EXPERIMENTAL STUDY OF PASSIVE COOLING OF A BUILDING USING LONG-WAVE NIGHT SKY RADIATION IN OWERRI, NIGERIA.

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CERTIFICATION

This is to certify that this project work, "**EXPERIMENTAL STUDY OF PASSIVE COOLING OF A BUILDING USING LONG-WAVE NIGHT SKY RADIATION IN OWERRI, NIGERIA.**" is a research work carried out by Engr. OKORONKWO CHUKWUNENYE ANTHONY Reg No 20074748238 a postgraduate student in the Department of Mechanical Engineering, Federal University of Technology, Owerri.

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DEDICATION

This work is dedicated to God almighty who has been my helper.

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First and foremost, I am most grateful to God almighty for the inspiration, knowledge and wisdom to write this thesis .I would like to appreciate the positive contributions of my supervisors: Engr Prof E.E Anyanwu, Engr Prof D.C.Onyejekwe and Engr Dr N.V Ogueke. I am very much indebted to them for their constructive criticism of my work which led to various changes and modifications of the work.

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A area (m ^{2})

B width (m)

C_p specific heat (J/kg K) D gap distance (m) d diameter (m)

F pump power factor

h convective heat transfer coefficient (W m $^{-2}$ K^{-1})

H height, (m)

K thermal conductivity (W m K^{-1})

L length (m) m mass (kg)

. mass flow rate (kg s $^{-1}$) q Heat flux (Wm $^{-2}$) Q heat transfer, W

R atmosphere or sky monochromatic radiation intensity

T temperature (°C or K) V velocity, m/s
σ Stefan–Boltzmann constant (5.67051 x 10-^{8}),W/m^{2} K^{4}

Subscript amb ambient conv convection
rad radiation *w water*
wind wind sky sky
i initial
f final
Greek symbols α monochromatic directional absorptance
μ Viscosity ε emissivity ρ
density(kg m $^{-3}$) β coefficient of thermal
expansion η efficiency

ABSTRACT

An experimental study of the passive cooling of a building using longwave night sky radiation in Owerri, Nigeria is presented. The experimental rig consists of a standard (3.0 x 3.0 x 2.5m) room, flat plate sky radiator, heat exchanger, storage tank, water pump, interconnecting pipes and another standard room to act as control for the experiment**.** This test rig is a rectangular building made from 15.24cm hollow blocks. It has two windows located in the east and the south facing walls respectively and an access door located directly opposite the south facing window. The roof is pitched at 12° to maximize the cooling potential of the building. Cold water from the radiator enters the heat exchanger through the inlet channel and fills its rectangular chamber. Heat from the room is transferred to the heat exchanger through convective heat exchange between the indoor air and the heat exchanger. This is made possible by the fact that cold air from the surface of the heat exchanger displaces the warm air in the room. During the daytime, the

cooled water stored in the storage tank is circulated between the heat exchanger and the storage tank, while the radiator is isolated with the aid of a control valve. Series of tests were conducted under the meteorological condition of the Federal University Of Technology, Owerri, Nigeria for the period spanning 13March 2010 to 30 November, 2010 and in April 2011. The periods are within the major climatic seasons of Nigeria. The results obtained showed that minimum water temperatures ranging over $21^{o}C – 23^{0}C$ were obtained through night sky cooling in Owerri. During the day, the cooled water was passed through the heat exchanger to cool the space. The results obtained for the daytime cooling yielded a maximum temperature depression of $3.5^{o}C$ below the recorded average room temperature of $28^{o}C$.This value translates to 1428kJ of useful cooling. The above results revealed a great promise for use of passive cooling technique in buildings in our tropical region. Application potentials for nocturnal cooling also exist for the storage of agricultural products like fruits, vegetable etc.

CHAPTER ONE
INTRODUCTION

Radiation cooling is an old idea which has recently gained popularity all over the world because it offers the potential to reduce cooling energy consumption and when coupled with building thermal mass, reduce peak cooling loads **Angelikl (2006).** Therefore, it is not surprising that the implementation of radiation cooling systems in commercial buildings in most civilized countries is currently under way. In the tropical regions like Nigeria, where grid electricity is not regular and in most cases not available, the need to explore alternative means becomes imperative and very urgent. Thus, the abundant solar radiant energy could be utilized in the area of electricity production, solar water heating and radiative passive cooling of buildings etc.

Radiative cooling of a surface of a radiator exposed to the atmosphere at night can be used to lower the temperature of the fluid in the radiator. Fluid, whose temperature have been lowered in such a procedure, could then potentially be used for space cooling in buildings, preservation of agricultural product, cooling of equipment and facilities which generates heat under the ground, as well as in other situations where the application of a cooling process is necessary or desirable **Boom-long(1981).**

1.1.0 Nocturnal Radiative Cooling

1.1.1 General Concept Descriptions`

The concept of radiative cooling is based on the heat loss by long-wave radiation emission from a body towards another body of lower temperature, (sky which plays the role of a heat sink) thus, resulting in an appreciable temperature reduction of the hotter body **Erel and Etzion(2000)**. This phenomenon uses the fact that the thermal energy emitted by a clear sky in the „„,window region‘‘‘‘ (8–13

μm), is much less than the thermal energy emitted by a blackbody at ground air temperature in this wavelength range. Hence, a surface on the earth facing the sky experiences an imbalance of outgoing and incoming thermal radiation and cools below the ambient air temperature. This is illustrated in Fig. 1.1.

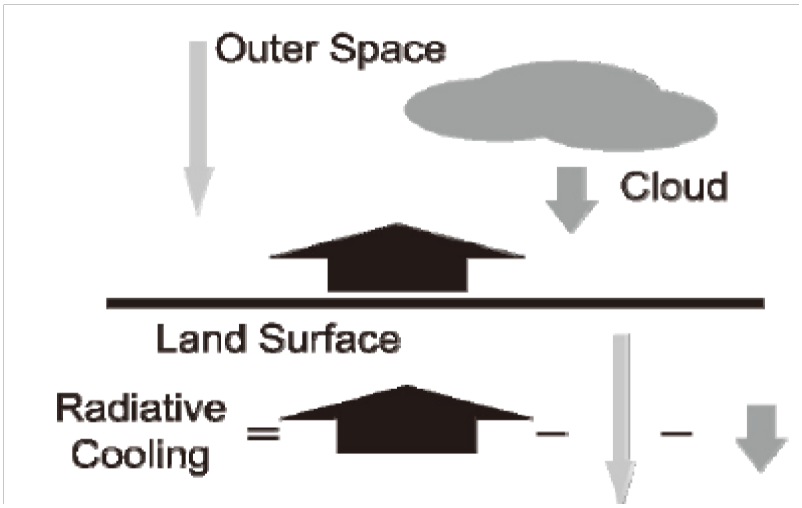


Fig 1.1 *Illustration of Simple Radiative Cooling*

In the case of buildings, the cooled body is the building surface and the heat sink is the sky, since during the night, the sky temperature is lower, than the temperatures of most of the objects upon earth, it is very possible that objects facing the night sky will lose its heat to the sky while its temperature drops. When this occurs, it becomes feasible to utilize this phenomenon for space cooling and water cooling at night. The cooled water can be circulated during the day to provide thermal comfort for the occupant of the building.

1.1.2 Physical and Technical Descriptions of Radiative Nocturnal Cooling

Theoretically, every object emits energy by electromagnetic radiation. This radiation is due to the molecular and atomic agitation associated with the internal energy of the material, which in the equilibrium state is proportional to the absolute temperature of the material measured in degrees Kelvin. This radiation is called thermal radiation. The major part of it is emitted within a narrow band of the electromagnetic spectrum - between 0.1 and 100 μm.

Thus an "ideal body" is the one that absorbs all the incident radiation impinging on it, for all wavelengths and all angles of incidence of radiation, and then emits radiation at one defined wavelength, which depends on its temperature. The emission from real bodies is evaluated relative to the emission of the black body under the same conditions, using a coefficient called emissivity ε**. Erel and Etzion (1996).**

More so, radiative cooling of an object is based on the physical principle of thermal equilibrium. If two bodies of different temperatures are facing one another without any medium between them, a net radiant heat flux from the hotter body will occur. This is illustrated in Fig. 1. 2.

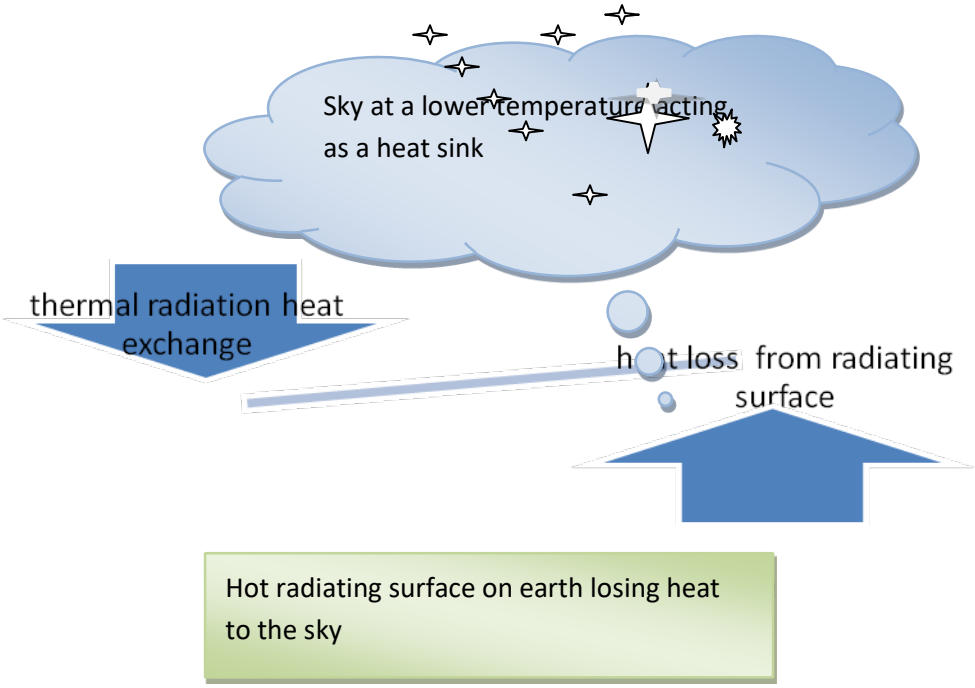


Fig 1. 2 showing how surfaces losses heat to the night sky to establish thermal equilibrium.

If the cooler element is kept at a fixed temperature, the other element will cool down to reach equilibrium. In the context of radiative cooling, the building envelope represents the body with the higher temperature; while the sky is the body with lower temperature. Without the atmosphere, the sky would be close to an ideal black body, absorbing all thermal radiation from the Earth.

The atmosphere, due to its specific composition, emits thermal radiation at all wavelengths except in the spectral region of 8-13 µm in the "atmospheric window". The atmospheric window which is often referred to as "the infrared atmospheric window" is a path from the land-sea surface of the earth to space. It is the overall dynamic property of the earth's atmosphere, that allows some infrared radiation from the cloud tops and land-sea surface pass directly to space without intermediate absorption and re-emission, and thus without heating the atmosphere . The spectral composition of window radiation varies greatly with varying local environmental conditions, such as water vapour content and land-sea surface temperature. The infrared absorptions of the principal natural greenhouse gases are mostly in two ranges **Goody and Yung (1989).** At wavelengths longer than 14 µm micrometers, gases such as CO_2 and CH_4 are absorbed due to the presence of relatively long C-H and carbonyl bonds, as well as water (H_2O) vapor absorbing in rotation modes. The bonds of H_2O and NH_3 absorbs at wavelengths shorter than 8 µm. This region is very close to the emitted wavelength of a black body at a temperature equal to the dry-bulb temperature of the air close to the ground. The observed variations from the black-body radiation to the atmosphere are attributed to the sky having its own wavelength dependent emissivity, known as the *skyemissivity.*

This emissivity is mainly due to the molecules of water vapour, CO_2, and ozone. The molecules of O_2 and N_2, which compose 99% of the atmosphere, are

transparent to infrared radiation (beyond 3μ m) - water vapour, CO2, and ozone have only few transitions in the spectral region of 8-13 μm **Argiriou et al(1994).** Since the atmospheric window will be closed by water vapour in the atmosphere, its "depth" depends on the weather and varies from day to day, as well as from one geographical location to another. Thus if an object on the earth‟s surface emits thermal radiation within the atmospheric window and the atmospheric conditions are such that the atmospheric window is open (low relative humidity and clear sky), then the object‟s temperature decreases.

1.2.0 Principles of Nocturnal Passive Cooling Techniques

Radiative cooling is a passive cooling method and passive cooling techniques in buildings have proven to be extremely effective and can greatly contribute in decreasing the cooling load of buildings. Efficient passive systems and techniques have been designed and tested. Passive cooling has also been shown to provide excellent thermal comfort and indoor air quality, together with very low energy consumption.

1.2.1 Classification of Passive Cooling Techniques

Passive cooling techniques can be classified in three main categories **Santamouris and Asimakopoulos (1996)**

1.2.1.1 **Solar and Heat Protection Techniques.**

Protection from solar and heat gains may involve: Landscaping, and the use of outdoor and semi-outdoor spaces, building form, layout and external finishing, solar control and shading of building surfaces, thermal insulation, control of internal gains, and use of vegetation to provide shades around buildings.

1.2.1.2 Heat Modulation Techniques.

Modulation of heat gain deals with the thermal storage capacity of the building structure. This strategy provides attenuation of peaks in cooling load and modulation of internal temperature with heat discharge at a later time. The cycle of heat storage and discharge must be combined with means of heat dissipation, like night ventilation, so that the discharge phase does not add to overheating.

1.2.1.3 Heat Dissipation Techniques.

These techniques deal with the potential for disposal of excess heat of the building to an environmental sink of lower temperature. Dissipation of the excess heat depends on two main conditions:

(i) The availability of an appropriate environmental heat sink.

(ii) The establishment of an appropriate thermal coupling between the building and the sink as well as sufficient temperature differences for the transfer of heat. The main processes of heat dissipation techniques are: ground cooling based on the use of the soil, and convective and evaporative cooling using the air as the sink, as well as water and radiative cooling using the sky as the heat sink. The potential of heat dissipation techniques strongly depend on climatic conditions. When heat transfer is assisted by mechanical devices, the techniques are known as hybrid cooling, **Martin (2000).**

1.3 Heat Dissipation and Hybrid Cooling

As previously mentioned, heat dissipation techniques are based on the transfer of a buildings‟ excess heat to a low temperature environmental sink. Main sinks are the ambient air, water, the ground and the sky. When heat is dissipated to the ambient air, the technique is known as convective cooling; when water is used the process is known as evaporative cooling; when the ground or the sky are the sinks, the techniques are known as ground and radiative cooling respectively.

Two distinct modes of radiative cooling are known for buildings. The first one is called *direct* or *passive radiative cooling* where the building envelope radiates towards the sky and gets cooler, thus enhancing the heat transfer out of the interior of the building. Although the radiant heat loss takes place day and night, the radiant balance is only negative during the night. During daytime, the absorbed solar radiation counteracts the cooling effect of the long wave emission and produces a net radiant heat gain. For the application of direct radiative cooling the position of the insulation layer (roof) has to be considered – in general the application of the direct method has to be initially respected in the construction concept of a building (e.g. insulated upper ceiling instead of insulated roof).

The second mode is called *hybrid radiative cooling*. In this case, a metal sheet is the radiator, which can be the roof of the building itself. In the cooling process, air or water is circulated under or in the radiator before it enters the building once again to cool it down with slab or ceiling cooling.

1.4.0 Radiating Surfaces

Both types of radiative cooling systems mentioned above, uses radiating surfaces to achieve cooling. Therefore, for cooling purposes, the ideal radiating surfaces should have emissivity, $\varepsilon_s = 1.0$ in the 8 μ m – 14 μ m atmospheric window, and zero elsewhere. **Santamouris and Asimakopoulos (1996).**

Considering a radiative exchange between an ideal radiator and the atmosphere, if it is assumed that only the primary atmospheric window is present, then the radiator will derive maximum advantage from this window, since it will radiate only in this wavelength range, and will not absorb any radiation from the atmosphere outside this range. Its equilibrium temperature will therefore be lower than that of the atmosphere. This temperature will, in fact, be that at which the radiator‟s emitted radiation in the 8-14 μm band; is equal to the amount of radiation

it absorbs from the atmosphere in the same band since no radiative exchange occurs outside this band.

A black surface will be less satisfactory for radiative cooling than an ideal surface, since a black surface will absorb atmospheric radiation outside the 8 – 14 μ m band as well. Therefore, its equilibrium temperature will be higher than that of the ideal radiator, **Granqvistc (1984)**. However, if the 17 – 22 μ m secondary windows are present, a black surface can radiate in this wavelength range, whereas the so-called ideal radiator cannot radiate at this wavelength range.

1.4.1 Radiator Materials:

Many spectrally-selective surfaces have been developed to maximize radiative cooling in the atmospheric window. One of the first, developed by **Catalanottic et al (1975)** , was aluminized polyvinyl fluoride plastic (PVF). Other materials subsequently developed include aluminum coated with 1.3 μ m of silicon oxy-nitride ($S.O_{0\ 6}NO_{.2}$), MgO and LiF on metal foils **Harrison and Walto (1978)**, and many others. Commercial white paint with high titanium dioxide (TiO_2) contents is also considered as a good radiator surface, although not very spectrally selective.

Gases also have been identified as potential radiators for cooling purposes **Boon-long (1981)**. The advantages of gaseous radiators have been cited as being relatively cheap, easy to transport, may be used as coolant medium directly, and also having the ability to be mixed together to obtain desired characteristics. Examples of gaseous radiators are ethylene (C_2H_4), ethylene oxide (C_2H_4O), and ammonia (NH_3).

1.4.3 Radiator Cover:

In order to reduce non-radiative heat gain into the radiator, in addition to side and back insulation, the radiator surface is usually covered with an infrared transparent screen, leaving a smaller air gap. The most common material used for radiator cover is polyethylene film. The stagnant air layer trapped between the radiation surface and the screen serves as a thermal resistance. Theoretically, no convection current will be present, where the air layer is in a stable configuration (colder lower boundary and warmer upper boundary). Therefore, the air layer becomes a very effective insulation. Nevertheless, in practice, conduction currents usually develop in the larger radiators due to end induced fluttering of the flexible cover, **Landro and Mccormick (1980)**, and significantly reduce the conductive insulation effect. So, the cover needs to be re-enforced for increased rigidity. **Nilsson et al (1985)** reported results from V-corrugated high-density polyethylene foils, design to reduce convective effects even in the presence of winds.

1.5.0 Physical Limiting Factors Affecting Nocturnal Cooling Activities

Several physical and ambient conditions influence or limit the radiative cooling process. They include the following: cloudiness, moisture and humidity, view factor, dew formation etc.

1.5.1 Cloudiness

As already noted, water vapour influences the emissivity. A cloud covered sky raises the emissivity and "narrows the atmospheric window" of the sky, so that the net radiating flux of the radiator towards the sky decreases – a cloudy sky has about 45% of the sky radiation potential of a clear sky. Clouds enhance the atmospheric radiation, and reduce the efficiency of radiative cooling. See fig 1.3

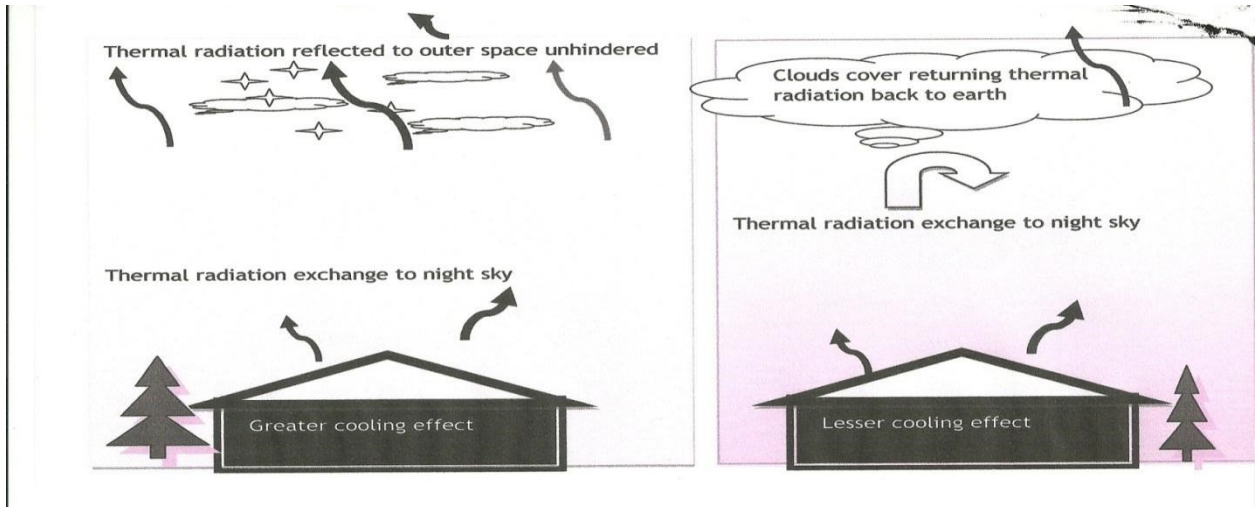


Fig 1.3 the effect of cloud formation on nocturnal cooling.

1.5.2 Humidity / Moist atmospheric conditions

Humidity plays a major role in radiative nocturnal cooling – humid locations see a higher effective sky emissivity and thus less radiative cooling potential. Often cloudiness and moist atmospheric conditions go together.

1.5.3.1 Convection / Wind

Night cooling can be strongly affected by winds. As the temperature of building surfaces fall below the ambient air temperature, they begin to gain heat by air convection. Still air and windless conditions produces the least heat added to exposed surface. Anything that reduces wind speeds around the radiating surfaces (without blocking the sky view) will help increase nocturnal radiation. Blowing winds with air temperatures higher than the radiator can decrease the cooling potential by up to 50%, **Santamouris and Asimakopoulos (1996).**

1.5.4 View factor

The extent to which a roof sees the sky for radiative heat exchange depends on the relative angle of the emitter (roof) and the absorber (night sky). Generally, horizontal surfaces will have the greatest exposure to the sky for the purpose of radiation, while those vertically inclined will not. The factor describing this is the view-factor, which is related to the geometry of a body relative to the sky. A simple rule of thumb is that a horizontal roof has a view factor equal to one, while a vertical wall with an unobstructed view will have a view factor of one third.

1.5.5. Dew Formation:

In more humid climates, where the dew point temperature of the ambient air is relatively high, it is possible that through radiative cooling, the temperatures of the radiator and/or the infrared transparent cover would fall below the ambient dew point. In such cases, dew will fall on the surfaces. Since water has a high absorptivity in the infrared, this dew film will reduce the net radiant cooling rate of the radiator. This dew formation is a self – accelerating process; the water film on the cover becomes the radiating surface itself, and due to its small thermal mass, cools rapidly, which causes more condensation on the surface, **Idso. (1981)**.
The cooling performance of the radiator is then drastically reduced. This phenomenon may render the use of selectively radiating surfaces ineffective in humid climates.

1.6 Statement of Problem

The cooling demands for cold production in recent years have increased with its high cost. During the past decades, the heat island effect and the consequential increase of temperature raised a high demand on cooling techniques. The problem of overheating is rather intense especially in Africa. The desire for a solution led to the widespread use of air conditioning in residential and commercial buildings. **Erel and Etzion (2000)** reported that the global sale of air conditioning equipment is still increasing from 39 million units in 1999 to 58 million forecasted in 2007. The European Union is about 7 % of the global market by number of units, with sales growing from about 3 million to about 5 million units. The major markets within Europe are the southern countries with Italy, Greece, and Spain accounting for about 75% of sales by number of units **Angelikl (2006).** The impact of usage of this air conditioning equipment on electricity demands is a serious problem for many developed countries. Peak electricity loads oblige utilities to build power

plants in order to satisfy the demand, thus increasing the average cost of electricity. The sales of air conditioning equipment in Nigeria and especially the entire African continent significantly increased the energy consumption, leading to a growing weakness of the countries to satisfy their energy needs.

Within the framework of achieving a more energy efficient built environment, many countries worldwide have started a broad discussion about issues like: the impact of urbanization on the cooling demand of buildings, the impact of temperature increase of cities on the cooling load of urban buildings, the problem of appropriate climatic data for design, the increased peak electricity load as well as the decreased efficiency of air conditioning equipment. As a result of these new phenomena, a great part of research and practice in the construction area is concentrated nowadays on techniques and methods that manage to reduce or even eliminate the cooling loads of a building as well as on their energy conservation potential. In the direction of achieving energy efficient built environment an increasing number of countries considered as an attractive approach the Energy Performance (EP) standardization and regulation. Several countries have already enacted such EP based regulations or are preparing one **Wouters (2004).**

For the above-mentioned reasons addressing successful solutions to counterbalance the energy and environmental effects of air conditioning is a strong requirement for the future. Possible solutions involve the use of nocturnal cooling techniques. However, the theoretical background for nocturnal cooling technique is not fully developed for effective implementation of this concept in modern architectural design. In summary, the following initiated the present study:

- High cost of cooling of residential buildings and its economic implication on the living standard of the people.

- High energy demand of countries resulting in low economic growth to satisfy the energy needs.
- Environmental effect of using air conditioning systems is quite enormous due to the depletion of the ozone layer caused by the use of some refrigerants like CFC.

However, some systems have been designed for nocturnal cooling in other climates. Therefore, the need to develop a standardized nocturnal cooling system suitable for Nigerian climate motivated this study.

1.7 Justification of the Study

Increasing concerns about global warming and greenhouse gas emission present the energy sector with a challenge to cut its energy consumption. In countries such as the UK and the US, for instance, the building sector consumes of the order of 40-50% of the total delivered energy. Of this, climate control systems, namely ventilation, cooling and heating can account for as much as 70% of the total energy use, **Martin (2000)**. However, this component of the energy consumption can be reduced significantly by employing passive environmental solutions instead of mechanical ones: for example, a well-designed passively cooled building can consume only a third of the energy consumed by an airconditioned building, while providing a comparable level of comfort. This is because passive design allows buildings to adapt more appropriately to their local climates and take better advantage of natural energy resources, such as wind and thermal buoyancy, to help condition their interior environments. Furthermore, passive, radiative cooled buildings have potential to provide more pleasant and healthier environments for the occupants compared to their mechanically ventilated counterparts. This will go a long way to reduce the effect of the menacing global warning and climate change

that is prevalent in the world. A well designed nocturnal cooled system can cut the energy need of the building drastically. Secondly, passive cooling systems are environmentally friendly with low maintenance cost, **Santamouris and Asimakopoulos (1996)**

1.8 Objective of the Present Study

The main objective of the present work is to investigate the nocturnal cooling of a building under the prevailing climatic conditions in Owerri, Nigeria. Accordingly, the specific objectives to realize the above include:

- Design, fabrication and installation of a nocturnal cooling system in a building in Owerri.
- Extensive field experimentation on the system designed.
- Evaluation of the seasonal variation in nocturnal cooling pattern in Under the climatic conditions prevalent in Nigeria.
- Comparison of the results obtained with others from other climates

1.9 Scope of the Present Study

The scope of work to achieve the overall objective falls into three categories: design, construction and experimentation.

In the design aspect, the existing radiative cooling theories would provide the data for the design, taking into account the relevant technical and economic constraints.

The construction materials used were sourced locally and the fabrication was carried at the Auto-lab of mechanical engineering workshop in the Federal University of Technology, Owerri. In the experimental aspect, the following measurements were undertaken - temperature measurements at specific points on the radiator surfaces, storage tank, room temperature, relative humidity of air.

Experimental data analysis involves the following activities.

- Use of pertinent equation for data reduction.
- Performance evaluation of the system data compared with theoretical results.

This experience is expected to be useful for subsequent development of large and economic radiative cooling system for use especially in areas where grid electricity is mostly unavailable.

CHAPTER TWO

Literature Review

2.1 Types of Radiative Cooling Method

Solar heating and radiative cooling systems have been studied in particular for the last 30 years through analytical and experimentation studies.

Two methods of radiative cooling have been identified in the cooling of buildings. The first is known as direct, or *passive radiative cooling* where the building envelope radiates towards the sky and gets cooler, thus enhancing the heat transfer out of the interior of the building. The second method is called *hybrid radiative cooling*. In this case, a metal sheet on the roof of the building can serve as a radiator. In the cooling process, air or water is circulated under or in the radiator before it enters the building once again to cool it down, with slab or ceiling cooling **Santamouris and Asimakopoulos (1996)**.

In this section, comprehensive reviews of some of the previous works done on various radiative cooling are presented.

2.2 Direct radiative cooling system

Parker (2005) reported a direct radiative cooling system used for an innovative residential cooling system, which uses nocturnal night sky radiation from a roof integrated radiator. The operation of the system is detailed in Fig. 2.1. The theoretical concept of the system has been recently evaluated in a project report which evaluates how the system operates against various parameters and under typical Florida weather conditions. **Parker et al (2008)**. The system utilizes a sealed attic covered by a highly conductive metal roof as the sky radiator. This radiator was linked by air flow to the main zone with the attic zone to provide cooling largely during night time hours. The experimental set-up consists of two highly instrumented side-by side 3.05 x 4.9 m test sheds located at the Florida Solar Energy Center. One of the test sheds was configured like a conventional home with a dark shingle roof and insulated ceiling under a ventilated attic. The experimental shed features a white reflective roof on battens with a sealed attic where the air from the interior of the shed was linked to the sealed attic and roof radiator. Series of tests were conducted to determine the performance of the system under varying climatic conditions.

Results obtained show that on a clear desert night, a typical sky facing radiator at 27^0 C cools at about 75 W/m^2 under completely over cast skies . The temperature of the radiator surface often time records a temperature of about 5^0C below the ambient air temperature. However, various physical limitations (differential approach temperature, fan power, convection and conductance) constrained what could have been achieved in this study. The big problem with night sky radiation cooling concepts has been that they have typically required exotic building configurations.

Fig 2.1 Night cool integrated system Parker et al (2008)

Khedari et al (2000) reported the feasibility of cooling by using night radiation under Thailand"s hot and humid climate. The system used is a direct radiative cooling system. Four types of roof radiators were made by using common construction materials. They were examined under three sky conditions: clear, cloudy and rainy, sky conditions.

Estimates were made, based mainly on the temperature of the different surface coatings of the roof radiator. These experimental results showed that the depression of different surface temperatures is in the range of 1–6° C below ambient temperature under clear and cloudy skies. On the other hand, with rainy skies, the temperature of the different surfaces of roof radiators and ambient air was fairly close. Measurement of temperature of the air below the radiator surface revealed that for "Type A" which is made of aluminium coated with white pigment, was in the range of 1 – 3°C. The other types, though made of aluminium but with different surface coatings, were however shown to perform better. From their work they concluded that surface coating effects nocturnal cooling greatly.

In general, their work showed that cooling using night sky radiation is mainly feasible during tropical winter season and also that sky condition affects night radiation cooling. Apart from sky condition however, other factor which could affect night radiation cooling are emissivity of the material and condensation on the radiator surface area

Apart from the influences of meteorological condition on night sky radiation cooling systems, the major problem with many night sky radiation cooling concepts have been their typically exotic building configurations

Michel and Biggs (1975) investigated the cooling of small buildings at night by radiation loss to the sky by monitoring the thermal performance of two huts: one roofed with galvanized steel decking, painted white; which acts as a „black body" for wavelengths greater than 3 μ m. The other hut had aluminum decking to which aluminized „Tedlar" sheet had been glued, the „Tedlar" acting as a selective surface absorbing and radiating mainly in the 8–13 μm band was also considered.

The hut with the painted roof was cooled marginally better than that with the „Tedlar" covered roof. The results showed that useful cooling power of 22 W/m^2 was achieved at a roof temperature of 5°C, at an ambient 10°C, while the gross cooling power probably exceeded 29 W/m^2. Calculations based on a simple simulation of the sky radiation yielded an upper limit of 40 W/m^2 for the cooling power of the surfaces. This suggests that an ideally selective surface operating under the best possible clear-sky conditions has little advantage over a black body radiator, unless the temperature of the surfaces is significantly lower than that of the ambient air temperature.

In another work**, Hay and Yellow (1970),** also reported a system that utilizes the nocturnal cooling concept. In their report, they described a system designed for

an area in the south western United States. The radiator also serves as a solar collector for the purpose of heating during the day time. The heat transfer fluid was stored in black plastic bags that are supported in the roof structure, while movable insulation was provided over the roof structure. During the day, the insulator is drawn aside so that the water and plastic can absorb radiation for heating season. Similarly, during the night, they are equally drawn aside to allow for radiative cooling to the night sky. The problem with this system is that extra costs are incurred in the course of moving the insulators.

Furthermore, **Beckham and Duffie (2000),** reported a nocturnal cooling storage system which was first tried in 1955, and has been recently developed and tested in Australia. The system consists of a large bed of rocks cooled by drawing cool night sky radiation. During the day, the dry warm air may be cooled by circulating it through the rock bed. At night, air is drawn through the rock bed.

Hay and Yellott (1969) also designed a movable insulation heating and cooling system. This system consists of a roof covered with approximately 20 cm of water sealed in clear polyvinyl chloride (PVC) bags, which are connected to inflatable air cells above the water for increased collection efficiency. These bags which are covered intermittently by sliding polyethylene insulation panels, 5 cm thick, are exposed during day to the sun. The concept of this passive system is based on the assumption that a water bag on top of a roof shielded by a moveable curtain can act as collector storage and heat exchanger.

The immediate application of the above design was carried out by **Hay and Yellott (1978)**, in 1972. The system was applied for immediate use in a three bedroom apartment in Atascadero, California. The house had a split level slab floor, with room height 2.4 and 3 m respectively. The external walls, walls and floors

were intended to act as thermal capacitors, to help damp radiations in room temperature resulting from variations of the water temperature in the bags.

The same moveable insulation heating and cooling system was further tested in 1975, over 9 months with normal family occupancy **Beckham and Duffie, (2000)**. The performance test showed that if the system had operated during the test, it would have saved the equivalent of a 500 litres of oil per annum. However, it could be implied that, presumably, these test results are very questionable because the maintenance cost of the large number of movable points required, could be high, unless the panels were moved by the occupants instead of a mechanical system.

Experimental and theoretical natural direct cooling applications of water at night in un-insulated tank were investigated by **Ali (2007)**. The experimental test rig consists of an uncovered tank and a covered tank. Both tanks were exposed to nocturnal cooling. The results obtained showed that lower temperatures were recorded for the open tank in comparison with covered tank due to evaporation heat transfer. A total net cooling power of 45 W /m^2 was obtained .The evaporative and convective mass transfer cooling for the open tank ranged from 38.7% and 57.4 % of the total net cooling. The analytical results revealed a 5% deviation from the experimental ones.

In another analytical study, **Ben Cheikh et al (2004)** proposed a roof design to provide comfort condition in hot climates. The proposed roof is made of aluminum plate on the top of an air gap, then a rock and water bed. The ceiling is made of Concrete. The top surface of the aluminum plate is painted with white titanium-based pigment to increase reflection to the sky. A finite difference dynamic model was solved to predict the indoor temperature. Temperature is kept

at reasonable value inside the room due to the heat pipe effect that carries the heat outside. Results show that cooling inside building can be improved by the application of such a roof. It was noticed that the maximum indoor temperature occur at around sunset.

Schneider and Berger (1981) have also designed a nocturnal cooling system called the Diode roof as illustrated in Fig 2.2. The system is a closed one, in which vapor condensation is used to enhance external heat exchange, by a heat pipe effect. The system is designed for hot dry tropical climates and is composed of a flat corrugated iron roof covered with inflated plastic sacks, which are partially filled with pebbles and a small quantity of water. The upper external surface is coated with cold selective pan bag. At night, the temperature of the radiator falls below the ceiling temperature. Water vapor made the sacks condensation slide along the covered radiator and is being transfer through the heat pipe effect.

An experimental study carried out on this system using a cubic test cell 2.5 m each, with a small window in Nice, showed that diode roof has a lot of promise in providing some comfort. The results obtained gave a temperature of about 23^0 C which is 5^0 C below the ambient temperature.

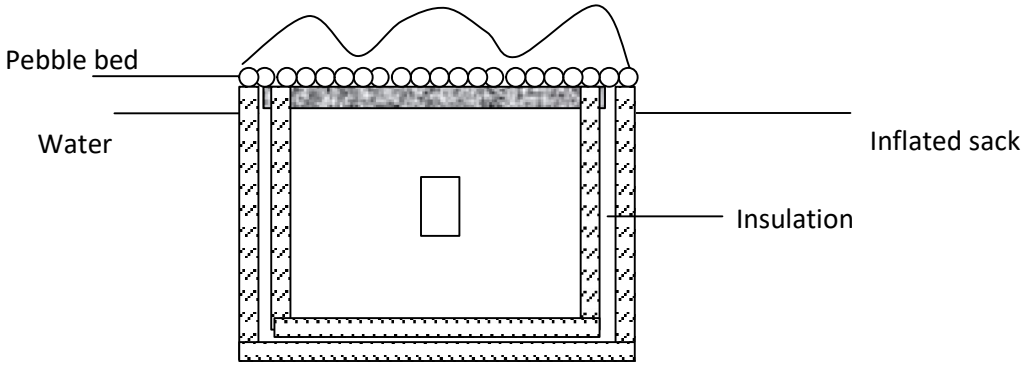


Fig 2.2 A diode roof experimental cell

A roof pond type of radiation cooler has also been built at Ha Makoe, Lesotho, by the Bethel Business and Community Centre. Fig 2.3 illustrates the principles of the roof pond. This system consists of plastic bags on a roof, filled with water. The water filled plastic bags are covered by a roof in the daytime, which at night times, it is moved away (on rollers) to expose the water to the nightsky.

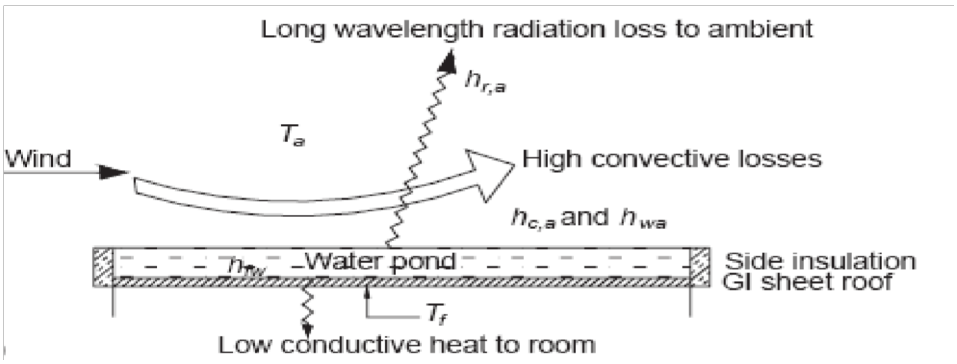


Fig 2.3 Illustration of the working principle of a roof pond

2.3 Hybrid Radiative Cooling Systems

Meir et al (2002) investigated the performance of a glazed polymer flat plate radiator with water as heat carriers and a large storage mass. This is illustrated in Fig 2.4. The system investigated is a modified solar heating system with polypropylene oxide and a twin walled parabolic collector as the major components.

The system utilizes water as the heat carrier and consists of a radiator roof, a reservoir, connecting piece, a pump and a control unit. The radiator is mounted on the roof of the building at a predetermined angle. The heat carrier is lifted by pump power to the upper part of the radiator. Driven by the force of gravity, the liquid trickled through the intrinsic channels of the radiators, releasing heat and returning to the reservoir.

The experiments carried out at the SO LAB, a small free standing outdoor test laboratory at the University of Oslo gave a minimum temperature of about 10^0 C for June 2 - 3 1999, at 3" o clock in the morning.

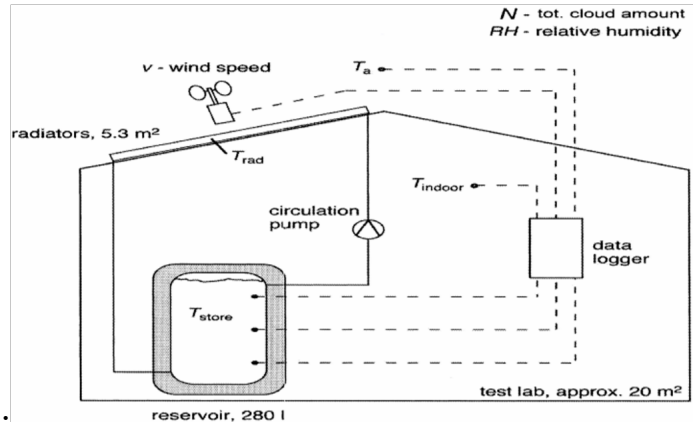


Fig 2.4 Schematic diagram of nocturnal cooling system using water as a heat carrier

This result clearly shows that nocturnal cooling took place since the ambient temperature was in the range of 15 – 20^0 C

Meir et al (2002) in a similar work, presented a concept for a combined system for solar heating and radiative cooling. The solar heating system is composed of a large and un-pressurized heat store, the solar collectors, an additional heating source, a heat distribution system in terms of floor heating and a controller unit. The solar loop and the floor heating loop are directly connected to the heat store, thus avoiding temperature gaps by heat exchangers and favoring a low temperature level. The floor tube system is connected directly to the cold store, and water with temperature of 2 - 3 ^{o} C below the desired room temperature circulates round the floor. The first small scale experiments revealed a potential for a low cost and simultaneously effective radiative cooling system.

These experiments were carried out in a small test house at the University of Oslo. The house was equipped with two radiative cooling systems mounted on the tilted roof. Preliminary results were obtained for the performance with and without

cover plate for the radiators; in order to study the impact of the flow rate in the radiator system. The radiator employed is a modular building element for roofs which is made of Polycarbonate twin-wall facades, based on a 60 cm module width and four standard lengths. The first series of measurements during clear sky conditions showed a stagnation temperature for the radiator 5 - 7 degrees below the air temperature during the night. A cooling power in the range of 40-60 W/m^2 was obtained. Though, the result obtained in both works show some promises in achieving comfort, however, they concluded that their system may not perform creditably in a hot humid climate.

Ito and Miura (1989) also investigated the performance of a radiator in a practical cooling system. In their study, they presented results for the experimental and analytical investigations on the thermal performance of a radiative cooling system for storing thermal energy. The radiative cooling system used consists of two radiator panels, a storage tank, a pump and a flow meter. Fig 2.5 illustrates the operation of the system. The heat transfer fluid circulates round the system with the aid of the pump. The flow rate was measured by a bearing-less flow meter. The liquid which was pumped out from the top of the storage tank flowed through the radiator and was pumped into the button of the storage tank. A 1960 mm long by 960 mm wide, flat plate type solar collector was used as the radiator panel. The panel was made of two spot welded plates of stainless steel material. Its effective cooling area was coated with the same thick black paint. The two radiator panels, whose backsides were insulated by 100 mm thick styrene foam, were arranged in a row on the shed at an angle of 30^0 to the horizontal. The storage tank, 250 mm inner diameter 1360 mm high, was made of vinyl chloride tube and had a volume of 66.7 litres. The upper part and the lower side of the storage tank were insulated by polyurethane foam of 150 mm thickness. A 300 W U shaped pipe heater, fixed at the bottom of the storage tank was used to set the initial temperature of the fluid in

the storage tank. Temperature was measured by thermocouples of 0.3 mm diameter at the inlet and outlet of the radiator, eight locations on one of the radiator panels, the inlet of the storage tank panels, and the inlet of the storage tank.

Results obtained showed that the cooling power of the black painted radiator panel at ambient temperature gave 40-60 W/m^2 in clear night in the summer and 60-80 W/m^2 in the fall. The fluid temperature in the tank for storage gave 2 – 5^0 C depression below the ambient temperature which is about 13 ^{o}C. The cooled water was later used for the cooling of a residential building.

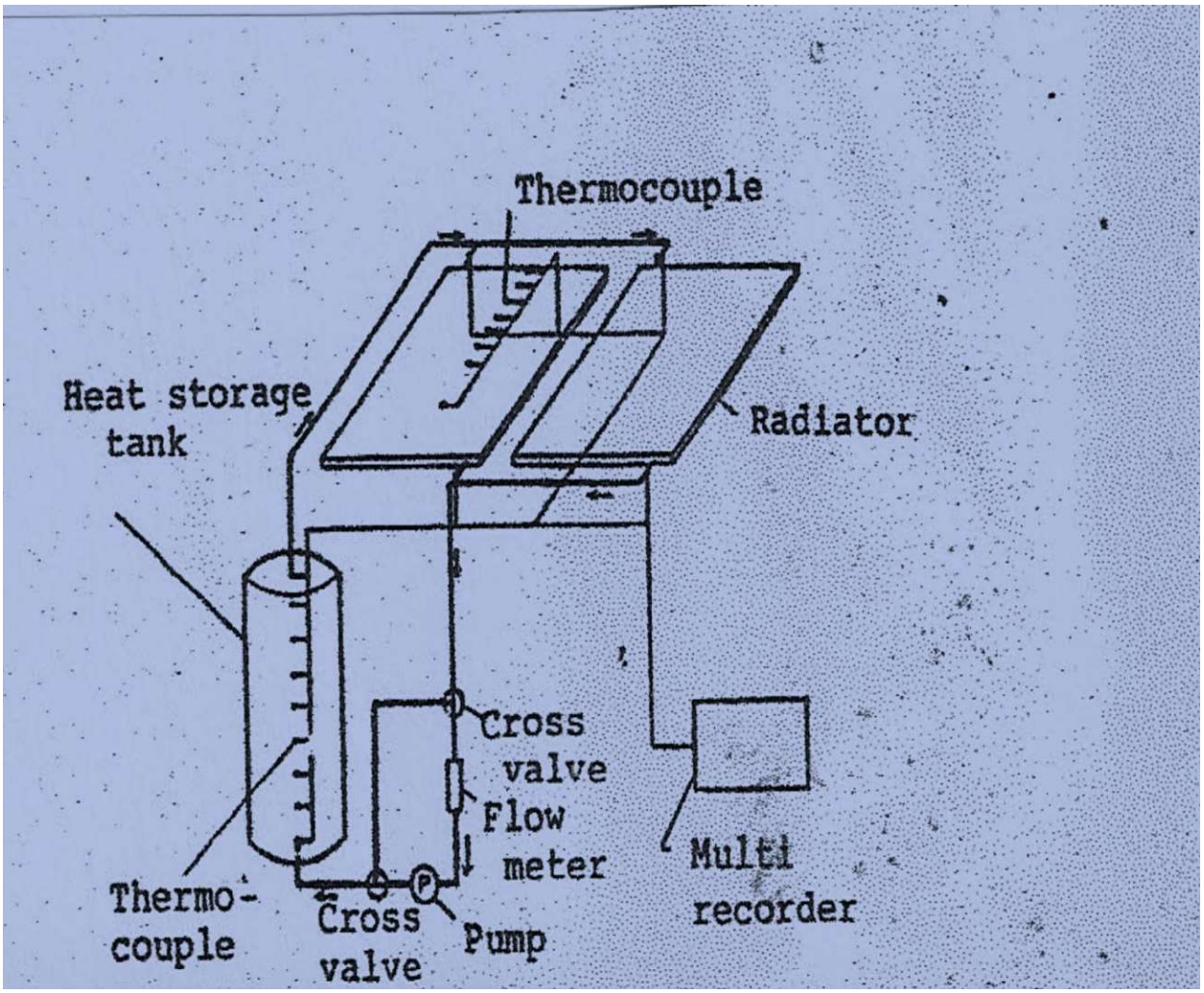


Fig 2.5 Schematic of the system by (Ito and Miura 1989)

The aforementioned works were carried out in temperate regions of the world. However, region along the tropics do experience weather conditions that are not favourable for radiative cooling due to high relative humidity. Nevertheless some efforts have been made by some researchers in these regions. Notable amongst them includes **Nammout and Kratsiriroral (2006), Santamouris and Asimakopoulos(1996).**

Nammout and Kratsiriroral (2006) developed a radiative nocturnal cooling system in Thailand. This system is illustrated in Figs. 2.6&2.7.

The system consists of a thermo-siphon heat pipe system whose condenser acts as a thermal radiator that transfers heat from its evaporator to the surrounding ambient as the radiator assist a set of thermo-siphon heat pipes made of 48 copper tubes each of 19.05 mm in diameter. Its condenser section of 6.3 m^2 acts as the radiator in the night time and its evaporators section are dipped in a well-insulated rectangular water storage tank of 1 m^3. The radiator is insulated on a 45^0 tilling roof of the tested room. Cool water in the storage tank is fed through six cooling coils each of 0.87 m^3 installed in a well-insulated room at the ceiling to extract artificial heat load obtained during the day time from a set of electrical heater. The room for the experiment has a dimension of 3.0m x 3.0m x 2.5m. The results obtained from the sets of experiments showed that a temperature depression of about 1.8^0 C below the ambient was achieved. This clearly indicates that the performance of the system is relatively low compared to the results obtained in **Meir et al, (2002) and Ito and Miura (1989).**

Fig 2.6 Experimental Test facility of Nammount and Kratsiriroral (,2006)

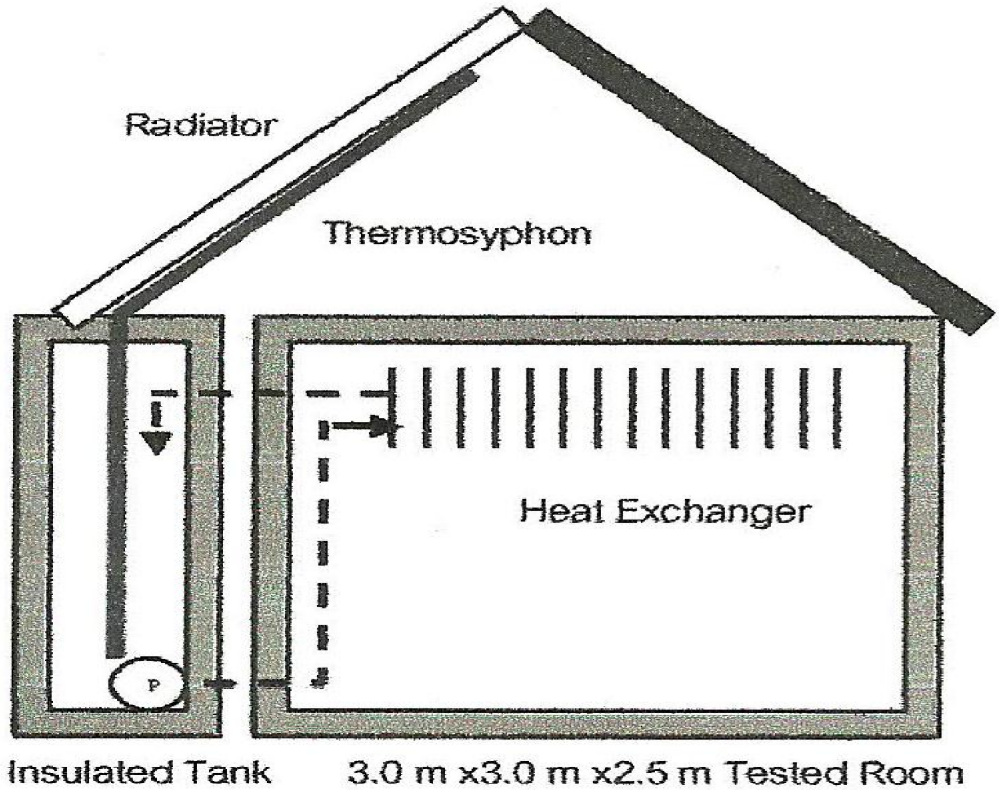


Fig 2.7 Schematic of the system described by *Nammount and Kratsiriroral (,2006)*

Bassindowa et al (2007) reported an experimental investigation on night radiative cooling in Jeddah, Saudi Arabia. Three sets of experiments were carried;

the first experiment was made to investigate the temperature history of two plates one of them in horizontal position and the other inclined at 45° with the horizontal and facing north. Both plates were painted black. In the second experiment, a plate was covered with a plastic sheet of 100 μm thickness, and insulated from the back; the third experiment was designed to test the effectiveness of radiative cooling to cool a shelter. The experimental test rig for the third experiment consists of two shelters of same size. The dimensions of each are 120 cm by 175 cm with a roughly 215 cm height. The roof of each shelter is inclined 10° with the horizontal and facing north. Figure 2.8 shows the two shelters. Shelter A is equipped with a 1500 liter cylindrical water tank. The tank is connected to a radiator on the roof of the shelter, and it is used to store cold water during the night. The radiator is fitted with 12.5 mm pipes. Piping is made between the top of the tank and the top part of the radiator. Water flows from the bottom of the radiator to the top of the tank. A provision was also made to fill the system with water. The wall of each shelter is made of two sheets of 3 mm plywood with 5 cm glass wool insulation sandwiched between them. The roof has two layers of 5 cm insulation. Figure 2.9 shows shelter A along with the radiator, the indoor tank, and the piping connecting the two. Thermocouple locations were also shown.

During the night time the water circulates naturally between the radiator and the tank. Therefore it could be inferred that the water will be cooled in the tank during night time. At the start of the day when the loads on the shelter increases, it was anticipated that natural convection between the tank and the indoor air will keep the indoor temperature at reasonable values. Shelter B is identical to shelter A in shape and size, and it serves as a reference. The indoor temperatures inside the shelters were also measured and recorded. The results obtained showed that for the second experiment, during the day, of temperature of the inclined plate released a temperature cover than the ambient and a comparable depression of about 4^{0} C

below ambient now released for the both plates. In this experiment, the effect of connective heat transfer was examined. They were done by covering the plate with thin plastic film of about 100 mm thickness. The effect of the convective heat transfer could be seen for the result obtained which gives a 5^0 C below ambient. It could be inferred that convection heat transfer inhibits nocturnal cooling.

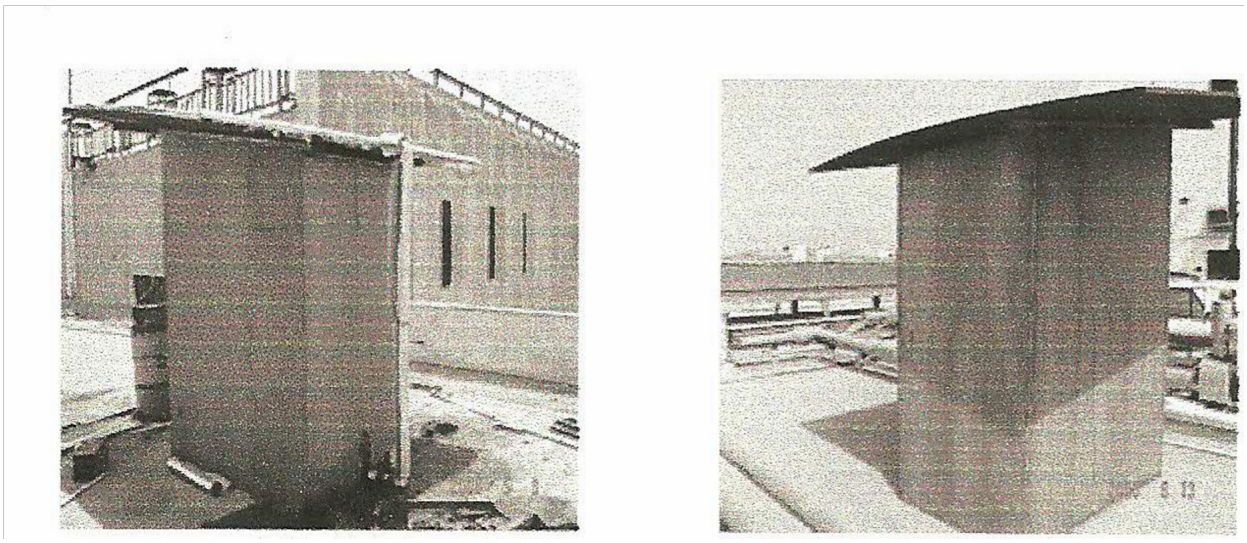


Fig 2.8 Experimental test rig by Bassindowa et al(2007)

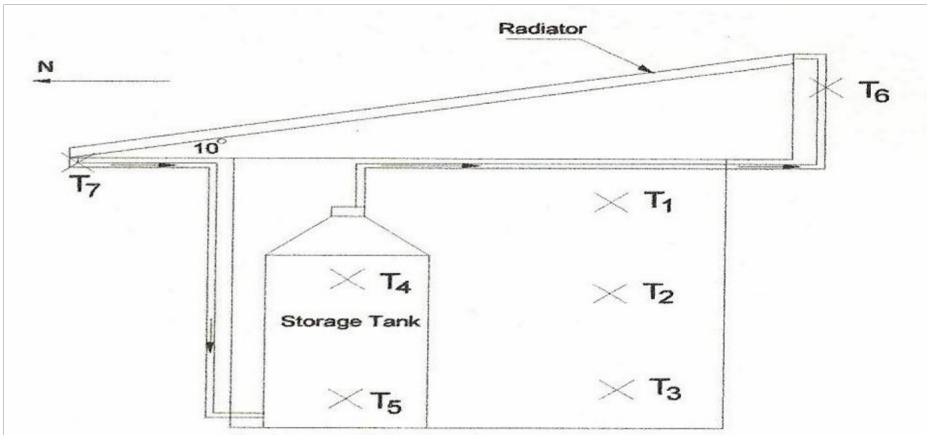


Fig 2.9 Schematic of the radiative cooling system

The heating performance of a radiation cooling system capable of a mean nightly cooling rate of 80 W/m^2 over an 8-hour operating period was experimentally investigated by **Erell and Etzion(1991)** . The system is made of a single 2.2 x 1.3 m commercially available flat plate solar water heater, which was

later modified for night time radiative cooling. The modified version has the glazing removed and a water pond put in place as part of the roof for thermal storage. The experimental rig was tested for a period of twelve hours during the night. The result obtained gave a temperature depression of 1.8^{o}C below the ambient condition.

Another radiation cooling system has been theoretically and experimentally investigated **by Al Nimr, et al (1998).** The radiating panel was made of 1,500 x 400 mm mild steel plates, with a 40 µm polyethylene cover and rock-wool backinsulation and a pump circulating water to a 120 litre water storage tank. The results showed that the radiation panel was able to reduce the temperature of the water by 15 °C under spring weather conditions. The experimental result was later validated with the analytical result. The result obtained showed a good agreement between the theoretical and experimental results as should be expected.

Al-Nimr et al (1999) also designed and constructed an experimental set up in Irbid-Jordan, to study the effectiveness of radiative cooling in cooling water during the night. The setup consisted of a black-painted steel radiator panel, a storage tank, and a pump. A dynamic mathematical model was also developed for the system. The model was later validated with the experimental results. The result obtained by comparison between the theoretical and experimental model showed good agreement with a 2% deviation under steady state conditions. This mismatch was as a result of the simulation tools used. The system‟s temperature depression was about 3° C below ambient, the report concludes.

Hamza et al (1992) designed and fabricated a two-parallel-plate radiator to study the cooling of water due to night sky radiation. The radiator plate was painted black and covered with a polyethylene wind screen cover. The mass flow rate of water and inlet and outlet temperatures was measured. A mathematical model was also

developed to examine the performance of the system. The system was later validated with experimental results which gave 5% deviation between the two results. Optimum values for the water mass flow rates were reported. In another study, commercially available flat plate collectors, with some minor modifications, were used at night as radiative coolers **Bassindowa et al (2007).** The temperature variation of fluid inside the radiator was predicted by an equation similar to the equation used for flat plate collector‟s analysis. Recommendations were presented to increase the effectiveness of the radiator. These recommendations include the use of polyethylene to cover the radiator to prevent heat addition due to convection; use of selective surfaces to enhance cooling to the night sky. A cooling rate of about 90 W/m^2 was reported for a clear sky condition.

A proposal for the roof design was suggested and experimented by **Dimoudi and Androutsopoulos(2006).** They made an experimental setup to test a radiator-based roof component. Water pipes were embedded in a concrete roof. The upper part of the roof was basically a radiator made up of steel plates attached to the steel pipes. Water circulated through the radiator and through the pipes inside the concrete roof. Their objective was to cool the room under the roof. Temperatures at several locations in the system were monitored along with the water flow rate. The results obtained shows the room temperature cooled to 3° C below the outdoor temperature.

Igor and Vladimir (2005) analyzed the performance of flat-plate radiative panels operation, using average hourly weather data (See Fig 2.10). Radiative panels, with high-emittance surface cover, were integrated in the space-ventilation system with air-cooling by means of a cold-water coil. The panels were used to cool sufficient quantity of cold water that is collected in a cold-water tank during the nighttime operation. The collected cold water was then used for cooling of the air

during day time. A simulation model for the parametric analysis of the system in summer operating conditions and the influence of its components on the system‟s operation was developed. The model includes the control of the system‟s operation, which prevents water circulation in the periods without cooling contributions.

The purpose of the research was to predict the system behavior in the Irish and the continental Croatian climatic conditions, to enable sizing and design of the test rig that is to be built for experimental validation of the system. The results of simulation were obtained for the small cooling system with a total panel aperture area of 6 m^2 and a volume of tanks of 300 litres.

The results obtained showed that the radiative cooling system is more efficient in maritime than in moderate continental climatic conditions during summer.

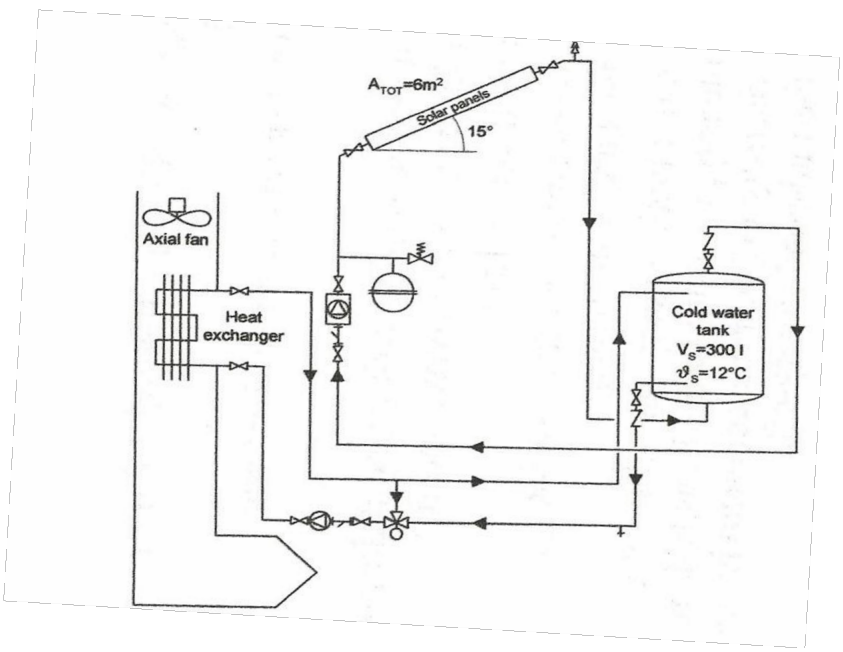


Fig 2.10 Experimental setup for nocturnal cooling system

Igor et al (2003) also designed another solar thermal system that is similar to the one described above .This system, which supplies the required amounts of

energy for the heat exchanger, is used for heating/cooling of the supply air. The system was initially designed for heating purposes but was later modified to provide cooling, and was to satisfy only a part of heating energy requirements. The system consists of a solar panels used for heating of water by means of collection of solar irradiation during daytime and also for cooling of water by means of radiation towards sky (and convection to outside air) during nighttime. Extensive analysis of flat-plate radiative panels operation using average hourly weather data for a maritime climate region was performed. The panels are integrated in the space ventilation system with air-cooling by means of a cold-water coil.

Their primary function is to prepare sufficient quantity of cold water, integrating radiative and convective cooling, that is collected in the cold-water tank during the nighttime operation. That cold water is used for cooling of the air during daytime. By small modification during daytime, solar panels could be turned into collectors and used to produce the hot water that is collected in a separate tank. See Fig 2.11 below. A simulation model for the parametric analysis of the system in summer operating conditions and influence of its components on the system‟s operation was developed. The model includes the control of the system‟s operation, which prevents water circulation in the periods without cooling/heating contributions.

The results obtained showed that the same system, with small modifications to the physical set-up, could provide a significant proportion of the hot water heating requirements in the daytime operation.

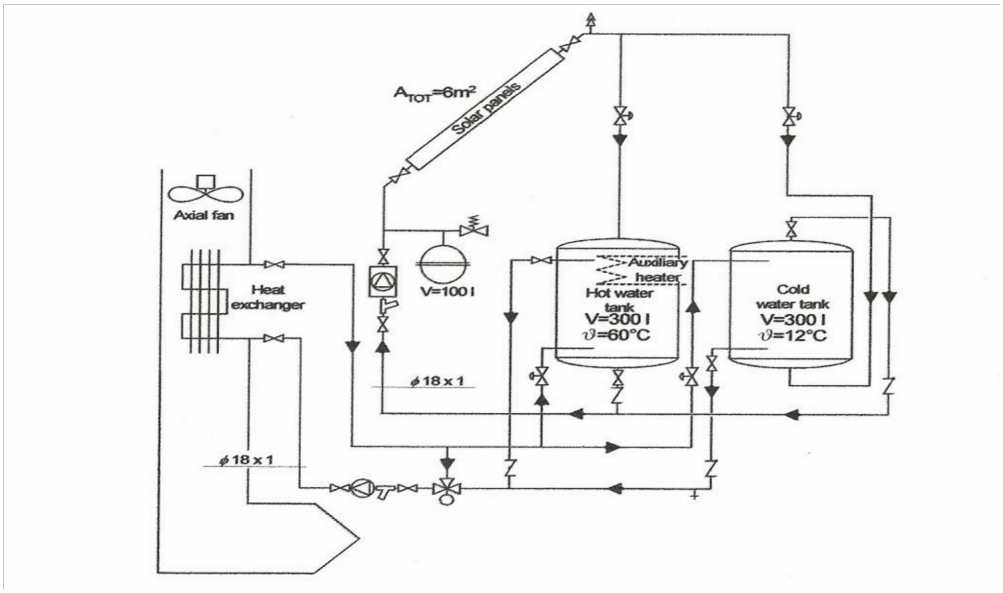


Fig 2.11 Schematic of the system described by Igor et al

Dobson (2005) has reported a night sky radiation cooling system suitable for a room situated in the Namib Desert at Gobabeb, Namibia. The is a hybrid radiative cooling system is consist of the following components: radiator panels, a single water storage tank, room air-to-water natural convection heat exchangers or convectors, circulating pump(s), interconnecting pipe work and temperature sensors and controls (see Fig 2.13). He used mathematical model equations to describe the thermal behavior of the various system components. These equations solve the hourly panel surface, water storage tank and room temperatures. The results obtained compared favorably with values reported in the literature. It is thus concluded that the thermal model presented can be used with confidence as a design tool for the sizing of a night sky radiation cooling system.

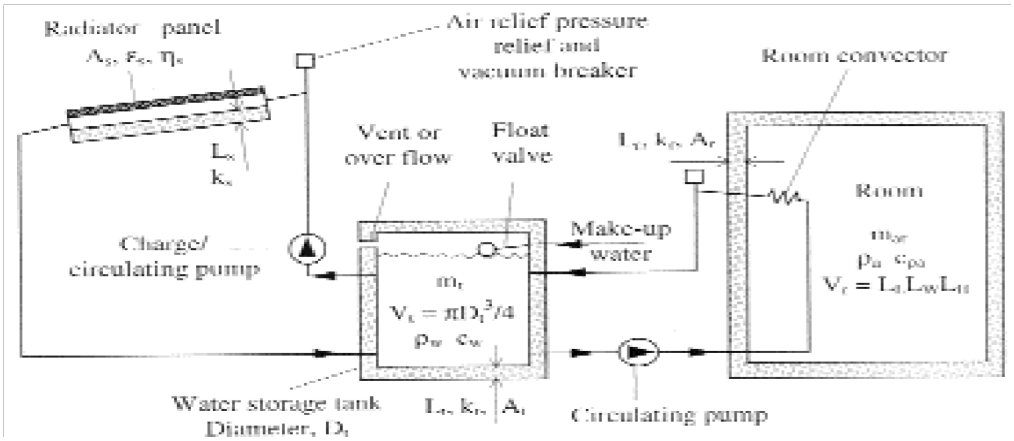


Fig 2.12 Thermal model of the night sky radiation cooling system showing the important components, variables and parameters

González and Givoni (2004) investigated the thermal performance of a passive cooling system in Maracaibo. The system investigated consists of two test cells designed to evaluate the thermal performance of passive cooling systems (Fig. 2.13). The test cells are theoretically identical except for the roof which is different. Both have an interior surface of 9 m^2 (3 m x 3 m) and a height of 2.45 m. They have features typical of a local construction. Their structure, is reinforced concrete (beams, columns and flooring), and their partitions is a hollow bricks of 0. 15 cm. of thickness covered with mortar of sand and cement on the two faces, which represents the traditional building characteristics prevalent in Maracaibo.

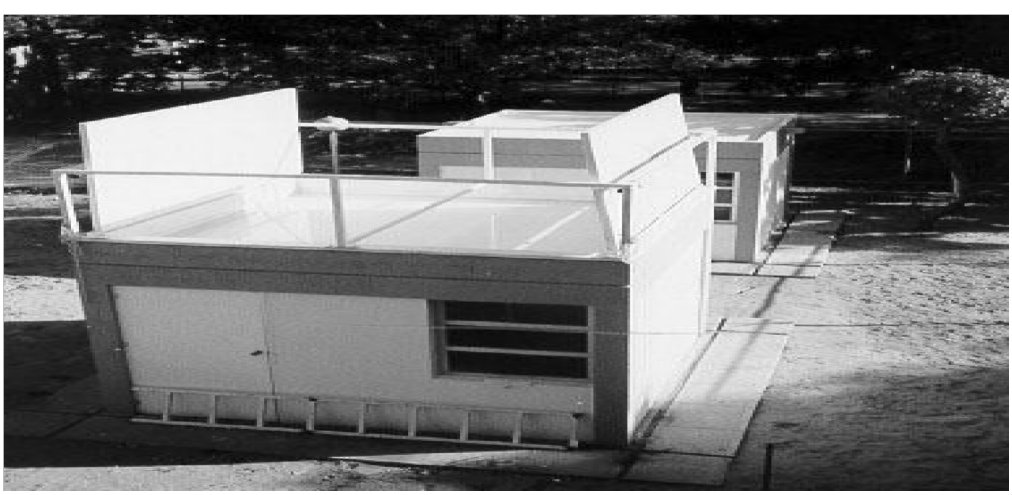


Fig 2.13 the test cell experimental rig by **González and Givoni (2004)**

Access to the cells is made of double door to limits infiltration and reduces heat gain by conduction. The cells also have two windows, one on the north façade and the other one on the south facade. These windows of aluminum and glass were closed and covered by panels of polystyrene of 10 cm of thickness, during the periods of tests. These experiments were carried to characterize the thermal performance and to determine the Mean Cooling Potential (MCP) of each system under the hot and humid climatic conditions of Maracaibo. The results obtained gave a higher MCP values ranging between 21.3 and 27.8 W/m2, in

Radiative/Evaporative system in the January series. While in the August series of the same system, the MCP values were between 14.7 W/m^2 and 22.9 W/m^2. The above variation in the MCP may be attributed to the relative humidity of air; because relative humidity affects radiative cooling.

A new prototype of a hybrid cold thermal storage system used for the preservation of agricultural product (Fig. 2.14) has been recently designed by the

Laboratory of Controlled Environment Agriculture, Dept. of Agricultural Environment Engineering, And National Institute for Rural Engineering, Japan. The system consists of a cold room, night sky radiator, a heat exchanger with PCM capsules. The system was designed to provide supplemental energy for the preservation of fresh vegetable. The system is equipped with a “cold room” for cooling at temperatures of 1-15°C, and a “medium-cold room” for cooling at temperatures of around 10°C or above. The most important feature of this system is to manufacture cold water by circulating water in the sky radiator utilizing night sky radiative cooling and cool outdoor air in nighttime, and then to cool the medium-cold room with the water, namely CFC-free coolant. The cold room is cooled with the ice thermal storage cooling system using inexpensive nighttime

power. The medium-cold room is about 1.6 m^2 and is cooled with a thermal load of 85-170 W(corresponding to the respiration heat of 160 kg of spinach at 1523°C), the room temperature was maintained at 13-15°C on a fine day in early November with cold water from a sky radiator.

The result obtained from an extensive laboratory testing gave a COP of about 1 and an average medium room temperature of about 10 ^{o}C. However, this system is expensive and does not use radiative cooling exclusively.

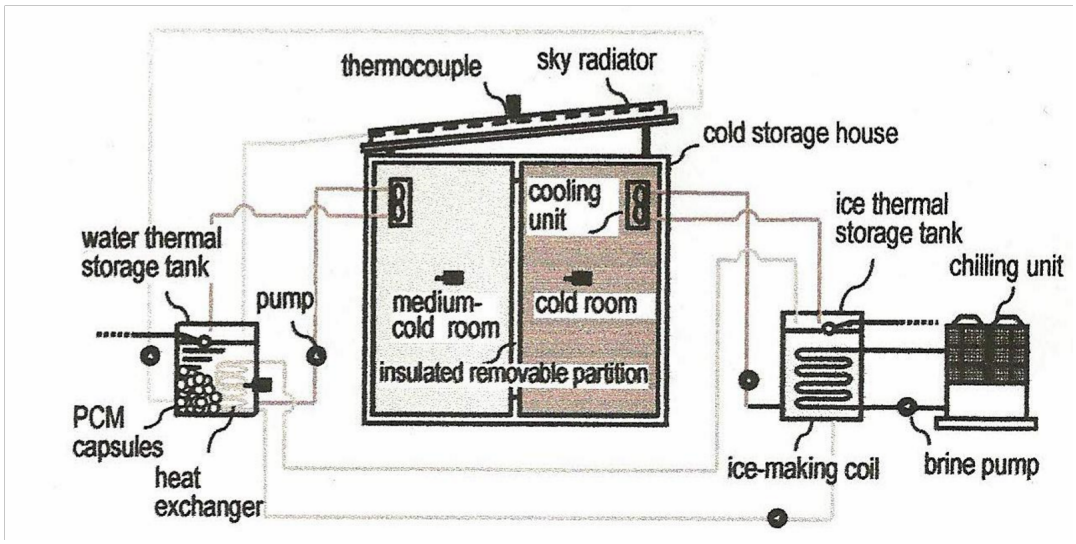


Fig 2.14 Schematic diagram of the hybrid cold thermal storage system

The literature survey was initiated as part of the overall investigation to examine extensively the extent of present work in nocturnal radiation cooling system. The exercise had two fold purposes. The first was to provide the necessary fundamental information on nocturnal radiative cooling theories, system designs and instrumentation. The second was to identify the specific and general problems associated with the various designs in actual field operations. The nocturnal cooling system installation first costs are very high, and depend strongly on the prevailing weather condition for its performance. Over 90% of the works reported in the

present study were tested under temperate climate where the ambient conditions are very low.

Furthermore, nocturnal cooling system installations are not easily carried out without changing the initial design configuration of the building, thus, sizing of the radiator, its orientation for the actual condition in the location of high ambient temperature and high solar heat gain becomes a serious design consideration. The most important limitation to nocturnal radiative cooling as reported in the literatures is the prevailing weather condition; humid climate poses a serious problem to nocturnal cooling due to high relative humidity. In this case, a thorough experimental evaluation is needed to ascertain the most suitable selective surface to be used. **Parker (2005) and Meir et al (2002)** reported some improvement using highly reflective surface and a polymer based radiator respectively. However, these added to the cost of installation of these systems significantly. There is, therefore, the need for a simpler and more maintenance free nocturnal radiative system. Such system should be easy to produce in the country of use from locally available material.

CHAPTER THREE

Methodology

The present study is an experimental investigation of nocturnal radiative cooling system under a tropical climate. The work involves experimental study of the nocturnal cooling of a building using the prevailing climatic conditions in Owerri, Nigeria. Accordingly, the space to be cooled and the nocturnal cooling system are designed using thermodynamics principles, relevant physical laws and meteorological conditions of Owerri, Nigeria. This is necessary to generate design specification for their construction.

Materials for the fabrication of the nocturnal cooling system, which include aluminum sheets, steel pipes, plastic tanks, as well as hollow cement blocks for construction of the test cell were sourced locally. The radiator coated with highly

reflective and the entire experimental units were fabricated according to the design specification

The method for data acquisition involves the use of a locally designed data logger fabricated for this work. The data logger measures the temperatures at regular intervals and records them in the computer memory. The basic parameters of interest are the ambient temperature, radiator surface temperature, and water temperature and control room temperature. The relative humidity, humidity ratios, and air velocity were obtained using specialized weather evaluation software called

"AIR MAP". This software computes the relative humidity, humidity ratios, air velocity, enthalpy of air, and the vapour pressure of the air using the wet and the dry bulb temperatures obtained directly from the field measurements.
The data obtained was analyzed and interpreted using the relevant equations of the system performance criteria. Consequently, the results obtained were then presented in tabular and graphical form.

3.1 Configuration of the Nocturnal Cooling System

The configuration of the nocturnal cooling system is illustrated in Fig 3.1. This arrangement illustrates the nocturnal cooling system for the day time and nighttime operations. The stored energy obtained during the nighttime is used to cool the space during the day when cooling demand is highest. This is achieved using control valves A and B. For day time cooling, the valve B is shut, thereby restricting water from flowing into the sky radiator. The water from the storage tank flows between the storage tank and the heat exchanger.

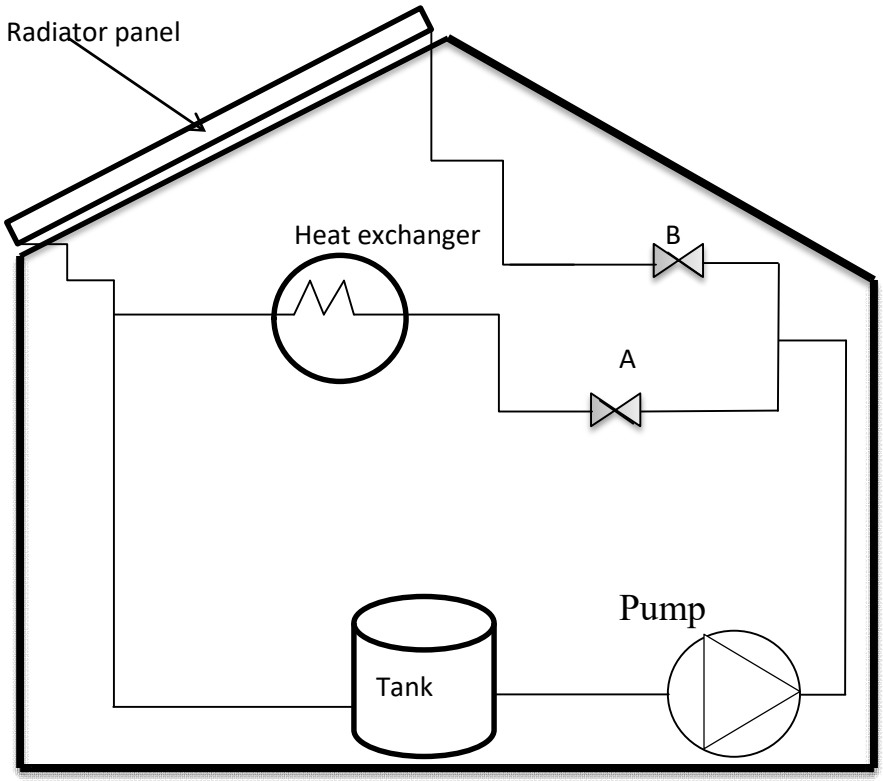


Fig 3.1 schematic diagram of the experimental nocturnal cooling system.

The experimental set-up is made up of a test cell, flat plate sky radiator, a heat exchanger unit, a storage tank, pump, interconnecting pipes and a control room. The test cell is a rectangular building made from 15cm hollow blocks. It has two windows and an access door while the roof is pitched at 12°.

The flat plate sky radiator is made of twelve cylindrical pipes which are connected to two headers feeder pipes. These headers feeder pipes allow the water to flow into and out of the radiator. The entire pipe network is sandwiched between an upper and lower anodized aluminum sheets. Anodizing involves placing a sheet of aluminum into a chemical acid bath, quite often acetone in laboratory experiments. In this manner, the sheet of aluminum becomes the positive anode of a chemical battery and the acid bath becomes the negative. An electric current passes through the acid, causing the surface of the aluminum to oxidize. The oxidized aluminum forms a strong coating as it replaces the original aluminum on the surface. Thus, this process reduces the tendency of the aluminum surface to degrade due to harsh environmental conditions.

Aluminum was used because according to **Parker (2005**), better cooling is achieved under tropical region from material with strong reflectivity in the short wave range between 8-14 um atmospheric windows. The common practice is to coat the surface of the material with black colour. However, black colour has been found not to be effective for night time cooling purpose under this atmospheric window. Nevertheless, under the secondary atmospheric window between the short-wave range of 17-22μm, black colours has been found to perform effectively under extreme dry climates like the desert region. The radiator was placed on the roof at a tilt angle of 12°. According to **Meir et al (2000),** this reduces cooling potential by 1% compared to 32° which reduces cooling potential by 7%.

The heat exchanger is made of a galvanized iron sheet painted with a rectangular shape. It is mounted horizontally between the lintel and the roof, which configuration is expected to enhance better cooling by convection. It has an inlet and outlet water channels. Cold water from the radiator enters the heat exchanger through the inlet channel and fills the rectangular chamber. At that point, heat from the room is transferred to the heat exchanger through convective heat exchange between the indoor air and the heat exchanger. This is made possible by the fact that cold air from the surface of the heat exchanger displaces the warm air below in the room. During the daytime, the cooled water stored in the storage tank is circulated round the heat exchanger while the radiator is isolated with aid of a control valve B. This is very essential because during the day, the radiator gains heat due to high insolation. This will result in the heating of the water in the storage tank.

The cooling tank which serves as a thermal storage medium is a Polyvinyl chloride (PVC) tank. The tank is insulated because it is anticipated that the stored cooled water when passed through the heat exchanger during the day will keep the

indoor temperature at reasonable value throughout the day **Bassindowa et al (2007)**.

When the pump is switched on, the cold water in the storage tank circulates between the heat exchanger and thermal radiator and the storage tank. This way, the cold water will be able to absorb heat from the room as it flows through the heat exchanger and lose it to the sky at the thermal radiator.

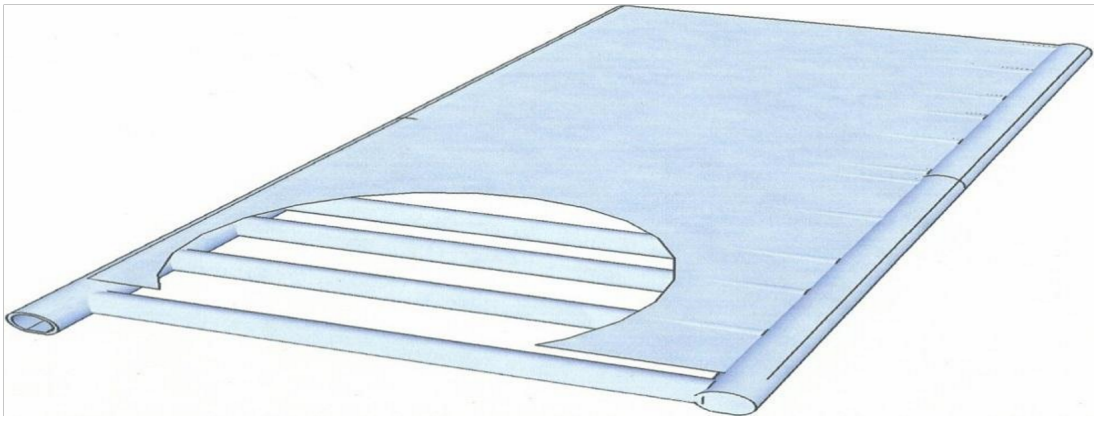


Fig 3.2 showing the arrangement of the radiator modules

The heat transfer fluid when lifted by pump power in the upper part of the radiator, losses heat as it passes through the radiator loop. This consequently reduces the temperature of the water as it gets back to the water tank. This process is continuously repeated as long as the pump is working. The space cooling is achieved mostly by the convective heat exchange between the indoor air and the heat exchanger which is cooled by circulating water from the night facing sky radiator.

3.2.0 Engineering Design of the Nocturnal Cooling System

3.2.1 Thermal Comfort Consideration Analysis in Nocturnal Cooling Design

The section serves as a prelude to what that is expected from the present study to achieve putting into serious consideration the prevailing factors that will

ensure a proper thermal comfort for the end user of the system. Thermal comfort as defined in **ASHRAE Standard 55a-(2004)**, is that condition of mind which expresses satisfaction with the thermal environment. Because there are large variations, both physiologically and psychologically, from person to person, it is difficult to satisfy everyone in a space. The environmental conditions required for comfort are not the same for everyone. However, extensive laboratory and field data have been collected that provides the necessary statistical data to define conditions that a specified percentage of occupants will find thermally comfortable **ASHREA, Fundamentals, (2001)**. The present nocturnal cooling system is expected to provide thermal comfort within a space through the convective heat exchange between the heat exchanger and the indoor air in which case, the cold air around the heat exchanger surface displaces the warm indoor air below.

Thermal comfort in a typical room according to **ASHREA, Fundamentals, (2004)**, is a function of the temperature, air velocity, the convective heat transfer coefficient etc. Enhanced air velocity may be achieved through proper ventilation of the test cell. **Fanger (1994)** relates comfort directly to the convective heat transfer coefficient, rather than to the average air velocity. According to **Fanger et al (1986),** draft is felt at an air temperature of 22 °C if the convective heat transfer coefficient is above 12 W/m^2K. Thus, the issue of upper limit and lower limit of thermal comfort need to be addressed in order to establish the extent different individuals can adapt to different room condition. The **ASHRAE Standard 55a(1995)** gives the upper limit of comfortable temperature as 26°C, and the wet bulb temperature upper limit as 20°C. However, many studies have concluded that the comfort temperature is higher in tropical regions, since humans have the ability to acclimatize to varying physiology and psychological conditions. Field study results show that the discrepancies between thermal comfort requirements amongst individuals vary with geographical location. A study carried out by **Busch (1992)**

shows that people living in air conditioned rooms in Thailand can accommodate upper limit temperature of 28^0C and 31^0C respectively in naturally ventilated rooms. In another experiment, **Kwok (1998)** conducted a survey of 3,544 students and teachers in 29 naturally ventilated and air-conditioned classrooms in Hawaii USA. He found that naturally ventilated classroom occupants accept a wide operative temperature range (22.0 – 29.5^0C).

Operative temperature according to **Fanger et al (1974),** is defined as the average of the ambient temperature and the mean radiant temperature inside the enclosure, weighed by their respective heat transfer coefficients. They show that supplying air at 22 °C with a velocity of 0.10 m/s and 30% turbulence intensity would elicit complaints of draft from 10% of building occupants. With the above suggestions, it becomes imperative to design a cooling system that will provide comfort having in mind the aforementioned constraints as outlined by **Fanger (1994) and Fanger et al (1986)**

3.2.2 Estimation of the Cooling Load of the Room

The building cooling load consists of heat transferred through the building envelope (walls, roof, floor, windows, door etc.) and heat generated by occupants, equipment, and lighting. The load due to heat transfer through the envelope is known as external load, while all other loads are known as internal loads. The percentage of external and internal load varies with building type, site, climate, and building design.

The cooling load of a room is made of two components; the sensible heat and the latent heat. Sensible heat represents only the heat that brings about an increase in the temperature of a substance. It is added to the test cell space by conduction, convection, and radiation. The sensible heat load component of the test cell includes the heat transmitted through the floors, ceilings, walls, and pump

circulating water through the sky radiator and infiltration and ventilation air. On the other hand , latent heat load involves the heat gain which occurs when moisture is added to the space either from internal sources (e.g. vapor emitted by occupants and equipment) or from outdoor air as a result of infiltration or ventilation to maintain proper indoor air quality.

This work, which is concerned with the nocturnal cooling system, is expected to produce cooling within the test cell that will yield thermal comfort for its occupant. This could be achieved through the direct convective heat exchange between the heat exchanger and the indoor air.

3.2.2.1 The Sensible Cooling Load

The test cell cooling load components are; the heat transmitted through the floor, walls, roof, and the pump circulating water through the sky radiator, infiltration of outside air, and air introduced by ventilation. Calculating all these loads individually and adding them up gives the estimate of total cooling load. The load, thus calculated, constitutes total sensible load. The normal practice is that, depending on the building type, certain percent of it is added to take care of latent load **Saitoh and Ono (1984).**

Step by step calculation procedure has been adequately reported in the literature **Michel and Briggs (1979), Dobson (2005) Ansari et al (2005)**.Therefore; the step by step calculation of the sensible load is presented as follows:

3.2.2.2 Heat gain through the floor:

The heat gain through the floor may be evaluated using the expression below:

$$Q_{rfl}=h_c A_f (T_r-T_a) \qquad (3.1)$$

The heat transfer coefficient between the air and the floor may be evaluated using Jacob"s correlation for horizontal surface in enclosed air spaces as (**Rajput,2008).**

$$Nu=0.195(Gr)^{0.25} \quad (3.2)$$

where Grashoff number Gr

$$Gr = \frac{L^3 \beta g \Delta t}{v^2} \quad (3.3)$$

Rajput (2008), also gives the general relation for obtaining the convective heat transfer coefficient using the Nussert number as:

$$Nu= \frac{h_{cf} L}{K} \quad (3.4)$$

Thus, when the Nusselt number in equation (3.2) is evaluated, equation (3.4) can then be used to find the heat transfer coefficient requested. To obtain the parameters required to solve equations (3.2), (3.3), and (3.4), the thermo physical properties of air needs to be evaluated at some certain temperature range. Therefore, since the present work aims at developing thermal comfort for the occupants of the room, we can choose the temperature region that will ensure some levels of thermal comfort. According to Kwok (1998) and **Fanger et al (1986),** the operative temperature that ensures thermal comfort falls within 21 ^{o} C and 22 ^{o} C. Therefore, choosing a temperature of 22 ^{o} C and assuming that the room temperature at the initial period falls between an averages of 27-28 ^{o} C; the thermophysical properties of air can then be evaluated using the mean temperature between 28 ^{o} C and 22 ^{o} C which is 25 ^{o} C. The thermo physical properties of air at 25 ^{o} C, taking from (Rajput, 2008) are given as: k=0.0863 W/m^{o} C,v=15.56 x 10^{-6},

While β is obtained as: $\beta = \frac{1}{(Tf+273)}$

By substituting these values into equation (3.3), the Grashoff number becomes:

$$Gr = \frac{3^2 x 0.00336 x 9.8 x 6}{1.556 x 10^{-7}}$$

$$Gr = 7.35 \times 10^{10}$$

If the Grashoff number obtained from the above expression is substituted into equation (3.2), the Nusset number becomes:

Nu=101.3

From equation (3.4), the heat transfer coefficient becomes

$$h = 3.02 W/m^2 \quad (3.5)$$

If h_c is substituted into equation (3.1), the heat gain through the floor becomes:

Q_f=3.02 x 9 x 6=162 W

3.2.2.3 Heat gain through the walls

The heat gain by the wall is given by the expression:

$$Q = \sum_{n=1}^{n=4} h_C A_{WALL} (T_{WALL} - T_{air}) \quad (3.6)$$

The simplified relation for the heat transfer coefficient for wall surface or vertical surface in enclosed air space at atmospheric pressure is given by **Rajput (2008) as:**

$$Nu=0.064(Gr)^{1/3}(H/L)^{-1/9} \quad (3.7)$$

$$Nu=0.064 \times (7.35 \times 10^{10})^{0.333} \times (2.5/3)^{-1/9}$$

Thus, h_{cw} for the wall can be obtained from equation (3.4) as

$$Nu=125=\frac{hL}{k}$$

h= (125 x 0.0863)/2.5 h=4.46

W/m ^{2o}C

By substituting the heat transfer coefficient h_{cw} and the combined areas of the walls, the heat gain due to the walls becomes:

Q_{walls} =4 x 4.46 x 7.5 x (28-22)

Q=802.9 W

3.2.2.4 Heat Gain through the roof

The correlation for the heat transfer coefficient for horizontal surface facing down is given by **Rajput (2008) as:**

$$h=1.32\left(\frac{\Delta T}{L}\right)^{0.25} \quad (3.8)$$

Where ΔT is the temperature difference between the roof surface and the indoor air temperatures respectively and L is the length of the room. The value for ΔT was obtained by measuring the temperature of the test room roof and the indoor air in the test room. The measured test room roof temperature gave 31^0C while the test room indoor air temperature gave 25^0C. Therefore, by substituting ΔT and L into equation (3.8), h_{cr} becomes:

h_{cr}=1.32 x (6/3)

h_{cr}=1.5 W/m $^{2\,0}$C

Therefore, the heat gain through the roof may then be evaluated using equation (3.8)

$$Q_r = h_{cr} A(\Delta T) \quad (3.9)$$

Q_r=1.5 x 9 x (31-25)

Q_r= 81 W

3.2.2.5 Heat gain from the water pump

The primary source of heat from the water pump comes from the electric motor that drives it. Calculation of this load component is not straightforward. The rate of heat gain at any given moment can be quite different from the heat equivalent of power supplied instantaneously to the electric motor. The heat generated by the operation of the electric motor is transmitted to the indoor air thereby adding to the heat gain to the test cell.

Generally, if the motor and the machine are in the room, the heat transferred can be calculated as **ASHREA,Fundamentals(2001),**

$$Q = 736 * (P / Eff) * FUM * FLM \qquad (3.10)$$

Where P = Horsepower rating from electrical power plans or manufacturer‟s data,

Eff = Equipment motor efficiency, as decimal fraction

F_{UM} = Motor use factor (normally = 1.0)

F_{LM} = Motor load factor (normally = 1.0),

For 0.25 hp pump, and design efficiency of 0.86 and by substituting these parameters, the heat gain due to the pump operation becomes: Q = 736 x (0.25/0.86) x 1 x 1 Q = 213 W.

The 0.25 hp pump was used because it is cheap and require little energy for it operation.

The heat gain due to infiltration and ventilation also contributes to the sensible heat load of the test cell. Since that actual rate of infiltration of air at this stage is unknown and based on rough approximation extracted from **ASHREA**

Fundamental (2001), 15% of the estimated heat load is added to be the combined heat load from air infiltration and ventilation. This combined load yields 188.8 W. Similarly, it is also a common practice to add 5 to 10% to the estimated sensible heat to represent the latent heat load. The value used depends on the reliability of the information used in determining the sensible heat load from contributing sources. In this work, 10% was used. Therefore the total cooling load becomes:

Sensible cooling + 10% sensible cooling load

$Q_T = Q_f + Q_w + Q_{rf} + Q_{pump} + Q_{others} + Q_{latent}$ *(3.11)*

= 162.2+802.9+81+213+188.8+145

Q_{total} = 1592.9 W.

3.3.0 Design of the Heat Exchanger and Sky Radiator

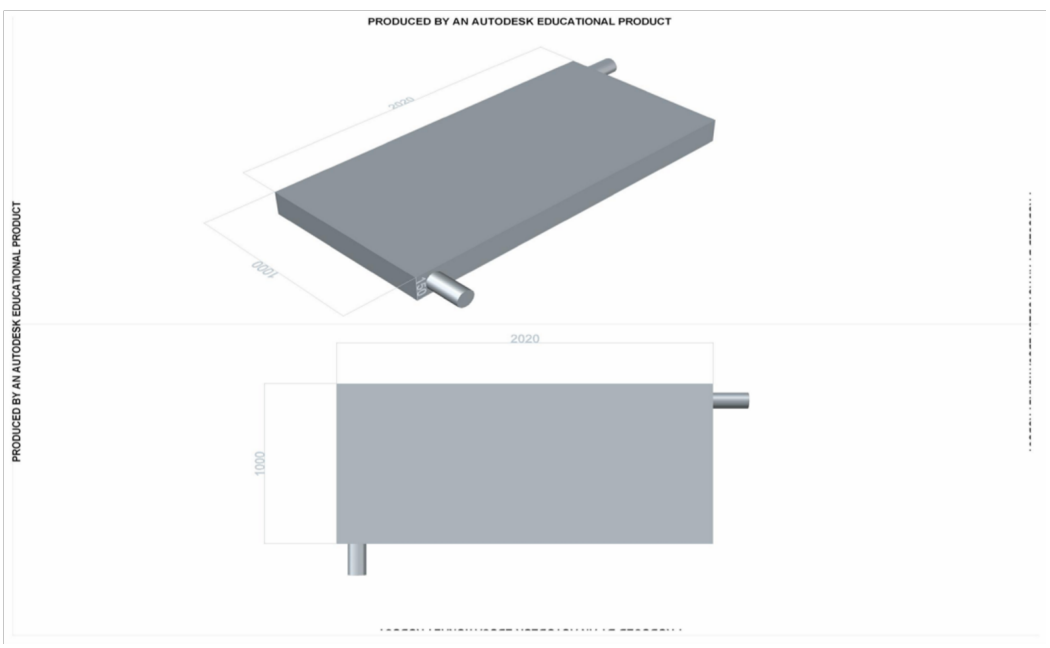


Fig 3.3 showing the heat exchanger configuration

In order to offset the cooling load of the test cell evaluated above, the heat exchanger is expected to provide enough cooling through the convective heat exchange between the indoor air and the heat exchanger which is cooled by the

water flowing into it from the radiator. However, the cooling obtained from a heat exchanger unit depends on the prevailing meteorological conditions and the effectiveness of the heat exchanger unit. This is because the prevailing meteorological conditions determine the temperature drop obtained from the circulating water as it passes through the night sky radiator. In this work, room temperature of 22 ^{o} C, in line with Fanger‟ s proposed operative temperature for adequate thermal comfort, is expected to be realized through the incorporation of the heat exchanger unit. In order to design the heat exchanger, it is necessary that the total cooling load transfer be related with its governing equation as illustrated in **Rajput (2008).**

$$Q_T = UA\,\theta_m \qquad (3.12)$$

Where U is the overall heat transfer coefficient of the various modes of transfer and A is the surface area of the heat exchanger, while θ_m is the logarithmic mean temperature difference. If the overall heat transfer coefficient U, flow condition, specific heat and mass flow rate of both fluids are assumed constant, within the limits of the operating temperature involved in this work, then the area of the heat exchanger that will offset the total cooling load of the test cell can be evaluated as follows:

The mass flow rate of water through the heat exchanger is assumed to be the same as the mass flow rate of the water pump, which is equivalent to 0.67 kg/s.

The effective air mass flow rate m is given by Allard. (1998) as

$$m = AH\,R\,\eta\,\rho \qquad (3.13)$$

Where A and H are the area and height of the room repectively, R is the air change rate and η is the temperature efficiency, defined as

$$\eta = (T_{out} - T_{amb}) / (T_b - T_{amb}), \qquad (3.14)$$

This takes account of the fact that the temperature of the air flowing out T_{out},(exit air temperature) is lower than the temperature of the building T_b. The density and the specific heat capacity of air are taken as ρ=1.2 kg/ m ^{3} and C_p = 1000 J/kg K. Assuming that the height of the room is 2.5 m, a constant effective air change rate given by Nikolai et al(2006) as R η =6 h^{-1} The mass flow rate of air in the room becomes

$\dot{}$ = *(3x3x2.5x6x1.2)/3600*

$\dot{}$ *=0.045kg/s*

The air velocity was taken from the average wind velocity for Owerri which ranges between 0-2 m/s **Anyanwu and Iwuagwu (1995).** The average water temperature of the heat exchanger should be lower than 22 ^{o} C if a temperature of 22 ^{o} C is to be maintained within the space. Assuming a temperature of 28 ^{o} C within the space before the cooling commences. This value is within the order of magnitude of room temperature in a typical tropical night immediately after sunset. **Bagirgas, and Mihalakakou (2008)** gave the water temperature under normal atmospheric temperature as 25 ^{o} C. This temperature varies depending on the weather conditions. Then, by using a simple energy balance, we have

Heat given up by air in the test cell = heat picked up by the fluid in the heat exchanger

$$Q= m\, C_p\, (T_{airi}-T_{airf}) = m_w\, C_{pw}\, (T_{wi}-T_{wf}) \qquad (3.15)$$

Substituting the various values of the parameter in equation (3.15), we obtain

$$0.45 \times 1.006(28-22) = 0.66 \times 4.187(25-T_{wf})$$

$$25 - T_{wf} = (0.45 \times 1.006 \times 5)/ (0.66 \times 4.187)$$

$$T_{wf} = 25-0.81=24.16\ ^o\ C$$

The logarithmic mean temperature difference LMTD is given by **Rajput(2008).** as:

$$\Delta m = \frac{(\Delta 1 - \Delta 2)}{In\left(\frac{\Delta 1}{\Delta 2}\right)} \qquad (3.16)$$

$$\Delta m = \frac{(28-22)-(25-24.16)}{In(5/0.82)}$$

$$\Delta m = \frac{4.18}{\left(In\frac{5}{0.82}\right)}$$

$$\Delta m = 2.31$$

The overall heat transfer coefficient of the various modes of heat transfer may be evaluated as follows:

The heat transfer coefficient of the air in the test room is mostly due to convection between the heat exchanger and indoor air. Accordingly, the heat transfer coefficient of the air within the room is obtained using the Jacob‟s correlation for horizontal enclosed air spaces.

The simplified relation for the heat transfer coefficient for vertical surface in enclosed air space at atmospheric pressure is given by **Rajput, 2008) as:**

$$Nu = 0.18(Gr)^{1/4}\left(\frac{H}{L}\right)^{-1/9} \qquad (3.17)$$

The *Grashoff* number can be evaluated using equ (3.3)

$$Gr = \frac{L^3 \beta g \Delta t}{v^2}$$

To obtain the parameters required to solve equations (3.17), the thermo physical properties of air needs to be evaluated at certain temperature range. Since the present work aims at developing thermal comfort for the occupants of the room, we can choose the temperature region that will ensure some levels of thermal comfort.

According to **Kwok (1998)** and **Fanger et al (1986),** the operative temperature that ensures thermal comfort falls within 22^0 C- 28 ^{o} C. Therefore, choosing a temperature of 22 ^{o} C and assuming that the room temperature at the initial period falls between upper limit temperatures of 27-28 ^{0} C; the thermophysical properties of air can then be evaluated using the mean temperature between 28 ^{0} C and 22 ^{0} C which is 25 ^{0} C. The thermo physical properties of air at 25 ^{0} C, taken from Rajput (2008) are given as: k=0.0863 W/m^o C,υ=15.56 x 10^{-6},

While β is obtained as: $\beta = \frac{1}{(Tf+273)}$

$$Gr = \frac{3^2 x 0.00336 x 9.8 x 6}{15.56 x 10^{-6}}$$

$$Gr = 7.35 x 10^{10}$$

$$Nu = 0.18 x (7.35 x 10^{10})^{-0.25} x (2.5/3)^{-0.111}$$

Nu =95.6

Therefore, by substituting this value into equation (3.17), the Nusselt number becomes:

$$Nu = 0.18 \ x \ (7.35 \ x \ 10^{10})^{0.25} x \ (0.8333)^{-0.111}$$

$$Nu = 95.6$$

Thus, h_{cw} for the wall can be obtained from the equation 3.4 as:

$$95.6 = \frac{hL}{k}$$

From the above expression, the required heat transfer coefficient becomes:

$$h_w = (95.6 \ x \ 0.0863)/2.5$$

$$h_w = 3.3 \ W/m^2.K$$

Similarly, the heat transfer coefficient of water flow through the heat exchanger can easily be estimated using the Blasius equation , **Rajput(2008).**

$$Nu = 0.0232(Re)^{0.8}(Pr)^{0.4} \quad (3.18) \text{ and}$$

$$Re = \frac{\rho V x}{\mu} \quad (3.19)$$

Where V is the average flow velocity of the water which is assumed to be the same as the flow velocity of the pump obtained from the manufacturer specified volume flow rate. Therefore, the thermo-physical properties of water at 25 ^{0}C are **Rajput (2008):** $k=0.60797\ W/m\ ^0C$, $\rho=996.65\ kg/m^3$

$\mu=0.862 \times 10^{-3}$

$Pr=5.924$

Substituting these parameter into equation (3.19), Reynolds number becomes

$$Re=\frac{996.65 \times 1.35 \times 3}{0.862 \times 10^{-3}}$$ Re=4,679,582

Substituting Re into equation (3.23) gives:

$$Nu = 0.0232(4679582)^{0.5}(5.92)^{0.333}$$

$$Nu=0.032 \times 2163 \times 1.807$$

$$Nu=38$$

The convective heat transfer coefficient h can calculated from the Nusselt number evaluated using equation (3.4) $h_w = \frac{0.6079 \times 38}{3}$ $h_w=7.76\ W/m^2\ ^oC$.

The overall heat transfer coefficient of a heat exchanger may be evaluated using the expression outlined by **Rajput(2008)** in equation (3.20) as.

$$U = \left(\frac{1}{hC} + \frac{1}{hW} + \frac{L}{k}\right)^{-1} \quad (3.20)$$

Therefore, substituting the various heat transfer coefficients into equation (3.20), we have,

$$U = ((3.30)^{-1} + (7.76)^{-1} + (2.5))^{-1}$$

$$U = 0.34\ W/m^{2\ o}C$$

The area of the heat exchanger surface may now be evaluated using equation (3.10) below as:

$$Q_T = UA\ \Theta_m$$

$$A = 1.59\ kW/(0.34 \times 2.31.)$$

$$A = 2.02\ m^2$$

This is the minimum surface area required to offset the total cooling load of the test room.

3.4.1 The Heat Exchanger Effectiveness

The heat exchanger effectiveness is the ratio of the actual heat transfer to the maximum possible heat transfer. Since the sky radiator provides the heat transfer to the heat exchanger, the maximum possible heat transfer is equivalent to the total cooling generated by the sky radiator.

Rajput (2008) gave the relation for evaluating effectiveness ε as ε

$$= \frac{\text{actual heat transfer}}{}$$

Maximum possible heat transfer

For a counter- flow type heat exchanger effectiveness is given by **Rajput (2008) as:**

$$\varepsilon = \frac{1-exp[-NTU(1-R)]}{1-Rexp[-NTU(1-R)]} \quad (3.21)$$

Where R is the capacity ratio and is given as:

$$R=C_{min}/C_{max} \quad (3.22)$$

Where C_{min} and C_{max} are the heat capacities of the two fluids The NTU is the number of transfer unit and is given as:

$$NTU=UA/C_{min} \quad (3.23)$$

But $C_{max}=m_wCp_w$ (3.24)

$=0.66 \times 4.2$

$C_{max}=2.772$

And $C_{min}=m_aCp_a$ (3.25)

$=1.24x1.006$

$=1.24$

Therefore, $NTU= \frac{(2.46x1.244)}{1.24}$

$=2.46$

$R= \frac{1.24}{2.772}$

$=0.447$

With these values, the effectiveness ε can easily be estimated from Fig. 3.8 below as:

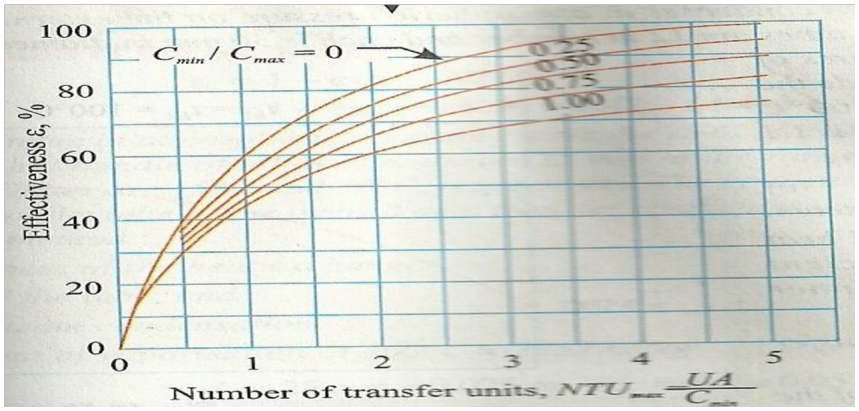


Fig 3.8 showing the graph of effectiveness against NTU

From the graph, ε is obtained as 0.83

3.5 Estimation of the Cooling Load of the Sky Radiator.

The heat exchanger receives the cooled water directly from the sky radiator, and at the point of entering into the heat exchanger; the cooling power obtained is expected to be equal to the cooling power of the heat exchanger. Thus,

$$\varepsilon = Q_T / Q_{Max} \qquad (3.26)$$

Since Q_T is known and Q_{Max} is assumed equal to the cooling load of the sky radiator, then from equation (3.26), the value of Q_{max} can be evaluated by substituting the value of the heat exchanger effectiveness and the total heat transfer by the heat exchanger as:

$Q_{max} = Q_T / \varepsilon$

$Q_{max} = 1.59 / 0.83$

$Q_{max} = 1.924\ kW$

3.5.1 Estimation of the Area of the Radiator Surface

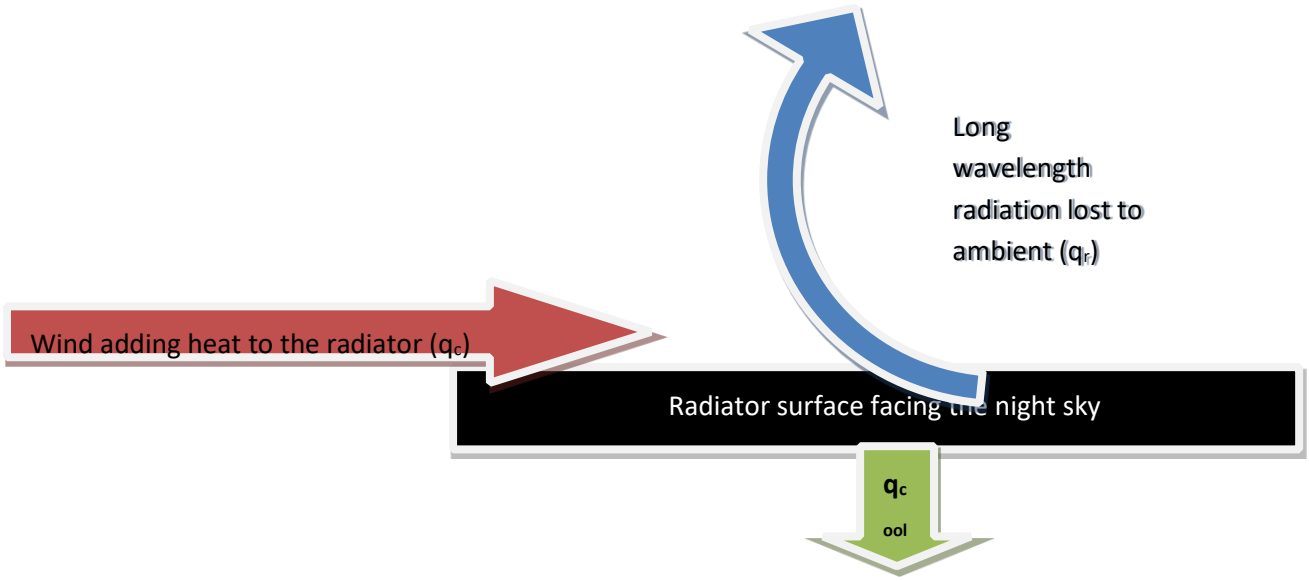


Fig 3.9 showing the radiator surface energy balance

Considering the diagram for the radiator as shown in Fig. 3.9 above, the heat loss of an uncovered night sky radiator is caused by radiation and convection. The heat loss related by conductive heat transfer between radiator and surrounding can be neglected **Meir et al(2002)** so that,

$$q=\sigma\,\varepsilon\,(T_r^4 - T_s^4) + hc\,(T_r - T_a) \qquad (3.27)$$

where

$$q=\frac{Qmax}{A} \qquad (3.28)$$

Temperature depressions of 3.1, 2.8 and 2.5 below ambient has been reported by **Parker (2005), Meir et al (2002) and Ito and Miura (1989)** respectively. With an average ambient temperature ranging from about 22-27 ^{o} C for a clear night, the radiator temperature is expected be in the range 1.3-2.6 ^{o} C below ambient depending on the prevailing weather condition. **Parker (2003)** gave this correlation for the T_s as:

$$T_{sky}=T_a\,\varepsilon^{0.25} \qquad (3.29)$$

For surfaces without wind screen the coefficient for convection h_c is in first order a linear function of the wind speed V which has the form h_c=a + b x V.

Previous studies by **Duffie and Beckmann, (1991), Clark and Berdahl (1980)** show a large variety in assigning values to "a" and "b".

Relations applied **by Martin and Berdahl (1984),** have been validated with experimental data. However, best fit was obtained by using h_c as suggested by the Australian standards of 1989, reported and discussed by **Meir et al (2002)** with V in m/s as:

$$h_c = 3.1 + 4.8\ V \qquad (3.30)$$

Where h_c is the convective heat transfer coefficient W/m $^{2\,o}$K

Meteorological data collected within Owerri gave an average wind speed in the range of 0.1-2.0 m/**s Anyanwu and Iwuagwu (1995). Fanger (1994)** recommends an average air velocity of about 1.35 m/s. Therefore, if we adopt this velocity in the present study, the heat transfer coefficient becomes:

$$h_c = 3.1 + 6.36\ h_c$$
$$= 9.46\ W/m^{2\,o}C$$

Assuming that T_a was taken from the point where temperature is at its highest point, that is 27 ^{o}C, then T_s which is the sky temperature becomes:

$$T_s = 300 \times 0.9^{0.25} = 292\ K.$$

The radiator temperature T_r is obtained by using the average of the results reported in **Meir et al(2002), Ito and Miura(1989), and Parker (2005)** and this translates to 2.5 ^{o} C below ambient. Thus

$$T_r = T_a - 2.5.$$

If we use 27 ^{o} C as a reference temperature, then T_r=24.5 ^{o} C.

The values of T_r varies with the prevailing weather condition.

Therefore, the area of the sky radiator can easily be estimated by substituting the various parameters in equation (3.24).

$q=\sigma\ \varepsilon\ (T_r^4 - T_s^4) + hc\ (T_r - T_a)$

$0.0000000567x0.9x\ (298^4 - 292^4) + 9.46x\ (297.2-300)$

$=27.14-26.49$

$=0.65\ kW/m^2$

Substituting q into equation (3.23) the area of the sky radiator is obtained as:

$0.65=\frac{192}{A}$

$A= 1.924/0.65$

$A \approx 2.9\ m^2$

3.5.3 Length of Radiator Tubes

The physical characteristics of the tubes selected are listed as:

(i) Inner tube: galvanized steel pipe with ID 19.5 mm and OD 25.27 mm The volume of the inner pipe per unit length is given as

$$V= \frac{\quad}{4(0.0195^2)} \qquad (3.31)$$

$V_1= 2.83 \times 10^{-3}\ m^3/m$

The quantity of water round the system at any point in time was calculated to around 40 litres (combined water volume in the entire system) Therefore, the length of pipe required for the water is given by:

$$Total\ length = \frac{Estimated\ volume}{Total\ volume\ per\ unit\ length}$$

$$= \frac{40\ litres}{2.83\ litres/m}$$

$$Total\ length = 14.5\ m$$

3.5.4 Width of the Radiator Plate

The pitch of the radiator tubes is obtained from as:

$$W = \frac{plate\ width\ b}{Number\ of\ tubes} \qquad (3.32)$$

If it is assumed that the tubes are spaced one and half diameter apart, then the pitch of the tubes becomes

$$W = 2.5\ D \qquad (3.33)$$

Combining the equation (3.26) and (3.27), we obtain

$$2.5\ D = \frac{plate\ width\ b}{Number\ of\ tubes}$$

$$b = 2.5\ D\ Nt\ b = 2.5\ x$$

$$29.27\ x\ 14\ b = 1023\ mm$$

The pitch of the radiator tube is

2.5 x 29.27 mm=73.1 mm

Consequently, radiator plate width =1.02 m

Radiator plate length =2.84 m

Radiator tube pitch=0.731 m

The tubes were welded together in a rectangular form.

3.6.0 Selection of the Water Pump

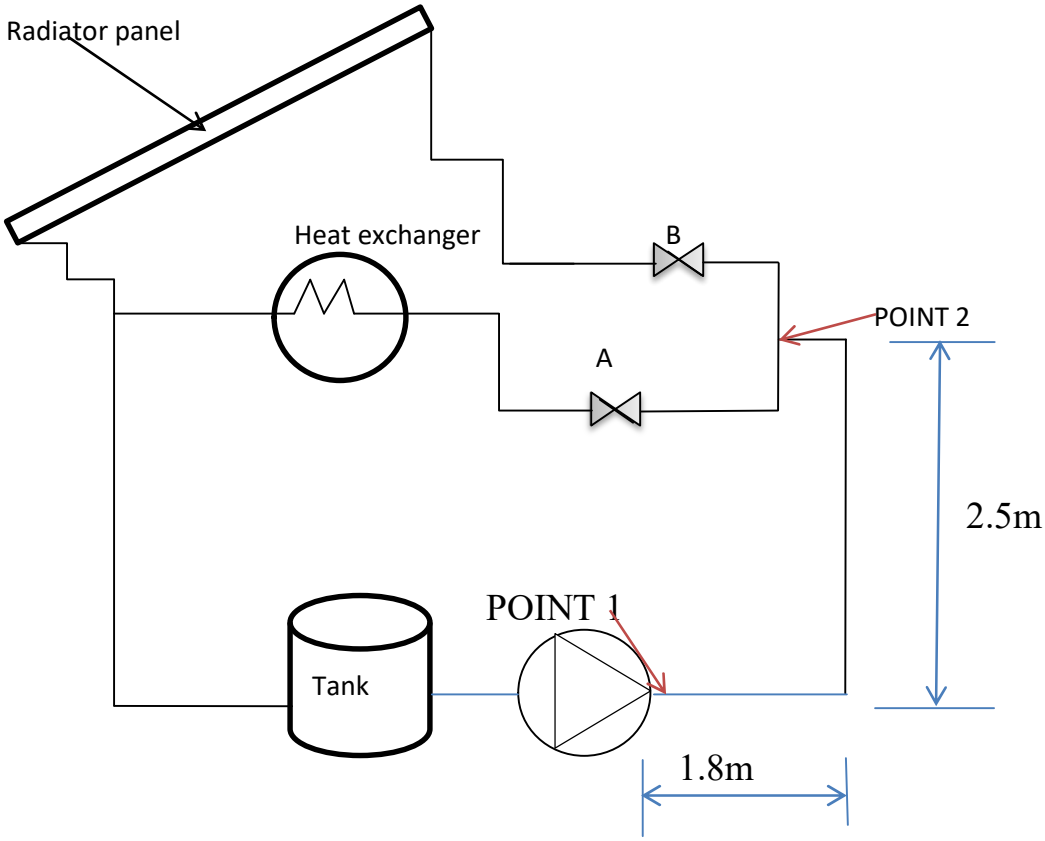


Fig 3.5 schematic diagram of the pipe network.

The energy balance between point 1 and 2 as shown in Fig 3.5 is given by the Bernoulli equation in the form :

$$\Sigma \frac{Pi}{\rho g AiVi} = \left(\frac{p2}{\rho g} - \frac{p1}{\rho g}\right) + \left(\frac{\alpha_2 V_2^2}{2g} - \frac{\alpha_1 V_1^2}{2g}\right) + (z_2 - z_1) + \Sigma fi \left(\frac{Li}{Di}\right)\left(\frac{V_1^2}{2g}\right) + \Sigma \frac{KLiV_1^2}{2g} \qquad (3.34)$$

It is assumed that $p_1 = p_2$ = atmospheric pressure; and $V_1 = 0$ because the initial velocity of a reservoir is assumed to be zero; $z_1 = 0$

If the diameter and the cross sectional area of the pipe are represented as D and A respectively, and the average velocity of the fluid in the downstream of the pump is assumed constant and equal to V_2. Then the Bernoulli equation in equation (3.28) reduces to:

$$H_p = \alpha_2 V_2^2/2g + z_2 + f\,(L/D)(V_2^2/2g) + V_2^2/2g\,(\Sigma K_{Li}) \quad (3.35)$$

Where Hp is the pressure head generated as a result of the flow of water from the reservoir. The energy components $\alpha V_2/2g$, $f\,(L/D)(V_2^2/2g)$, $V_2^2/2g\,(\Sigma K_{Li})$ and z_2 represents , kinetic energy and potential energy changes .

The velocity of flow is related to volumetric flow rate from continuity equation as follows:

$$V_1A_1=V_2A_2=Q. \qquad (3.36)$$

Since V_1 is zero, the equation 3.30 reduces to

$$V_2= Q/A_2 \qquad (3.37)$$

This further reduces to

$$V_2 = Q / A = Q / (\pi D^2/4)$$

By substituting V_2 into equation 3.29 and multiplying the equation by the specific weight (specific weight is the product of density and gravitational acceleration) ρ g Q, we obtain the power as $\rho\, g\, Q\, H_p = (8\, \rho\, Q^3/\pi^2 D^4)\,(\alpha_2 + f\,(L/D) + \Sigma K_{Li}) + \rho\, g\, Q\, z_2$ *(3.38)*

The known variables are:

$\rho = 998.2$ *at 20 ^{o} C, $Q = 0.04\ m^3/s$,* α_2

$= \begin{cases} 1 \text{ for uniform velocity profile ,} \\ >1 \text{ for non-uniform velocity profile;} \end{cases}$

f depends on the Reynolds number of the flow ,L_r = 3.0 m(Length of room) ,K_{Li} for 90° threaded elbows = 1.5 ,K_{LV} for fully open glove valve = 10 ,K_{LT} for tab = 2 ,g = 9.81 m/s^2 ,

z_2 = 2.5 m (height of the building), the friction factor, f, is dependent on the Reynolds number, Re. Re is given in **Rajput(2008)** as:

$$Re = \rho V_2 D/\mu = 4\rho Q/\pi D\mu \qquad (3.38)$$

Substituting the value of the parameters Q, D, ρ and μ into equation 3.34, Re becomes:

$$Re = 4(998.2\ kg/m^3)(0.001\ m^3/s)/\pi D(1.002 \times 10^{-3}\ Ns/m^2)$$

$$Re = 1268.411 / D$$

For Re < 2100, D > 0.604 m = 60.4 cm

For Re > 4000, D < 0.317 m = 31.7 cm

A pipe of diameter D = 0.317 m is much larger than what would even be suitable for the system. Therefore, every pipe less than this value are acceptable.

The above statement entails that any value of D less than 0.317 m will be appropriate for the system. However, the manufacturer specification of the exit outlet of the pump determines the value of D to be chosen.

Design equation therefore becomes

$$P = (8.0911 \times 10^{-7}/D^4)[f(3.0/D) + 19] + 97.923 \qquad (3.39)$$

Then, Re = 1268.411/(0.025 m) = 50720

For PVC pipe tubing $\varepsilon = 0.0015$ *mm*
Thus, $\varepsilon/D = 0.0015/2.5 = 0.0006$

From the Moody chart, at Re = 50720 and $\varepsilon/D = 0.0006$*), the friction factor f becomes f = 0.0195.*

Substituting D = 0.0025 m and f = 0.0195 into the above equation 3.39, we obtain

$$P = (8.0911 \times 10^{-7} / (0.0025\ m)^4)[0.0195\ (3.0/(0.0025\ m)) + 19] + 97.923$$

$$= 103.25\ W$$

To generate an exit volumetric flow rate of Q = 0.04 m^3/s, drawn PVC tubing of 0.0025 m diameter (D) pipe and 103.25 W power pump (P) are required. Hence the 0.25 hp pump was chosen because it is very close to the design requirement and the smallest pump available in the market during the period under review.

3.7.0 Construction Details for the Nocturnal Cooling System

The fabrication of the nocturnal system"s components like the sky radiator and the heat exchanger involved the cutting and welding of metal sheets and pipes. The operations were performed in a standard engineering workshop. The components were linked by network of pipes, using the available plumbing system and technique in the country. The procedures involved in the construction of the various components of the nocturnal cooling system are discussed in detail below.

3.7.1 Construction of the Night Sky Radiator:

The sky radiator is the major component of the system discussed and it consists of aluminum sheet and metal grill. The radiator flat plate which is made of aluminum was cut into dimensions 1020 x 2840 mm using hand shearing machine. With the hand hacksaw, twelve absorbers tubes each of 15 mm OD x 13 mm ID x 1020 mm long were cut from commercially available galvanized steel pipes. Two larger pipes of 25 mm OD x 23 mm ID x 2840 mm long were also cut from commercially available pipes. On the larger outer pipes, twelve equally spaced holes were made to accommodate the small pipes. The perforations were of 15 mm diameter spaced 120 mm. One of the open ends of each of the header tubes was sealed by welding a cover plate onto it. In order to assemble the grill of pipes, the inner perforated tubes were first coupled into the holes created and consequently were welded onto the header tube by manual arc welding.

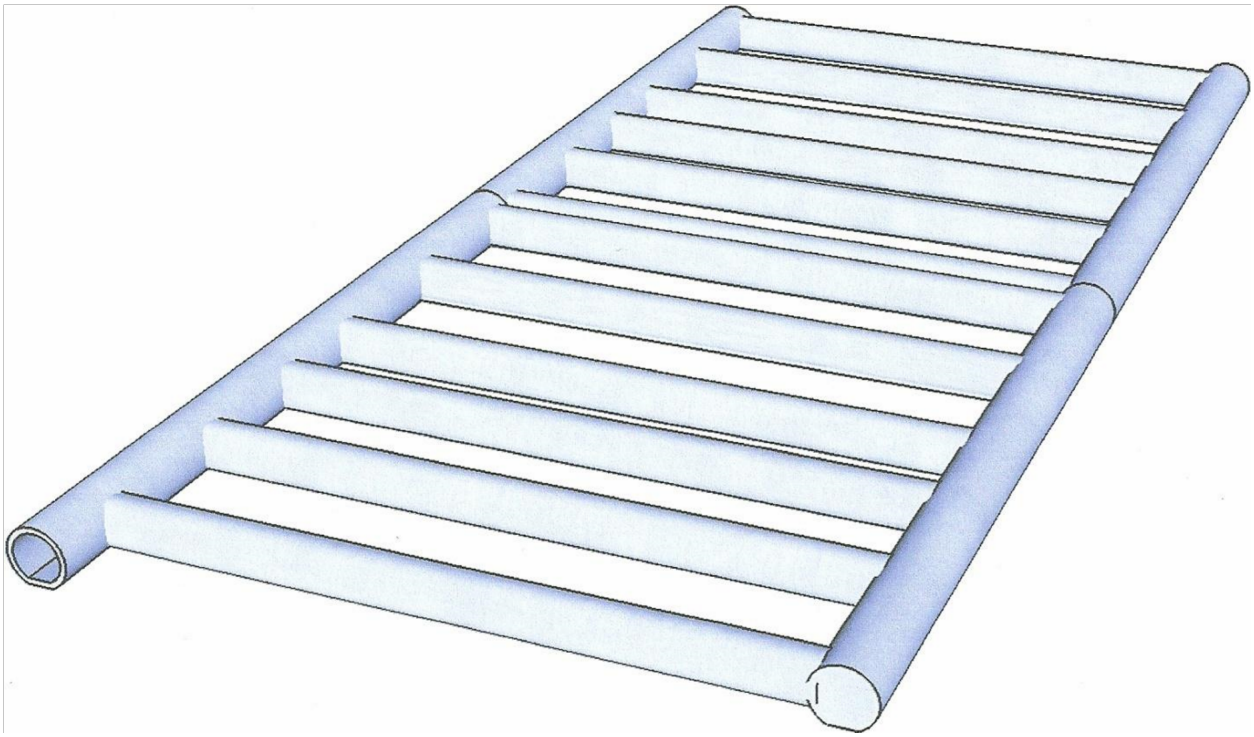


Fig 3.6 the radiator metal grill assembly

The pipes were welded to the delivery pipes. Fig 3.6 shows the metal grill. The joined metal tubes were tested for leakages by pouring water into the metal grill. The points of leakages were sealed with cold weld materials. The aluminum sheets for the radiator surface were cut into shape using a cutter. Two sheets each of 1020 mm x 2840 mm were cut out from the roll of aluminum sheet according to the dimension of the radiator metal grill. The entire pipe grill was covered with aluminum sheets and soldered to the delivery pipes using cold weld metal binding wires; to ensure that the sheet metal is firmly fixed to the metal grills.

The function of the pipe grill is to allow the passage of water from the storage tank through the radiator. It also absorbs heat from the water and transfers it to the surface of the radiator for onward radiation to the night sky.

3.7.2 Construction of the Heat Exchanger

The heat exchanger receives cold water from the sky radiator. It was constructed of 16 gage sheet metal. First the dimensions of 50 mm x 2020 mm x 1000 mm were marked on two metal sheet simply laid together (along their width) on the ground. The 20 mm sheets were marked and cut off from the galvanized steel into four, for both the length and the width. Thereafter, the metal sheets were held in an anvil so that they can be joined together through the process of welding (Brazing).

Accordingly, the sheet metals were welded to specification using an arc weld.

On the 1000 mm surface, two holes of 25 mm at a distance of 700 mm were made on one surface, while underneath the other surface, another holes of the same dimension was made to accommodate the inlet from the sky radiator and water pump for day time cooling and outlet to the water tank respectively. The heat exchanger was filled with water to check leakages. Points of leakages were sealed using special substance called cold weld.

3.7.3 Construction of the Pipe Network

The PVC pipes were cut from commercially available pipes. From the water pump, a length of about 2500 mm was cut and fixed from the water reservoir to the inlet of the pump. At the pump section about 2300 mm was cut and fitted to the pump exit section. A reducer was used to link the pipe to the sky radiator and from the sky radiator; another of 90 ^{o} elbow reducer was used to channel the water from the radiator to the pipe linking the heat exchanger. From the heat exchanger, another pipe of about 1000 mm was used to connect the water from the heat exchanger to the water storage tank.

3.8.0 Experimental Program and Data Reduction

3.8.1 Experimental Instrumentation

The measured performance of the nocturnal cooling system is usually assessed by such parameters as the climatic cooling potential, cycle COP. These parameters are determined from measurement of nigh sky radiation, radiator surface temperature and ambient air temperature.

The measurement of these parameters is usually monitored using the following instruments;

The ambient air temperature, radiator surface temperature, room air temperature were all measured using a microcontroller based data logger. The data logger measures and stores this temperature at a predetermined interval. The data logger was interfaced in a Pentium III computer which indicates and records these

temperatures at 30 minutes intervals. In the format shown below in Fig. 3.7

File Help **Temperature Recorder**

8:23 am

38.5	45.46	46.68	36.12	37	32.45	34.53
Temp1	**Temp2**	**Temp3**	**Temp4**	**Temp5**	**Temp6**	**Temp7**
37,7	35.2	39.2	92	87.4	56.6	76.8
36.5	35.4	37.3	87.3	85.4	65.5	59.6
24	36.4	36.8	49.7	60.2	40.4	47.9
15.5	37	34.6	33.3	44.6	35.4	36.8
8.4	18.2	22.5	14.6	30.4	34.1	35.2

Fig 3.7, showing the displayed reading of the data logger.

In order to validate the accuracy of the data logger a separate manual reading was taken using a Hopmen digital thermometer. This thermometer measures the instantaneous temperatures, and the corresponding wet bulb temperatures. The values obtained are then processed using weather monitoring software called AIRMAP .It uses the values the dry bulb and wet bulb temperatures to evaluate the specific humidity, relative humidity, vapour pressure present etc and plot them on a psychometric chart.

The reading accuracy of the data logger is usually within 10 – 20% percent. The data logger was designed and fabricated at the Federal University of Technology, Owerri.

3.8.2 Experimental Methods

The experimental procedure for running a nocturnal cooling system includes the following;

1. Filling the thermal storage tank with water
2. Connecting the thermocouple terminals to the measurement panels.
3. Powering the electric pump to circulate water round the system 4 Acquisition of data.

3.9.0 Experimental Set Up

Fig 3.8 the experimental set up test rig

The nocturnal cooling system investigated has a flat plate sky radiator with aluminum sheet as major radiator component. The radiator has twelve cylindrical pipes which are joined together to a common feeder inlet and outlet pipe. An aluminum sheet of surface area 2.9 m^2 was used to cover the metallic pipes. The radiator was placed on the roof at tilt angle of 12 ^{o} .The cooling tank which serves as a thermal storage medium is a 100 litres PVC plastic tank placed inside the building. The pump used is a 0.25 hp "Petriolar" pump with a maximum discharge rate of 40 litres per minutes. The pump is connected to the storage tank and it draws water from the tank and circulates it round the radiator and the heat exchanger and back to the tank.

The building is a 3.0 x 3.0 x 2.5 m^3 cement block house which represents a standard bed room for an average man in the tropics. The roof was made from a special rust resistant material with high reflectivity. The highly reflective roof reduces the temperature of the interior of the room at a reasonable value during the day thereby enhancing the efficiency of the system during the night time. The building has an access door and two windows for cross ventilation. It has no ceiling in it; this is to facilitate direct heat exchange between the special roof and the

indoor air in addition to the cooling achieved using the heat exchanger unit. In the interior of the building, there is a 100 litres water tank and the 0.25 hp pump. The pump was connected directly to the water tank through a 2.5 mm PVC pipe. It draws water from the tank and circulates it through the radiator modules and the heat exchanger back to the tank.

The heat transfer fluid when pumped to the radiator by the nocturnal cooling process, the temperature of the water consequently reduces and it is stored in the water tank. This process is continuously repeated as long as the pump is working.

Temperature is measured using digital thermometers, thermocouples and wet and dry bulb thermometers at the inlet and the outlet of the radiators, and the inlet of the storage tank. In addition a locally designed data logger which measures and records the temperature at intervals was equally employed to record the temperature every five minutes. The thermocouples were soldered on the surface of the radiators at two different portions of the radiator (on the surface of the radiator and outlet where water flows out of the radiator intrinsic pipe network). There is another thermocouple or heat sensors attached to the storage tank (inside the storage tank) to measure the temperature of the heat transfer carrier.

All the temperature sensors were interfaced to a Pentium III computer which indicates and stores the temperatures at intervals. The relative humidity was evaluated from the temperatures obtained using a computer software called AIRMAP. This software computes the relative humidity, air vapour pressure, and other properties of the ambient air and plots them in a psychometric chart.

Series of experiments were conducted using the experimental test rig shown Fig. 4.3 above. The parameters measured included the ambient temperature, radiator

surface temperature, stored water temperature, test room temperature and the wet bulb temperatures.

3.9.1 Setting Up Of the System

The water tank was filled with water at the beginning and its initial temperature measured. The system was energized so that water can flow round the system in order to detect leaks. As soon as this was done with, the water was allowed to settle over again in the tank and its temperature measured again.

The same procedure was repeated again to ensure that the control system is responding to the set up. When this had been done, the data logger which was interfaced to the computer was energized to ensure that it was responding to the change in temperature. This was usually done by pressing the reset button so that the temperature displayed will display the accurate reading. This practice is essential because the data logger reads the default value at inception and if it is not reset, that reading will not be displayed on the computer screen.

3.9.2 Connecting the Thermocouple Terminals

The radiation surface, the water tank and the temperature of test rig room were monitored simultaneously. Thermocouples were attached to these points. The terminals were interfaced to the data logger, which is directly connected to the Pentium III computer.

3.10.0 Data Reduction Methods

The temperatures of the sky and radiator surface were used to compute the cooling power using equation (3.40).

$$Q_{net} = Q_{rad} + Q_{con} \qquad (3.40)$$

Where Q_{net} is the total available energy for cooling and Q_{con} is the total convective heat. The cooling produced by the sky radiator cools the space through the heat

exchanger unit and as well as the water stored in the storage tank. This is achieved by cooling the water stored in the tank and then using it to cool the space by passing it through the heat exchanger. In order to achieve this, control valves were used to by-pass the water flowing through the sky radiator so that the water circulates between the heat exchanger and the storage tank.

To be able to assess the feasibility of the system to substantially reduce the cooling load in a residential building, it is imperative that the performance of the system be evaluated to ascertain whether it is efficient to use the nocturnal cooling system as an alternative to conventional cooling system.

Some of the parameters adopted for the present study to assess the system performance include; the system COP and the climatic cooling potential.

3.10.1 Total Available Energy for Cooling

The energy available for cooling is obtained as a result of the heat loss of an uncovered night sky radiator which is caused by radiation and convection. Both heat transfer modes relates to each other in such a way that the total energy available for cooling is obtained as,

Cooling power of the radiator and it is given as

$$Q_{con}=hA(T_{amb}-T_{rad}). \quad (3.41)$$

The long –wave radiative cooling power Q_{rad} of a radiator with aperture area A and an emittance ε_r is given by Meir et al (2002) as:

$$Q_{rad} =A.\ \varepsilon_r(\sigma T^4_{rad} - R) \quad (3.42)$$

R is the long wave radiation incident on the radiator‟s surface and this is given by **Meir et al (2002)** as

$$R= \sigma\varepsilon T^4 a. \quad (3.43)$$

Therefore, the total energy available for nocturnal cooling is obtained by substituting equations (3.41), (3.42) and equation (3.43) into equation (3.40) as

$$Q_{net} = hA(T_{amb} - T_{rad}). + A.\ \varepsilon_r(\sigma T^4_{rad} - \sigma\varepsilon_r T^4_{amb}) \qquad (3.44)$$

3.10.2 Maximum Possible Cooling

The maximum cooling available is obtained when the sky radiator is completely covered, to avoid heat gain due to convection.

Thus $Q_{con}=0$

Therefore, equation (3.44) is reduced to the form:

$$Q_{max} = A\square\square_r(T_{rad}^4 - \sigma\varepsilon r T^4 a\ ) \qquad (3.45)$$

3.10.3 Useful Energy Delivered To the Water

The useful energy delivered to the water to cool it is determined using the equation *(3.46)*

$$Q_w = \dot{\ }_w Cp\ \Delta T. \qquad (3.46)$$

Thus, the quantity of water (m) placed in thermal contact with the radiator undergoes a temperature reduction ΔT intermittently. Therefore, the equation (3.46) can further be expanded as shown below:

$$Q = \dot{\ } Cp(T_{w2} - T_{w1}) \qquad (3.47)$$

Where T_{w2} and T_{w1} are the temperatures of the water recorded at the end and beginning of the experiment.

3.10.4 Total Useful Energy Available For Space Cooling

The useful energy available for space cooling is determined using equation (3.46) as: $Q_{sp} = \dot{\ }_a\ Cp\ \Delta T.$ (3.47)

Thus (m) is the mass flow rate of air in the room and ΔT is the change in the room temperature. Therefore, equation (3.47) can further be expanded as shown below:

$$Q = m\,Cp\,(T_{r2} - T_{r1}) \qquad (3.48)$$

Where T_{r2} and T_{r1} are the temperatures of room recorded at the end and beginning of the experiment.

From the above considerations the performance indices of the nocturnal cooling system may be obtained as follows:

(i)Radiator efficiency,

$$\eta = Q_{net}/Q_{max} \qquad (3.49)$$

(ii)Useful coefficient of performance (water cooling)

$$COP_u = Q_w/Q_{net} \qquad (3.50)$$

(iii)Useful coefficient of performance (space cooling)

$$COP_{sp} = Q_{sp}/Q_{net} \qquad (3.51)$$

3.10.5 Calculation of the Climatic Cooling Potential

Climatic cooling potential measured in degree-hour is used to characterize the impact of the climate on the thermal behavior of a building **Nikolai et al(2006).** In this study, the climatic cooling potential (CCP) for the space cooling is used to characterize the level of cooling to be achieved under the climatic conditions in Owerri.

To evaluate the CCP for the present study, the sensible heat gain (solar heat gain) by the building during the day, which is at its maximum at the beginning of the night time cooling, and its minimum value at the dawn of the morning, is computed using the relevant equations.

Therefore, to calculate the CCP, the thermal capacity of the building is assumed to be sufficiently high so as not to limit the heat storage process. If the

building is in the same state after one cycle, the heat which charges the building structure Q_{charge} during the time it is occupied t, is equal to the heat released through night cooling $Q_{cooling}$ Therefore, the mean heat flux during the storage process q, per room area can then be calculated as

$$q = \frac{Q}{At} = \frac{\dot{} \; CpCCP}{A \; t_{occ}} \qquad (3.52)$$

The effective mass flow rate $\dot{}$ is given by **Allard (1998)** as eqn (3.14)

$$\dot{} = AH\,R\,\eta\,\rho$$

Where H is the height of the room, R is the air change rate and η is the temperature efficiency, defined as

$$\eta = (T_{out} - T_{amb}) / (T_b - T_{amb}),$$

This takes account of the fact that the temperature of the air flowing out, Tout is lower than the temperature of the building T_b. The density and the specific heat capacity of air are taken as ρ=1.2 kg/ m ^{3} and C_p = 1000 J/kg K.

Since the height of the room is 2.5 m, a constant effective air change rate given **by Nikolai et al (2006)** as R η =6 h^{-1}

Assuming that the average internal heat gain of 20 W/m^2 and solar gain of 30 W/m^2 was accumulated by the building during the day, the CCP needed to discharge the stored heat is 71 K h per day. For building in the tropics, internal heat gain of the building and solar heat gain are usually higher; the sum of the internal heat gain and solar heat gain is usually in the range of 250 W/m^2-450 W/m^2 for Owerri climate **Anyanwu and Oteh(2003).** Year round prediction of the solar radiation intensity in Owerri can be analytically obtained using appropriate mathematical relation as specified by **Beckman and Duffie, (1991).**

For the same building configuration as the one described by **Nikolai et el (2006),** the CCP required to discharge the stored heat is 400 K h which is quite enormous. The significant of the above result is that more energy is exhausted in trying to bring about cooling in the building. Places with lower values of CCP will achieve more space cooling.

CHAPTER FOUR

Results and Discussion

4.1 Experimental Observations

Apart from the experimental methods explained in the previous section, the instruction for use of this nocturnal cooling system include: cleaning the surface of the radiator at least once every week and filling the poly vinyl chloride (PVC) storage tank with water at the beginning of the nighttime cooling operation.

The operation of the nocturnal cooling system started at 6 pm on 29th March, 2010. Soon after (about 5 min) the water was circulated through the system, it was notice that water was leaking profusely from some joints. As a result of this, the pump was immediately stopped to correct all observed leakages at the joints. The leakages were arrested by sealing with plumbing gum. Thereafter, the system was allowed to run for the whole night.

The data logger which was interfaced with a computer was observed to be giving readings above 35 ^{o} C which was quite high during night time in Owerri. This high temperature measurement obtained was as a result of faulty instrument and it was immediately rectified .To ensure that the readings were accurate and reliable; an electrical thermocouple was brought to record the temperature

alongside the data logger at the experimental test rig. Though there was an initial excessively high temperature due to the faulty meter, it was observed that the radiator surface temperature was consistently lower than the ambient air temperature. Another observation for the first night is that the ambient air temperatures around the test building, and other areas contiguous to the test site, such as the students‟ hostel environment and around the mechanical engineering administrative building were all different.

The temperature of the circulating water was noticed to have increased above the initial temperature. Ordinarily it should have been lower after a while because the ambient air temperature at this point was gradually reducing. However, when the pump was stopped from circulating water round the system, the temperature started dropping gradually.

This is very indicative of the fact that the heat accumulated by the radiator component was yet to be dissipated at that time of the day hence it affected the temperature of the water passing through it. The following day (30th march, 2010), no positive change was observed especially in the depression of the surface temperature of the radiator below the ambient air.

Similarly, the water temperature was observed to be constant throughout without any appreciable change in its temperature. The radiator surface temperature was consistently lower than the ambient air throughout the night. The weather was observed to be initially overcast and as time progressed it became partially clear.

4.2 Presentation of Results

The results of the experimental observations are presented graphically in Figs. 4.14.50. Figs 4.1- 4.33 illustrate the results for the nocturnal cooling obtained using the system designed. They show the measured radiator surface temperature, test

room temperature, ambient air temperature, stored water temperature and the relative humidity. The system operating conditions and the performances are also presented in tables 4.1 and 4.2. The experimental period spanned over one year.

During this time, there were clear days, some overcast and rainy days. The analysis of the results obtained

4.4 System Performances

Table 4.1 Measured System Performances for (29/03/2010)

Time	T_{rad}	T_{amb}	T_{room}	T_{dewp}	RH	T_{water}	q_{net}(W/m2)	COP(sp)	COP(W)
6.00 pm	25.3	24.1	27.3	23.5	80.6	26.6	27.47247	0.012438	0.0133
6.30 pm	25	24.4	27	23.1	83.4	25.9	24.37094	0.011111	0.0134
7.00 pm	24.8	24.4	26.7	23.1	83.6	25.3	19.73182	0.025197	0.0123
7.30 pm	24.5	24.3	26.9	23.1	84.5	25	21.21763	0.071322	0.0141
8.00 pm	24.2	24.2	26.8	23	85.3	24.8	16.64582	0.012516	0.0134
8.30 pm	24.1	24.3	24.8	22.9	86.5	24.5	15.09451	0.018318	0.0144
9.00 pm	24	24.4	25.6	22.9	87.7	24.2	15.04893	0.025596	0.0211
9.30 pm	24	24.3	24.8	22.9	88.1	24.1	7.505498	0.011136	0.0222
10.00 pm	24.2	24.1	24.7	22.8	88.2	24	8.983894	0.017084	0.032
10.30 pm	24.2	23.5	24.6	22.7	88.3	24	14.88262	0.054665	0.012
11.00 pm	23.7	23.5	24.3	22.6	88.5	24.2	17.76894	0.054665	0.023
12.00	23.7	23.8	24.3	22.5	88.7	24.2	20.68845	0.070688	0.022
12.30 am	23.1	23.2	24.6	22.5	89.7	23.7	19.18147	0.011156	0.0133
1.00 am	21.7	22.7	24.5	22.4	89.8	23.7	19.12312	0.011156	0.0212
1.30 am	22.4	22.6	24.4	22.3	90.2	23.1	23.42875	0.020760	0.0122
2.00 am	22.8	23.1	24.4	22.2	90.3	23.7	19.00677	0.011156	0.0133
2.30 am	23	23.1	24	22.1	90.4	23.4	20.45843	0.012685	0.0144

3.00 am	22.1	22.8	23.9	22	90.5	23.8	17.51792	0.012703	0.0244
3.30 am	22.2	22.5	23.8	21.9	90.5	23	17.50008	0.028465	0.0234
4.00 am	22.6	22.9	23.7	21.9	90.8	23.1	17.48225	0.029343	0.0122
4.30 am	21.7	22.5	23.2	21.8	91.2	23.2	17.48225	0.022487	0.0133
5.00 am	21.9	22.5	23.8	21.8	92.0	22.6	14.56853	0.018488	0.0134
5.30 am	21.8	22.4	23.5	21.8	94.5	22.7	16.01722	0.015207	0.0134
6.00 am	21.9	22	23	21.7	95.3	22.9	14.58338	0.015164	0.0135

Table 4.2 comparison with other experimental test results

Name of designer(s)	**Country**	**Cooling power(W/m^2)**	**Ta-Tr**	**Area of radiator (m^2)**	Application
Meir et al	Norway	30	5 °C	5.2	Space cooling
Nammont	Thailand	25	1.8 °C	6.0	Space cooling
ITO AND MIURA	JAPAN	35	3 °C	6.0	Water cooling
DANNY PARKER	USA	55	1.4 °C	22.9	Space cooling
THIS STUDY	NIGERIA	40	2.3 °C	2.9	Space cooling
Dobson	NAMIBIA	35	5 °C	6.0	Water cooling
Bassindowa et al	SAUDI ARABIA	40	4 °C	5.0	Space cooling
Khedari, et al	THAILAND	40	1-6 °C	3.9	Space cooling

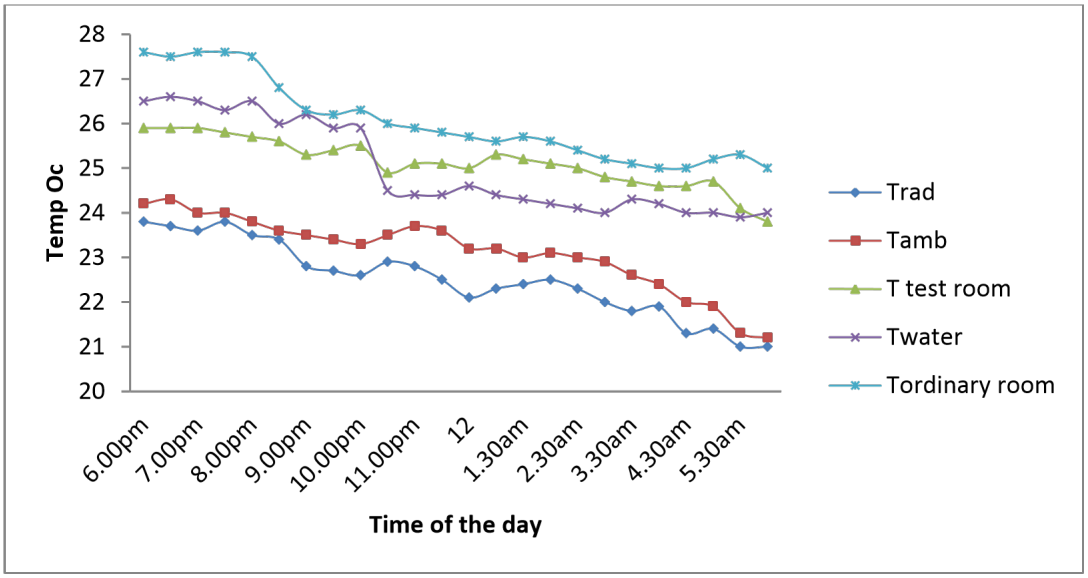


Fig 4.1 the variation of the temperature with time of the day on 29th march – 30th March l 2010.

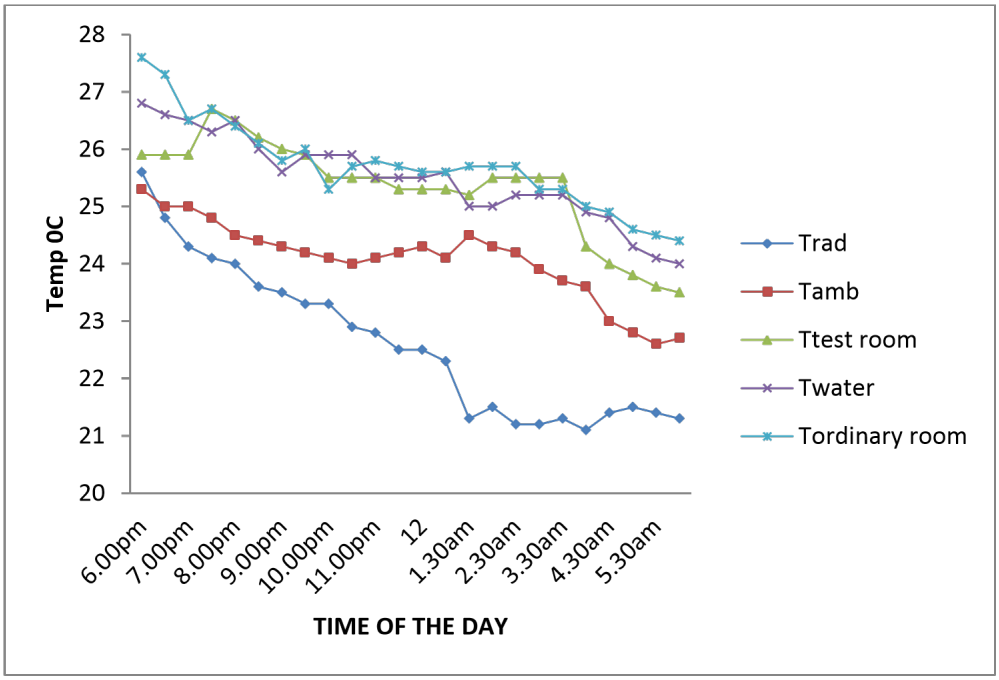
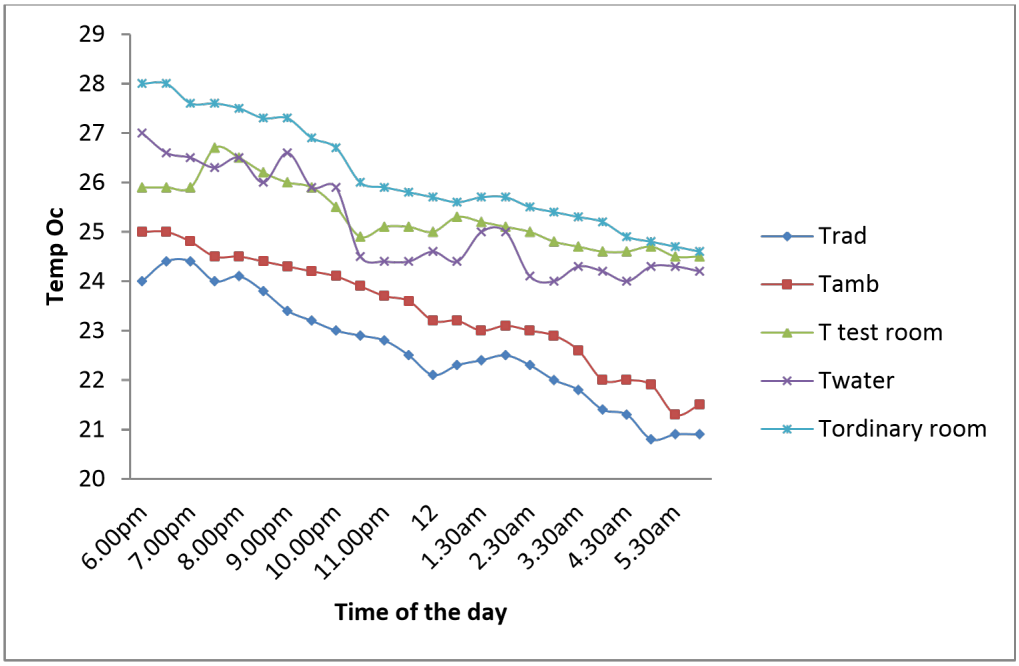


Fig. 4.2 Radiative cooling experiments carried out at the federal University of technology, Owerri during April–2nd-3rd 2010.



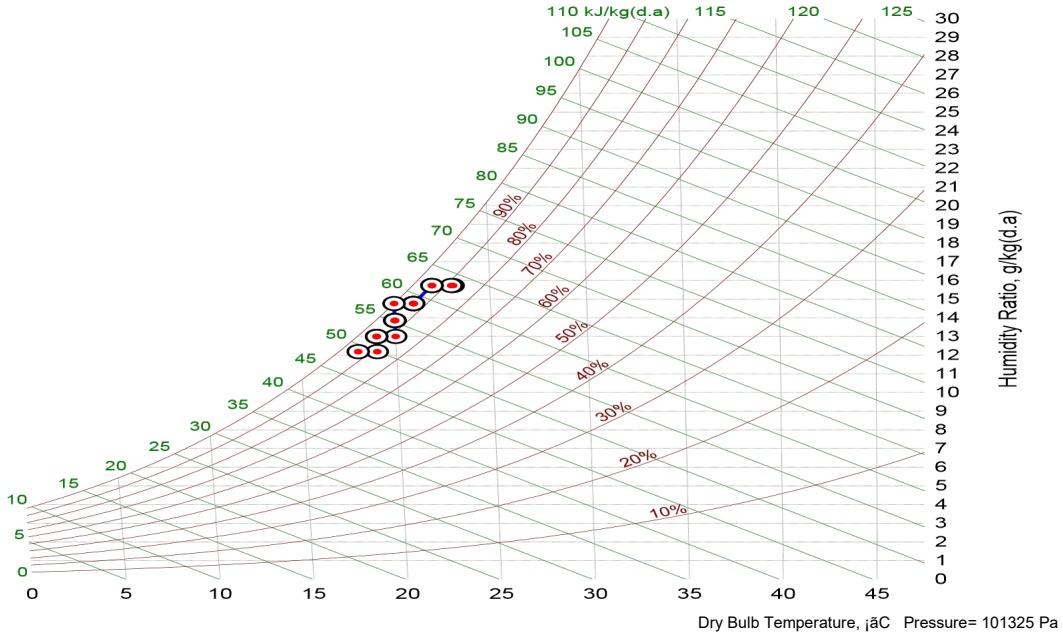


Fig 4.3 the variation of the temperature with time of the day on 4th- 5th April, 2010.

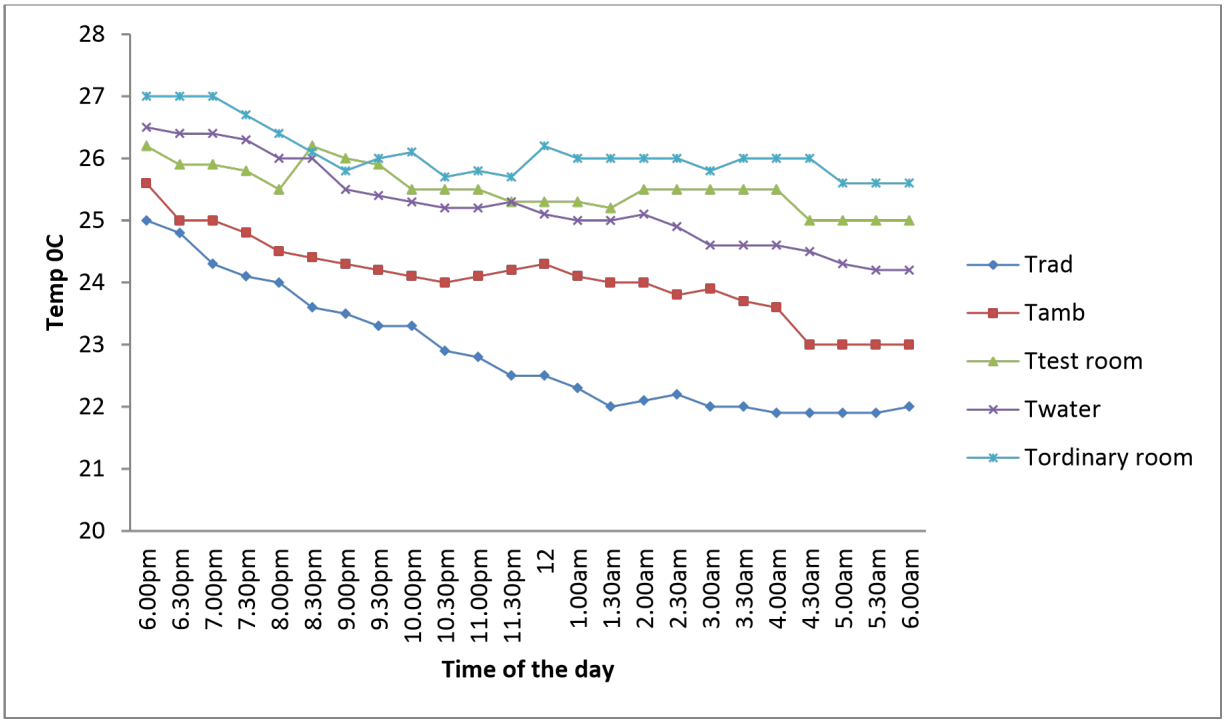
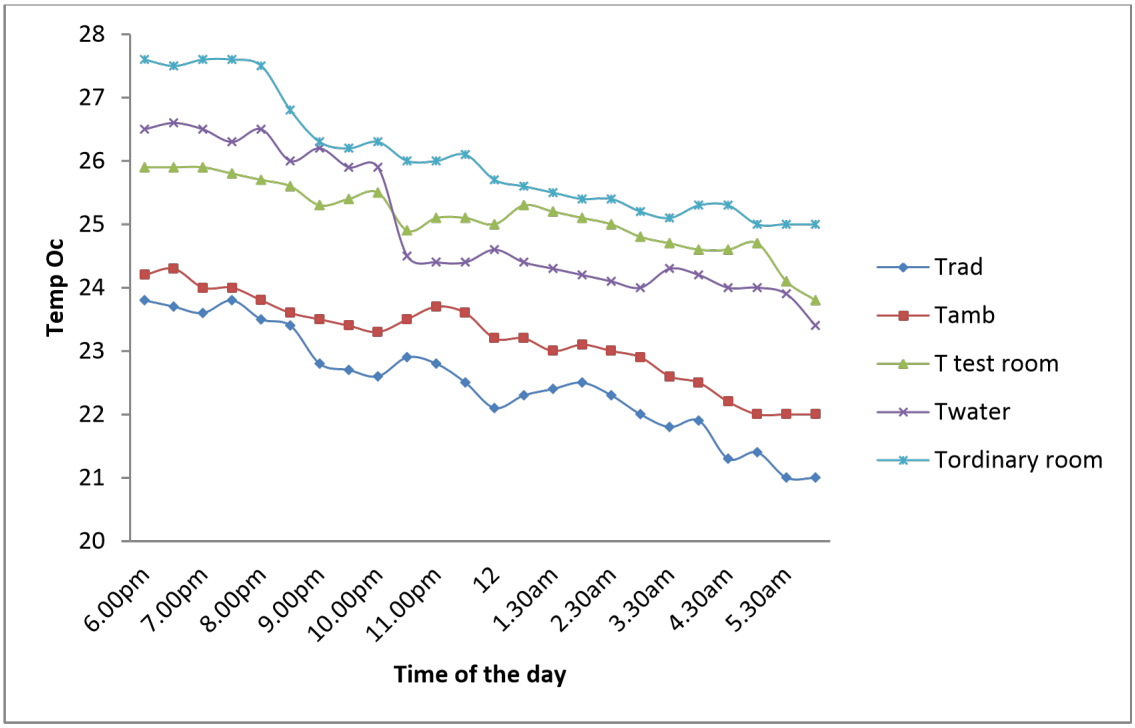


Fig 4.4 the variation of the temperature with time of the day on 5th -6th April 2010.



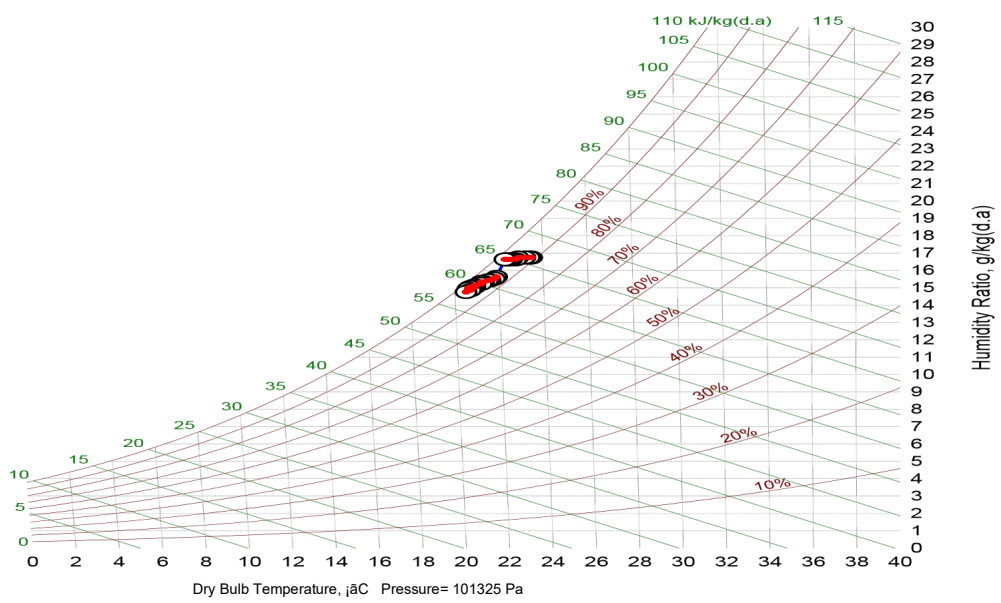


Fig. 4.5 Radiative cooling experiments carried out at the Federal University Of Technology, Owerri During April–10-11 2010.

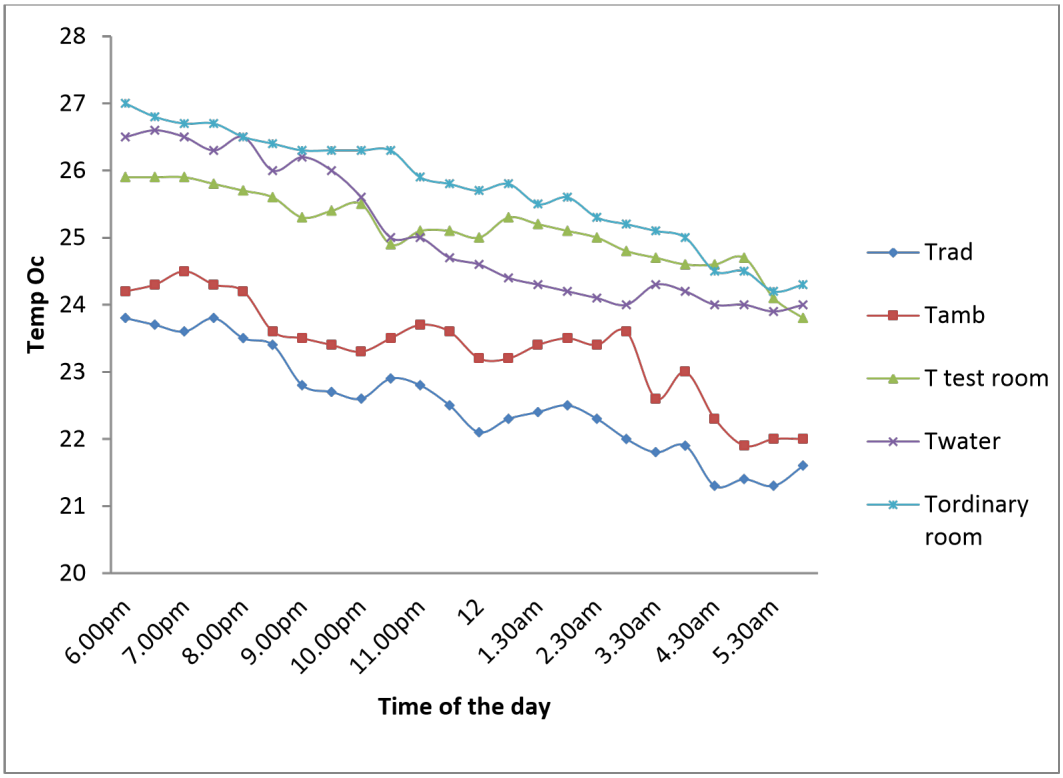
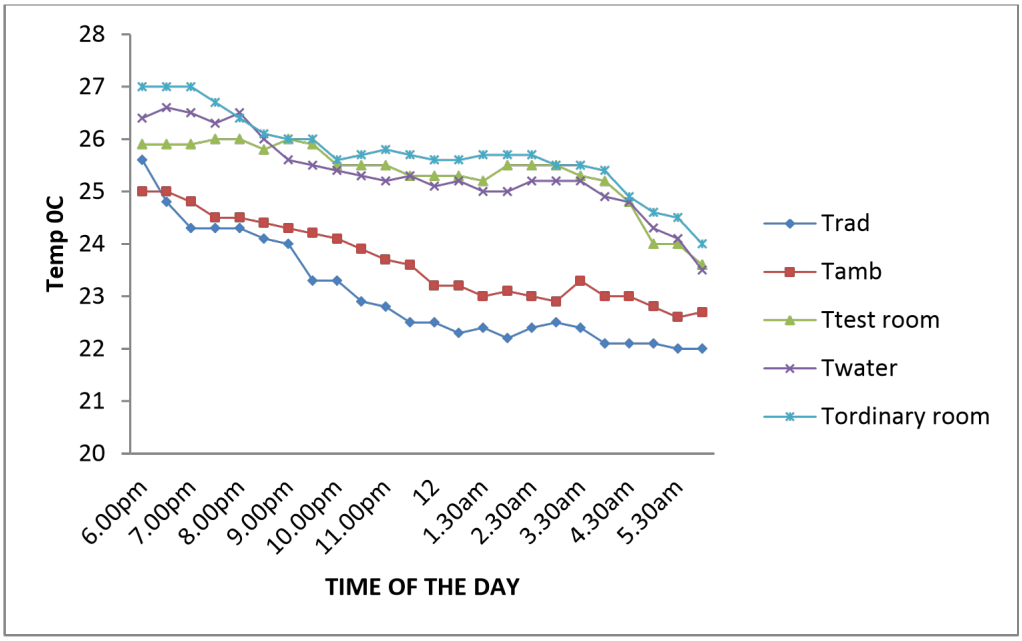


Fig 4.6 *Radiative cooling experiments carried out at the Federal University of Technology, Owerri on April–12-13, 2010.*



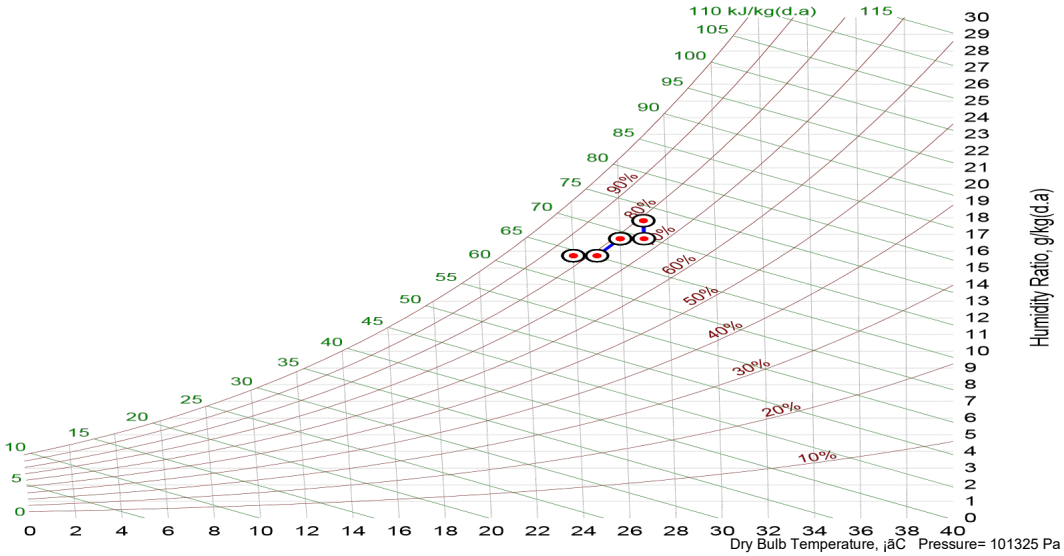
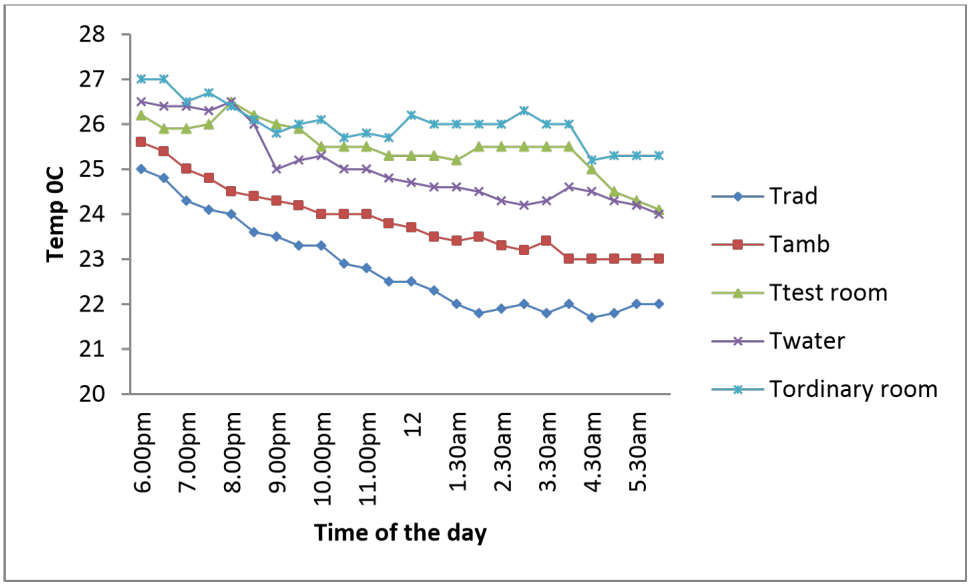


Fig 4.7 *Radiative cooling experiments carried out at the Federal University of Technology, Owerri on April–14-15 2010.*



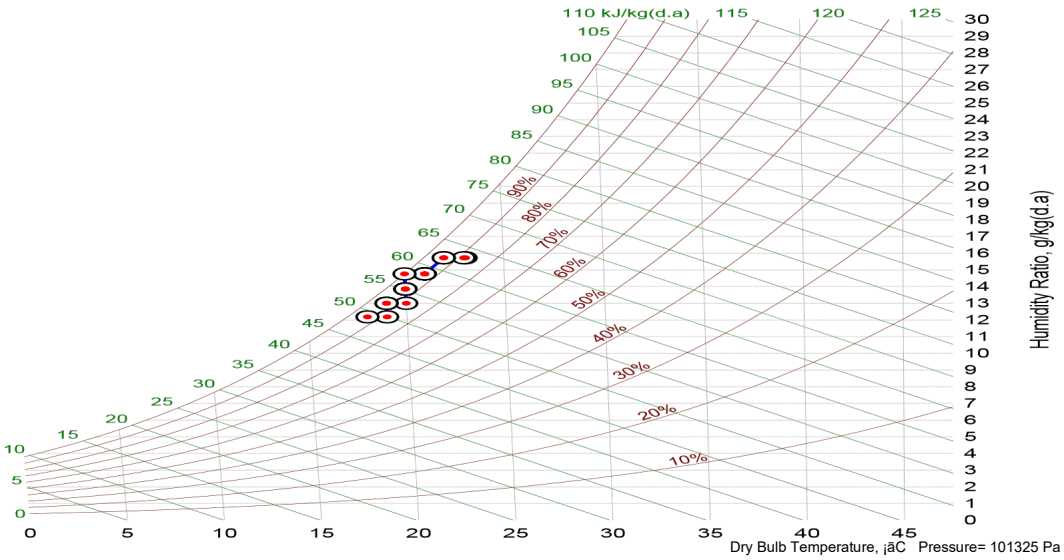
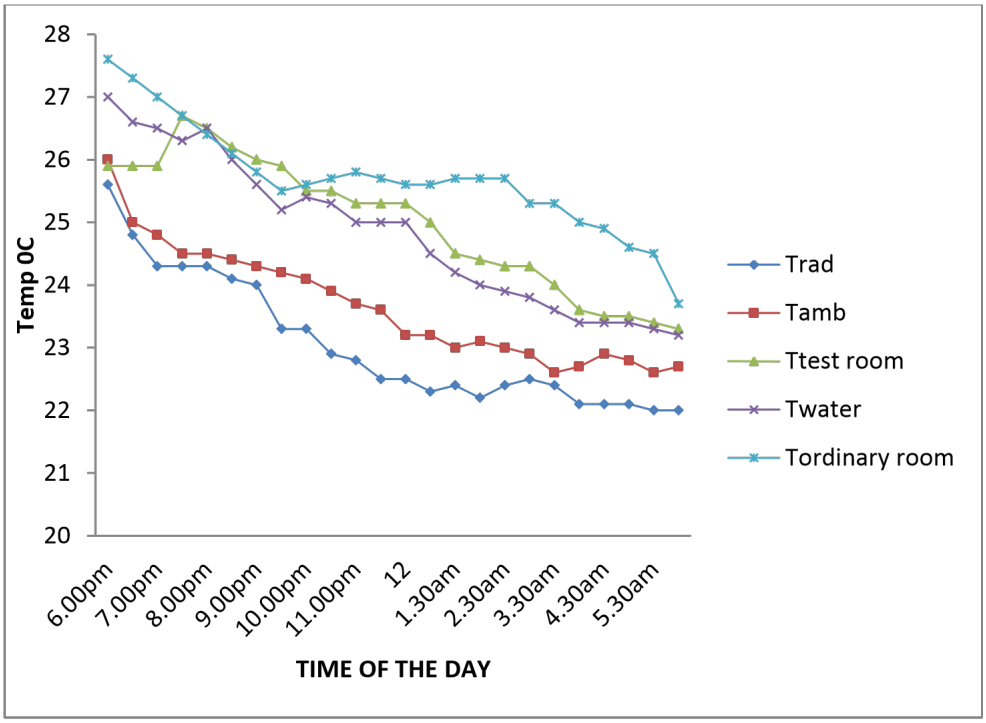


Fig 4.8 *Radiative cooling experiments carried out at the Federal University of Technology, Owerri on April–19-20 2010.*



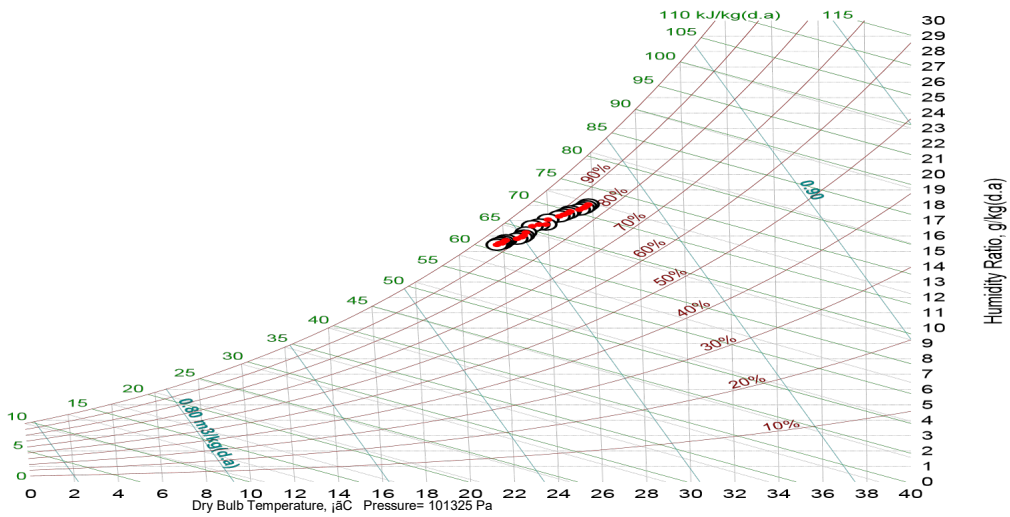
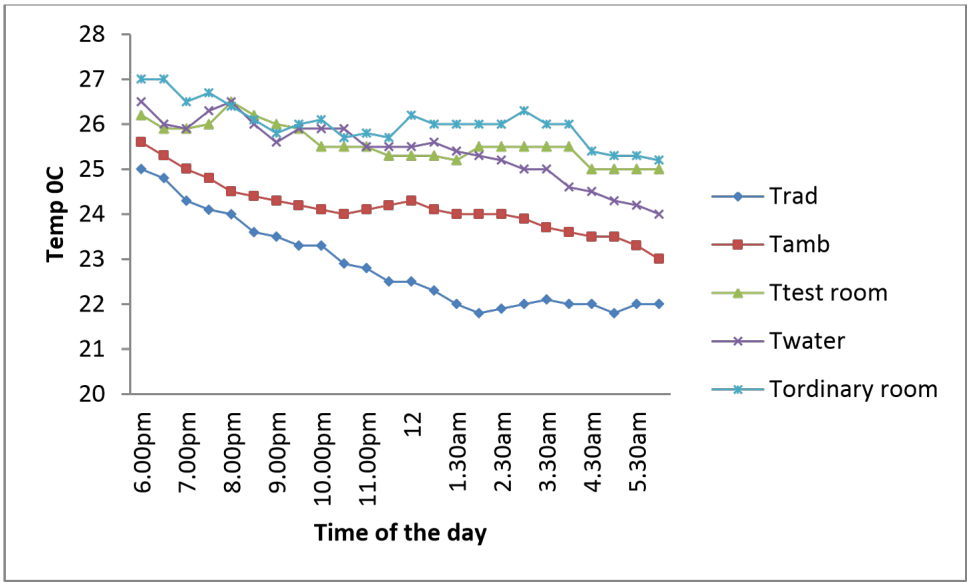


Fig 4.9 *Radiative cooling experiments carried out at the Federal University Of Technology, Owerri on April–21-22 2010.*



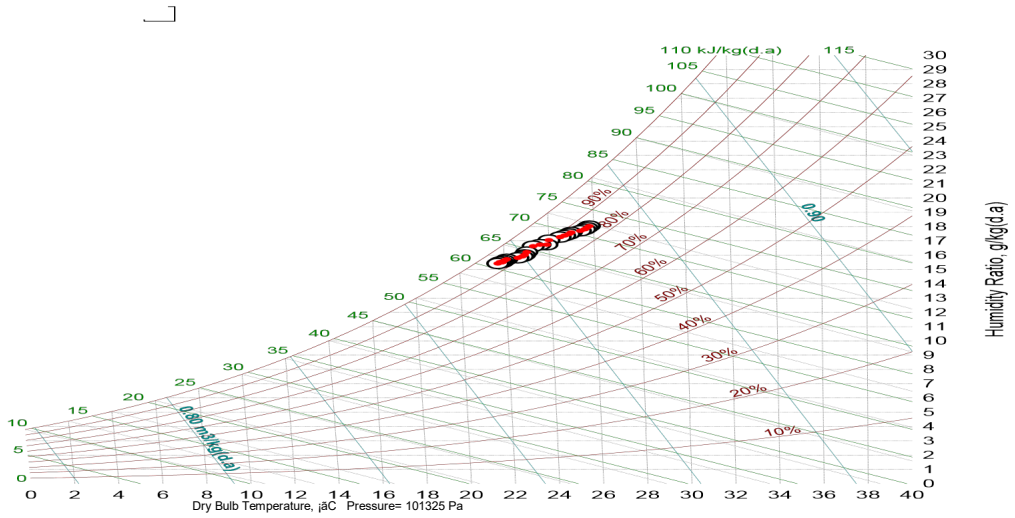


Fig 4.10 *Radiative cooling experiments carried out at the Federal University of Technology, Owerri on April–22-23 2010.*

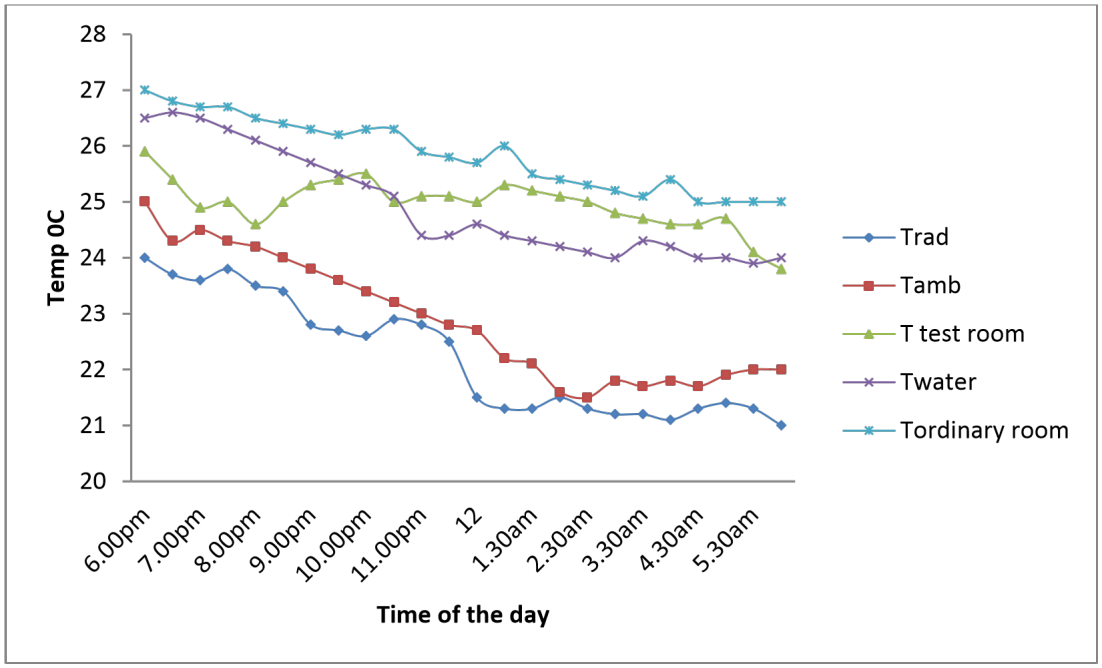


Fig 4.11 Radiative cooling experiments carried out at the Federal University of Technology, Owerri on April–29-30 2010

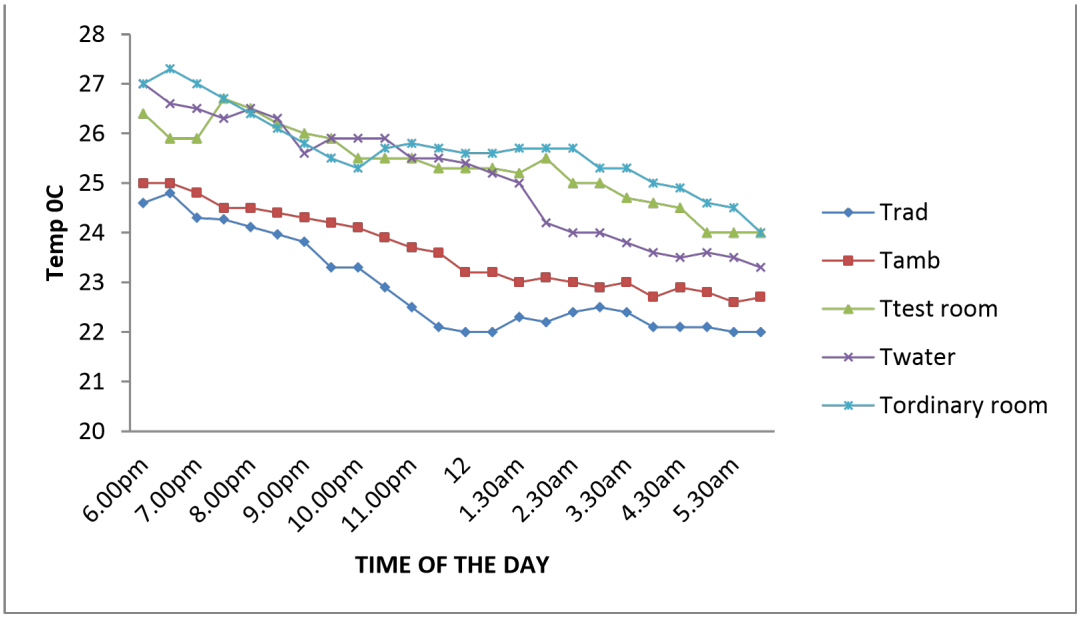


Fig 4.12 *Radiative cooling experiments carried out at the federal University of technology, Owerri during May 1-2, 2010.*

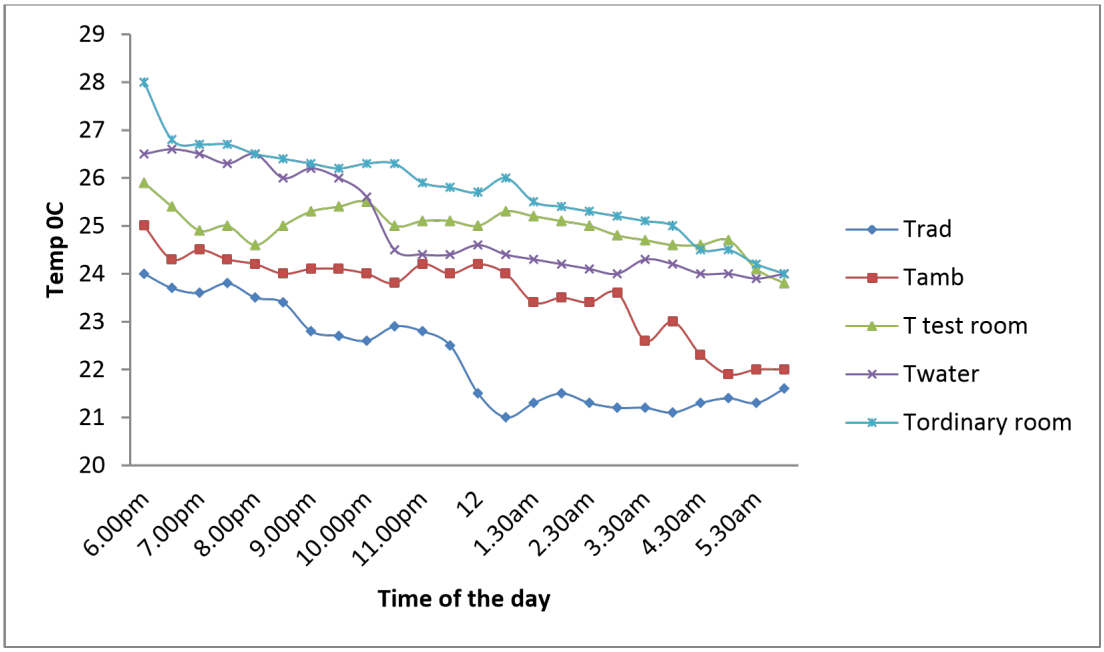
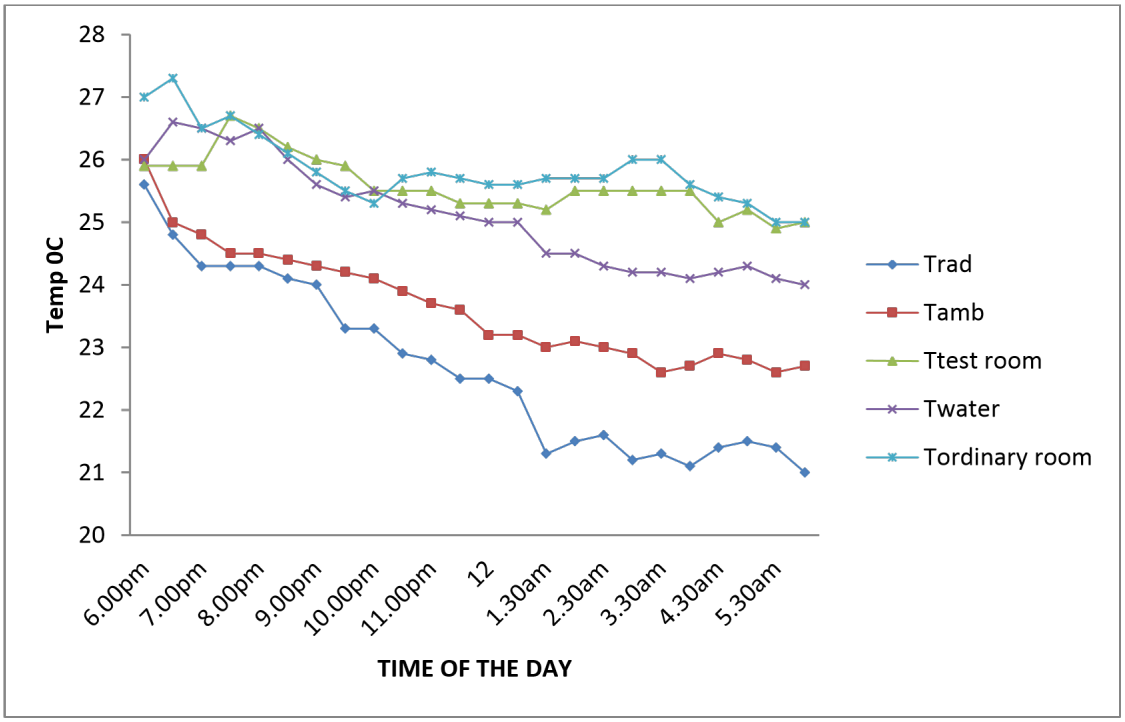


Fig 4.13 *Radiative cooling experiments carried out at the Federal University of*

Technology, Owerri on May4-5, 2010.



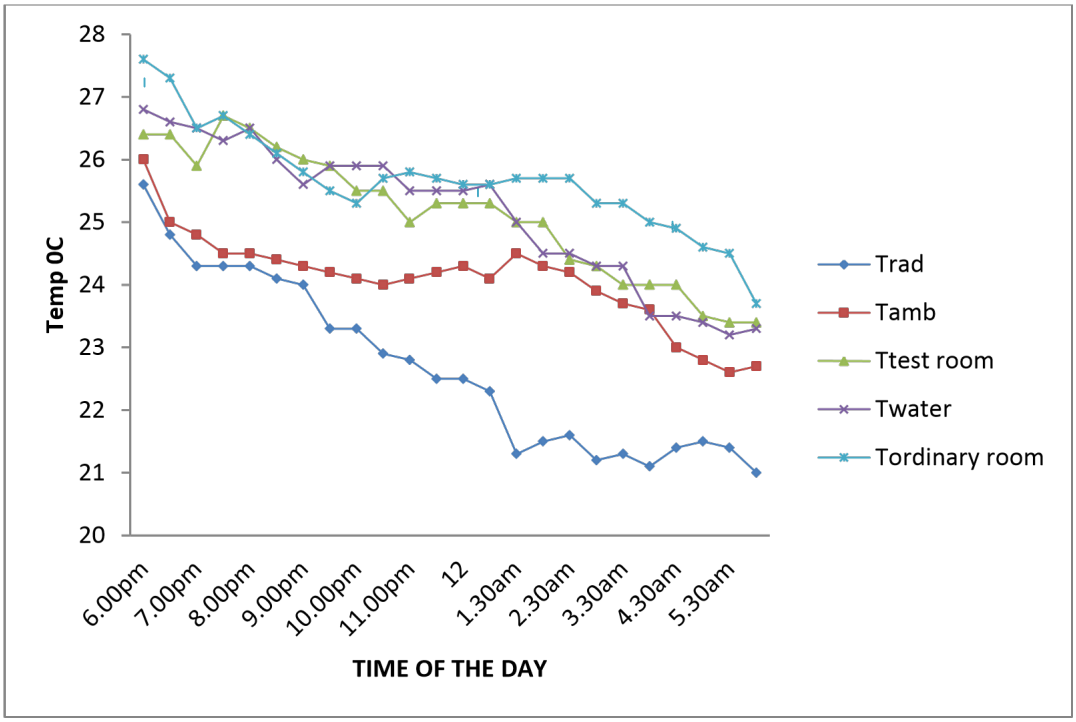


Fig 4.14 Radiative *cooling experiments carried out at the Federal University of Technology, Owerri on June –12-13 2010.*

Fig 4.15 Radiative cooling experiments carried out at The Federal University of

Technology, Owerri on June –14-15 2010.

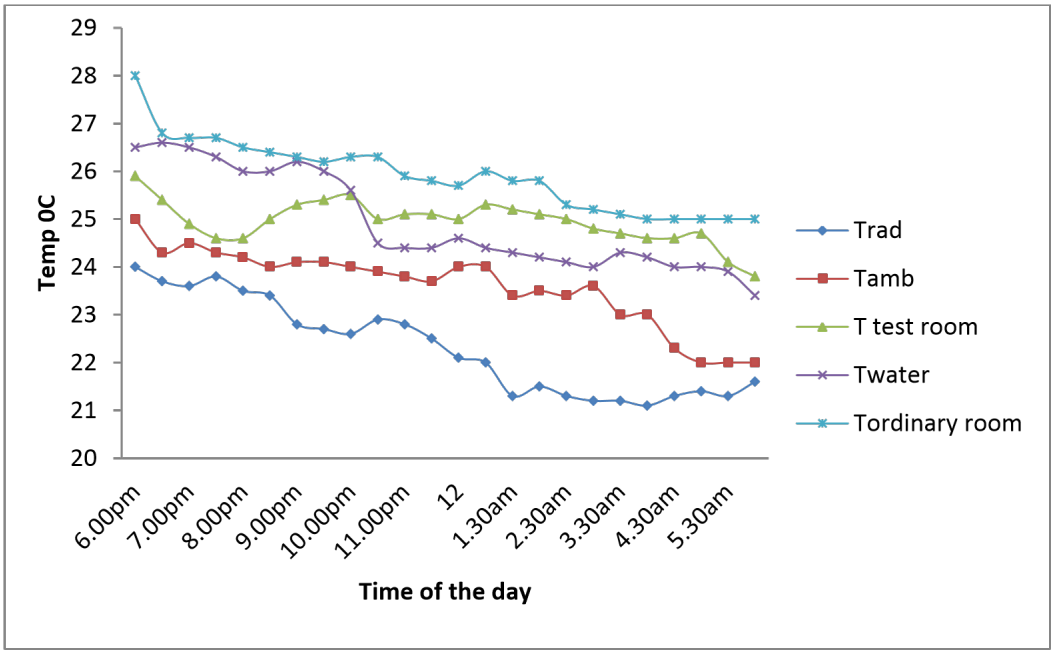


Fig 4.16 *Radiative cooling experiments carried out at the Federal University of Technology, Owerri on June –16-17, 2010.*

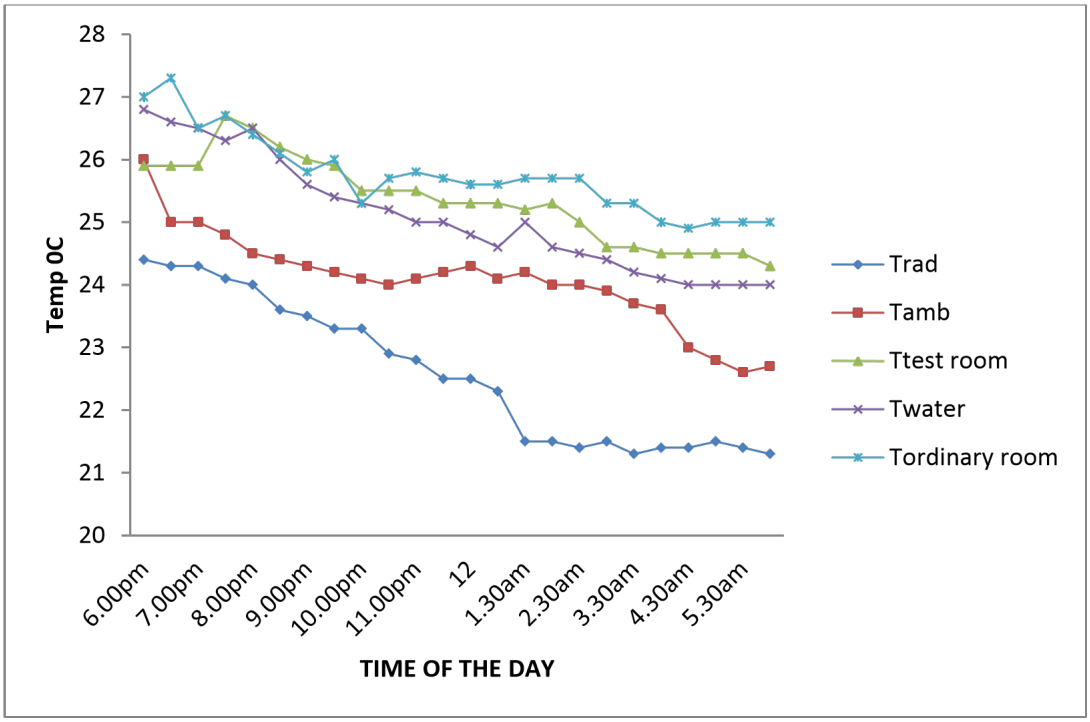


Fig 4.17 *Radiative cooling experiments carried out at the federal University of*

technology, Owerri on June –22-23, 2010.

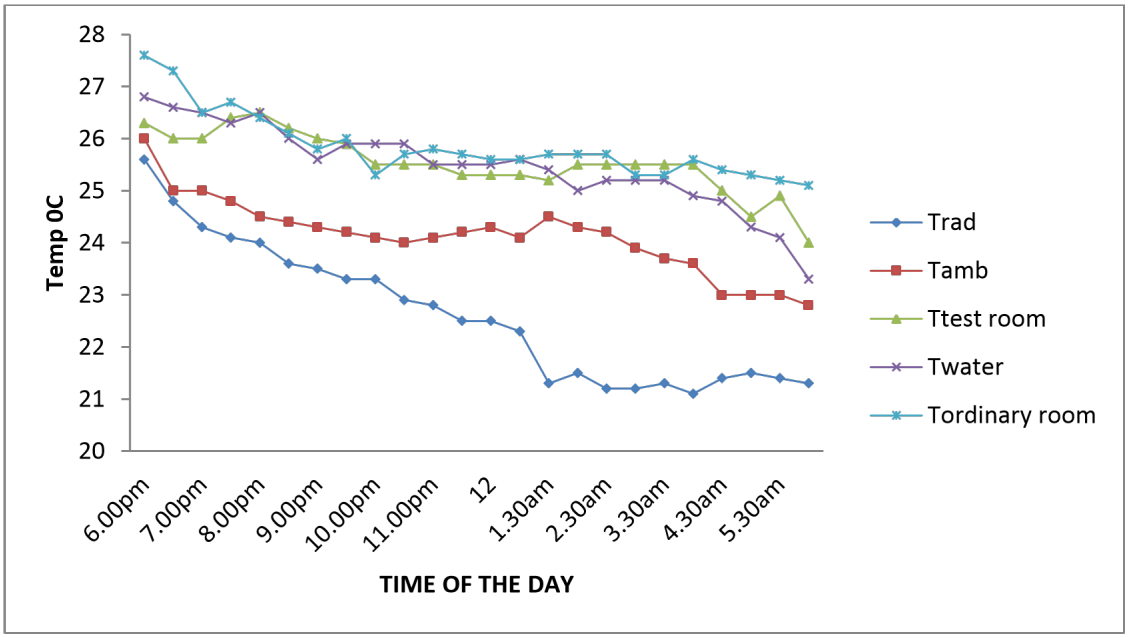


Fig 4.18 *Radiative cooling experiments carried out at the federal University of technology, Owerri on June –26-27, 2010.*

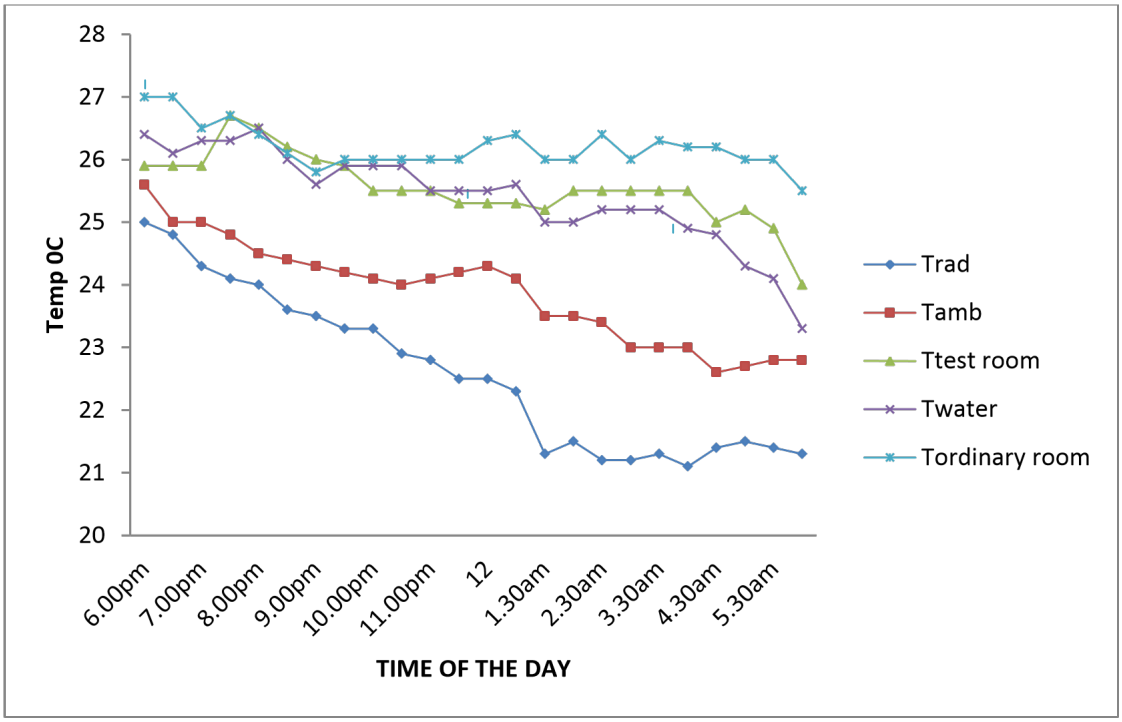


Fig 4.19 *Radiative cooling experiments carried out at the Federal University of*

Technology, Owerri On July –2-3, 2010.

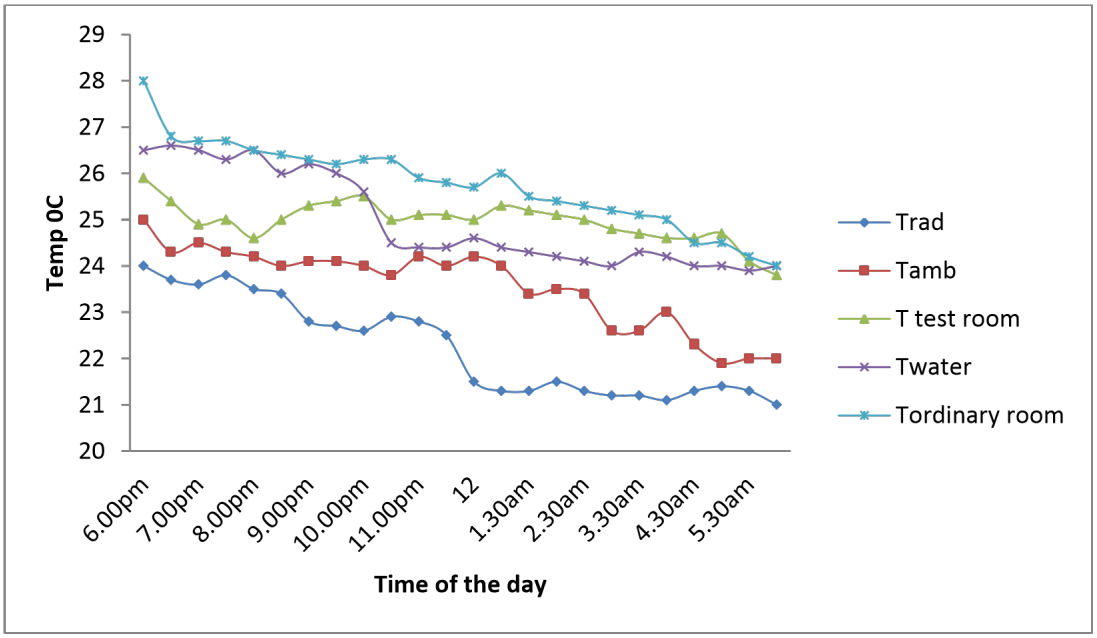


Fig 4.20 Radiative cooling experiments carried out at The Federal University of Technology, Owerri on July –12-13, 2010.

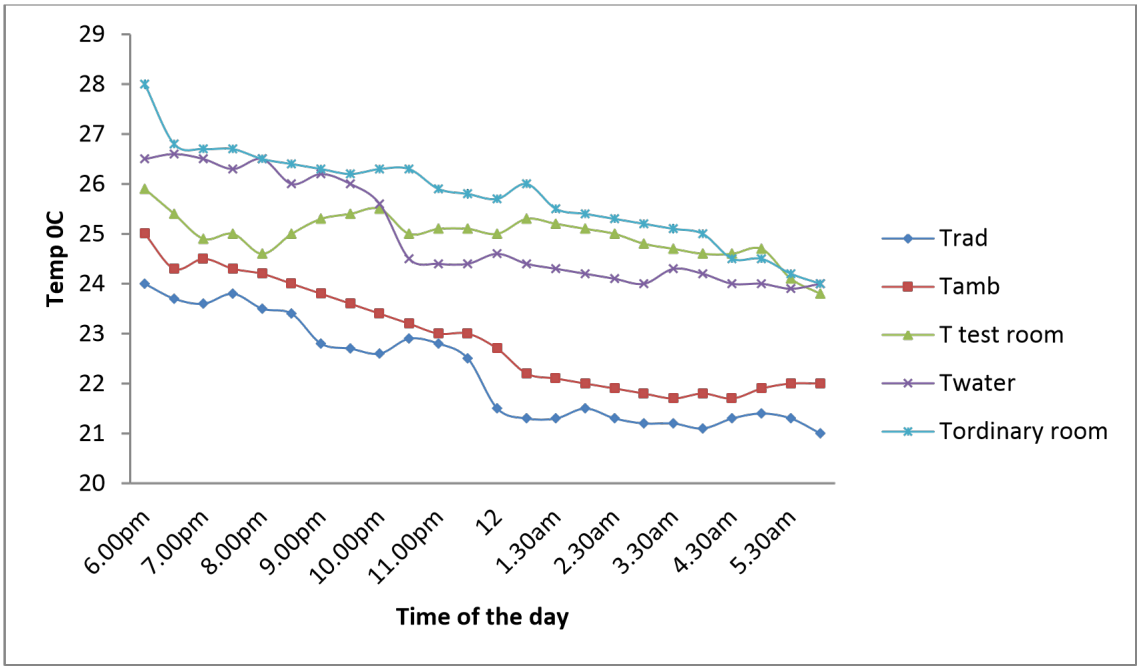


Fig 4.21 *Radiative Cooling Experiments carried out at The Federal University of Technology, Owerri on July –25-26, 2010.*

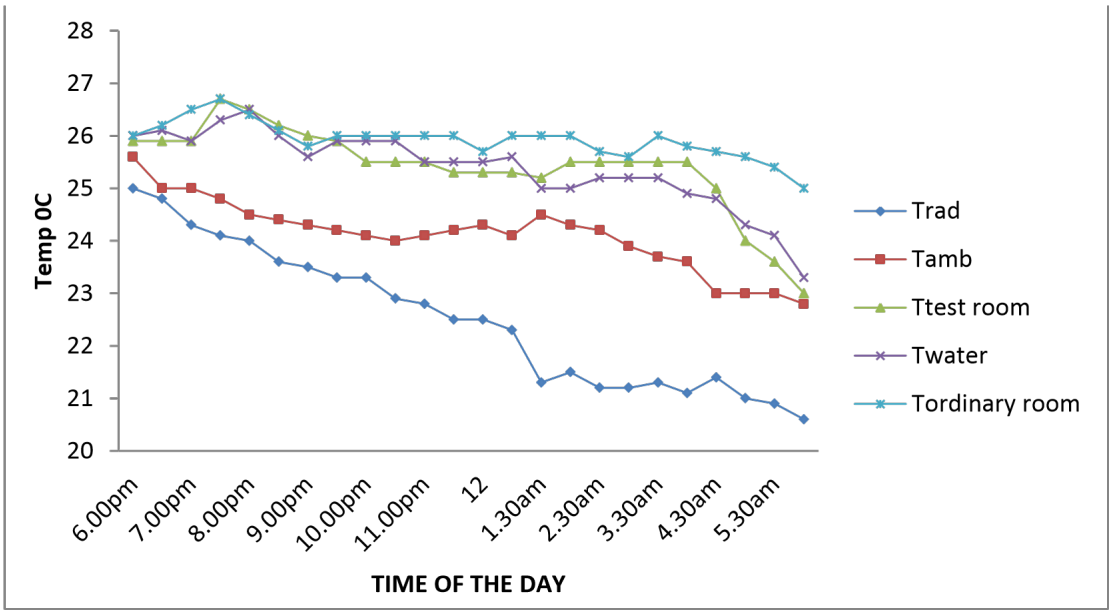


Fig 4.22 Radiative Cooling Experiments Carried Out At The Federal University Of Technology, Owerri on August –6-7, 2010.

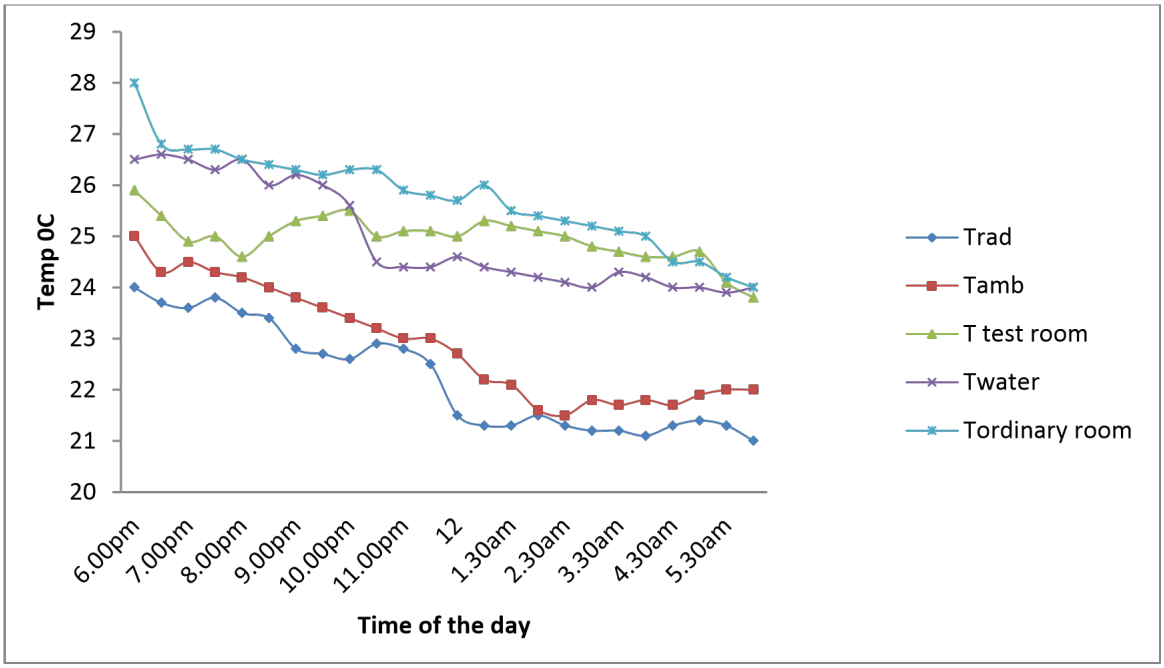


Fig 4.23 *Radiative Cooling Experiments Carried Out At The Federal University Of Technology, Owerri on August –12-13 2010.*

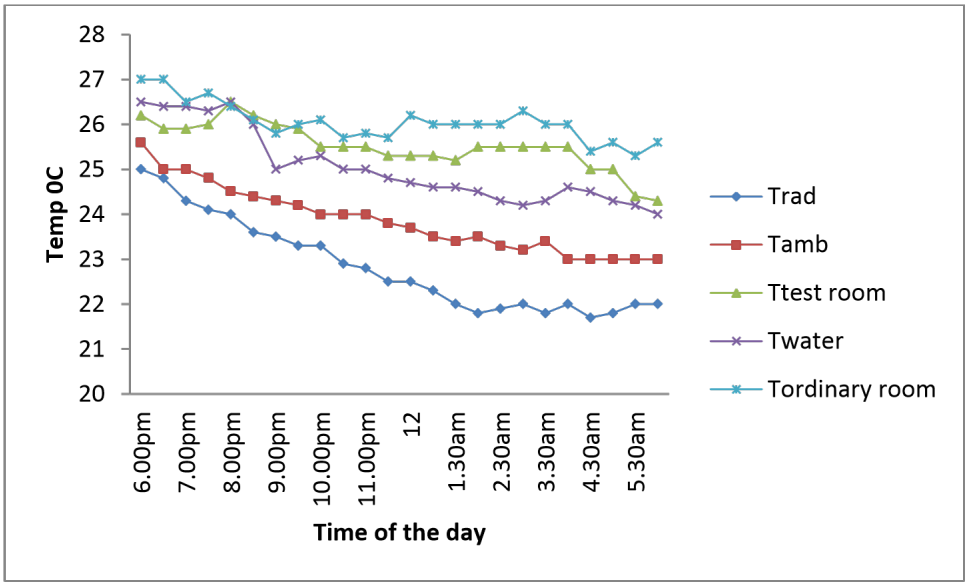


Fig 4.24 *Radiative Cooling Experiments Carried Out At The Federal University Of Technology, Owerri on August –15-16, 2010.*

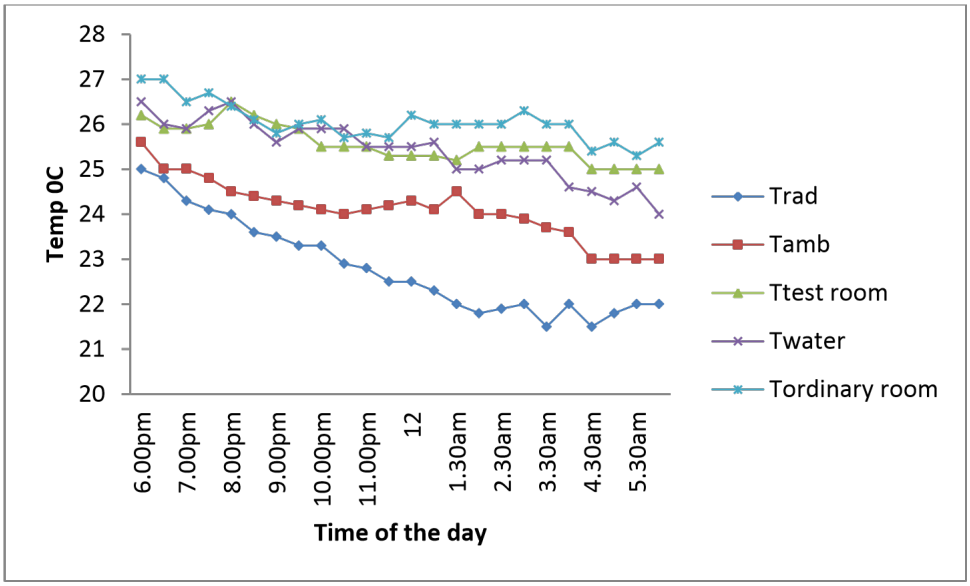
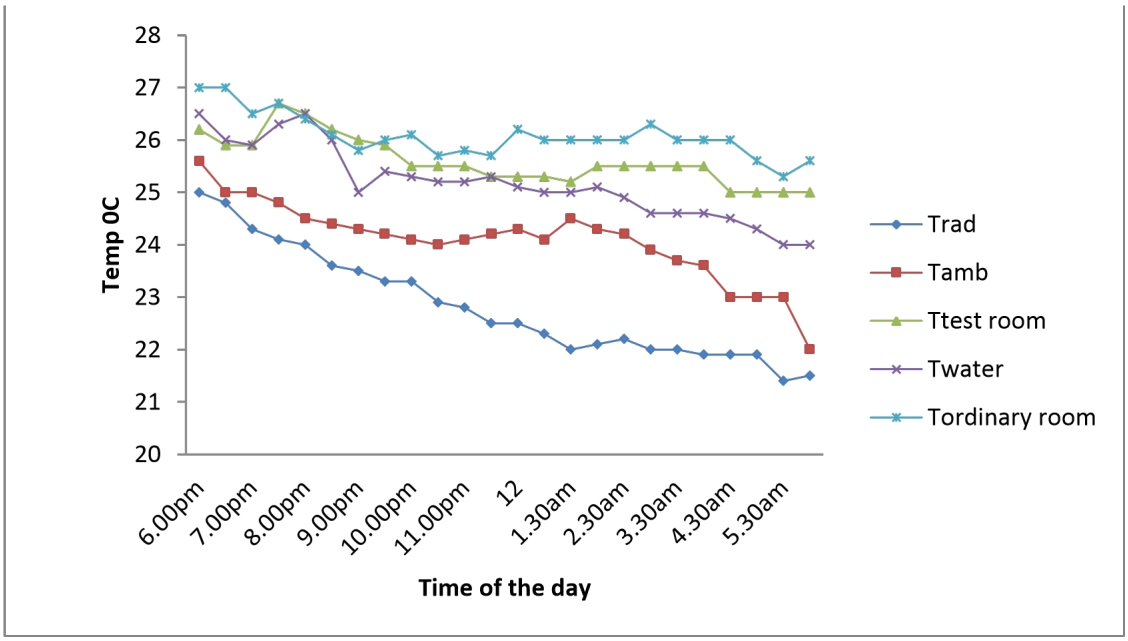


Fig 4.25 *Radiative Cooling Experiments Carried Out At The Federal University Of Technology, Owerri on August 20-21, 2010.*



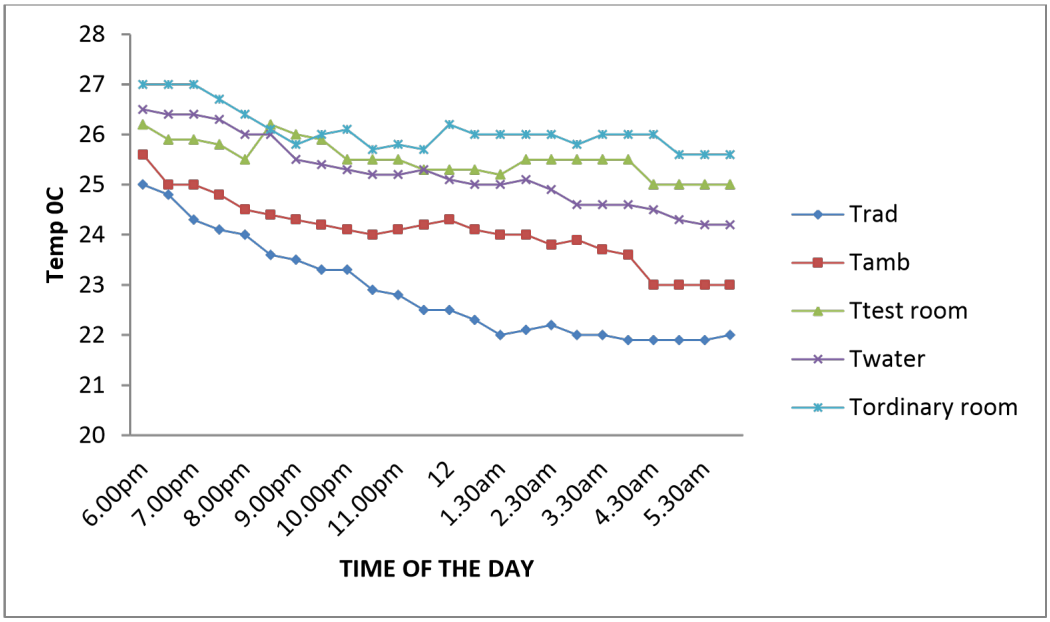


Fig 4.26 *Radiative Cooling Experiments carried out at the Federal University of Technology, Owerri on August 25-26, 2010.*

Fig 4.27 *Radiative Cooling Experiments carried out at the Federal University Of Technology, Owerri on September 2-3, 2010.*

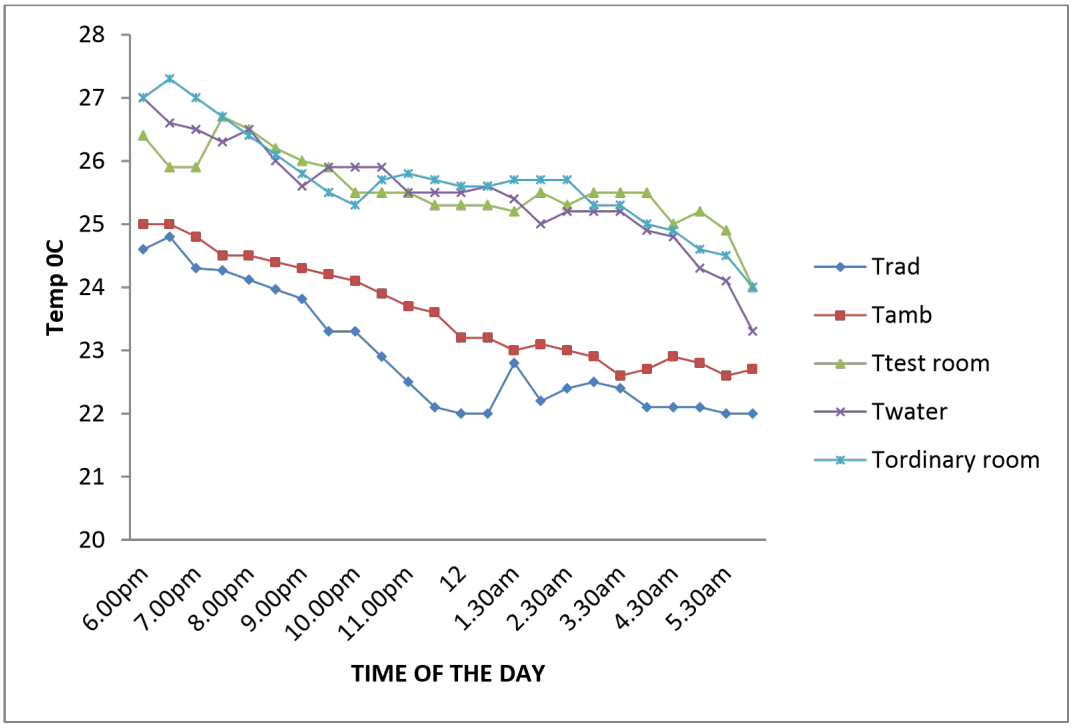


Fig 4.28 *Radiative Cooling Experiments carried out at the Federal University of Technology, Owerri on September 4-5, 2010.*

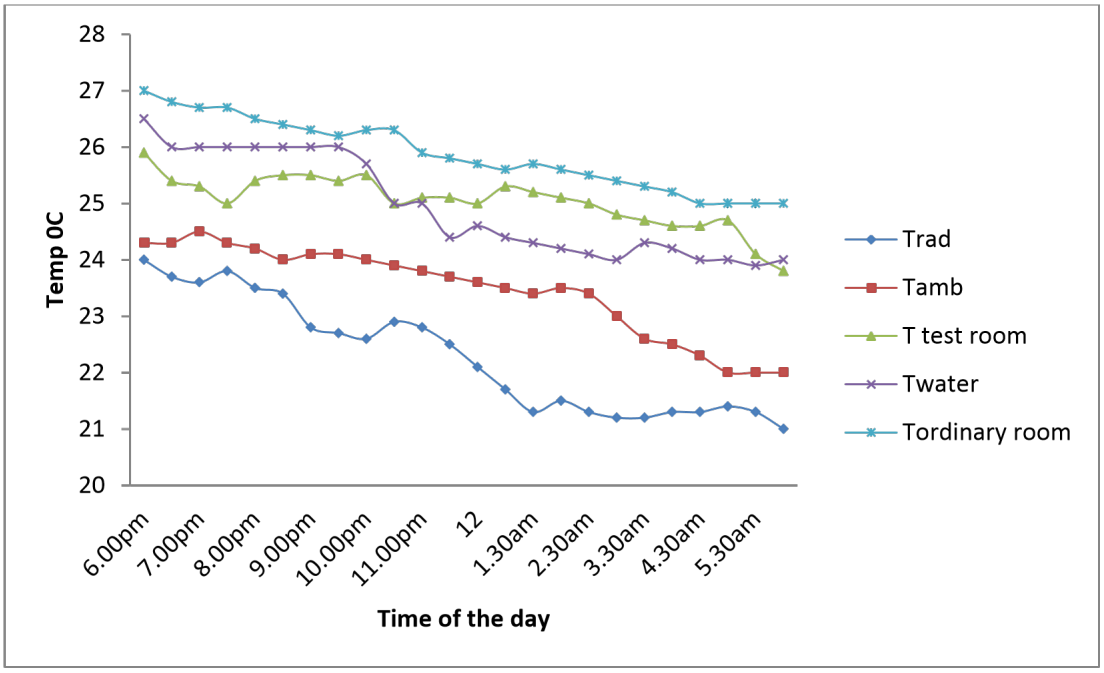


Fig 4.29 *Radiative Cooling Experiments Carried Out At The Federal University Of*

Technology, Owerri on September 12-13, 2010.

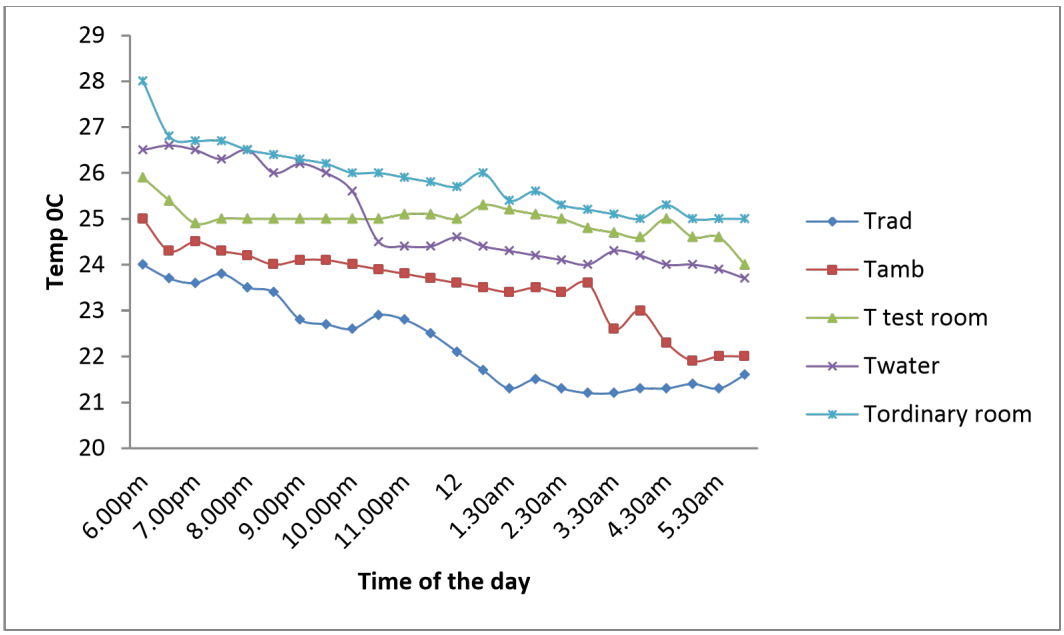


Fig 4.30 *Radiative Cooling Experiments carried out at the Federal University of Technology, Owerri on September 20-21, 2010.*

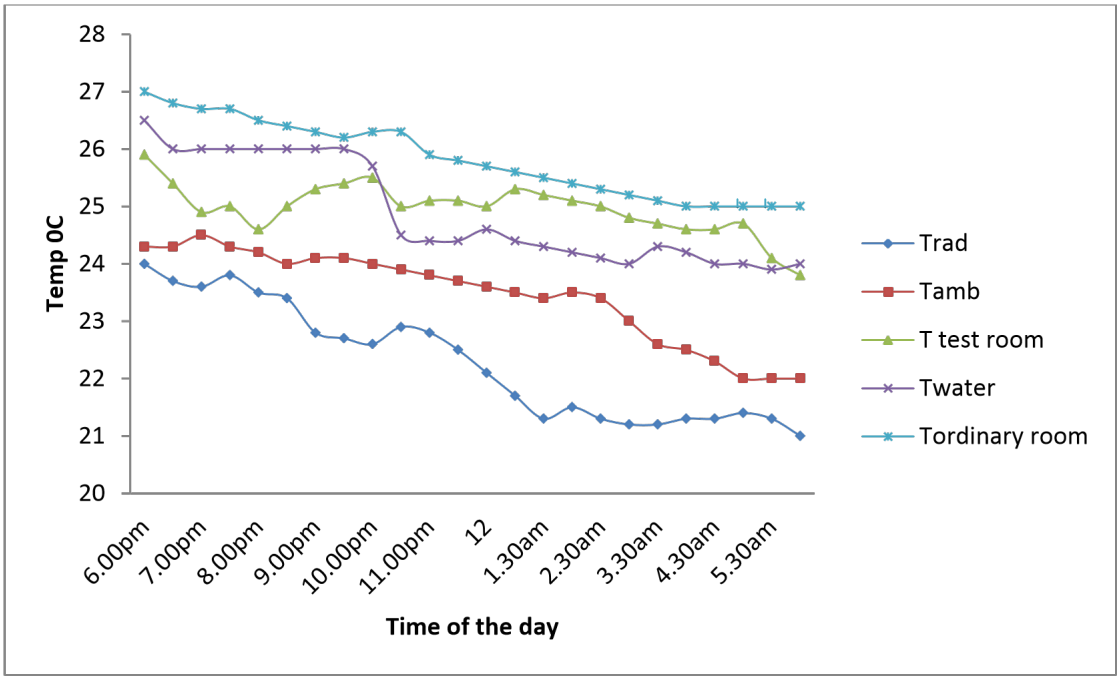


Fig 4.31 *Radiative cooling experiments carried out at the federal University of*

technology, Owerri during September 24-25, 2010.

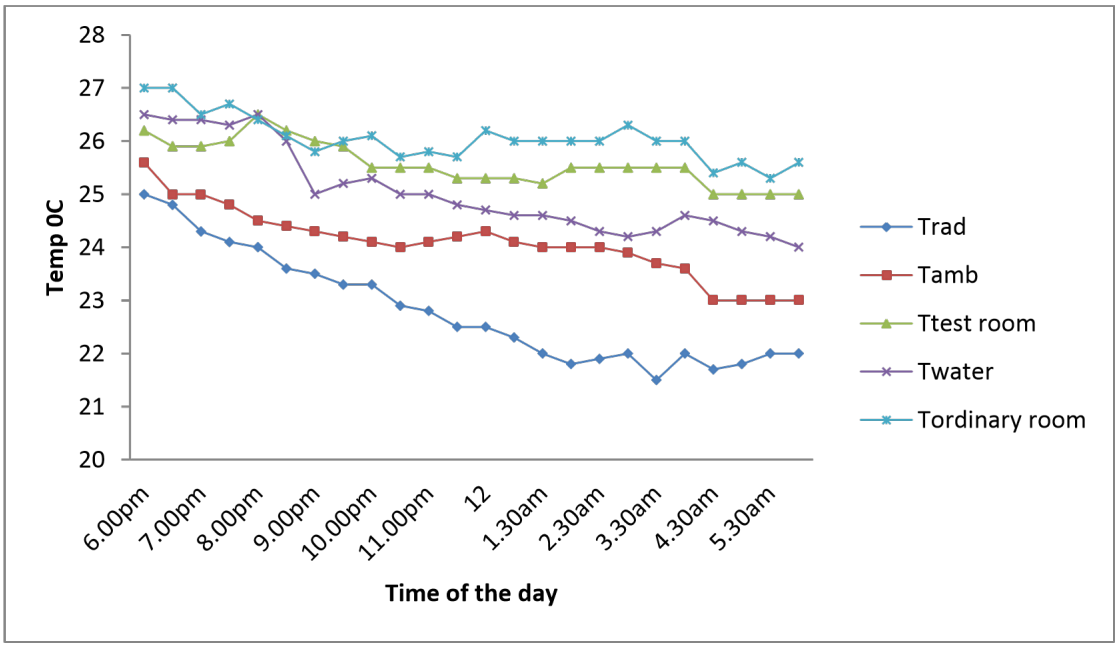


Fig 4.32 *Radiative cooling experiments carried out at the federal University of technology, Owerri on September 29-30, 2010.*

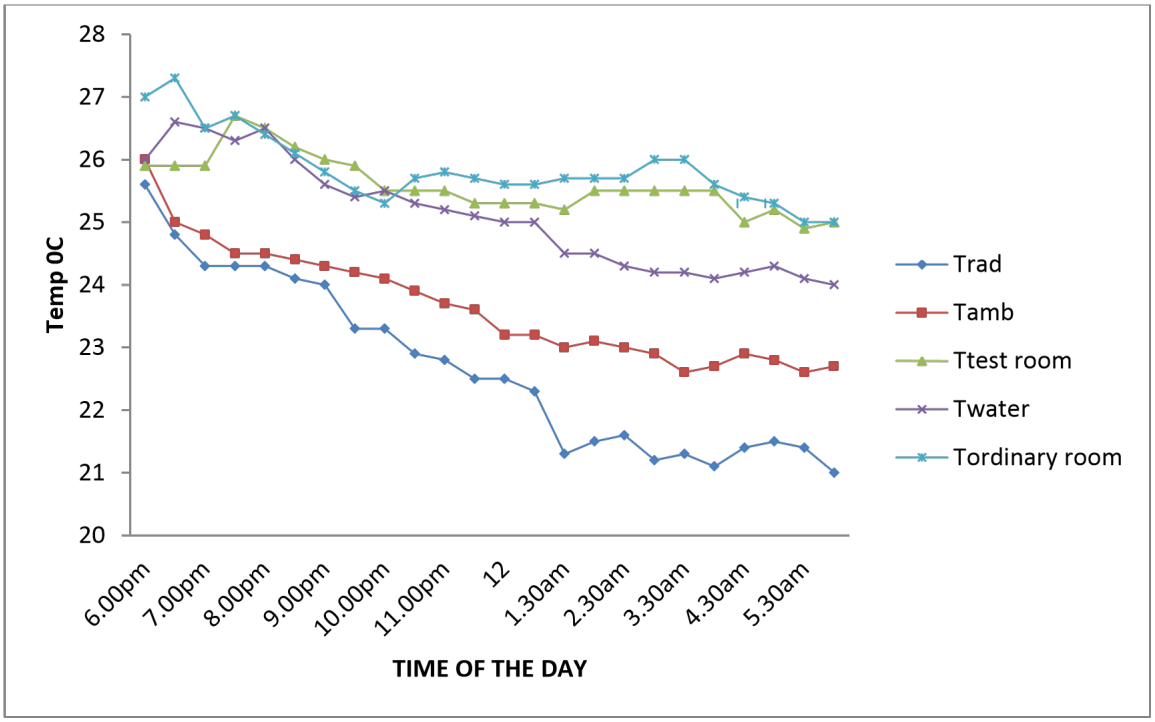


Fig 4.33 *Radiative cooling experiments carried out at the federal University of technology, Owerri during October 9-10, 2010.*

4.5 ANALYSIS OF THE NOCTURNAL COOLING PROFILE

Table 4.1 shows the system operating performance for the period 29th March 2010. It indicates the values for the radiator surface temperature, ambient air temperature, dew point temperature, relative humidity, net cooling power computed and system coefficient of performance. Similarly, fig 4.1-4.33 shows the temperatures of the radiator surface, test room temperature, water storage tank temperature and the control room temperature. The role of the radiator is to cool the water that passes through it. The efficiency of the radiator unit was observed through computation to vary over the range of 4.5 to 30.0%. The Overall efficiency values of the radiator unit was rather low compared to those of similar units used by **Meir et al, (2002)** and **Ito and Miura(1989).** However, such low value may be attributed to the effect of environmental degradation of the aluminum surface, thus reducing the emittance of the surface **Ali et al, (1995).** Improvements could be achieved by using a selective surface coating and by providing a screen cover over the radiator surface. In addition, dust particles also contributed to the low efficiency of the system. The period of night time cooling varied over 6 to 11 hours. Depressions in the temperatures of the water and the test room were not observed until around 9.00 pm -11.00 pm which means that some percentage of the cooling produced was used up to offset the accumulated solar heat gain during the day. Since the re-radiation of the earth surface and the building mass back to the atmosphere usually occurs when solar insolation is no longer available, the accumulated solar radiation heats up the ambient air temperature reasonably.

The sky radiator temperature, which depends on the ambient condition, rate of heat loss to the sky and the rate of heat loss from the circulating water which is being cooled by the radiator surface, was equally affected. As reported by **Meir et al (2002),** this heat loss from the system is approximately proportional to the temperature difference between the radiator surface and ambient temperature. The

temperature variations of the ambient air matched approximately to the radiator surface temperature. See fig 4.1 -4.33. This is because, the sky radiator being exposed to the atmosphere gains heat due to convection heat transfer. As the ambient air temperature fluctuates due to fluctuating ambient conditions, the heat transfer to the radiator due to convection changes in equal proportion. Gradual decrease in the radiator surface temperature, water temperature, room temperature respectively occurred always between 11.00 pm to 2.00 am after the system has been operated for some hours. (See fig 4.1) It would be observed that initially in fig 4.1 that the water temperature remained high despite the low temperature recorded by the radiator surface. Afterwards, a sharp drop could be observed in the entire graph for the water. This is so because at that period, the circulated water has released its heat to the atmosphere by circulating round the sky facing radiator whose temperature at that point is far lower than the temperature of water.

The useful cooling produced is a function of selective surface used and the configuration of the radiator system. For space cooling, the useful cooling is a function of the roof sheet and the thermal mass of the walls, air velocity. The heat exchanger facilitates day time cooling when the cooled water is redirected through it by shutting the valve passing water to the sky radiator.

After this period, a slight increase in temperature is usually observed for some days, this may be attributed to cosmic radiation which causes temperature inversion (see fig 4.8). This is usually noticeable between 3.00 am to 4.00 am. Temperature inversion according to **Encyclopedia Britannica (2010),** is a reversal of the normal behavior of temperature in the troposphere (the region of the atmosphere nearest the Earth's surface), in which a layer of cool air at the surface is overlain by a layer of warmer air. (Under normal conditions air temperature usually decreases with height.) Inversions play an important role in determining cloud forms (clear,

overcast, cloudy etc), precipitation, and visibility. An inversion acts as a cap on the upward movement of air from the layers below. As a result, convection produced by the heating of air from below is limited to levels below the inversion. Diffusion of dust, smoke, and other air pollutants is likewise limited.

Under the rain, a different scenario ensues, thus, a close examination of the graphs obtained during the period between May and September, 2010 shows that the temperatures recorded for the radiator surface temperature, ambient air are almost the same. A closer observation on the fig 4.26-4.30 showed that the moisture in the atmosphere was excessively high hence the temperatures of the radiator, ambient and water were all the same. This is because the moisture on the surface of the radiator forms a layer above it thereby making it impossible for the surface to radiate directly to the sky. The cooling of the radiator surface in this case was not due to radiation to the atmosphere; rather the surface was cooled through evaporation. Usually after the rain, the excess moisture in the atmosphere resulted to low radiative cooling because dew formation.

Dews formed on the radiator surface contributed to ineffective cooling of the system. Dew formation as we may recall is more predominate in humid climatic, where the dew point temperature of the ambient air is relatively high. It is possible that through radiative cooling, the temperatures of the radiator would fall below the ambient dew point. In such situation, dew will fall on the surfaces. Since water has a high absorptivity in the infrared, this dew film will reduce the net radiant cooling rate of the radiator. Thus, the dew formed has a self – accelerating process; the water film on the cover becomes the radiating surface itself, and due to its small thermal mass, cools rapidly, which causes more condensation on the surface **Idso(1981).** The cooling performance of the radiator is thus, drastically reduced.

This phenomenon may render in effective the use of selectively radiating surfaces in humid climates.

The net cooling power is usually at its maximum at the beginning of the experiment because the factor that inhibits cooling has its effect at its barest minimum. This occurs between 9.00 pm to 12.00 pm in most cases. After this period, condensation of the dew on the surface of the sky radiator results in sensible heating of the radiator surface thereby reducing the cooling potential of the radiator.

Generally, the surface temperature of the radiator reached a minimum of 21.0 ^{o} C most of the time except during the harmattan period where the temperature of 17 ^{o} C was recorded. The temperature depression between the radiator surface and ambient air temperature was generally low, which is indicative of the unfavorable weather condition which generally manifest due to the high relative humidity of the air around the system. The effect of high relative humidity could also be felt more predominantly during the wet season. A close examination of the graphs obtained during May to October showed a little temperature difference between the radiator surface temperature and the ambient air with maximum temperature depression below ambient being in the range of 0.4-1.5 ^{0} C. (see fig 4. 26).

Ambient temperatures during this period ranged over 22-29 ^{0} C. This observation clearly indicates that high moisture content in the atmosphere adds to sensible heating of the air thereby rendering radiative cooling of surfaces facing the sky ineffective. Similarly, during the dry seasons, the radiator surface was observed to witness a high temperature depression below the ambient in the range of 1.2- 3.5 ^{0} C. This is very indicative that low relative humidity of the atmosphere played a major role in facilitating passive cooling in our local climate. This could be seen from the results obtained during the March and April period.

The total net cooling power of the system varied over the range of 10 W/m^2 – 52.5 W $/m^2$ of the expose radiator area. The value of the total net cooling power depends largely on the surface and night sky conditions, high relative humidity occur during the rainy season thereby increase the effect of the atmosphere constituent. The useful heat energy delivered to the water in the storage tank depends on the amount of energy delivered to it from the radiator.

The available system coefficient of performance COP for the space cooling was found to be fluctuating within the range of 0.015 to 0.09. The water cooling COP is expected to be higher than the value obtained for the space cooling because; it is the cooled water that will in turn cool the room. This is expected since COP is a function of the radiator surface temperature and the ambient air temperature. The cooling obtained from the radiator cools the water. The water in turn cools the room .Thus; the maximum cycle COP of 0.0912 was realized during the March run where the relative humidity is very low. The minimum value of

0.011 was realized where the radiator surface temperature was 23.5 ^{o} C in Sept, 2010 and March, 29 2010. For this nocturnal cooling system, the cooling effect realized was within 6-15% of the total energy delivered to the radiator. Whereas the cooling effect for the control room was observed to be to be within 1- 5% for room without natural ventilation and with 4 – 10% for room with natural ventilation. These observations simply imply that the large thermal capacities of the water, wall structure, affect the cooling effect of the system.

The useful cooling produced Q_{net} is a function of the mass of water placed in contact with the radiator steel rod modules. Production of cold water using the system would generally result in low cooling because of the effect of the accumulation of heat generated of solar heat gain which is not taken into account.

The available system cycle COP was found to be approximately constant in most days during the wet season. This was expected since COP is the function of the radiator and ambient air temperature.

The actual COP is seen to depend on the useful cooling produced and quantity of energy delivered to the cooled water. The effect of the various season was equally observed to effect cooling. Net cooling power obtained during the rainy and dry seasons indicates that dry seasons are well suited for nocturnal radiative cooling. Thus the maximum cycle COP of 0.09 and 0.25 was realized during the dry season where the temperature depression of 3.5 ^{o} C below ambient for the radiator was attained for the space and water cool respectively. The minimum value of 0.011 and 0.09 was realized when the temperature depression of 0.1 ^{o} C was attained during the rainy season. The net cooling effect for the space cooling is within 3-7% of the total energy available. This was partly due to the lack of proper ventilation. However, when the doors and windows of the system were opened, the cooling effect rose to about 10-15% of the total energy available. These observations imply that to achieve proper cooling effect, natural ventilation system must be adequately provided.

4.6 ANALYSIS OF DAY TIME COOLING PROFILE

The capability of the nocturnal cooling system designed to substantially reduce cooling load of the building during the day was further verified in an experiment. The cooled water from the previous night experiment was circulated between the storage tank and the heat exchanger as shown in fig 4.34

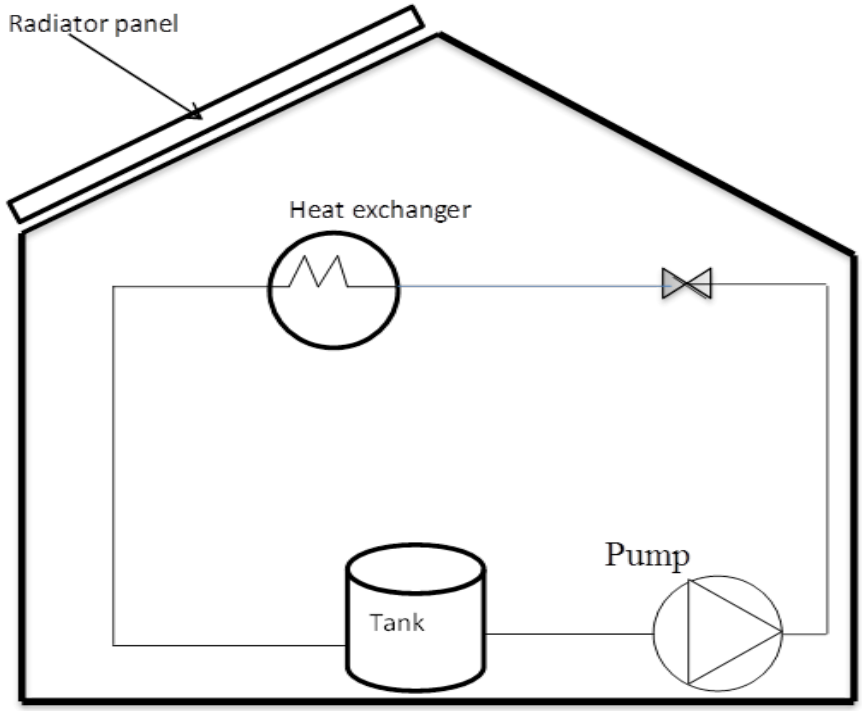


Fig 4.34 The *Daytime Cooling Arrangement*

This was made possible by employing the services of valve B which isolates the flow into the sky radiator. As the water passes through between the heat exchanger, and the water tank, the heat accumulated in the room is consequently transferred to the heat exchanger. As the process continues, the room and the heat exchanger tend to achieve a state of thermal equilibrium thereby reducing considerably, the temperature of the room. Series of tests were carried out on the 12th through the 30th of April, 2011 at The Federal University of Technology, Owerri.

The following results were obtained as shown in fig 4.34-fig 4.40. From fig 4.34, it could be observed that the temperature of the test room was usually lower than the temperature of the control room and the ambient air during the day. Temperatures in the range of 25.6 - 24.4^{o}C were obtained during the test in the test room whereas the control room temperature was in the range of 28.5^{0}C-25.9^{0}C.

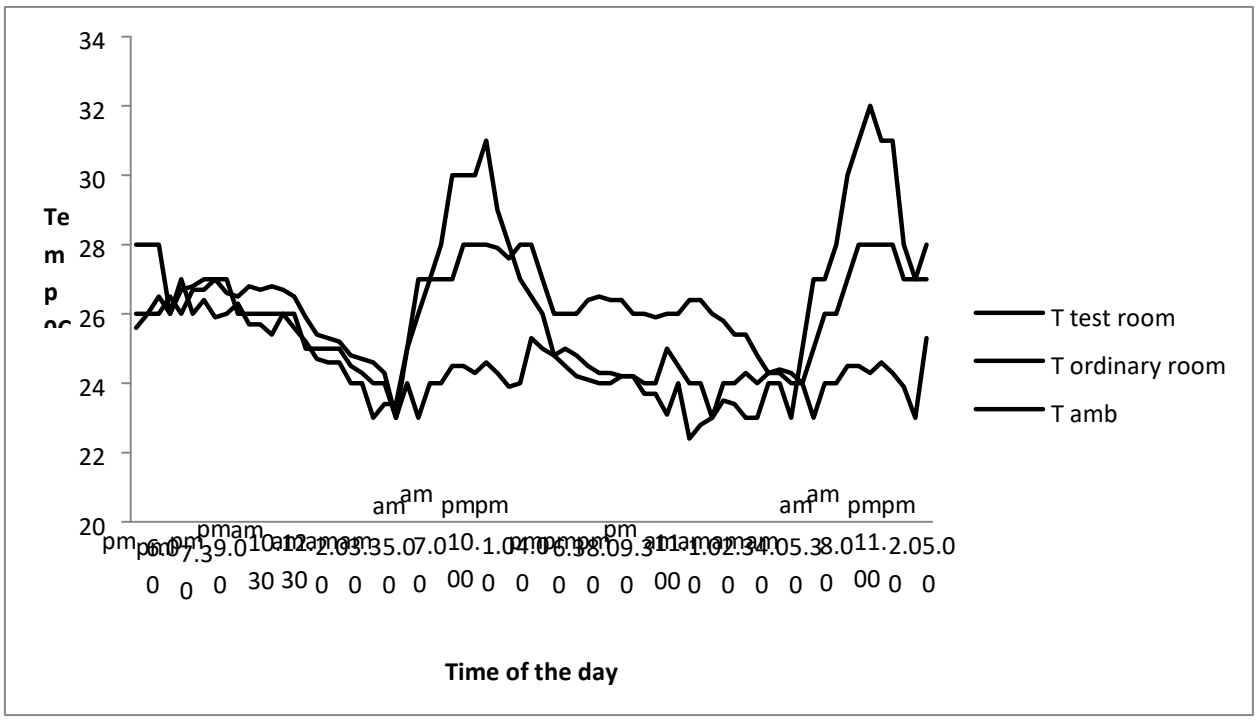


Fig 4.35 the temperatures of the test room, ordinary room and ambient air for12th 14th April, 2011.

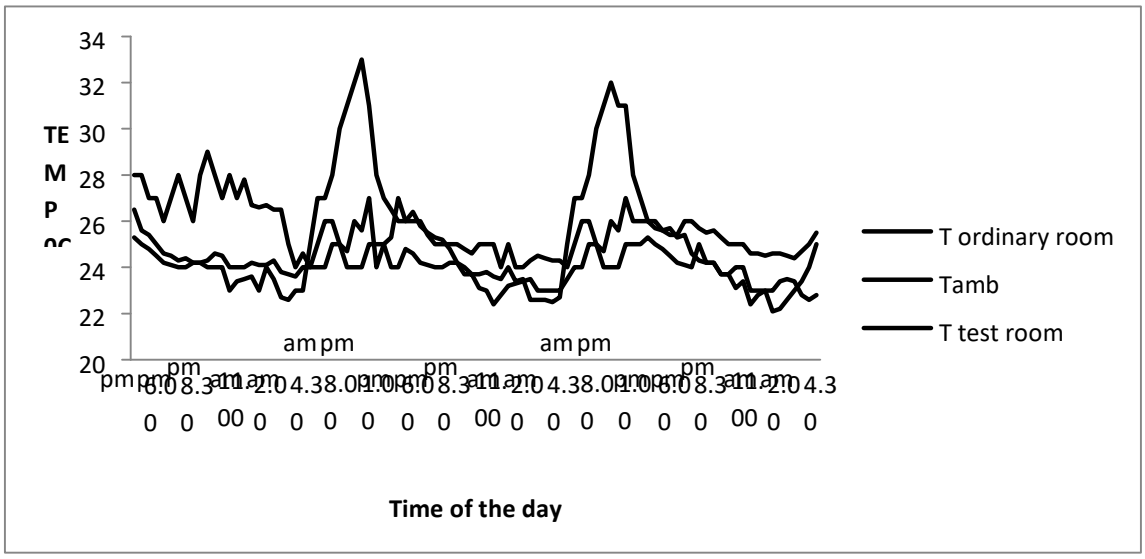


Fig 4.36 the temperatures of the test room, ordinary room and ambient air for17th -

19th April, 2011

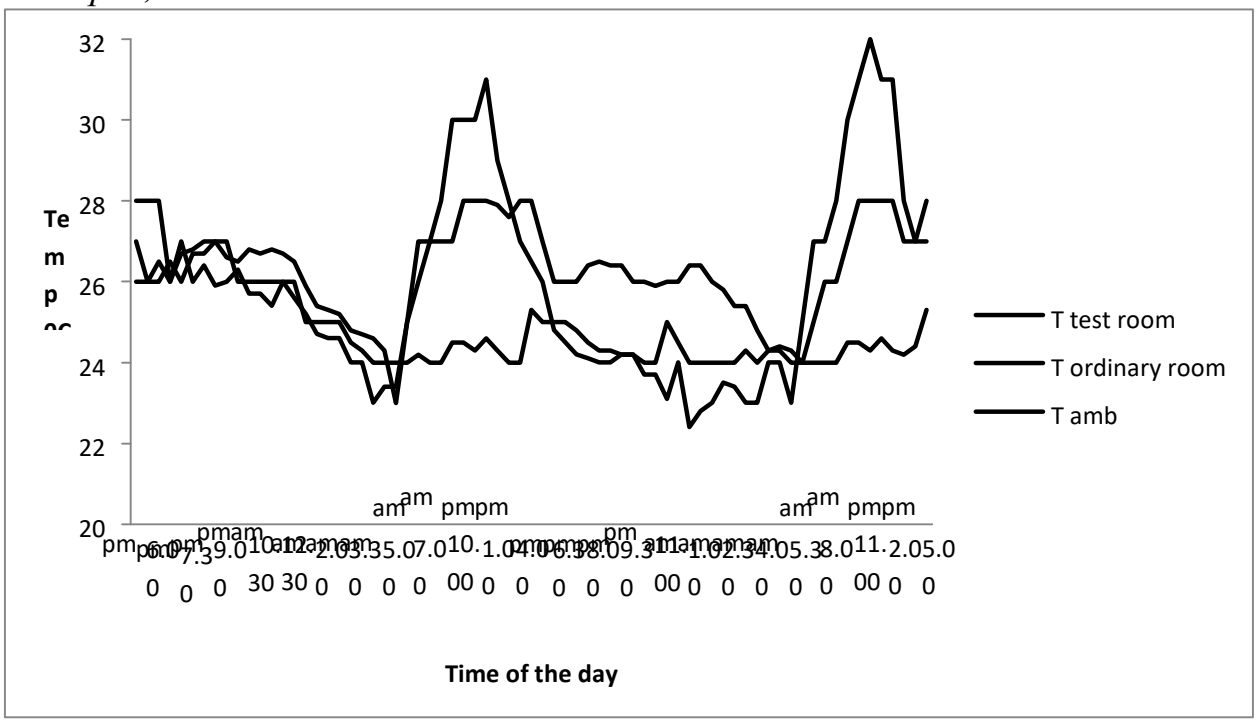


Fig 4.37 the temperatures of the test room, ordinary room and ambient air (20th 22nd April, 2011)

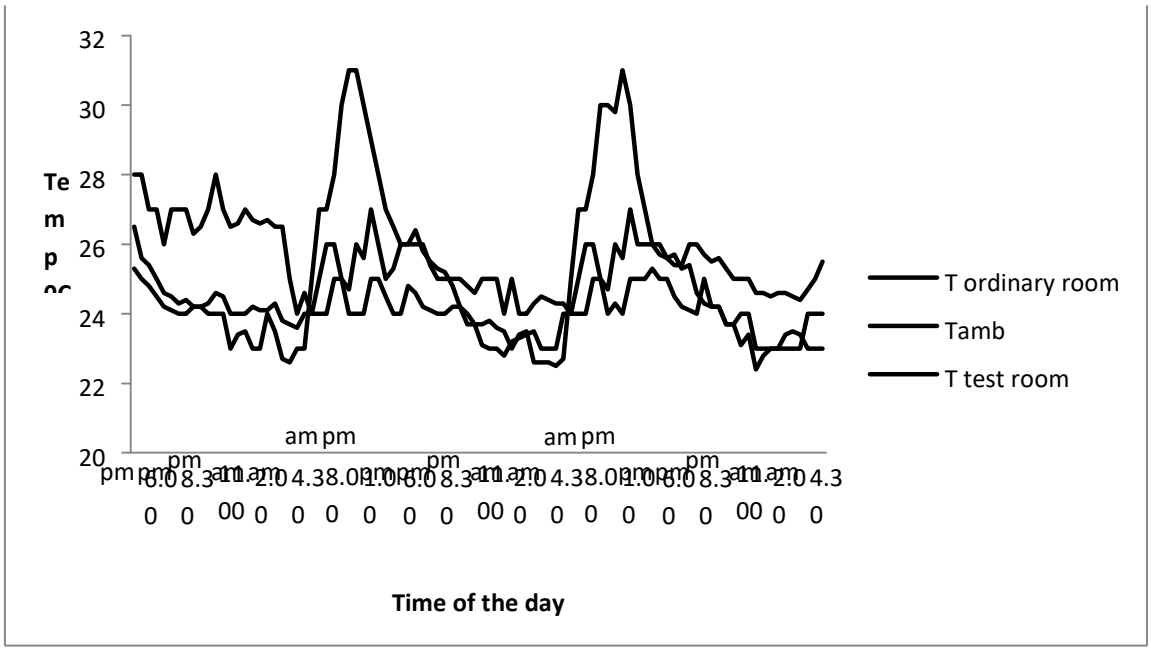


Fig 4.38 the temperatures of the test room, ordinary room and ambient air for23rd -25th April, 2011

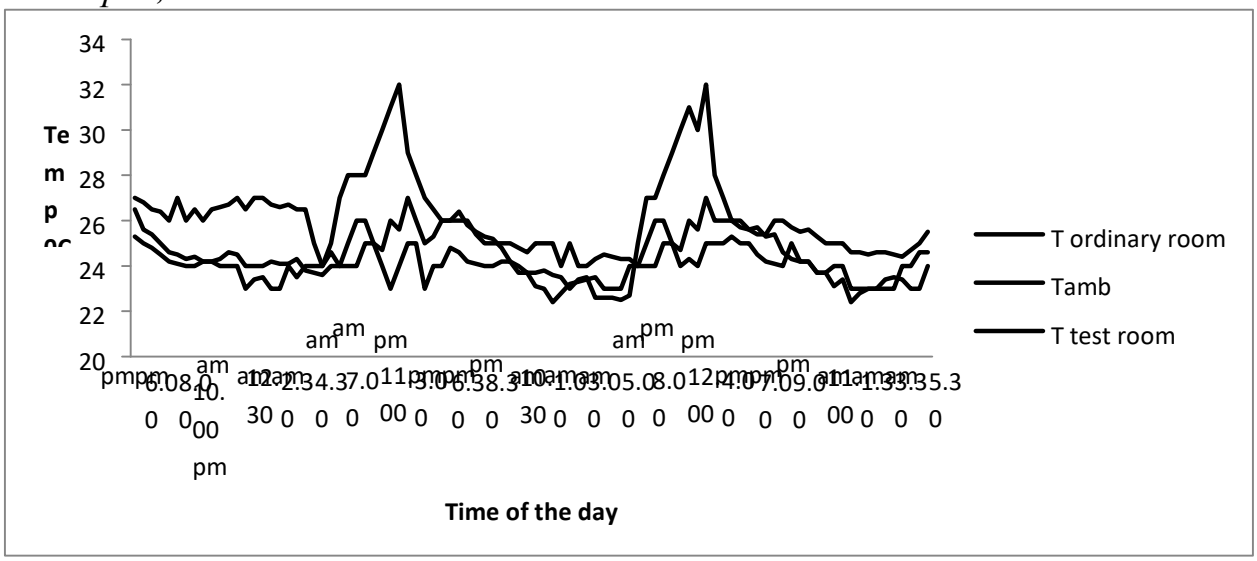


Fig 4.39 the temperatures of the test room, ordinary room and ambient air for26th 28th April, 2011

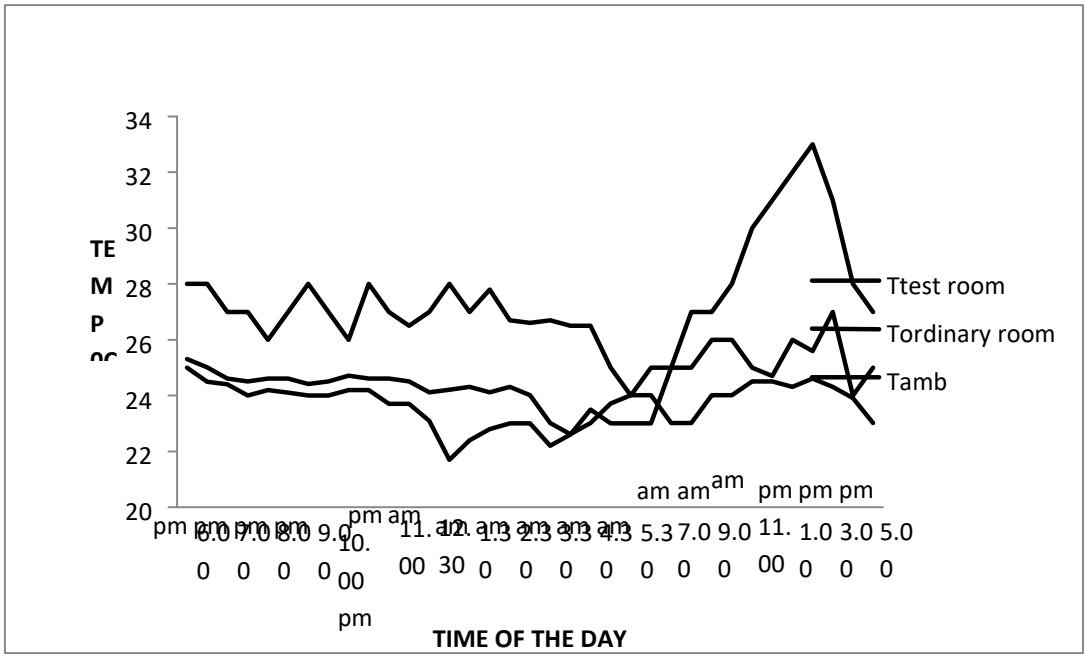


Fig 4.40 the temperatures of the test room, ordinary room and ambient air for29th 30th April, 2011

A closer look at figs 4.35- 4.40 reveals that the test room temperature is constantly below the control room temperature. Temperature depression of the test room below the ordinary room gave an average depression in the range of 1.2-3.5^{0}C. Fig

3.35 shows that the difference between the $T_{test\ room}$ and $T_{ordinary\ room}$ at 1.00pm on the 12th of April ,2011 is 3.5^{0}C corresponding to a net cooling of about 1428kJ. The incorporation of the room convector actually provided sufficient energy to cool the test room.

The series of experimental result reported in the preceding section revealed that the performances of the nocturnal cooling were not as expected in the design stage. The theory and experimental measurements indicate, however a rather low overall COP for the nocturnal cooling system using aluminum radiator surface. The expected cooling load was 100 W/m^2 of radiator area used on minimum radiator surface temperature of 20.5 ^{o} C.

The experimental result however indicated that only water temperature of around 21.0 ^{o} C could be obtained and the best useful cooling achieved was about 52.5 W/m^2 based on the radiator aperture area or the equivalent of about 40% of the expected value. The cooling potential of the system has been shown to be severely affected by the dynamic of the system. Generally the large thermal mass of the radiation/water/wall units is responsible. The walls in particular were found to be too massive and inadequately insulated such that high proportion of the circulating cooling capacity of the nocturnal cooling system was used for its cooling.

This implies that reducing the thermal mass of the wall should significantly improve the cooling of the room interior. One method of accomplishing this is by using material with low thermal mass and providing adequate ventilation.

The low overall COP values obtained could be attributed to the technical inefficiencies in the construction of the nocturnal cooling system.

4.7 Comparison with Other Test Results under Different Climatic Condition

Table 4.2 shows that the temperature depression obtained in this work are significantly lower than similar nocturnal cooling system operating on flat plate sky facing radiator except for the results obtained by **Santamouris and Asimakopoulos (1996)** in Thailand where there is a high humid climate. The other results obtained from America and Europe and some other climate were significantly lower.

In addition, the operating condition of a nocturnal cooling system varies with its geographical location. Systems in temperate climates usually have high COP compared to system in tropical climate like Nigeria.

Fig 4.41 shows the cooling power variation with mean temperature depression obtained in this work together with those of **Meir et al(2002), Ito and Miura, (1989), Parker (2005).** This comparison was based on the average net cooling power obtained during the month of June. The temperature variation may not be the most appropriate comparison tool because the other work belongs to a different climate where the maximum ambient temperature is far below the minimum

temperature obtainable under Owerri climate.

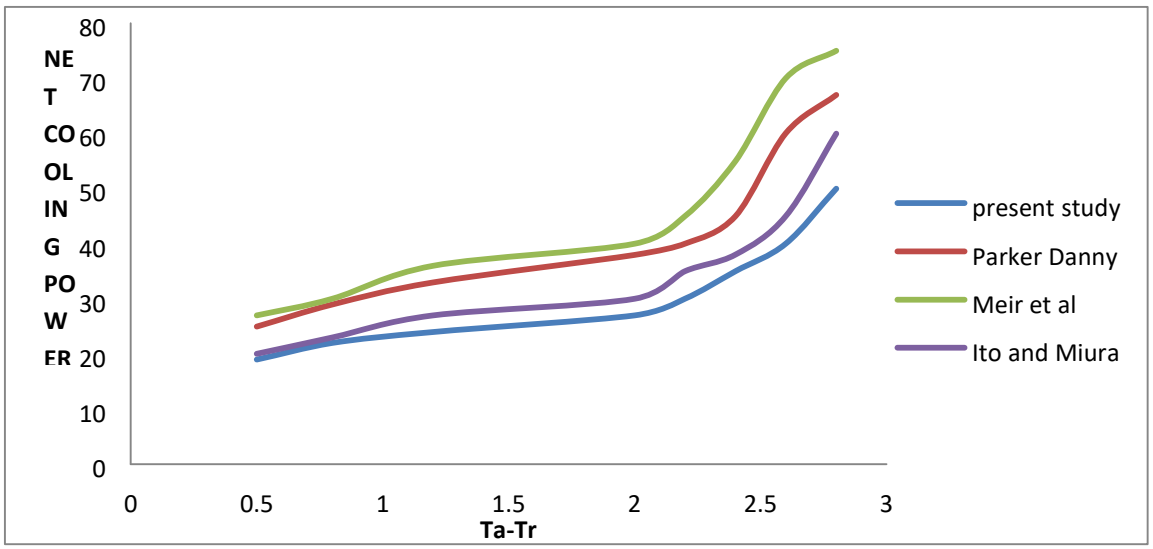


Fig 4.41 Comparison of the cooling power against temperature depression below ambient for the present work and other results from other climate

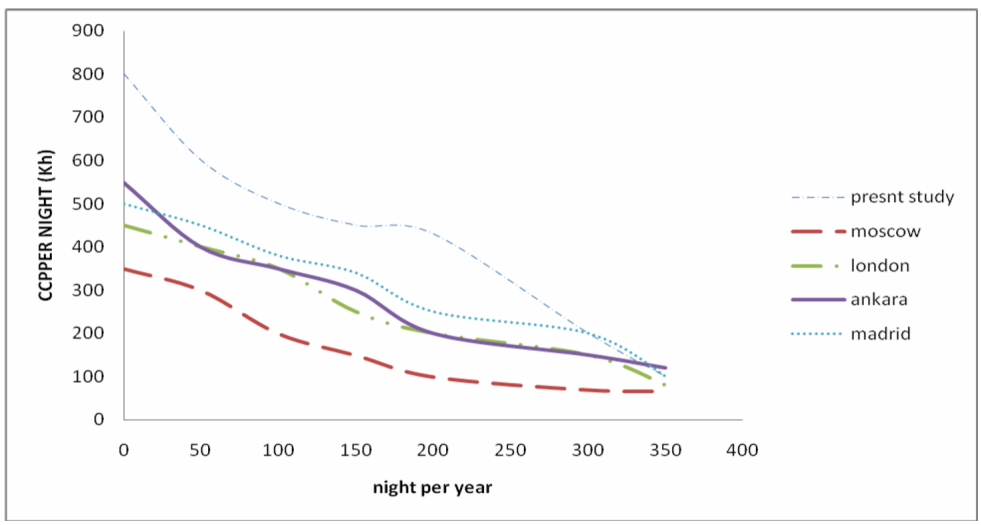


Fig 4.42 climatic cooling potential for different countries

Minimum temperatures are seen to be 20.5 ^{o}C, 15 ^{o}C, 13 ^{o}C, 23 ^{o}C. and 5 ^{o}

C for this work,**Meir et al(2002), Ito and Miura(1989),Parker(2005)** and**Angeliki(2005)** respectively.

The general trend of the temperature profile is similar for all the experimental measurements for this work and **Chotivisarut and Kiatsirioat, (2006)** a better similarity is noticed in the curve. This observation may be attributed to the similar climate condition (tropical).The climatic cooling potentials for this work vary significantly with that **of Nikolai et al(2006),Meir et al(2002) and Artmann et al(2006)** because the climatic conditions are different hence different climatic cooling potential was observed. This observed variation is mainly due to different climatic regions where these works were carried out. Note that the climatic cooling potential depends on the solar heat gain during the day. The all year round solar heat gain for Owerri was computed analytically and its value was compared with the work of **Nikolai et al(2006),** as shown in fig 4.42.

COP recorded for this work is lower compared to that obtained from **Meir et al(2002), Ito and Miura(1989),Parker(2005)** and **Chotivisarut and Kiatsirioat, (2006)** . This may be attributed to technical configuration of the radiator and the materials used.

4.8 The Cooling Power obtained for Some Selected Days

The net radiative cooling power for some selected days are presented in fig 4.43–fig 4.50. The figures presented in the graphs show cooling pattern for the prevailing seasons namely; dry, rainy and harmattan seasons respectively. The net power was plotted against the temperature depression of the radiator surface below the ambient air. These temperature depressions were obtained during the night in an interval of between two to three hours. The maximum net cooling obtained is about 52.5 W/m^2 . Actually, some graphs gave values of about $70W/m^2$, these values may not be reliable because of the faulty readings of the data logger, Though this

was later corrected using thermocouple reader and a Hopman digital thermometer. A close observation of the cooling pattern shows that the cooling pattern varies as the seasons changes.

This clearly indicates that nocturnal cooling activities are weather dependent.

The cooling was also observed to be decreasing and increasing intermittently as the cooling period progresses and the temperature depression below ambient was equally observed to follow the same pattern. This may be attributed to the fact that the aluminum used for the construction of the radiator surface was losing it selectivity properties due to adverse weather condition. Dust particles have been known to hinder the radiative properties of some radiator material **Ali et al (2002).** Secondly, the March period was the tail end of the dry season which entails that the change in weather may have affected the performance of the radiator.

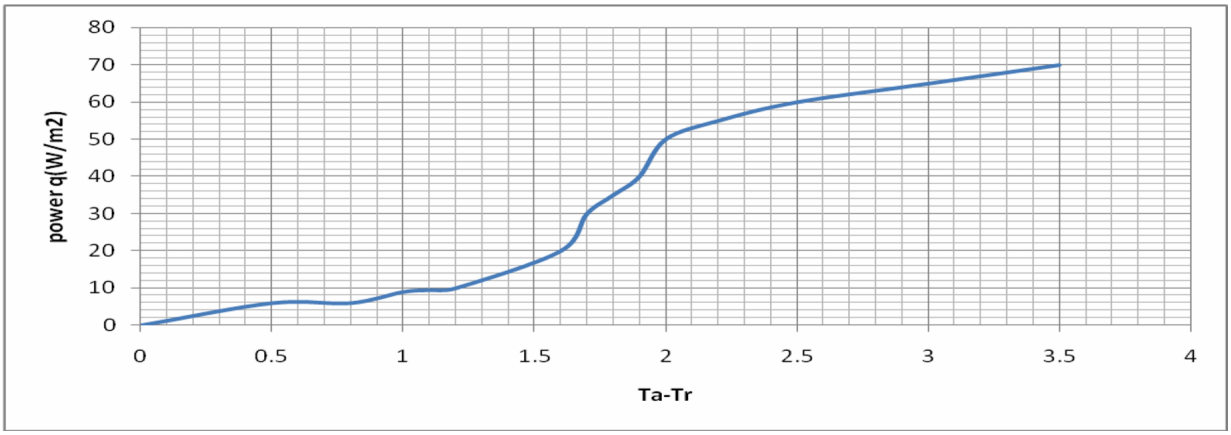


Fig 4.43 the net cooling power against the average temperature (Ta-Tr) on the 29th of March 2010

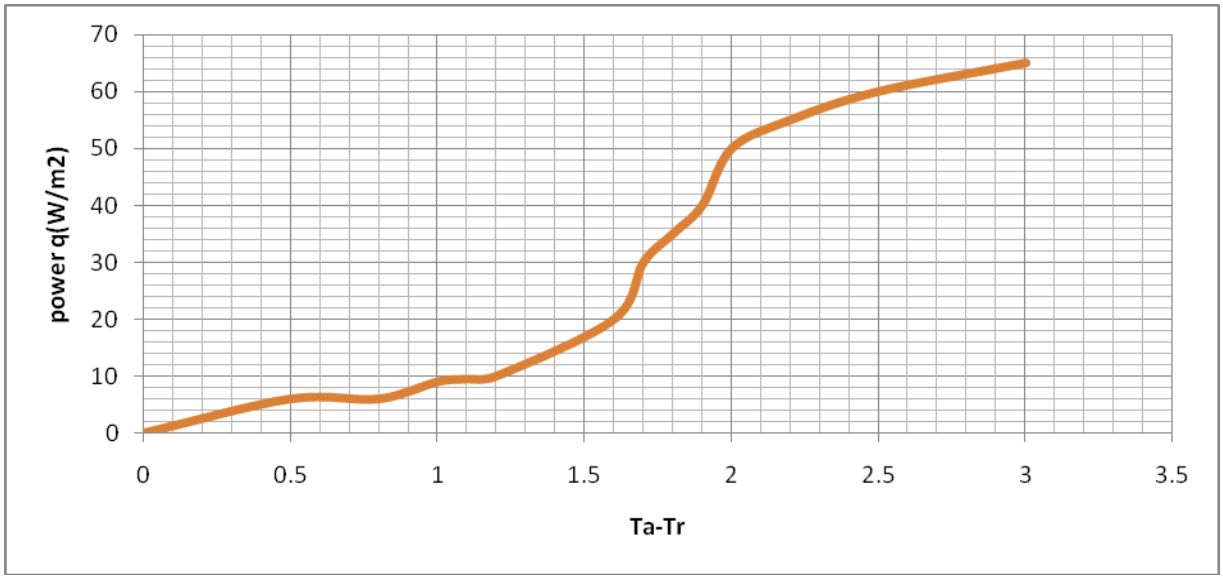


Fig 4.44 *the net cooling power against the average temperature depression (TaTr) on the 30th of March 2010*

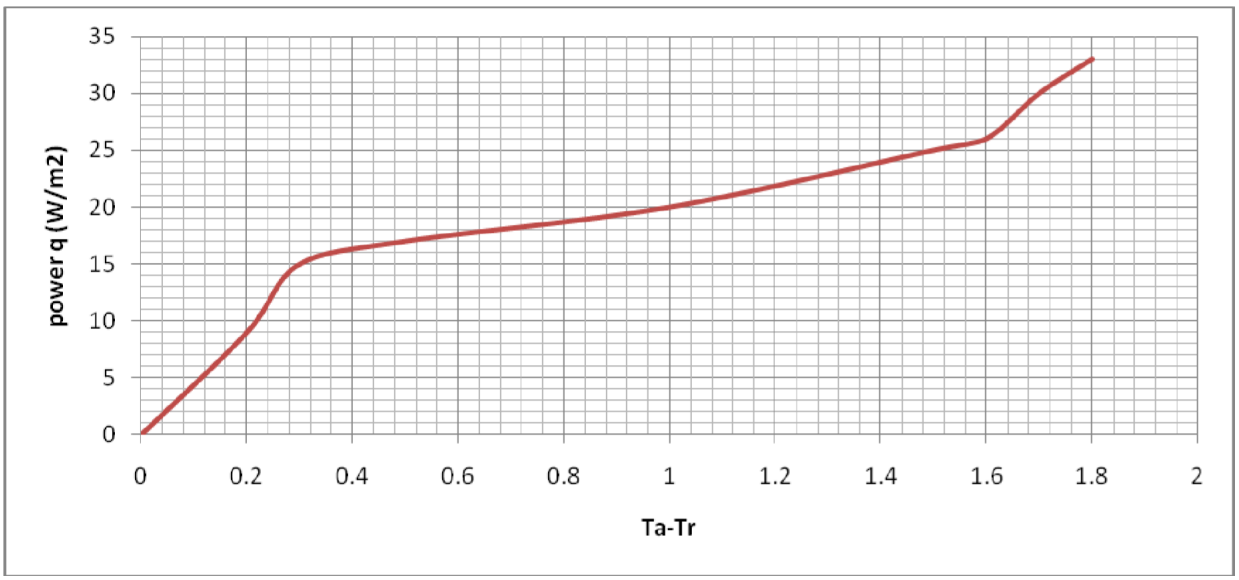


Fig 4.45 the net cooling power against the average temperature depression (TaTr) on the 29th of April, 2010

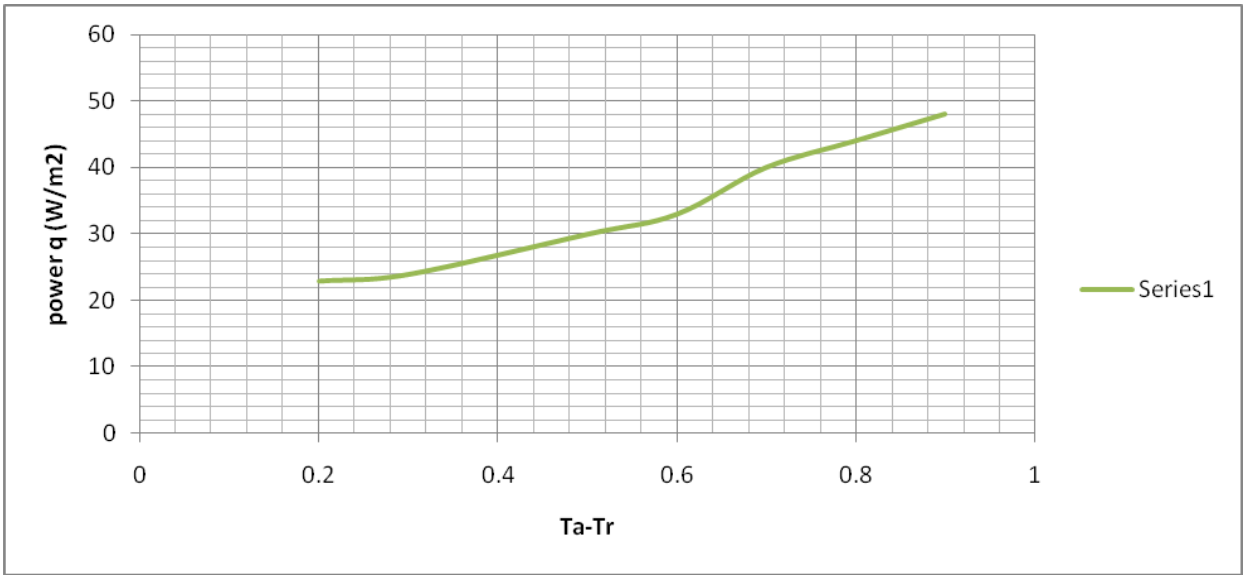


Fig 4.46 the net cooling power against average temperature depression (Ta-Tr) on the 9th of July, 2010

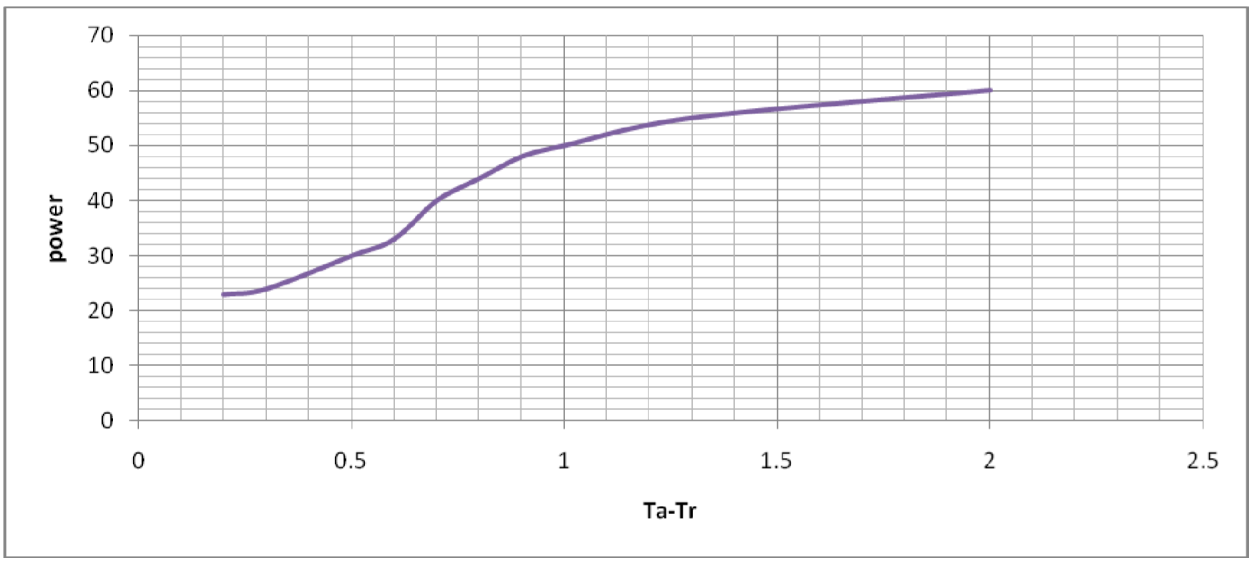


Fig 4.47 the net cooling power against the average temperature depression (Ta-Tr) on the 5th of August, 2010

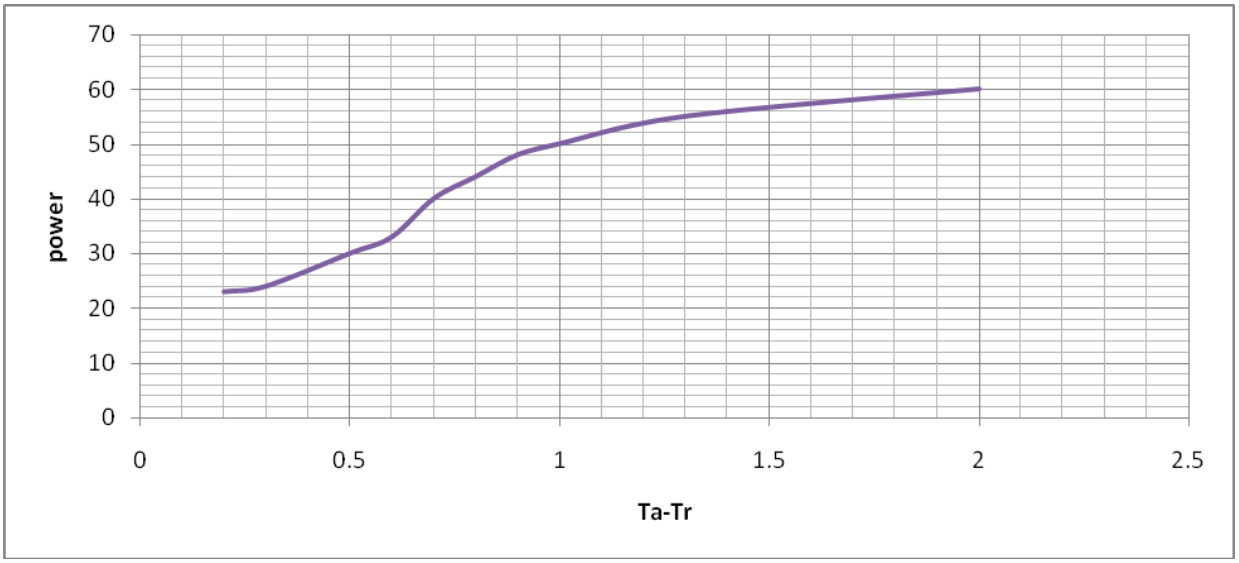


Fig 4.48 the net cooling power against the average temperature depression (TaTr) on the 5th of September, 2010

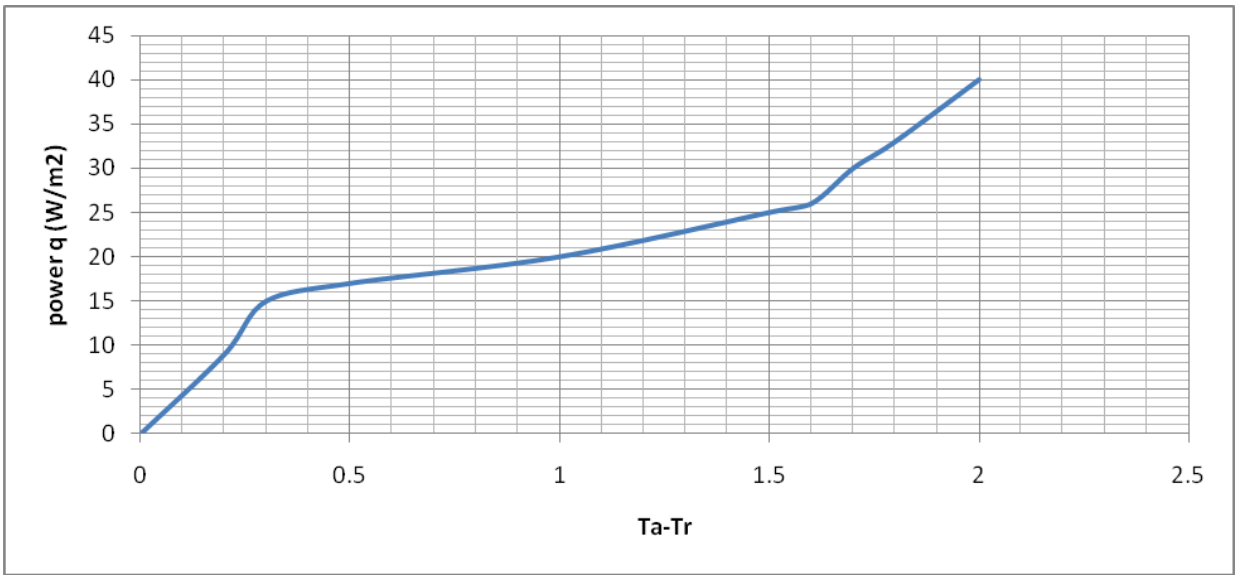


Fig 4.49 the net cooling power against the average temperature depression (Ta-Tr) on the 9th of September, 2010

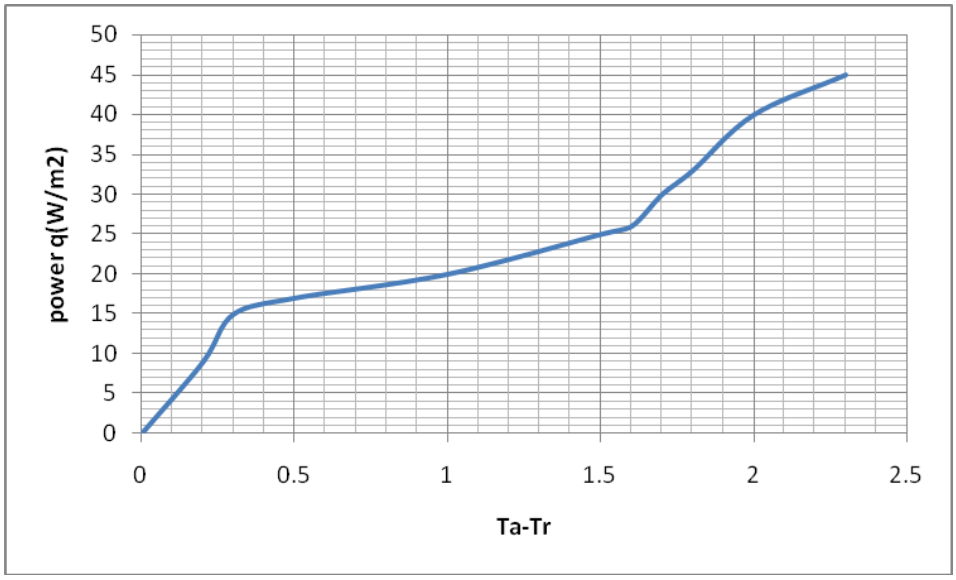


Fig 4.50 the net cooling power against average temperature depression (Ta-Tr) on the 29th of September, 2010

4.9 The Effect of Seasonal weather Variation on Nocturnal Cooling profiles

The present work examined the performance of system under the prevalent weather condition encountered in the typical tropical environment. Nigeria is a typical tropical region of the world hence is subject to different climatic conditions. There are two major seasons namely the dry and the rainy seasons. During the dry season, there is another weather condition known as harmattan. Harmattan is a hot, dry wind that blows from the northeast or east in the Western Sahara and is strongest in late November to mid-March. It usually carries large amounts of dust, which it transports hundreds of kilometers over the Atlantic Ocean; the dust often interferes with aircraft operations and settles on the decks of ships.

The rainy season is characterized by excess rainfall and too much moisture in the atmosphere. It starts from April to late October. Rain bearing clouds dominates most days and night. Seasonal variation in nocturnal cooling pattern within these

periods varies depending on the prevailing weather condition. The dry season is characterized by a high ambient condition with high ambient temperatures in the range of 21 ^{o} C to 27 ^{o} C and high temperature depression of about 3.5 ^{o} C below ambient for the radiator surface. While the rainy season has low ambient condition with low ambient temperature in the range of 22.0 ^{o} C to 29

^{o} C and ^{o} low temperature depression of about 1.05 C below ambient. The effect of relative humidity was pronounced during the rainy season, hence the low temperature depression. The temperatures of the ambient air, radiator surface and dew point were observed to be constant during the rain and vary slightly immediately after the rain.

Experimental test were undertaken at the federal university of technology, Owerri in the eastern part of Nigeria on Latitude 5.23 ^{o} N, longitude 6.58 ^{o} E during periods spanning the two prevailing seasons between (February 2010 and December, 2010) to evaluate the performance of the system under these seasons.

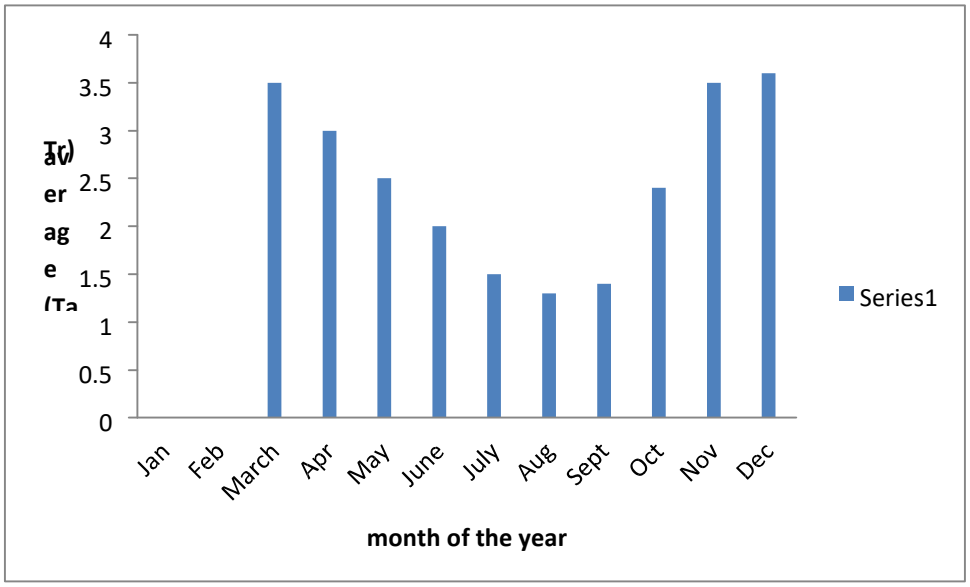


Fig 4.51 showing the average temperature depression (Ta-Tr) for the period of the experiments.

Fig 4.51, show the temperature depression profile recorded for the nocturnal cooling under the dry and raining season. From fig 4.51, it may be observed that the nocturnal temperature depression for the month of March, April, November and December which falls into the typical dry season under Owerri climate is relatively higher than those of the May, June, July, August, September and October. These months of the year represent a typical rainy season hence the presence of moisture in the atmosphere hinders night time radiative cooling activity within these months.

The temperature depression for March, April period is about 3.5 ^{o} C below the ambient air temperature. While the same system under the wet season has average temperature depression of about 1.05 ^{o} C below the ambient air temperature for the measurement carried out on the 6th of August 2010 .The radiator temperature when compared to the ambient air temperature under the tropical rain gave no appreciable difference in most instances. This may be attributed to high moisture content of the atmosphere which increases the relative humidity of the atmosphere thereby increasing the atmospheric radiation. We may recall that high relative humidity increases the atmospheric radiation which covers the 8µm-14µm atmospheric window. This atmospheric window is responsible for nocturnal radiative cooling. When it is covered due to high relative humidity, radiative cooling is usually hindered.

From the figures it could be observed that the ambient conditions for the rainy season are usually low when compared to the dry seasons. The ambient air temperature measured especially in the evening after a heavy ranged from 24 ^{0} C at

the beginning of the experiments for the rainy season as against 27 ^{o} C during dry season. Similarly, the net cooling power of the system also varies as can be seen in fig 4.52 below

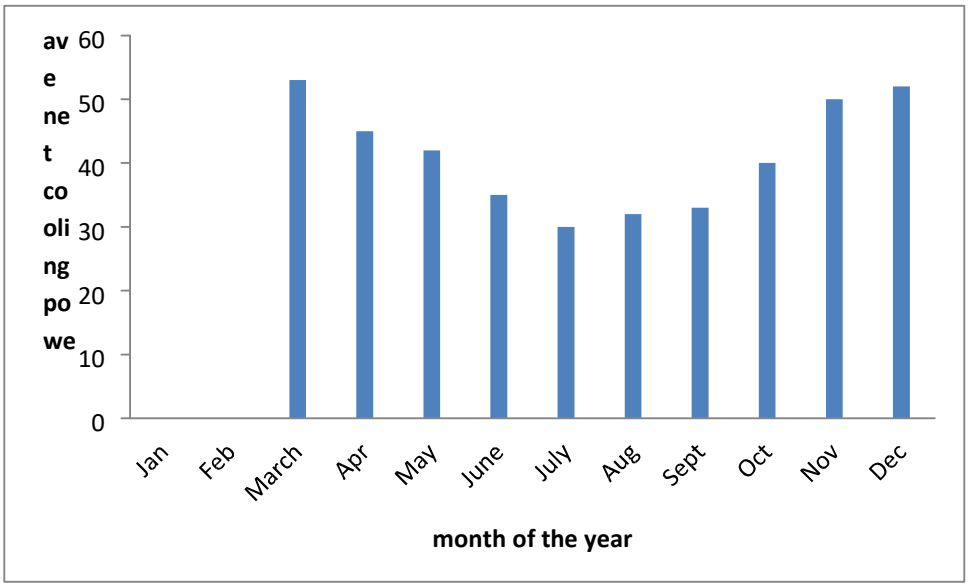


Fig 4.52 the variation of the net cooling power for the period of the experiment.

From fig 4.52, it is quite obvious that nocturnal radiative cooling is more effective during the dry season. This is because the dry season unlike the wet season is characterized by low moisture presence in the atmosphere. Where as in the wet season the moisture level increases resulting in high relative humidity

which hinders cooling.

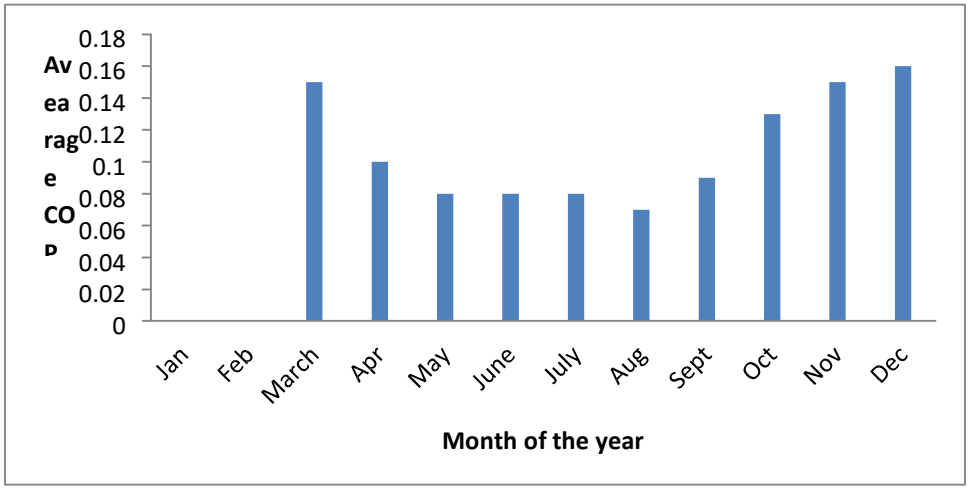


Fig4.53 the variation of the COP for the nocturnal radiative cooling system for the rainy and dry season for owerri climate.

The COP also varies as shown in fig 4.53. However, the variation fluctuates at interval showing that the COP is strongly affected by seasonal variation in the weather condition.

The effect of relative humidity is very pronounced in the wet season and in the dry season. From the appendix B, the psychometric chart produced shows that the value of the relative humidity ranges from 58% to 69% in the dry season while in the wet season the value ranges from 85% to 100% under the rain.

4.10 The Observed Effect of Natural Ventilation on Space Cooling

Natural ventilation is an important and simple technique that when appropriately used may improve thermal comfort conditions in indoor spaces, decreases the energy consumption of air conditioned buildings, and contribute to fight problems of indoor air quality by decreasing the concentration of indoor pollutants.

It must be recognized that natural ventilation is not just an alternative to air conditioning; it is a more effective instrument to improve indoor air quality, protect health, provide comfort, and decrease unnecessary energy consumption. Natural ventilation is usually provided in living spaces through the introduction of welldesigned doors and windows. By opening the doors and windows, the flow of air ensures that the room temperature is well maintained. It is known that natural ventilation can be generated by two methods: by thermal force or buoyancy effect, and by wind pressure force or wind-driven effect. In general, wind-driven natural ventilation is easier to achieve because it only needs a low wind speed to create adequate indoor air velocities that help people‟s heat transfer by means of evaporation. Tantasavasdi et al. (2001) study natural ventilation for houses in

Thailand and find that the buoyancy effect can create indoor air velocities only as high as 0.1 m/s because the height of a two-storied house is generally not enough to create a strong stack effect. On the other hand, the study finds that wind-driven effect can easily create higher indoor air velocities up to 0.4 m/s. Although other studies suggest that buoyancy effect can be more effective, it needs extra efforts, for example, venting towers .These are not in common practice yet. Therefore, this present study focuses on natural ventilation caused only by wind pressure force, which is more practical in creating thermal comfort for occupants in hot-humid tropical climates. The main purpose of natural ventilation as a passive cooling strategy is to achieve high indoor air velocities with the air that has appropriate temperature and relative humidity. Factors that influence these parameters can generally be divided into two parts: outdoor environment and building component. It is known that landscape elements such as trees and water bodies can reduce the air temperature while hard-surface elements such as concrete grounds raise the air temperature. According to Givoni (1998), building components that affect natural ventilation include shape of the building, geometrical configuration, orientation of opening, window size and type, and subdivision of interior space.

In the present study, efforts were made to study the effect of ventilation on nocturnal cooling systems by regulating the air movement using the door and the windows in the test room. At the beginning of the experimental test, the indoor temperature in the experimental test rig was observed to be very high above the

ambient. This observation continued for a long time but when the door and the windows of the room were opened in another experimental day, the indoor temperature was observed to be lower in most case than the ambient air. The purpose of this exercise is to determine the viability of cooling a living space naturally without employing a mechanical ventilator such as fan and air conditioning system. Series of test conducted between the periods from the 30th of April, 2010 to 5th of May, 2010 reveal that the test room temperature was continuously lower than the control room. Results obtained are shown in fig 4.54 4.57.

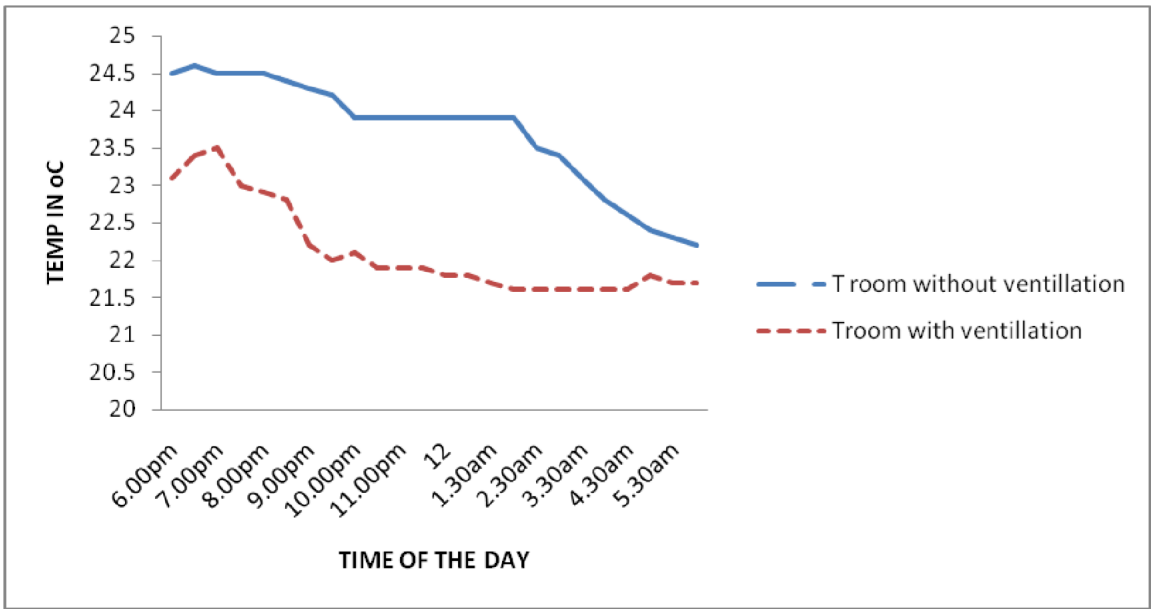


Fig 4.54 the effect of natural ventilation (30th April, 2010)

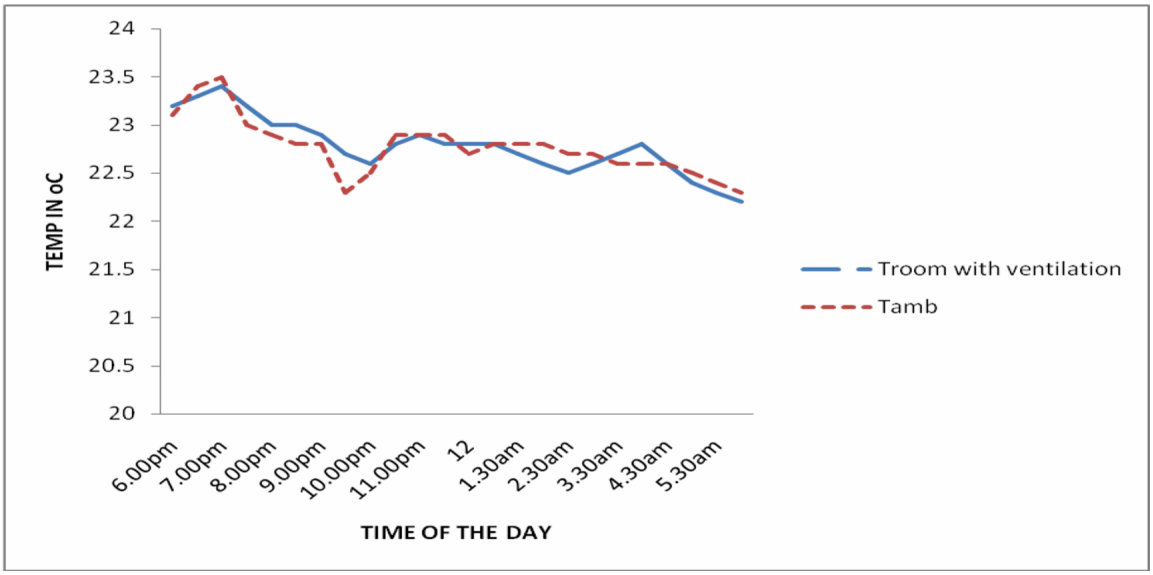


Fig 4.55 comparison between T_{amb} and T_{room} with ventilation (5th May,2010)

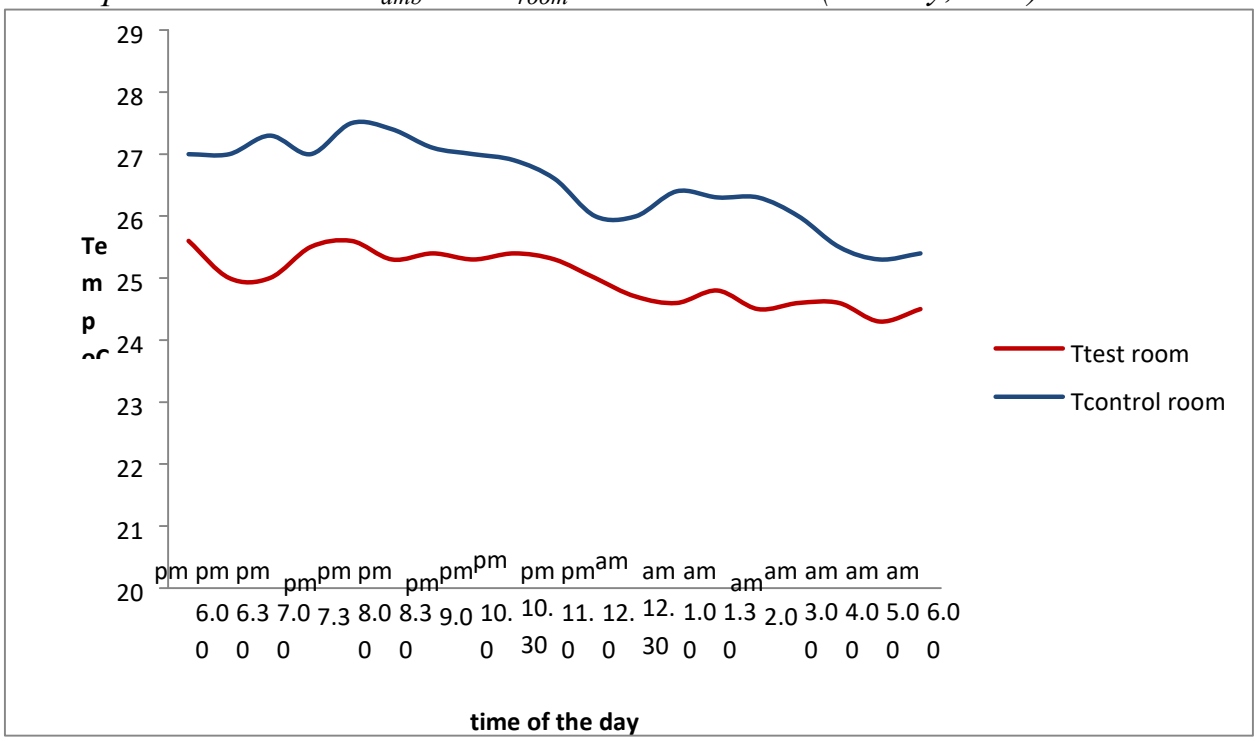


Fig 4.56 the effect of natural ventilation (3rd May, 2010)

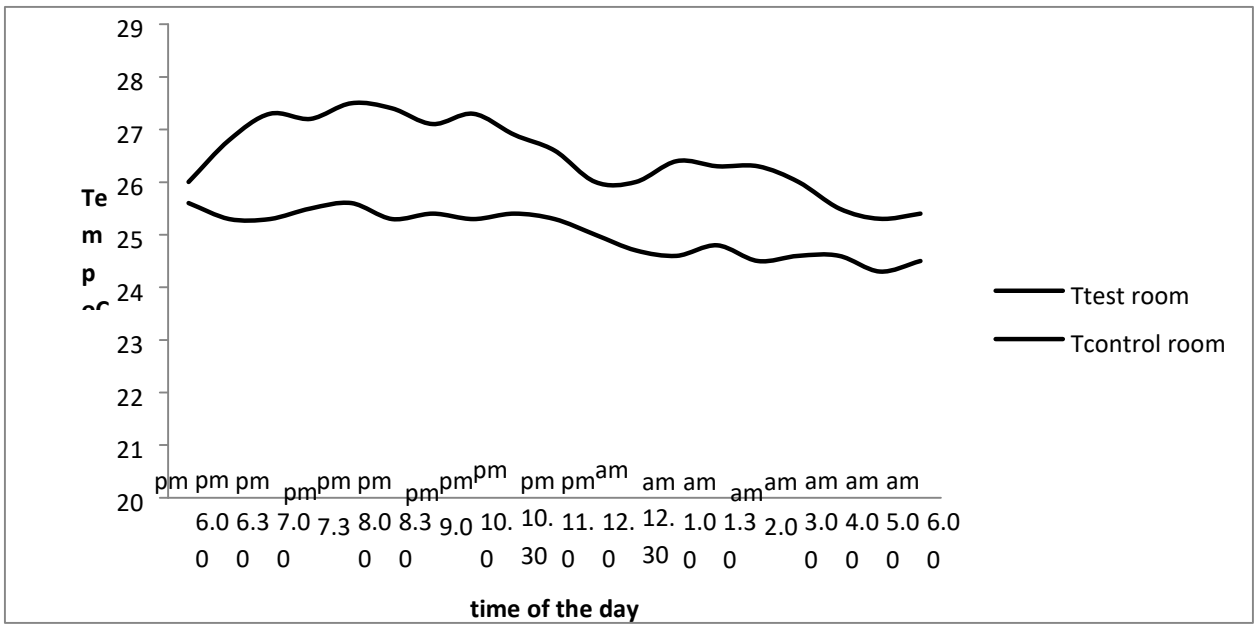


Fig 4.57 the effect of natural ventilation 5th May, 2010) The observed variations in fig 4.54- fig 4.57 underscore the need for the installation of natural ventilation in buildings. The above may stimulate low energy strategies for buildings especially in the tropical regions of the world where cooling loads are much higher. However, the architectural designs of building should aim at ensuring that low energy building designs are put in place to save cost in the cooling and heating needs of the buildings. The imperatives to strategize and optimize energy for building project performance in the tropics is very crucial and urgent as we know that energy is a major factor in propelling the wheel of economic development of nations. For a building to perform it must provide comfort to the occupants in case of residential occupation and stimulate services for other uses like industrial, agriculture, mining etc. When conventional energy sources are scare or inadequate, then the best alternative is to develop natural means of providing energy to homes.

4.12 The Effect of Relative Humidity

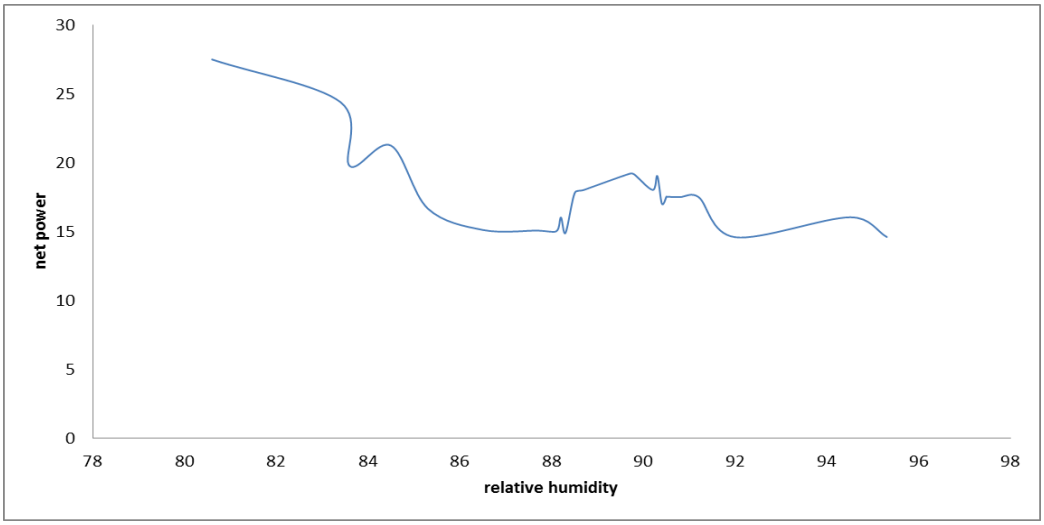


Fig 4.58 the effect of relative humidity on the net power (30th March,2010).

From the figure above, the net power decreases as the relative humidity increases. The above is very indicative that relative humidity plays a crucial role in the nocturnal radiative cooling of surfaces on the earth. The role of relative humidity has been exhaustively treated in so many literatures**, Meir et al (2002),** and its effect is very predominant in topical humid regions like Nigeria. It may not be surprising to say that arid regions are better disposed for radiative cooling than tropical regions because of the prevalence of moisture in humid climates.

4.13 System Economic Analysis

This section examines the cost of setting up a nocturnal cooling system. The costs of the components listed are the market value between October 2009 and May 2010, period during which the material were procured. The major application of the nocturnal cooling system is expected to be in remote rural area without grid connected electricity supplies. This economic analysis is carried out to ascertain the

feasibility of developing nocturnal cooling system to substantially reduce the cooling need of homes in the tropics.

Table 4.3 the bill of quantity of the nocturnal cooling system components

BILL OF QUANTITY FOR THE CONSTRUCTION OF THE NOCTURNAL COOLING SYSTEM					
Material Procured	QUANTITY	UNIT	RATE	LABOUR COST	TOTAL COST
One Inch PVCPipe s	10	In	250		2500
Petriolar Pump	1/4 HP				12000
Storage Tank	100	M^3			5000
Aluminum Sheet	11	M^2	2000		22000
Steel Tubes	12		200		2400
Saw Blades	6				400
Reducers	10				600
Construction Of Room					250000

Total	294900

Table 4.4 showing cost of installing the control unit

COST OF THE CONTROL UNIT FOR DATA ACQUISITION					
Material Procured	QUANTITY	UNIT	RATE	LABOUR COST	TOTAL COST
PENTIUM III COMPUTER	1		250		2500
DATA LOGGER	4		5000		20000
THERMOCOUPLES	6		500		3000
TEMPERATURE READ OUT	4		10000		40000

COMPUTER SOFTWARE PSYCHROGEN 2.0 FOR RELATIVE HUMIDITY, AIR VELOCITY CALCULATION	1	FREE	
Total			65500

The total cost of the system is the sum of the individual components which is made up of bill of quantity and the control unit cost is ₦ 350, 000.00. The cost of installing the nocturnal cooling system only is about ₦44900 However, for practical use, the incorporation of natural ventilation system is expedient, though it will increase the cost of the system. The cost of establishing this system on a commercial scale will include labour cost. The building structure account for more than 80 percent of the total cost . It a 3.0x 3.0 x2.5 m one bed room apartment, the walls are not shaded from solar insolation falling on them resulting in very high cooling potential value to reduce it during the night time.

From the bill of quantity presented above, a thorough economic analysis is therefore enumerated using the payback period method.

Payback period is the number of years required to recover the cost of the project. The payback period can be evaluated thus **Donald (2011)**

Payback period = Initial investment / Annual Cash inflows (4.1)

The payback period is dependent on cash inflows which are hard to predict. Therefore, payback period only considers revenue, and does not consider profits. To calculate the payback period for the nocturnal cooling system, it is expedient to

evaluate the annual cash inflows required to reduce the cooling load of a one bed room apartment. The cost of using a cooling system for the whole year may be obtained if the annual electricity consummation of that system is known.

Typical cooling systems include the fan, and the air conditioners. Manufacturers of this equipment usually specify their annual electricity consumption p. The consumption of the air conditioner ranges from 324 to 435kwh per annum and that of fan is in the range of 30 to 60kwh per annum. The present electricity tariff by the PHCN is about N10 per kWh.

The average electricity consumption of an air conditioner is about 380kwh while that of an electric fan is in the neighborhood of about 45kwh. Ordinarily, most households in Nigeria use both appliances at the same time. Therefore, the successful installation of an effective cooling system will eliminate the use of conventional electricity. Since the source of the alternative source of cooling is renewable, the saving from electricity bill due to the use of fans and air conditioners will be assumed to be equivalent to the cash inflow. If the present project should perform optimally, the electricity bill is eliminated and the total electricity cost will automatically become savings or return on investment since there will be no need to use the equipment. If the above fact is established, the cash inflow is therefore obtained as:

Cash inflows= power consumption x cost of electricity per kWh +cost of the Conventional cooling system. *(4.2)*

Cost of electricity for one year = (380+30) kWh x N10=₦4100.00.

Cost of procurement of the mechanical system=N30, 000

Therefore, cash inflow= N4100.00+N30000

Payback period= Initial investment / Annual Cash inflows

$= ₦44900/₦34100$

$=1.3$ *years*

CHAPTER FIVE

Conclusion and Recommendation

An experimental study is performed to investigate nocturnal radiation cooling of a building in Owerri, Imo State. The feasibility of using the cooled water from tank which has been previously cooled through nocturnal cooling for space cooling was studied. The experimental study identified effect natural ventilation and the various seasonal periods in Nigeria on nocturnal radiative cooling of buildings in Owerri as well as the effect of the ambient weather conditions on the total net cooling of a building.

The results are summarized as follows:

- The experimental results showed that plastic tank containing water with an open surface area of 1 m^2 and depth of 0.5 m can be cooled from an initial temperature of about 25.8 to 22.1^oC using nocturnal cooling.

- The study also showed that the radiator surface temperature can go below the ambient air temperature under the Owerri climate. In most cases, the temperature depression ranges from 1.0 to 3.5^oC below the ambient.

- The cooling power obtained during the dry season was shown to be greater than that of the raining season. A maximum net power of about 52.5W/m^2 was obtained during the dry season while for the wet season the cooling power dropped to 37W/m^2.

- The coefficient of performance for the system ranges over 0.01-0.09. The poor COP value may be attributed to high relative humidity value for Owerri region, which affects the net cooling of the system.
- The thermal capacitance of wall components has a large effect on the performance of the system and should be minimized.

- The daytime cooling experiment showed that a temperature depression of about 3.5^oC can be achieved by the incorporation a heat exchanger (room convector).

- The performance of the nocturnal cooling system is highly dependent on the prevailing weather condition and as such environmental limitation may be a major constraint to the performance of the system within regions of similar climatic condition.

The incorporation of the heat exchanger actually facilitated the day-time cooling of the building. For an average room temperature of about 28°C, the introduction of the heat exchanger actually reduced the room temperature to about 3.5^{o}C below the control room temperature. This was validated using an adjacent room temperature within the vicinity of the experimental test rig. While the adjacent room temperature was about 28.7^{o}C the test room temperature gave 24.6^{o}C. This value translates to 1428kJ of useful cooling.

The above results therefore indicate that:

- Nocturnal radiative cooling can be used to cool buildings in the tropical region.

- Daytime cooling is possible with the incorporation of a heat exchanger unit.

- The system performance can be improved significantly by incorporating natural ventilation unit.
- Economically, with natural ventilation, the nocturnal cooling system developed in this study will compete favorably with common vapour compression refrigeration system, and depending on geographical location, could become the preferred choice due to high cost of energy.

Nocturnal cooling system may be required in the following applications;

- Space cooling

- Production of cold for the preservation of foodstuff

- Water chilling.

The need for a nocturnal cooling system is greatest in the rural area of the world where conventional fuel and grid electricity are either non-existent or too expensive.

Its technology is cheap and inexpensive with little awareness; people can actually adopt the concept for space cooling, which is the greatest need of the African region. Moreover, it is a low grade energy which does not require an external source of energy for it operation. Though nocturnal cooling practice is an age long practice yet people are still ignorant of its great and vast potentials in combating the menace of global warming and the reduction of the greenhouse gas emission.

The concept and development of nocturnal cooling system is still at the research stages. The various possible techniques of nocturnal cooling system to provide cool can be classified into the direct and hybrid radiative cooling systems.

However, these techniques at their present state of development do not compare favorably in performance with the conventional vapour compression air conditioners and refrigerating system s. Here in Nigeria no meaningful work has been done on nocturnal cooling systems except for the preliminary investigation of passive cooling potential by **Ezekwe, (1986)** at the University of Nigeria Nsukka.

5.2 Achievement of the Present Effort and Contribution to Knowledge

The achievements of the work described in this report include the following:

- Nocturnal radiative cooling system has been successfully designed and constructed using locally available technology.

- The present study has proved that a nocturnal cooling activity is very possible in Owerri climate.
- The result of the actual field test this system have demonstrated that climate climatic cooling potential(CCP) of Owerri Nigeria is such that more cooling energy is needed to effectively cool a building unlike those in temperate climate where the environment is sufficiently cold enough to encourage nocturnal cooling of the living space. The significance of the climatic cooling potential is that regions with higher values of the climatic cooling potential require more energy for space cooling than regions with little value.
- The present study has shown that dry season is more appropriate for nocturnal radiative cooling than the wet season.
- The nocturnal cooling system shows good promise as an alternative to air conditioning for space cooling, especially in regions without grid connected electricity supply.
- The present study achieved a useful cooling load of about 1428.0kJ during the day without the aid of a conventional electric fan or vapour compression refrigerating system.
- This work has contributed to the growing body of knowledge on the performance characteristics of the nocturnal cooling system.
- In the extensive literature review presented, no example of previous work on this region of the world was noted except the work by Ezekwe (1986) which evaluated the long-wave radiation for night cooling at the University of Nigeria Nsukka.

- The work reported has also contributed towards the creation of awareness on the application of nocturnal radiative technology development in Nigeria.

5.3 Recommendation for Future Research

Nocturnal cooling has been shown to be effective in cooling residential building, and cold storage for agricultural products. The main limitation of the technique is that it is strongly dependent on the local climate. Therefore, to utilize the technique effectively, three factors are supposed to be put into serious consideration: location of usage, season of usage, and user‟s requirements in term s of temperature and humidity. The major problem which remains in the practical utilization of nocturnal cooling is the lack of better radiator materials. This aspect should be the focus of future research efforts. To date, the use of currently available materials in nocturnal cooling system has limited their temperature below the minimum or slightly above the minimum ambient. If this temperature should be lowered to 5-10 ^{o} C further, a much wider application would result. Better materials must therefore be developed for the radiator surface, to maximize cooling in the atmospheric window, and minimize absorption outside it. Materials developed for the above, must be durable, resistant to deterioration under adverse climatic conditions, and relatively cheap.

The result of the present work described in this report led to the following recommendations:

Although highly reflective surface coating gives better performance than a black surface according to **Parker (2002),** for this highly reflective surface to be effective, a transparent cover should be incorporated to prevent dust particles from tarnishing the radiator surface. This would imply additional cost to the system, its used could, significantly improve the radiator performance as heat gain through

convention will be eliminated completely. Where such cover is not available or too expensive, transparent polyethylene sack can be used.

The storage tank stores the water at a temperature below the room temperature. Heat gain through convection cannot be totally eliminated. However, a porous medium could be used in place of plastic tank. The porous pot will further cool the water through evaporative cooling. In that case, the temperature of the water will remain constant throughout because as the stored water gains heat, evaporative cooling removes such heat thereby maintaining a constant temperature within the tank.

The walls which are made of cement blocks have large thermal mass. In that case, the large thermal mass absorbs excess heat during the day adds heat to the system. Future work should consider using porous bricks lining sandwiched between the inner wall surfaces. As water flows between the wall and the porous bricks, evaporative cooling will help maintain a stable temperature within the building. Though this would imply additional cost and extra work to eliminate leakages, its use will significantly improve space cooling within the building interior

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APPENDIX

29th March,2011

	T rad	T amb	T room	T water	T dewp		comment
6.00pm	24.1	25.3	27.3	26.6	21.9		Clear weather
6.30pm	24.4	25	27	25.9	21.8		
7.00pm	24.4	24.8	26.7	25.5	21.7		
7.30pm	24.3	24.5	26.9	25.2	21.7		
8.00pm	24.2	24.2	26.8	25.6	21.6		
8.30pm	24.3	24.1	24.8	26.5	21.6		
9.00pm	24.4	24	25.6	26	21.6		
9.30pm	24.3	24	24.8	26	21.6		
10.00pm	24.1	24.2	24.7	24.5	21.5		
10.30pm	23.5	24.2	24.6	24.5	21.5		
11.00pm	23.5	23.7	24.3	24.5	21.5		
11.30pm	23.8	23.7	24.3	24.5	21.5		
12	23.2	23.1	24.6	24.5	21.4		
1.00am	22.7	23.7	24.5	24.5	21.4		
1.30am	22.6	23.4	24.4	24.5	21.4		
2.00am	23.1	23.8	24.4	24.5	21.4		
2.30am	23.1	23	24	24.1	21.3		
3.00am	22.8	23.1	23.9	24.2	21.3		
3.30am	22.5	23.2	23.8	24.1	21.3		
4.00am	22.9	22.6	23.7	24.5	21.2		
4.30am	22.5	22.7	23.2	24	21.2		
5.00am	22.5	22.9	23.8	24.1	21.1		
5.30am	22.4	22.8	23.5	24.1	21.1		
6.00am	21.8	22.9	23	23.6	21.0		

14th April, 2011

TIME	Trad	Tamb	Troom	Twater	Comment
6.00pm	24	25	25.9	27	**Rainy/ cloudy weather. Ambient condition very cool temperatures of radiator and ambient air almost the same. This may be attributed to the high moisture content in the atmosphere**
6.30pm	24.4	25	25.9	26.6	
7.00pm	24.4	24.8	25.9	26.5	
7.30pm	24	24.5	26.7	26.3	
8.00pm	24.1	24.5	26.5	26.5	
8.30pm	23.8	24.4	26.2	26	
9.00pm	23.4	24.3	26	26.6	
9.30pm	23.2	24.2	25.9	25.9	
10.00pm	21.9	22	25.5	25.9	
10.30pm	21.9	22	24	24.5	
11.00pm	21.5	21.8	24	24.4	
11.30pm	21.2	21.8	24	24.4	
12	22.1	23.2	23.8	24.6	
1.00am	22.3	23.2	25.3	24.4	
1.30am	22.4	23	25.2	25	
2.00am	22.5	23.1	25.5	25	
2.30am	22.3	23	25.5	24.1	
3.00am	22	22.9	25.5	24	
3.30am	21.8	22.6	25.5	24.3	
4.00am	21.9	22	25.5	24.2	
4.30am	21	21.3	25	24	
5.00am	21	21.9	25.2	24.3	
5.30am	20.9	21.3	24.9	24.1	
6.00am	20.9	21.2	24.9	24	

27/07/2010

Time	Trad	Tamb	T water	T room	comment

6.00pm	23.1	25.3	25.9	27	At the beginning the sky was overcast. Intermittently, the weather changes from clear to overcast to cloudy. The temperature variation was obvious in all these changes.
6.30pm	23.4	24.6	25.9	26.6	
7.00pm	23.5	24.5	25.9	26.5	
7.30pm	23	24.5	26.7	26.3	
8.00pm	22.9	24.5	26.5	26.5	
8.30pm	22.8	24.4	26.2	26	
9.00pm	22.2	24.3	26	26.6	
9.30pm	22	24.2	25.9	25.9	
10.00pm	22.1	23.9	25.5	25.9	
10.30pm	21.9	23.8	25.3	24.5	
11.00pm	21.9	23.5	25.2	24.4	
11.30pm	21.9	23.6	24.8	24.4	
12	21.8	23.2	23.8	24.6	
1.00am	21.8	22.8	25.3	24.4	
1.30am	21.7	22.7	25.2	25	
2.00am	21.6	22.6	25.5	25	
2.30am	21.6	23	25.5	24.1	
3.00am	21.6	22.9	25.5	24	
3.30am	21.6	22.6	25.5	24.3	
4.00am	21.6	22.5	25.5	24.2	
4.30 am	21.6	22.5	25	24	
5.00 am	21.8	22.3	25.2	24.3	
5.30 am	21.7	22.4	24.9	24.1	
6.00 am	21.7	22.2	24.9	24	

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TIME	Trad	Tamb	Twater	Troom	comment

6.00pm	24	25	25.9	27	The weather condition at the beginning was clear with high ambient condition. However, towards the midnight, the weather condition changed to a rainy cloudy sky.
6.30pm	24.4	25	25.9	26.6	
7.00pm	24.4	24.8	25.9	26.5	
7.30pm	24	24.5	26.7	26.3	
8.00pm	24.1	24.5	26.5	26.5	
8.30pm	23.8	24.4	26.2	26	
9.00pm	23.4	24.3	26	26.6	
9.30pm	23.2	24.2	25.9	25.9	
10.00pm	23	24.1	25.5	25.9	
10.30pm	22.9	23.9	24	24.5	
11.00pm	22.8	23.7	24	24.4	
11.30pm	22.5	23.6	24	24.4	
12	22.1	23.2	23.8	24.6	
1.00am	22.3	23.2	25.3	24.4	
1.30am	22.4	23	25.2	25	
2.00am	22.5	23.1	25.5	25	
2.30am	22.3	23	25.5	24.1	
3.00am	22	22.9	25.5	24	
3.30am	21.8	22.6	25.5	24.3	
4.00am	21.9	22	25.5	24.2	
4.30am	21.3	22	25	24	
5.00am	21.4	21.9	25.2	24.3	
5.30am	20.9	21.3	24.9	24.1	
6.00am	20.9	21.2	24.9	24	

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6.00pm	**24.7**	**25.3**	**25.9**	**27**	**comment**
6.30pm	24.4	24.6	25.9	26.6	Clear weather throughout

7.00pm	24.4	24.5	25.9	26.5	
7.30pm	24	24.5	26.7	26.3	
8.00pm	24.1	24.5	26.5	26.5	
8.30pm	23.8	24.4	26.2	26	
9.00pm	23.4	24.3	26	26.6	
9.30pm	23.2	24.2	25.9	25.9	
10.00pm	23	23.9	25.5	25.9	
10.30pm	23.2	23.8	25.3	24.5	
11.00pm	23.1	23.5	25.2	24.4	
11.30pm	22.7	23.6	24.8	24.4	
12	22.5	23.2	23.8	24.6	
1.00am	22.4	22.8	25.3	24.4	
1.30am	22.4	22.7	25.2	25	
2.00am	22.5	22.6	25.5	25	
2.30am	22.3	23	25.5	24.1	
3.00am	22	22.9	25.5	24	
3.30am	21.8	22.6	25.5	24.3	
4.00am	21.9	22.5	25.5	24.2	
4.30am	22.4	22.5	25	24	
5.00am	22	22.3	25.2	24.3	
5.30am	22	22.4	24.9	24.1	
6.00am	22.2	22.5	24.9	24	

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Time	**Trad**	**Tamb**	**Twater**	**Troom**	**T dewp**	**comment**
6.30pm	21.9	22.9	23.5	23	21.9	

7.00pm	21.9	22.9	23.5	23	21.8	**Rainy and cloudy weather throughout the night**
7.30pm	21.9	22.9	23.5	23	21.7	
8.00pm	21.9	22.9	23.5	23	21.7	
8.30pm	21.9	22.9	23.5	23	21.6	
9.00pm	21.9	22.9	23.5	23	21.6	
9.30pm	21.9	22.9	23.5	23	21.6	
10.00pm	21.9	22.9	23.5	23	21.6	
10.30pm	21.9	22.9	23.5	22.9	21.5	
11.00pm	21.9	22.9	23	22.9	21.5	
11.30pm	21.8	22.9	23	22.9	21.5	
12	21.6	22.9	23	22.9	21.5	
1.00am	21.6	22.9	23	22.9	21.4	
1.30am	21.6	22.9	23	22.9	21.4	
2.00am	21.6	22.9	22.9	22.9	21.4	
2.30am	21.6	22.8	22.9	22.8	21.4	
3.00am	21.6	22.7	22.9	22.3	21.3	
3.30am	21.6	22.6	22.9	22.4	21.3	
4.00am	20.9	22.5	22.9	22.4	21.3	
4.30am	20.9	22.1	22.9	22.4	21.2	
5.00am	20.8	21.9	22.8	22.5	21.2	
5.30am	20.8	21.9	22.7	22.5	21.1	
6.00am	20.7	21.9	22.5	22.4	21.1	
					21.0	

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TIME	**Trad**	**Tamb**	**Twater**	**Tdwep**	**Troom**	**T dewp**		**comment**
6.00pm	24.5	25.7	23.5	21.9	23	22		
6.30pm	24.5	25.7	23.5	21.8	23	21.8		

7.00pm	24.3	25.7	23.5	21.7	23	21.7		
7.30pm	24.3	24.9	23.5	21.7	23	21.7		
8.00pm	24.3	24.5	23.5	21.6	23	21.6		
8.30pm	23.8	24.2	23.5	21.5	23	21.6		
9.00pm	23.8	24.2	23.5	21.5	23	21.6		
9.30pm	23.8	24.2	23.5	21.5	23	21.6		
10.00pm	23.3	24.2	23.5	21.5	23	21.5		
10.30pm	23.1	24.2	23.5	21.5	22.9	21.5		
11.00pm	22.9	23.4	23	21.4	22.9	21.5		
11.30pm	22.8	23.7	23	21.3	22.9	21.5		
12	22.1	23.5	23	21.3	22.9	21.4		
1.00am	21.6	23.2	23	21.3	22.9	21.4		
1.30am	21.6	23.2	23	21.2	22.9	21.4		
2.00am	21.5	23.2	22.9	21.2	22.9	21.4		
2.30am	21.5	23.2	22.9	21.2	22.8	21.3		
3.00am	21.5	23.2	22.9	21.1	22.3	21.3		
3.30am	21.5	22.8	22.9	21.1	22.4	21.3		
4.00am	21.5	22.8	22.9	21.1	22.4	21.2		
4.30am	21.3	22.8	22.9	21	22.4	21.2		
5.00am	21.2	22.8	22.8	21	22.5	21.1		
5.30am	20.8	21.7	22.7	21	22.5	21.1		
6.00am	20.7	21.9	22.5	20.9	22.4	21.0		

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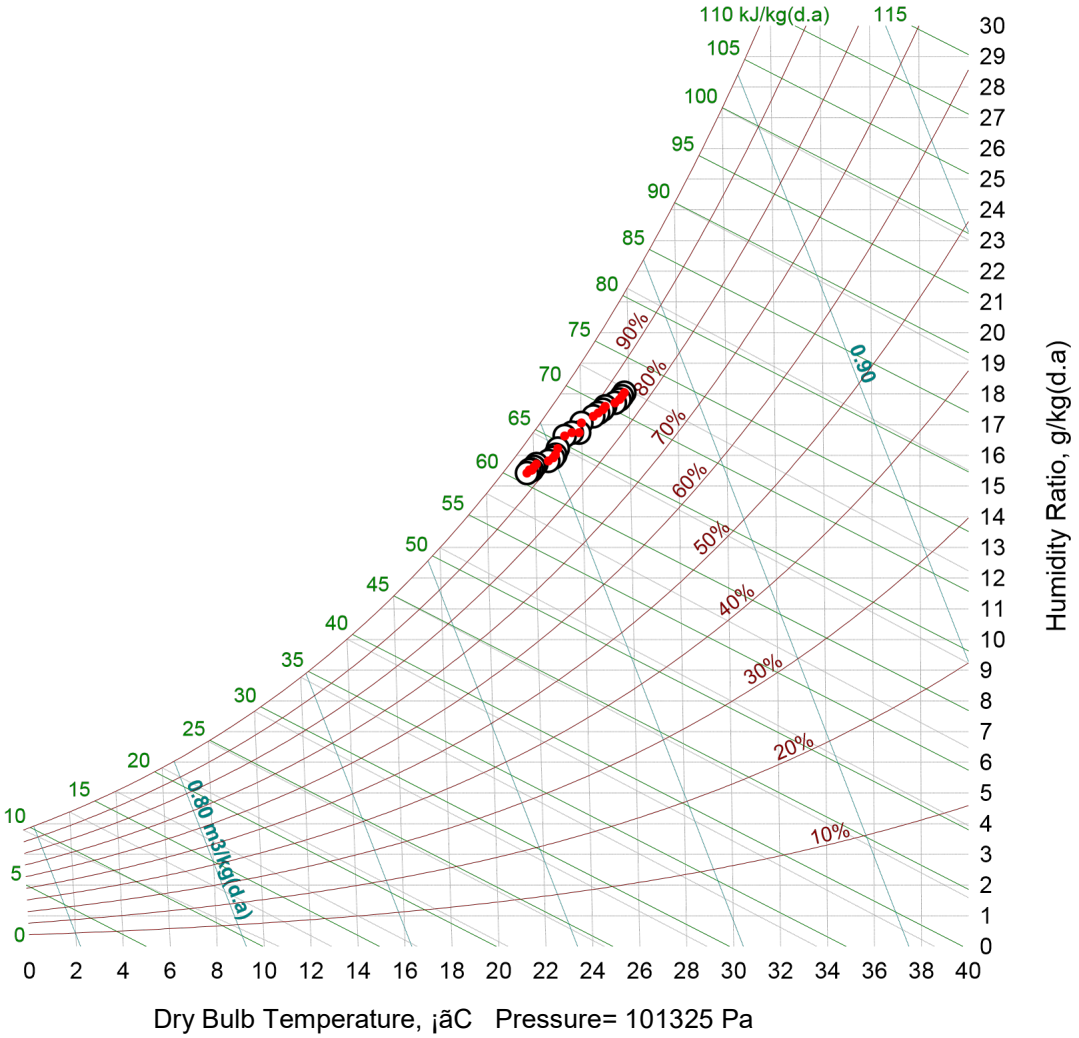
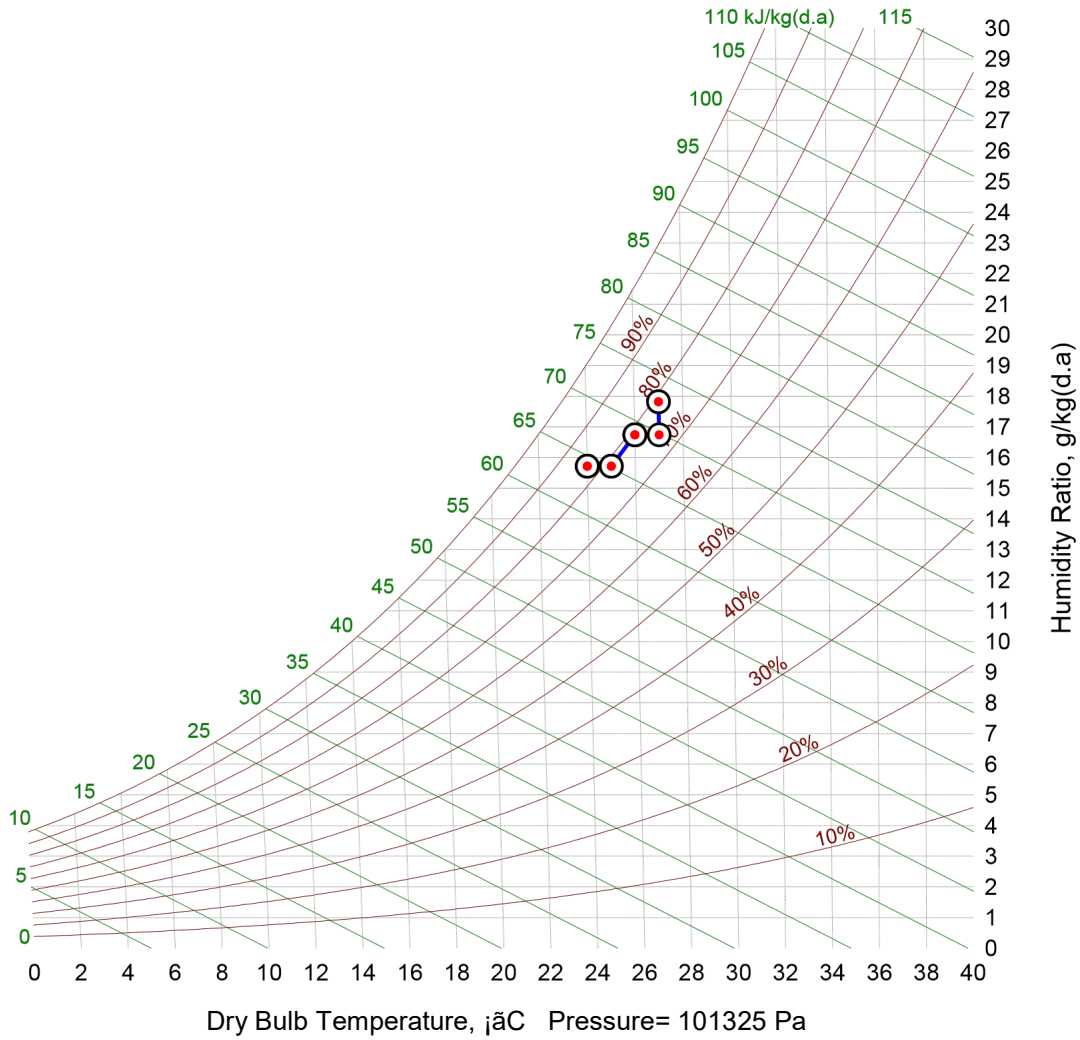
TIME	**Trad**	**Tamb**	**Twater**	**Troom**	**COMMENT**
6.00pm	25.6	26	25.9	27	Clear weather
6.30pm	25.3	25	25.9	26.6	throughout
7.00pm	25	24.8	25.9	26.5	

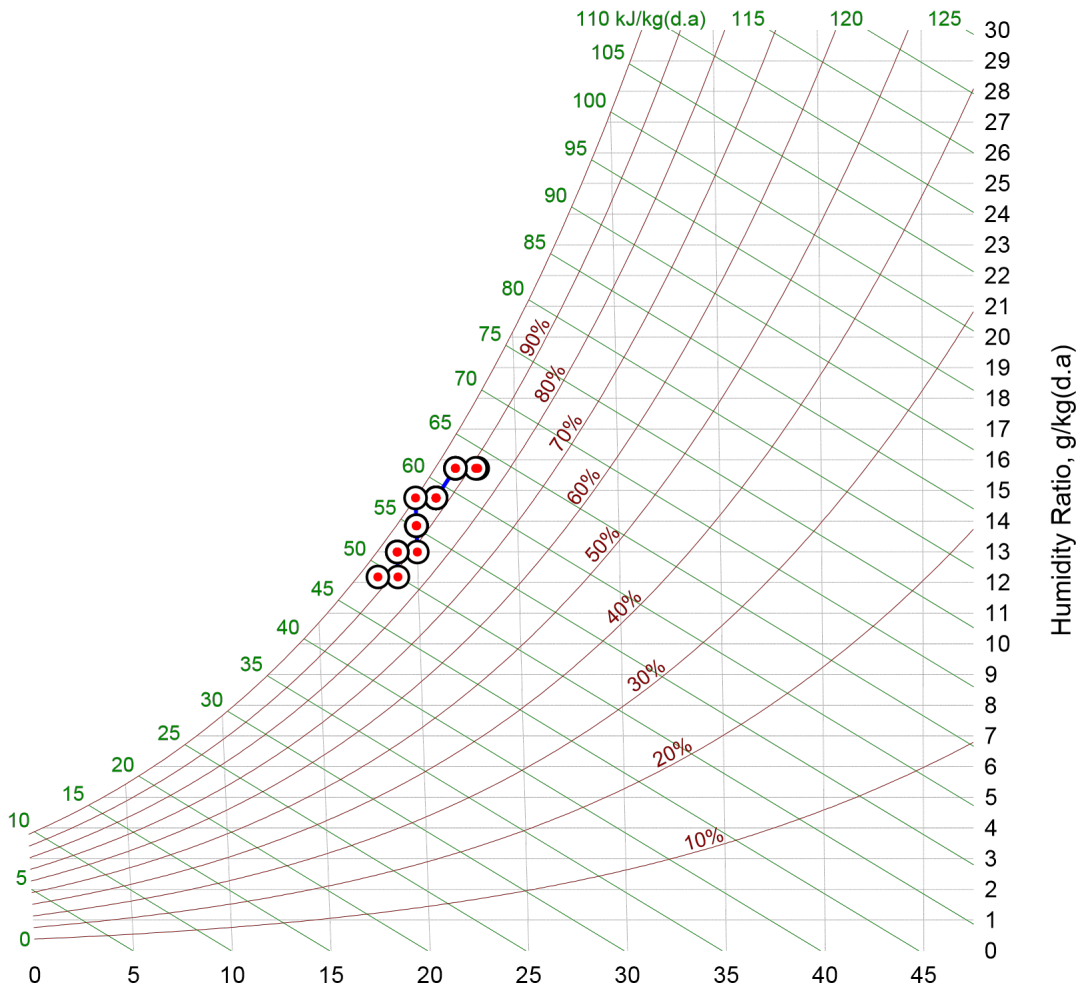
7.30pm	25	24.5	26.7	26.3	night the
8.00pm	24.9	24.5	26.5	26.5	radiator
8.30pm	24.4	24.4	26.2	26	surface
9.00pm	24	24.3	26	26.6	temperature
9.30pm	23.3	24.2	25.9	25.9	was
10.00pm	23.3	24.1	25.5	25.9	constantly
10.30pm	22.9	23.9	25.5	25.9	lower than
11.00pm	22.8	23.7	25.5	25.5	that of the
11.30pm	22.5	23.6	25.3	25.5	ambient
12	22.5	23.2	25.3	25.5	
1.00am	22.3	23.2	25.3	25.6	
1.30am	22.4	23	25.2	25	
2.00am	22.2	23.1	25.5	25	
2.30am	22.7	23	25.5	25.2	
3.00am	22.8	22.9	25.5	25.2	
3.30am	22.5	22.6	25.5	25.2	
4.00am	22	22.7	25.5	24.9	
4.30am	22.1	22.9	25	24.8	
5.00am	22.1	22.8	25.2	24.3	
5.30am	22	22.6	24.9	24.1	
6.00am	22	22.7	24.9	24	


110 kJ/kg(d.a)
115
105
100
95
90
85
80
75
70
65
60
55
50
45
40
35
30
25
20
15
10
5
0
90%
80%
70%
60%
50%
40%
30%
20%
10%
0.90
0.80 m3/kg(d.a)
0 2 4 6 8 10 12 14 16 18 20 22 24 26 28 30 32 34 36 38 40
Dry Bulb Temperature, ¡ãC Pressure= 101325 Pa
Humidity Ratio, g/kg(d.a)
30 29 28 27 26 25 24 23 22 21 20 19 18 17 16 15 14 13 12 11 10 9 8 7 6 5 4 3 2 1 0

07/05/10



12/05/10


110 kJ/kg(d.a)
115
120
125
105
100
95
90
85
80
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70
65
60
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40
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90%
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19
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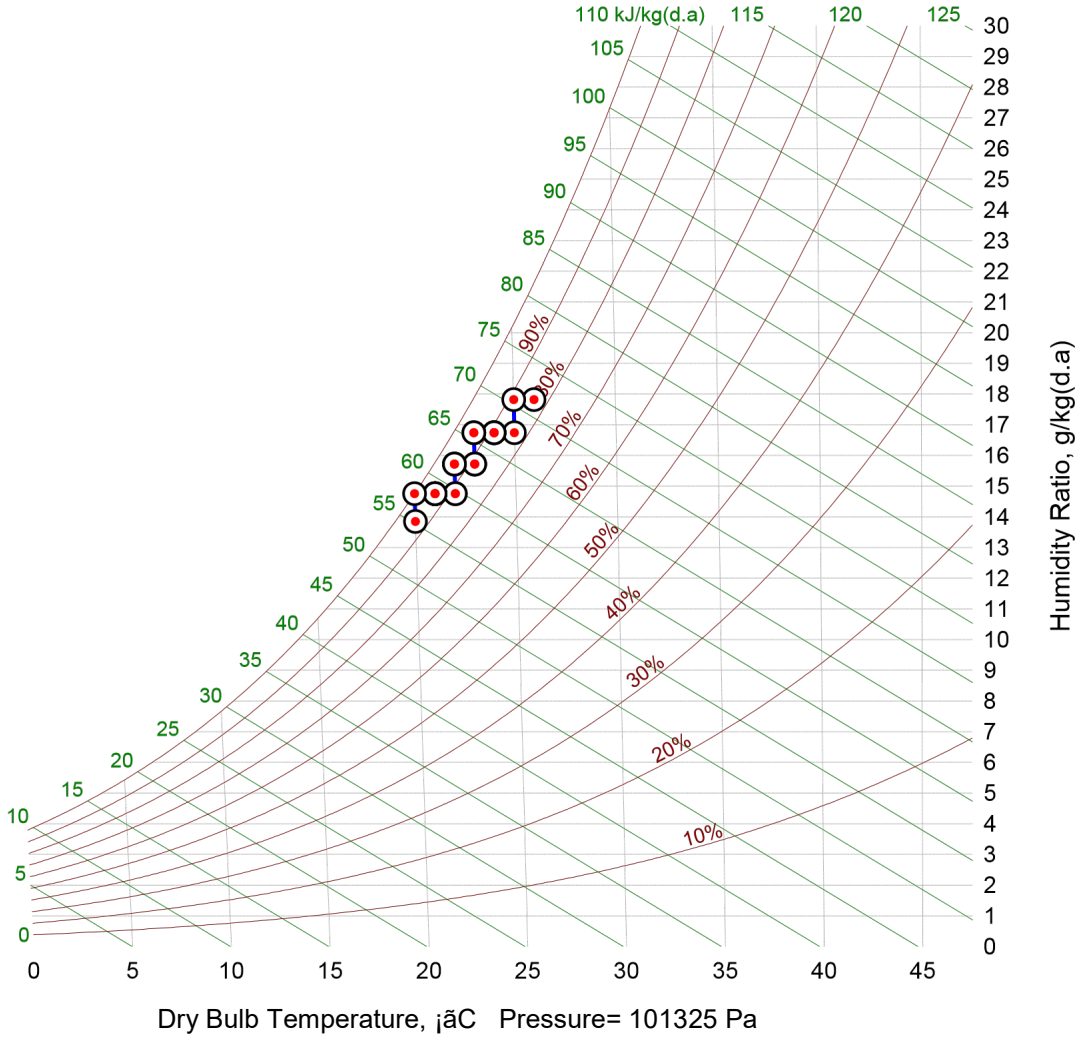
17
16
15
14
13
12
11
10
9
8
7
6
5
4
3
2
1
0
Humidity Ratio, g/kg(d.a)
0
5
10
15
20
25
30
35
40
45
Dry Bulb Temperature, ¡ãC Pressure= 101325 Pa

15/07/10


110 kJ/kg(d.a)
115
120
125
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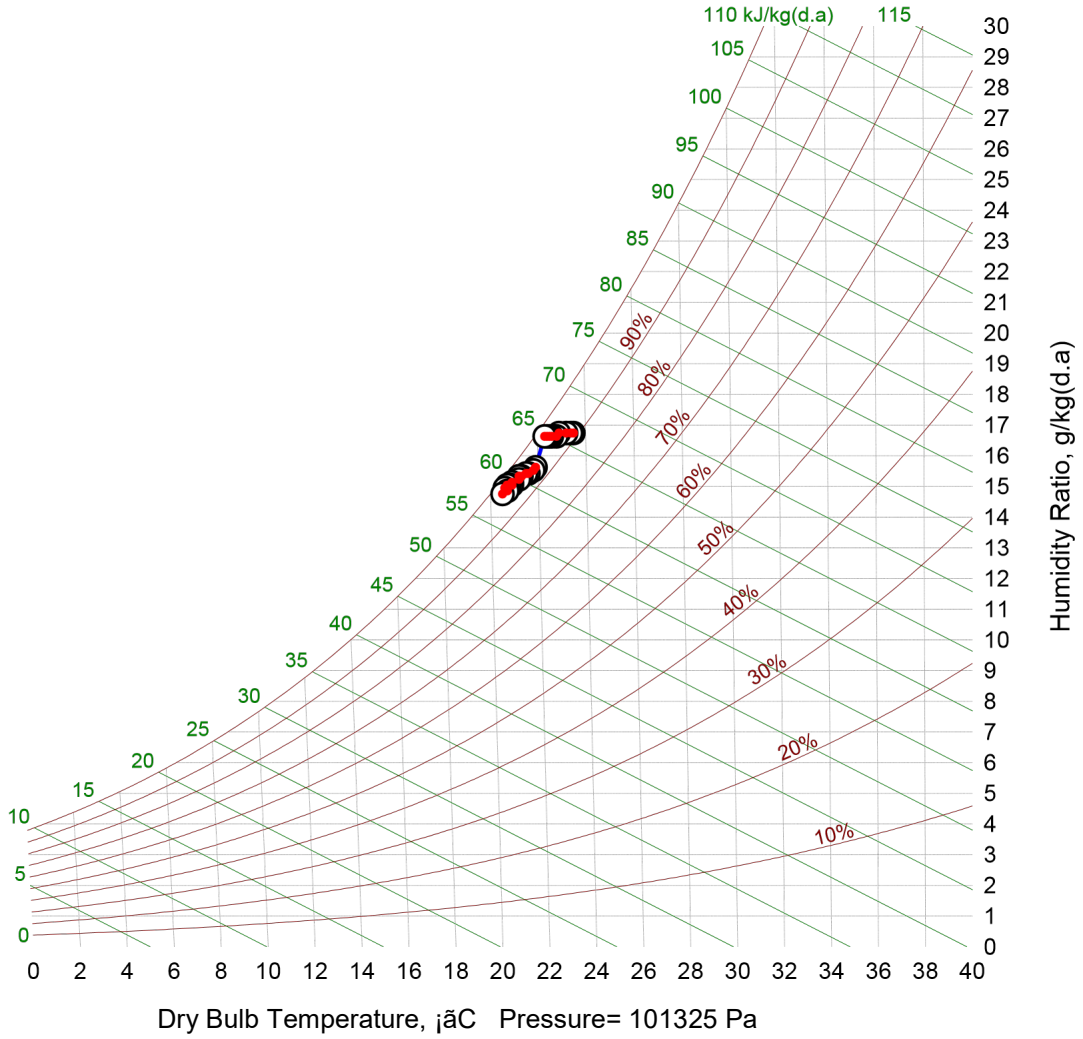
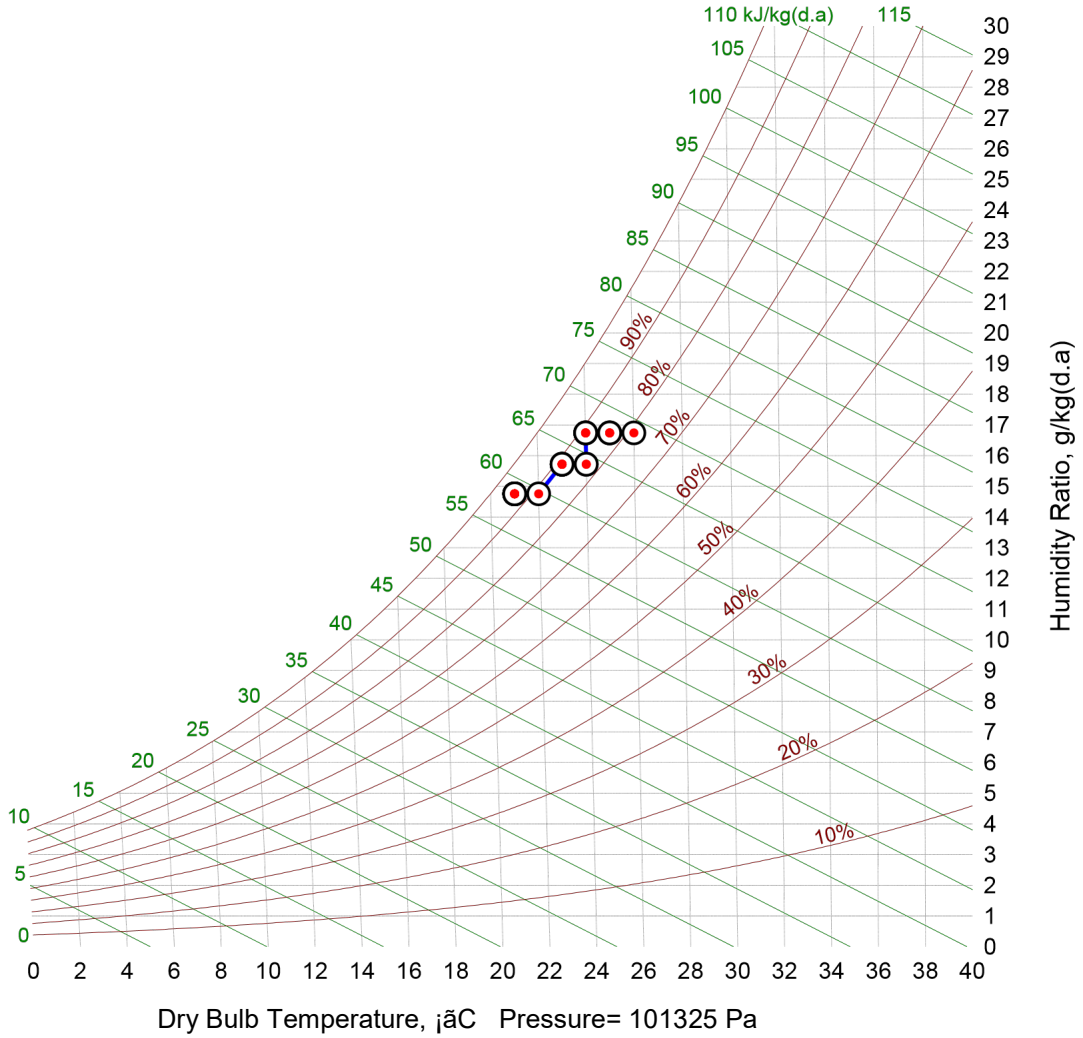
Dry Bulb Temperature, ¡ãC Pressure= 101325 Pa
Humidity Ratio, g/kg(d.a)

20/07/10



29/07/10


110 kJ/kg(d.a)
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60%
50%
40%
30%
20%
10%
0 2 4 6 8 10 12 14 16 18 20 22 24 26 28 30 32 34 36 38 40
Dry Bulb Temperature, ¡ãC Pressure= 101325 Pa
Humidity Ratio, g/kg(d.a)

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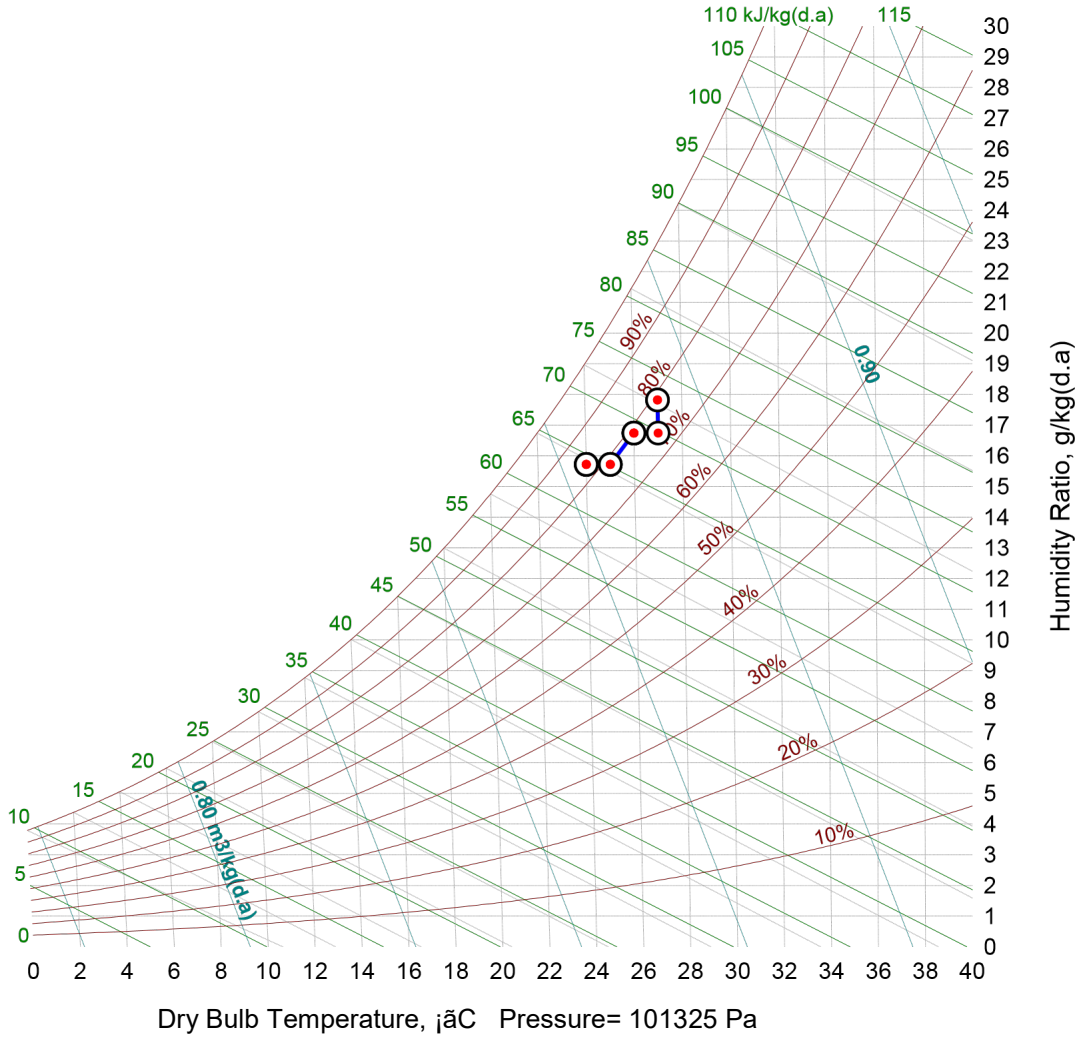
110 kJ/kg(d.a)
115
105
100
95
90
85
80
75
70
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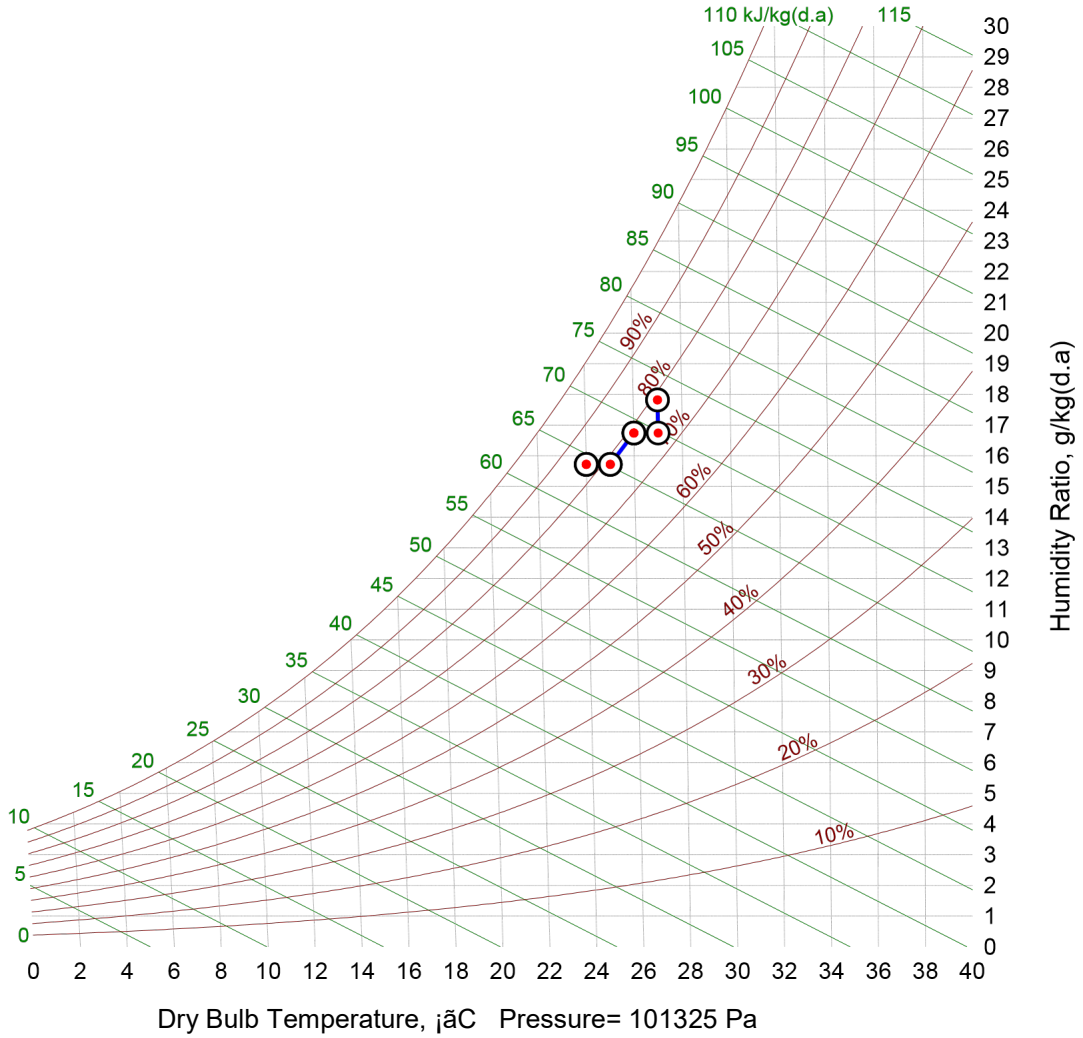
60
55
50
45
40
35
30
25
20
15
10
5
0
90%
80%
70%
60%
50%
40%
30%
20%
10%
0.90
0.80 m3/kg(d.a)
0 2 4 6 8 10 12 14 16 18 20 22 24 26 28 30 32 34 36 38 40
Dry Bulb Temperature, ¡ãC Pressure= 101325 Pa
Humidity Ratio, g/kg(d.a)
30 29 28 27 26 25 24 23 22 21 20 19 18 17 16 15 14 13 12 11 10 9 8 7 6 5 4 3 2 1 0

20 /08/ 10


110 kJ/kg(d.a)
115
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90%
80%
70%
60%
50%
40%
30%
20%
10%
0 2 4 6 8 10 12 14 16 18 20 22 24 26 28 30 32 34 36 38 40
Dry Bulb Temperature, ¡ãC Pressure= 101325 Pa
Humidity Ratio, g/kg(d.a)

Absorptivity & Emissivity table of some materials

Material	Solar Absorption	Surface Emissivity □
Silver, Highly polished		0.02 - 0.03
Gold, Highly polished		0.02 - 0.04
Barium Sulphate with Polyvinyl Alcohol	0.06	0.88
Aluminum polished	0.09	0.03
Magnesium Oxide Paint	0.09	0.90
Magnesium/Aluminium Oxide Paint	0.09	0.92
Aluminum quarts overcoated	0.11	0.37
Aluminum, Highly polished		0.04 - 0.06
Snow, Fine particles fresh	0.13	0.82
Zinc Orthotitanate with Potassium Silicate	0.13	0.92
Aluminum anodized	0.14	0.84
Aluminum foil	0.15	0.05
Potassium Fluorotitanate White Paint	0.15	0.88
Zinc Oxide with Sodium Silicate	0.15	0.92