PRE-FEASIBILITY ANALYSIS OF INCORPORATING NON-CONCENTRATING SOLAR THERMAL ENERGY SYSTEMS IN THE KENYAN TEA INDUSTRY

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Abstract

This paper emerged in the context of reducing the carbon footprint of the Kenya tea industry. The tea factories consume thermal energy at low temperatures (< 35 °C) during the withering process and at medium temperatures (90 °C to150 °C) during the drying process. Currently firewood is used to produce steam to meet the thermal energy requirement of both the withering and the drying processes. This paper investigates the viability of displacing some of the steam used for the withering process by considering the use of flat plate solar air collectors with rock bed storage. Using Arroket tea factory in Kenya as case study, the results show that a solar air collector field with an area of 55.2 m^2 and a rock bed storage size of 15 m^3 could be used to meet 24.2 % of the annual thermal energy requirement of one trough in a tea factory processing black tea. The corresponding LCOH and carbon offset are respectively 0.0353 Euro/KWh and 33 tonnes CO_2-equivalent per year.

Keywords: Tea; Thermal; Solar; Withering.

1. Introduction

Tea production is an important economic activity in the agricultural sector of Kenya. While Kenya is the third largest tea producer in the world after India and Sri Lanka, about 95 % of the tea processed in Kenya is exported [1]. Many Kenyans earn their living from tea related activities such as cultivation, processing and sale of tea. The tea sector in Kenya is contributing about 4 % of the country's GDP [2].

The tea growing regions in Kenya are endowed with a tropical climate, volcanic red soils and distributed rainfalls which favour the production of quality tea throughout the year [2]. Tea harvesting occurs all-year round and consequently tea processing factories in the country also operate almost every day of the year. Most tea factories rely on wood fuel and electricity from the grid to meet their thermal and electrical energy demands respectively. Thermal energy represents about 85 % of the energy demand in a tea factory [3] and this makes the tea industry one of the main industries consuming wood fuel in Kenya. Wood fuel is burned in the boilers located on factory sites for steam generation purposes and consequently the factories emit Greenhouse Gases (GHG) to the atmosphere. Because of global warming, GHG emissions from the factories constitute a concern to the tea industry in Kenya, in the context of sustainable development. Some tea factories in the country grow their own tree plantations to produce sustainable wood fuels. To accelerate sustainable practices within the tea industry, tea factories are encouraged to use energy more efficiently and seek alternative and clean energy sources such as solar and wind. This paper investigates the viability of using non-concentrating solar thermal energy in the Kenyan tea industry.

Thermal energy consumption in a tea factory depends on the processing method and the type of tea being produced. Black tea is the main type of tea produced in Kenya, and the common processing method used is the Cutting Tearing and Curling (CTC) method. When processing black tea using the CTC method, low temperature air with drying capability (< 35 °C) is used during a stage called withering, to reduce the moisture content of fresh leaves from about 80 % wet basis (wb) to about 66 % wb [4]. In most Kenya tea factories, air has sufficient drying capability when the wet-bulb depression[1] (WBD) is above 3 °C. After the withering stage, the tea leaves are cut, teared and curled between a series of rollers, hence the name CTC method. After the CTC stage, the curled tea particles go

[1] Difference between dry and wet bulb temperatures

through a fermentation stage and are then dried using hot air at a temperature around 150 °C.

The dried tea particles,with moisture content of about 4 % wb are sorted, graded and packed. All the thermal energy requirement in a tea factory is therefore in the form of warm and hot air: low temperature warm air for the withering stage and medium temperature hot air for the drying stage. Steam is sometimes needed at the fermentation stage to keep the operating temperature constant. The processing line of black tea is summarized in figure 1.

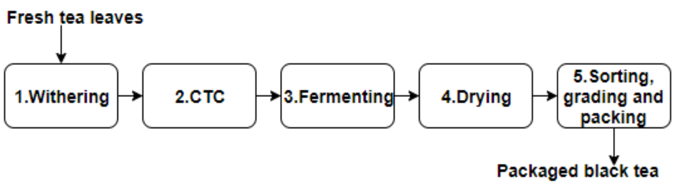

Fig. 1. Processing line of black tea

The focus of this paper is on the low temperature thermal energy requirement during the withering stage. In most Kenyan tea factories, ambient air is used directly for withering during the day when its WBD is above 3 °C [5]. When this is not the case, often at night, steam is used to raise ambient air temperature by sucking air through heat exchangers. Because the maximum temperature of the air used at the withering stage is 35 °C, steam is not necessarily needed. This paper considers the use of non-concentrating solar thermal technology to reduce or eliminate the need for steam for the withering stage in black tea processing. Arroket tea factory in Kenya is used as a case study.

2. Withering thermal energy demand and proposed solar thermal systems

2.1. Arroket tea factory: location, solar resource assessment and meteorological data

Arroket tea factory, which is owned by Sotik Tea Estates, is located west of the Rift Valley in Kenya, at a latitude of - 0°37'8.15" with a longitude of 35°4'8.29". The factory operates all year-round and the total annual duration of withering is estimated to be 5387 h.

The annual global horizontal solar irradiation (GHI) at the factory location in 2015 was 2180.8 kWh / m^2 [6]. As explained earlier the demand for external thermal energy in the withering process depends on the WBD of ambient air, in other words, the demand depends on meteorological data such as the relative humidity, the dew point and the atmospheric pressure. Based on the 2015 meteorological data at the factory location [6] , the number of hours for which the WBD of ambient air is below 3 °C represents about 43 % of the total annual duration of withering. Figure 2 displays the variation per hour of the WDB of ambient air over a typical wet and dry seasons day at the factory location. During the wet season, the WBD of ambient air is low for most periods of the day as shown in figure 2. But during the dry season, the WBD of ambient air during daytime is usually above the minimum required WBD and steam is only required during nightime.

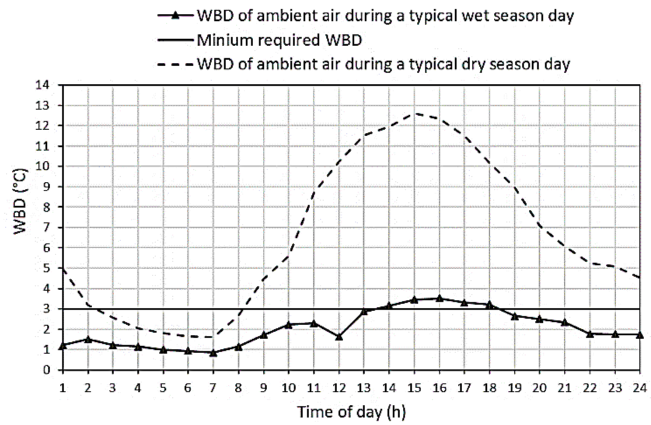

Fig. 2. Variation of the WBD of ambient air over a typical wet and dry season day

2.2. Withering description and thermal energy demand

Withering consists of blowing air through freshly plucked tea leaves which are spread on troughs, as shown in figure 3. Each trough is equipped with a fan and the outlet of a duct coming from a steam-air heat exchanger is located in front of the fan.When ambient air needs to be heated, steam is circulated through the heat exchanger and since the duct in front of the fan is connected to the heat exchanger, hot air coming from the heat

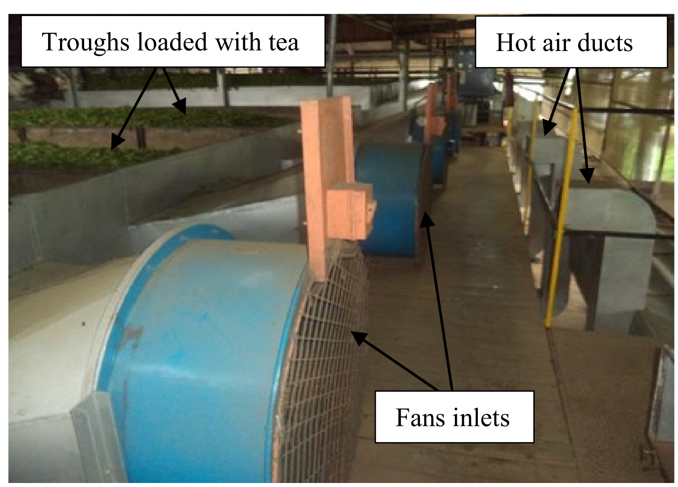

Fig. 1. Withering system set-up

exchanger side is mixed with ambient air to raise its temperature. A manual damper is used to control the percentage of air flow coming from the heat exchanger side. The main purpose of the damper is to ensure that the temperature of the air going to the withering trough does not exceed 35 °C. Higher temperatures would negatively impact the quality of the produced tea. All the troughs at the factory have the same the design and the air flow going into each trough when external energy is required is a mix of hot air and cold air. Because of the similarity between the troughs, only one trough is analysed in this study.

Table 1 summarises the parameters used to estimate the annual energy consumption of one withering trough. It is assumed that when the WBD of ambient air is less than 3 °C, its dry bulb temperature should be raised by 6 °C before it is used in the withering process. The wet bulb temperature also rises when ambient air is being heated. In fact, Keegel [7] noted that the wet bulb temperature rises by 1 °F for every 3 to 4 °F increase in the dry bulb temperature. He recognised that it is impossible to set a hard rule for how much the dry bulb temperature should be increased because of the many variables involved when withering tea. As a recommendation, he proposed an increase of about 10 °F in ambient air's dry bulb temperature. A temperature difference of 10 °F is approximately equivalent to a temperature difference of 5.6 °C. Raising the dry bulb temperature of ambient air by 6 °C is therefore a reasonable conservative approach.

Parameters	Values	Units
Withering of fresh tea leaves capacity of one trough	1 000	kg
Air flow per kg of fresh leaves	0.01 [3]	$m^3/(s.kg)$
Required rise in dry bulb temperature when WBD of ambient air is less than 3 °C	6	°C
Average withering time	8	h

Table 1. Parameters used to determine the thermal energy requirement of one withering trough

For every hour, when ambient air needs to be heated, the energy demand Q_{demand} is calculated as

$$Q_{\text{demand}} = \dot{m}_{\text{total.air}} \times c_{\text{p}} \times \Delta T_{\text{db}} \times \Delta t \quad \text{(Wh)} \qquad (1)$$

where $\dot{m}_{\text{total.air}}$ is the total mass flow of air going to the withering trough, c_{p} is the specific heat capacity of air and Δt is 1h. As per Table 1, $\Delta T_{\text{db}} = 6$ °C. The monthly energy demand for one trough is calculated and shown in figure 4, based on the energy demand equation and the assumptions that withering is done in the trough twice per day (8 h each), every day of the year except on Sundays when the factory is not in operation.

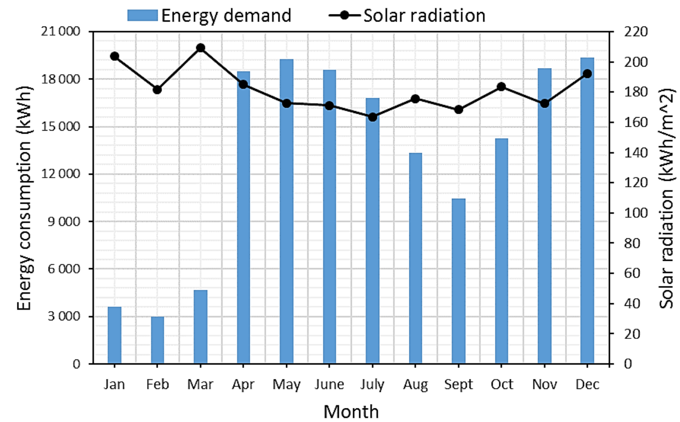

Fig. 4. Estimated monthly withering energy consumption and GHI at the factory in 2015

In figure 4, the months of high energy consumption coincide with the rainy season in Kenya and it is also the period where large quantity of fresh tea leaves are harvested. There is less energy consumed in January and February, which are periods of dry season in Kenya. During the dry season, tea harvest is low and the WBD of ambient air is usually above the required threshold. It can be observed from figure 4 that the solar radiation is high during the months of low energy consumption and low during the months of high energy consumption. Based on these observations, the required storage in this study would be similar to a seasonal storage [8].

2.3. Proposed method for the integration of solar systems in the withering process

The thermal energy demand in a tea factory is in the form of hot air so the first decision when considering solar energy would be the use of solar air collectors. In fact, most tea factories [9] which, with the aim of reducing their coal fuel consumption, have considered the use of solar air collectors without storage to pre-heat the hot air required for both the withering and drying stages. Solar air collectors are also considered for this study, with focus specifically on the withering process. Figures 5 shows the considered solar air heating system. This project went further by considering the use of thermal storage in order to have solar heat available for night operations as well. In fact, thermal energy is mostly needed for the withering process during night-times. In the proposed system, ambient air temperature is raised through

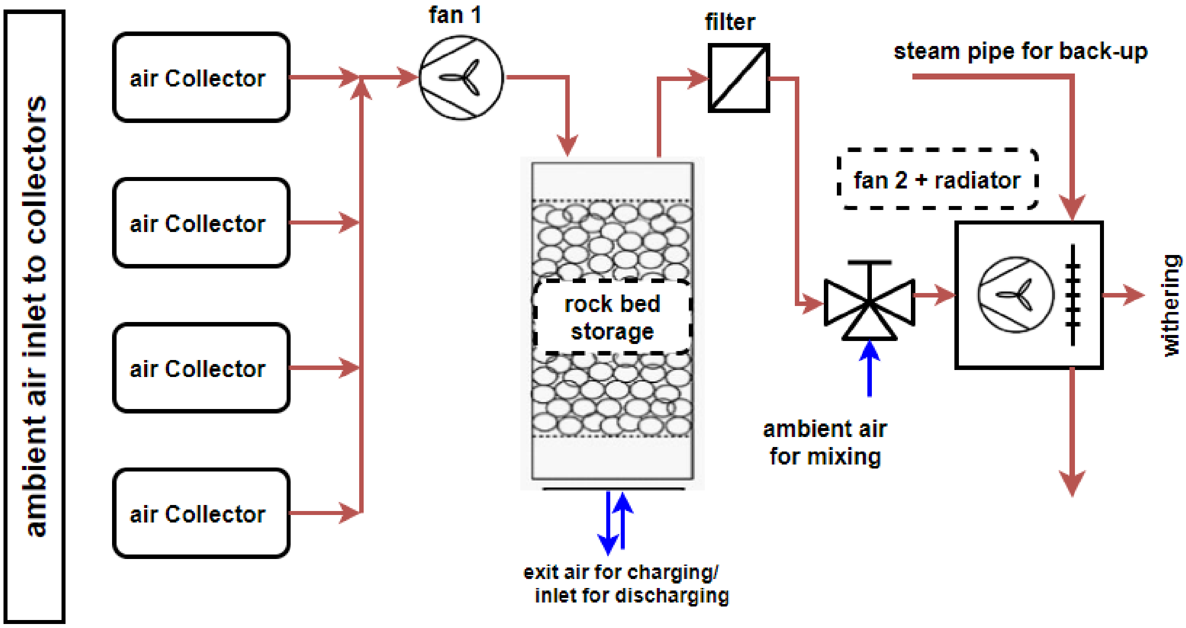

Fig. 5. Schematic diagram of the solar air system

solar air collectors during the day. The hot air is pushed through a rock bed storage where thermal energy is stored. When external heat is required in the withering process, ambient air is sucked through the rock bed storage and mixed with cold ambient air to produce the desired air temperature and WBD.

3. Theoretical design

The theoretical calculations performed in this study can be grouped into four main categories: the collector thermal outputs system, the storage system, the ducting system and the electrical energy input system.

3.1. Collector thermal output

The thermal output of the air collector is calculated based on the Hottel-Whillier-Bliss equation [10] which is expressed as

$$\dot{Q}_u = A_c F_R [S - U_L (T_{pm} - T_a)] \tag{2}$$

where A_c is the collector aperture area, F_R is the heat removal factor, S is the effective absorbed solar radiation, U_L is the overall heat loss coefficient from the collector plate. T_{pm} is the mean plate temperature and T_a is the ambient air temperature.

The effective solar radiation absorbed and the overall heat loss coefficient of the air collector were calculated as recommended in Duffie and Beckman [8]. The heat removal factor which is known to be the ratio of the actual useful energy gained from the collector to the maximum possible energy that could be gained based of the absorbed solar radiation, is evaluated based on the method described in detail by Karim and Hawlader [11].

The temperature of the air at the outlet of the collector is calculated based on equation (2) and the relationship:

$$\dot{Q}_u = \dot{m} c_p (T_o - T_i) \tag{3}$$

$\dot{m}$ is the mass flow rate of air through the collector, c_p is the specific heat constant of the fluid, T_i and T_o are respectively the inlet and outlet temperature of the fluid.

Given that the heat removal factor and the overall heat loss coefficient are a function of the mean plate temperature, an iterative process is used in the numerical modelling of the collector thermal output.

3.2 Rock bed storage

The rock bed storage is modeled by dividing the storage into n identical control volumes (cv) and an energy balance is performed across each cv. A node is located at the centre of each cv. The energy balance equation used in this study for the rock bed storage is based on the infinite NTU model proposed by Hughes et al (1976) as referenced in Duffie and Beckman [8]. The NTU is evaluated as described in [8]:

$$NTU = \frac{h_v A L}{(\dot{m} c_p)_f} \tag{4}$$

$$h_v = 650 \left(\frac{G_o}{d}\right)^{0.7} \tag{5}$$

Where h_v is the volumetric heat transfer coefficient, A is the cross- sectional area of the storage, L is the height of the storage, G_o is the mass velocity and d is the particle diameter.

Duffie and Beckman [8] state that the infinite NTU model could be use to predict the long-term performance of packed bed storage when the NTU is above 25 or even when the NTU is as low as 10.

From equations (4) and (5), it can be deduced that the minimum *NTU* would occur at the maximum mass flow rate and the minimum storage volume. An initial estimation of the *NTU* calculated using the maximum expected flow rate through the storage and a minimum storage size of 2 m^3 as inputs gives an *NTU* of 24. As per the recommendation in [8], this confirms that the infinite NTU model can be used in this study.

As with any thermal storage model, the level of stratification obtained when using the infinite NTU model depends on the number of nodes used. An experimental study done in [12] shows that a good thermocline agreement could be obtained with experiment results when the infinite NTU model with 10 nodes is used. In this pre-feasibility study, 10 nodes are used to model the rock bed storage.

3.3 Thermal losses in ducts and electrical energy input

3.3.1 Thermal losses in ducts

The thermal losses in the ducts are evaluated by considering the convection heat transfer between the working fluid and the duct inner surfaces, the heat conduction through the duct walls and insulation, the natural convection and radiation from the duct outside surfaces.

3.3.2 Electrical energy input

The electrical energy consumed by the fan in the solar system depends on the pressure drops throughout the system.

The main pressure drops are expected across the collectors and the rock bed storage. The pressure drop across each collector is calculated based on the graph of pressure drop vs flow rate provided by the manufacturer. The pressure drop across the rock bed storage is evaluated using the correlation provided by Dunkle and Ellul (1972) as referenced in Duffie and Beckman [8].

The pressure drops due to friction inside the pipes are also evaluated in this study.

4. Simulation parameters and methodology

The theoretical equations in section 3 are coded in the computer software Matlab.

4.1 Technical data

Finned solar air collectors are used in this pre-feasibility study and their technical data is summarised in table 2.

Parameters	**Values**	**Units**
Length	2500	mm
Width	1003	mm
Height	175	mm
Aperture Area	2.3	m^2
Absorber sheet	0.6	mm
Glass cover thickness	4	mm
Insulation thicknesses	60 (back) and 20 (on the sides)	mm
Insulation thermal conductivity	0.04	W/(m.K)
Maximum flow rate	100	$m^3/(h.m^2)$
Minimum flow rate	30	$m^3/(h.m^2)$

Table 2. Technical parameters of the air collector [13]

4.2 Flow rates

The fan in the solar air system is to be powered by solar PV panels and a PWM[2] controller. The role of the PWM is to ensure that the speed of the fan changes with the output power of the PV panels [14]. Since the output of PV panels varies with solar radiation, the air flow rate through the collectors will vary as well with solar radiation.

An experimental study done on an air collector having a fan powered by a PV panel and PWM controller showed a linear relationship between the air flow rate through the collector and the available solar radiation and an increased in efficiency for most part of the day [15].

Based on the findings in [15], the following relationship is used to estimate the flow rates $\dot{V}$ through the collectors in this study:

$$\dot{V} = aI \tag{6}$$

$$a = \frac{\dot{V}_{\max}}{I_{\max}} \tag{7}$$

Where $\dot{V}_{\max}$ is the maximum flow rate as per the collector's manufacturer catalogue and $I_{\max}$ is the maximum hourly solar

[2] PWM: Pulse-Width Modulation

radiation in the year at the factory location. In practice, the PV panels and the fan can be sized accordingly.

4.3 Methodology

The solar air heating system was simulated with the following 8 different effective collectors areas: 18.4 m^2, 36.8 m^2, 55.2 m^2, 73.6 m^2, 92 m^2, 110.4 m^2, 128.8 m^2 and 138 m^2.

To visualise how the solar fraction changes when the storage size is increased, a parametric study is performed. For a given collector area, 12 simulations with different storage sizes were performed and the solar fractions are plotted. The solar fraction represents the amount of useful thermal energy delivered by the solar system to the withering process, as a percentage of the total thermal energy consumed by the withering process.

The levelized costs of heat (LCOH) are calculated afterwards and plotted against collector areas and storage sizes. The energy savings and carbon offsets are then calculated for the point of minimum LCOH. The LCOH in this study is calculated as recommended in [16]:

$$LCOH = \frac{C_0 + \sum_{i=1}^{N} C_i.(1+r)^{-i}}{\sum_{i=1}^{N} E_i.(1+r)^{-i}} \tag{8}$$

Where C_0 is the total capital cost of the solar system. C_i and E_i are respectively the operation & maintenance costs and the delivered thermal energy from the system in the year *i.* The discount rate r is assumed to be 10 % and the lifespan N of the system is assumed to be 20 years.

The specific costs used for the air collector and rock bed storage are respectively 50 Euro/m^2 [17] and 8.66 Euro/kWh[18].The current electricity tariff at the factory is 8 KES/KWh [19] and based on the conversation rate of 1 Euro equals 118.88 KES, the tariff could be expressed as 0.0673 Euro/KWh.

5. Results and discussion

Figure 6 shows how the solar fraction changes as the storage size increases for the same collector field. As the collector area increases, the solar fraction does not improve much for smaller storage sizes (< 5 m^3). Such behaviour of the system could be explained by the fact that for small storage sizes, the storage is the limiting element. It can only store a certain amount of energy even if more solar energy is collected. From the storage size of about 60 m^3, the differences between the solar fractions for all the collector areas become more relevant. For each collector area, the solar fraction increases with the storage size until it appears to reach an asymptote after a certain storage size. The collector area becomes the limiting parameter of the system in the latter scenario. Given that storages are expensive features of solar heating systems, it would be desirable to know which combination of collector area and storage size provides the minimum LCOH. The LCOHs for all the combinations of collector area and storage size considered in this study are plotted in figure 7. For a fixed collector area, the LCOH displays a local minimum value at a certain storage size. As the collector area increases, the storage size for which the local minimum LCOH is observed, also increases. For a fixed storage size, the LCOH seems to display similar profile as in the case of a fixed collector area.

From the considered combination of collector areas and storage sizes, the value of the absolute minimum LCOH is 0.0353 Euro / KWh and it is obtained with a collector area of 55.2 m^2 and storage size of 15 m^3. The results obtained when considering the point of minimum LCOH as a case study are summarized in table 3.

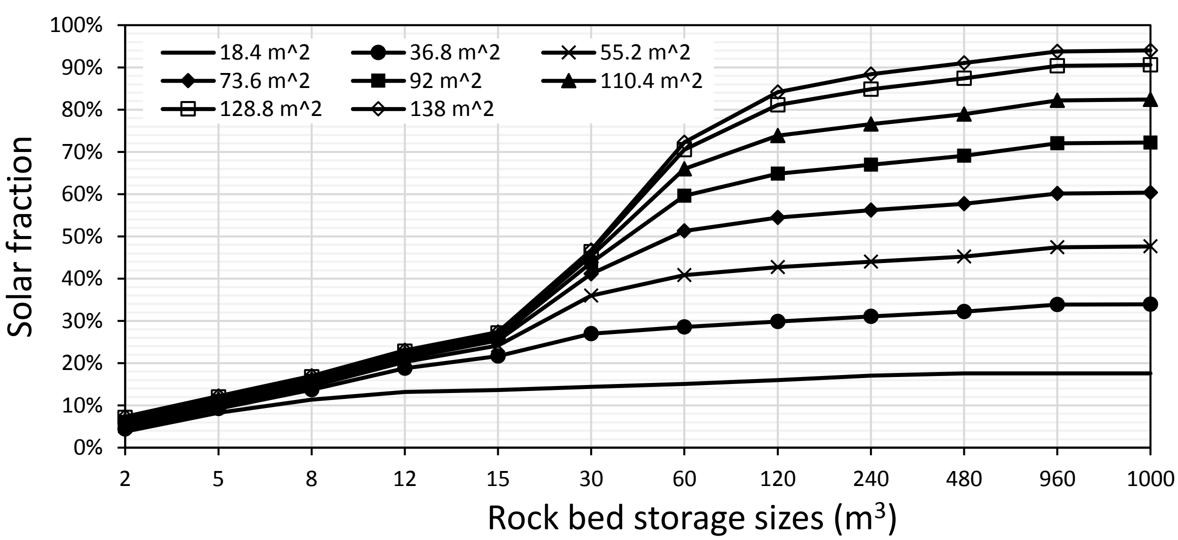

Fig. 6. Plot of solar fraction vs storage size for 8 different collector areas

Parameters	Values	Units
Collector area	55.2	m^2
Rock bed storage size	15	m^3
Delivered thermal energy from solar system	39.64	MWh_{th}
Equivalent thermal energy savings[3]	84.45	MWh_{th}
Solar fraction	24.2	%
LCOH	0.0353	Euro/KWh
CO_2-equivalent offset	33	tonnes

Table 3. Results obtained from the annual simulation displaying the minimum LCOH

In 2015, the factory spent about 1 KES/KWh_{th} [20] on primary wood energy and it is equivalent to 0.0085 Euro/KWh_{th}. Based on the cost of primary wood energy, the thermal energy savings and the estimated financial parameters in table 4, the payback period is about 14 years and an IRR[4] of 4.9 %.

Parameters	Values	Units
Capital cost	11 293	Euro
Payment method (equity)	100	%
Discount rate	10	%
Escalation rate of electricity price	3.5 [20]	%
Escalation rate of wood energy price	6.3 [21]	%
Maintenance cost as percentage of capital cost	1	%
Tax rate	30 [22]	%

Table 4. Parameters used to calculate the financial indicators

When considering a scenario of 30 % of subsidy on the capital cost, the payback period drops to 10.8 years and the IRR is 8.66 %. However, there is currently no indication of subsidies for such solar thermal projects in Kenya.

[3] Useful thermal energy is 47% of input woodfuel thermal energy [23]

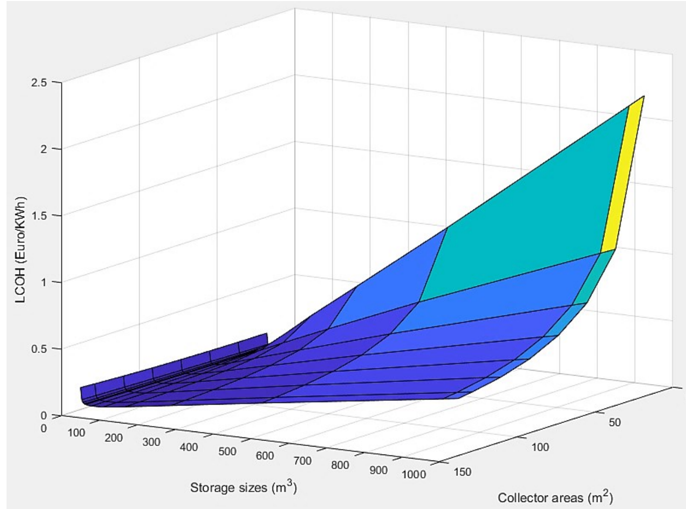

Fig. 7. Plot of LCOH vs Collector areas and storage sizes

6. Conclusion

When processing black tea, thermal energy is consumed mainly during the withering and drying stages. Thermal energy is consumed at a low temperature during withering while drying is a medium-temperature thermal energy consuming process. Most Kenyan tea factories currently use steam for both withering and drying. In the context of reducing the carbon footprint in the Kenyan tea industry, this study has proposed the use of flat plate solar air collectors with rock bed storage to reduce the steam consumption in the withering process.

A numerical model of the proposed system was built in Matlab. Based on the annual thermal energy demand of one withering trough, simulations were performed using a combination of different collectors' areas (from 18.4 to 138 m^2) and storage sizes (from 2 to 1000 m^3). The minimum LCOH is 0.0353 Euro / KWh and it is obtained for a solar collector area of 55.2 m^2 with a storage size of 15 m^3. The corresponding solar fraction and carbon offset are respectively 24.2 % and 33 tonnes CO_2-equivalent per year. Financing such project with 100 % equity gives a payback period of about 14 years and an IRR of 4.9 %. The financial indicators of the viability of the project could be improved under suitable subsidy schemes. Additional incomes could also be obtained by selling carbons credits from the project.

[4] IRR: Internal Rate of Return

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