore, the distance between two pipes of system should be more of 240 cm.

In the study case, approximately 90% of the heat transfer occurs in the first 50 cm of soil while Deglin *et al.* (1999) have obtained approximately 75% of the heat transfer in only 30 cm in dry sand but the same percentage was reached in 45 cm in saturated silt.

4.1 Summer 2013

In the typical day of summer, the dry line of the experimental system in semi-arid land had a capacity to reduce the external air temperature of 6.20°C (from 46.10 to 39.90°C) while the wet line reduced the external air temperature of 7.40°C (from 46.10 to 38.70°C).

Results demonstrated that WE had a greater capacity to cool and to heat the air than the DE. Wetting the soil around the pipe led to increase heat dissipation by raising the thermal conductivity of the soil. Hence, the outcomes were in line with many researches stating that the effect of moist soil on the heat exchange is a significant parameter affecting thermal performance (Mathur *et al.*, 2015; Bansal *et al.*, 2013; Ozgener *et al.*, 2013). Use the EAHE is viable to reduce costs for heating and cooling the air in poultry facility or other barns (Krommweh *et al.*, 2014).

Negative sign in the difference between outlet temperature and external temperature (Table 3.8) indicates a switch from cooling to heating mode. The air conditioning process changed from heating to cooling at 6:30 and vice versa at 19:30 because of external temperature variation compared with the soil temperature. In South Iraq in summer, the internal temperature of the barn during the night is still high so the heating may not be necessary. Anyway if the system works the air continues to pass through the pipes reducing the temperature of the soil surrounding the pipes and so increasing the efficiency of cooling performance during the next day time.

During the hottest day of summer when the outside air temperature reached 51.10°C, the capacity of EAHE systems (dry and wet) increased. At the same time the difference between WE and DE increased too (Table 3.8).

In this study, the effect of the pipe length was very remarkable during the hottest period of summer 2013, as shown in Table 3.10 and Figure 3.9. The first third of the length of the pipe (12.5 m) reduced the air temperature by 57.7% of the total reduction of the entire pipe. The second third (24.5 m) and last third (37 m) instead, reduced the air temperature of 22.2% and 20.1% of the total reduction for wet exchanger.

At the same time, the dry exchanger recorded reduction of 68.32%, 19.57% and 12.11%, respectively. These percentages are in accord with the results of Deglin *et al.* (1999), which observed that 70% of heat transfer came from the first 10 m of the pipe. Thus, it confirms that the effect of the pipe length is greater in the first half of pipe.

Moreover, with the increase of the pipe length, the difference between WE and DE rises due to rising of the thermal exchange of soil surrounding the pipe artificially wetted (Figures 3.9 and 3.10). The capacity of cooling and heating within the length of pipe for wetted ground in the WE system remains more effective compared with the DE system. This result confirms that a high moisture in the soil increases the heat exchange regardless of other factors as investigated by Ascione *et al.*, 2011; Ozgener *et al.*, 2013; Mathur *et al.*, 2015.

During the summer, the high air temperature led to rise the soil temperature around the pipe diminishing with the distance from the pipe. This effect is more evident in the summer respect to other seasons (Table 3.13). The soil at 25 cm of distance from the pipe has a temperature of about 2.81°C, higher compared with undisturbed soil. Figure 3.12 shows the connection between the passing air temperature and distance from the pipe in summertime. The regression line which describes this relationship has a coefficient of determination R^2=0.6525.

The main result of the research was the significantly difference between the cooling COP of WE and DE earth-to-air heat exchangers. In particular, the maximum and minimum values of the WE system cooling COP was always higher than the values of the DE system (Table 3.16). Figure 3.17 confirms the difference of COP values of the two systems during the whole day. The impact of artificial wetting determined 1.7 point of COP more for WE than DE. Thanks to the increased coefficient of thermal conductivity of the soil around the pipe, the WE system resists better to the variation of ground temperature. Alghannam (2012) observed similar results of the coefficient of performance of earth tube heat exchanger in a greenhouse on sandy soil on desert arid climate of Saudi Arabia. In August, in that environment he recorded a maximum COP of 5.5 during the cooling tests. In the night period, the environmental conditions were reversed. Baxter (1994), as well, obtained COP about 6.30, investigating the performance of cooling mode operation of an earth-tube heat exchanger when operated continuously on the Agricultural Experiment Station at the University of Tennessee, Knoxville. Furthermore, Dubey *et al.* (2013) investigated the coefficient of performance characteristics of a pipe in parallel connection with variable speed in the summer climate. They observed COP reduction from 6.4 to 3.6 with increasing of the velocity from 4.16 to 11.2 m/s. In the same way, for summer cooling in Jaipur (India) Bansal *et al.* (2010) observed a range of COP of 1.9 – 2.9 with the velocity increased from 2 to 5 m/s in EAHE of 23.42 m length.

4.2 Autumn 2013

During the test, in autumn the mean of external air temperature decreased from 37.66°C to 22.28°C respect summer while the humidity raised from 14.55% to 19.22%. In the same time, the undisturbed soil was very little affected by fluctuations of external air. While the ambient temperature change was 15.38°C, the variation in the soil temperature was only 0.57°C. The passage from summer to autumn climate led to change completely the operation way of EAHE system. During all day and all night, it worked in the heating mode.

During the autumn, the capacity heating of EAHE systems had greater value in the night time than in the day time. The main reason for this result is a significant decline of the outside air temperature which achieved values lower than the soil. Therefore, the amount of heat transfer from soil to air raised increasing the air temperature at the outlet of pipes. Figure 3.2 shows that WE system had a maximum capacity to rise air temperature of 10.50°C compared with 7.80°C of the DE system.

Consequently, the soil surrounding the pipe continues to be influenced by temperature of the air passing inside the pipe during time in autumn (Figure 3.13). However, in autumn the difference between the external air temperature and the ground temperature was low. In particular, at 100 cm of distance an increase of 0.09°C respect to the undisturbed soil temperature is remarked.

4.3 Winter 2013

During the winter 2013, the undisturbed soil temperature fell 11.01°C compared to autumn season, while external air temperature dropped 9.97°C. The passage inside the pipe of air at a very low temperature led to a reduction of the surrounding soil temperature especially in the night. In this condition, during some hours of the day the external air temperature was greater than wall pipe temperature. Since that, the EAHE systems changed mode of operation from heating to cooling. Several researchers (Krommweh *et al.,* 2014; Ozgener, 2011) obtained similar results. However, during the daytime the air at outlet should not enter in the poultry barn but the system should continue to work in order to take heating advantage of the surrounding soil for the next night functioning.

The analysis of coldest day indicated that the earth-to-air heat exchanger can provide an effective method for preheating in cold period, and that the heating capacity of the plant during the winter is significantly important especially during the coldest hours of the day. Moreover, during the coldest day, the system operated always in heating mode both in nighttime and in daytime. Even in winter time, the difference between WE and DE techniques during coldest day was significant.

The WE system heated the air more than the DE system both in the night and in the day. Anyhow, both the techniques have demonstrated to be effectively useful for heating or at least for preheating the air for the poultry barn (Figures 3.7 and 3.8).

In a similar study, Ghosal *et al.* (2005) found that the air temperature of a greenhouse in winter was on average 6 – 7°C higher than in a comparable greenhouse located in New Delhi (India) which operated without EAHE. In another research, Sethi and Sharma (2007) found that heating was effectively realized with an aquifer coupled to cavity flow heat exchanger system. They confirmed this system was able to maintain the air temperature inside the greenhouse room from 7 to 9°C above outside air in winter months.

In addition, Xamán *et al.* (2015) investigated the pseudo transient thermal behavior of an earth-to-air heat exchanger in three different climatic conditions of México. They found that the maximum heating potential achieved by the EAHE in Juárez, Mérida and México City was 6.3°C, 12.5°C and 3.2°C, respectively.

In winter, the air temperature drop to 0°C affected significantly the undisturbed soil temperature, which decreased about 10°C from autumn. However, despite the underground temperature reduction, the subsoil at 2 m depth still worked as a heating resource. On the other hand, the impact of cold air that passes in the pipe was remarkably high, especially at 25 cm. The average soil temperatures at 25, 50 and 100 cm from the pipe were less than the temperature of undisturbed soil (Figure 3.14).

The average heating COP in the coldest four hours of the coldest day of the year was 11.08 for WE and 9.86 for DE proving that the heating capability of EAHE systems in cold environments was equal to the cooling capacity in hot case. Wetting the soil surrounding the pipe in cold period had similar effect as in the hot period. An increase of conductibility of soil led to a rise of the efficiency of EAHE systems. In fact, the difference of COP between DE and WE was significant.

In similar environment condition**,** Sharan and Jadhav (2003) studied the performance of a EAHE system with pipes 50 m length, 10 cm diameter and 3 m depth, which was installed near Ahmedabad campus in India. The average heating COP recorded was 3.8 for the 14 hours of continuous work in the night. Baxter (1992), instead, obtained COP average hourly values from 1.6 to 4.2 during the entire period of test and the greatest hourly values of COP were from 3.2 to 10.3.

4.4 Summer 2014

After one year of operating, WE and DE systems achieved results of thermal performance similar to the previous year (2013), as shown in Figure 3.4. In August 2014 the EAHE systems had the same capacity to cool or to heat the air passing inside the pipe respect to August 2013. Howsoever, the effect of wetting the soil nearby the pipe was clearly remarkable. Because the external air temperature was higher in 2014 than 2013 especially during the nighttime, the cooling performance was superior as well. For this reason, the heating mode during the night had very short time.

The EAHE systems were still able to accumulate or remove heat in the soil surrounding the pipe by cooling mode and by heating mode. In summer condition, the temperature fluctuation between day and night times was very useful to improve the cooling performances for both the WE and DE systems. The EAHE systems helped the soil around the pipe to dissipate away the accumulated heat during the cooling mode. This result is in line with the numerical investigation study conducted by Mathur *et al.* (2015), which demonstrated that the intermittent mode operation of EAHE system allows the accumulated heat in the soil surrounding the pipe to dissipate away from the pipe. They investigated the thermal performance of earth-air tunnel heat exchanger under three operating modes. The first one worked continuously for 12 h, the second run for 60 min and remained off for 20 min and the third was 60 min on and 40 min off. Then, they found that the second and third mode increased thermal performance of the system of 3.35% and 3.56% in terms of heat transfer rate compared with first mode in the same soil.

5 CONCLUSION

The relatively cold soil can reduce the electrical costs for cooling and heating the air in a semi-desert area as the Basra Province (Iraq). Both DE and WE cooling systems can be considered useful solutions to create better thermal conditions inside the barns, so reducing heat stress of animals in poultry barns during the hottest period. Anyway, the use of WE can give better results in reduction of temperature in livestock barns. The WE and DE techniques were able to reduce the air temperature by 12.60°C and 10.60°C in July during hottest period, and to rise the air temperature by a similar amount 12.50°C and 11.20°C during the coldest period in winter in study year, respectively.

This work confirms that wetting soil technique around EAHE can improve the heat exchange efficiency. By adding a drip water tube, the WE system reduced the temperature of the incoming air more efficiently than the DE system, especially when the difference between the temperatures of the outside air and the soil was lower. In particular, during the hottest hours, the average cooling COP of WE was 1.70 points higher than the one of the DE, whereas the average ΔT was 0.76 degrees higher in WE than the one in DE. Whereas, the WE system raised the temperature of the passing air more efficiently than the DE system. Particularly, the average heating COP was 11.08 for WE system and 9.86 for DE in coldest time in winter.

The length of pipe had a significant impact on the performance of EAHE systems, but with artificial wetting, this effect became more remarkable compared of DE system. The differences between the two systems were 0.08°C, 1.08°C and 1.40°C at 12.5 m, 24.5 m and 37 m, respectively.

The effect of air temperature on the soil surrounding the pipe was marked into 120 cm from the pipe. Therefore, the smaller distance between the pipes must be not less than 250 cm.

Therefore, it can be clearly concluded that the performance of EAHE for WE increases during running operation thanks to wetting of soil surrounding the pipe.

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