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# Performance Evaluation of Native-Kankan Padded Evaporative Space Cooler Using Arduino Mega

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**Abstract**
Performance evaluation of native-kankan padded evaporative space cooler using Arduino Mega is presented. Materials for fabrication were both locally improvised and conventionally sourced for such as the Nigerian Native-Kankan fibre sponge used as the wet and dry filter pads and expanded polystyrene which is used as thermal insulation material. Dry air from the outdoor surrounding is passed through the soaked Kankan fibre pad using a reverse DC fan and cooled by evaporative means where the sensible heat of dry air is converted to latent heat accumulation in the circulating working fluid. 150Watts solar power was supplied to the constructed cooler for a test run and results showed that the peak temperature drop of 27□ to 24.5□ was experienced in the indoor space / cooler exit of the test room facility. The average system COP of 1.224, evaporation rate of 0.066GPH and cooling capacity of 95.23W was achieved from the conducted experiment. The native-kankan dry filter pad reduced the humidity level of cooled air entering the indoor test room cubicle by 2%. A comprehensive cooling load analysis of the test room facility was carried out and the results were applied using sensible heat removal method to achieve cooler size rating of 319.02CFM. All experimental results were collected using Arduino Mega, type K thermocouple sensors and DHT11 humidity sensors installed on the constructed evaporative cooling unit.
**Keywords:** Latent heat; Air conditioning; Relative Humidity, Arduino Mega; Evaporative Cooling; Native-Kankan fibre sponge.

## 1. Introduction

The automatic control of an atmospheric environment either for the comfort of human beings or animals or for the proper performance of some industrial or scientific process summarizes what is accepted to be air conditioning (Jones 2001). Space cooling either for industrial process applications, fruit preservation, human thermal comfort applications etc is currently dominated by conventional compression refrigeration system, which is not desirable for energy use and global climate concerns. The conventional compression refrigeration system is highly energy intensive due to extensive use of energy for the operation of the the compressor unit and its environmental unfriendliness (Raymond et al 2017). On the contrary, it has been proven that evaporative cooling technologies are energy-efficient and are viable alternatives to compressor-based cooling methods. Bhatia (2012), maintains that evaporative cooling can meet most or all building cooling loads using one-fourth of the energy of conventional equipment. Bhatia (2012), also stated that evaporative cooling technologies are more cost effective when integrated with conventional chiller systems. Evaporative air cooling in recent times, has become environmentally friendly and less in energy consumption, hence highly attractive to renewable energy researchers. An Evaporative cooling system is divided into three types: Direct, Indirect and two stage Direct and Indirect Evaporative cooling systems (Raymond et al 2017). The principal science underlying evaporative cooling is the fact that liquid absorbs significantly more heat to become vapour hence, evaporation occurs (baltimoreaircoil.com). The aim of this work is to use locally available materials such as the Native-Kankan fibre sponge, conventionally sourced for materials and other improvised methods to achieve direct evaporative cooling for a typical Nigerian cubicle room space. The coinage 'Kankan' is typical to the Yoruba speaking part of Nigeria (West-Nigeria) where it is predominantly used as sponge for various purposes. The performance of the constructed Native-Kankan padded evaporative space cooler is measured using the Aurdino Mega instrumentation set up. Arduino is an open-source physical computing platform based on a simple input/output board and a development environment that implements the Processing/Wiring language (sparkfun.com). Arduino can be used to develop stand-alone interactive objects or can be connected to software on your computer e.g. Flash, Processing, MaxMSP (sparkfun.com). According to (sparkfun.com), the Arduino Mega is a microcontroller board based on the ATmega 2560 and it has 54 digital input/output pins (of which 14 can be used as PWM outputs), 16 analog inputs, 4UARTs (hardware serial ports), a 16 MHz crystal oscillator, a USB connection, a power jack, an ICSP header, and a reset button. A picture view of the Arduino Mega board is presented in figure 4.

## 2. Working Principle

The designed evaporative cooling system is similar to the conventional evaporative coolers with some modifications like the incorporation of Native-Kankan fibre sponge to serve for both wet and dry filter pads, creation of a separate chamber for the dry filter pad to enhance moisture reduction content of processed air flow into indoor space, and the incorporation of expanded polystyrene called Styrofoam as a thermal insulation material. In the design of this evaporative cooler, two chambers were created namely: the mixing chamber and the moisture reduction or the dry chamber. When water is pumped from the tank, which is kept at ground level with the help of a submersible pump, the water flows through to the mixing chamber where the wetting of the first filter pad takes place and the evaporation of the water molecules caused by humid air intake occurs. This chamber accounts for 90% of the total volume of the box where cooling takes place. The moisture reduction process occurs as the dry filter pad processes the moist cooled air to improve the air quality. The water is sprayed on the first placed dry filter pad to saturation point where water begins to drip and drained out of the mixing chamber through a collection channel. The reversed suction fan plays a key role by sucking ambient air through the wet filter pad porous spaces called void fractions. As the ambient air passes through the first placed wet filter pad, it exchanges its sensible heat with the latent heat of water and evaporation occurs which eventually causes the desired cooling effects on the air intake into the test room cubicle. At this point the reduced temperature and relative humidity of the processed air quality is envisaged in the indoor space of the test room facility.

## 3. Evaporative Cooling Test Rig Description

The test room cubicle is $27.35m^3$ in volume. The constructed evaporative cooler has embedded in it the Native-Kankan filter pads, enclosed with wire gauzes and positioned by means of binding wires to an aluminium bar making an overall area of 1600 $cm^2$ for each pad. The pump selected for the working fluid transport is a submersible 12 DC volt pump which has a power rating of 0.035hp. The tank is made of plastic and can contain approximately 17 litres of water when filled to the brim. The tank is improvised from common water dispensers readily available. The cooling box is made of gauge 18 aluminium steel internally insulated with Styrofoam. The box is made of aluminium casing with volume $96000cm^3$. The dry filter pad is placed 10cm away from the suction fan. The water collection channel is rectangle in shape, fastened with the help of rivets directly under the cooling box. The base of this collection channel is left open, but slanted to allow for free flow of dripping water from the saturated filter pad to a particular collection spot. A return hose line is connected from the collection channel to the storage tank for flow of water via gravity. Two hose lines were fitted into the tank; The first hose fitted to the submersible pump which supplies water for soaking the first filter pad while the other hose serves as a return flow line from the collection channel to the storage tank.

### *3.1 Design Considerations and Analysis*

In the determination of the cooler size, a comprehensive cooling load analysis was carried out for the particular test room facility needed for cooling. According to Watt (1986) two methods apply for the design of evaporative coolers namely: Air Change Method and the Sensible Heat Removal method. This work adopts the sensible heat removal method where a comprehensive cooling load analysis was carried out and the outdoor dry/wet bulb temperature, saturation efficiency and wet bulb depression were measured and used to calculate the evaporation rate (GPH) in terms of heat removal rate represented in standard cubic feet per minute (SCFM). The Standard cubic feet per minute (SCFM) is a volumetric flow-rate corrected to a set of 'standardized' conditions of pressure, temperature, and relative humidity while the CFM is a measure that describes the rate of air flow in and out of a space (Wikipedia.org). The detailed cooling load analysis is not presented in this article, however energy formulas presented here were solved based on results from the cooling load analysis. The cooling load energy balance accounted for both sensible and latent heat gains of the test room namely; heat gain from occupants, heat gain from natural infiltration of humid air, heat gain from lighting and room appliance, heat gain though exterior wall, roof, door, floor, window and solar radiation heat gains. Most of the heat equations and constants used for the detailed cooling load analysis where adopted form ASHRAE (2007), CIBSE (2006), and Jones (2001).

#### *3.1.1 Heat gain from occupants*

$$Q = n \times X + n \times Y \quad (1)$$

Where Y and X are rate of latent and sensible heat gains (W), given by $Y=45\times N_L$ and $X=70\times N_L$; $N_L$ is the total number of humans anticipated occupants of the test room facility.

#### *3.1.2 Heat gain from lighting*

$$Q_l = \text{no of bulbs} \times \text{fluorescent factor} \times \text{use factor} \times \text{power rating} \quad (2)$$

#### *3.1.3 Heat gain from one window glass*

$$Q_g = AU(T_0 - T_r) \quad (3)$$

#### *3.1.4 Heat gain from roof*

$$Q_r = AU(T_{rm} - T_r) + AU(T_o - T_{rm})f \quad (4)$$

#### *3.1.5 Heat gain from walls and floor*

$$Q_f = AU(T_{oav} - T_r) + AU(T_r - T_{oav})f \quad (5)$$

#### *3.1.6 Latent and Sensible heat gains by natural infiltration through other small openings*

$$Q_L = 0.8 \times n \times V \times (g_0 - g_r) \quad (6)$$

$$Q_s = 0.33 \times n \times V \times (T_0 - T_r) \quad (7)$$

$$\text{Total Heat Gain in test room } (Q_t) = \text{Total sensible heat gain} + \text{Total latent heat} \quad (8)$$

$$Q_t = 1323.06W + 101{,}13W = 1424.19W$$

*3.1.7 Heat gains from solar radiation*

The heat gained from solar consists majorly of the heat absorbed by walls and roofs exposed to solar radiation and heat transmitted directly through window glass and some variables like declination, solar altitude, azimuth of the sun, wall solar azimuth, hour angle. These factors contribute immensely in the determination of solar radiation in the test room space. In calculating these solar radiation parameters, temperature data for the test facility roof and soil were obtained from the Centre for Atmospheric and National Space Research Development Agency (CAR-NASRDA), Nsukka on Latitude 6.89°N, longitude 7.40°E and altitude of 416m.

*3.2 Calculating the CFM*

In the sensible heat removal method, the calculated heat gain of the test room is investigated to determine the required CFM. The solutions of equations 9 – 12 below were used to determine the constructed evaporative cooler size and the power input required to run the cooler.

$$\text{Standard CFM} = \frac{\text{Sensible Heat Gain}}{[1.08\times(\text{IDB}-\text{LDB})\times\text{Density Ratio}]} = 350.58\text{SCFM} \quad (9)$$

$$\text{CFM} = 350.58\text{SCFM} \times 0.91 = 319.02$$

*3.3 Calculating the Evaporation Rate*

$$\text{LDB} = \text{ODB} - [\text{SE} \times (\text{ODB} - \text{OWB})] \quad (10)$$

$$\text{WBD} = \text{ODB} - \text{OWB} \quad (11)$$

$$\text{Evaporation Rate(GPH)} = \frac{\text{CFM}\times\text{WBD}\times\text{SE}}{8700} = 0.066\text{GPH}$$

$$\text{Saturation Efficiency(SE)} = \frac{T_i - T_o}{T_i - T_{wb}} \quad (12)$$

*3.4 Energy Balance Analysis*

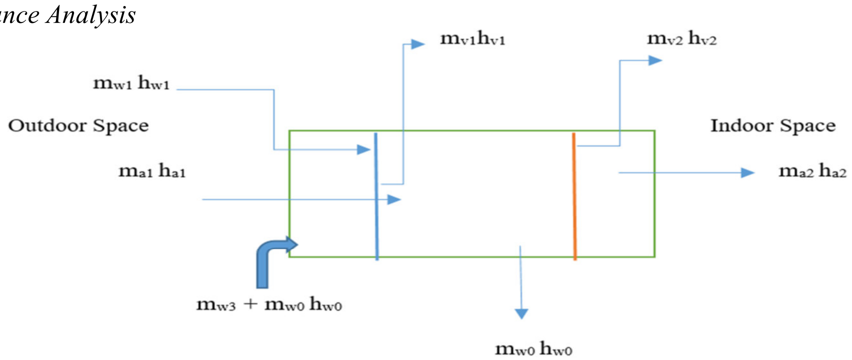



Fig. 1. Energy balance analysis of the evaporative cooling test rig

The following assumptions are adopted to simplify the analysis:

1. The detailed dimensional momentum, heat and mass transfer analyses of the problem is ignored, hence the energy and mass transfer in the system is considered in terms of specific enthalpies, heat capacities and flow rates at steady state.
2. Mass flow rate of air is equal at the cooler inlet/outlet, wet, mixing and dry chambers of the examined cooler; $m_a = m_{a1} = m_{a2}$
3. The working fluids flowing through the constructed cooler are atmospheric air and water at ambient conditions.

The applicable energy equation describes the energy interactions of the air, feed water, vaporized moist air, recirculated feed water and make up water. This can only be expressed mathematically as;

Energy inflow from outdoor space into the cooling unit = Energy outflow from cooling unit into the indoor space room facility.

$$m_{w1}h_{w1} + m_{a1}h_a + m_{w3} + m_{w0}h_{w0} - m_{v1}h_{v1} = m_{a2}h_{a2} + m_{w0}h_{w0} - m_{v2}h_{v2} \quad (13)$$

By employing assumption 2, equation 17 yields the mass flow rate of air from fan entering the cooler inlet at point 1. Equations 15-16 are the final measures to which the constructed cooler performance is evaluated.

$$m_{a1} = \frac{m_{v1}h_{v1} - m_{w3} - m_{w1}h_{w1} - m_{v2}h_{v2}}{h_{a1} - h_{a2}} \quad (14)$$

$$\text{COP} = \frac{m_{a1} \times C_{pma}(T_{db} - T_s) \times 1000}{\text{Power input}} \quad (15)$$

$$\text{Cooling Efficiency } (\ell) = \frac{T_{db}-T_s}{T_{db}-T_{wb}} \times 100 \quad (16)$$

$$\text{Cooling Capacity } (Q_i)\text{kW} = \frac{Q_v \times \rho_a \times C_{pma}(T_{db}-T_s) * 3.6}{3600} \quad (14)$$

*3.5 Thermodynamic Procedures*

Assuming the working fluids flowing through the constructed cooler are at ambient conditions as stated in assumption three (3), hence equations 15 – 16 is applicable for use in determining important parameters in equations 14-15. Table 1 shows the corresponding results obtained by hand calculations. All thermodynamic formulas and tables used here are adopted from Eastop et al (2004).

$$P = P_{a1} + P_{v1} \quad (15)$$

In equation 15, $P_{v1}$ is the partial pressure of the evaporating liquid vapour, $P_{a1}$ is the partial pressure of dry air and P is the barometric pressure 1.01325bar. The relative humidity, ∅and the specific humidity, ω is computed with available measured experimental data in Table 2 and the respective partial pressures in the cooler wet (Point 1) and dry (Point 2) chambers are consequently calculated using the relationships in equation 20 and 21.

$$\emptyset_{1,2} = \frac{P_{v1,2}}{P_g \text{ at } T_{db} \,\&\, T_s} \quad (16)$$

$$\omega_{1,2} = 0.622\left[\frac{P_{v1,2}}{P-P_{v1,2}}\right] P_{v1} = P_{a1} \,\&\, P_{v2} = P_{a2} \quad (17)$$

Table 1: Summary of calculated and constant values required for the psychometric mixture analysis

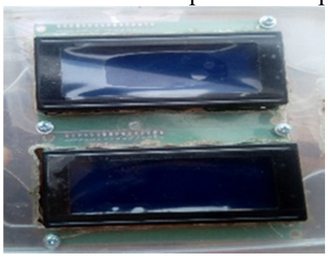
| No. | Description | Constant values | Symbol | Unit | Calculated Values |
|---|---|---|---|---|---|
| 1 | Volume of dry air at cooler inlet point 1 | - | $V_{a1}$ | $m^3/\text{dry air}$ | 0.8788 |
| 2 | Volume of dry air at cooler inlet point 2 | | $V_{a2}$ | ✓ | 1.0712 |
| 3 | Fan power input | 74.16 | - | $J/s$ | - |
| 4 | Pump work input | 26.16 | - | ✓ | - |
| 5 | Partial pressure of the evaporating moist air leaving cooler inlet point 1(dry chamber) | - | $P_{v1}$ | bar | 0.0335 |
| 6 | Partial pressure of air at cooler inlet point 1 | - | $P_{a1}$ | ✓ | 0.9797 |
| 7 | Partial pressure of the evaporating moist air leaving cooler at inlet point 2(dry chamber) | - | $P_{v2}$ | ✓ | 0.2152 |
| 8 | Partial pressure of air at cooler inlet point 2 | - | $P_{a2}$ | ✓ | 0.7977 |
| 9 | Mass flow rate of air from fan entering cooler to mid-point 0 (mixing chamber) | - | $m_a$ | $kg/s$ | 0.0364 |
| 10 | Mass flow rate of air from fan entering cooler inlet at point 1 | - | $m_{a1}$ | ✓ | 0.1502 |
| 11 | Mass flow rate of the evaporating moist air leaving cooler at point 1 (wet chamber) | - | $m_{v1}$ | ✓ | $7.7407\times10^{-4}$ |
| 12 | Mass flow rate of the evaporating moist air leaving cooler at point 2 (dry chamber) | - | $m_{v2}$ | ✓ | $6.306\times10^{-4}$ |
| 13 | Mass flow rate of water drained out through the collection chamber outlet for recirculation | - | $m_{wo}$ | ✓ | $1.1053\times10^{-4}$ |
| 14 | Mass flow rate of possible make up-water due to water loss by evaporation effect | - | $m_{w3}$ | ✓ | $1.4347\times10^{-4}$ |
| 15 | Mass flow rate of pumped water entering cooler at point 1 (wet chamber) | - | $m_e$ /$m_{w1}$ | ✓ | $2.54\times10^{-4}$ |
| 16 | Enthalpy of the evaporating moist air leaving cooler at point 1 | - | $h_{v1}$ | $kJ/kg$ | 2550.34 |
| 17 | Enthalpy of the evaporating moist air leaving cooler at point 2 | - | $h_{v2}$ | ✓ | 2546.15 |
| 18 | Enthalpy of water drained out at mid-point 0, through the collection chamber outlet for recirculation | - | $h_{w0}$ | ✓ | 103.75 |
| 19 | Enthalpy of air entering cooler inlet at point 1(wet chamber) | - | $h_{a1}$ | ✓ | 24.873 |
| 20 | Enthalpy of air entering cooler inlet at point 2(dry chamber) | - | $h_{a2}$ | ✓ | 27.135 |

## 4. Experimental Setup and Testing

The Experimental set up incorporated the following resources: Arduino Mega board, Data logger, DHT 11-humidity sensors, MAX6675 plug, Type K thermocouples, Two LCD Screens, Vero board, Jumper wires, and a data logger which has a port for memory card used for data storage. The DHT 11-humidity sensor has inbuilt capacitive humidity sensor and a thermistor to measure the surrounding air hence producing a digital signal. Two LCD screens were used to visualize the displayed information of temperature and humidity fluctuations in the cooler test rig. The Jumper wires were used to connect Arduino components to the Vero and Mega boards. The Arduino Mega microcontroller used has in it stored written codes that programmed the inputs/outputs of the Arduino system to ensure measurements. These codes were written via the open-source IDE software application downloadable for windows. The microcontroller is powered using a USB cable connected to a computer.

Ten type k thermocouples and four humidity sensors were procured and placed at specific points on the evaporative cooler. The points were labelled as indoor space/cooler exit air temperature-$T_1$, out-door ambient/cooler inlet Temperature of the room-$T_2$, wet Kankan fibre pad-$T_3$, dry Kankan pad-$T_4$, pump water temperature -$T_5$, ambient humidity-$H_1$, humidity of the cooled air released into the room-$H_2$, humidity in the moisture reduction chamber-$H_3$, humidity in the mixing chamber - $H_4$. Picture preview of the experimental setup and Arduino components are presented in figures 2-5


(a)

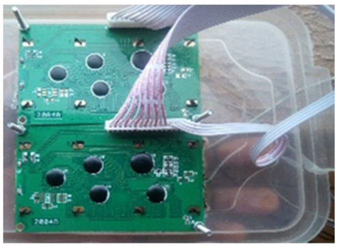
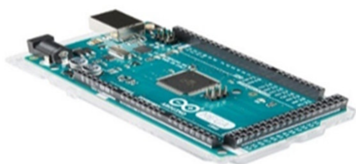

(b)

Fig. 2. (a) LCD top view (b) LCD bottom view


(a)


(b)

Fig. 3. (a) Inside view of complete Arduino setup box, (b) Complete Arduino setup box

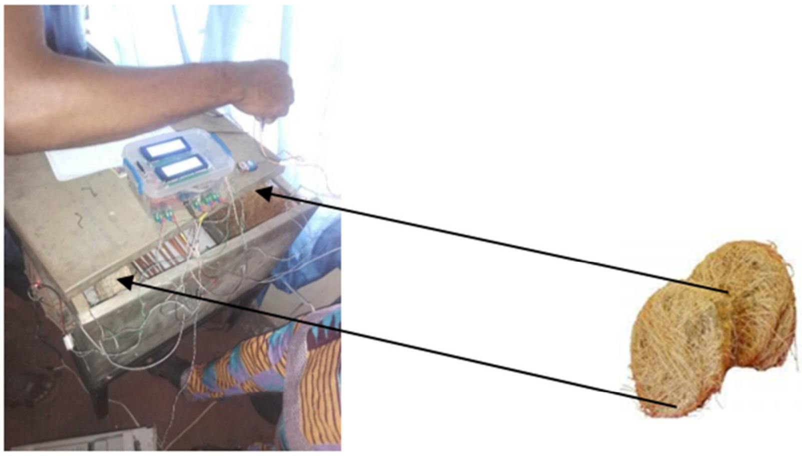

(a)


(b)


(c)

Fig. 4. (a) Arduino Mega board, (b) MAX6675 plug assembly, (c) MAX6675 plug /Type-K Thermocouple, Source: (sparkfun.com).



(a) (b)

Fig. 5. (a) Cooling Test Rig-Arduino Set up, (b) Native-Kankan fibre sponge used as wet and dry filter pads.

**5. Results and Discussion**

Results obtained from a cooling load test period of 30 minutes are presented in figures 6-9. In Figure 6, The temperature profile of the indoor space/cooler exit air temperature $–T_s$ and the out-door ambient/cooler inlet temperature of the test room-$T_{db}$ is plotted against time. The consistent drop in their temperatures clearly shows the evaporation rate and hence, decrease in the space/cooler exit air temperature $–T_s$. The plots and measured data presented in table 1, shows the temperature of the indoor space of the test room dropped from 27℃ to 24.5℃ while the temperature of the cooler inner chambers dropped from 26.25℃ to 24℃. The decrease in the cooler air temprature resulted to the temperature regulation of the indoor test room space over time. In Figure 7, the variations of exit temperature from the cooler test rig showed little or no effects on the humidity levels and the careful study of figure 8 revealed that as the out-door temperature reduces or increases the humidity levels changes insignificantly. Table 2, shows the results for cooling capacity, cooling efficiency and COP of the system during its test run period of 30 minutes. The plots for cooling capacity, cooling efficiency and COP are presented in figure 9.

**6. Conclusion**

Results showed that the Native-Kankan fibre-dry filter pad used reduced the humidity level of cooled air entering the indoor test room space only by 2% which is still not desirable for human thermal comfort. The average coefficient of performance COP of 1.224, peak performance COP of 3.76 and cooling capacity of 95.23W were obtained from measured experimental data in Table 2 collected for a test run period of 30 minutes.

Table 2. Measured and Calculated Results

| Time(min) | Ts(°C) | Tdb(°C) | Twb(°C) | 1(%) | Ø2(%) | *l* | *Qi* | COP |
|---|---|---|---|---|---|---|---|---|
| 1 | 26.25 | 27 | 25.75 | 92 | 88 | 60 | 103.8755 | 1.254402 |
| 2 | 26.25 | 27 | 25.75 | 93 | 88 | 60 | 103.8755 | 1.254402 |
| 3 | 26.5 | 27.25 | 26 | 93 | 89 | 60 | 103.8755 | 1.254402 |
| 4 | 26 | 27 | 25 | 93 | 89 | 50 | 138.5006 | 1.672536 |
| 5 | 26 | 27 | 25.75 | 93 | 89 | 80 | 138.5006 | 1.672536 |
| 6 | 26 | 26.75 | 25.75 | 93 | 89 | 75 | 103.8755 | 1.254402 |
| 7 | 26 | 26.25 | 25.75 | 93 | 89 | 50 | 34.62515 | 0.418134 |
| 8 | 26.25 | 27.5 | 26 | 93 | 88 | 83.33333 | 173.1258 | 2.09067 |
| 9 | 25.5 | 26.25 | 25.25 | 93 | 89 | 75 | 103.8755 | 1.254402 |
| 10 | 25.75 | 26.25 | 25 | 93 | 89 | 40 | 69.2503 | 0.836268 |
| 11 | 26 | 26.25 | 25.25 | 93 | 89 | 25 | 34.62515 | 0.418134 |
| 12 | 25.75 | 26.25 | 25.5 | 93 | 88 | 66.66667 | 69.2503 | 0.836268 |
| 13 | 25.75 | 26.25 | 25.5 | 93 | 89 | 66.66667 | 69.2503 | 0.836268 |
| 14 | 25.25 | 27 | 25 | 93 | 89 | 87.5 | 242.3761 | 2.926938 |
| 15 | 25.25 | 26 | 25 | 93 | 89 | 75 | 103.8755 | 1.254402 |
| 16 | 25.25 | 25.75 | 25 | 93 | 88 | 66.66667 | 69.2503 | 0.836268 |
| 17 | 25.25 | 25.75 | 25 | 93 | 88 | 66.66667 | 69.2503 | 0.836268 |
| 18 | 25.25 | 25.75 | 25 | 93 | 88 | 66.66667 | 69.2503 | 0.836268 |
| 19 | 25 | 25.5 | 25 | 93 | 88 | 100 | 69.2503 | 0.836268 |
| 20 | 25 | 25.5 | 25 | 93 | 87 | 100 | 69.2503 | 0.836268 |
| 21 | 25 | 25.5 | 25 | 93 | 88 | 100 | 69.2503 | 0.836268 |
| 22 | 25 | 25.75 | 24.75 | 92 | 89 | 75 | 103.8755 | 1.254402 |
| 23 | 25 | 25.75 | 24.75 | 93 | 89 | 75 | 103.8755 | 1.254402 |
| 24 | 24.75 | 25.75 | 24.75 | 93 | 88 | 100 | 138.5006 | 1.672536 |
| 25 | 24.75 | 27 | 24.5 | 94 | 88 | 90 | 311.6264 | 3.763205 |
| 26 | 24.75 | 25 | 24.5 | 95 | 88 | 50 | 34.62515 | 0.418134 |
| 27 | 24.5 | 25 | 24.25 | 93 | 88 | 66.66667 | 69.2503 | 0.836268 |
| 28 | 24.5 | 25 | 24 | 94 | 89 | 50 | 69.2503 | 0.836268 |

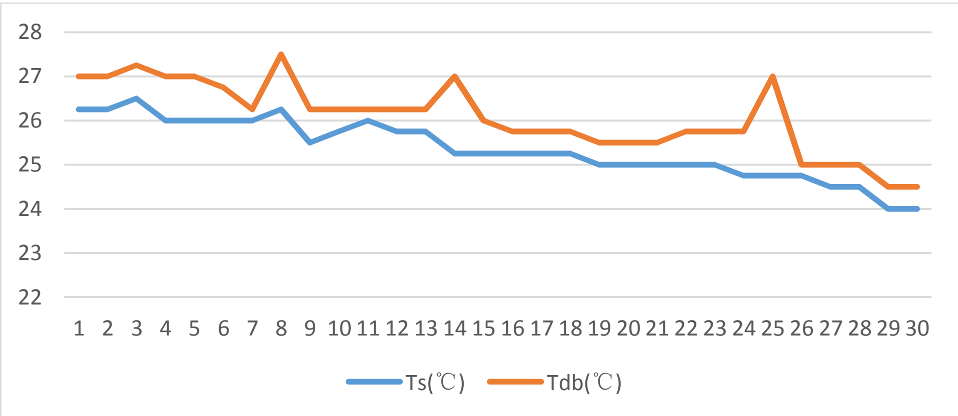



Figure 6. $T_s$℃ & $T_{db}$℃ against time.

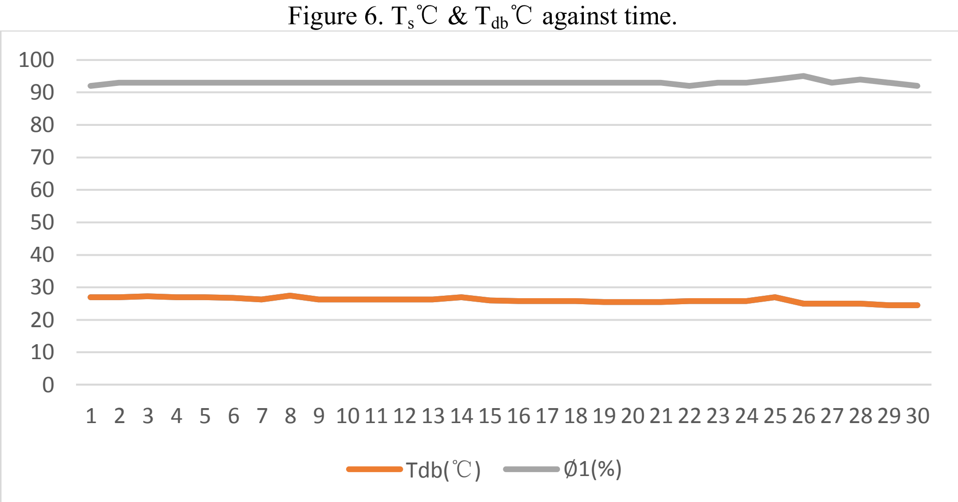



Figure 7. $T_{db}$℃ & $\varnothing_1$ against time.

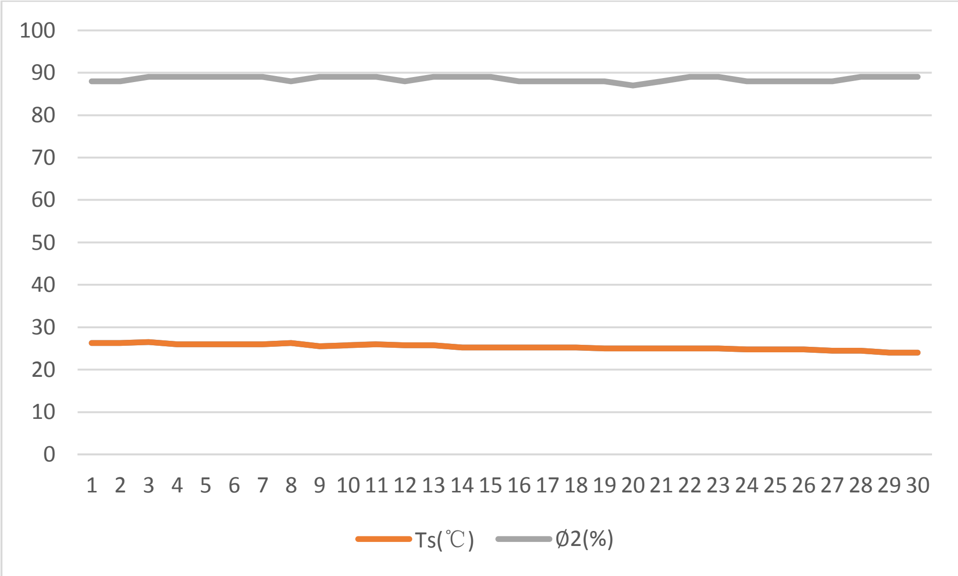



Figure 8. $T_s$℃ & $\varnothing_2$ against time.

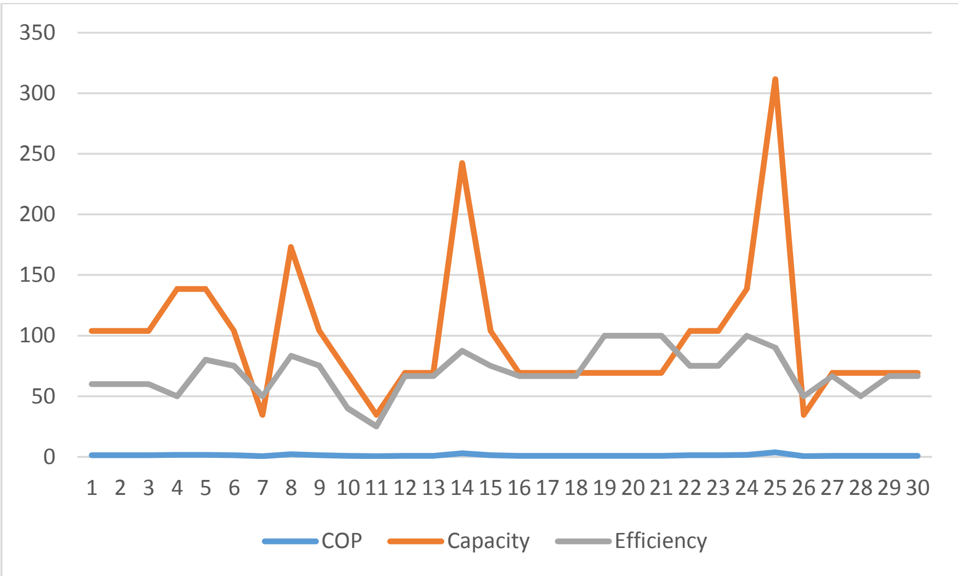



Figure 9. Cooling Efficiency (%), Cooling Capacity (kW) and COP against time.

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NOMENCLATURE

$m_e$ – rate of water consumption, kg/s
$Q_v$ – volumetric flow rate, m$^3$/s
$V_a$ – velocity of flowing cool air, m/s
$V_r$ – volume of test room space, m$^3$
$\rho_w$ – density of water, kg/m$^3$
$C_{pa}$ – specific heat capacity of air, kJ/kg.k
$C_{pw}$ – specific heat capacity of water, kJ/kg.k
$C_{pv}$ – specific heat capacity of vapour, kJ/kg.k
$C_{pma}$ - specific heat capacity of mixture, J/kg.k
$A_p$ – area of filter pad, m$^2$
A – area of roof, wall or floor, m$^2$
$\ell$ – cooling efficiency, %
$Q_i$ – cooling capacity, kW
$T_0$ – outdoor dry bulb temperature, °C
$T_i$ – indoor dry bulb temperature, °C
$T_r$ - conditioned test room temperature, °C
$T_2$ , $T_{wb}$ – wet bulb temperature, °C
$T_1$ , $T_{db}$ – dry bulb temperature, °C
$g_0$ – outdoor air moisture content, g/kg-dry air
$P_g$ – saturation pressure, bar
$g_r$ – indoor air moisture content, g/kg-dry air
$T_{oav}$ – mean outdoor temperature, °C
$T_{rm}$ – mean sol-air temperature, °C
f – decrement factor
ω – specific humidity, kg/kg dry air
ΔT – measured temperature, °C
P – Barometric pressure, bar
$H_1$, $\phi_1$– Humidity at cooler inlet/indoor space,%
$H_2$,$\phi_2$– Humidity at cooler outlet/indoor space,%
COP – Coefficient of Performance
GPH – Evaporation Rate
LCD – Liquid Crystal Display
LDB – Leaving air Dry Bulb, Temperature, °C
IDB – Indoor Dry Bulb Temperature, °C
ODB – Outdoor Dry Bulb Temperature, °C
OWB – Outdoor Wet Bulb Temperature, °C
CFM – Cubic Feet Minutes
SE – Saturation Efficiency, %
WBD – Wet Bulb Depression, °C
COP – Coefficient of Performance
Subscripts
a – air
w – water
v – moist air