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Evaluating coefficient of performance and rate of moisture loss of some biomass humidifiers materials with a developed simple direct stand-alone evaporative cooling system for farmers

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ABSTRACT

The purpose of this study was to investigate the feasibility of deploying a direct evaporative cooler with Jute fibre, palm fruit mesocarp fibre and wooden charcoal as a humidifier in Nigeria. The high cost of imported evaporative coolers and humidifiers can discourage farmers and limit the adoption of evaporative cooling technology. For this purpose, an experimental direct evaporative cooling test rig was developed and assembled in southwestern Nigeria with a biomass humidifier. The evaluation parameters were the humidifying efficiency, the rate of moisture loss, coefficient of performance and sensible heat ratio. The humidifier presented average cooling efficiency of 55.9 to 78.62%, an average rate of moisture loss of 1.37×10^{-3} to 2.61×10^{-3} kg/s, an average COP of 8.48 to 23.42 (EER of 11 to 78) and an average sensible heat ratio of 1.28×10^{-4} to 4.06×10^{-4} for the air velocity of 3.0 to 4.5 m/s. The obtained performance can be better in a dryer month or nearly impossible in a very wet month as the humidifying efficiency of direct evaporative coolers is found to diminish at high humidity. To avoid casting doubt on the effectiveness of direct evaporative coolers by farmers, they should only serve as stop-gap preservation equipment and be deployed during favourable weather conditions especially during the winter when the air is dry. These can be identified from the metrological charts of the chosen location.

1. Introduction

The ever-increasing global energy demand with the consciousness of the negative environmental impact caused by high carbon emissions has aroused global interest in cheap, alternative and renewable energy systems [35]. There is a global need for technologies that will substantially decarbonize the ecosystem [31]. In 2019 the estimated global carbon emission rose by 20% from the estimated value of 36.1 Gt. Per year in 2013 to 43.2 GT. Per year [33]. According to the authors, this value is estimated to increase by 2200% by 2040. In the year 2000 alone, about 40–70% of global energy utilization was expanded in cooling systems for air conditioning or storage [12,19]. In some regions like the Gulf States, cooling makes up tom 50- 70% of electrical energy consumption [8,13]. Therefore, this makes compression-based cooling systems that include, air-conditioners and refrigerators among the major sources of CO_2 emissions. Apart from high energy cost and CO_2 emissions, when mechanical vapour compression refrigerators are used to preserve fruits and vegetables, it might lead to chilling injury that harms the stored fruits and vegetables [4,15,16,21]. Therefore, the use of zero energy system (evaporative coolers) that is affordable and renewable has been advocated to reduce energy cost and protect the environment. Additionally, the deployment of direct or indirect evaporative cooling has been reported to save 40 to 50% of energy costs in crop preservation [16]. Thus researchers have advocated the use of evaporative coolers for cooling or a hybrid cooling systems combination where mechanical vapour compression refrigerators or air conditioners can be combined with evaporative coolers [2,8,9,31].

Evaporative coolers can lower the ambient temperature up to 10 °C [3]. To extend the shelf life of fruits or vegetables, these coolers humidify the product environment (up to 90% humidity) and at the same time lower the temperature [3]. They are most useful to subsistence farmers and small-scale producers or fruit sellers who need to keep their products cool and humid to maintain their freshness [10]. The reason is that what most small farm holders need especially in the case of fruits and vegetable farming is a technology that can extend the freshness within a short period for them to sell their products [15]. Thus, they need a

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Nomenclature

A:	area of walls (m^2)
b:	base area (m^2)
C_p:	specific heat capacity (J/kg k)
E_c:	energy input
h:	height (m)
H_t:	humidifier thickness (m)
H_A:	humidifier area (m^2)
K:	thermal conductivity (W/ mK)
L:	length(m) or latent heat of vaporization (J/kg K)
m_H:	mass (kg) of the biomass humidifier
m_w:	mass (kg) of water
Mp:	mass of product (kg)
P_{atm}:	atmospheric pressure (Pa)
P_v:	vapour pressure (Pa)
Q_{air}:	volumetric air flow rate (m^3/s)
Q_v:	respiration heat (W)
$Q_{v/hr}$:	respiration heat per hour (W/hr)
$Q_{f/hr}$:	field heat per hour (W/hr)
Q_f:	field heat (W)
Q_c:	heat gained by conduction (W)
Qc/hr:	heat gained by conduction per hour (W/hr)
Qs:	service heat (W)
Qs/hr:	service heat per hour (W/h)
Rh:	relative humidity (%)
THL:	total sensible heat load (KW or kJ/s)
T:	temperature (°C)
v:	face velocity (m/s)
V:	volume of the cooler (m^3)
x:	thickness (m)

Greek letters

θ:	angle of the triangle (°)
$\dot{m}_a$:	mass flow rate of air
ω:	specific humidity of the air
ϵ:	gas constant ratio between wet and dry air
ρ:	density (kg/m^3)

Subscripts

o:	outer wall or outlet
in:	insulator wall
i:	inner wall or inlet
db:	dry bulb
c:	cooler
w:	wet bulb
a:	air

simple and cheap evaporative cooler for short-term fruit and vegetable preservation [16]. Furthermore, achieving low-cost cooling in the face of rising energy costs is a motivation for researchers to seek a viable alternative.

Evaporative coolers come in different types which include, direct, indirect, dew point types etc [6,17,22]. However, the complexity of designs adds to the cost of evaporative coolers and might require extra training for individuals to use them [4]. This is a limiting factor in using evaporative coolers, especially in food preservation in developing countries with low literacy and income level [22]. As a consequence, a lot of farmers cannot benefit from this environmentally friendly cooling technology. Sometimes it is not economically viable to deploy a sophisticated evaporative cooling system at a higher cost whereas a simple direct evaporative cooler can do the job at a lower cost input. However, research in evaporative cooling technology is limited and most evaporative cooler research available focused more on using them for cooling homes [29].

The major component of an evaporative cooler is the humidifier which comes in different commercially available types like Celdek, Aspen, Kraft paper, porous ceramics, cotton fabric etc [22,28,36]. These exotic pads add to the overall cost of evaporative coolers putting the cost beyond the average purchasing power of most homes in Africa and other developing countries [21]. Additionally, they are not biodegradable since the pads are required to be changed often. Therefore researchers have found biomass materials with good wetting ability as an alternative [15]. The use of biomass waste will solve the disposal issue, reduce cost and make the system more renewable. One of the challenges of evaporative cooling is that the various designs in most cases are environmental specific [37]. The performance of the coolers is a function of the variable external environment, thus the design must adapt to the various environment [2]. Sub-Saharan Africa needs a simple, cheap and effective evaporative cooler they can afford. These coolers can be used during favourable conditions to reduce dependence on compression refrigeration within certain periods of the year [22]. Though the southern part of a country like Nigeria is mostly wet the weather is partly dry from November to March which is the seasonal harvesting period of most crops. These months provide a climate in which direct evaporative cooling can be used [1,20]. Active evaporative cooling systems are now feasible in some part less windy, hot and humid parts of the world. This research, therefore, is justified in evaluating the suitability of this technology in southern Nigeria with different biomass humidifiers. We, therefore, present a very simple, affordable direct evaporative cooler using biomass humidifier as a cost-effective humidifier for short-term cooling and shelve life extension of fruits and vegetables

2. Design of the evaporative cooling systems

The stand-alone direct evaporative cooler was designed using the climatic condition of Akure, South-West Nigeria (7°10′N and 5°05′E). The climatic data were deduced from the metrological data from the Nigerian institute of metrology (NIMET). Climatic data shows that direct evaporative cooling can be applied in this area around September to March period when the ambient humidity is low.

2.1. Design features of the evaporative cooling system

The evaporative cooler will consist of the following:

1 Double-walled hexagonal cooler housing lagged in the middle with fibreglass. The outer wall was a mild steel plate, while the inner wall was aluminium to prevent corrosion since it will likely encounter moisture.
2 Plastic (thermoplastic) top and bottom water tanks were used. Plastics are corrosion-resistant and light. They are available and relatively cheap.
3 The humidifier holder was constructed with galvanized steel material to prevent corrosion.
4 Palm fruit mesocarp fibre (density 130 kg/m^3; Water holding capacity 2.05 kg of water/kg of solid), wood charcoal (bulk density 267 kg/m^3; Water holding capacity 0.5 kg of water/kg of solid) and jute fibre (bulk density 89.8 kg/m^3; Water holding capacity 2.1 kg of water/kg of solid) was selected. This is because they can be locally sourced as waste material, therefore relatively cheap. Also, they have a good water holding capacity with enough pore spaces for air passage [1,5,11].
5 Axial fans were used because it is relatively cheap. Sometimes lower in profile. They are less affected by problems in the installation.
6 A sump water pump was used for the recirculation of water during the operation of the cooler. This was chosen because the pump will be located below the water line and will not be submerged in water. They are cheaper than submersible water pumps
7 PVC rubber flexible hose was used. This was selected due to its lightweight and cannot corrode. In addition, it can be easily bent to take the desired shape.

8 A water distribution header was used to maintain a steady water flow rate.
9 Water flow float and control valves were used to control water flow through the pad.
10 Electric power was used since the cooler fan and pump requires electricity, though at comparative low energy than refrigeration.

2.2. Products to be stored

For the fruit and vegetable, the equipment is designed for its storage including but not limited to tomato, cut pumpkin leave(*Telferia Occidentalis*), Amaranths, orange, water leave (*Talinum triangulare)* and fairly pawpaw (*Carica papaya*). The above crops were selected because of their availability all year.

2.3. Storage capacity

The shape of the cooler is assumed regular hexagonal to present a larger projected surface for air circulation [18]. The cooler capacity (m^3) is calculated with a simple geometrical equation given in Eq. (1) as follows

$$V = h \times b. \tag{1}$$

The base area of the regular hexagon is divided into six equilateral triangles. The Base area is the area of a triangle multiplied by 6

$$b = \left(\frac{1}{2} \times L^2 \times \sin\theta\right) \times 6 . \tag{2}$$

Substituting: $h = 1.0$ m, $L = 0.3$ m, into Eq. (2) with $\theta = 60\ ^0$, $b = 0.24$ m^2, $V = 0.24$ m^3

2.4. Sources of heat load into the cooler to be removed

The following are the sources of heat load into the evaporative cooler which need to be removed.

2.4.1. Heat of conduction

This is the heat entering through the walls of the evaporative cooler.

2.4.2. Field heat of the produce

This is the heat accumulated by the fruits to be stored from harvest. This heat is extracted from the fruits and vegetables as it cools to the storage temperature. It is proportional to the mass of the produce and storage temperature.

2.4.3. Heat of respiration

Even at harvest, agricultural products respire. The heat of respiration is the heat generated by the product due to its respiration.

2.4.4. Service load

This is the heat from moist air that can enter through openings or when the door is opened.

2.5. Calculation of heat loads

2.5.1. Field heat of the produce

The field heat of the product per hour is given deduced with Eq. (3) as follows [7]

$$Q_f = M_p C_p \Delta T. \tag{3}$$

Using tomato of 2 kg, $M_P = 2$ kg, $C_p = 4.02$ J/kg°C, $T_s = 17$ °C, $T_b = 38$ °C, $Q_{f/hr} =$ 152.76 W/hr

Taking residence cooling time to be 12 h [32], $Q_f = 1833.12$W

2.5.2. Respiration heat load of produce product

This is the energy by the product as it respires. This heat decreases during the initial cooling of the product and stabilizes after the required storage temperature is achieved. The respiration heat per hour is expressed as Eq. (4) as follows [7]

$$Q_v = M_p P_v. \tag{4}$$

$P_v = 0.130$ W/kghr, $M_p = 2$ kg, $Q_{v/hr} = 0.26$ W/hr. Taking residence cooling time of 12 h, sec [32], $Q_v = 3.12$ W.

2.5.3. Heat gained by conduction

Heat is conducted through the walls, roof and floor. The amount of heat flowing through these surfaces is a function of their thermal resistance (R-value), their area, and the temperature difference between one side and the other. The heat flow through the system can be represented with the heat balance equation given in Eq. (5) as follows [7]:

$$Q_c = \frac{Q_o x_o}{A_o K_o} + \frac{Q_{in} x_{in}}{A_{in} K_{in}} + \frac{Q_i x_i}{A_i K_i}. \tag{5}$$

$A_0 = A_{in} = A_i = 1.98$ m^2, $x_o = 0.001$ m, $x_{in} = 0.01$ m, $x_i = 0.001$ m, $K_i = 250$ W/ mK, $K_{in} = 0.048$ W/mK, $K_o = 35$ W/mK, Qc/hr = 198 W/hr. Taking residence cooling time to be 12 h [32] and 6 sides of the hexagonal shaped cooling chamber, Qc = 2376W

2.5.4. Service heat load (air infiltration heat)

This comprises several miscellaneous items and is called air infiltration. It includes heat given off by equipment. This is taken as 10% of the total sub-heat load (Boyette et al., 2010) and is calculated as Qs/hr = 35.1 W/hr. Taking residence cooling time to be 12 h Qs = 421.2 W

2.5.5. Total heat load (THL)

The total sensible load is given from Eq. (6) as follows

$$THL = Qf + Qv + Qc + Qs \tag{6}$$

THL is calculated as 4630.32W (4.63kW).

2.6. Air requirement

This is the total volume of air (m^3/s) required to remove the total sensible heat load from the cooler. It is given in Eq. (7) as follows [14]:

$$Q_{air} = \frac{(THL)}{\rho \times C_{pa} \Delta T} . \tag{7}$$

Taking 17 °C for the average storage temperature for tomatoes and the extreme temperature of 38 °C.

$Q_{air} = 0.19 m^3/s.$

2.7. Area of the humidifier

The humidifier area was calculated from the Eq. (8) below

$$H_A = \frac{Q_{air}}{\text{Recommended air face velocity}}. \tag{8}$$

Using air face velocity of 0.62 m/s [14]

$Q_{air} = 0.19 m^3/s, H_A = 0.3 m^2$

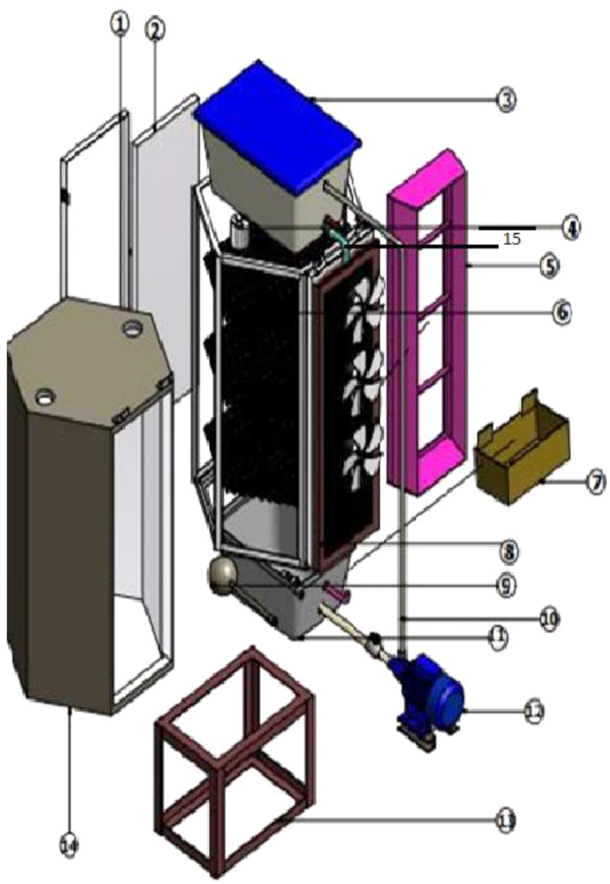
2.7.1. Humidifier thickness

A constant humidifier thickness (H_t) of 30 mm was selected for best performance (Richard et al., 1991).

2.7.2. Humidifier volume

The humidifier volume is given from the Eq.(9) as follows

$$H_v = H_t x H_A$$
$$H_v = 0.009 m^3 \tag{9}$$



COMPONENTS OF THE EVAPORATIVE COOLING SYSTEM		
S/N	DESCRIPTION	QTY
1	DOOR FRAME	1
2	DOOR	1
3	UPPER WATER TANK	1
4	AIR VENT	2
5	AIR CHANNEL	1
6	FRUIT AND VEGETABLE TRAY	1
7	WATER COLLECTOR	1
8	COOLING PAD HOLDER	1
9	WATER CONTROL SWITCH	1
10	WATER RETURN PIPE	1
11	SUMP	1
12	WATER PUMP	1
13	STAND	1
14	COOLER HOUSING	1
15	WATER DISTRIBUTION HEADER	1

Fig. 1. The exploded form of the prototyped developed evaporative cooling system showing the arrangements of the components to form the evaporative cooling system.

2.8. Water circulation

Water is continuously circulated over and through the pad during operation. The minimum water flow rate through a vertically mounted pad of 50 mm thickness; per unit length of the pad is 10 L/min.

2.8.1. Water delivery pipe

Water is delivered from the top tank to the cooling pad through a 20 mm diameter PVC pipe, with about 2.5 mm holes drilled in single rows.

2.9. Water tank

A 20L tank capacity (Th_c) was chosen for “top” and “bottom” tanks, respectively. The height of the water tanks was estimated from Eq. (10) as follows

$$H = \frac{Th_c}{Base\ area} \tag{10}$$

The base area is the same as the cooler

$H = 7.69$ cm; 8 cm selected

2.10. Water recirculation pump

A 1/10 h.p sump pump having a water delivery capacity of 40 L/min with a maximum delivery head of 30 m was used to lift water from the bottom tank to the top tank.

2.11. Exhaust air vent

The exhaust air outlet removes air from the cooler. The airflow re of the exhaust outlet should not exceed 2 m/s [14]. For a face velocity of 1.5 m/s and airflow rate of 0.5 m^3/s. The area for the air outlet can be calculated using Eq. (11).

$$\text{Qair} = \text{Av.} \qquad A = 0.75m^2 \tag{11}$$

2.12. Power requirement

The cooler consists of three small axial air inlet fans of 0.02 kW power each (0.06 kW total) and a sump water pump of power range 0.075–0.37 kW. The total energy consumption is gotten by the addition of all the components.

$$P_{total} = 0.135 to 0.43 kW$$

3. Description of the tested direct evaporative cooler

The exploded schematic picture and isometric view of the evaporative cooler are presented in Figs. 1 and 2, respectively. The cooler is made up of a 0.24 m^3 double jacket hexagonal-shaped aluminium storage chamber mounted on a steel frame 0.3 m high (2) with wire mesh vegetable trays for storing fruits and vegetables. This storage chamber is detachable from the steel frame. The inner walls of the cooler and the outer wall were separated with fibreglass to provide lagging for e system. The outside of the wall is silver-coloured to increase the reflectivity of the material and decrease the rate of absorption of heat. To draw in

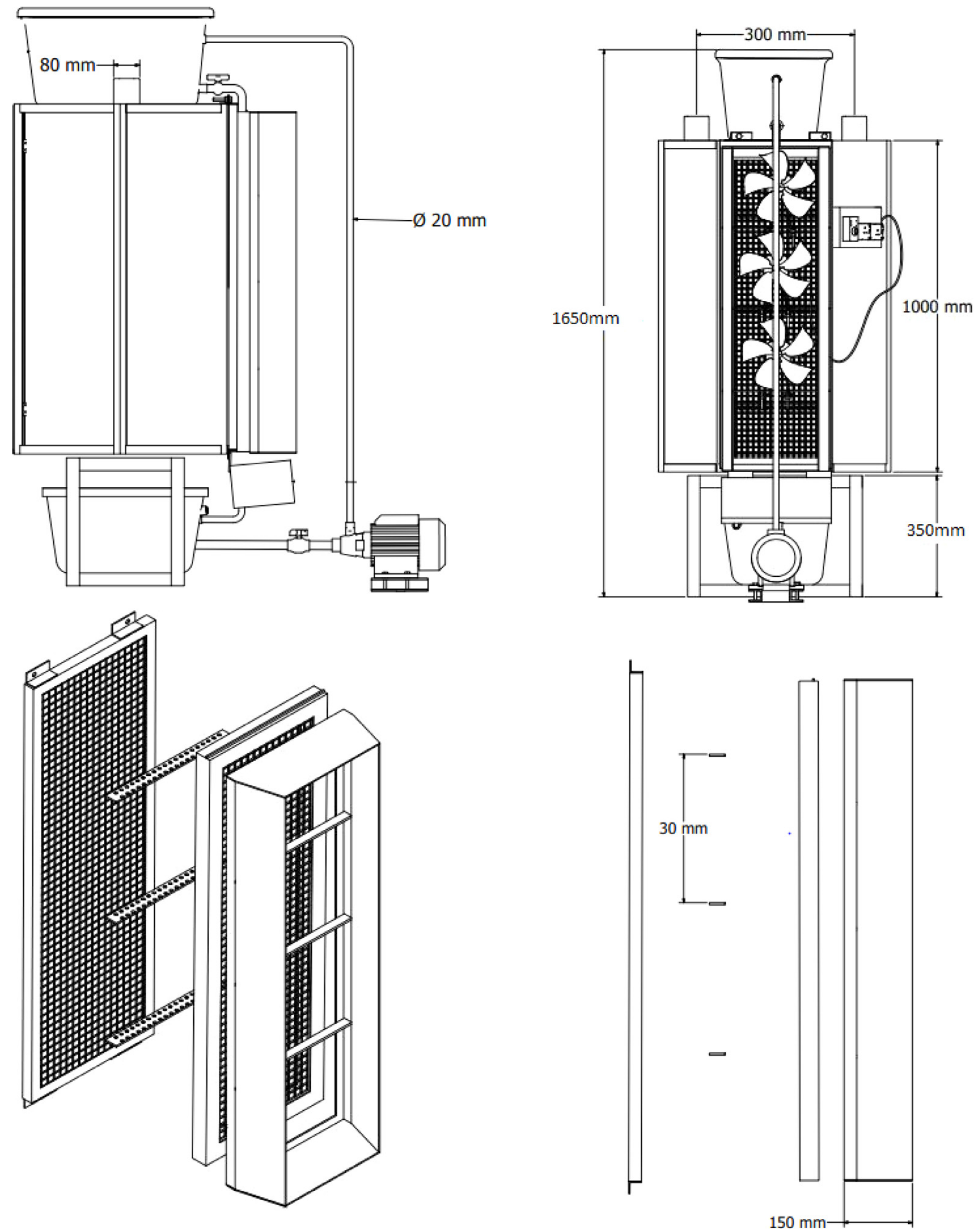


Fig. 2. Isometric view of the developed evaporative cooling system and cooling pad holder showing the front and side elevations.

the air through the humidifier, the storage chamber is equipped with three axial fans (0.02 hp each) mounted vertically as shown in Fig. 2. The design is in such a way that the fans can be mounted behind the humidifier (pad) holder at the inside of the storage chamber or outside depending on the direction of airflow of the fan. Variation of the airspeed (3, 4 and 4 m/s) is through the use of a rheostat connected to the fans. These fan speeds provided a measured air flow rate of about 0.41. 0.086, 0.370.077 and 0.280.081 kg/s respectively. The detachable humidifier holder (0.3 m^2) was mounted on one side of the cooler with bolt and nut assembly. The inner wall of the humidifier holder opens into the cooling chamber equipped with storage trays to place the fruits or vegetables. To provide for moist air escape from the storage chamber which is one of the conditions for evaporative cooling, the conditioned air passes through two vents 8 cm in diameter, located 0.3 m from each other. The vents open to the atmosphere through 1 mm vent holes. Directly at the bottom of this 0.24 m^3 housing is a plastic 20 L water storage tank for storage of water (Fig. 1). A 0.075 kW electric water pump (Fig. 1) lifts the water from the bottom tank through a 2 cm PVC pipe (Fig. 2) to 20 L upper plastic water tank.

The water returns to the humidifier through a 2.5 cm flexible hose equipped with a ball valve to control the rate of water flow. The valve opens into the humidifier through a perforated 2 cm plastic water distribution header lined with pinholes that spray water on the humidifier at 10 cm^3/s. Water sprayed at the top edges of the humidifier is further distributed by gravity and capillarity and drains into a trough under the humidifier. The trough opens into the sump through a 2 cm PVC pipe. The bottom tank is equipped with a floating valve. The humidifier is parked inside the 0.009 m^3 cuboids-shaped humidifier holder (Fig. 3) made of a double-walled galvanized steel net. To ensure ease of maintenance, the two walls were held with two hinges on one side (Fig. 3) to ensure an opening for parking the wetting material. The holder was divided into three equal compartments, separated with a perforated gal-

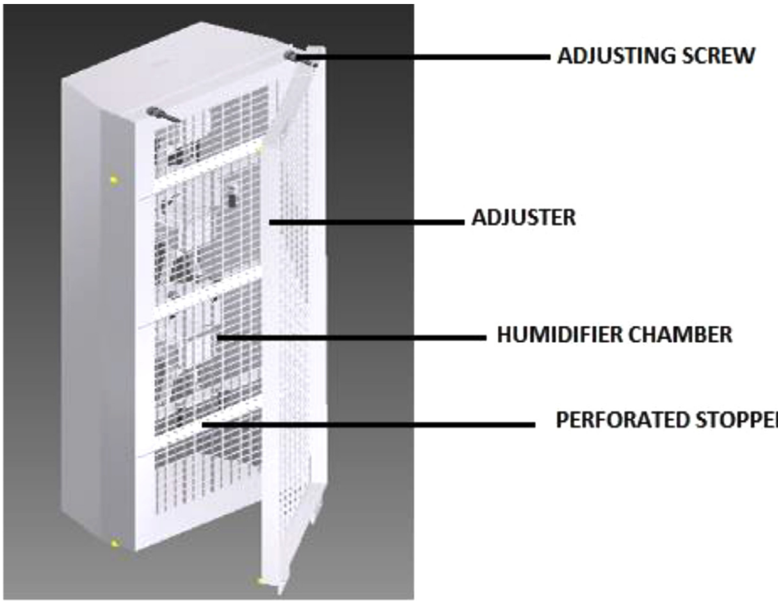


Fig. 3. Inner view of the pad holder compartment showing the arrangement of the stopper.

Table 1
Bill of materials.

Item	Description	Qty	Unit cost (₦) N(((	Total cost(₦)
1	Mild steel	1 sheet	4500	4500
2	Sump Water pump	1	6000	6000
3	Plastic tank	2	500	1000
4	2cm PVC pipe	1 length	250	250
5	Elbow joint	2	30	60
6	Ball valve	2	50	100
7	PVC hose	$\frac{1}{2}$ length	100	100
8	Aluminium sheet	1 sheet	2000	2000
9	Fibreglass	-	4000	4000
10	Suction fan	3	1000	3000
11	Floating valve	1	200	200
12	Angle iron	1 length	1500	1500
13	Mild steel pipe	1/5 length	1000	1000
14	1 gang switch and socket	2pair	250	500
15	Rheostat	1	5500	5500
16	Electric wire	4 yards	100	400
17	Wood shaven	-	-	-
18	Wood charcoal	-	-	-
19	Waste Jute	-	-	-
20	Labour	-		4000
21	Total			53500 ($89)

vanized steel stopper to prevent the sagging of the pad and allow water to pass through. The cost of fabricating the cooler is $89 as shown in Table 1. This cost can go down considerably under mass production.

4. Assembling steps and operation of the evaporative cooler

Some of the components as described in parts or whole in Figs. 1 to 3 can be fabricated in the workshop or purchased in the market. To assemble the components requires minimal knowledge of plumbing and hand tools craft. Fig. 4 can be used as a reference figure after assembly. First, select a position where to keep the cooler and firmly slide the storage chamber on the steel frame and hold it with a bolt and nut. Next, align the three fans behind the humidifier holder before fixing the holder on the opening created for it on one side of the storage chamber wall. Then, fix the spray nozzles with a ball and socket valve on the top tank with the help of a flexible hose, PVC gum, elbows and joints. Keep the tank on top of the cooling chamber between the two vents and sets the spray nozzles on the upper sump on top of the humidifier holder. Next, unlock the outer part of the humidifier holder and load the biomass humidifier and close it with the nuts and bolt. Assemble the floating valve and fix it through the hole created for it with the help of a rubber seal inside the bottom tank. The two terminals of the floating valves should be fitted with a copper hook connected to the pump and the power source to serve as the water pump sensor. Next fix the bottom tank firmly into the space on the steel frame under the storage tank. Connect the bottom tank to the pump inlet with a PVC pipe. Hold the pump with a screw and bolt on the steel stand linked to the steel frame. Next link the pump discharge with the upper tank with the help of PVC pipe as shown in Fig. 4. Once the assembling is completed, the bottom tank can be filled with water until the floating valve activates the pump to pump water into the top tank. During the pumping, the ball and socket valve should be opened to allow water to spray on the edges of the biomass humidifier. Initially, the humidifier will absorb the water until it is saturated before releasing the excess back to the bottom tank through the bottom sump. The next switch on the three fans is controlled through a single rheostat switch. Allow the cooler to stabilize for 15 to 30 min before loading your fruits and vegetables.

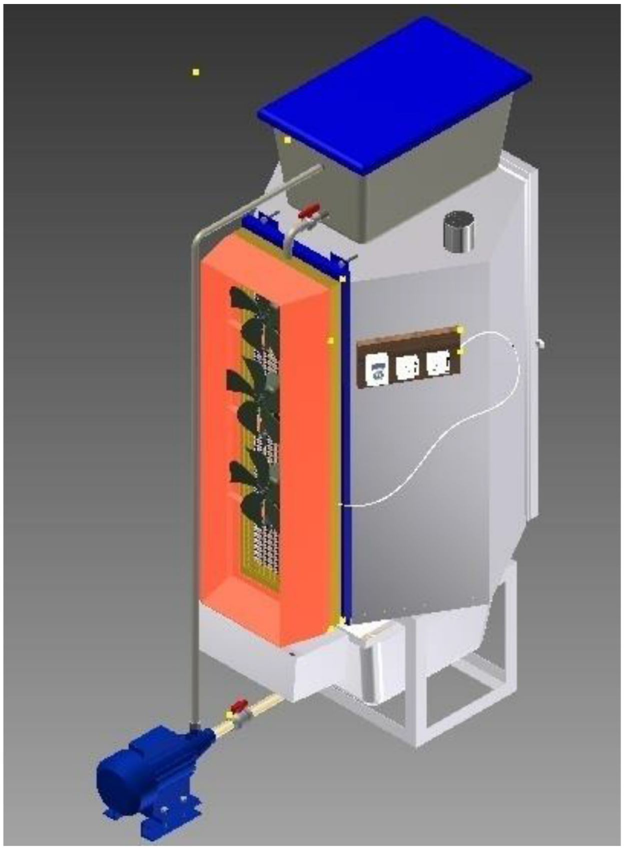


Fig. 4. Reference Evaporative cooling system assembly.

5. Experimental tests procedure

Experimental tests were undertaken with, 2.8 kg of dry wood charcoal (0.8374 kJ/kg K), 1.3 kg of dry jute fibres (1.6 kJ/kg K) and 0.98 kg of very dry palm fruit mesocarp fibre (2.816 kJ/kg K) at three different fan velocities of 4.5, 4.0 and 3.0 m/s to study the effectiveness of the design and the biomass humidifiers in the reduction of the temperature and improvement of the relative humidity of the ambient air. This is also to establish the transient response to variations in prevailing weather conditions. The test was carried out in February periods at Akure South Western Nigeria. This period presented the most challenging period for evaporative cooling within the possible climatic period when evaporative cooling can be applied in this location. The idea is to obtain minimal performance which is expected to improve in December and January with lower humidity. During this period, the relative humidity varied from 28 to 80% with the ambient temperature reaching 45 °C. The loading thickness of the humidifiers was 0.03 m and they were parked inside the humidifier holder with the bulk density ranging from 200 -220 kg/m^3. The water (20 L / 20 kg) wets the humidifier at a maximum water flow rate of 10 cm^3/s from the top edges through the nozzles (pinholes) directly lay on top of the horizontal edges of the humidifier. On saturation, the water drains into the bottom sump which contains a floating valve that automatically triggers re-circulation of the water back to the water tank. Before data collection, the humidifier was allowed to be wet for 24 h to remove any possible hot spots. The cooling chamber temperature data collection is done hourly with a thermocouple located at four points and connected to a data logger (omega data logger, HH1147) (±0.1 °C). The air velocity was measured at three points with a digital anemometer (AM-4826) (±0.1 m/s). The ambient temperature and humidity were determined with a humidity clock (±0.1 °C and 1.0%). A liquid in a glass thermometer is inserted into the transparent polycarbonate water tank and the sump measures the inlet and outlet water temperature. The experiment was carried out under an open shade (2.5 × 2 × 2.5 m) with only the roof covered and supported with mild steel pipes. The shed has the advantage of reducing considerably the effects of direct solar radiation on the test facility and exposes it to conventional airflow. The cooler was positioned facing the most frequent direction of the wind.

6. Performance evaluations

The cooling efficiency defined by Eq. (12) is a widely used index for evaluating the performance of direct evaporative cooling systems. It is given as follows [20,37].

$$\varepsilon = \frac{T_{db} - T_c}{T_{db} - T_w}. \tag{12}$$

where T_{db} is the shade dry bulb temperature, T_c is the evaporative cooler dry bulb temperature, and T_w is the wet bulb temperature calculated with Eq. (13) as follows [30,34];

$$\begin{aligned} T_w = \; & T_{db} \times \arctan\left[0.151977 \times \sqrt{Rh + 8.313659}\right] + \arctan\left(T_{db} + Rh\right) \\ & - \arctan\left(Rh - 1.67633\right) + 0.00391838 \times (Rh)^{3/2} \\ & \times \arctan(0.023101 \times Rh) - 4.686035) \end{aligned} \tag{13}$$

Where Rh is the relative humidity (%).

The rate of moisture lost due to evaporation is determined with Eq. (14) as follows

$$Mrr = \left(\omega_i - \omega_o\right)\dot{m}_a. \tag{14}$$

where $\dot{m}_a$. is the mass flow rate of air, ω_i is the specific humidity of air inlet and ω_o is the specific humidity of outlet air.

The specific humidity is deduced with Eq. (15) as follows [27]

$$\omega = \frac{\epsilon . P_v}{P_{atm} - P_v.(1- \in)} \tag{15}$$

ϵ is the gas constant ratio between wet and dry air given as 0.622 kg kg^{-1}, and P_v is the vapour pressure (Pa) calculated with Eq. (16) as follows [19]

$$P_v = 610.8 \; exp\left(\frac{17.27\,T}{T + 273}\right) \tag{16}$$

The coefficient of performance (COP) is calculated with Eq. (17) and (18) as follows

$$\mathrm{COP} = \frac{\dot{m}_a C_{pa}\left(T_{in} - T_{out}\right)}{E_c} \tag{17}$$

$$E_c = W_p + W_f \tag{18}$$

where W_p and W_f are the power input for the water pump and the fan respectively.

The ratio of sensible heat added to the air supplied by the humidifier is given in Eq. (19) as follows

$$SHR = \frac{m_H C_p \Delta T}{m_H C_p \Delta T + m_w L} \tag{19}$$

7. Uncertainty error analysis

The measured data and their values obtained are associated with errors of uncertainty. The experimental uncertainties of measurement of temperatures, air velocities, and relative humidity were used to calculate the uncertainties from, cooling efficiency, rate of moisture loss, COP, and sensible heat ratio using Eq. (20) as follows

$$U_R = \left[\left(\frac{\partial R}{x_1}\right)y_1^2 + \left(\frac{\partial R}{x_2}\right)y_2^2 + \ldots + \left(\frac{\partial R}{x_2}\right)y_n^2\right]^{1/2} \tag{20}$$

where y is the uncertainty in variables x. The calculated uncertainties for evaporative efficiency, moisture removal rate and COP were 0.342,

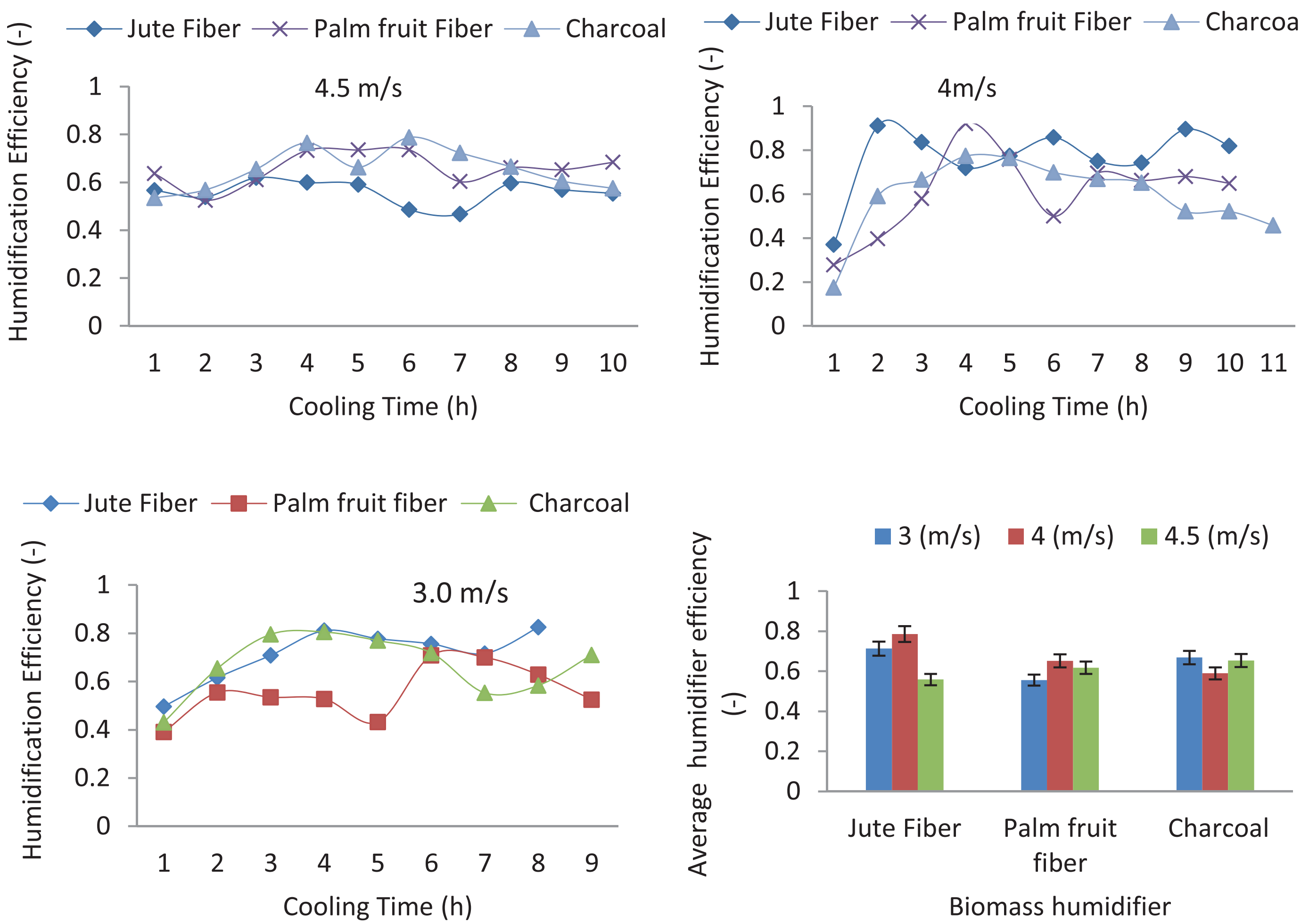


Fig. 5. Average humidifier efficiency and its variation with time at different air velocities.

0.0013 and 0.042, respectively. The uncertainty results indicate the level of confidence in the obtained data. The closer the values to zero the better the results. Therefore the obtained values showed a higher level of confidence in the primary data measured.

8. Results and discussion

Theoretically, direct evaporative cooling can only cool down the temperature of the inlet air to its wet bulb temperature. However, in most cases, this wet bulb temperature is not always attained in direct evaporative cooling. The reason is that the air coming in contact wh the humidifier is hotter and dryer due to solar irradiance and ambient conditions and some of this air might escape the humidifier without coming in contact with the water due to a possible hot spot on the humidifier. Other parameters like the inlet air velocity and thickness of the humidifier have also been confirmed as a factor. Therefore, the degree of temperature depression as the air exits from the humidifier into the cold storage chamber is associated with the evaporative effectiveness (humidification efficiency) of the humidifier. Typically, the value of the humidification efficiency ranges from 0 to 1.0 (0 – 100%). This parameter is an indicator of cooling that can be extracted from the stored product to extend its post-harvest shelf life. Thus from Fig. 5, these values varied with time irrespective of the fan speed or air flow rate or biomass humidifier type. This is because of the vagaries of weather conditions which alter fluidly the inlet temperature and humidity of the inlet pad. Thus, if the ambient humidity or temperature is constant, the depression of the temperature will be higher with increased inlet temperature or lower humidity. From the results, the humidification efficiency ranged from 0.49–0.62, 0.53–0.73 and 0.53–0.78 for jute fibre, palm fruit mesocarp fibre and wood charcoal respectively at 4.5 m/s (0.41 kg/s, average value). Also for the same biomass humidifiers, the values of humidification efficiency were 0.37 – 0.91, 0.28 – 0.92 and 0.18 – 0.77 respectively at 4 m/s (0.37 kg/s, average value). Furthermore at 3 m/s (0.28 kg/s, average value) the range of values for the humidifiers were 0.50–0.83, 0.39–0.71 and 0.43–0.8, respectively. However, on average the jute fibre provided a humidification efficiency of 0.71, 0.79 and 0.56 for air velocities of 3, 4 and 4.5 m/s respectively. In contrast, this value was 0.56 0.65 and 0.61 respectively for palm fruit mesocarp fibre. Additionally, for wood charcoal, the average values of the humidification efficiency were 0.67, 0.60 and 0.65 for air velocities of 3, 4 and 4.5 m/s respectively. Overall the result showed the capacity of the simple cooler and the biomass humidifiers to lower the temperature of the inlet air closer to the wet bulb value which is an indicator of the shelve live extension capability of the cooler. Jute fibre had the best performance at 4 m/s compared to other evaluation parametric conditions. This might be because the jute fibre had higher moisture holding capacity than the others. Therefore it has higher moisture content for evaporation to occur. Theoretically, we can use the obtained values and metrological data to determine if the ambient condition will be suitable for direct evaporative cooling application by empirically expressing the evaporative cooler temperature to be achieved using different biomass humidifiers. Therefore we fitted the cooling effectiveness with the airflow rate, humidifier volume and the inlet air relative humidity for each biomass humidifier to obtain the outlet temperature.

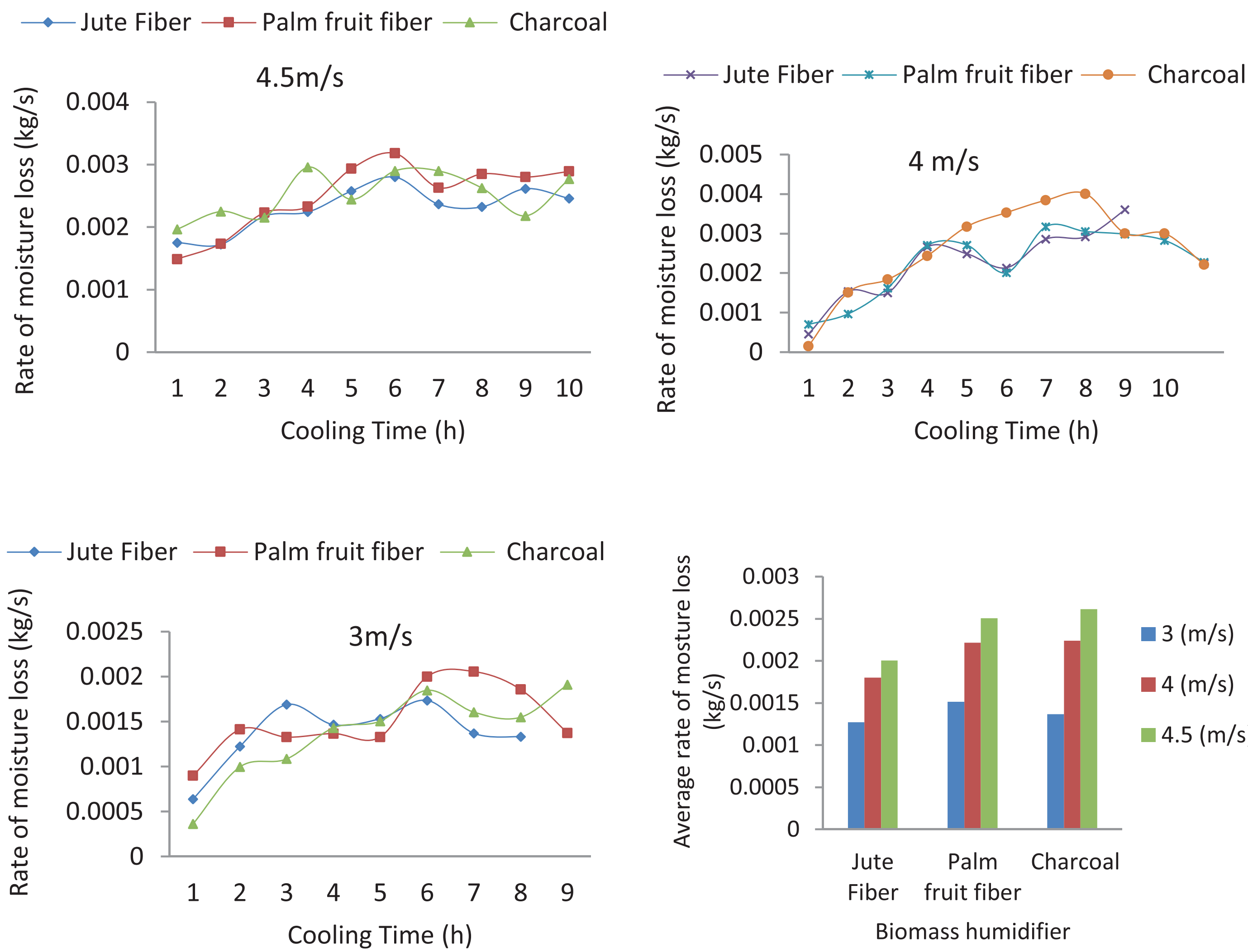


Fig. 6. Average moisture removal rate and its variation with time at different air velocities.

The obtained empirical equation is given in Eqs. (21)–(23) for jute, palm fruit mesocarp fibre and wood charcoal humidifier respectively as follows

$$T_c = T_{ab} - \left[\left(T_{ab} - T_w\right)\left(0.226 - 0.0094RH + 2.55\dot{m}_a + 0.00203V_H\right)\right]. \quad R^2 = 0.81 \tag{21}$$

$$T_c = T_{ab} - \left[\left(T_{ab} - T_w\right)\left(0.182 + 0.0131RH - 0.0546\dot{m}_a + 0.0016355V_H\right)\right]. \quad R^2 = 0.78 \tag{22}$$

$$T_c = T_{ab} - \left[\left(T_{ab} - T_w\right)\left(1.52 - 0.006288RH - 1.739\dot{m}_a - 0.00912V_H\right)\right]. \quad R^2 = 0.76 \tag{23}$$

where T_c is the direct evaporative cooler outlet air temperature (°C), T_{ab} is the inlet air temperature (°C), T_w is the wet bulb temperature (°C) of the inlet air, RH is the relative humidity (%) of the inlet air, $\dot{m}_a$. is the mass flow rate of air (kg/s) and V_H is the volume of the humidifier (m^3).

With the above regression equation, it will be easier to make an indicative assessment for the best season and location with the location metrological data where we can deploy direct evaporative cooling using any of the evaluated biomass humidifiers and also the product it can store based on the storage temperature of the product. The lower the value of T_c, the higher t capacity of the inlet air to cause more cooling of the stored product and extend the shelf life.

Moisture loss from the humidifier leads to the presence of a hot spot which introduces hot air into the cooling chamber. The presence of moisture on the humidifier enhances the cooling capacity due to the large body of moisture for evaporation. Therefore, we evaluated the rate of moisture loss for each biomass humidifier used to ascertain the transient response of each material to moisture loss at a constant humidifier thickness of 0.03 m. From the average rate of moisture loss in Fig. 6, the rate of moisture loss increased with an increase in air velocity though this value varies with time of the day due to continuous changes in the ambient conditions at different air speeds as also shown in Fig. 6. This showed that less water is lost at lower air velocity. This is because the rate of moisture loss is a product of the specific humidity of the air streams at the inlet and outlet and also the air stream flow rate. It is worth noting that less water will be lost at higher humidifier thickness and constant air velocity because of additional water for wetting the extra thickness. Higher moisture loss is an indication of more water consumption by the humidifier. Water usage is an issue in evaporative cooling due lack of water in some places. Therefore it is important to make choices in design and biomass humidifier material that can guarantee the desired cooling need at efficient water consumption. This also depends on the meteorological condition of each location and the product to be stored. Lesser moisture available for the humidifier translates to lower cooling performance and vice versa due to lower available moisture for evapora-

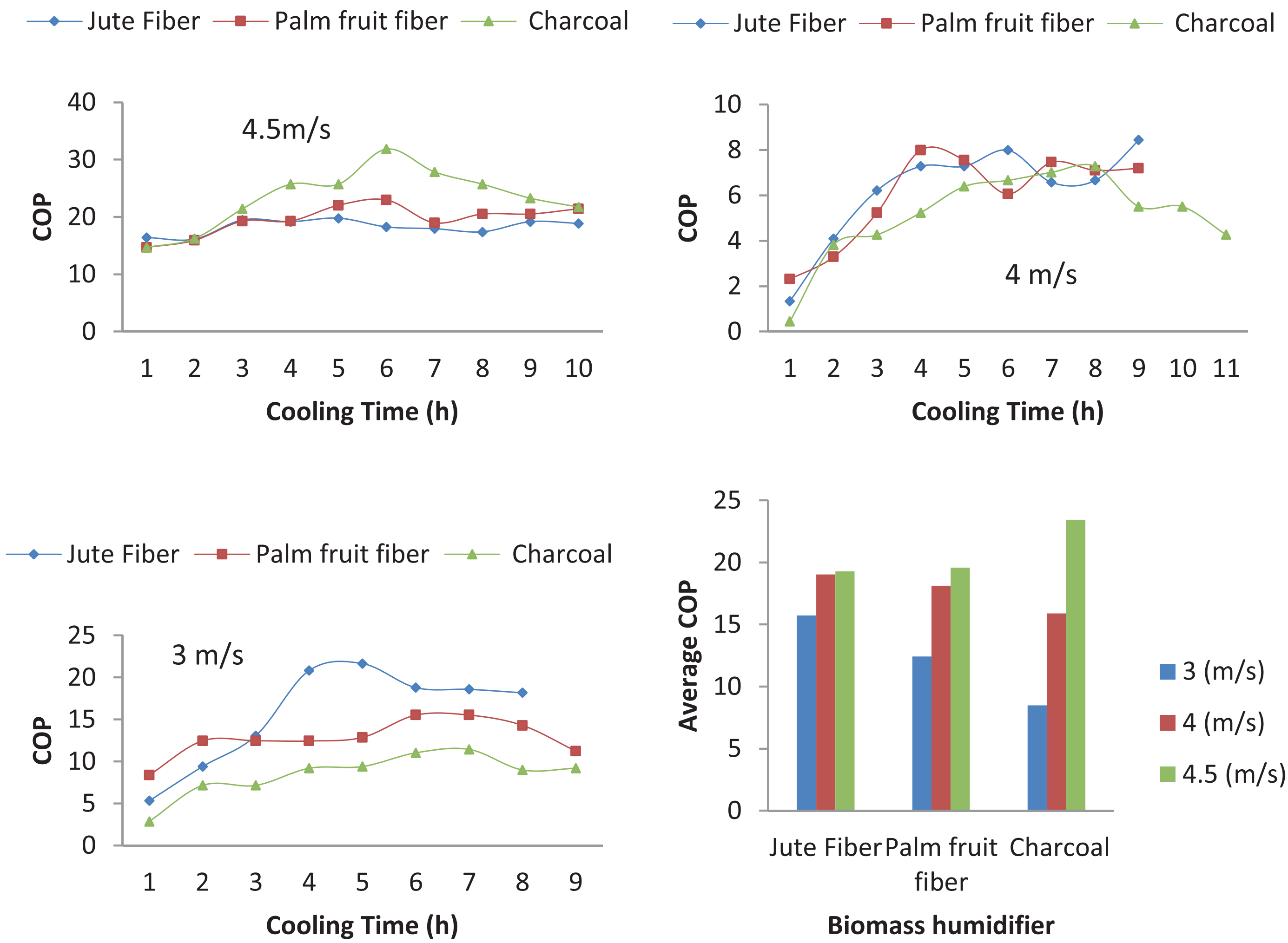


Fig. 7. COP and its variation with time at different air velocities.

tion. Jute fibre had the lowest average rate of moisture loss compared to palm fruit fibre and charcoal as shown in Fig. 6. This showed that jute fibre is capable of providing a better cooling blanket per thickness of humidifier than others because of its high water holding ability. In contrast, wood charcoal had the highest rate of moisture loss at 4.5 m/s. This might be due to large pore spaces which remain even after wetting and also the lower moisture holding capacity of the wood charcoal compared to others. Higher moisture loss will result in more energy consumption from the pump as this will require the pump to supply the lost moisture back to the humidifier. Using empirical relationships, we can obtain the values of the rate of moisture loss with the air mass flow rate, the relative humidity, ambient temperature and humidifier volume for Jute fibre, palm fruit mesocarp fibre and wooden charcoal as shown in Eqs. (24)–(26) respectively.

$$Mrr = 0.00001115RH + 0.00001973T_{db} - 0.00499\dot{m}_a + 0.0450V_H + 0.0033. \quad R^2 = 0.76 \tag{24}$$

$$Mrr = 0.0002053T_{ab} - 0.000006753RH + 0.006194\dot{m}_a - 0.72V_H + 0.0006461. \quad R^2 = 0.82 \tag{25}$$

$$Mrr = 0.0001397T_{ab} - 0.000004462RH + 0.00729\dot{m}_a - 0.25V_H - 0.000074. \quad R^2 = 0.91 \tag{26}$$

Where T_{ab} is the inlet air temperature (°C), RH is the relative humidity (%) of the inlet air, $\dot{m}_a$ is the mass flow rate of air (kg/s) and V_H is the volume of the humidifier ().

The coefficient of performance measures the ratio of energy output due to cooling and the overall energy supplied to achieve cooling. For all the biomass humidifiers tested, the COP increased with air velocity as shown in Fig. 7. This showed that the heat transfer rate was higher at higher air velocity which resulted in higher moisture rate loss at higher air velocity, as shown in Fig. 6. Jute fibre had the highest COP at 3 and 4 m/s while wood charcoal had the highest COP at 4.5 m/s. Fig. 7 also showed that the COP varied strongly with time due to the transient response of heat transfer with the weather variations. The behaviour is similar to all calculated evaluation parameters which are common with direct evaporative coolers. The highest COP of 31.85 was obtained for wood charcoal at 4.5 m/s while the same wooden charcoal produced the lowest COP of 2.85 at 3 m/s. If this calculation is based on an energy efficiency rating ($EER = \frac{\Delta H}{Input\ power} = \frac{(T_{db}-T_c)C_p}{Input\ power}$) for the system, it will translate to the highest EER of 78 at 4.5 m/s and the lowest EER of 11 at 3 m/s. Comparing the EER to that of mechanical vapour refrigeration (EER = 6–12), [12] this can translate to energy savings of about 45 to 85%. These values will increase in drier weather conditions in the month of December to February in the study location. The COP increased with time which corresponds to an increase in dry bulb temperature due to increased solar intensity in the afternoon. However, it decreased towards the evening time when solar intensity is adjudged to go down in the studied location zone [23–26].

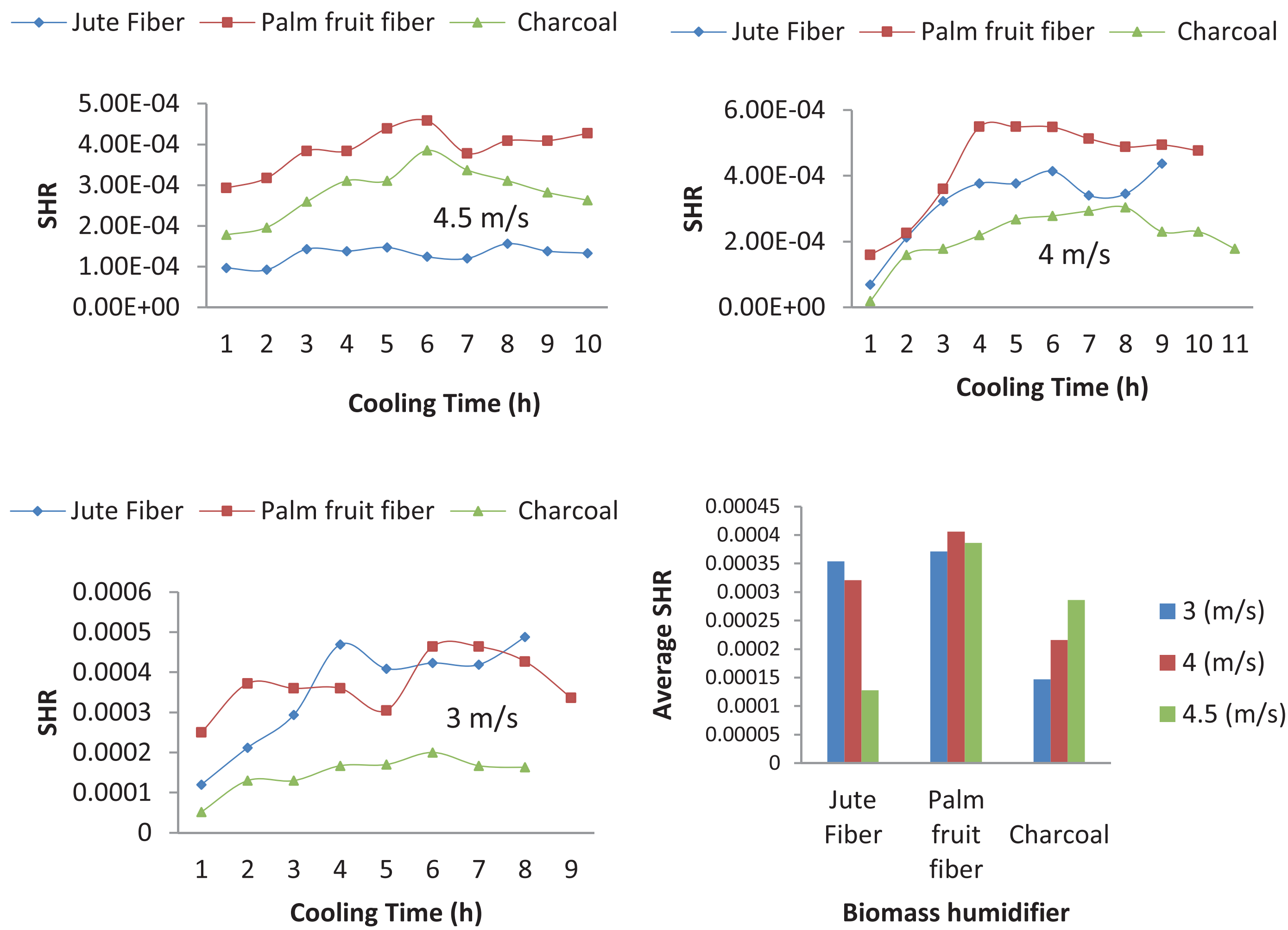


Fig. 8. Average removal rate and its variation with time at different air velocities.

The sensible heat transfer ratio (SHR) shows the fraction of extracted heat from the biomass to the overall heat extracted from both the biomass and evaporated moisture. The biomass will lose heat which is acquired by the moisture and the air however humidification reduces the value. Lowering this value will reduce the sensible heat and increases the cooling capacity/efficiency of the evaporative cooler. The values of SHR are shown in Fig. 8 for all the biomass humidifiers. The palm fruit mesocarp fibre has the highest sensible heat ratio due to its higher specific heat capacity. The highest value obtained was 5.49×10^{-4} for palm fruit mesocarp fibre at 4.0 m/s. On average, wooden charcoal had the lowest SHR for all the air velocities while palm fruit mesocarp fibre had the highest.

9. Conclusions

The feasibility of deploying a direct evaporative cooler with Jute fibre, palm fruit mesocarp fibre and wooden charcoal as a humidifier was investigated in the February period in Akure Nigeria. An experimental stand-alone evaporative cooler was built for the above purpose. The built evaporative cooler and the three biomass humidifiers were capable of humidifying inlet air and lowering the temperature. The humidifiers presented average cooling efficiency of 55.9 to 78.62%, an average rate of moisture loss of 1.37×10^{-3} to 2.61×10^{-3} kg/s, an average COP of 8.48 to 23.42 (EER of 11 to 78) and average sensible heat ratio of 1.28×10^{-4} to 4.06×10^{-4} for the air velocity of 3.0 to 4.5 m/s. Jute fibre had the highest average humidifying efficiency of 78.62 at 4.0 m/s with a lower rate of moisture loss followed by palm fruit mesocarp fibre. These cheap biomass humidifiers can replace the synthetic costly humidifiers to reduce the cost of cooling. The obtained performance can be better in a dryer month or nearly impossible in a very wet month as the humidifying efficiency of direct evaporative coolers is found to diminish at high humidity. To avoid casting doubt on the effectiveness of direct evaporative coolers by farmers, they should only serve as stop-gap preservation equipment and be deployed during favourable weather conditions especially during the winter when the air is dry. These can be identified from the metrological charts of the chosen location.

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Declaration of Competing Interest

The authors declare no conflict of interest and declare that the corresponding author signs every author's agreement regarding this article on behalf of all the authors

Data Availability

Data will be made available on request.

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