# A TRANSIENT PERFORMANCE EVALUATION OF A POROUS EVAPORATIVE COOLER FOR PRESERVATION OF FRUITS AND VEGETABLES.

**BY**

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**REG NO: 20174078678**

**A THESIS SUBMITTED TO THE POSTGRADUATE SCHOOL, FEDERAL UNIVERSITY OF TECHNOLOGY OWERRI, IMO STATE.**

**IN PARTIAL FULFILMENT OF THE REQUIREMENTS FOR THE AWARD OF MASTERS OF ENGINEERING (M.ENG.) IN MECHANICAL ENGINEERING.**

**NOVEMBER, 2023**

## CERTIFICATION

This is to certify that this work **"A Transient Performance Evaluation of a Porous Evaporative Cooler for Preservation of Fruits and Vegetables"** was carried out by **Rosemary Oluchi Paul-Okore (Reg. Number: 20174078678)** in partial fulfilment for the award of Degree of Master of Engineering in Mechanical Engineering, Federal University of Technology, Owerri.

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## Dedication

I dedicate this project to God almighty for giving me the desired wisdom, knowledge and understanding required to complete this work.

I also dedicate this to my late Dad Sir Kieran Azuwuike Nwaofor. Daddy, this is for you and may your gentle soul continue to find peace in the bosom of the Lord, Amen!

## Acknowledgement

The completion of this project work has indeed been a knowledge packed experience as I was exposed to different aspects of conducting a research work. Therefore, I owe everyone that went through this journey with me a profound gratitude for extending their hands of assistance. I wish to extend my warm gratitude to my supervisor and Lecturer Engr. Prof. N.V. Ogueke for his immeasurable and continuous support, patience, motivation, enthusiasm, immense knowledge and dedicated involvement throughout the whole process of my M. Eng project. Thank you Prof. for being very thorough with the reviews of my thesis from the beginning to the final level, your encouragement, guidance and assistance helped me to develop a better understanding of this work. I would also like to thank my other supervisor, Engr. Dr. G.N. Nwaji for his excellent supports, suggestions, for providing the FlexPDE computational fluid dynamic analyzer for my analysis and for his detailed and concise contributions towards the actualization of this work.

I wish to also acknowledge Engr. Prof. E.E. Anyanwu for his fatherly role, advice and consistent encouragement. Prof., I very much appreciate all your efforts in ensuring I actualize this work by walking me through the right path whenever I drift away from the actual aim of this project and making sure I receive any necessary assistance I desire towards the completion of this work. Your indispensable fatherly role is second to none. Also, I won't fail to acknowledge immediate past H.O.D Engr. Dr. G. O. Obiukwu for his support and advice and also the current H.O.D Engr. Dr. G.O Onuoha. To our very experienced technologists in the department of Mechanical Engineering; Engr Ebuka Nwagu and Engr Emuka for their wonderful technical supports. My heartfelt gratitude goes to my Lecturers Engr. Prof. J.O. Igbokwe, Engr. Dr. O.C. Nwufo and Engr. Dr. C. Okoronkwo for impacting the right knowledge required to see through this project work. I am highly indebted to many of my very supportive colleagues in the department of Mechanical Engineering, Federal University of Technology Owerri. They are Engr. Dr. O.C. Nwufo for always giving me listening ears whenever I approach him with my numerous academic challenges, Engr. Dr. J.O. Azubuike, Engr. B.U. Okonkwo, Engr. C. Ike, Engr. K. Chukwuemeka and Mr. D.A. Ekpechi for their supports and contributions.

My gratitude also goes to my amiable course mates Engr. (Miss) .C. Ezeaku, Mr. C.R. Dimson and Mr. C.S Amuta for their excellent support, advice, contributions and assistance. I acknowledge my ever supportive friend and professional colleague Mr. A. Waribo, your informal support and encouragements have been indispensable. My warm and sincere gratitude goes to my lovely husband Engr. Paul Okore Kalu who has been a constant source of support emotionally, financially and morally and without whom this project work would certainly not have existed. I wish to also extend my heartfelt gratitude to my amiable mom Mrs. G.U. Nwaofor and my siblings Bar. Kennedy .O. Nwaofor and Mr. Macdonald . C. Nwaofor for their selfless and endless love, care and encouragement. I would not have achieved so much without them.

Above all, I acknowledge my maker, the almighty God for the wisdom, knowledge, understanding and strength to pull this project work through and I pray that this thesis and any publications from it should be a useful guide for other researchers.

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## NOMENCLATURE

| Symbol | Description | Unit |
|---|---|---|
| $\Delta U$ | Change in internal energy | $J$ |
| $R$ | Specific gas constant | $\frac{J}{Kg}.K$ |
| A | Area of the porous wall | $m^2$ |
| $K_1$ | Thermal conductivity of clay | $W/m°K$ |
| $K_2$ | Thermal conductivity of water | $W/m°K$ |
| $\varepsilon$ | Porosity of clay | |
| $DF$ | Driving force of the system | |
| $h_{fg}$ | Latent heat of vaporization of water | $KJ/Kg$ |
| $h$ | Heat transfer coefficient | $W/m^2°K$ |
| $\bar{A}$ | Amplitude | $m$ |
| $\bar{B}$ | Annual mean temperature | $°K$ |
| $\xi$ | Time in day | $hrs$ |
| P | Period of time | $days$ |
| N | Number of day at month | |
| $Nu$ | Nusselt Number | |
| $Gr$ | Grashof number | |
| $Pr$ | Prandtl number | |
| $g$ | Acceleration due to gravity | $m/s^2$ |
| $\beta$ | Coefficient of volume expansion of air | $°K^{-1}$ |
| $\nu$ | Kinematic viscosity of air | $m^2/_s$ |
| $\mu$ | Dynamic viscosity of air | $kg/_{m.s}$ |
| $D$ | Diffusion coefficient of water | $m^2/s$ |
| $M$ | Molecular mass | |
| P | Total Pressure | Pa |
| $G$ | Universal gas constant | $KJ/_{kg} - Mol.K$ |
| $P$ | Partial pressure | Pa |
| $h_m$ | Convective mass transfer coefficient | $W/m^2°K$ |
| $C$ | Concentration | |
| Le | Lewis number | |
| $\alpha$ | Thermal Diffusivity | $m^2/_s$ |
| D | Mass Diffusivity | $m^2/_s$ |
| $\omega$ | Specific humidity | g/kg |
| $\rho$ | Density | $kg/m^3$ |
| $C_p$ | Specific heat capacity | $KJ/KgK$ |
| $Q$ | Heat transfer | $W$ |
| $L$ | Characteristic length of the porous wall | $m$ |
| $K$ | Thermal conductivity of the porous wall | $W/_{m.°K}$ |

| | | |
|---|---|---|
| $\dot{m}$ | Mass flow rate | $Kg/s$ |
| $V$ | Volume | $m^3$ |
| $T$ | Temperature | $°K$ |
| $W$ | Wind velocity | $m/s$ |
| T | Time | $hrs$ |

***Subscripts***

| | |
|---|---|
| *A* | *Air* |
| *V* | *water vapour* |
| *ha* | *humid air* |
| *O* | *outer porous wall* |
| *I* | *inner porous wall* |
| *opm* | *outer porous medium* |
| *ipm* | *inner porous medium* |
| *int* | *internal energy* |
| *conv* | *Convection* |
| *D* | *dry air* |
| *ch* | *Chamber* |
| *sc* | *storage chamber* |
| *W* | *Water/moisture* |
| *cond* | *Conduction* |
| *eff* | *Effective* |
| *rep* | *Replacement* |
| *sf* | *Surface* |
| *amb* | *Ambient* |
| *C* | *Clay* |
| *isp* | *Interspace* |
| *ipw* | *inner porous wall* |
| ∞ | *Natural(free)* |
| E | *Experimental* |
| N | *Numerical* |
| *T* | *Time* |
| *dp* | *dew point* |
| *ev* | *Evaporated* |
| *wd* | *water diffused* |
| *max* | *Maximum* |
| *min* | *Minimum* |
| *mn* | *Mean* |
| *In* | *inflow/influx* |
| *out* | *outflow/efflux* |
| *G* | *internal generation* |
| *S* | *Sensible* |
| *L* | *Latent* |

**Abstract**

Post harvest loss is a major factor affecting commercial farming. Efforts to reduce it using the conventional cooling systems have not been successful to cost and unavailability of grid connected electricity. Evaporative cooling is a promising alternative but this work is required to improve on its overall performance. To do this, the transient performance under different climatic conditions is required for better understanding and possible system components optimization. This work therefore presents the transient performance evaluation of a porous evaporative cooler carried out using a mathematical model developed from first principle. The model is based on an energy balance and mass transfer analysis on different parts of the evaporative cooler. The developed model was solved using FlexPDE computational fluid dynamics analyser based on the finite element numerical approach. The numerical solution was validated using experimental data. Results obtained showed that the model very closely predicted the actual system performance with a Root Mean Square Error (RMSE) of 0.205. In general it was observed that the evaporative cooler maintained a significantly lower storage temperature (20.9 – 24 )$^{O}$C compared to the ambient temperature (27 – 33)$^{O}$C for all climatic seasons of the year with the best performance recorded during the late dry season (January). Temperature difference between ambient and storage space during the hot periods of the day was in the range of 3 – 9 $^{O}$C. Thus, the evaporative cooler has good potentials for all year round reduction of post harvest losses and the developed model is a good tool for the evaporative cooler performance optimization.

**Keywords:** Temperature; Evaporative Cooling; Model; Preservation; Fruits and Vegetables.

# CHAPTER ONE

# INTRODUCTION

## 1.1 Background of the Study

Generally, food is essential to support life and good health. Some variants of food such as fruits and vegetables are very essential because of their very high vitamins content. Therefore, their year-round availability is important for healthy living.

In Africa, majority of the populace living in the rural areas depend mostly on proceeds from their farm for survival. Also, both those living in rural and urban areas require farm produce as food for nourishment. However, most of the freshly harvested products rot away during their periods of harvest and therefore, sold at cheaper prices owing to lack of affordable storage facilities (Yahaya and Mardiyya, 2019). Some are lost during the times of handling and transportation. Fruits and vegetables are the most susceptible to spoilage. They are easily affected by heat and easily attacked by pest, fungi, bacteria and viruses hence reducing their shelf life. Basediya et al. (2013) reported that low temperature and high humidity are essential for the preservation of fruits and vegetables. This is so because under these conditions, pathological activities are slowed down hence extending the shelf life of the products.

Mechanical refrigerators are very useful storage facilities as they provide the necessary space conditions that preserve perishable food products. However, they are quite expensive for local farmers to purchase and maintain. Lack of constant grid power supply is also a huge constraint. In many parts of Africa, for instance, Nigeria, nonexistent/irregular electricity supplies or the high cost thereof deny many rural dwellers access to the needed mechanical cooling for thermal comfort, food and medicine preservation and storage (Nwaji & Anyanwu, 2022). However, ancient methods of passive cooling are available to satisfy human comfort and preserve the shelf life of farm produce. One such method is cooling using clay pots, Mohammed Bah Abba developed a pot-in-pot refrigeration system for the preservation of fruits and vegetables (Strauss, 2016). Unfortunately, it is not able to extend shelf life beyond a few days (Nwaji & Anyanwu, 2022).

In order to meet the global food demand, agricultural produce losses must be reduced to the barest minimum. Evaporative cooling method presents one of the cheapest and reliable techniques of satisfying this requirement due to its low energy requirement and simple technology. Evaporative cooling is a cold production process which uses the cooling effect of evaporated water to cool airstream within an enclosure. It is based on the traditional method of cold production by placing water in porous jars exposed to ambient environment after sunset. The night breeze evaporated the moisture, which seeped through the jars, making the water inside cooler (Anyanwu, 2004). The traditional system has progressed from this rudimentary process to gain wide applications for space conditioning in comfort cooling and food preservation.

The study of psychrometry has armed us better with ideas and options on how to employ evaporative cooling effectively to condition hot space. With the increasing demand for climate control and environmental sustainability, evaporative cooling seems one of the easiest techniques to employ to achieve the twin goals.

Evaporative cooling systems performance is highly dependent on the following:

i. High Temperature,
ii. Available air movement from wind or electric fans.
iii. Low humidity of the surrounding air
iv. Availability of water

The schematics of an evaporative cooling system is as shown in figure 1.1.

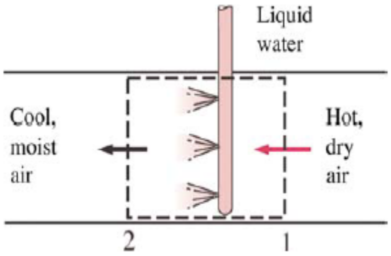


Figure 1.1: Schematics of Evaporative Cooling System (Bansal & Mathur, 2009)

## 1.2 Problem Statement

Heat induced post harvest losses is a major factor responsible for the high cost of agricultural produce and lack of interest in large scale farming as a result of loss of revenue occasioned by the high losses. The use of conventional systems like the vapour compression systems have not been successful due to the high cost associated with its use and the unreliable or unavailable grid connected electricity. Passive cooling approach, like the porous wall evaporative cooler, despite showing good potentials, has not gained wide market acceptance due to the current low level of performance (Abdullah et al., 2023). There is therefore need for the studies to improve the performance. To do this, the mechanism and dynamics of the system need to be well studied using a model that can give its transient behaviour. Previous models over simplified the system and as such will not provide the needed information to improve the design. This work address this by developing a transient model that can give better insight into the systems components behaviour during operation and subsequently use the model to study the transient performance of the evaporative cooler.

## 1.3 Objectives of Study

The major objective of this research is to carry out a transient performance evaluation of a porous evaporative cooler for preservation of fruits and vegetables.

The specific objectives are:

i. To develop a transient thermal model of a porous evaporative cooler for preservation of fruits and vegetables.
ii. To determine the performances of the model in (i) above using Flex PDE finite element model builder and numerical solver version 7.21.
iii. To validate the numerical data with available experimental data.
iv. To study the evaporative cooler performance during different seasons of the year.

### 1.4 Justification of the Study

The current poor performance of the porous evaporative cooler is largely due to the not very clear understanding of the contributions of the various components of the evaporative cooler to the over all system performance. the model developed in this work will enable a better understanding of the contributions of the components, thus leading to design of system with improved performance. it will also help to better predict the systems performance during different climatic seasons and in different meteorological environment

### 1.5 Scope of Study

This work is restricted to the development of one dimensional porous evaporative cooler model equations, simulation and prediction of the system performance during different climatic seasons using Flex PDE finite element model builder and numerical solver version 7.21. It does not include parametric study of the system's components.

# CHAPTER TWO

# LITERATURE REVIEW

## 2.1 Traditional Evaporative Cooling

Evaporative cooling technique has been applied in comfort cooling for thousands of decades. This technique evolved from many ancient practices by Egyptians and Persians(Iran) who fanned water-filled porous pots to produce cool air (Davis, 2016). Also, ancient paintings by Egyptians portrayed scenarios where slaves fanned large porous clay pots and individuals sleeping outdoors with wet sheets hung beside them just to get relief from the summer heat (Davis, 2016). In Rome, people circulated water from channels through building walls to get relief from summer heat (Davis, 2016). The less privileged survived summer heat waves just by hanging wet clothes over the doors of their tents so that fresher and cooler breeze will be driven in (AzEvap, n.d.)

In ancient times, our forefathers lived in cave houses as a defense and security measure (McFadden, 2017), but they discovered that the cave houses were always cool all year round hence, they started living in it for thermal comfort just to control heat during hot periods. A picture of the cave is shown in plate 2.1. In Cappadocia (i.e. Turkey), whole cities were built underground just to control heat and provide the desired comfort conditions.

Plate 2.1.: Typical Traditional Cave House (Nwaji & Anyanwu, 2022)

Recently, these cave houses have been modernized into today's underground structures e.g. basements, split-level homes, and homes built into hillsides (Nwaji & Anyanwu, 2022) as shown in plate 2.2.


Plate 2.2: Hillside built homes as underground pavements for all year-round cool space (Nwaji & Anyanwu, 2022)

Researchers have studied the physics behind these practices and engineered it into functional and useful evaporative cooling systems classified according to their cooling technology as direct, indirect and combined indirect/direct evaporative coolers respectively.

Direct evaporative coolers (DECs) are the oldest evaporative cooling systems. They cool an airstream through the process of evaporation initiated when water absorbs the sensible heat in air. For this system, outside air is directly passed through a wetted pad, the water in the pad absorbs the sensible heat required for water evaporation from the air thus reducing the air dry bulb temperature while increasing its relative humidity. DECs differ according to the electrical power consumption required for their operation. Some are operated naturally hence require zero electrical power for their operation. Such systems are referred to as Passive DEC while others are actively operated using an electrical blower which is power driven. They are referred to as Active DEC. Although active coolers are power driven, they have energy saving efficiency of up to 90% thus they are more energy efficient when compared to other traditional cooling techniques 90% (Lechner, 2009). DECs are most suitable for hot and dry climates since the humidity of such systems can rise up as high as 80% in moist condition. However, such humid air cannot be directly supplied into building spaces for thermal comfort because it can cause warping, rusting and mildew of susceptible materials (Lechner, 2009). A schematics of passive and active direct evaporative cooling processes are shown in figures 1.2(a) and (b).

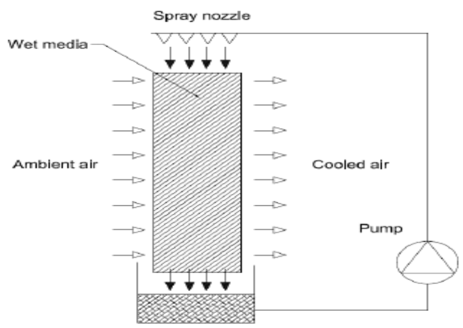


Fig. 2.1(a) Schematics of a Passive DEC (Alam et al., 2015)

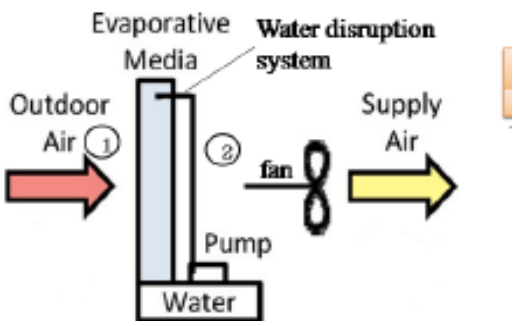


Fig. 2.1(b) Schematics of an Active DEC (Amer et al., 2015)

Similarly, indirect evaporative coolers (IEC) produce cooling by lowering the sensible heat and dry bulb temperature of air without adding moisture to the air. Indirect evaporative cooling systems are classified into two; the wet bulb temperature and the sub-wet bulb temperature IEC. The sub-wet bulb temperature IEC is also called the Dew point temperature IEC. Wet bulb temperature IEC are packaged flat plate stack, cross flow heat exchanger while sub wet bulb temperature IEC are multi perorated flat plate cross flow heat exchanger. They operate based on the Maisotsenko - cycle(m-cycle) and have two working fluids, the primary and the secondary working fluid. In a sub wet bulb temperature type, there are two channels that separates airstream e.g wet channel for rejecting spent air and dry channel for admitting supply air. The temperature of the supply air in the dry channel is reduced indirectly by releasing its sensible heat to the airstream in the wet channel through a thin non-permeable channel wall. To attain sub-wet bulb temperature, part of the cool air in the dry channel is diverted to accomplish the evaporation process in the wet channel. The schematics diagram of a typical IEC is shown in figure 1.3 while figures 1.4(a) and (b) shows the schematics of a wet bulb and sub wet bulb temperature IEC, respectively.

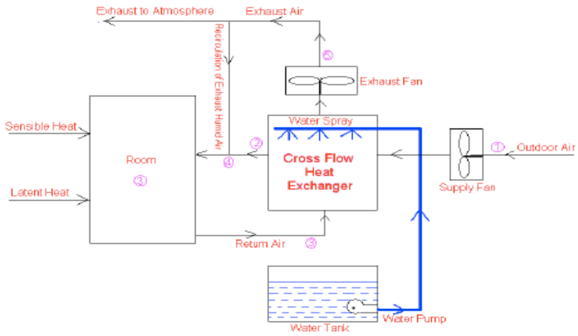


Fig. 2.2: Schematics of a simple IEC (Bisoniya et al., 2012)

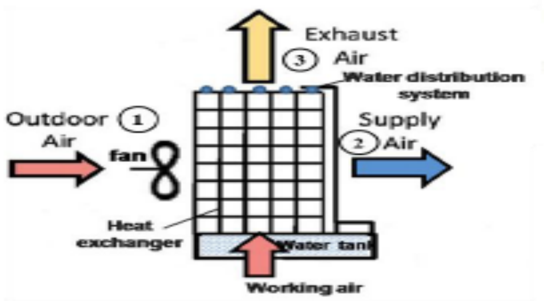


Fig. 2.3(a): Schematics of a Wet bulb Temperature IEC (Amer et al., 2015).

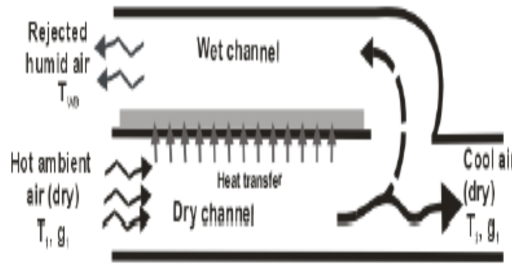


Fig.

Fig. 2.3(b): Schematics of a simple Sub Wet bulb Temperature IEC (Boukhanouf et al., 2014)

Likewise, a combined Indirect/Direct evaporative cooler (IDEC) combines DEC and IEC in a single unit. This is done since DEC systems have high effectiveness but with a drawback of high indoor humidity and IEC systems have low effectiveness with constant humidity, it is believed that a combination of both systems or in joint application with other cooling technologies can help actualize the best cooling characteristics of both systems in terms operating conditions (e.g. cooler supply air at a lower relative humidity, higher efficiency and controlled humidity). The main components of IDEC system are heat exchanger of IEC unit, evaporative pad of DEC unit, water recirculation system, water reservoir, and blowers, as shown in figure 1.5.

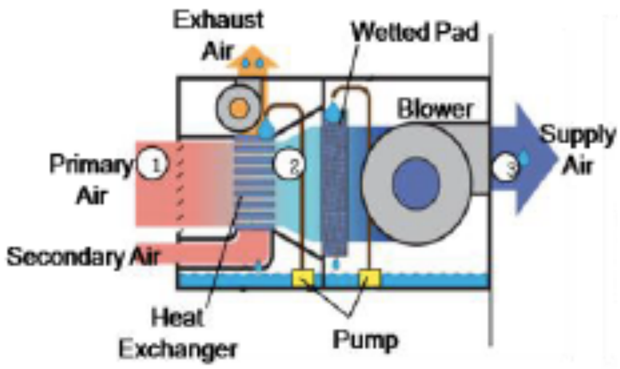


Fig. 2.4: Schematics of a Two Stage IDEC (Amer et al., 2015)

Several types of evaporative cooler designs have been reported (Anyanwu, 2004). The most common being the cabinet evaporative cooler constructed from metallic materials with charcoal placed adjacent to the sides, thereby inhibiting heat and mass flows between the storage space and the environment Taha et al., (1994). With development in the designs of evaporative cooler utilizing porous construction materials, water seepage through the walls becomes a critical factor for improved performance. this can best be studied using models that will describe the transient behaviour of the system. This work therefore presents one such model developed from first principle.

**2.2 Evolution of Evaporative Coolers**

In the Middle East, the first manufactured evaporative cooler became evident during the ancient times. The coolers had towers that trapped wind and blew it past water (beneath the system housing) into a building keeping the space cool at all times (Dehghani-Sanij, 2015). Also, Leonardo Da Vinci contributed immensely to the development of evaporative cooling. As an innovative fellow, he improved on the existing concept of evaporative cooling systems.

In the 1800s, when summer heat become intolerable, New England textile manufacturers began to use evaporative cooling systems to cool the air in their mills. Also, same technique was adopted in the 1920s and 1930s in western America to lower ambient temperature for thermal comfort.

Apart from using evaporative cooling of air for comfort, it can also be used for food preservation. Since early historical time, preservation of food items after production have been a global concern. People began preserving food in sterilized containers that are hermetically sealed (Nicholas, 1900). Olives and meats were preserved by salting, drying and smoking. Sugar and vinegar were used to preserve fruits and vegetables.

Thomas, 1803 a dairy farmer in Maryland designed an icebox and used it to transport butter to the new capital city. Amazingly, by the 1840's Americans had already began storing and preserving food items in iceboxes. This technique was further developed into refrigerated cold stores and happened to be the best alternative for preserving perishable food products but appeared quite expensive to buy and run.

Unfortunately, with all the aforementioned techniques, majority of the post harvest losses of perishable food products are still common in many countries worldwide. However, such areas unarguably require simple and affordable designs with low-cost features to replace the high cost refrigerated cold stores and the complexities of ice production. Evaporative cooling techniques therefore presents better alternatives since they do not require external electrical energy for power supply. This technology has attracted great research interest due to their ability to offer environmentally friendly operation with low energy utilization when compared to mechanical cooling systems

Oscar Palmer of Phoenix built the first evaporative cooler otherwise known as drip type evaporative cooler in Arizona (Cook, 1979). Thereafter, various types of evaporative coolers have been developed. The most common types are used for preserving perishable food products. They are basically a double chamber designs consisting of an inner chamber (i.e storage chamber) housed in a bigger chamber. The interspace or annular space between the chambers is usually filled with a material capable of absorbing and desorbing water. Water is introduced into the interspace to keep the absorbent material constantly moist through a water reserviour. Perishable food products to be preserved are loaded in the storage chamber. Figure 2.1 shows the schematic diagram of a typical passive evaporative cooling system. It operates on the principle of evaporation.

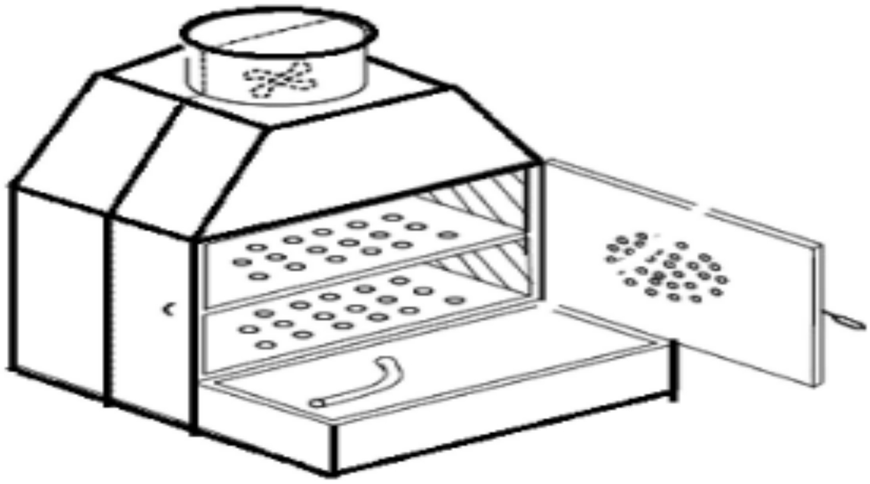

Fig. 2.5: Schematic diagram of an Evaporative Cooler [Mogaji & Fapetu, 2011).

There are simple designs of evaporative coolers that can be used in the home and these basic designs consist of a storage clay pot placed inside a bigger clay pot that holds water. Plate 2.3 and 2.4 shows a picture of a typical clay pot and a pot –in- pot evaporative cooler.

Plate 2.3: Traditional clay pots for water cooling (Nwaji & Anyanwu, 2022)

Plate 2.4: Pot –in- Pot Evaporative Cooler (Abba, 1985)

Some other simple designs of evaporative coolers that can be used in homes are;

### 2.2.1 Bamboo Cooler

The bamboo cooler has its base made from a large diameter tray filled with water. Inside the tray are bricks with an open weave cylinder of bamboo placed on top of them. The bamboo frame is wrapped with hessian cloth to ensure that the cloth is properly dipped into the water so that water can be drawn up the cylinder's wall. Food is therefore kept in the cylinder with a lid placed on top (Mangaraj, 2014)

### 2.2.2 Almirah Cooler

This is a more sophisticated type that has a wooden frame covered with cloth. This cloth dips into a water tray at the base and on top of the frame thus, keeping this cloth wet. It incorporates a hinged door and interval shelves which allow easy access to the stored produce (Practical Action, 2009).

### 2.2.3 Charcoal Cooler

This is made from an open timber frame of approximately 50mm x 25mm (2" x 1") in section. The door is made by simply hinging one side of the frame. The wooden frame is covered with mesh, inside and out, leaving a 25mm(1") cavity which is filled with pieces of charcoal sprayed with water so that when wet, it provides evaporative cooling. This frame work is mounted outside the house on a pole with a metal cone to deter rats and a good coating of grease to prevent ants from getting to the food (Sharma & Rathu, 1991) . Also, the top is usually solid and thatched with an overhang to deter flying insects. The schematics of the charcoal cooler is shown in figure 2.2.

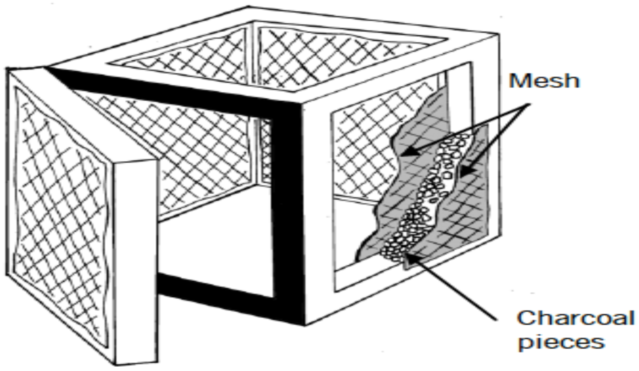


Fig. 2.6: Charcoal Cooler (Sharma & Rathu, 1991)

### 2.2.4 Naya Cellar Storage

Practical Action Nepal has been successful in transferring cooler technology, similar to the Indian Agricultural Research Institute design, especially to rural areas. This is called the Naya Cellar Storage and was originally designed by Dr GyanShrethra from the Green Energy Mission and Mr Joshi. This construction is made from locally available materials and the results have been encouraging for rural food processors who had little or no income and have been unable to acquire costly refrigerator. This is so since it is easy to adapt the design to the user's requirements (Mangaraj, 2014).

### 2.2.5 Static Cooling Chamber

The Indian Agricultural Research Institute has developed a cooling system that can be built in any part of the country using locally available materials (Roy, 1985). Its basic structure can be built from bricks and river sand, with a cover made from cane or other plant material and sacks or cloth. There must also be a nearby source of water, several bricks of about 400 are needed to build a chamber which has a capacity of about 100kg.

After construction, the walls, floor, sand in the cavity and cover are thoroughly saturated with water. Therefore, once the chamber is completely wet, a twice daily sprinkling of water is enough to maintain the moisture and temperature of the chamber. A covering for the chamber is made with canes covered in sackings, all mounted in a bamboo frame. The whole structure is protected from the sun by making a roof to provide shade (Odesola & Onyebuchi, 2009). Figure 2.3 shows a diagram of a static evaporative cooling system.

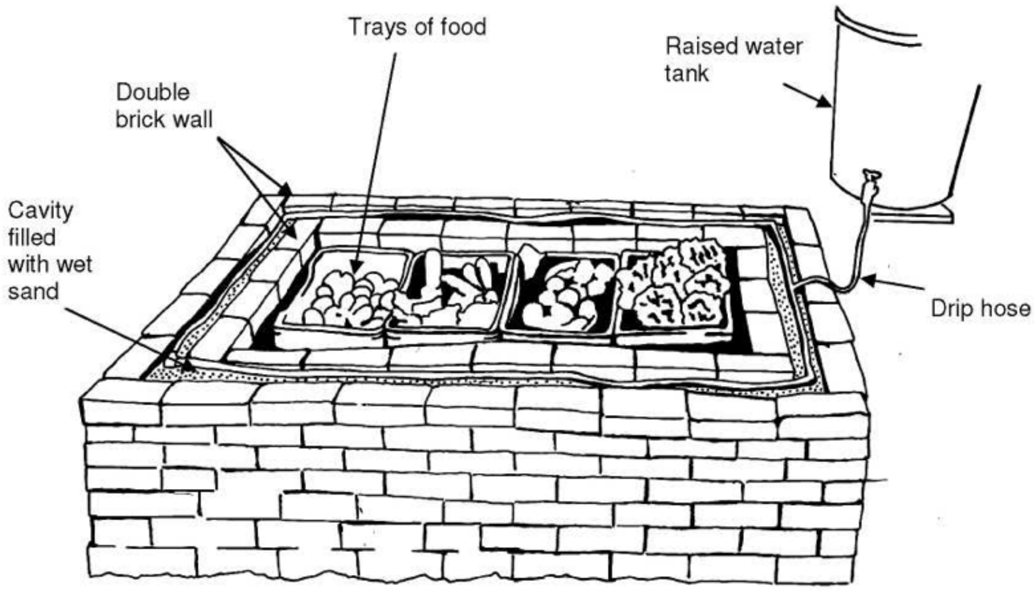


Fig. 2.7: Static Cooling System (Roy, 1985)

### 2.2.6 Evaporative Cooling Storage of Tomato in Cambodia and Laos

A simple brick-walled evaporative cooler with moistened sawdust (Cambodia design) or sand (Lao design) as wall insulation was tested for storage of mature-green and breaker fruits of two promising tomato AVRDC varieties, CLN1462A and TLCV15. During fruit storage, the temperature in the evaporative cooler decreases by about 1-10$^{o}$C lower than the ambient while relative humidity increased by about 10-35% higher than the ambient relative humidity. As a result, weight loss of stored fruits of both varieties decreased considerably. However, the evaporative cooler has no appreciable inhibitory effect on ripening, except in TLC15 fruits as obtained in the Cambodia experiment. Decay incidence showed variations due to treatment conditions. In the Cambodia experiment, decay in both varieties was higher in the evaporative cooler than at ambient. Pre-storage wash in 2% bicarbonate solution (2g ordinary baking soda in 100ml water) for 3 minutes reduced decay during evaporative cooling storage. In the Lao experiment, decay did not vary with storage condition. Evaporative cooling storage had no adverse effect on the physiochemical and sensory quality attributes of the fruits at the re-ripe stage but these attributes differed with variety. In general, CLN146A fruits had lower soluble solids and were rated lower in sensory quality than TLCV15 fruits (Vanndy et al., 2008).

### 2.2.7 Traditional Evaporative Coolers

The operation of traditional clay pots coolers is based on the principles of evaporative cooling and functions best in hot and dry ambient condition. This is the basis of the zeer pot, commonly used in West Africa for keeping fruits and vegetables fresh. In the zeer pot, a smaller clay pot fits inside a lager one. The gap between the two pots is filled with sand and then with water on a regular basis (usually once or twice a day). Fruits, vegetables and leftovers are placed in the smaller pot, which is covered with a lid or cloth. The water in between the pots slowly evaporates, keeping the food cool (Abbah, 1985)

Another example of traditional evaporative coolers is the coolgardie safe (McCormick, 1890), an evaporative cooler that was quite common in Australia at the beginning of the twentieth century. Both these systems work best in drier climate.

The relevance of evaporative cooling transcends its use for domestic cooling to some industrial application. It was in vogue for aircraft engines in the 1930's, for example with the Bearrmore Tornado airship engine. Here the system was used to reduce or eliminate completely the radiator which would otherwise create considerable drag. In these systems the water in the engine was kept under pressure with pumps allowing it heat to temperatures above 100$^{o}$C as the actual boiling point is a function of the pressure. The superheated steam was then sprayed through a nozzle into an open tube where it rapidly released its heat. The tubes could be placed under the body of the aircraft resulting in a zero drag cooling system.

However, these systems also had serious disadvantages. Since the amount of tubing needed to cool the water was large, the cooling system covered a significant portion of the plane even though it was hidden. This resulted to all sorts of added complexity and the systems were always terribly unreliable. In addition, this large size meant it was very easy for it to be hit by fire.

## 2.3 Recent discoveries and Development on Evaporative Cooling Technology

Evaporative cooling technology have in recent times received research interest. This is because of its low energy technique and off grid line electricity operation. Unfortunately, the system design is so simple that it is yet to receive wide application and market acceptance. Experimental and model based studies have been reported on improvement in the system design as detailed below.

### 2.3.1 Experimental Studies

Roy and Pal (1994) reported the performance of a low-cost cooling chamber developed using bricks. The interspace was filled with water saturated sand and the chamber covered with rush mat framed with bamboo as shown in plate 2.5

Plate 2.5: Pictorial View of the Brick Cooling Chamber [Roy & Pal, 1994)

The cooling chamber was tested during hot summer period in India and results indicated that the cooler can maintain storage temperature within the range of (15 – 18) ℃ and a relative humidity of 95%. Tables 2.1 and 2.2 shows the performance of the cooling chamber when loaded with fruits and vegetables.

**Table 2.1 Storage of Fruits in Cool Chamber (Roy & Pal, 1994)**

| **Crop** | **Cool Chamber** | | **Room Temperature** | |
|---|---|---|---|---|
| | **Shelf Life (days)** | **PLW (%)** | **Shelf Life (days)** | **PLW (%)** |
| Aonla | 18 | 1.72 | 9 | 8.70 |
| Banana | 20 | 2.50 | 14 | 4.80 |
| Grape Fruit | 70 | 10.20 | 27 | 4.94 |
| Guava | 15 | 4.00 | 10 | 13.63 |
| Kinnow | 60 | 5.30 | 14 | 16.10 |
| Lime | 25 | 6.00 | 11 | 25.00 |
| Mango | 9 | 5.04 | 6 | 14.99 |
| Sapota | 14 | 9.46 | 10 | 20.87 |

PLW = Physiological Loss in weight

**Table 2.2 Storage of Vegetables in Cool Chamber (Roy & Pal, 1994)**

| Crop | Cool Chamber | | Room Temperature | |
|---|---|---|---|---|
| | Shelf Life (Days) | PLW (%) | Shelf Life (days) | PLW (%) |
| Amaranth | 3 | 10.98 | <1 | 49.82 |
| Okra | 6 | 5.00 | 1 | 14.00 |
| Parwal | 5 | 3.89 | 2 | 32.86 |
| Carrot | 12 | 9.00 | 5 | 29.00 |
| Potato | 97 | 7.67 | 46 | 19.00 |
| Mint | 3 | 18.6 | 1 | 58.5 |
| Turnip | 10 | 3.4 | 5 | 16.00 |
| Peas | 10 | 9.2 | 5 | 29.8 |
| Cauliflower | 12 | 3.4 | 7 | 16.9 |

PLW = Physiological Loss in weight

A Nigerian teacher by name Mohammed Abbah developed a small scale pot-in-pot evaporative cooler otherwise referred to as zeer pot as shown in plate 2.4 and 2.6. The cooler chambers were molded with clay while the interspace between the pots was filled with damped sand. However, Sudan's Woman's Association for Earthenware Manufacturing and Practical Action began performing empirical studies on Mohammed Abbah's design with the aim of discovering the effectiveness and economical benefits of adopting the design for preserving perishable food products (Longmone, 2003). The results of their experiment are shown in Table 2.3.

Plate 2.6: Pot-in-pot Evaporative Cooler (Longmone, 2003)

**Table 2.3 Shelf Life of Perishable Products Using the pot-in-pot Cooler (Longmone, 2003).**

| Produce | Shelf-life of produce without using the zeer. | Shelf-life of produce using the zeer. |
|---|---|---|
| Tomatoes | 2days | 20days |
| Guavas | 2days | 20days |
| Rocket | 1day | 5days |
| Okra | 4days | 17days |
| Carrots | 4days | 20days |

Following their results, the Woman's Association for Earthenware Manufacturing began to manufacture and sell pots for the preservation of food.

Mordi and Olorunda (2003) built an evaporative cooling system for preserving fresh tomatoes. The cooler was able to reduce storage temperature by 8.2°C from ambient condition of 33°C. Similarly, relative humidity was increased by 36.6% from 60.4%. Also, rate of discoloration was slowed in samples stored in the cooling system relative to those exposed to ambient conditions. Overall, evaporative cooling extended the shelf life of tomatoes from 4days to 11days. However, when the samples were packaged in perforated polyethylene bags, the fresh tomatoes remained fresh for 18days when stored in the evaporative cooler and 13days when exposed to ambient conditions.

Anyanwu (2004) designed, fabricated and studied the evaporative cooling performance of a cuboid shaped porous clay container located inside another clay container. The interspace was filled with wetted coconut fibre. Results obtained during testing indicated a depression from ambient condition within the range of 0.1 to 12℃. The ambient conditions during the test periods was between 22 - 38℃, thus the cooler is a better technique of preserving vegetables for short period soon after harvest than open air preservation.

Babarinsa (2006) conducted an investigative study on the performance of an evaporative cooler made of brick walls in a semi-arid climate. The interspace was filled with water saturated riverbed soil. The study was carried out without loading the storage chamber. Experimental results showed that unloaded cooler conditions varied with local climatic conditions. Readings were taken during periods of cold and hot dry days. During cold dry days, cooler temperature reduced from 14°C - 26°C ambient temperature to 1.5°C - 10°C respectively with an average cooling efficiency of 69%. However, the cooler showed a better performance during hot dry days by lowering the cooler temperature from 26.5°C - 35°C ambient temperatures to 8°C - 12.5°C respectively with a cooling efficiency of 71%. Overall, the storage chamber's relative humidity varied consistently between 82% and 100%.

Thiagu et al. (2007) conducted a study to compare the build up of lycopene in tomatoes ripened by evaporative cooler and those ripened under room conditions. The evaporative cooler (EC) storage conditions were between 20°C - 25°C and 92% - 95% relative humidity whereas the room conditions were between 28 °C – 33 °C and 45 – 65% relative humidity. The experiment was carried out in Mysore during summer period. Results showed that tomatoes stored in the EC were 100% ripe as at day fifteen of storage whereas those exposed to room conditions were 83.3% ripe. Also, the lycopene content of tomatoes ripened in EC doubled relative to those exposed to room conditions. Overall, it was observed that tomatoes exposed to room conditions showed moisture loss of 6.5 times more than those stored in EC. Hence, evaporative cooling lowers the rate of rupture and shear stress in fruits and vegetables.

Rayaguru et al. (2010) investigated the suitability of a zero energy cool chamber for short term preservation of fruits and vegetables. This study was carried out in the coastal district of Orissa and was aimed at achieving a steady and conducive storage environment for storage of fruits and vegetables. Standard quantity of water was applied during the experiment and results indicated an optimum requirement of 75L/day and 90L/day in summer and winter months respectively. Also, the cool chamber achieved a depression in average ambient temperature within the range of (5 – 8) °C and maintained over 90% relative humidity. Overall, the cool chamber was effective in prolonging the shelf life of potato, tomatoes, mango, banana and spinach by 3 to 15 days as compared to the ambient conditions.

Kachhwaha and Prabhakar (2010) proposed a simple design methodology and performed a heat and mass transfer analysis for a direct evaporative cooler. The objective of the analysis was to predict the performance of the cooler and establish pad thickness and height for achieving maximum cooling using climatic condition parameters e.g: Dry bulb temperature (DBT), humidity ratio, water temperature and flow rate, air velocity and geometric properties of the evaporative cooler. Obtained results indicate that this approach could be useful for the design and analysis of direct evaporative coolers.

Ndukwu (2011) conducted an experimental study on an evaporative cooler he developed. Its cooling chamber was molded with clay. The interspace between the chambers measured 15cm and was filled with wetted wood shaving cooling pads. The top of the cooler was covered with perforated aluminum foil due to its ability to reflect solar radiation while the perforation was for easy exit of exhaust air. Matured freshly harvested tomatoes were used to carry out the test. Experimental and control tests were performed for the tomatoes in order to evaluate the cooling efficiency of the cooler. Results taken bihourly between 600hrs and 2200hrs local time indicated that the cooler was capable of preserving the tomatoes for 19days post harvest. Transient response showed that the ambient temperature was lowered by evaporative cooling process from (32 – 40) °C to (24 – 29) °C and relative humidity increased from (40.3 – 92) %. It was observed that the performance of the cooler was at its peak during the day between noon and 1600hrs local time.

Sunmonu and Alabadan (2012) reported the findings of an empirical study on the performance of a multipurpose passive evaporative cooler. The interspace was filled with loamy sand and both chambers made from galvanized steel. A water reserviour was connected to the annular space to ensure continuous flow of water. A water outlet was created beneath the system housing so that warm water will exit the system carrying the rejected heat from the air stream hence lowering the temperature and raising the relative humidity in the cold chamber. The performance evaluation was carried out with fresh bananas. Test results indicated a significant reduction in storage chamber temperature from 28.8$^{o}$C to 24.5$^{o}$C and relative humidity increased from 69.41% to 88.87%. Overall, the cooling efficiency of the system was 55%.

Ndukwu et al. (2013) reported the experimental results of an active evaporative cooling system for short-term storage of fruits and vegetables. The experiment was carried out in southern Nigeria and the cooler featured a hexagonal shape. The outer chamber was made with mild steel plate while the inner chamber was made with aluminum sheet. The interspace between the chambers was filled with palm fruit fibre. Three suction fans were connected to the cooler compartments for enhanced airflow rate. The cooling system was tested for transient response, cooling efficiency and capacity. Results indicated a significant drop in ambient temperature in the range of 4 -13 °C, which was close to the wet bulb depression of ambient air, an increase in relative humidity to 96.8%, an evaporative cooling efficiency of 98% with a maximum cooling capacity of 2.529kw respectively. The prototype of the cooler is shown in plate 2.7.

Plate 2.7: Prototype of the Active Evaporative cooling system (Ndukwu et al., 2013)

Sunmonu et al. (2017) developed a battery operated evaporative cooler for storing perishable fruits. The inner storage chamber was made of aluminum, the outer chamber made of galvanized steel while the interspace measured 7cm and was filled with river bed sand. The river bed sand was continuously wetted with room temperature water. An electric fan was connected to a 12 voltage battery to enhance the airflow rate. The system was tested with freshly harvested matured pears. Results showed that the average storage chamber temperature reduced by 20% from 31.14°C to 24.93°C while the average relative humidity increased from 61.14% to 89.5%.

Gutu (2020) developed two different cooling chambers made from brick and charcoal. He then conducted an experimental study to compare the effect of the storage chambers on the shelf life of freshly harvested tomatoes. Results indicated that brick made cooler lowered average daily temperature within the range of 4 -5.9°C while charcoal made cooler lowered the temperature within the range of (4 – 8.1) °C. Similarly, brick made cooler enhanced the relative humidity within the range 66 – 94% while charcoal made cooler enhanced the relative humidity from ambient conditions of 70% to 94% and 40% to 70% for the two coolers. Overall, both coolers prolonged the shelf life of tomatoes to twenty-three days. However, as the storage days increased, charcoal made cooler showed better performance with a smaller cumulative post harvest loss of tomatoes compared to the losses from the brick cooler.

Kale and Sundaram (2015) reported the results of a comparative study carried out on Natural and Forced updraft evaporative coolers. The natural draft cooler has an insulated enclosure with well defined height mounted on top of the cooler. The enclosure is to provide the updraft chimney effect which will enhance the effect of adiabatic evaporation of water. In the forced updraft cooler, the chimney is replaced by a suction fan. The conditions of samples stored in the coolers were compared with those exposed to open air conditions. Results indicate that the natural draft cooler performed better than the forced updraft cooler probably due to interruption of power supply. For the Forced updraft cooler, as air velocity and room temperature increased, the amount of water evaporated showed a sharp increase relative to the natural draft cooler. Between 150 - 190grams of water evaporated within 24hrs in the natural draft cooler whereas (240-310) grams of water evaporated within 7hrs in the forced updraft cooler. Also, the relative humidity of the storage chamber increased by 15 - 25 % in the natural draft cooler and by 10 - 12% in forced updraft cooler. Vegetables loaded into natural draft cooler showed better condition after 7 days than those in forced updraft cooler and in open air conditions. Overall, evaporative cooling technique was able to extend the shelf life of vegetables. Figure 2.4 and plate 2.8 shows the schematic diagram and picture of the coolers.

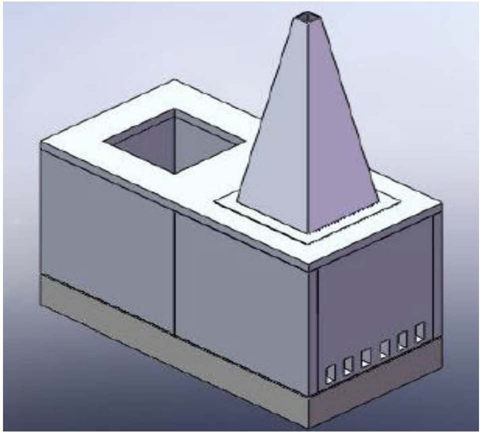

Fig. 2.8: Combined Natural and Forced Updraft Evaporative Cooler (Kale and Sundaram, 2015)


Plate 2.8: Forced Updraft Evaporative Cooler (Kale and Sundaram, 2015)

### 2.3.2 Model Based Studies

Mathematical models of energy conservation and heat and mass transfer analysis for humid air and water were developed for a direct evaporative cooling air conditioning system. These equations were then used to determine the effectiveness of the system. Calculated results obtained were compared with experimental results. Overall, both results were in good agreement with insignificant differences (Camargo et al., 2003).

Mogaji and Fapetu (2011) developed and evaluated the performance of a pyramidal shaped active evaporative cooler for preserving fruits and vegetables. The outer chamber was made with mild steel while the storage chamber was made with stainless steel and insulated on the inside with polystyrene foam. The interspace between the chambers was filled with jute cloth cooling pad. The system was tested for transient response and also the ability to increase the shelf life of the stored products. Results showed that the cooler prolonged the shelf life of the products by fourteen days relative to open air storage. Also, the the ambient temperature was lowered from 26°C -16 °C and 32 °C - 26 °C for different hours of the day while the ambient relative humidity was increased from 18% - 33% and 31% - 88%.

Ndukwu et al. (2013) developed a simple mathematical model for experimental validation of the performance of a small evaporative cooling system in a tropical climate. The model was used to determine the cooling efficiency of the evaporative cooler and considered the thermal conductivity and thickness of the cooling pad material. Results obtained showed cooling efficiency range between 92 – 99.2% respectively.

Livio and Kim (2016) reported the performance of an evaporation fridge with integrated solar chimney. A thermal network modelling and heat and mass transfer analysis was used to investigate the cooling process of the system. A solar chimney was integrated to enhance airflow thus increasing cooling capacity. MATLAB software was used to perform steady and transient simulations of the cooling system. Obtained simulation results indicate that the evaporation fridge performed better with the solar chimney. This is evident from the cooling potential of the evaporation fridge. With an enhanced airflow velocity of over 0.5m/s, the evaporation fridge maintained a 25% lower storage temperature compared to the ambient temperature of 30°C.

Adetunji et al. (2023) performed a thorough analysis to find out the influence of geographical and climatic factors on the performance of evaporative coolers for the storage of postharvest produce. Passive evaporative cooling systems used in various nations for the preservation of fresh farm goods were sampled and modelled for performance. Study results indicates the need to conduct analysis on the impact of air velocity, heating element plate texture and vapour loss on the temperature profiles of the cooler.

Halasz (1998) conducted a general non-dimensional mathematical model to describe all the different types of evaporative cooling devices in use today (e.g. water cooling towers, evaporative condensers and evaporative fluid coolers, air washers, dehumidifying coils etc.) The structure of differential equations defining non-adiabatic evaporation processes were converted to a pure non-dimensional form by introducing non-dimensional coordinates and parameters and substituting a straight air saturation line in place of the real one. As a result, describing the whole process becomes very simple and the overall performance of any such system can be defined in terms of a few parameters and presented in one or few diagrams. This non-dimensional model forms a basis for simple and rather accurate methods for rating various types of evaporative cooling systems. For any such system, a distinctive rating technique can be established, irrespective of the relative flow direction of the fluids involved, which is very convenient when cross-flow devices are considered.

Chelabchi (2017) developed an integrated modeling of interrelated heat and mass transfer processes in evaporative air coolers using indirect evaporative air coolers. Water was used as the evaporating agent and the basic airflow schemes (e.g. cross-flow and counter-flow) were considered. Partitioning of the problem was proposed and this was done in subtasks (e.g. air flow distribution among channels, heat and mass transfer in the channel system). The system configuration was decomposed into a number of elements. A system of non-linear one-dimensional ordinary differential equations were developed as the mathematical models of interrelated processes of heat and mass transfer. The developed efficient methods of joint solution of the mathematical model equations were used in simulation. An improved and vigorous technique for the simulation of media flows in the channels of a complex profile (two-dimensional fields of velocity and temperature) was suggested. A combined solution of continuity equations, Navier-Stokes equation (in projections to the coordinate axes), Fourier equation and pressure equation were performed. The method does not depend on the channel shape and the properties of the working media. The method of calculation of heat exchange coefficients with the use of the obtained velocity and temperature fields were established and this helps to refine results of modeling of the systems proceeding processes. Computational and full-size experiments for a number of designs and operation conditions of air coolers were conducted and comparative analysis of the obtained results indicated reliability of the proposed methods of mathematical modeling. The difference between the numerical results and available experimental values for the cooled air temperature was not more than 0.5$^{o}$C. The description of mathematical modeling helps in obtaining reliable information required to optimize the design and operating conditions of coolers. This developed method for the mathematical modeling of the heat and mass transfer processes can be applied in the study of diverse heat exchangers.

Liberati et al. (2017) conducted a performance analysis using a phenomenological model developed for the components of an indirect evaporative cooler with cross flow heat exchanger. The model was validated using standard summer operating conditions, at different air streams temperature, humidity ratio and flow rates. Results indicates the benefit of using the Indirect Evaporative Cooling unit, which leads to significant energy savings in almost all the investigated conditions.

Gao et al. (2015) developed a heat and mass transfer mathematical model, for the evaluation of the performances of a hybrid system in terms of coefficient of performance (COP). The models results were validated by comparing calculated results with the experimental results. Developed models were used to investigate the effects of the operating parameters, and the number of transfer units (NTU) on the performance of both the heat exchanger and indirect evaporative cooler. Obtained results indicates that the optimum process inlet temperature and humidity should not be up to 35$^{o}$C and 18 g/kg, respectively. Increase in Number of Transfer Units Heat Exchanger (NTUHE) or Number of Transfer Units Indirect Evaporative Cooler (NTUIEC) is within a certain range. While NTUHE is less than or equal to three (3), NTUIEC is less than or equal to six (6). Therefore, both can improve the COP of the system; The working air mass ratio in indirect evaporative cooler can affect the amount of air supplied to the room and the COP of the system, but can rarely affect the basic air conditions.

Liu et al. (2013) developed a simplified model using a modified effectiveness–Number of Transfer Units (NTU) technique for thermal performance simulation of indirect evaporative heat exchangers. Numerical results were then validated using experimental data from literatures. The developed differential equations were solved using computerized simulation packages and results on the impact of Lewis factor on evaporation processes using this model indicates that the assumption of unity Lewis factor may not be valid when the inlet air to the wet side is cold and dry.

Lekwuwa et al. (2012) developed a mathematical model of an evaporative cooling pad using sintered Nigerian clay. Applying the laws of conservation of continuum mechanics, a physical model of the evaporative cooling phenomenon was developed and then the governing equations which describes the energy and mass transfer for the clay model were derived. Some simplifying assumptions were applied to the physical model while constitutive relationships were also developed for further analysis of the developed equations. The developed energy transfer governing equations were solved using a finite element approach to determine the transient temperature response of the exposed boundary surface. Results obtained indicate that surface temperature differences could be as much as $6^{o}$C in the first cycle of evaporative cooling with the possibility to reduce further.

Fahmy et al. (2012) provided a complete mathematical modeling and simulation of evaporative cooling system used for greenhouse application. It was desired to maintain the greenhouse condition within $20^{o}$C indoor temperature and 70% relative humidity to support the growth of plants. They noted that the condition could be achieved by adjusting the air volume flow rate in pad-fan (evaporative) cooling systems to ensure continuous operation of the greenhouse. The model developed was based on MATLAB SIMULINK software which was used to predict the temperature and relative humidity profiles inside the greenhouse. Results indicates that the desired greenhouse condition was realized. Regenerative indirect evaporative cooling (RIEC) systems are a potential alternative to conventional air-cooling systems. In a study by Comino et al., (2021), which main objective was to experimentally determine the performance of a

RIEC air-cooling system under different inlet air conditions, a mathematical RIEC model based on a modified ε-NTU numerical method was developed and validated. The experimental RIEC results showed a high cooling capacity, with dew point effectiveness values up to 0.91. Obtained results indicates that the mathematical model can be properly used to study the global behavior of a RIEC.

Heidarinejad et al. (2010) presented the results of the performance analysis of a ground-assisted hybrid evaporative cooling system in Tehran. A Ground Coupled Circuit (GCC) was used to provide the desired pre-cooling effects, assisting a Direct Evaporative Cooler (DEC) to cool the air below its wet-bulb temperature. The GCC includes four vertical ground heat exchangers (GHE) arranged in series configuration. To effectively and accurately predict the optimum performance of a GCC, a computational fluid dynamic simulation was performed. Simulation results indicates that the integration of GCC with a DEC system provided significant comfort cooling whereas only DEC did not. Considering the simulation results the cooling effectiveness of a hybrid system is more than 100%. This innovative hybrid system was able to reduce the ambient air temperature below its ambient wet-bulb temperature. Therefore, this environmentally clean and energy efficient system can surely be considered as a potential alternative to the mechanical vapor compression systems.

### 2.4 Psychrometric Analysis of Evaporative Cooling

Psychrometry is the study of the thermodynamic properties of moist air. The chart describes the interactions between moisture and air. The basic knowledge and understanding of psychrometry helps Heating Ventilation, Air Conditioning and Refrigeration (HVAC-R) engineers in changing air from one condition to another. The Psychrometric charts makes it easy for state points to be fixed or established during any HVAC-R design process.

In evaporative cooling, the dry bulb temperature, humidity, specific volume and vapor pressure of air changes along the wet bulb temperature constant lines as shown by line AB in figure 2.5. Air enters the evaporative cooler at state A and is passed through a cooling pad wetted with water. As a result, a part of the water evaporates after absorbing heat energy from the air equivalent to the latent heat necessary to evaporate the water. The amount of water evaporated is dependent on the amount of heat absorbed. This process results to a reduction in the dry bulb temperature of the airstream and an increase in humidity. The cool and humid air leaves the cooler and is supplied to the desired space at state B while state C is the state of saturation when the airstream can no longer absorb more water. This state should be avoided because the cooler stops to create cooling when the air is saturated with moisture and the space temperature begins to build up.

Point C is the temperature achievable when the air is completely saturated. Since the cooling pad allows some quantity of air to bypass the pad without being cooled, this means that a fraction of the entry air passes through the cooling pad without being cooled. This fraction of air is referred to as the By-pass factor (BPF) mathematically given by equation (2.1).

$$BPF = \frac{T_B - T_C}{T_A - T_C} \tag{2.1}$$

On the psychrometric chart for evaporative cooling, since the wet bulb temperature lines almost coincides with the constant enthalpy lines, it can be inferred that the enthalpy lines for evaporative cooling is constant (i.e. adiabatic process) as shown by line AB in figure 2.5. The bypassed air will again mix with the airstream in contact with the cooling pad as shown in figure 2.6 and this process is known as Adiabatic mixing since the heat transfer with the environment is considered negligible. Also, there is no work interactions and kinetic and potential energies are also considered negligible.

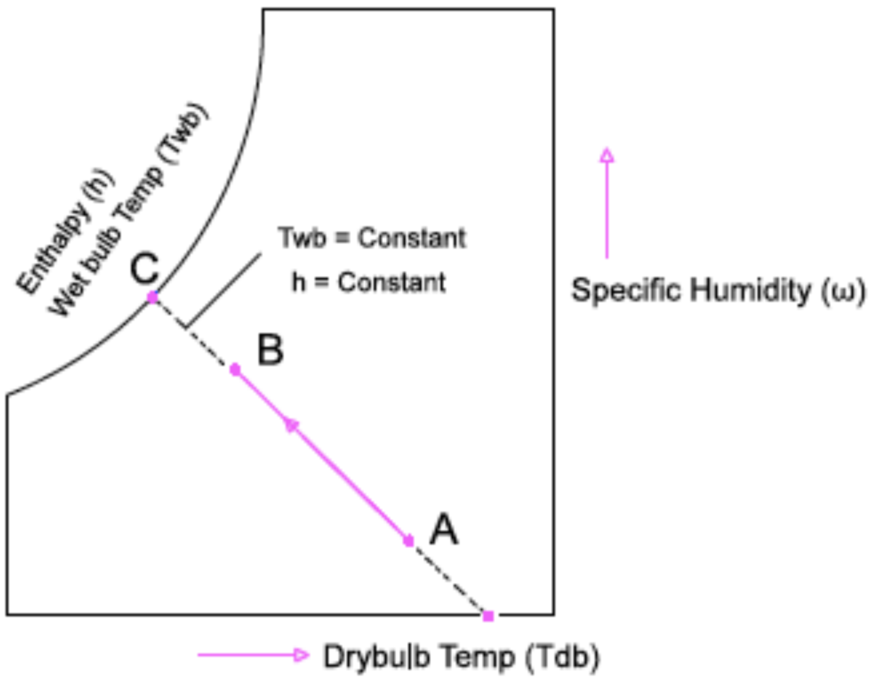


Fig. 2.9: Psychrometric Representation of Evaporative Cooling

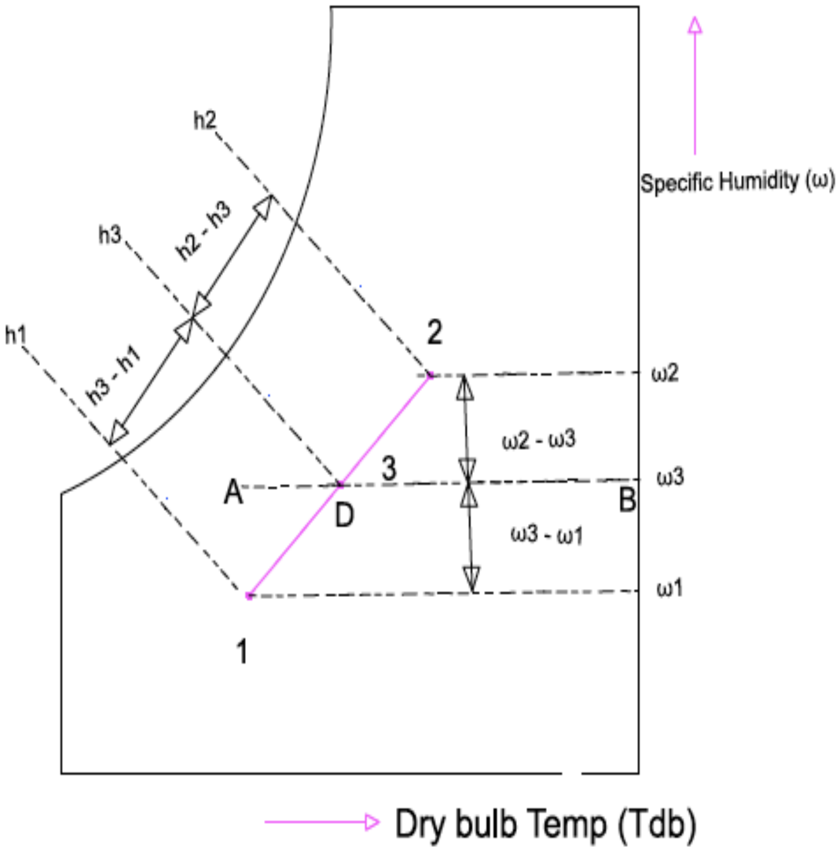


Fig. 2.10: Psychrometric Representation of Adiabatic Mixing Process

**2.5 Summary of Literature Review**

Elaborate work has been done on the performance analysis of indirect evaporative cooling systems for comfort space cooling with major interest on the flow configuration of the heat exchanger, thickness of cooling pads and varying the coefficient of Performance (COP). Some modeling works considered the integration of a hybrid system and checked for increased performance. Evaporative coolers applied in greenhouses has also been reported. Some evaporative coolers were modeled using climatic data different from what is obtainable in the tropical areas like Nigeria. Most of the models featured both cooling chambers made of galvanized steel, mild steel outer chamber and aluminum sheet inner chamber, and two different chambers made from bricks and charcoal. Evaporative coolers with solar chimney were also reported. Studies which considered cooling efficiency by considering thermal conductivity and thickness of cooling pads have also been reported while performance analysis was also conducted for battery operated coolers.  These areas were of interest because indirect evaporative cooling systems with their constant humidity have shown characteristics that suits most cooling requirements (e.g. for comfort cooling) when compared to direct evaporative cooling systems. A few works have modeled the performance analysis of passive direct evaporative coolers for preservation of fruits and vegetable. However, their models mostly established the steady state performance of the systems. Interestingly, most of the reported work on evaporative cooling for agricultural produce preservation are experimental works with results suggestive of the systems needing further performance enhancement although the present performance reported can considerably extend the shelf life of the products stored in them.

**2.6 Research Gap**

The literature review has shown that most published research works on evaporative coolers were mainly experimental. The models were mostly for evaporative coolers for comfort cooling, Very few models were reported for evaporative coolers for fruits and vegetables preservation and such models were mostly for steady state performance prediction of the coolers. For performance optimization studies that will lead to overall improved evaporative cooler performance, transient models are required. This project addressed this established gap by developing a transient model of a passive evaporative cooler for fruits and vegetables preservation.

# CHAPTER THREE

# RESEARCH METHODOLOGY

## 3.1 Description of the Model

The evaporative cooler modelled in this study is of a square cabinet Box-in-box type experimentally studied by Anyanwu (2005). The cabinets were made from locally sourced clay which comprised feldspar clay, fire clay and Aroclay in the ratio 1: 1: 1. The schematic illustration of the cooler is shown in Fig. 3.1.

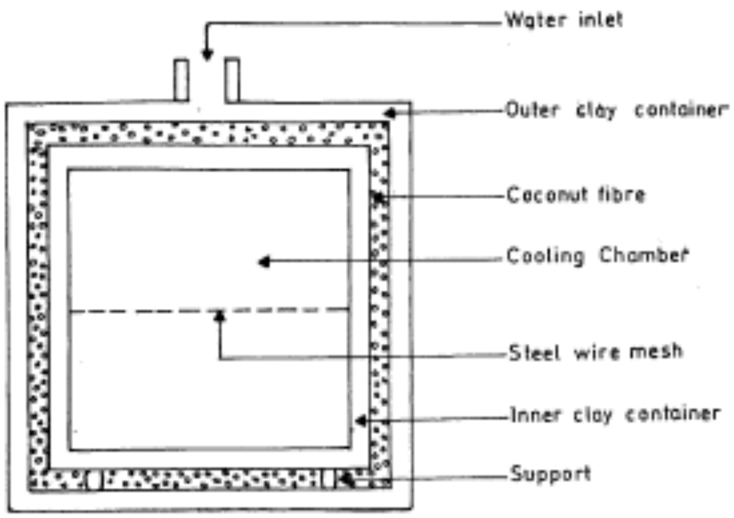


Fig. 3.1: Schematic illustration of the Evaporative Cooler components (Anyanwu, 2005)

The porous cabinets were moulded in 20mm thickness, reinforced with 3mm thick steel wire mesh for rigidity and subsequently used to form square boxes. The inner box which is the storage chamber is housed by the outer box. A uniform gap of 50mm is provided between the top, vertical and bottom side walls of the cooler. The interspace between the inner and outer boxes are filled with locally sourced coconut fibre to improve the water retention capacity of the clay walls. The overall dimension of the inner box is (240 X 240 X 240) mm. Similarly, the overall dimension of the outer box is (380 X 380 X 380) mm. A side door was integrated on the box to gain access to the storage space. A water reservoir is connected to a flexible pipe introduced at the top of the cooler. This pipe continuously supplies the required amount of water for the cooling process. The soaked water then seeps through the porous wall to the outer surface where it evaporates on coming in contact with ambient air due to a difference in humidity. The water vapour produced is at a lower temperature and is absorbed by the surrounding air resulting in cooler and more humid air. The cooling effect produced by the moist air keeps the storage space consistently cold and moist throughout the operation of the cooler.

The main reasons evaporative coolers have attracted research attention over the years is because of its efficiency, cost effectiveness and use of simple passive cooling techniques to achieve low temperature for preservation of perishable fruits, vegetables, food, etc. It can be used both in rural and urban areas but most suitable in rural areas since it requires no special skills, no grid electricity and external energy source for its operation.

### 3.2 Principle of Operation

Evaporative cooling is an adiabatic cooling process that uses the principles of evaporation to reduce the temperature of air. Cooling systems of such type are referred to as evaporative coolers. They can be used for both air conditioning and refrigeration. Evaporative coolers use water as their cooling agent to achieve temperature reduction of dry air.

The operation of the cooler begins with the continuous introduction of cooling water into the cooler to consistently wet the coconut fibre absorbent. As water is introduced into the cooler, the water first soaks the absorbent and then seeps through the outer porous medium to the adjacent side walls of the cooler where it then evaporates. Water evaporation is induced by the difference in the humidity of air. The evaporated water is absorbed in vapour form by the surrounding air, removing the heat from the air, and reducing air temperature; meaning the more moisture in the air, the greater the cooling effect. The cooling effect produced is naturally circulated within the cooler for refrigeration. It is important to note that the cooling effect produced is not dependent on the temperature of circulating water but rather depends on the following;

i. The surface area available for air to travel and evaporate water.
ii. The quantity of water available for evaporation
iii. The relative humidity of ambient air: The lower the relative humidity, the more moisture air can absorb/hold, the greater the cooling effect.
iv. The airflow velocity of surrounding air: The higher the quantity of air drawn towards the cooler, the greater the air carrying capacity and the greater is the cooling effect produced.

Evaporative coolers do not require recirculation of water therefore adequate planning should be put in place for continuous supply of water. Evaporative coolers are most effective in hot dry regions. When positioned at the appropriate location with adequate air movement, the cooler can yield a cost-efficient method of cooling.

The amount of evaporative cooling achievable is measured through the wet bulb depression of air (i.e., the difference between the dry and wet bulb temperatures of air). The minimum temperature attainable on the outer surface of the porous medium through passive evaporation as in the case of the modelled cooler, is the wet bulb temperature of air. The lower the wet bulb depression, the higher the evaporative cooling achieved.

### 3.3 FORMULATION OF MODEL EQUATIONS

The fundamental governing equations of heat and mass transfer for evaporative cooling are: Fourier's equation, Fick's law of diffusion and convective heat and mass transfer equations. Each of these equations would therefore be applied at the appropriate points where they reasonably represent the process to properly analyse the system.

### 3.3.1 Simplifying Assumptions

In order to model the considered evaporative cooler, the following assumptions were made:

i. No internal heat generation

ii. The porous wall is homogenous and isotropic, therefore, energy transfers from each of the exposed surface is considered uniform and in a direction perpendicular to the surface. Hence, transient one-dimensional heat transfer process is assumed across the porous walls.

iii. Humidity inside the cooling chamber is high thus the rate of water evaporation from the inner wall surface to the cooling chamber is very low, hence evaporative cooling is considered negligible within the cooling chamber.

iv. The quantity of water absorbed by the coconut fibre stocked in the interspace is very small hence convective heat transfer is neglected within the interspace.

v. Water flows continuously at constant velocity from the reservoir into the interspace housing the coconut fibre. Therefore, coconut fibre absorbent is uniformly wetted.

vi. Water flows in a streamlined manner due to the effect of gravity, hence the flow is laminar.

vii. Air flow is not enhanced. There are no eddies, swirls or current normal to the flow direction. Therefore, air flow is considered to be laminar.

viii. The coconut fibre stocked into the interspace is considered to be in perfect thermal contact with the walls. Therefore, Energy change within the interspace is as a result of convection, conduction and due to the differences in temperature of water within the inner porous media wall surfaces and the temperature of water replaced from the reserviour

ix. The porous m is a solid component and allows interaction between particles of the system; therefore, mass transfer through the porous media is by conduction.

### 3.3.2 ENERGY AND MASS BALANCE EQUATIONS

A typical control volume for the energy transfer through the porous walls is as represented in Figure 3.2.

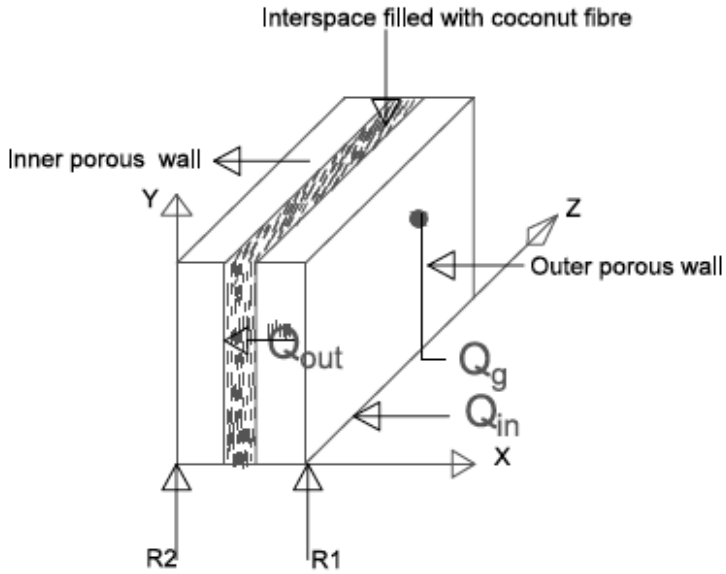


Fig 3.2: System Control Volume

Considering assumption (i), Figures 3.3 shows the 1-D heat flow directions on a section of the porous wall.

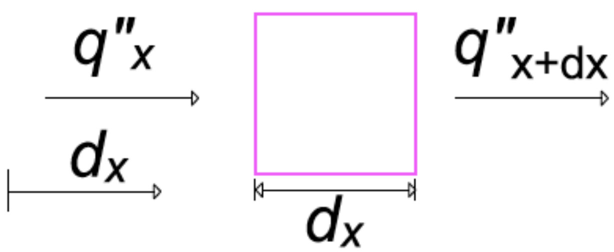


Fig. 3.3: 1-D Heat Conduction in Evaporative Cooler

The general energy balance equation given as Equation (3.1) and it applies to all the walls of the porous media.

$Heat\ inflow\ + Internal\ energy\ generation = Heat\ outflow\ +$

$change\ in\ internal\ energy$ (3.1)

Mathematically; equation (3.1) is presented as;

$$Q_{in} + Q_g = Q_{out} + \Delta U \tag{3.2}$$

where $Q_{in}$ is the heat inflow into the cooler, $Q_g$ is the Internal heat generation, $Q_{out}$ is the heat outflow from the cooler $and$ $\Delta U$ is the Change in Internal energy

### 3.3.3 Heat Flow Through the Outer Porous Wall of the Cooler

The porous wall is a material containing pores or spaces in or around it to permit movement of fluid through or around the pores.

Considering assumption (viii), equation (3.2) is reduced to equation (3.3).

$$Q_{in} = Q_{out} + \Delta U \tag{3.3}$$

$where$ $Q_{in}$ is the energy inflow in the x-direction which is as a result of conduction and is proposed by the Fourier as equation (3.4) (Cengel, 2007)

$$Q_{in} = Q_x = -KA\frac{\partial T}{\partial x} \tag{3.4}$$

where $K$, $A$ and $T$ represents thermal conductivity of the porous wall, area of the porous wall and temperature of the porous wall, respectively. In the case of the outer porous wall which is a composite wall moulded from clay, the thermal conductivity will be the effective thermal conductivity of the composite wall as presented in equation (3.5)

$$Q_{in} = Q_x = -K_{opm}A_o\frac{\partial T_{opm}}{\partial x} \tag{3.5}$$

$$K_{opm} \equiv K_{eff}$$

where $K_{opm}$, $K_{eff}$, $A_o$ and $T_{opm}$ represents the thermal conductivity of the outer porous wall, the effective thermal conductivity of the porous wall, area of the outer porous wall and temperature of the outer porous wall.

By considering the solid and pore phases of the porous wall as thermal resistors in parallel, for a two phase mixture problem, Collishaw & Evans (1994) proposed equation (3.6) for establishing the effective thermal conductivity of the composite wall.

$$K_{eff} = \left[\frac{K_2+2K_1+2\varepsilon(K_2-K_1)}{K_2+2K_1-\varepsilon(K_2-K_1)}\right] \quad (3.6)$$

where $K_1$ is the thermal conductivity of the clay =(1.28W/mK), $K_2$ is the thermal conductivity of fluid(water) =( 0.630W/mK), $\varepsilon$ is the porosity of clay = (0.55) (Bejan and Kraus, 2003).

Similarly, $Q_{out}$ is the energy outflow in the x-direction as shown in figure 3.2. Since the outer porous wall surface is in contact with the ambient air, the heat transfer from the outer surface is as a result of conduction, convection and evaporation. The energy required for water evaporation is provided in the form of latent heat of vaporization of water. Therefore, the energy outflow from the outer porous is given by equation (3.7).

$$Q_{out} = q_{x+dx} + Q_{fo} \quad (3.7)$$

Where $Q_{fo}$ is the driving force of the system and it is equal to the sum of the sensible heat ($Q_s$) and latent heat (($Q_{l)}$ contributions as presented by equation (3.8). The sensible heat portion is due to temperature difference between ambient and the surface of the porous wall and also due to the wind effect as presented by equation (3.9). Also, the latent heat is due to differences in humidity between ambient and the surface of the porous wall as presented by equation (3.10).

$$Q_{fo} = Q_s + Q_l \quad (3.8)$$

$$Q_s = hA\Delta T \quad (3.9)$$

$h$ is the convective heat transfer coefficient, $A$ is the area of the porous wall and $\Delta T$ is the difference in temperature between ambient air and the outer wall surface temperature.

$$Q_l = \dot{m}_{ev} h_{fg} \quad (3.10)$$

$\dot{m}_{ev}$ is the mass of water evaporated at the surface of the exposed wall, $h_{fg}$ is the latent heat of vaporization of water.

#### 3.3.3.1 Sensible Heat Outflow Contributed by Conduction

The sensible heat outflow contributed by conduction according to equation (3.7) is given by equation (3.11).

$$Q_{cond} = q_{x+dx} \tag{3.11}$$

$$q_{x+dx} = q_x - \frac{\partial}{\partial x}(q_x)dx \tag{3.12}$$

$$\frac{\partial}{\partial x}(q_x)dx = \frac{\partial}{\partial x}\left(-K_{eff}A_o\frac{\partial T_{opm}}{\partial x}\right) = -K_{eff}A_o\frac{\partial^2 T_{opm}}{\partial x^2} \tag{3.13}$$

Substituting for equations (3.5), (3.6) and (3.13) into equation (3.12), we have;

$$q_{x+dx} = q_x - \frac{\partial}{\partial x}(q)dx = -\left[\frac{K_2+2K_1+2\varepsilon(K_2-K_1)}{K_2+2K_1-\varepsilon(K_2-K_1)}\right]A_o\frac{\partial T_{opm}}{\partial x} - \left[\frac{K_2+2K_1+2\varepsilon(K_2-K_1)}{K_2+2K_1-\varepsilon(K_2-K_1)}\right]A_o\frac{\partial^2 T_{opm}}{\partial x^2}$$

$$q_{out} = Q_{x+dx} = -\left[\frac{K_2+2K_1+2\varepsilon(K_2-K_1)}{K_2+2K_1-\varepsilon(K_2-K_1)}\right]A_o\left[\frac{\partial^2 T_{opm}}{\partial x^2} + \left(\frac{\partial T_{opm}}{\partial x}\right)\right] \tag{3.14}$$

#### 3.3.3.2 Sensible Heat Outflow Contributed by Convection

The convective heat transfers ($Q_{conv}$) is given by Cengel as shown in equation (3.15)

$$Q_{conv} = h_\infty A_o\left(T_{amb} - T_{opm}\right) \tag{3.15}$$

Where $h_\infty$ is the the convective heat transfer coefficient at the wall- ambient interface, $T_{amb}$ is the ambient temperature of the surrounding air and $T_{opm}$ is the outer porous wall surface temperature. But the ambient temperature is not a constant value hence equation (3.16) is applied to account for the variability in ambient temperature with time (Mirzabozorg, 2014).

$$T_{amb}(t) = \bar{A}\sin\left[\frac{2\pi(t-\xi}{p}\right] + \bar{B} \tag{3.16}$$

Where $\bar{A}$ is the amplitude, $\bar{B}$ is the annual mean temperature, t is time in day, $\xi$ is the time in day in which $T_{amb}(\xi) = \bar{B}$, p is the period of time function which is equal to 365days. In this function $\bar{A}$ can be determined using equation (3.17) (Mirzabozorg, 2014).

$$\bar{A} = \frac{1}{2}\left[|T_{max} - T_{mn}| + |T_{min} - T_{mn}|\right] \tag{3.17}$$

Where $T_{max}$, $T_{min}$ and $T_{mn}$ are presented by equations 3.18), (3.19) and (3.20) respectively.

$$T_{max} = \frac{\sum_{i=1}^{n}(T_{max})_i}{n} \tag{3.18}$$

$$T_{min} = \frac{\sum_{i=1}^{n}(T_{min})_i}{n} \quad (3.19)$$

$$T_{mn} = \frac{\sum_{i=1}^{p} T_i}{P} \quad (3.20)$$

n is the number of day of the month.

The natural heat transfer coefficient, $h_\infty$, can be calculated using equation (3.21) (Rajput & Chand, 2007).

$$h_\infty = \frac{K_a}{L} * Nu \quad (3.21)$$

$h_\infty$ is the convective heat transfer coefficient of ambient air, $K_a$ is the thermal conductivity of air, $L$ is the characteristic length which is the height in this case. $Nu$ is the Nusselt number which is dependent on the Grashoff ($Gr$) and Prandtl ($Pr$) numbers and can be obtained from the empirical equations provided by Churchill and Chu (1975) for two different cases.

**Case I**

For $r.\mathrm{Pr} < 10^9$ , $Nu = 0.68 + \frac{0.67(Gr.Pr)^{1/4}}{\left[1+(0.492/Pr)^{9/16}\right]^{4/9}}$ (3.22)

**Case II**

For $r.\mathrm{Pr} > 10^9$ , $Nu = \left\{0.825 + \frac{0.387(Gr.Pr)^{1/6}}{\left[1+(0.492/Pr)^{9/16}\right]^{8/27}}\right\}^2$ (3.23)

For this study, the product of $Gr \; and \; Pr$ is calculated to be less than $10^9$ i.e. $Gr.\mathrm{Pr} < 10^9$, hence case one as shown in equation (3.22) applies. The correlations for the Grashoff number ($Gr$) and Prandtl (Pr ) number are given by equations (3.24) and (3.25).

$$Gr = \frac{g\beta\Delta T L^3}{\nu^2} \quad (3.24)$$

$$Pr = \frac{\mu C_{pa}}{K_{air}} \quad (3.25)$$

Where g is the acceleration due to gravity in $m \;/\; s^2$, $L$ is the characteristic length which is the height in meters of the outer porous wall in this case, $\Delta T$ is the temperature difference between the outer porous wall surface and the ambient temperature and $\beta, \nu, \mu$ and $C_{pa}$ are properties of air and represents $\beta = \frac{1}{T_{amb}}$ is the coefficient of volume expansion of air in ($^1/_K$), $\nu$ is the kinematic viscosity of air in $^{m^2}/_s$, $\mu$ is the Dynamic viscosity of air in $^{kg}/_{m.s}$ and $C_{pa}$ is the specific heat capacity of air in $KJ/kgK$ respectively. These properties of air are determined at the average annual temperature of the city which is Owerri in this case with an average annual temperature of 25.9°C (Climate-data.org). Therefore, Table 3.1 presents the properties of air in Owerri.

**Table 3.1 Properties of air at 1atm and ambient temperature in Owerri (Engineers Edge, n.d.)**

| Parameter | Value | Unit |
|---|---|---|
| Kinematic Viscosity ($\nu$) | $1.57028 \times 10^{-5}$ | $^{m^2}/_s$ |
| Dynamic Viscosity ($\mu$) | $1.85314 \times 10^{-5}$ | $^{kg}/_{m.s}$ |
| Specific Heat Capacity($Cp_a$) | 1.007 | $^{kJ}/_{kgK}$ |
| Thermal Conductivity ($k$) | 25.5766 | $^{W}/_{m.K}$ |

Source: Data generated from the EES software developed by S. A. Klein and F. L. Alvarado. Original sources: Keenan, Chao, Keyes, Gas Tables, Wiley, 1984; and Thermophysical Properties of Matter, Vol. 3: Thermal Conductivity, Y. S. Touloukian, P. E. Liley, S. C. Saxena, Vol. 11: Viscosity, Y. S. Touloukian, S. C. Saxena, and P. Hestermans, IFI/Plenun, NY, 1970, ISBN 0-306067020-8

Therefore, substituting for equations (3.21) and (3.22) into equation (3.15), the convective sensible heat transfer contribution is obtained as presented in equation (3.26).

$$Q_s = Q_{conv} = \frac{K_a}{L}\left[0.68 + \frac{0.67(Gr.Pr)^{1/4}}{\left[1+(0.492/Pr)^{9/16}\right]^{4/9}}\right]A_o\left(T_{amb} - T_{opm}\right) \quad (3.26)$$

Similarly, the ambient wind velocities vary from time to time and this variability is accounted for by applying equation (3.27).

$$W_{(t)} = -2 \times 10^{-(0.6t^6 + 0.0001t^5 - 0.0027t^4 + 0.0298t^3 - 0.1062t^2 - 0.11t + 2.167)} \quad (3.27)$$

Where W is wind velocity and t is time.

#### 3.3.3.3 Latent Heat Contributed by Evaporation

The energy required to initiate the evaporation process on the exposed surfaces of the evaporative cooler is provided in the form of latent heat of vaporization of water denoted by ($Q_L$ ). This happens when water seeps from the interspace through the inner surface of the outer porous wall to the adjacent exposed surfaces of the clay pot where evaporation occurs.

Therefore, Latent Heat of vaporization of water ($Q_L$ ) is given by;

$$Q_l = \dot{m_{ev}} h_{fg} \tag{3.28}$$

$h_{fg}$ is the specific enthalpy of vaporization of water, $\dot{m}_v$ is the mass of water evaporated at the exposed surfaces of the clay pot.

The mass of water diffused through the porous wall to the adjacent side exposed walls before the evaporation process can be determined using Fick's law of diffusion.

The mass of water diffused is calculated in differential form by applying Fick's law of mass diffusion equation to obtain the Stefan's law for diffusion equation as presented in equation (3.29), (Rajput, 2007).

$$\dot{m}_{wd} = -\frac{D_w A_O M_w}{G.T} * \frac{dP_v}{dx}\left(\frac{\text{Pv}}{P-P_v}\right) \tag{3.29}$$

Where $\dot{m}_{wd}$ is the mass flow rate of water diffused, $D_w$ is the diffusion coefficient of water, $A_o$ is the area of the outer porous wall, $M_w$ is the molecular mass of water, Pv is the total pressure of water vapour and surrounding air, $G$ is the universal gas constant, T is the temperature of the film filled exterior wall surface and $P_v$ is the partial pressure of water vapour at the film temperature. The mass of water evaporated is dependent on the quantity diffused through the porous walls to the outer surfaces. Thus, two scenarios exist;

a. $\dot{m}_{wd} > \dot{m}_{ev}$

For this situation, the rate at which water is diffused to the surface is greater than the rate of evaporation. Therefore, the rate of water evaporation can be computed from;

$$\dot{m}_{ev} = h_m A_o(\Delta C_w) \tag{3.30}$$

By applying the convective mass transfer equation.

Where $\Delta C_w$ is the difference in moisture concentration between the film and ambient temperature and $h_m$ is the convective mass transfer coefficient which can be obtained using the Chilton – Colburn analogy provided by Equation. (3.31)

$$h_m = \frac{h_w}{\rho_w C_{pw} Le^{2/3}} \tag{3.31}$$

Where $\rho_w$, $h_w$ and $C_{pw}$ are properties of water and represent the density of water , the heat transfer coefficient of water and specific heat capacity of water at ambient air condition (Kosky & Wise, 2021). $Le$ is the Lewis number and can be obtained from equation (3.32).

$$Le = \frac{K_w}{\rho_w D C_{pw}} \tag{3.32}$$

Where $K_w$ and $D$ are properties of water and represent the thermal conductivity of water and the mass diffusivity of water respectively.

b. $\dot{m}_{wd} < \dot{m}_{ev}$

For this situation, the rate at which water is diffused to the surface is less than the evaporation rate. Hence, the rate of water evaporation can be assumed to be equal to the rate of water diffusion to the surface as presented by equation (3.33).

$$\dot{m}_{wd} = \dot{m}_{ev}$$

(3.33)

Hence, the mass flow rate of water diffused is given by equation (3.32)

$$\dot{m}_{wd} = \frac{D_w A_o M_w}{GT} * \frac{dP_v}{dx}\left(\frac{\mathrm{Pv}}{P - P_v}\right) = \dot{m}_{ev} \tag{3.34}$$

Taking the differential of equation (3.30), we have equation (3.35)

$$d\dot{m}_{ev} = h_m A_o \partial C_w \tag{3.35}$$

However, Camargo et al. (2003) proposed equation (3.36) as the mass of water evaporated from the outer surface of the clay pot in differential form.

$$dm_{ev} = h_m \rho_w (\omega_v - \omega_{amb}) dA \tag{3.36}$$

Where $w_v$ and $\omega_{amb}$ are specific humidity of water vapour at ambient temperature and the specific humidity at ambient air condition. Integrating equation (3.36), we obtain equation (3.37).

$$\dot{m_{ev}} = h_m A_o \rho_w (\omega_v - \omega_{amb}) \tag{3.37}$$

But the specific humidity of ambient air is not a constant value and depends on the dew point temperature. It is given by the correlation presented in equation (3.38), (Banaszak, 2021).

$$\omega_{amb} = 6.11 \times 10^{\frac{(7.5 \times T_{dp})}{(237.3+T_{dp})}} \tag{3.38}$$

Where $T_{dp}$ is the dew point temperature

Substituting appropriate equations into equation (3.28), we obtain equation (3.39)

$$Q_L = h_m A_o \rho_w \left( \omega_v - \left| 6.11 \times 10^{\frac{(7.5 \times T_{dp})}{(237.3+T_{dp})}} \right] \right) h_{fg} \tag{3.39}$$

Similarly, Eq. (3.7); becomes

$$q_{out} = -\left\{ \left[ \frac{K_2+2K_1+2\varepsilon(K_2-K_1)}{K_2+2K_1-\varepsilon(K_2-K_1)} \right] A_o \left[ \frac{\partial^2 T_{opm}}{\partial x^2} + \left( \frac{\partial T_{opm}}{\partial x} \right) \right] \right\} + \frac{K_a}{L} \left[ 0.68 + \frac{0.67(Gr.Pr)^{1/4}}{\left[ 1+(0.492/Pr)^{9/16} \right]^{4/9}} \right] A_o \left( T_{amb} - T_{opm} \right) + h_m A_o \rho_w \left( \omega_v - \left| 6.11 \times 10^{\frac{(7.5 \times T_{dp})}{(237.3+T_{dp})}} \right] \right) h_{fg} \tag{3.40}$$

Also, the change in the internal energy of the outer porous wall due to heat and mass transfer is obtained using equation (3.41).

$$\Delta U_{opm} = \rho_c V_{opm} C_{pc} \frac{\partial T_{opm}}{\partial \tau} \tag{3.41}$$

where $\rho_c = 1450 kg/m^3$ is the density of the porous clay, $C_{pc} = 0.880 KJ/KgK$ is the specific heat capacity of porous clay (Bejan and Krauss, 2003), $V_{opm}$ is the volume of the outer porous clay wall, $T_{opm}$ is the outer porous media temperature, $\tau$ is time. Substituting equations (3.5), (3.40) and (3.41) into Equation (3.3) gives equation (3.42)

$$-K_{eff}A_o\frac{\partial T_{opm}}{\partial x} = -\left\{\left[\frac{K_2+2K_1+2\varepsilon(K_2-K_1)}{K_2+2K_1-\varepsilon(K_2-K_1)}\right]A_o\left[\frac{\partial^2 T_{opm}}{\partial x^2}+\left(\frac{\partial T_{opm}}{\partial x}\right)\right]\right\}+\frac{K_a}{L}\left[0.68+\frac{0.67(Gr.Pr)^{1/4}}{\left[1+(0.492/Pr)^{9/16}\right]^{4/9}}\right]A_o(T_{amb}-T_{opm})+h_mA_o\rho_w\left(\omega_v-\left[6.11\times 10^{\frac{(7.5\times T_{dp})}{(237.3+T_{dp})}}\right]\right)h_{fg} + \rho_cV_{opm}C_{pc}\frac{\partial T_{opm}}{\partial \tau} \tag{3.42}$$

Also, with respect to assumption (i); heat flows in a direction perpendicular to the wall surfaces resulting in a one-dimensional flow as shown in figure 3.3.

The expression for the change in outer porous media temperature with time is given by equation (3.43).

$$\frac{\partial T_{opm}}{\partial \tau}=\frac{1}{(\rho_cA_oH_oC_{pc})}\left|\left\{\left[\frac{K_2+2K_1+2\varepsilon(K_2-K_1)}{K_2+2K_1-\varepsilon(K_2-K_1)}\right]A_o\left[\frac{\partial^2 T_{opm}}{\partial x^2}+\left(\frac{\partial T_{opm}}{\partial x}\right)\right]\right\}\right.$$
$$-\left[\frac{K_2+2K_1+2\varepsilon(K_2-K_1)}{K_2+2K_1-\varepsilon(K_2-K_1)}\right]A_o\frac{\partial T_{opm}}{\partial x}$$
$$-\frac{K_{air}}{L}\left[0.68+\frac{0.67(Gr.Pr)^{1/4}}{\left[1+(0.492/Pr)^{9/16}\right]^{4/9}}\right]A_o(T_{amb}-T_{opm})$$
$$\left.-h_mA_o\rho_w\left(\omega_w-\left[6.11\times 10^{\frac{(7.5\times T_d)}{(237.3+T_d)}}\right]\right)h_{fg}\right| \tag{3.43}$$

### 3.3.4 Heat Transfer Through the Inner Porous Wall of the Cooler

Similar to the outer porous wall, the inner porous wall is a material containing pores or spaces in or around it to permit movement of fluid through or around the pores. Considering assumption (viii), the energy balance within the inner porous wall is also as presented by equation (3.3) above. While the energy influx to the inner porous wall is as presented by equation (3.44), the energy efflux is as presented by equation (3.45).

$$q_{in} = q_x = -K_{eff}A_i\frac{\partial T_{ipm}}{\partial x} \tag{3.44}$$

Where $A_i$ and $T_{ipm}$ represents the area of the inner porous wall and temperature of the inner porous wall.

Also, the inner porous wall is a composite wall moulded from clay, so the thermal conductivity will be the effective thermal conductivity of the composite wall denoted by $K_{eff}$ with the correlation as presented in equation (3.6).

$$q_{out} = q_{x+dx} = q_x - \frac{\partial}{\partial x}(q_x)dx = -K_{eff}A_i\frac{\partial T_{ipm}}{\partial x} + \left(K_{eff}A_i\frac{\partial^2 T_{ipm}}{\partial x^2}\right) + q_s \tag{3.45}$$

Where $Q_{conv}$ is the convective heat transfer at the liquid-solid interface of the inner porous wall surface and it is as presented in equation (3.46). Every other parameter has the same meaning.

But $Q_{conv} = h_w A_i(T_{ipm} - T_{ch})$ (3.46)

Where $T_{ch}$ is the storage chamber temperature and $h_w$ is the convective heat transfer coefficient of humid air and is described by the correlation as presented by equation (3.47).

$$h_w = \frac{K_w}{L} * Nu \tag{3.47}$$

Where $K_w$ is the thermal conductivity of humid air and the Nusselt number ($Nu$) is described by equation (3.48).

$$Nu = 0.508(Pr)^{\frac{1}{2}}(0.952 + Pr)^{-\frac{1}{4}}(Gr)^{\frac{1}{4}} \tag{3.48}$$

$Gr$ is the Grashoff's number described by the correlation in equation (3.24). Substituting appropriate equations, we obtain the correlations for the heat transfer coefficient of humid air and the convective heat transfer on the inner porous wall surface as presented by equations (3.49) and (3.50).

$$h_w = \frac{K_w}{L} * 0.508(Pr)^{\frac{1}{2}}(0.952 + Pr)^{-\frac{1}{4}}(Gr)^{\frac{1}{4}} \quad (3.49)$$

$$Q_{conv} = \frac{K_w}{L} * 0.508(Pr)^{\frac{1}{2}}(0.952 + Pr)^{-\frac{1}{4}}(Gr)^{\frac{1}{4}}A_i(T_{ipm} - T_{ch}) \quad (3.50)$$

Substituting equation (3.50) into equation (3.45), we obtain the energy efflux from the porous wall as given in equation (3.51)

$$q_{out} = q_{x+dx} + q_{conv} = q_x - \frac{\partial}{\partial x}(q_x)dx = -K_{eff}A_i\frac{\partial T_{ipm}}{\partial x} + \left(K_{eff}A_i\frac{\partial^2 T_{ipm}}{\partial x^2}\right) + \frac{K_w}{L} *$$
$$0.508(Pr)^{\frac{1}{2}}(0.952 + Pr)^{-\frac{1}{4}}(Gr)^{\frac{1}{4}}A_i(T_{ipm} - T_{ch}) \quad (3.51)$$

Again, the correlation for the change in the internal energy of the inner porous wall is as presented by equation (3.52).

$$\Delta U_{ipm} = \rho_c V_{ipw} C_{pc}\frac{\partial T_{ipm}}{\partial \tau} \quad (3.52)$$

Where $\Delta U_{ipm}$ is the change in the internal energy of the inner porous wall and $V_{ipw}$ is the volume of the inner porous wall.

Substituting equations (3.44), (3.51) and (3.52) into equation (3.3), the energy balance is as presented by equation (3.53).

$$\rho_c V_{ipw} C_{pc}\frac{\partial T_{ipm}}{\partial \tau} = \left(-K_{eff}A_i\frac{\partial T_{ipm}}{\partial x}\right) + \left\{K_{eff}A_i\left(\frac{\partial T_{ipm}}{\partial x} - \frac{\partial^2 T_{ipm}}{\partial x^2}\right)\right\} - \left\{\frac{K_w}{L} *\right.$$
$$\left.0.508(Pr)^{\frac{1}{2}}(0.952 + Pr)^{-\frac{1}{4}}(Gr)^{\frac{1}{4}}A_i\left(T_{ipm} - T_{ch}\right)\right\}$$

(3.53)

The expression for the transient change in temperature of the inner porous wall can be obtained using equation (3.54).

$$\frac{\partial T_{ipm}}{\partial \tau} = \frac{1}{\rho_c V_{ipw} C_{pc}} \left\{ -K_{eff} \frac{\partial^2 T_{ipm}}{\partial x^2} - \left[ \frac{K_w}{L} 0.508 (Pr)^{\frac{1}{2}} (0.952 + Pr)^{-\frac{1}{4}} (Gr)^{\frac{1}{4}} A_i (T_{ipm} - T_{ch}) \right] \right\}$$

(3.54)

### 3.3.5 Energy Balance Within the Interspace

Considering assumption (vi), heat is transferred through the interspace by conduction, convection and due to the difference in the temperature of water within the inner porous media wall surfaces and the temperature of water replaced from the reserviour. Conduction accounts for heat transfer through coconut fibre while convection accounts for heat transfer due to water absorbed by the fibre within the section. However, the coconut fibre is loosely bound together such that it forms an area equal to the area of the inner walls of the porous media. It serves as a sponge absorbing the water before water seeps to the side porous walls. Considering assumption (x), the quantity of water absorbed by the coconut fibre is very small and hence convection heat transfer is neglected.

Therefore, change in energy within the interspace is as a result of heat conduction from the outer and inner porous media wall surfaces and energy change due to the presence of water as presented by equation (3.55).

$$\Delta U_{int,isp} = Q_{cond} + Q_{conv} + Q_w \qquad (3.55)$$

Where $U_{int,isp}$ is the change in the internal energy of the interspace, $Q_{conv}$ is the convective heat transfer due to water absorbed by the coconut fibre absorbent, $Q_w$ is the energy change due to the presence of water and $Q_{cond}$ is the heat conducted into the interspace from the inner and outer porous wall surfaces. The change in internal energy of the interspace is given by equation (3.56) while the conduction heat transfers from the inner and outer porous wall surfaces is given by equation (3.57).

$$\Delta U_{int,isp} = \rho_w V_i C_{pw} \frac{\partial T_{sf}}{\partial \tau} \qquad (3.56)$$

Where $\rho_w$ is the density of water, $V_i$ is the volume of the interspace, $C_{pw}$is the specific heat capacity of water and $T_{sf}$ is the Surface temperature.

$$Q_{cond} = -K_{opm}A_o \frac{\partial T_{sf}}{\partial x}|_{x=R_2} - K_{ipm}A_i \frac{\partial T_{sf}}{\partial x}|_{x=R_1} \tag{3.57}$$

However, from figure 3.2, the following boundary and conditions applies;

<u>Boundary Conditions</u>

$$At\ x = R_1;\ K\frac{\partial T_{sf}}{\partial x} = K\frac{\partial T_{ipm}}{\partial x} \tag{3.58}$$

$$At\ x = R_2;\ K\frac{\partial T_{sf}}{\partial x} = K\frac{\partial T_{opm}}{\partial x} \tag{3.59}$$

<u>Initial Conditions</u>

$$T_{sf\ @\ t=0} = T_{amb}$$

$$T_{opm\ @\ t=0} = T_{amb}$$

$$T_{ipm\ @\ t=0} = T_{amb}$$

$$T_{ch\ @\ t=0} = T_{amb}$$

But the porous walls are homogenous and isotropic, hence;

$$K_{opm} \equiv K_{ipm} \equiv K_{eff} = \left[\frac{K_2+2K_1+2\varepsilon(K_2-K_1)}{K_2+2K_1-\varepsilon(K_2-K_1)}\right]$$

Substituting equations (3.58) and (3.59) into equation (3.57), we obtain equation (3.60) which is the expression for the conductive heat transferred into the interspace from the inner and outer wall surfaces.

$$Q_{cond} = -K_{eff}\left[A_o \frac{\partial T_{opm}}{\partial x}|_{x=R_2} + A_i \frac{\partial T_{ipm}}{\partial x}|_{x=R_1}\right] \tag{3.60}$$

The energy change due to the presence of water ($Q_w$) is given by equation (3.61).

$$Q_w = \dot{m}_w C_{pw}\left(T_{ipm} - T_{rep}\right) \tag{3.61}$$

Where $\dot{m}_w$ is the mass flow rate of water and $T_{rep}$ is the temperature of the replacement water from the reserviour and is described by the correlation $T_{rep} = (T_{amb} - 3)$.

Substituting equations (3.56), (3.60) and (3.61) into equation (3.55), the expression for the energy balance within the interspace is obtained and is as presented by equation (3.62).

For $Q_{conv} = 0$;

$$\rho_w V_i C_{pw} \frac{\partial T_{sf}}{\partial \tau} = -K_{eff}\left[A_o \frac{\partial T_{opm}}{\partial x}|_{x=R_2} + A_i \frac{\partial T_{ipm}}{\partial x}|_{x=R_1}\right] + \dot{m}_w C_{pw}\left(T_{ipm} - T_{rep}\right)$$

(3.62)

Hence, the transient change in the surface temperature of the interspace is as presented in equation (3.63).

$$\frac{\partial T_{sf}}{\partial \tau} = \frac{1}{\rho_w V_i C_{pw}}\left\{-K_{eff}\left[A_o \frac{\partial T_{opm}}{\partial x}|_{x=R_2} + A_i \frac{\partial T_{ipm}}{\partial x}|_{x=R_1}\right] + \dot{m}_w C_{pw}\left(T_{ipm} - T_{rep}\right)\right\} \quad (3.63)$$

### 3.3.6 Heat Transfer from the Inner Porous Wall Surface

The inner porous wall surface interacts with the storage chamber. Therefore, heat is transferred by convection and evaporation between the humid air within the storage chamber and the inner surface of the porous medium. However, overtime there is a build up of humidity within the storage chamber making the rate of water evaporation from the inner wall surface to the cooling chamber is very low hence evaporative cooling becomes insignificant; thus the driving force that continues to maintain the cooling within the chamber remains only the convection heat transfer. Therefore, considering assumption (ii), evaporation heat transfer is negligible. The change in the internal energy of the storage chamber is given by equation.

$$\Delta U_{int,sc} = Q_{conv,sc} \quad (3.64)$$

Where $\Delta U_{int,sc}$ is the change in the internal energy of the storage chamber and it is described by equation (3.65) and $Q_{conv,sc}$ is the convective heat transferred from the inner porous wall surface to the storage chamber and it is described by equation (3.68).

$$\Delta U_{int.sc} = \rho_{ha} C_{pha} \frac{\partial T_{ch}}{\partial \tau} \quad (3.65)$$

Where $\rho_{ha}$ is the density of humid air in the storage chamber described by equation (3.66), $C_{pha}$ is the specific heat capacity of humid air and $T_{ch}$ is the storage chamber temperature.

$$\rho_{ha} = \left[\frac{p_d}{R_d T_{ch}} + \frac{p_v}{R_v T_{ch}}\right] \tag{3.66}$$

Where $p_d$ is the partial pressure of dry air (Pa), $R_d$ is the specific gas constant of dry air = 287.05J/Kg.K , $p_v$ is the partial pressure of water vapour (Pa) and $R_v$ is the specific gas constant of water vapour = 461.495J/Kg.K.

Similarly, the specific heat capacity of humid air is described by equation (3.67).

$$C_{pha} = C_{pa} + C_{pv}\omega \tag{3.67}$$

Where $C_{pa}$ is the specific heat capacity of dry air, $C_{pv}$ is the specific heat capacity of water vapour and $\omega$ is the specific humidity in kg water vapor per kg dry air in the mixture.

$$Q_{conv,sc} = h_{ipm} A_i \left(T_{ipm} - T_{ch}\right) \tag{3.68}$$

But $h_{ipm} = h_w$ which is described by equation (3.47). All other parameters have same description.

For a vertical wall, the Nusselt number (Nu) is given by the correlation below;

$$Nu = 0.508(Pr)^{\frac{1}{2}}(0.952 + Pr)^{-\frac{1}{4}}(Gr)^{\frac{1}{4}} \tag{3.69}$$

Where $Gr$ and $Pr$ are as described by equations (3.24) and (3.25).

Therefore, the transient change in energy within the storage chamber is given by equation (3.70).

$$\frac{\partial T_{ch}}{\partial \tau} = \frac{h_w A_i \left(T_{ipm} - T_{ch}\right)}{\left[\left(\frac{p_d}{R_d T_{ch}} + \frac{p_v}{R_v T_{ch}}\right)\left(C_{pa} + C_{pv}\omega\right)\right]} \tag{3.70}$$

### 3.4 Solution to the Model Equations

The developed model equations were solved using FlexPDE computational fluid dynamic analysis model builder and numerical solver version 7.2.1 to generate numerical results. The numerical results are presented and discussed in chapter 4. Data obtained for the numerical study were validated using experimental data. The root mean Square Error (RMSE) method given as equation 3.71 was used for the validation study.

$$RMSE = \sqrt{\frac{\sum_{i=1}^{N} \| E_{(i)} - N_{(i)} \|^2}{N}} \tag{3.71}$$

N is the number of data points, $E_{(i)}$ is the i$^{th}$ experimental measurement, $N_{(i)}$ is the i$^{th}$ numerical measurement.

# CHAPTER FOUR

# RESULTS AND DISCUSSION

## 4.1 Results

FlexPDE computational fluid dynamic analysis software was used to generate numerical solutions of the models developed in chapter three. They are presented below;

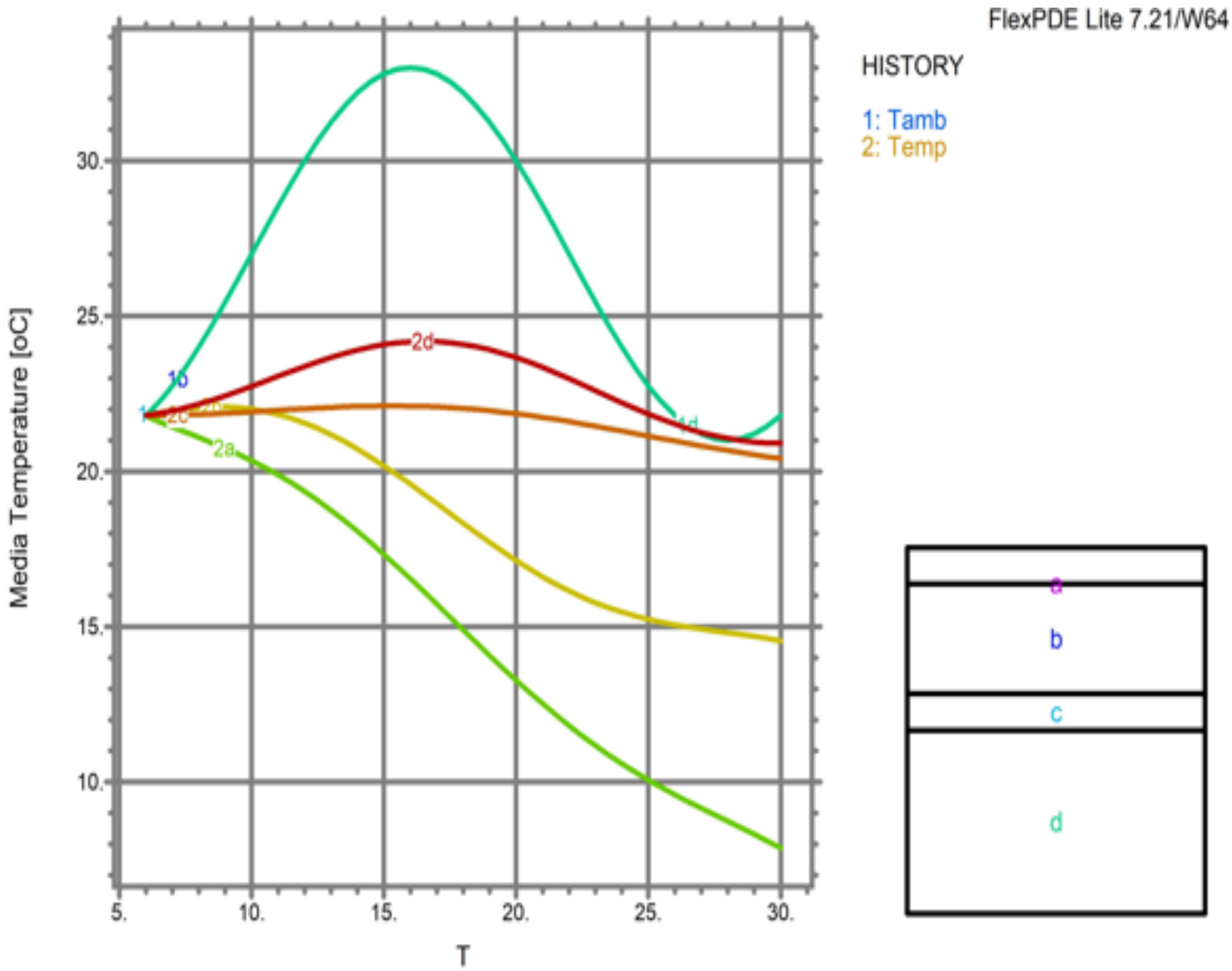


Fig.4.1: Temperature Profile of Ambient and Cooler Sections in January

**LEGEND**

| | |
|---|---|
| A | Outer porous media |
| B | Interspace |
| C | Inner porous media |
| D | Storage Chamber |

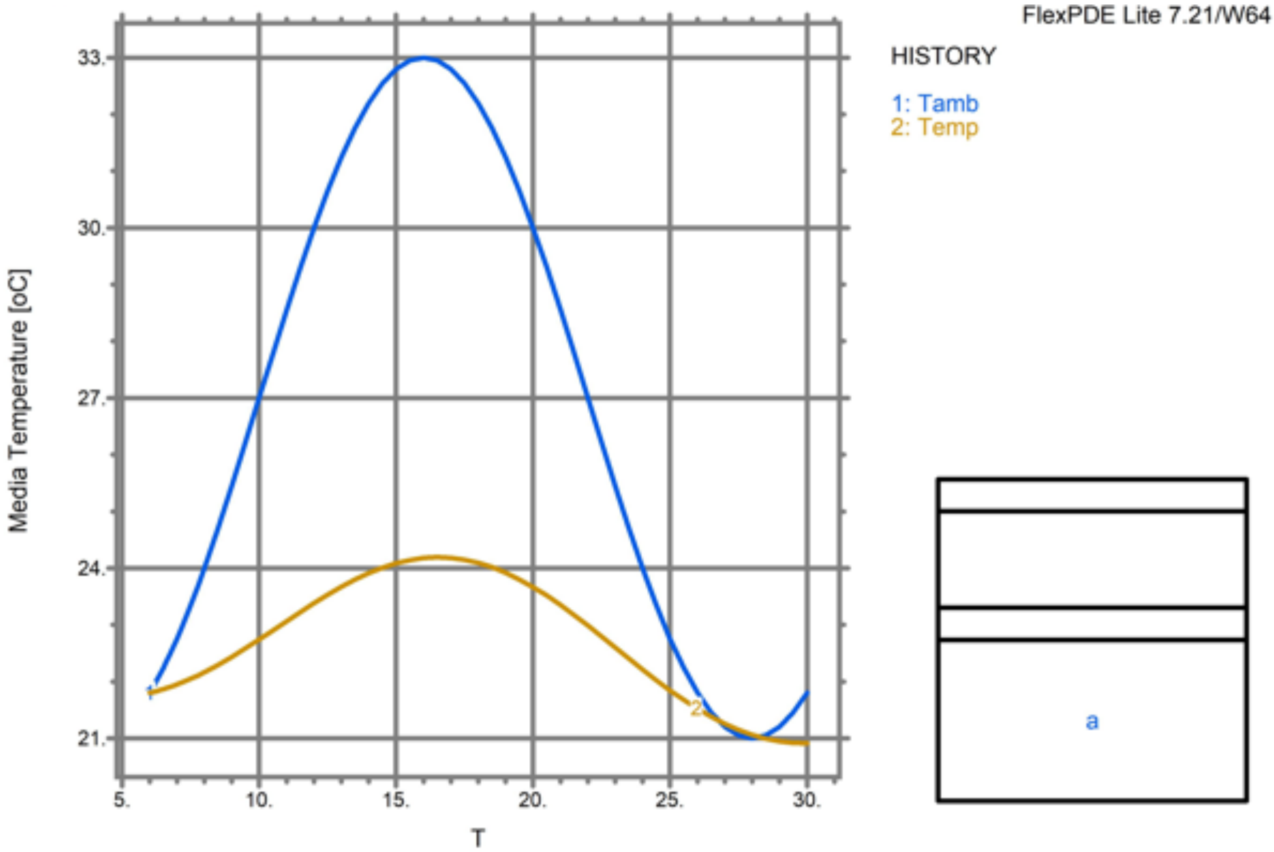


Fig. 4.2: Temperature Profile of Ambient and Cooling Chamber in January

**LEGEND**

a Storage Chamber

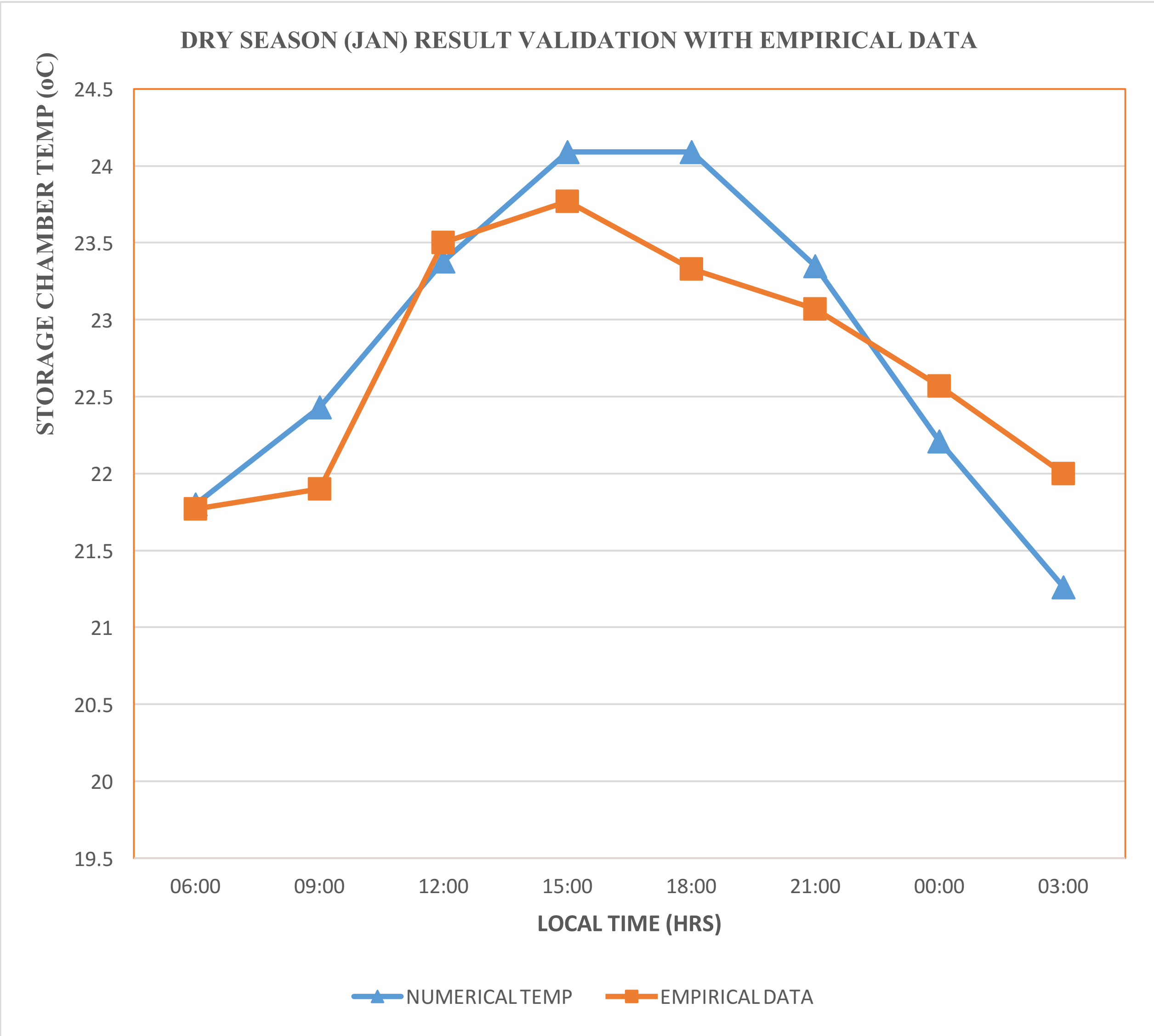

Fig. 4.3: Validation of Numerical Results with Experimental Data

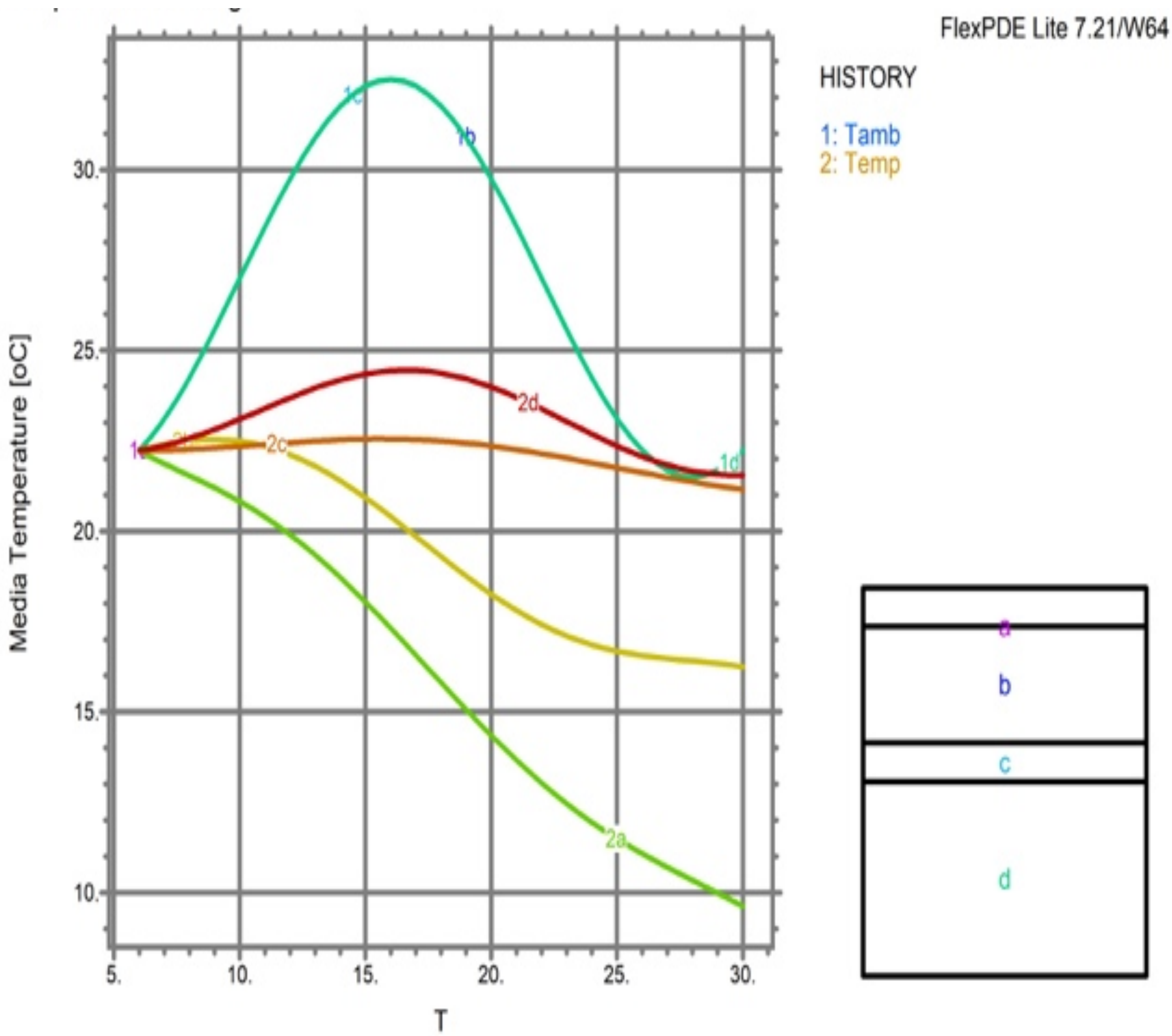


Fig.4.4: Temperature Profile of Ambient and Cooler Sections in November

**LEGEND**

| | |
|---|---|
| a | Outer porous media |
| b | Interspace |
| c | Inner porous media |
| d | Storage Chamber |

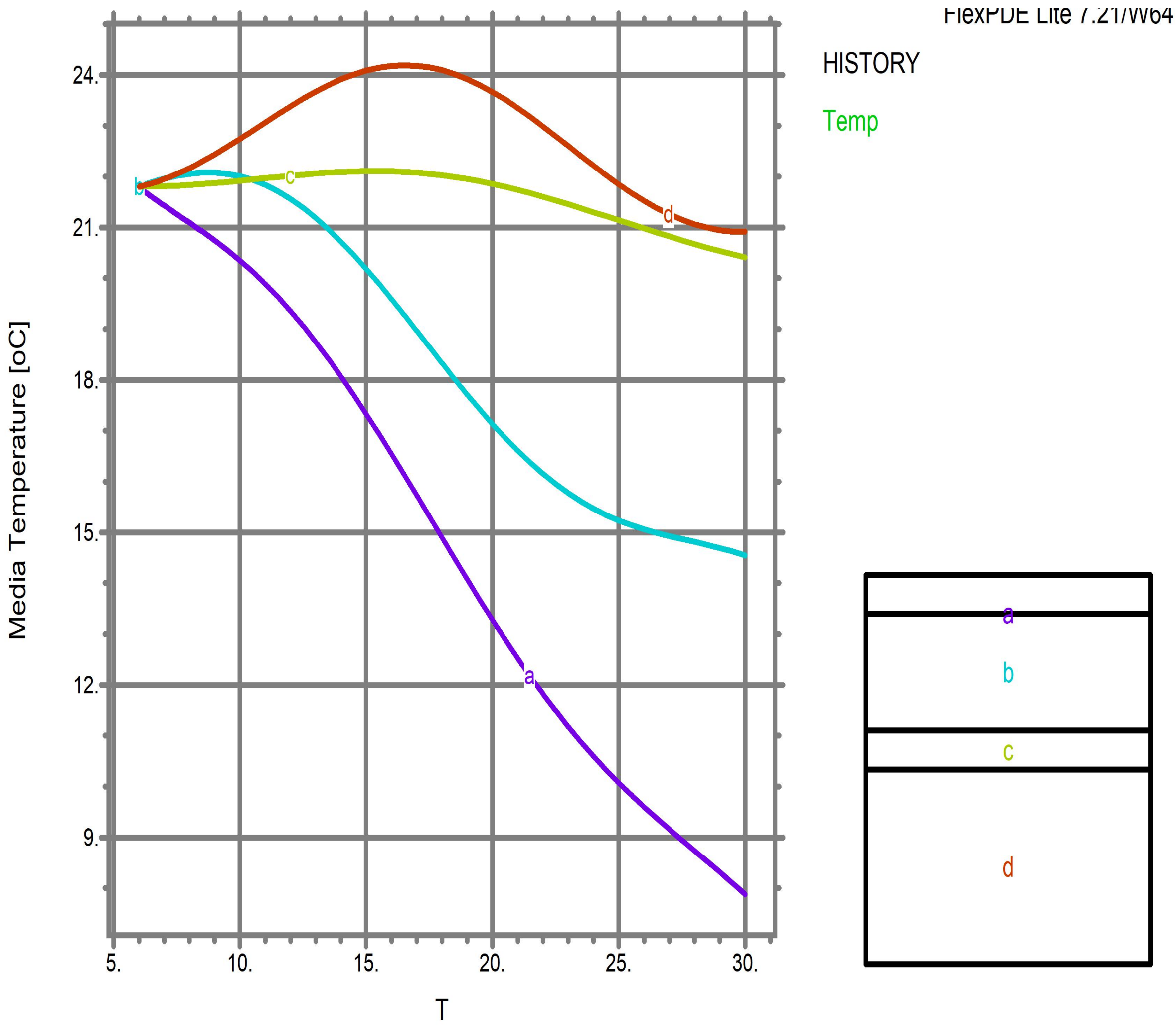


Fig. 4.5: Temperature Profile of Cooler Sections in January

**LEGEND**

| | |
|---|---|
| a | Outer porous media |
| b | Interspace |
| c | Inner porous media |
| d | Storage Chamber |

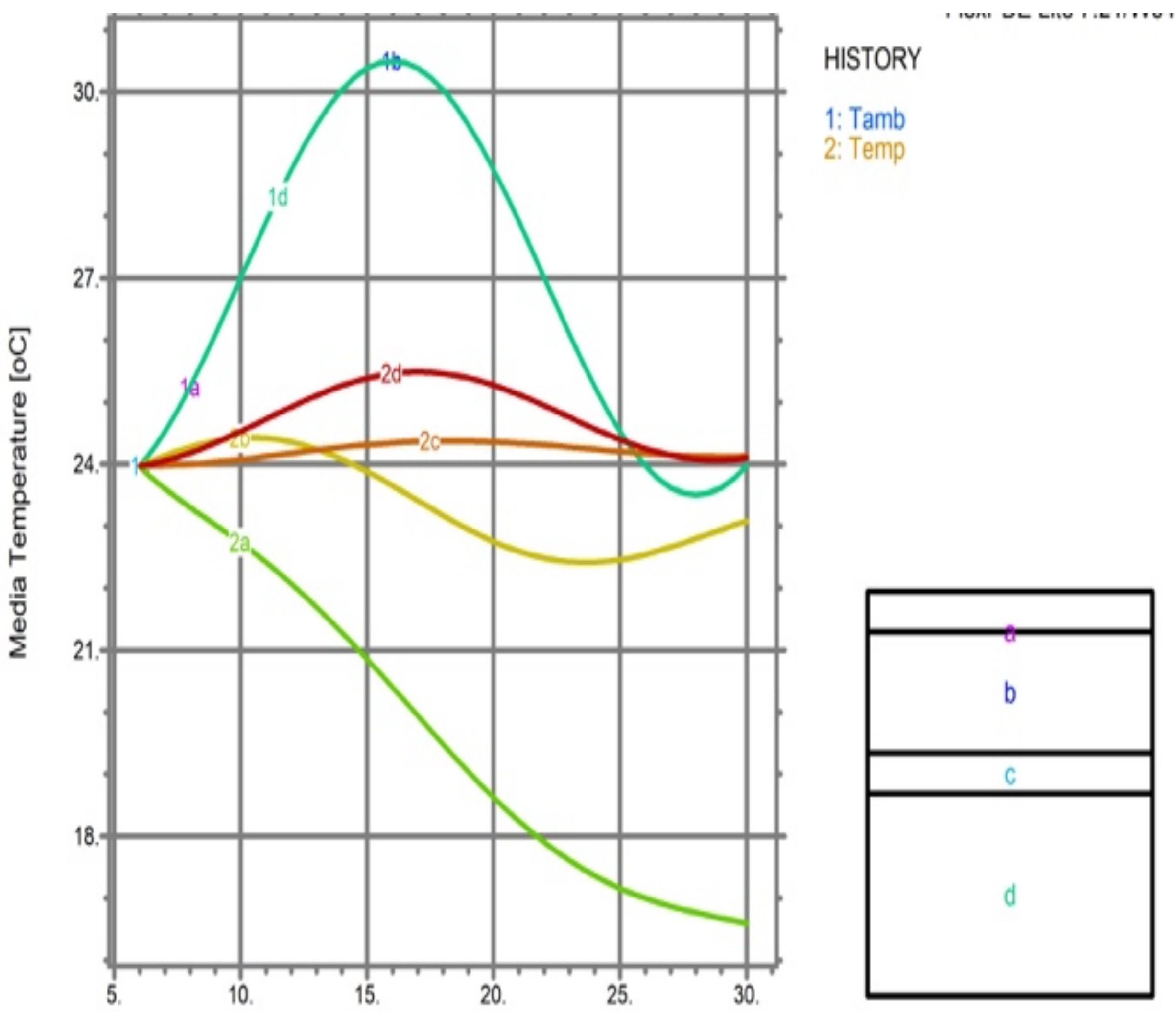


Fig.4.6: Temperature Profile for Ambient and the Cooler Sections in September

**LEGEND**

| | |
|---|---|
| A | Outer porous media |
| B | Interspace |
| C | Inner porous media |
| D | Storage Chamber |

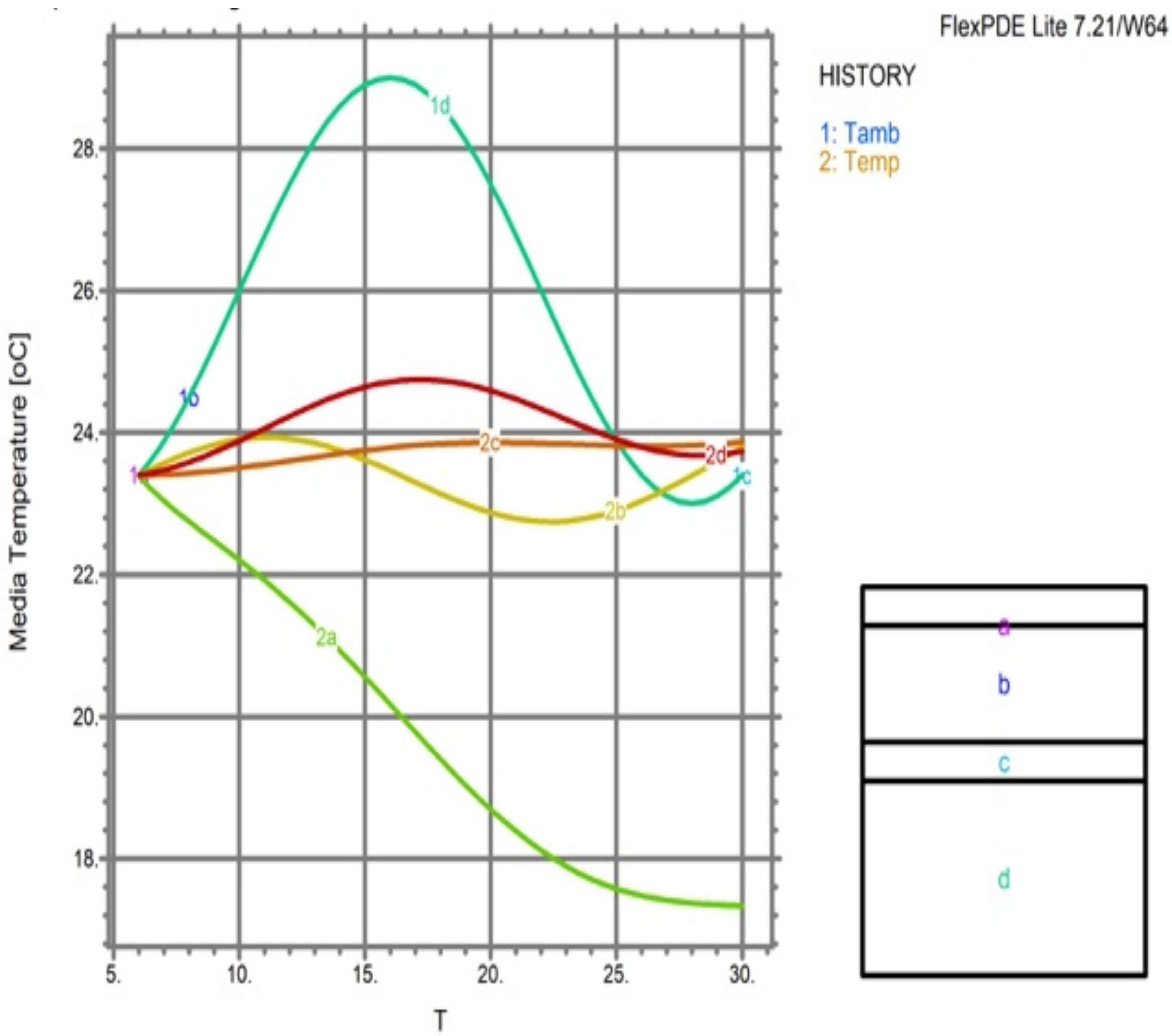


Fig. 4.7: Temperature Profile for the Ambient and Cooler Sections in April

**LEGEND**

| | |
|---|---|
| a | Outer porous media |
| b | Interspace |
| c | Inner porous media |
| d | Storage Chamber |

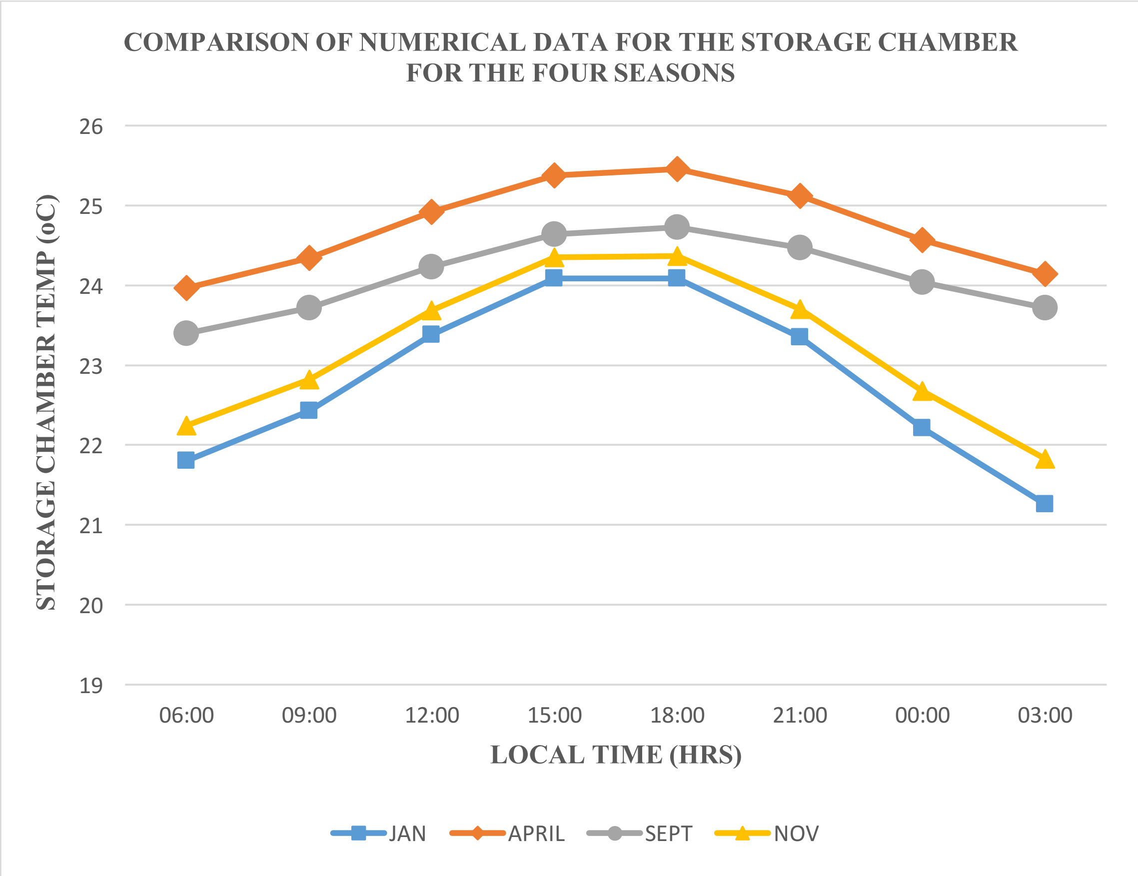


Fig. 4.8: Plot of Storage Chamber Temperatures for the Four Seasons

## 4.2 Discussion

Figures 4.1 and 4.2 show the numerical solutions obtained using meteorological data corresponding with that for one of the days in January when the physical system was tested. While figure 4.1 shows the temperature profile for the different sections of the evaporative cooler (i.e. outer/inner porous walls, interstitial space and cooling chamber) and relates these temperatures to ambient air temperature, figure 4.2 relates the cooling chamber temperature and the ambient air temperature. Taking a close look at figure 4.1, it can be seen that there is a continuous drop in the outer wall outer surface temperature below ambient which is indicative of the evaporative cooling process taking place on that surface as a result of continuous water seepage through the pores of the porous wall to its surface. A closer examination of the temperature profile for the outer surface shows that the rate of temperature drop of the surface was highest  between 1100hrs and 1700hrs. This is because of increasing insolation, ambient temperature increases and relative humidity of ambient air becomes much lower hence increasing the rate of water evaporation from the surface. From figure 4.2, the high ambient temperature observed within the same period (i.e. 1100hrs – 1700hrs) is responsible for the increasing temperature trend within the cooling chamber. These high ambient temperatures tend to increased heat ingress into the cooling chamber thus cancelling out the evaporative cooling effect happening on the outer porous wall.

The effect of the high ambient temperature was also felt by the interstitial space and inner porous wall but the degree was not as high as the cooling chamber temperature because these other sections were in direct contact with the outer porous wall experiencing evaporative cooling.  However, it can be seen from the graph that despite the effect of the high ambient temperature, the cooling chamber remained consistently below ambient temperature values. A temperature depression between 2 – 9$^{o}$C was observed between cooling chamber and the ambient air temperature with the maximum of 9$^{o}$C occurring at  about 1500hrs. Analyzing the profile, it can be seen that during the hottest hours of the day (i.e. between 1100hrs and 1700hrs) when cooling is most desired, the outer porous media temperature indicated a depression within the range of 5 - 17$^{o}$C from the ambient air temperature which is  very good since the outer media temperature drives the cooling effect felt within the entire system. Also, between same period (i.e. 1100hrs -1700hrs of local time) when the ambient air temperature is somewhat at its peak, a significantly high depression occurred. This indicates that during the hottest hours of the day when radiant energy was at its highest and cooling most needed, the storage chamber temperatures continuously showed lower values when compared to the ambient air temperature. From meteorological data available for the wind velocities in Owerri, Imo State, the month of January is usually not very windy, the average wind velocity is 2.2m/s (“Average Wind Speed in Owerri,” n.d.). Hence it is assumed that the temperature parameters were greatly affected by the weather condition. Therefore, the performance profile of the evaporative cooler during the month of January can be attributed to the low humidity and high temperature of ambient air.

Figure 4.3 shows the validation of numerical solutions with experimental data. Analyzing the profile of figure 4.3, it can be seen that the numerical solution closely predicted the experimental results. Using the Root Mean Square Error (RMSE) statistical tool defined by equation (3.71) to predict how accurately the numerical data predicted the experimental data, the calculated result showed that the maximum error between the two available data was 0.205 which is indicative of the accuracy of the model as a predictive tool for the system performance. Based on this established accuracy of the model, it was subsequently used to predict the performance of the evaporative cooler at different climatic seasons experienced in Nigeria as shown in figures 4.4 – 4.7.

The figures (4.4 – 4.7) showed the same profile as figure 4.1 but with different cooling chamber and outer porous wall temperatures. Cooling chamber was between 24 – 25.5$^{O}$C for the early wet season (Figure 4.6), 23.68 – 24.72$^{O}$C for the late wet season (figure 4.7), 21.5 – 24.3$^{O}$C for the early dry season (figure 4.4) and 21 – 24$^{O}$C for the late dry season (figure 4.1).These correspond to temperature depression of 5.2$^{O}$C, 4.48$^{O}$C, 8.2$^{O}$C and 9$^{O}$C respectively for the early wet season, late wet season, early dry season and late dry season. This observed temperature depression trend is expected since the cooling achieved is largely dependent on the relative humidity (RH). Relative humidity is generally lowest during the late dry season. This is followed by the early dry season which presents slightly higher RH. Late wet season has the highest RH.

# CHAPTER FIVE

# CONCLUSIONS AND RECOMMENDATIONS

## 5.1 Conclusions

In this work, the transient performance of a porous evaporative cooling system has been evaluated and reported. The system models have been developed and validated using experimental data obtained from a properly designed, constructed and tested evaporative cooler. The developed models were used to generate numerical solutions which showed that the temperature depression in storage chamber temperature from ambient air condition reached 3-9$^0$C for ambient air temperature within the range of 22 – 33$^o$C. This result is very well encouraging as vegetables and fruits could be preserved for a period of time to prevent rotting and wastage after harvest.

The results of this analysis showed that evaporative cooling systems can be an effective and affordable technology to achieve significant cooling and energy savings for the preservation of fruits and vegetables in urban and rural areas without grid line electricity. A Transient Performance evaluation of the numerical results for the four seasons of the year i.e. early wet and dry seasons (e.g. April and November) and late wet and dry seasons (e.g. September and January), showed that the system can be comfortably applied all year round but functions best during periods of low humidity and elevated ambient air temperature (i.e. dry season). This means that with improved performance, better results can be obtained over a prolonged test hours and optimum performance parameters can be established for better cooling effect following the level of cooling obtained during the 24hour period.

Also, results showed that evaporative cooling is not very suitable in the tropics where air is moist and mostly still which is especially during the wet seasons. During the dry season, the harmattan haze blowing dry cold air from the Northern hemisphere reduces the specific humidity of the ambient air considerably and also increases air flow velocity. During this period, the evaporative cooling technique becomes more effective and can be employed for cooling purposes. This implies that the wind velocity plays a significant role in achieving effective evaporative cooling.

## 5.2 Recommendations

This work recommends that a parametric study and optimization of the design parameters of the evaporative cooler be carried out in further research.

## 5.3 Contribution to Knowledge

In this work, a model for transient performance analysis of a porous evaporative cooler have been developed and the model can be used for the system performance optimization.

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# APPENDIX

## Appendix A: Temperature Data for Late Dry Season (January)

| Local Time (hrs) | Storage Chamber Temperature $T_{ch}$ (°C) | | Temperature of the Cooler Sections(°C) | | |
|---|---|---|---|---|---|
| | $T_N$ | $T_E$ | $T_{opm}$ | $T_i$ | $T_{ipm}$ |
| 06:00 | 21.8 | 21.77 | 21.80 | 21.80 | 21.80 |
| 06:30 | 21.87 | 21.88 | 21.60 | 21.89 | 21.81 |
| 07:00 | 21.95 | 21.89 | 21.43 | 21.96 | 21.82 |
| 07:30 | 22.05 | 22 | 21.26 | 22.02 | 21.82 |
| 08:00 | 22.17 | 22.1 | 21.09 | 22.06 | 21.83 |
| 08:30 | 22.29 | 22.15 | 20.92 | 22.08 | 21.85 |
| 09:00 | 22.43 | 22.4 | 20.74 | 22.08 | 21.87 |
| 09:30 | 22.58 | 22.42 | 20.55 | 22.06 | 21.89 |
| 10:00 | 22.74 | 22.64 | 20.35 | 22.01 | 21.92 |
| 10:30 | 22.9 | 22.85 | 20.13 | 21.93 | 21.94 |
| 11:00 | 23.07 | 23.02 | 19.89 | 21.84 | 21.97 |
| 11:30 | 23.23 | 23.25 | 19.63 | 21.72 | 21.99 |
| 12:00 | 23.38 | 23.5 | 19.36 | 21.56 | 22.02 |
| 12:30 | 23.53 | 23.5 | 19.06 | 21.38 | 22.04 |
| 13:00 | 23.67 | 23 | 18.75 | 21.19 | 22.06 |
| 13:30 | 23.8 | 23.5 | 18.42 | 20.96 | 22.08 |
| 14:00 | 23.91 | 23.17 | 18.08 | 20.72 | 22.09 |
| 14:30 | 24.01 | 23.98 | 17.71 | 20.46 | 22.10 |
| 15:00 | 24.09 | 23.99 | 17.33 | 20.18 | 22.11 |

| | | | | | |
|---|---|---|---|---|---|
| 15:30 | 24.14 | 24.12 | 16.94 | 19.89 | 22.11 |
| 16:00 | 24.18 | 23.99 | 16.54 | 19.59 | 22.10 |
| 16:30 | 24.19 | 24.05 | 16.14 | 19.28 | 22.09 |
| 17:00 | 24.18 | 24.11 | 15.72 | 18.96 | 22.08 |
| 17:30 | 24.15 | 24 | 15.31 | 18.64 | 22.06 |
| 18:00 | 24.09 | 23.33 | 14.89 | 18.33 | 22.03 |
| 18:30 | 24.02 | 24 | 14.48 | 18.02 | 21.99 |
| 19:00 | 23.92 | 23.99 | 14.01 | 17.71 | 21.95 |
| 19:30 | 23.8 | 23.74 | 13.67 | 17.42 | 21.91 |
| 20:00 | 23.67 | 23.64 | 13.27 | 17.13 | 21.86 |
| 20:30 | 23.51 | 23.42 | 12.89 | 16.86 | 21.80 |
| 21:00 | 23.35 | 23.27 | 12.52 | 16.61 | 21.74 |
| 21:30 | 23.17 | 23.02 | 12.16 | 16.37 | 21.67 |
| 22:00 | 22.99 | 22.85 | 11.82 | 16.15 | 21.60 |
| 22:30 | 22.8 | 22.65 | 11.49 | 15.95 | 21.53 |
| 23:00 | 22.6 | 22.7 | 11.17 | 15.77 | 21.46 |
| 23:30 | 22.41 | 22.38 | 10.87 | 15.61 | 21.39 |
| 00:00 | 22.21 | 22.12 | 10.59 | 15.47 | 21.30 |
| 00:30 | 22.03 | 22.01 | 10.32 | 15.34 | 21.22 |
| 01:00 | 21.85 | 21.8 | 10.07 | 15.23 | 21.14 |
| 01:30 | 21.68 | 21.65 | 9.83 | 15.14 | 21.06 |
| 02:00 | 21.52 | 21.5 | 9.59 | 15.06 | 20.98 |
| 02:30 | 21.38 | 21.24 | 9.37 | 14.99 | 20.90 |
| 03:00 | 21.26 | 21.35 | 9.16 | 14.93 | 20.82 |

| | | | | | |
|---|---|---|---|---|---|
| 03:30 | 21.15 | 21.27 | 8.95 | 14.87 | 20.75 |
| 04:00 | 21.06 | 21.02 | 8.74 | 14.81 | 20.67 |
| 04:30 | 20.99 | 20.93 | 8.53 | 14.75 | 20.60 |
| 05:00 | 20.95 | 20.9 | 8.32 | 14.69 | 20.54 |
| 05:30 | 20.92 | 20.88 | 8.10 | 14.62 | 20.47 |

**Appendix B: Temperature Data for Early Dry Season (November)**

| Local Time (hrs) | Storage Chamber Temperature $T_{ch}$ ($^oC$) | | Temperature of the Cooler Sections ($^oC$) | | |
|---|---|---|---|---|---|
| | $T_N$ | $T_E$ | $T_{opm}$ | $T_i$ | $T_{ipm}$ |
| 06:00 | 22.24 | 21.77 | 22.37 | 22.37 | 22.37 |
| 06:30 | 22.29 | 21.88 | 22.04 | 22.32 | 22.24 |
| 07:00 | 22.37 | 21.89 | 21.86 | 22.40 | 22.24 |
| 07:30 | 22.46 | 22 | 21.70 | 22.46 | 22.25 |
| 08:00 | 22.57 | 22.1 | 21.53 | 22.50 | 22.27 |
| 08:30 | 22.69 | 22.15 | 21.37 | 22.53 | 22.28 |
| 09:00 | 22.82 | 22.4 | 21.20 | 22.54 | 22.30 |
| 09:30 | 22.95 | 22.42 | 21.16 | 22.52 | 22.33 |
| 10:00 | 23.1 | 22.64 | 20.82 | 22.49 | 22.35 |
| 10:30 | 23.25 | 22.85 | 20.62 | 22.43 | 22.37 |
| 11:00 | 23.4 | 23.02 | 20.39 | 22.35 | 22.40 |
| 11:30 | 23.55 | 23.25 | 20.15 | 22.25 | 22.42 |
| 12:00 | 23.69 | 23.5 | 19.90 | 22.12 | 22.45 |
| 12:30 | 23.83 | 23.5 | 19.63 | 21.97 | 22.47 |
| 13:00 | 23.96 | 23 | 19.34 | 21.80 | 22.49 |
| 13:30 | 24.08 | 23.5 | 19.03 | 21.60 | 22.51 |
| 14:00 | 24.18 | 23.17 | 18.71 | 21.39 | 22.53 |
| 14:30 | 24.27 | 23.98 | 18.38 | 21.16 | 22.54 |
| 15:00 | 24.35 | 23.99 | 18.04 | 20.92 | 22.55 |
| 15:30 | 24.4 | 24.12 | 17.68 | 20.67 | 22.55 |

| | | | | | |
|---|---|---|---|---|---|
| 16:00 | 24.43 | 23.99 | 17.31 | 20.40 | 22.55 |
| 16:30 | 24.45 | 24.05 | 16.94 | 20.13 | 22.55 |
| 17:00 | 24.44 | 24.11 | 16.57 | 19.85 | 22.54 |
| 17:30 | 24.42 | 24 | 16.19 | 19.57 | 22.52 |
| 18:00 | 24.37 | 23.33 | 15.81 | 19.30 | 22.50 |
| 18:30 | 24.3 | 24 | 15.44 | 19.02 | 22.47 |
| 19:00 | 24.21 | 23.99 | 15.07 | 18.76 | 22.44 |
| 19:30 | 24.11 | 23.74 | 14.70 | 18.50 | 22.40 |
| 20:00 | 23.99 | 23.64 | 14.34 | 18.26 | 22.36 |
| 20:30 | 23.85 | 23.42 | 14.00 | 18.02 | 22.31 |
| 21:00 | 23.7 | 23.27 | 13.66 | 17.80 | 22.26 |
| 21:30 | 23.55 | 23.02 | 13.34 | 17.60 | 22.20 |
| 22:00 | 23.38 | 22.85 | 13.03 | 17.42 | 22.15 |
| 22:30 | 23.21 | 22.65 | 12.74 | 17.25 | 22.08 |
| 23:00 | 23.03 | 22.7 | 12.46 | 17.10 | 22.02 |
| 23:30 | 22.86 | 22.38 | 12.19 | 16.97 | 21.95 |
| 00:00 | 22.68 | 22.12 | 11.94 | 16.86 | 21.89 |
| 00:30 | 22.52 | 22.01 | 11.71 | 16.76 | 21.82 |
| 01:00 | 22.36 | 21.8 | 11.49 | 16.68 | 21.75 |
| 01:30 | 22.21 | 21.65 | 11.27 | 16.61 | 21.68 |
| 02:00 | 22.07 | 21.5 | 11.07 | 16.56 | 21.62 |
| 02:30 | 21.94 | 21.24 | 10.88 | 16.51 | 21.55 |
| 03:00 | 21.83 | 21.35 | 10.70 | 16.47 | 21.49 |
| 03:30 | 21.74 | 21.27 | 10.52 | 16.44 | 21.42 |

| | | | | | |
|---|---|---|---|---|---|
| 04:00 | 21.66 | 21.02 | 10.34 | 16.41 | 21.36 |
| 04:30 | 21.61 | 20.93 | 10.17 | 16.38 | 21.31 |
| 05:00 | 21.57 | 20.9 | 9.99 | 16.34 | 21.25 |
| 05:30 | 21.55 | 20.88 | 9.81 | 16.30 | 21.20 |

**Appendix C: Temperature Data for Late Wet Season (April)**

| Local Time (hrs) | Storage Chamber Temperature $T_{ch}$ ($^oC$) | | Temperature of the Cooler Sections ($^oC$) | | |
|---|---|---|---|---|---|
| | $T_N$ | $T_E$ | $T_{opm}$ | $T_i$ | $T_{ipm}$ |
| 06:00 | 23.4 | 23.6 | 23.40 | 23.40 | 23.40 |
| 06:30 | 23.43 | 23.22 | 23.20 | 23.49 | 23.40 |
| 07:00 | 23.47 | 23.57 | 23.04 | 232.57 | 23.41 |
| 07:30 | 23.53 | 23.48 | 22.89 | 23.65 | 23.41 |
| 08:00 | 23.58 | 23.53 | 22.75 | 23.72 | 23.43 |
| 08:30 | 23.65 | 23.62 | 22.61 | 23.78 | 23.44 |
| 09:00 | 23.72 | 24.83 | 22.48 | 223.83 | 23.46 |
| 09:30 | 23.8 | 24 | 22.34 | 23.87 | 23.48 |
| 10:00 | 23.88 | 24 | 22.21 | 23.90 | 23.51 |
| 10:30 | 23.97 | 23.53 | 22.06 | 23.92 | 23.53 |
| 11:00 | 24.06 | 25.02 | 21.92 | 23.93 | 23.55 |
| 11:30 | 24.14 | 24.25 | 21.77 | 23.93 | 23.58 |
| 12:00 | 24.23 | 24.93 | 21.61 | 23.91 | 23.61 |
| 12:30 | 24.31 | 24.42 | 21.45 | 23.89 | 23.63 |
| 13:00 | 24.39 | 24.83 | 21.28 | 23.85 | 23.66 |
| 13:30 | 24.47 | 24.58 | 21.11 | 23.81 | 23.68 |
| 14:00 | 24.53 | 23.83 | 20.93 | 23.75 | 23.71 |
| 14:30 | 24.59 | 24.62 | 20.74 | 23.69 | 23.73 |
| 15:00 | 24.64 | 24.6 | 20.56 | 23.62 | 23.75 |
| 15:30 | 24.68 | 24.52 | 20.37 | 23.54 | 23.77 |
| 16:00 | 24.72 | 23.67 | 20.17 | 23.46 | 23.79 |

| | | | | | |
|---|---|---|---|---|---|
| 16:30 | 24.74 | 23.98 | 19.98 | 23.38 | 23.81 |
| 17:00 | 24.75 | 23.77 | 19.78 | 23.30 | 23.82 |
| 17:30 | 24.75 | 23.99 | 19.59 | 23.22 | 23.83 |
| 18:00 | 24.73 | 25.3 | 19.40 | 23.14 | 23.84 |
| 18:30 | 24.71 | 24.65 | 19.22 | 23.06 | 23.85 |
| 19:00 | 24.68 | 23.97 | 19.04 | 22.99 | 23.85 |
| 19:30 | 24.64 | 24.62 | 18.86 | 22.93 | 23.86 |
| 20:00 | 24.59 | 24.1 | 18.69 | 22.87 | 23.86 |
| 20:30 | 24.54 | 24.49 | 18.54 | 22.83 | 23.86 |
| 21:00 | 24.47 | 24 | 18.39 | 22.79 | 23.86 |
| 21:30 | 24.41 | 24.35 | 18.25 | 22.76 | 23.85 |
| 22:00 | 24.33 | 24.2 | 18.12 | 22.75 | 23.85 |
| 22:30 | 24.26 | 24.12 | 18.00 | 22.74 | 23.84 |
| 23:00 | 24.19 | 24 | 17.89 | 22.75 | 23.84 |
| 23:30 | 24.11 | 23.99 | 17.80 | 22.77 | 23.83 |
| 00:00 | 24.04 | 23.9 | 17.71 | 22.80 | 23.83 |
| 00:30 | 23.97 | 23.92 | 17.64 | 22.85 | 23.82 |
| 01:00 | 23.91 | 23.9 | 17.58 | 22.90 | 23.82 |
| 01:30 | 23.85 | 23.85 | 17.52 | 22.97 | 23.81 |
| 02:00 | 23.8 | 23.5 | 17.48 | 23.04 | 23.81 |
| 02:30 | 23.75 | 23.8 | 17.44 | 23.12 | 23.81 |
| 03:00 | 23.72 | 23.2 | 17.41 | 23.21 | 23.81 |
| 03:30 | 23.7 | 23.4 | 17.39 | 23.30 | 23.81 |
| 04:00 | 23.68 | 23.59 | 17.38 | 23.39 | 23.82 |

| | | | | | |
|---|---|---|---|---|---|
| 04:30 | 23.68 | 23.58 | 17.36 | 23.49 | 23.83 |
| 05:00 | 23.69 | 23.5 | 17.35 | 23.59 | 23.84 |
| 05:30 | 23.71 | 23.64 | 17.34 | 23.69 | 23.85 |

## Appendix D: Temperature Data for Early Wet Season (September)

| Local Time (hrs) | Storage Chamber Temperature $T_{ch}$ ($^oC$) | | Temperature of the Cooler Sections ($^oC$) | | |
|---|---|---|---|---|---|
| | $T_N$ | $T_E$ | $T_{opm}$ | $T_i$ | $T_{ipm}$ |
| 06:00 | 23.97 | 23.6 | 23.97 | 23.97 | 23.97 |
| 06:30 | 24.01 | 23.22 | 23.77 | 24.06 | 23.97 |
| 07:00 | 24.05 | 23.57 | 23.60 | 24.14 | 23.98 |
| 07:30 | 24.11 | 23.48 | 23.45 | 24.21 | 23.98 |
| 08:00 | 24.18 | 23.53 | 23.30 | 24.27 | 24.00 |
| 08:30 | 24.26 | 23.62 | 23.16 | 24.33 | 24.01 |
| 09:00 | 24.34 | 24.83 | 23.02 | 24.37 | 24.03 |
| 09:30 | 24.43 | 24.00 | 22.88 | 24.40 | 24.05 |
| 10:00 | 24.53 | 24.00 | 22.73 | 24.42 | 24.07 |
| 10:30 | 24.62 | 23.53 | 22.58 | 24.43 | 24.10 |
| 11:00 | 24.72 | 25.02 | 22.41 | 24.42 | 24.12 |
| 11:30 | 24.82 | 24.25 | 22.25 | 24.39 | 24.15 |
| 12:00 | 24.92 | 24.93 | 22.07 | 24.36 | 24.17 |
| 12:30 | 25.02 | 24.42 | 21.89 | 24.31 | 24.20 |
| 13:00 | 25.11 | 24.83 | 21.69 | 24.24 | 24.22 |
| 13:30 | 25.19 | 24.58 | 21.49 | 24.17 | 24.25 |
| 14:00 | 25.26 | 23.83 | 21.29 | 24.08 | 24.27 |
| 14:30 | 25.33 | 24.62 | 21.07 | 23.98 | 24.29 |
| 15:00 | 25.38 | 24.6 | 20.85 | 23.88 | 24.31 |
| 15:30 | 25.43 | 24.52 | 20.63 | 23.77 | 24.33 |
| 16:00 | 25.46 | 23.67 | 20.40 | 23.65 | 24.34 |

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|---|---|---|---|---|---|
| 16:30 | 25.48 | 23.98 | 20.17 | 23.53 | 24.35 |
| 17:00 | 25.49 | 23.77 | 19.94 | 23.41 | 24.36 |
| 17:30 | 25.48 | 23.99 | 19.71 | 23.29 | 24.37 |
| 18:00 | 25.46 | 25.3 | 19.48 | 23.17 | 24.37 |
| 18:30 | 25.43 | 24.65 | 19.26 | 3.06 | 24.37 |
| 19:00 | 25.39 | 23.97 | 19.04 | 22.95 | 24.37 |
| 19:30 | 25.33 | 24.62 | 18.83 | 22.84 | 24.37 |
| 20:00 | 25.27 | 24.1 | 18.62 | 22.75 | 24.36 |
| 20:30 | 25.2 | 24.49 | 18.43 | 22.67 | 24.35 |
| 21:00 | 25.12 | 24 | 18.24 | 22.59 | 24.34 |
| 21:30 | 25.03 | 24.35 | 18.07 | 22.53 | 24.32 |
| 22:00 | 24.94 | 24.2 | 17.90 | 22.48 | 24.31 |
| 22:30 | 24.85 | 24.12 | 17.75 | 22.45 | 24.29 |
| 23:00 | 24.76 | 24.00 | 17.61 | 22.42 | 24.27 |
| 23:30 | 24.66 | 23.99 | 17.48 | 22.41 | 24.26 |
| 00:00 | 24.57 | 23.9 | 17.36 | 22.42 | 24.24 |
| 00:30 | 24.48 | 23.97 | 17.25 | 22.43 | 24.22 |
| 01:00 | 24.4 | 23.9 | 17.16 | 22.46 | 24.20 |
| 01:30 | 24.32 | 23.85 | 17.07 | 22.49 | 24.19 |
| 02:00 | 24.25 | 23.5 | 17.00 | 22.54 | 24.17 |
| 02:30 | 24.19 | 23.8 | 16.93 | 22.60 | 24.16 |
| 03:00 | 24.14 | 23.2 | 16.87 | 22.66 | 24.15 |
| 03:30 | 24.1 | 23.4 | 16.82 | 22.73 | 24.14 |
| 04:00 | 24.08 | 23.59 | 16.77 | 22.80 | 24.13 |

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|---|---|---|---|---|---|
| 04:30 | 24.06 | 23.58 | 16.72 | 22.87 | 24.12 |
| 05:00 | 24.06 | 23.5 | 16.68 | 22.94 | 24.12 |
| 05:30 | 24.08 | 23.64 | 16.64 | 23.01 | 24.12 |