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ANALYTICAL MODELS AND DESIGN OF COMBINED DEHUMIDIFICATION AND EVAPORATIVE COOLING SYSTEM FOR MANGO STORAGE IN GHANA

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ABSTRACT

In Ghana, the production, consumption and commercialisation of mangoes are substantially low. The fruits are highly perishable and require storage temperatures of 10-15^{o}C, but most mango producing areas in Ghana are remote, poor and do not have access to electricity and so cannot use conventional cold storage systems. The wet bulb temperatures in these areas are between 14 and 27^{o}C; and relative humidities vary between 16 and 77%, so a combined desiccant dehumidification and evaporative cooling system has been designed and used at these areas. The system consists of the storage chamber, dehumidifier, heat exchanger and saturator. Analytical models of these components have been developed, the components designed and experimental procedure explained in this paper. Three key assumptions were made in solving the model equations with a computer simulation programme, namely no heat transfer through the walls, no water loss by drift and steady flow conditions. The system's performance was found to be dependent on incoming air conditions and water flow rate and its overall performance produced a minimum cooling of 0.31 tons and maximum cooling of 2.00 tons for water flow rates of 0 and 0.252 l/s respectively. The observed storage temperature of 12^{o}C and relative humidity of 90% were found to be conducive for the long term storage of mango.

Keywords: Evaporative cooling, Dehumidification, Analytical models, Mango storage, Ghana

INTRODUCTION

Mangoes are one of the finest fresh fruits in the world, and are higher in vitamin C than citrus fruits, as well as other health benefits [1]. In Ghana, there is a huge potential for mango cultivation, which can make the fruit become a major foreign exchange earner for the country, but production levels are currently far below expectations. For example, the total production in 2007 was only 1,071 metric tonnes [2]. One of the reasons which has been suggested as a disincentive for increased mango production is the lack of accessible storage structures, both on-farm and off-farm. This lack leads to various types of postharvest losses and high storage costs.

Mango farmers in Ghana are mainly poor, and live in remote locations where there is no grid electricity [3]. Conventional methods of mango storage such as controlled atmosphere, modified/controlled atmosphere, thermal quarantine, vapour compression, which are mainly used in developed countries, are too expensive and also not practically feasible for the rural farmer [4]. In [5], one way that has been found to be efficient and economical in reducing temperature and increasing the relative humidity in an enclosure, which has been extensively tried for increasing the shelf life of horticultural produce in some tropical and subtropical countries is evaporative cooling principle. [6] notes that evaporative cooling involves cooling dry air by blowing it across a wet surface; and the conversion of sensible heat to latent heat is the underlying principle of this concept. Systems designed using this principle have the advantages of low energy consumption, easy maintenance, installation and operation, and are environmentally friendly since water and air are the main working fluids.

Although the benefits of evaporative cooling may be easily obtained, its application has performance limitations. Theoretically, the minimum temperature that can be achieved with a very efficient evaporative cooling system is the wet bulb temperature of the location where the system is used. However, climatic data recorded by [7] revealed that average ambient wet bulb temperatures in the transitional zones of Ashanti and Brong Ahafo regions and the Northern Savannah areas of Northern, Upper East and Upper West Regions of Ghana, which are the some of the main mango growing areas, are between 14 and 27°C; and average ambient relative humidity vary between 16 and 77%.

Generally, the storage temperature of matured green mango is 10-15°C [8]. This means that a stand-alone evaporative cooling system cannot achieve the required storage temperature. To achieve the desired storage temperature of mango in the mango growing areas in Ghana, [9] suggests combined dehumidification and evaporative cooling. With this process, the high moisture content of the incoming atmospheric airstream is first dehumidified with a desiccant material, which also lowers the wet bulb temperature.

The objective of this study therefore, was to develop analytical models, design and conduct experimental studies of the combined dehumidification and evaporative cooling storage system.

Description of the Cooling System

The cooling structure consists of four main parts, namely; the storage chamber, dehumidifier, air –to – air heat exchanger and evaporative cooler (or saturator).

A flow diagram of the representation of the air process as on a psychrometric chart (a) and the air flow path (b) is shown in figure 1.

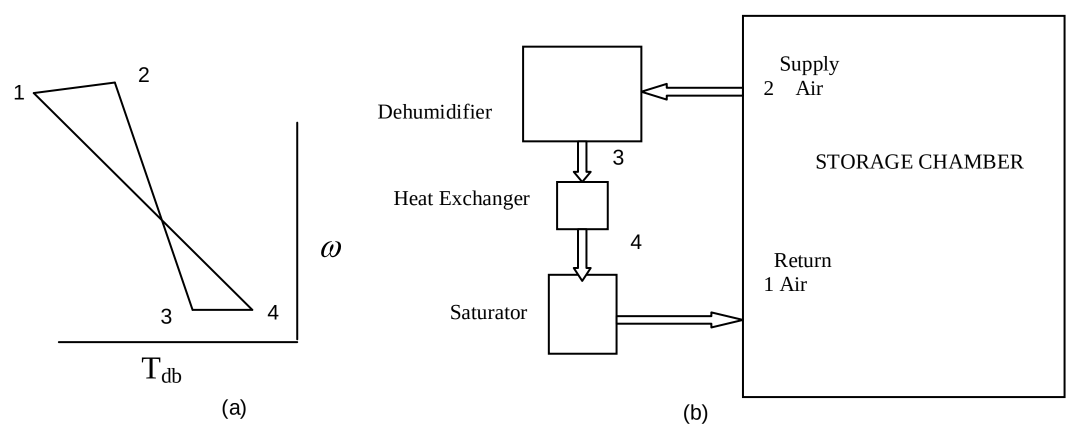


Fig.1. Flow diagram of combined dehumidification with evaporative cooling: (a) schematic of air process as on a psychrometric chart (b) air flow path

Source: [9]

From figure 1, for the (b) part, the air stream at state 2 is passed through a desiccant wheel. Its moisture is partly but significantly adsorbed by the desiccant material and the heat of adsorption elevates its temperature so that a warm and rather dry air stream exits at the state 3. The air stream is then cooled successively in the heat exchanger from state 3 to state 4, and then in an evaporative cooler from state 4 to state 1. At state 1, the air is much cooler and very moist (lower dry bulb temperature and higher relative humidity) and it is then released into the storage chamber containing the mangoes.

Analytical Models of the Components of the Cooling System

Dehumidifer

The dehumidification unit is also called absorber unit in this study. It is analysed using the first law of thermodynamics, and conservation of mass assuming steady flow as in equations (1) and (2).

The first law is written as

$$Q - \dot{W} = \sum m_e \left(h_e - \frac{V^2{}_e}{2} + gz_e\right) - \sum m_i \left(h_i - \frac{V^2{}_i}{2} + gz_i\right) \qquad (1)$$

and

Conservation of mass as

$$\sum \dot{m} = 0 \qquad (2)$$

The effectiveness of dehumidification was used in this study to analyse the performance of the absorber unit. As shown in equation (3), the effectiveness of absorber to remove moisture is defined as the actual moisture removed over the maximum possible change in moisture content.

$$\epsilon = \frac{W_i - W_e}{W_i - W_{e,minimum}} \qquad (3)$$

The maximum possible change in moisture is a property that is defined by the configuration of the absorber when operating under adiabatic conditions. Since the configuration is a counter flow pattern, the vapour pressure of TEG at the top of the absorber is the lowest because it has the lowest temperature and highest concentration when it enters. The air vapour pressure is also the lowest at the top since its temperature is the highest and the humidity the lowest at the exit. As long as the air vapour pressure is higher than the liquid vapour pressure, dehumidification will take place. Therefore the best the unit can perform is to lower the vapour pressure of the air to that of the incoming liquid and once the vapour pressures are equal, dehumidification stops. Therefore the overall effectiveness of the unit can be calculated if it is assumed that the best operation is when dehumidification stops. This means, dehumidification stops when $P_{air,exit,minimum}$ is equal to $P_{liquid,inlet}$ (i.e. equal vapour pressures). The minimum specific humidity (humidity ratio) can be calculated from the vapour pressures desired which corresponds to the TEG vapour pressure at the entrance and is written as shown in equation (4).

$$W = \frac{m_v}{m_a} = \frac{P_v V / R_v T}{P_a V / R_a T} = \frac{P_v / R_v}{P_a / R_a} = 0.622 \frac{P_v}{P_a} = 0.622 \frac{P_v}{P - P_v} \qquad (4)$$

The TEG vapour pressure can be determined from the method described by [10], based on concentration and temperature.

Air –to – Air Heat Exchanger

In the heat exchanger analysis, the "Effectiveness – Number of Transfer Units (NTU) " method was used for rating the heat exchanger. In [11], effectiveness is defined as

$$\varepsilon = \frac{actual\ heat\ transferred}{\begin{array}{c} maximum\ heat\ that\ could\ possibly\ be \\ transferred\ from\ one\ stream\ to\ the\ other \end{array}}.$$

Using the analyses in that text, the effectiveness of the heat exchanger is adopted as

$$\epsilon = \frac{1-\exp\left(-NTU(1-C_r)\right)}{1-C_r\exp\left(-NTU(1-C_r)\right)} \tag{5}$$

Effectiveness is a function of the NTU and the ratio of $C_{min}/C_{max} = C_r$, that is

$$\epsilon = f\left[NTU, \frac{C_{min}}{C_{max}}\right] = f(NTU, C_r) \tag{6}$$

and NTU is defined as

$$NTU = \frac{UA}{C_{min}} \quad \text{and} \tag{7}$$

The heat exchanger was tested for cross-over contamination to ensure it was operating satisfactorily.

Saturator

The saturator, also known as evaporative cooling unit, is used to evaporate moisture into the air and its performance is described by its saturation efficiency. When the temperature of the water that is being evaporated approaches the wet bulb temperature of the incoming air, in the process of adiabatic saturation, equation (8) can be used to evaluate the saturation efficiency of the unit,

$$S.E. = \frac{T_{db,i}-T_{db,e}}{T_{db,i}-T_{wb,i}} \tag{8}$$

If the water being supplied is warmer than the wet bulb temperature of the air entering (which was the case in this study) then the equations in the sections following are valid for a cooling tower operation. A cooling tower was not built for this work. Water was supplied using tap water, which was modelled as a cooling tower to determine the volume. Figure 2 shows a schematic diagram of a counter-flow cooling tower for which the differential volume element was used to evaluate the mass and energy balance. As part of the analysis, the heat transfer through the walls and water loss by drift were ignored and steady flow conditions used.

Therefore, for differential volume element in figure 2 the energy balance is:

$$m_a dh = -[m_w - m_a(W_2 - W_1)]dh_{f,w} + m_a dWh_{f,w} \tag{9}$$

or neglecting $m_a(W_2 - W_1)$ term in the right hand side:

$$m_a dh = -m_w dh_{f,w} + m_a dWh_{f,w} \tag{10}$$

Also, from the volume element the water and air mass balance is

$$-m_w dh_{f,w} = h_c A_v dV(t_w - t) + h_D A_v dV(W_{s,w} - W)h_{fg,w} \quad (11)$$

also $$m_a dW = h_D A_v dV(W_{s,w} - W) \quad (12)$$

substituting $Le = h_g/h_D C_{p,a}$ into equation (11) gives

$$-m_w dh_{f,w} = h_D A_v dV[Le\, C_{p,a}(t_w - t) + (W_{s,w} - W)h_{fg,w}] \quad (13)$$

Combining equations (10), (12) and (13) results

$$\frac{dh}{dW} = Le\, C_{p,a} \frac{(t_w - t)}{(W_{s,w} - W)} + h_{g,w} \quad (14)$$

Approximate h_g by 2501.3 and assume constant $C_{p,a}$:

$$h_{s,w} - h = C_{p,a}(t_w - t) + 2501.3(W_{s,w} - W) \quad (15)$$

Equation (14) becomes

$$\frac{dh}{dW} = Le \frac{(h_{s,w} - h)}{(W_{s,w} - W)} + (h_{g,w} - 2501.3\, Le) \quad (16)$$

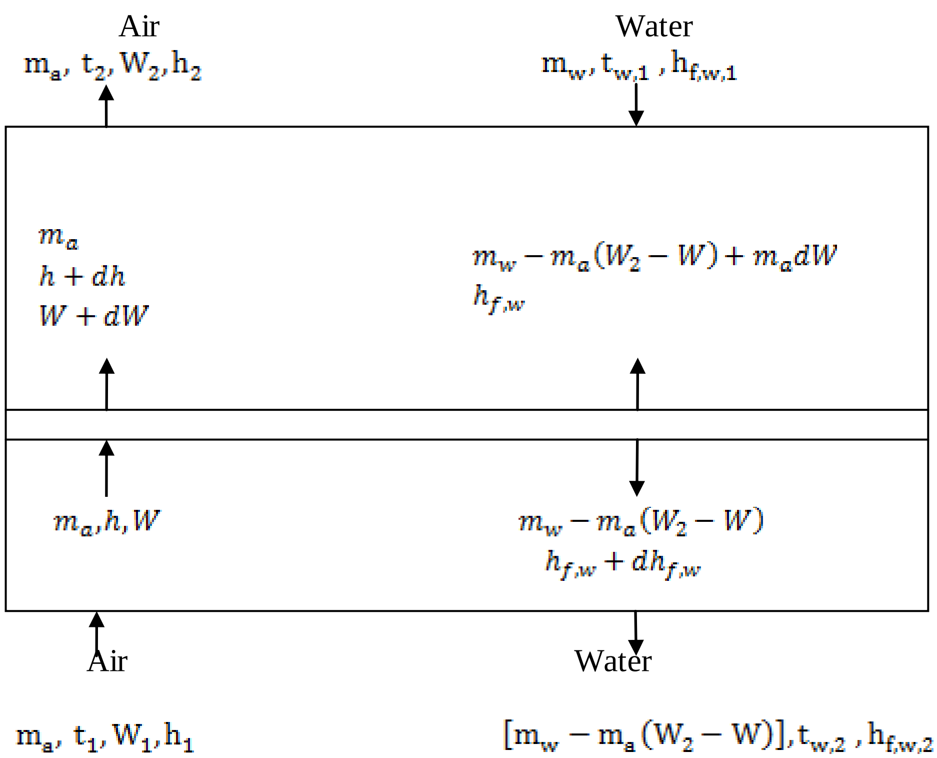


Fig. 2. Counter- flow Cooling Tower Schematic *Source:* [9]

Equation (16) describes the condition line between incoming and exiting air states. This equation was solved on a computer using FORTRAN codes. In the solution, the inlet and exit temperatures, water flow rate and entering air state are assumed to be known. Equation (16) was solved at state one using the inlet air and exiting water states. State two was arbitrarily located along the condition line and the properties of air at this new state can be determined. To calculate the water temperature corresponding to the state two, equation (10) was rewritten as

$$-\Delta T_w = \frac{m_a}{m_w c_{p,w}}\left(\Delta h - \Delta W_{hf,w}\right) \qquad (17)$$

Equation (16) was then solved at state two and the above procedure was continued until the water temperature, calculated from equation (17), reaches the inlet temperature. This procedure also determined the exit air state. The accuracy of this solution depends on the size of incremental changes at the air state when arbitrarily locating state two.

The equation for the entire cooling tower is given by equation (18)

$$h_2 = h_1 + \frac{m_w c_{p,w}}{m_a}\left(T_{w,1} - T_{w,2}\right) + (W_2 - W_1)h_{f,w,2} \qquad (18)$$

This equation was used to calibrate the accuracy of the stepwise solution. Given the exiting air state and the average mass transfer coefficient $h_D A_v$, it is possible to rewrite equation (12) to get equation (19) and hence determine the cooling tower volume required.

$$V = \frac{m_a}{h_D A_v}\int_{W_1}^{W_2} \frac{dW}{W_{s,w} - W} \qquad (19)$$

Equation (19) was solved by multiple application of the trapezoidal rule.

Evaluation of the Cooing System

When evaluating the performance of a specific component or combination of components, the coefficient of performance (COP) is a useful value. The COP is defined as ratio of useful refrigeration effect to the energy input, and is represented in equation (20).

$$COP = \frac{\text{useful refrigeration effect}}{\text{net energy input}} \qquad (20)$$

This is what was used to evaluate the performance of the cooling system.

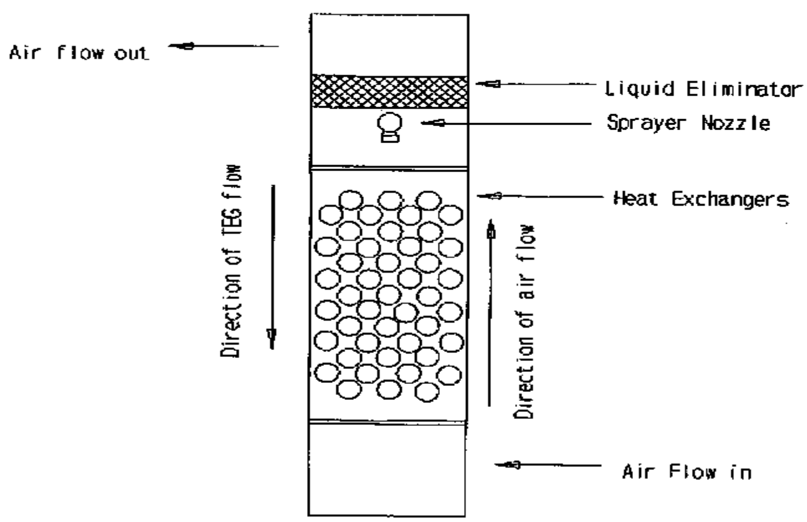
Design of the Cooing System Components

The refrigeration system components in this study were designed and constructed or selected from manufacturers stock. The saturator and absorber units were two units that had to be designed and constructed. Other pieces of equipment required such as the heat exchanger, fans, etc. were purchased and assembled to complete the experimental apparatus.

Dehumidifier

As shown in the schematic diagram of the absorber unit (dehumidifier) in figure 3, air enters at the bottom and flows to the top while the TEG enters at the top and is spread into the air stream and flows over the finned tube heat exchangers. Chilled water from a cooling tower was supplied to the heat exchanger at the bottom. But, in the actual experiments the cooling tower was not built, instead tap water was used and a cooling tower volume estimated from heat transfer in the absorber. The water flowing through the heat exchangers from the bottom to the top, parallel to the air flow, picks up heat and is returned to the cooling tower.

Fig.3. Absorber Unit (Parts and Flow Directions) ***Source:*** **[9]**

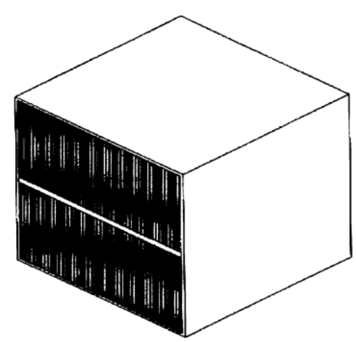


The unit was designed to handle 0.118 m^3s^{-1} to 0.236 m^3s^{-1} of air and have inside flow dimensions set at 0.61 m by 0.2 m. The internal structure of the absorber unit consists of two continuous finned tube heat exchangers. The frame is galvanised steel. The tubes are 1.59 cm copper tubes with 0.06 cm wall thickness. The plate fins are 0.03 cm thick aluminium with a waffle pattern at 3.9 fins per centimetre.

Air –to – Air Heat Exchanger

Combined with the evaporative cooler, the heat exchanger provides indirect cooling to the process air stream. The heat exchanger was designed to handle air flows up to 0.708 m^3s^{-1} with an effectiveness of about 0.8. The physical structure data and appearance is shown in figure 4. It was constructed from different types of materials. The casing was constructed from galvanised steel, the heat transfer plates from aluminium, and nylon was used as a sealer. The materials do not readily deteriorate and the components are biologically inert thus do not promote bacteria growth.

Fig. 4. Heat exchanger physical data ***Source:*** **[9]**



Dimension: 595 × 595× 610 mm
Weight: 34 kg
Plate Spacing: 3 mm
Plate area: 59 m^2

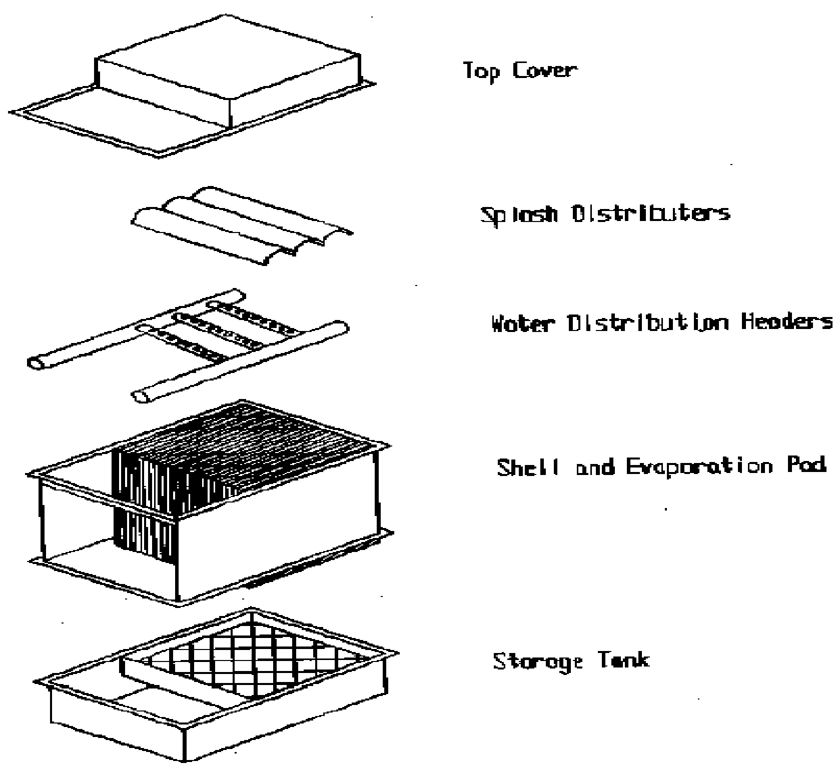
Evaporative Cooling Unit

The evaporative cooling unit is the primary device used in this experiment and was the first stage of cooling in the system. The unit consists of several components such as reservoir, pump, distribution header, splash cover and evaporative pad (or media). The water was collected in the reservoir where it was pumped to the distribution header and spread over the media. The water then flows through the media while air passes perpendicular to the water flow. The air picks up moisture as it passes through the media and was saturated to 95-100%.

The media was sized to handle a maximum flow of 0.47 m^3s^{-1} with estimated saturation effectiveness of up to 95%. The media is like an honey comb structure with the flutes angled at 45^o. These flutes were situated such that they were sloping downward into the direction of flow. This allows falling water a longer residence time in the media with the result of an increase in effectiveness, while preventing water carry over through the media.

As shown in figure 5, water was supplied to the top of the media through a water distribution system. The distribution header is a lattice of piping that distributes water over the top of the media. Holes are drilled in the top of the top distribution header and a splash cover is fastened above the media to break up the water jet. The lattice of piping was designed to allow even distribution over a wide range of flows.

Fig.5. Evaporative Cooling Unit Components *Source:* [9]



Regenerator Unit

The primary function of the regenerator was to remove the moisture from the TEG after each test. A detailed study of the operation and efficiency of this component was not made except that it adequately regenerated the TEG.

Experimental Procedure

The components were oriented as shown in figure 1 such that dehumidification and cooling of the air was obtained. Humidity measurements of the incoming and exiting air were taken. Dry bulb measurements were taken before and after the dehumidifier, heat exchanger and evaporative cooler. Wet bulb temperatures were taken after the evaporative cooler and on both sides of the heat exchanger. Water and TEG temperatures were taken entering and exiting the dehumidifier unit. Pressure drop readings were taken before and after the dehumidifier, heat exchanger and evaporative cooler.

Initially the concentration of the TEG was determined using the refractive index. If the concentration was 98% ± 2% the experiment was started. The fan was started to circulate air through the system. The pump to the saturator was turned on and the water flow was adjusted to 0.05 ls^{-1}. The water flow through the absorber was set at 0.0, 0.063, 0.126, 0.189 or 0.252 ls^{-1}. The system was allowed to run until the water temperature in the saturator reached a temperature within 1^{o}C of the incoming wet bulb temperature of the air. The water entering the absorber was adjusted to this temperature by balancing hot and cold water from the tap until the required water temperature was reached. The system was then allowed to run approximately one and half hours to develop and ensure steady readings.

At this point a data logger was started to record temperatures, pressures and humidity at 5 minutes intervals. After 3 readings without TEG flowing, the TEG pump was started. The TEG was allowed to flow until dehumidification had discontinued. Dehumidification was considered to have reached its limit when the exiting temperature and humidity readings were unchanged for 15 minutes. Total run time of the system was approximately 3 hours. After the system was turned off the TEG was pumped to the regenerator. The regeneration of the TEG usually required 3 hours of operation. The TEG was then cooled to room temperature for reuse.

CONCLUSION

Analytical models were developed to enable a comprehensive and accurate design of the refrigeration system. Mathematical modelling and simulation approach to design offers horticultural and storage or preservation industries and researchers a rapid means of assessing cold storage system modifications without expensive trial and error experimentation, and this was the reason for using analytical models to first analyse the system components.

The analytical models used to analyse the performance of each of the components constructed and tested were used together with the storage requirements of mangoes and climatic conditions of the savannah and transitional zones, which are some of the main mango growing areas in Ghana.

The cold storage system’s performance was found to be dependent on the incoming air conditions and the water flow rate. Ambient air temperatures varied from 20.0°C to 36°C dry bulb temperatures and 14.0°C to 25.0°C wet bulb temperatures. In this range the units produced a maximum of 2 tons of cooling with a water supply of 0.252 l/s and a minimum of 0.31 tons with a water supply of 0 l/s. The system’s overall performance varied for water supplies of 0, 0.63, 1.26, 1.89 and 0.252 l/s from 0.31 to 0.96, 0.66 to 1.29, 1.02 to 1.72, 0.96 to 1.81 and 1.31 to 2 tons respectively. The cooling was affected by the systems dependence on the inlet conditions thus resulting in a range of cooling performances for each flow rate.

Freshly harvested mango fruits were used for the validation and the observed average temperature in the cooler was found to be approximately 12^oC and the corresponding average relative humidity was 90%. The observed storage temperature and relative humidity were found to be within the range of the recommended storage conditions of mangoes. The fruits in the storage structure still looked green-yellow, firmly fibrous and same sweet-mild flavour after four weeks in the storage system.

Three key assumptions were made in solving the models with a simulation programme (the FORTRAN codes): no heat transfer through the walls, no water loss by drift and steady flow conditions. The accuracy of the programme depended on the size of the incremental changes in the air state when arbitrarily locating state two.

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NOMENCLATURE/NOTATIONS

$\dot{m}_a$	air flow rate (kgs^{-1}), $\dot{m}_a = \frac{V}{\upsilon}$
$\dot{m}_f$	rate of evaporation ($kgmin^{-1}$)
$\dot{m}_w$	water vapour removed ($kgmin^{-1}$)
$h_D A_v$	mass transfer coefficient (ms^{-1})
A	flow area (m^2)
COP	coefficient of performance
C_p	specific heat capacity of air ($kJkg^{-1o}C^{-1}$)
h	specific enthalpy ($kJkg^{-1}$dry air)
h_{fg}, h_g, h_f	Enthalpy ($kJkg^{-1}$),
Le	Lewis number
$\dot{m}$	mass flow rate ($kg\ s^{-1}$)
NTU	number of transfer units
P	pressure ($kgm^{-1}s^{-2}$)
R	the gas constant ($J\ K^{-1}\ mol^{-1}$)
Q	heat transfer (kJs^{-1})
S.E.	saturation efficiency (%)
T/t	Temperature (^0C)
T_{db}	dry bulb temperature (^0C)
TEG	triethylene glycol
U	overall heat transfer coefficient
u	velocity (ms^{-1})
V	volume flow rate of air (m^3s^{-1})
$\dot{W}$	work done (J)
x	concentration of TEG
z	height (m)
ρ	density (kgm^{-3})
υ	specific volume of air (m^3kg^{-1} dry air)
ω/W	humidity ratio (kg H_2O kg^{-1} dry air)
ϵ	heat exchanger effectiveness (%)

Subscripts

1,2,3,4	position/state
A	Air state point
a	referencing air
c	referencing cold path
e	exit
f	enthalpy of saturated liquid
g	enthalpy of saturated vapour
g	referencing TEG
h	referencing hot path
i	inlet
s	base
v	vapour
w	referencing water