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Transient Analysis and Performance Prediction of Nocturnal Radiative Cooling of a Building in Owerri, Nigeria

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Abstract: A study aimed at a Transient analysis and performance prediction of passive cooling of a building using long wave nocturnal radiation in Owerri, Nigeria is presented. The system modeled consists of the room of a building with a radiator panel attached to its roof, water storage tank located inside the room, pump to circulate water through the radiator panel at night and through a heat exchanger in the room during the day. The mathematical model is based on the thermal radiation properties of the local atmosphere, the heat exchange equations of the radiator panel with the sky during the night and the equations incorporating the relevant heat transfers within the space to be cooled during the day. The resulting equations were transformed into explicit finite difference forms for easy implementation on a personal computer in MATLAB language. This numerical model permits the evaluation of the rate of heat removal from the water storage tank through the radiator panel surface area, $Q_{wt,out}$, temperature depression between the ambient and room temperatures (T_{amb}-T_{rm}) and total heat gained by water in the storage tank from the space to be cooled through the action of the convector during the day, $Q_{wt,in}$. The resulting rate of heat removal from the radiator gave a value of 57.6 W/m^2, temperature depression was predicted to within 1-1.5ºC and the rate of heat gain by the storage water was 60 W/m^2. A sensitivity analysis of the system parameters to ±25% of the base case input values was carried out and the results given as a percentage variation of the above system performance parameters showed consistency to the base case results. An optimal scheme for the modeled 3.0×3.0×2.5 m^3 room showed a radiator area of 18.2 m^2, a convector area of 28.62 m^2 and a tank volume of 1.57 m^3.These results show that passive nocturnal cooling technique is a promising solution to the cooling needs for preservation of food and other agricultural produce. It is also useful in small office space cooling. Thus the model developed is undoubtedly a useful design tool for the development of passive cooling systems that can reduce considerably the huge cooling cost requirements of mechanical air conditioning systems.

Keywords: Finite difference, nocturnal cooling, radiative, temperature, transient

INTRODUCTION

When a surface on the earth faces the night sky, it loses heat by radiation to the sky and gains heat from the surrounding air by convection. If the surface is a relatively good emitter of radiation, it radiates more heat to the sky at night than it gains from the surrounding air. The net result is that the surface temperature drops below that of the surrounding air. Since at night the insolation is zero, the surface cooling goes on throughout the night. This night-time cooling is called radiative cooling. Figure 1 is a schematic diagram of radiative cooling process.

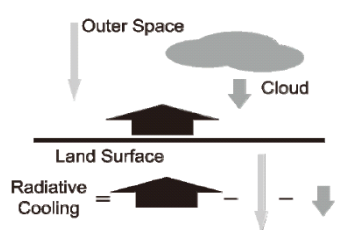

Fig. 1: Radiation cooling at night

Radiative cooling is one passive cooling technique utilizing the transmittance of the earth's atmosphere for thermal radiation in the wavelength interval from approximately 8-14 μm (Meir *et al*., 2003). The thermal radiation of a black body at ambient temperature on the earth's surface interacts with higher and colder atmospheric layers and may cool down below the ambient air temperature under optimal conditions. The maximal cooling power of a body at ambient temperature with high

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infrared emittance is of the order of 100 W/m^2 for clear night sky and low air humidity.

Radiative cooling is a very popular and well-known phenomenon and has been observed and referred to by many authors. Despite its many potential applications, including storage of food, seed and medicine, air-conditioning of buildings and water desalination, its commercial exploitation is still largely untapped (Dobson *et al*., 2003; Armenta-Deu *et al*., 2003). According to Hardy (2008), night time cooling is feasible if any or a combination of the following occurs:

- Peak indoor zone temperature exceeds 23ºC
- Average zone indoor temperature exceeds 22ºC
- Average afternoon outdoor temperature is higher than 20ºC

Similarly, night time cooling should continue to be used if all of the following criteria occur:

- The indoor zone temperature is higher than the outside air temperature by +2ºC
- The indoor zone temperature is higher than indoor heating set point
- The outside air temperature is higher than 12ºC

Various works have been undertaken towards studying the nocturnal cooling phenomena. Among them include the works by Etzion and Erell (1991) and Erell and Etzion (1992) in which they demonstrated that the key to improving radiative cooling systems for buildings lay in the recognition that sustaining a high cooling rate was possible only if the radiating surface remained relatively warm. This required a means of extracting the energy absorbed in the thermal mass of the building during the daytime to a radiator, where it might be dissipated to the environment at night. The cooling system proposed consisted of a shallow roof pond insulated from the environment and flat plate collectors exposed to the sky, through which the water was circulated at night to be cooled by long wave radiation and convection. Building on this, Erell and Etzion (1999) conducted a systematic analysis of the characteristics of a radiator designed specifically for the nocturnal, long wave radiative cooling of buildings. The objective was to maximize the cooling output per unit area of radiator, neglecting further improvements in the integration of the cooling system in the test building. They concluded that the following specifications were needed for the design of a radiator specifically for nocturnal cooling by long wave radiation:

- The distance between the pipes in the radiator should be kept to a minimum
- A turbulent flow regime within the pipes is required for a slightly better heat exchange between the fluid and the pipe walls
- If needed, the length of the radiator may be adapted to the geometry of the roof on which it is installed.

Also, since the cooling power produced by a flat plate radiator depended on its surface temperature, the cooling system as a whole should be designed to control the following factors:

- The radiator inlet temperature should ideally be as warm as the warmest part of the building to be cooled.
- The mass flow rate should be controlled so as to achieve a fairly flat longitudinal temperature profile, so that the average surface temperature of the radiator is high.

The mean nightly cooling output of the radiator achieved-due to the combined effect of radiation and convection-was over 90 W/m^2 under typical desert meteorological conditions.

Ali *et al*. (1996) studied nocturnal cooling of water flowing through a night sky radiator. They used two parallel plates night sky radiator with the top plate made from aluminium and painted black. The radiator plate was covered by a polyethylene windscreen cover. The performance of such unit was studied by the water temperature difference, the cooling power and the overall efficiency. They discovered that for typical hot dry summer nights and for open flow systems (gravity flow) having water mass flow rate ranging from 4.8-20.2 kg/h, the value of the optimum water mass flow rate for maximum cooling power was about 17 kg/h. However, the lowest water temperature of 16.3ºC was obtained at lowest mass flow rate of 4.8 kg/h, as expected. Also, evaporation produces a slight enhancement in performance of the whole system. However, cooling power and efficiency of the radiator were lower for uncovered supply tank than those of covered supply tank. The thermal capacitance of the radiator components was observed to have has large effect on the overall performance of the system and therefore should be minimized for improved performance.

In another work, Dobson (2005) studied numerically the nocturnal cooling phenomena in a typical tropical region. The analysis was based on a steady state phase of the system flow. This present work is building on this analysis by approaching it from the transient flow state condition and using the finite difference analysis scheme. A case study of Owerri in Nigeria will be utilized to evaluate this study.

The fundamental approach to this numerical problem lies in the replacement of the differential equation with a finite-difference equation. A thermodynamics energy balance approach is utilized to present temperature as a time depending variable. Mathematical modeling (finite difference scheme) is employed to generate effective relationships (models) towards predicting the night cooling resource of a typical system under consideration.

SYSTEM REPRESENTATION/DESCRIPTION

Figure 2 is a typical integrated nocturnal cooling system under consideration. Subsystems within the system include a radiator panel, circulating pump, water storage tank, connecting pipe network, data loggers/temperature sensors and room convector. Water from the storage tank is circulated through the radiator

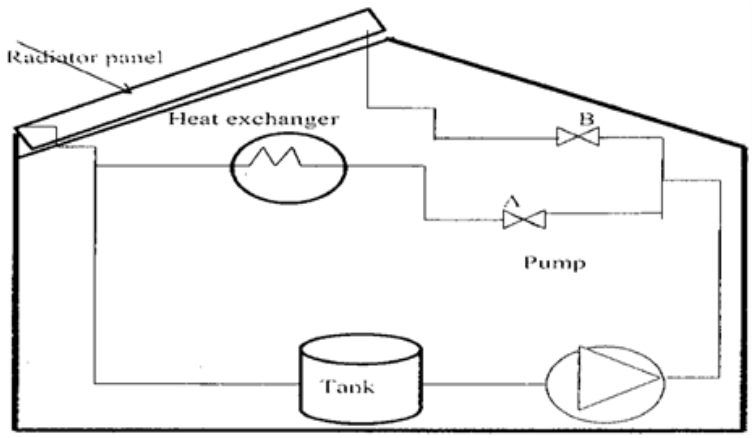

Fig. 2: Configuration of an integrated nocturnal cooling system

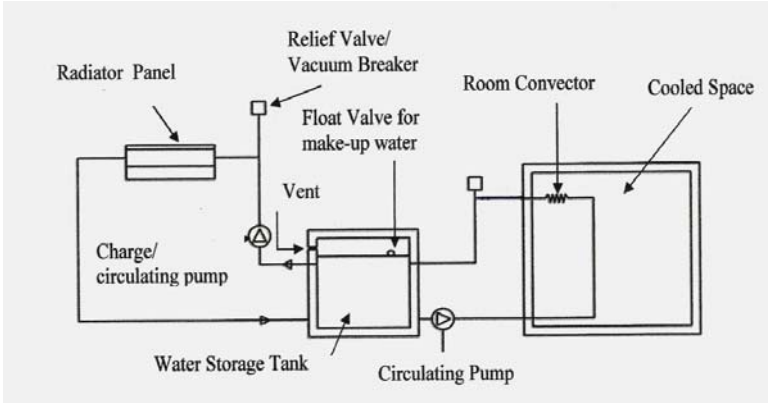

Fig. 3: Process diagram of the nocturnal cooling system

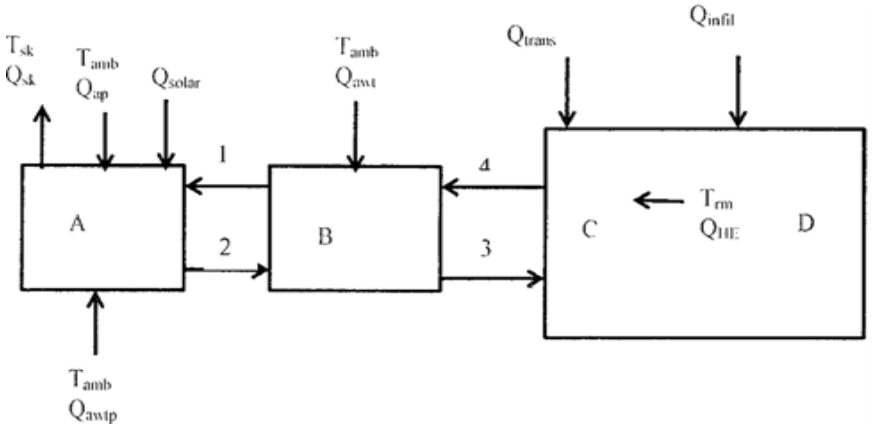

Fig. 4: Integrated control volume of subsystems of A = radiator panci, B = water storage tank, C = room heat exchanger and D = cooled room space

panel at night where it is cooled by the night sky radiation. The cooled water is stored in the tank and subsequently used to cool the air in the room during the day by natural convection. This is achieved by circulating the cold water through the convector in the room.

Figure 3 is a process diagram of the cooling system showing the loop/network arrangement of the component parts of the nocturnal cooling system used in this analysis. Figure 4 is an integrated control volume representation of the subsystems using block diagrams

FINITE DIFFERENCE SCHEME

Using the thermodynamics energy balance approach: Energy flow into a system control volume, Q_{in} + Energy generated internally in the system control volume, Q_{ixn} = Energyloss from the control volume, Q_{out} + change in internal energy of the system control volume, H.

$$i,e.,\ \dot{Q}_{in} + \dot{Q}_{ixn} = \dot{Q}_{out} + H \quad (1)$$

where,

$$H = mc\ (\delta T/\delta t) \quad (2)$$

Applying the forward finite difference scheme to the partial derivative of (2) as below:

$$(\delta T/\delta \tau)_{\xi,\tau,fwd} \approx 1/\delta\tau\ [T(\xi,\tau + \delta\tau) - T(\xi,\tau)] \quad (3)$$

The partial derivative in linear form with respect to time becomes:

$$\delta T/\delta t = (T^{t+1} - T^{t})/\ \Delta t \quad (4)$$

Therefore, change in internal energy of the system is given as:

$$H = (m_t c_w/\Delta t)\ (T^{t+1} - T^{t}) \quad (5)$$

Radiator panel: In analyzing the heat transfer modes crossing the boundary of the radiator panel, the following assumptions are made:

- The radiator panel is made of a thin Aluminium plate with high thermal conductivity, thus the resistance to conduction across the plate thickness is negligible. This is typically true for solids of large thermal conductivity with surface areas that are large in proportion to their volume such as plates and thin metallic wires.
- The lumped parameter analysis is used on the panel so that at any given interval of time, the temperature distribution on the radiator is the same.
- Axial heat conduction and viscous dissipation for the working fluid (water) are neglected.
- The internal heat generation in the system is negligible. Following assumption 1 above, the internal resistance of the radiator panel (L/kA) can be assumed to be small or negligible in comparison with the convective resistance (1/hA) at the surface.
- The thermal resistance between the water and the radiating surface is relatively small, so that the temperature of the water in the panel will thus be more or less equal to the radiating surface temperature.
- The radiator panel circulation pump is turned on before mid-night at about 11 pm to utilize the heat lost by the radiator panel to the sky to cool the water in the storage tank and then switched off at about 7 am the next day.

Energy balance for the radiator control volume:
Energy balance about this control volume requires that: Energy flow into the control volume, Q_{in} + energy generated internally in the control volume, Q_{ixn} = Energy loss from the control volume, Q_{out} + change in internal energy of the control volume, H:

$$i.e\quad \dot{Q}_{in} + \dot{Q}_{ixn} + \dot{Q}_{out} + H \quad (6)$$

Following assumption 4 above, the internal energy generation is negligible and is eliminated from the equation.

Thus, Eq. (6) reduces to:

$$\dot{Q}_{in} = \dot{Q}_{out} + H \quad (7)$$

From the control volume of Fig. 4, the energy balance equation for the parameters of Eq. (7) becomes:

$$\dot{Q}_{in} = \dot{Q}_{ap} + \dot{Q}_{solar} + \dot{Q}_{awtp} + \dot{m}_{w1} c T_{w1} \quad (8)$$

$$\dot{Q}_{out} = \dot{Q}_{sk} + \dot{m}_{w1} c T_{w2} \quad (9)$$

Using the finite difference method earlier explained in section 3.0, the change in internal energy, H of the control volume is:

$$H = mc\ (\delta T/\delta t) \quad (10)$$

$$H = (m_t c_w/\Delta t)\ (T^{t+1} - T^{t}) \quad (11)$$

Energy flow into the control volume consists of the convective heat transfer between the air and the panel, Q_{ap}; the overall solar radiation absorbed by the panel, Q_{solar}; the back insulation heat transfer from the ambient air to the panel through the back insulation, Q_{awtp} and energy due to water from the storage tank. The radiator

surface losses heat to the night sky, Q_{sk}; and from water returning to the storage tank; while there is no internal energy generation in the radiator panel as earlier explained.
Substituting Eq. (8), (9) and (11) into (7), we have:

$$\dot{Q}_{ap} + \dot{Q}_{solar} + \dot{Q}_{awtp} + \dot{m}_{w1}cT_{w1} = \dot{Q}_{sk} + \dot{m}_{w1}cT_{w2} + (m_t c_w/\Delta t)(T^{t+1} - T^t) \quad (12)$$

Simplifying further, we have:

$$\dot{Q}_{ap} + \dot{Q}_{solar} + \dot{Q}_{awtp} + \dot{m}_{w1}cT_{w1} - \dot{m}_{w1}cT_{w2} - \dot{Q}_{sk} = (m_t c_w/\Delta t)(T^{t+1} - T^t) \quad (13)$$

And rearranging further still, Eq. (13) becomes:

$$\dot{Q}_{ap} + \dot{Q}_{solar} + \dot{Q}_{awtp} + \dot{m}_{w1}c\,(T_{w1} - T_{w2}) \dot{Q}_{sk} = (m_t c_w/\Delta t)\,(T^{t+1} - T^t) \quad (14)$$

To solve the energy balance Eq. of (14), recall from heat transfer principles that heat transfer rate between two points is given as:

$$\dot{Q} = (T_1 - T_2)/\,(R_{th}) \quad (15)$$

Using Eq. (15) to analyze the heat transfer modes in Eq. (14), the resultant energy balance equation from (Eq. 14) now becomes:

$$(m_t c_w/\Delta t)(T^{t+1} - T^t) = \{[(T_{amb} - T_p)/R_{awtp}] + [(T_{amb} - T_p)/R_{ap}] - [(T_p - T_{sk})/R_{psk}] + [(\theta\alpha\rho G A_s)] + [\dot{m}\,c\,(T_{w1} - T_{w2})]\} \quad (16)$$

Rearranging further,

$$T^{t+1} = T^t + (\Delta t/m_t c_w)\{[(T_{amb} - T_p)/R_{awtp}] + [(T_{amb} - T_p)/R_{ap}] - [(T_p - T_{sk})/R_{psk}] + [(\theta\alpha\rho G A_s)] + [\dot{m}\,c\,(T_{w1} - T_{w2})]\} \quad (17)$$

Equation (17) is the functional relationship for the lumped parameter analysis of the radiator panel.

Water storage tank: The water storage tank is well insulated on both sides and water flow in and out of the tank is uniform. The water storage tank, considered as a control volume, has an average temperature T_{wt}. The tank has a layer of insulation of thickness L_{wt} and thermal conductivity K_{wt}. The tank dimensions are chosen so that its height equals its diameter, therefore it is possible to express its surface area and volume in terms of only one variable, its diameter D_t.

Energy balance of water storage tank control volume: From the above diagram of Fig. 3, the energy balance for this control volume is:

$$\dot{Q}_{awt} + \dot{m}_{w2}cT_{w4} + \dot{m}_{w1}cT_{w2}\ \dot{m}_{w2}cT_{w3} \dot{m}_{w1}cT_{w1} = H \quad (18)$$

Substituting the value of the heat transfer rate between the water in the tank and the ambient air, Q_{awt}, using Eq. (15) and (18) now becomes:

$$(T_{amb} - T_{wt})/R_{awt} + \dot{m}_{w1c}\ (T_{w1} - T_{w2}) + \dot{m}_{w2c}\ (T_{w4} - T_{w3}) = H \quad (19)$$

Recall from Eq. (11);

$$H = mc\,(\delta T/\delta t) = (m_t c_w/\Delta t)\,(T^{t+1} - T^t)$$

Hence,

$$(T_{amb} - T_{wt})/R_{awt} + \dot{m}_{w1}c\,(T_{w1} - T_{w2}) + \dot{m}_{w2}c\,(T_{w4} - T_{w3}) = (m_t c_w/\Delta t)\,(T^{t+1} - T^t) \quad (20)$$

Simplifying further, we obtain Eq. (21) and (22):

$$T^{t+1} - T^t = (\Delta t/m_t c_w)\{(T_{amb} - T_{wt})/R_{awt} + \dot{m}_{w1}c\,(T_{w1} - T_{w2}) + \dot{m}_{w2}c\,(T_{w4} - T_{w3})\} \quad (21)$$

$$T^{t+1} = T^t + (\Delta t/m_t c_w)\{(T_{amb} - T_{wt})/R_{awt} + \dot{m}_{w1}c\,(T_{w1} - T_{w2}) + \dot{m}_{w2}c\,(T_{w4} - T_{w3})\} \quad (22)$$

Equation (22) represents the new temperature of the water in a finite time Δt:

Conversely,

$$T^{new} = T^{old} + (\Delta_t/m_t c_w)[\dot{m}_{w1}c\,(T_{w1} - T_{w2}) + \dot{m}_{w2}c(T_{w4} - T_{w3}) + \dot{Q}_{awt}] \quad (23)$$

Room convector heat exchanger: Heat is transferred from the air in the room to the cold water circulating in the convector. The convector is made of copper tubes with aluminium fins. The fin spacing is large enough to allow for natural convection. In analyzing the room convector, it is assumed that the temperature of the water circulating through the convector is approximately equal to the temperature of the room. This is as the convector pump is turned on in the morning hours (about 8 am) to use the heat lost by the room convector through cool water from the water tank to cool the room; before it is turned off in the evening hours (about 6 pm). The room convector is assumed to operate at steady state; its function is mainly to balance heat exchange between the room and the storage tank.

Other assumptions in analyzing the room convector include (Rajput, 2005):

- No heat generation within the convector fins
- Uniform heat transfer coefficient over the entire surface of the fin
- Homogenous and isotropic fin material of aluminium with constant thermal conductivity
- Negligible contact thermal resistance
- One dimensional heat conduction and negligible radiation

Energy balance for the room convector control volume: Energy balance for the convector is:

$$\dot{Q}_{HE} \; \dot{m}_{w2} c \, (T_{w4} - T_{w3}) = 0 \tag{24}$$

The resultant energy present in the convector is energy due to the heat transfer from the air in the room to the water convector and energy due to water flowing from the cold tank to the room, while the heat transferred from the air in the room to the water in the convector, Q_{HE} is evaluated using Eq. (15).

Hence, the steady state energy balance for the convector becomes:

$$0 = (T_{rm} - T_{cv})/R_{rmcv} - \dot{m}_{w2} c (T_{w4} - T_{w3}) \tag{25}$$

Room options: To design the room space for thermal comfort, the room walls will be insulated, a mechanism will be incorporated to cool the air inside the room by extracting heat from the room space and the interior wall surfaces.

Energy balance for the room options control volume: Room heat sources include:

- Internal heat source, Q_{ixn}
- Infiltration of hot outside ambient air, Q_{infil}
- Heat transmission from the outside ambient air into the room, Q_{trans}

Recall that energy balance relationship from Eq. (7) is:

$$Q_{in} + Q_{ixn} = Q_{loss} + H$$

Similarly, recall that change in internal energy, H from Eq. (11) is:

$$H = mc\,(\delta T/\delta t) = (m_t c_w/\Delta t)\,(T^{t+1} - T^t)$$

From the control volume above:

$$\dot{Q}_{in} = \dot{Q}_{infil} + \dot{Q}_{trans} \tag{26}$$

$$\dot{Q}_{loss} = \dot{Q}_{HE} \tag{27}$$

The energy balance equation for the room option is therefore:

$$\dot{Q}_{infil} + \dot{Q}_{trans} + \dot{Q}_{ixn} - \dot{Q}_{HE} = (m_t c_a/\Delta t)\,(T^{t+1} - T^t) \tag{28}$$

The infiltration load and transmission load are evaluated using Eq. (15) while the internal heat generation load, Q_{ixn}, is due to the number of occupants, electrical appliances, lights, etc in the room.

The overall energy balance equation therefore becomes:

$$c_{infil}\,(T_{amb} - T_{rm}) + (T_{amb} - T_{rm})/R_{arm} + \dot{Q}_{ixn} - (T_{rm} - T_{cv})/R_{rmcv} = (m_t c_w/\Delta t)\,(T^{t+1} - T^t) \tag{29}$$

Simplifying further:

$$T^{t+1} = T^t + (\Delta t/m_t c_a)\,[c_{infil}\,(T_{amb} - T_{rm}) + (T_{amb} - T_{rm})/R_{arm} \; \dot{Q}_{ixn} - (T_{rm} - T_{cv})/R_{rmcv}] \tag{30}$$

Equation (30) is the functional relationship for the cooled room space.

RESULTS AND DISCUSSION

Test runs were carried out for four different weather seasons prevalent in Nigeria-early (March/April) and late (July/September) rainy seasons; and early (November/December) and late (January/February) harmattan seasons. The input variables include the ambient weather conditions for the various seasons as shown in Fig. 5, the solar irradiation as shown in Fig. 6,

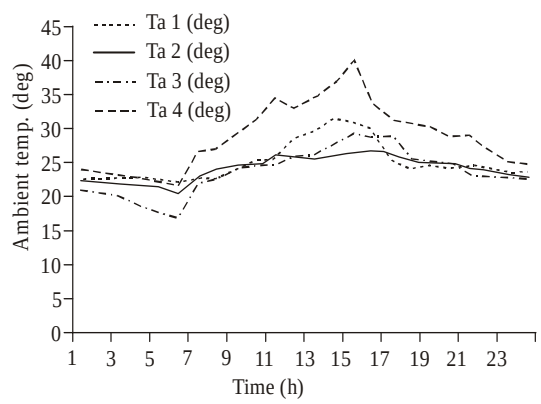

Fig. 5: Average input ambient temperature values (Ta1: Early rainy season (March/April, Ta2: Late rainy season; Ta3: Early harmattan season (Nov/Dec, Ta4: For Late harmattan season (Jan/Feb)

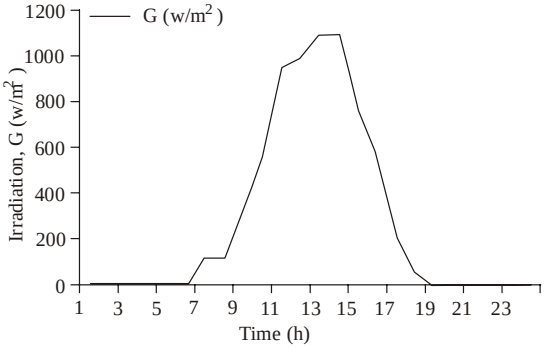

Fig. 6: Average irradiation input values for Owerri

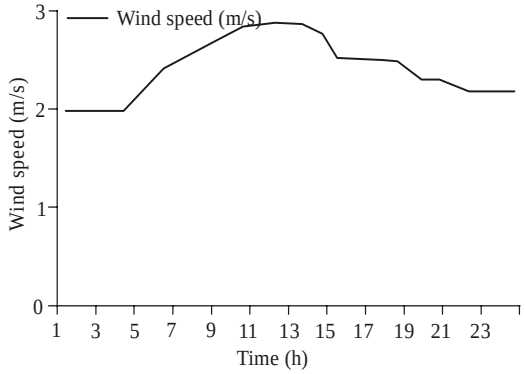

Fig. 7: Average wind speeds for Owerri

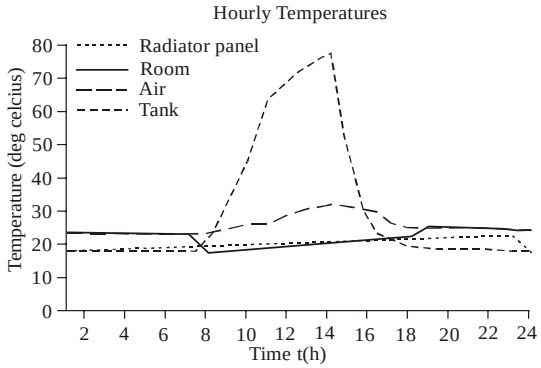

Fig. 8: Plots of temperatures of radiator panel, cooled roomspace, ambient air and storage tank against hourly time interval for early rainy season

and the wind speeds as shown in Fig. 7. According to Anyanwu and Iwuagwu (1995), the average wind speed for Owerri is 2.80 m/s ±0.81; annual mean wind power density is 12.91 ± 0.26 W/m^2.

In developing the simulation program, actual ambient values are used to achieve a true temperature profile.

MATLAB outputs and discussion: Figure 8 to 11 are results from the test run using MATLAB programming language. The results cover the seasons prevalent in Nigeria.

Figure 8 to 11 are different temperature plots for different ambient temperature conditions representing early rainy season, late rainy season, early harmattan season and late harmattan season, respectively.

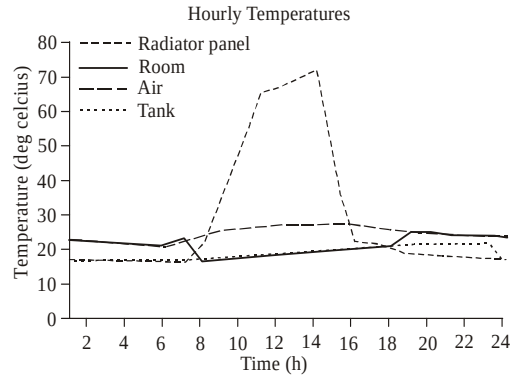

Fig. 9: Plots of temperatures of radiator panel, cooled roomspace, ambient air and storage tank against hourly time intervals for late rainy season

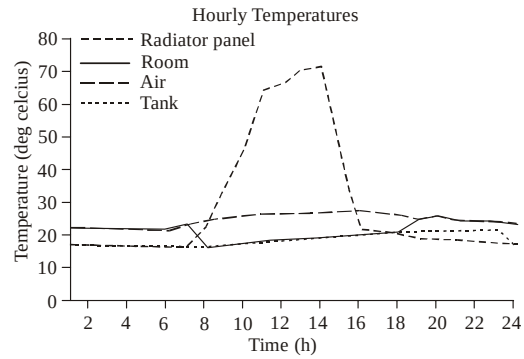

Fig. 10: Plots of temperatures of radiator panel, cooled roomspace, ambient air and storage water tank against hourly time intervals for early harmattan season

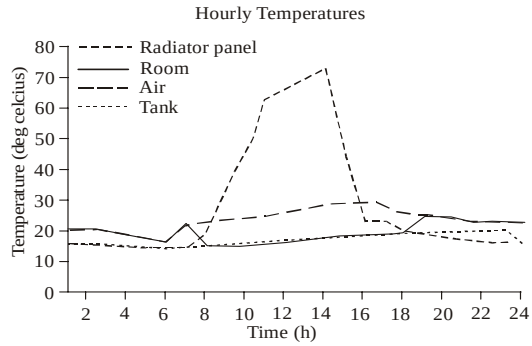

Fig. 11: Plots of temperatures of radiator panel, cooled roomspace, ambient air and storage water tank against hourly time intervals for late harmattan season

To generate these profiles, the radiator circulation pump was turned on at 23 h (11 pm), to utilize the heat lost by the radiator panel to cool the water in the storage tank and then switched off by 7 h (7 am) the next day. Then the room convector circulation pump was turned on by 8 h (8 am), to utilize the heat lost by the room convector to cool water from the water tank, so as to cool the room. The convector pump is switched off at 18 h (6 pm).

From the figures, the room temperature, which from the night was programmed to 'mirror' the ambient temperature, begins to drop significantly from around 6 am, achieving the lowest daytime cooling at about 8 am when the room convector is switched on. This is expected as the room convector extracts any heat within the space, given that the ambient temperature is only gradually increasing. As the ambient temperature continues to rise during the day, the room temperature experiences a steady increment but at a rate that is insignificant due to the building configuration and the action of the room convector. This trend continues till towards evening (about 5 pm) when the room temperature has built up nearly equaling the ambient temperature which is steadily decreasing towards the evening into the night.

Another reason for the rise in room temperature towards evening is the build-up of heat in the water storage tank due to repeated extraction of heat through the action of the room convector. Even though the storage tank is properly insulated, the water flowing through the convector is getting 'hotter' after each successive pumping through the room convector loop. The ability of the water to further extract heat from the cooled space is steadily decreasing as the water temperature is gradually rising. At this point, the convector pump is turned off and the radiator pump is switched on to expel the accumulated heat in the water storage tank. More so, the radiator temperature which was hitherto high, has now decreased significantly even below the ambient temperature. The relatively hot water in the storage tank is used to defrost the radiator panel surface by passing it through the fins of the radiator at night while still maintaining thermal comfort within the space. This continues until the morning, dropping the temperature of the water in the storage tank to about the same as that of the radiator panel surface. This equilibrium is important both to avoid freezing of the panel surface, considering the dew point temperature and to reduce the water temperature to the possible minimum in readiness for heat extraction during the day. Cold water is needed in the storage tank to contain possible heat generation within the cooled space during the day.

From Fig. 8, temperature depression of as much as 12ºC is achieved. This is a good result as the early rainy season is still considerably hot. The ambient temperature is still significantly high. From Fig. 9, the temperature depression is at a maximum of about 7ºC. This is the late rainy season characterized by reduced ambient temperatures. Room temperature is as low as 16ºC very early in the morning.

From Fig. 10, the temperature depression gets narrower. The ambient temperature is now significantly low. The room architecture is still capable of maintaining depressions of about 10ºC. The harmattan is set and the season is generally cold. From Fig. 11, the ambient temperature has risen tremendously and a peak temperature depression of about 18ºC is achieved. However, the room temperature is considerably comfortable during most of the day. This period is characterized by a very hot weather and this accounts for the reduced performance of the nocturnal cooling system.

Using these models to predict nocturnal cooling during the day, room temperatures of between 17ºC minimum and 22ºC maximum are achieved. This temperature range is thermally comfortable for human habitation.

Daytime cooling, which is the major objective of nocturnal cooling systems, is significantly achieved as shown in Fig. 8 to 11. During the day, between 7 am-6 pm, the maximum room temperature is below 25ºC. Considering the human body temperature average of about 37ºC, a temperature of 25ºC for a conditioned space is capable of providing significant cooling of up to 12ºC depression to the human body. This is obviously thermally comfortable. Considering other factors affecting thermal comfort such as the mean radiant temperature of the surroundings, the plots show that during the day, the room temperature is far less than the surrounding air temperature, i.e., $T_r << T_a$. This is another significant factor in thermal comfort of occupants of a cooled space.

Sensitivity analysis: In performing a sensitivity analysis, a ±25% of the system variables including the radiator panel surface, storage tank volume, room wall thickness, convector fin surface area and the mass flow rate analyses were conducted. Figure 12 and 13 shows the resultant plots for the observed variations in radiator temperature and tank temperature.

The results of this sensitivity analysis are given in Table 1 as a percentage variation of a number of the system performance parameters which include the sum of the difference between the outside ambient air temperature and the inside room air temperature multiplied by the time step for each of the hours of the nine hour period (8 am-5 pm) that the room circulation pump was switched on, i.e., $\sum(T_{amb}-T_{rm})\Delta t$. Others are the total amount of heat gained by the storage water during

Table 1: Sensitivity analysis of performance parameters with a variation of ±25% from base case input values

Variable	$Q_{wt,in}$ (MJ)	$Q_{wt,out}$ (MJ)	$\sum(T_{amb}-T_{rm})\Delta t$ [°C.hour]
Base case	80.2	79.5	38.07
0.75 A_p	77.5	70.5	30.52
1.25 A_p	84.6	87.2	44.87
0.75 V_{wt}	78.4	68.2	36.8
1.25 V_{wt}	81.9	88.6	38.09
0.75 L_{wt}	80.3	79.7	29.6
1.25 L_{wt}	77.8	79.6	41.6
0.75 A_{co}	71.9	76.4	1.26
1.25 A_{co}	84.7	79.9	46.3
0.75 mw	80.1	78.8	36.7
1.25 mw	80.2	79.3	37.5

Table 2: T_{amb}-T_{rm} values for the base case and ±25% of the base case (9 am-5 pm)

Time (h)	Tamb−-T_{rm} base case	T_{amb}− T_{rm} for −25 (%)	T_{amb}− T_{rm} for +25 (%)
9 am	2.56	3.57	1.89
10 am	2.75	3.8	2.05
11 am	2.93	4.02	2.21
12 noon	3.12	4.24	2.37
1 pm	3.3	4.46	2.52
2 pm	3.48	4.68	2.67
3 pm	3.66	4.9	2.83
4 pm	3.83	5.11	2.98
5 pm	4.01	5.33	3.13

the room cooling operating period, $Q_{wt,in}$ and the heat removed during the night sky cooling period $Q_{wt,out}$,

For an 8 h period (11 pm-7 am), the radiator panel surface area was able to remove 79.5 MJ from the water storage tank as shown in the base case values. This corresponds to an average heat removal rate of 57.6 W/m^2. This heat removal rate compares favourably with the value of 80 W/m^2 as reported by Erell and Etzion (1999) and the value of 60.8 W/m^2 as reported by Dobson (2005).

The variations due to the parametric analyses show that the models are stable as an analysis tool. Variations of ±25% of the chosen variables showed significant changes in the values of the performance parameters. Similarly, the heat gained from the room space through the action of the room convector into the storage tank during the day (8 am-6 pm) shows a value of 80.2 MJ. This value is slightly more than the value of the heat lost to the ambient from the system. This is practically correct as the temperature of the water in the storage tank is never zero; there is always some amount of sensible heat in the water within the storage tank.

The values of $\sum (T_{amb}-T_{rm})\Delta t$ from the Table is indicative of the functionability of the modeled system. These values are comparable to values from Dobson (2005). The depression between the values of the ambient air and the room is critical in determining the cooling power available to a system; hence the values are useful in analyzing a nocturnal system. Table 2 shows the

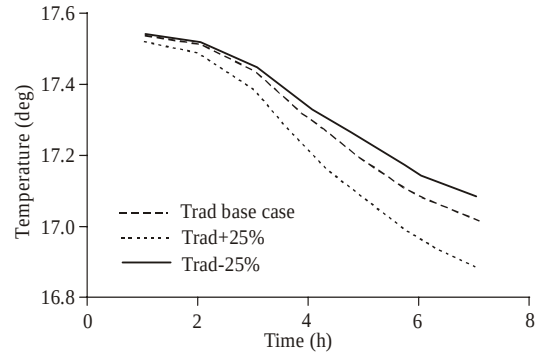

Fig. 12: Plots showing the values of the radiator panel temperature for the base case inputs and ±25% of the base case inputs

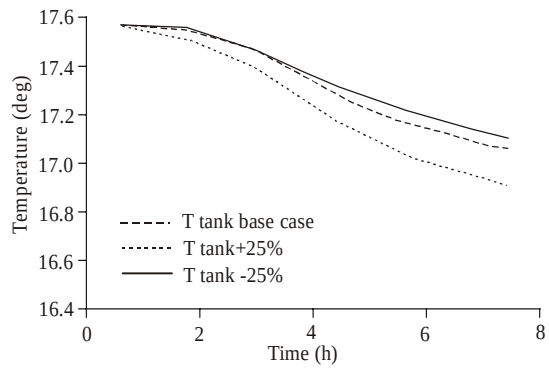

Fig. 13: Plots showing the values of the water storage tank temperature for the base case inputs and ±25% of the base case inputs

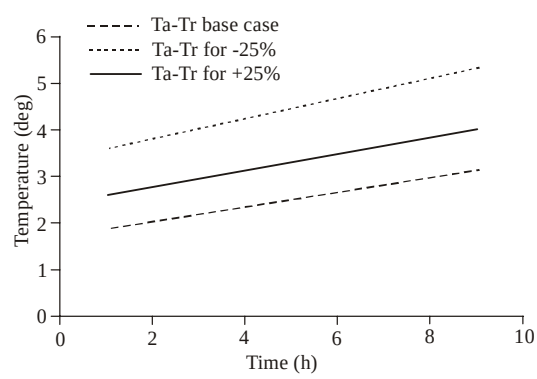

Fig. 14: Plots of T_{amb}-T_{rm} for base case and ±25% of the base case values

difference between the ambient air temperature and the room temperature against time between 8 am-5 pm for the base case, and ±25% of the base case and plotted in Fig. 14.

For the base case situation, the maximum temperature difference is 4.01, which is comparable to values in literature. This depression is indicative that nocturnal

Table 3: Temperature depression (T_{amb}-T_{rm}) comparison for different locations

Temperature depression (T_{amb}-T_{rm})	Country	Author
5ºC	Norway	Meir *et al*. (2003)
3ºC	Thailand	Ito and Miura (1989)
5ºC	Namibia	Dobson (2005)
4.01ºC	Nigeria	Present study

Table 4: Optimal values of design parameters

Parameters	Values
Radiator surface area (m^2)	18.20
Convector area (m^2)	28.62
Tank volume (m^3)	1.570
Room volume (m^3)	22.50
Surface area of room (m^2)	48.00

cooling is possible using this model. A very large value of T_{amb}-T_{rm} is indicative of impossibility of nocturnal cooling. This is because it implies that much heat is added to the cooled space during the day and it could be due to poor insulation. This difference contributes the heat added to the room during the day and extracted to the water reservoir through the room convector. It is also indicative of the possible amount of heat to be removed from the cooled space at night.

During the nocturnal hours when cooling is added to the room space (9 am-7 am), the temperature depression of (T_{amb}-T_{rm}) has a maximum value of -0.69. This very low temperature value makes it possible for heat to flow from the water reservoir, now at a higher temperature, to the radiator panel which is obviously at a lower temperature. Also it is important to note that the negative direction is an added impetus to cooling at night, as some heat is also further lost due to this temperature depression. This can be viewed as natural cooling of the room space.

Similarly, a comparison of the temperature depression, as shown in Table 3, obtained in this study to the ones in literature show that the model is a robust tool for the prediction of night cooling.

Following the sensitivity analysis, the optimal design parameters for the modeled 3.0×3.0×2.5 room is as given in Table 4:

CONCLUSION

The mathematical model of a nocturnal cooling system using an integrated system comprising a radiator panel, water storage tank, room convectors and a cooled space was developed from first principle. The model was developed for a transient system. The resulting equations were transformed into the numerical format using the finite difference scheme and subsequently used to carry out a transient analysis and performance evaluation of the nocturnal system using a computer program written in Matlab 7.0. The numerical solution was validated with data from the actual field performances of a similar system tested at a site in the Federal University of Technology, Owerri (FUTO). The predicted temperature values showed agreement with the measured values. The mean deviations between the predicted and measured temperature values varied over the ranges of 3.67-5.02ºC for the radiator temperature and 0.53-2.14ºC for the room temperature, while that between the predicted and measured water tank temperatures varied over the range of 3.06-4.68ºC.

On the heat removal rate, the radiator panel surface area was able to remove 79.5 MJ from the water storage tank which represents an average heat removal rate of 57.6 W/m^2 over an 8 h period of radiator circulation pump operation (11 pm-7 am). This is very comparable to other works such as Erell and Etzion (1999) with heat removal rate of 80W/m^2 and Dobson (2005) with heat removal rate of 60.8W/m^2.

From the above and in line with the objectives of this study, it is summarized as follows:

- A thorough study on the principle of nocturnal cooling for the purposes of cooling room spaces was carried out.
- Mathematical models governing the relationship of the various variables involved in the integrated system were developed in the transient phase.
- A computer simulation program in Matlab 7.0 was developed to solve the objective functions of the subsystems involved in the setup.
- The available cooling due to the design was evaluated and a heat removal rate of 57.6 W/m^2 was achieved.
- A sensitivity analysis carried out to within ±25% showed a consistency in heat removal rate with a deviation of 0.54 representing 0.94% deviation. This further validates the model.
- An optimal scheme for commercialization showed the following component sizes: a radiator area of 18.2 m^2, a convector area of 28.62 m^2 and a tank volume of 1.57 m^3.

It is therefore concluded that the thermal model presented in this study can be used with confidence as a design tool for the sizing of a cooling system using night sky radiation.

The overall thermal management strategies adopted in order to keep the cooled space to within the acceptable limits were as follows:

- The room was insulated properly from heat ingress due to environmental conditions.
- Internally generated heat was removed from the cooled room space during the day by allowing the heated air to flow over a finned room convector heat

exchanger through which cold water from the storage tank is circulated.

- Heat extracted into the storage tank during the day is expelled during the night by passing the heated water through the finned heat exchanger of the radiator panel to the ambient. This prevents the radiator from forming frost on the surface and removes heat within the system to achieve desired cooling during the day.

NOMENCLATURE

A = Area, m^2
c = Specific heat, J/kgK
D = Diameter, m
G = Solar radiation, W/m^2
g = Gravitational constant, 9.81 m/s^2
h = Heat transfer coefficient, W/m^2K
k = Thermal conductivity, W/mK
L = Length, m
m = Mass, kg
$\dot{m}$ = Mass flow rate, kg/s
$\dot{Q}$ = Heat transfer rate, W
R = Thermal resistance, K/W
t = Time, s
T = Temperature, ˚C
V = Volume, m^3
v = Velocity, m/s

Greek symbols:

α = Absorptivity
β = Coefficient of thermal expansion
Δ = Difference
ϵ = Emissivity
θ = Angle, °
ρ = Density, reflectivity
μ = Viscosity, m/kgs
σ = Stefan-Boltzmann constant = $5.67 \times 10^{-8} W/m^2K^4$
τ = Transmissivity
η = Efficiency

Subscripts:

amb = Air = ambient
cv = Convector
infil = Room air infiltration
rm = Cooled room space
p = Radiator panel surface
sk = Sky
solar = Solar
wt = Water storage tank
trans = Wall heat transmission

REFERENCES

Ali, A.H.H., I.M.S. Taha and I.M. Ismail, 1996. Cooling of water through a night sky radiator. Solar Energ., Elsevier Science Ltd., USA, 55(4): 235-253.

Armenta-Deu, C., T. Doaire and J. Hernando, 2003. Thermal analysis of a prototype to determine radiative cooling thermal balance. Renew. Energ., 28: 1105-1120.

Anyanwu, E.E. and C.J. Iwuagwu, 1995. Wind characteristics and energy potentials for Owerri, Nigeria. Renew. Energ., Elsevier Science Ltd., Great Britain, 6(2): 125-128.

Dobson, K.A., G. Hodes and Y. Mastai, 2003. Thin semi-conductor films for radiative cooling applications. Sol. Energ. Mat. Sol. C., 80: 283-296.

Dobson, R.T., 2005. Thermal modelling of a night sky radiation cooling system. J. Energ. S. Afr., 16(2).

Erell, E. and Y. Etzion, 1992. A radiative cooling system using water as heat transfer medium. Arch. Sci. Rev., 35: 35-49.

Erell, E. and Y. Etzion, 1999. Analysis and experimental verification of an improved cooling radiator. Renew. Energ., 16: 700-703.

Etzion, Y. and E. Erell, 1991. Thermal storage mass in radiative cooling systems. Build. Environ., 26(4): 389-394.

Hardy, M., 2008. A Practical Guide to Free Cooling, Alternative Cooling, Night Cooling and Low Energy systems for Air Conditioning Systems. Ambthair services Ltd. Retrieved from: http//www.ambthair.com.

Ito, S. and Miura, N. 1989. 'Studies of radiative cooling systems for storing thermal energy'. J. Solar Energ., 111, 251.

Meir, M.G., J.B. Rekstad and O.M. Lovvik, 2003. A study of a polymer based radiative cooling system. Solar Energ., 73(6): 403-417.

Rajput, R.K., 2005. Heat and Mass Transfer in SI Units. Published by S. Chand and Company Ltd., New Delhi, India.