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In Chapter: 3, we have discussed characterization (proximate analysis, ultimate analysis and higher heating value) of ten locally available biomass samples and performance testing of a WBG - 10 downdraft biomass gasifier. It had been observed that *Dalbargia sissoo* has highest fixed carbon (15.60%). *Bambusa tulda* has second highest fixed carbon (15.20 %) and *Ficus lepidosa* had minimum fixed carbon (12.20 %). Ash value of *Dalbargia sissoo* was minimum 4.4% and *Ficus lepidosa* was maximum 5.8% respectively. It was found that *Psidium guajava* had the highest calorific value (18.403 MJ kg\(^{-1}\)) and *Ficus lepidosa* has the lowest (15.952 MJ kg\(^{-1}\)) among the tested samples. Calorific value of *Bambusa tulda* was (18.401 MJ kg\(^{-1}\)) and *Camellia sinensis* was (18.400 MJ kg\(^{-1}\)). The gasifier performed satisfactorily with uprooted tea shrub (*Camellia sinensis*) and producer gas calorific value was (4.5 MJ m\(^{3}\)) at air fuel equivalence ratio for gasification (0.27) with uprooted tea shrubs as a gasification feedstock.

In Chapter: 4 we had considered black tea drying experiment and drying kinetics modelling with producer gas generated from a mixture of *Camellia sinensis*, *Bambusa tulda* and *Psidium guajava* as gasification feed stock in equal proportion. An improved producer gas burner was redesigned for appropriate mixing of producer gas and air for combustion. At air fuel equivalence ratio (A: F = 1:1), the producer gas burner had best thermal efficiency (57%) as obtained by water boiling test. Tea drying modelling results revealed that Modified Page model was the best fit for local variety black tea while using producer gas combustion product mixed with air as a drying medium. The specific energy consumption per kilogram of made was obtained as 25.50 MJ kg\(^{-1}\) of made tea.

In the Chapter: 5, we will discuss on local solar thermal energy resources, efficient conversion technology (Solar air heater) and its performance studies. Based on previous black tea drying experimental data with producer gas, an analysis will be made for black tea drying possible hybridization of solar air heater with producer gas energy. Therefore, the details of the analytical procedure for hybridization will be
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discussed in this chapter. Further, the economics of renewable thermal energy application in tea drying is also analyzed through a standard procedure.

5.1 Analytical procedure of estimating component shares of solar biomass hybrid renewable energy for black tea drying

The hybrid black tea drying was considered by mixing hot air from solar air heater with combustion products of producer gas. The conceptual layout of solar biomass hybrid thermal energy application for black tea drying has been presented in Fig. 5.1:

![Solar-biomass energy hybridization scheme for tea drying](image)

where, $Q_{ST}$ is thermal energy flow rate from solar air heater, $Q_{BT}$ is biomass thermal energy flow rate from biomass gasifier and $Q_{HD}$ is hybrid drying energy flow rate from the combined system for black tea drying. Now thermal energy available from biomass through gasification and solar air heater is estimated from Eq. (5.1) and Eq. (5.2) below:

$$Q_{BT} = m_{bm} \times \eta_{gasification} \times CV_{bm} \times \eta_{combustion} \quad (5.1)$$

$$Q_{ST} = I \times A \times \eta_{thermo\_hydraulic} \quad (5.2)$$

where, $m_{bm}$ is biomass consumption rate (kg h$^{-1}$) of gasifier, $\eta_{gasification}$ is gasification efficiency (65%) of biomass gasifier, $CV_{bm}$ is calorific value (18.40) MJ
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kg\(^{-1}\) of biomass sample, \(\eta_{\text{combustion}}\) is producer gas burner combustion efficiency (90%), \(I\) is average solar radiation (W m\(^{-2}\)), \(A\) is area (m\(^2\)) of solar air heater, and \(\eta_{\text{thermo-hydraulic}}\) is thermo-hydraulic efficiency (74%) of solar air heater. It may be mentioned that producer gas combustion product and air mixture were used for tea drying and therefore no heat exchanger was used. Hence, combustion efficiency was considered about 90% for producer gas burner. The total thermal energy for hybrid mode black tea drying is given by Eq. (5.3).

\[
\dot{Q}_{HD} = \dot{Q}_{ST} + \dot{Q}_{BT}
\]  

(5.3)

5.2 Computation of drying efficiency in individual and hybrid modes

For low humidity and low temperature convective drying, energy efficiency of dryer may be approximated with its thermal efficiency that is given by Eq. (5.4).

\[
\eta_{\text{thermal}} = \frac{T_1 - T_{wb}}{T_1 - T_{amb}}
\]  

(5.4)

Where \(T_1\) is inlet air temperature to the dryer, \(T_{wb}\) is wet bulb temperature corresponding to dryer outlet air condition, and \(T_{amb}\) is ambient air temperature.

The system efficiency for producer gas fired solar assisted hybrid dryer is given by Eq. (5.5).

\[
\eta_s = \frac{WL}{(I\eta_{ar} + P_f + V_{pg}\times LCV)}
\]  

(5.5)

Where \(V_{pg}\) (Nm\(^3\) h\(^{-1}\)) is volume flow rate of producer gas combusted and LCV is lower calorific value (kJ Nm\(^{-3}\)) of producer gas. \(P_f\) is dryer suction blower energy consumption (kWh), \(I\) is solar radiation (kW m\(^{-2}\)), \(A\) is area (m\(^2\)) of air heater, \(\eta_{ah}\) is thermal efficiency of improved air heater, \(W\) is the mass of water removed per unit time and \(L\) is theoretical amount of heat (2700 kJ kg\(^{-1}\)) required for evaporation of one kg of bound moisture. This will give the additional energy input with appropriate hybridization ratio [234].
5.3. Assessment of locally available solar energy resources

Automatic Weather Station (AWS) was used to monitoring and record climatic data at 30 days interval. AWS provided data concerning (i) wind speed (m s^{-1}), (ii) solar radiation (W m^{-2}), (iii) air temperature (°C), (iv) dew point (°C), and (v) humidity. It may be noted that AWS could provide data for the parameters at any time interval as per requirement of the experiments.

Therefore, five years (2008 to 2012) solar radiation data for Sonitpur district (Assam: India) was taken from AWS located at Tezpur University campus (latitude 26° 42´ 03ʺ N and longitude 92° 49´ 49ʺ E). The solar radiation data were available in Wm^{-2} at hourly basis. The daily average data for a particular day were calculated and then converted to kWm^{-2}day^{-1}. From the daily average data, monthly average of solar radiation data were calculated and analyzed to investigate the prospect of solar thermal energy utilization. Also from AWS data, the total availability of solar radiation hour (Fig.5.2) in Sonitpur District was calculated.

![Solar insolation in Sonitpur district](image)

Fig.5.2 Monthly variation of solar insolation in Sonitpur district, Assam

The solar radiation data measured with the AWS indicated that on an average more than five hours per day solar radiation was available above 630 W m^{-2} and wind velocity was less than 4.5 m s^{-1}. This radiation was sufficient for testing and
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Performance evaluation of improved designed solar air heater. Performance studies of the solar air heater was experimented by measuring continuously the velocity of hot air at outlet of duct, inlet air temperature, outlet air temperature, ambient air temperature, incident solar irradiation, wind speed and direction. The availability of solar radiation hour per day for Tezpur, Assam during the year 2008 to 2012 is shown in the Fig. 5.3 below.

![Graph showing monthly variations of solar radiation hours per day at Tezpur](image)

Fig. 5.3 Monthly variations of solar radiation hours per day at Tezpur

Data were available from July month for year 2008 as shown in Fig. 5.3. It was observed that the highest and lowest solar radiation hours were obtained in September (6 h/day) and in December (3 h/day) months in 2008. However, the data were available for the entire year in the year of 2009, 2010, 2011, and 2012. In the year 2009, it was observed that during the month of July, availability of solar radiation hour per day was highest (7 h/day) and lowest values were in the months of January and December (3 h/day). The availability of maximum solar radiation hours per day was observed during the months of April and May and minimum radiation for the months of October and November in the year 2010. The maximum and minimum
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Solar radiation hours per day were in the months of May, July (6.5 h/day) and January (3.5 h/day) in the month of 2011. The corresponding radiation hours were in the month of May (6.5 h/day) and December (3 h/day) in the year 2012.

5.4 Monthly variation of tea drying thermal load

It has been observed from studies that black tea production in tea processing industries in Assam varies over the years. General trend is November, December and January to March of the succeeding year; the black tea production is almost insignificant. From middle of April to early November is considered peak period for black tea production. Accordingly, thermal load for tea drying also varies. Average variation of thermal load (%) over the year is presented in Fig.5.4. Now comparing Fig. 5.3 (Solar radiation pattern in Tezpur over the year) with Fig.5.4 (Thermal energy consumption pattern in a two million kilogram black tea production tea estate), it has been observed that a fraction tea drying thermal energy may be supplemented by solar radiation if appropriately designed solar air heater exists.

Fig.5.4 Monthly variation of processing load in a two million kg made tea factory
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5.5 Solar air heater

It has been observed that some air heaters are available in India market. However, site-specific designed and low cost air heater for assisting black tea drying is still lacking. Moreover, literature on research and development and performance testing of improved solar air heaters are available [123–133]. These products are not available in market particularly for tea drying application. It has been observed that most of solar air heater absorber plate in mentioned literature had used galvanized iron or steel sheet. Therefore, a different solar thermal absorber material (Aluminum plate of 2 mm thickness) had been selected for the design. In this Chapter, an improved solar air heater has been considered for its performance evaluation in actual outdoor condition at Tezpur University campus (latitude 26° 42´ 03˝ N and longitude 92° 49´ 49˝ E) in the month of May-June 2012. Both the thermal performance and thermo hydraulic performance would be evaluated for the hemispherical protruded solar air heater at variable dimensionless protrusion height and pitch (roughness parameter) and Reynolds number. This series of exercise have been performed with an aim to examine the prospect of solar thermal energy in black tea drying.

5.5.1 Major components and development of solar air heater

The different components of a solar air heater had been discussed in Chapter: 2 [123-133] (literature review). Therefore, two rectangular ducts measuring 2400 (L) × 375 (W) × 37.5 (H) mm³ was fabricated (Fig 5.5a- Fig.5.5d) at Tezpur University. The top of the air heater was covered with two number of 5 mm thickness commercial transparent glass. The other five sides of the air heater were covered with 10 mm thickness plywood boards. Moreover, the lengths of entry and exit sections were provided with 900 mm and 500 mm as per established standards [242]. The hydraulic diameter ($D_h$) of the solar air heater duct was calculated from Eq. (5.6):

$$D_h = \frac{2WH}{W+H}$$  \hspace{1cm} (5.6)

where $W$ is the duct width (mm) and $H$ is duct height (mm). The design methodology of the solar air heater has been presented in Fig.5.5 below.
Design flowchart of solar air heater

Development of a Solar Air Heater

Material Selection

Design Parameter Computation

Aluminum Plate
2400 × 750 × 2, MS angle/flat

5 mm thickness glass, 10 mm thickness ply-board

Hydraulic diameter
(D = 68 mm)
computation

Protrusion height (e = 2.4, 3.0, 3.7) mm

Pitch (p = 24, 36, 52) mm

Relative roughness height (e/D = 0.035, 0.045, 0.055) mm

Relative roughness pitch (p/e = 10, 12, 14) mm

Fig.5.5 Design methodology of solar air heater thermal energy absorber

The designed and developed solar air heater absorber

The absorber plate is considered as a critical component of solar air heater design.
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Fig. 5.6a Hemispherical protrusion on aluminum sheet

Fig. 5.6b Dimension of hemispherical protrusion
The protrusion height ($e$) was varied as (2.4, 3.0, 3.7) mm (Fig. 5.6a and Fig.5.6b). The long way length ($p$) was varied as (24, 36, and 52) mm respectively (Fig.5.6c).

Nusselt number for the hemispherical protruded absorber may be calculated from experiment data of average heat transfer coefficient ($\bar{h}$), thermal conductivity ($k$) and hydraulic diameter ($D_h$) of duct as presented in Appendix: A3. Nusselt number for smooth rectangular duct is given by Dittus-Boelter (Eq. (5.7)) [233]. Roughness parameters namely $\frac{e}{D}$ and $\frac{p}{e}$ are strong function of Nusselt number for artificially roughen solar air heater. The cross section of air heater is shown in Fig.5.6d below.

$$Nu_s = 0.034Re^{0.8}Pr^{0.4}2\left(\frac{\rho_{av}}{\rho}\right)$$

(5.7)

Fig. 5.6c Different dimensions of solar air heater absorber
5.5.2 Experimental methodology

ASHRAE 93-2003 standard was followed for testing of high performance solar air heater for tea drying air [235]. One improved solar air heater of gross absorber area 1.8 m² was considered for present studies. The collector temperature and pressure measurement points were made to close to the collector rigid duct section. The measured variables included inlet and outlet air temperatures, ambient temperature, airflow rates, wind velocity, pressure drop and solar radiation.

The ten numbers of PT-100 type thermocouples (24 SWG) with digital display units (Electra, made in India), temperatures accuracy of 0.1°C had measured the duct air temperatures along the flow length of air. Similarly, fifteen thermocouples were pasted on the plate to measure average plate temperatures.
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The mass flow rate of air was measured with the help of a calibrated hotwire anemometer (Testo 425, Made in Germany, [(0-20) m s\(^{-1}\), resolution = 0.01 m s\(^{-1}\), Accuracy = 0.03 m s\(^{-1}\)]. Average pressure drop across the air heater duct was measured with micro-manometer (Testo 525, Made in Germany), [(0- 200) hPa, resolution = 0.1 hPa]. Air velocity was varied by using a variable speed blower (Black and Decker, India, Maximum discharge = 3.5 m\(^3\) min\(^{-1}\), maximum rpm = 16000). The hotwire anemometer was calibrated with a gas turbine flow meter (Discharge: 6- 2500 m\(^3\) h\(^{-1}\), Linearity = ± 0.5%, Make: Rock-win, India).

The computed hydraulic diameter \(D\) was 68 mm for this roughen duct air heater. The relative roughness height \(\frac{e}{D}\) (Fig.5.6a, Fig.5.6b) varied as 0.035, 0.045, 0.055 and relative roughness pitch \(\frac{P}{e}\) varied as 10, 12, 14 during the experiments (Fig.5.6c) for total nine absorbers. The Reynolds number was varied from (3500-17000) for the experiments by using a variable speed blower. Flow of air was measured with a gas turbine flow meter.

To analyze the performance solar air heater, the following assumptions had been made. (1) The temperature difference between the plate and protrusion was neglected due to the large thermal conductivity of the absorber plate and hemispherical protrusion. (2) The thermal process in roughened air collector was approximately in steady state. (3) Centrifugal blower caused negligible rise in air temperature. (4) The glazing material had negligible heat capacity. The improved air heater performance testing experiments were normally conducted on sunny days from 9.00 a.m. to 15.00 p.m. at Tezpur University campus (latitude 26° 42´ 03“ N and longitude 92° 49´ 49“ E). The air heater experimental set up is given in Fig.5.7a and Fig.5.7b below.
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Fig. 5.7a Improved solar air heater: experimental setup
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5.6 Economic analysis

An effort had been made to investigate economic feasibility of woody biomass based gasifier and solar air heater hybrid renewable energy system for partial substitution of conventional thermal energy in tea manufacturing. There are different procedures available in literatures for economic feasibility analysis of a new energy system. However, in the present investigation a specific procedure was followed where (i) Net present value (ii) Benefit cost ratio and (iii) Payback period pertaining to a new renewable energy technology were assessed. The difference between the present value of all future returns ($F_{n1}$) and present money required to make an investment ($F_{n2}$) with rate of interest ($i$) for ($n$) years are related with net present worth by Eq. (5.8).

$$NPV = \sum_{n=1}^{n} \frac{F_{n1} - F_{n2}}{(1+i)^n}$$  \hspace{1cm} (5.8)

Benefit cost ratio defined as the present worth of benefit stream to present worth of cost stream. An acceptable project must have benefit cost ratio greater than one. Mathematically, benefit cost ratio is expressed as in Eq. (5.9).
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\[
\text{Benefit cost ratio} = \frac{\sum_{n=1}^{n} F_{n1}}{\sum_{n=1}^{n} F_{n2}} \quad (5.9)
\]

The principal amount capital \((P)\) with rate of interest \(i\) (minimum attractive rate of return, \(MARR\)) for \(n\) years yield future amount of money \(F_n\) given by the following Eq. \((5.10)\) [236].

\[
P = \frac{F_n}{(1+i)^n} \quad (5.10)
\]

The necessary condition for attractive payback period for an investment \(C_0\), the amount accumulated \(A_t\) in \(m\) years is given by the inequality \((5.11)\) and a project investment has to be attractive, internal rate of return must be greater than minimum attractive rate of return \((IRR > MARR)\).

\[
PBP = \text{the smallest } m \text{ such that } \sum_{t=1}^{m} A_t > C_0 \quad (5.11)
\]

The payback period is the total length of time from beginning of the project until the net value of the incremental production stream recovers total amount of capital investment. The following parameters were considered to carry out economic analysis of a gasifier (454 kW\text{thermal}) cum tea dryer system [236-238].

**5.7 Results and discussions**

The computation for collector efficiency was performed for incident of solar radiations (average 790 W m\(^{-2}\)). Data were measured from 9.00 a.m. to 3.00 p.m. at automatic weather station of Tezpur University. Therefore, the Fig. 5.8 shows variation of solar radiation and improved air heater air temperature with time from 9.00 a.m. to 3.00 p.m. The maximum outlet air temperature was 65 °C around 12.00 p.m. at solar irradiance of (950) W m\(^{-2}\). The testing was performed as per established standards [235].

Fig 6.9 shows variation of solar air preheater output air temperature and efficiency with hot air mass flow rate (kg s\(^{-1}\) m\(^{-2}\)) against collector area. It is clear that thermal efficiency increases with increase in air mass flow rate. The output temperature of hot
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Fig. 5.8 Variation of solar radiation and improved air heater air temperature with time

Fig. 5.9 Variation of collector outlet temperature and efficiency with air mass flow rate
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air decreases with increase hot air mass flow. Beyond hot air mass flows rate 0.028 kg s\(^{-1}\) m\(^{-2}\), the falling rate of outlet temperature of air heater become steady, although collector efficiency still increases. From these data, the best operating point of solar air heater may be found out around air mass flow rate of 0.028 kg s\(^{-1}\) m\(^{-2}\). Beyond this mass flow rate, even if there is an increase of collector efficiency, due to fall in outlet air temperature, it not economical to operate the solar air heater. Therefore, performance studies have been made at mass flow rate of 0.028 kg s\(^{-1}\) m\(^{-2}\).

Fig. 5.10 shows the effect of variable \(\frac{p}{e}\) (10-14) for fixed value of roughness parameter, \(\frac{e}{D}\) (0.055). It is clear that Nusselt number is maximum for \(\frac{p}{e}\) value of 12 and it decreases in either side of 12. This might be due to separation of airflow over hemispherical protruded surface and reattachment of free shear layer occurs for \(\frac{p}{e}\) (12). This gives rise to maximum heat transfer near reattachment region. Reattachment may
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not occur near \( \frac{P}{e} \) (10 or 14) and therefore, Nusselt number is smaller in these regions. For higher value of \( \frac{e}{D} \) (0.055) ratio, more reattachment of free shear layer might occur that enhances Nusselt number.

The effect of \( \frac{P}{e} \) on performance of hemispherical protruded absorber has been discussed at Fig. 5.11. It is seen that maximum thermal efficiency of 82% is achievable for \( \frac{P}{e} = 12 \) and \( \frac{e}{D} = 0.055 \). The corresponding \( F_0 U_l \) and \( F_0(\alpha \tau) \) are \((10, 09 \text{ and } 07) \, \text{W} \, \text{m}^{-2} K^{-1}\) and 0.837, 0.721, and 0.572 for \( \frac{P}{e} \) (12, 10 and 14). Similarly, Fig. 5.12 shows the effect of \( \frac{e}{D} \) (0.055, 0.045, 0.035) of hemispherical protruded absorber on constant \( \frac{P}{e} = 12 \). The values of \( F_0 U_l \) and \( F_0(\alpha \tau) \) are \((10, 09, 08) \, \text{W} \, \text{m}^{-2} K^{-1}\) and 0.884, 0.746 and 0.664 respectively. Average solar radiation was above (790) W m\(^{-2}\) for all experiments.

![Graph showing thermal efficiency vs. 1000(\( (T_o - T_i)/I_c \))](image)

Fig. 5.11 Effect of p/e on performance of hemispherical protruded air heater
Fig. 5.12 Effect of $e/D$ on performance of hemispherical protruded solar air heater

Fig. 5.13 presents variation effective efficiency of hemispherical protruded solar air heater with Reynolds number. It is clear that effective efficiency increases with Reynolds number for all three values of $\frac{e}{D}$ (0.035, 0.045, and 0.055) and attains maximum value for Reynolds number around 12000. Effective efficiency for smooth solar air heater absorber was minimum up to Reynolds number 140000 and beyond this Reynolds number, effective efficiency of smooth solar air heater become maximum. Therefore, beyond this region, there is no gain in effective efficiency of artificially roughen air heater. It is also clear that with $\frac{e}{D}$ value 0.035 and 0.055 gives maximum and minimum effective efficiency (74% and 64%) respectively around Reynolds number 12000.
Fig. 5.13 Variation of effective efficiency hemispherical protruded solar air heater with Reynolds number

Fig. 5.14 Variation of useful energy of solar air heater with Reynolds number
Fig.5.14 shows variation of useful energy of hemispherical protruded solar air heater with Reynolds number. It is clear that with decrease in dimensionless protrusion height ($e/D$ : 0.055, 0.045, 0.035), useful heat gain by the hemispherical protruded solar thermal absorber increases. All three solar thermal energy absorber gain maximum energy around Reynolds number 12000 and beyond this useful energy gain decreases because of more high-grade energy is required to propel air than it acquires from roughen air heater.

Fig.5.15 presents comparison of effective efficiency of hemispherical protruded solar air heater with smooth air heater. The maximum efficiency enhancement takes place for dimensionless protrusion height ($e/D = 0.035$) that is about 8.9% around Reynolds number 6000-10000. The minimum efficiency enhancement about 5.5% takes place for protrusion height ($e/D = 0.055$).
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For performance testing of solar air heater, it has been observed that hemispherical protrusion with \( \frac{\varepsilon}{D} = 0.035 \) and \( \frac{P}{e} = 12 \) geometry gave best thermo-hydraulic efficiency 74% around Reynolds number 12000. Therefore, this configuration improved solar air heater has been selected for producer gas solar hybrid studies of black tea drying analysis.

\[
Nu_{rh} = 5.2 \times 10^{-4} Re^{1.27} \left( \frac{P}{e} \right)^{3.15} \exp(-2.12) \left[ \log \left( \frac{P}{e} \right)^{1/6} \left( \frac{\varepsilon}{D} \right)^{0.33} \exp(-1.30) \left( \log \frac{\varepsilon}{D} \right)^2 \right] \quad (5.12)
\]

A modified correlation of Nusselt number had been developed from analysis of our experimental data of hemispherical protruded solar air heater as given by correlation Eq. (5.12) similar to that developed by Saini and Verma [133].

5.8 Hybridization of improved solar thermal air heater and biomass gasifier for black tea drying

To perform analytical studies of hybridization of improved solar air heater and woody biomass gasifier, the best operating condition for both the renewable energy system has been considered from experimental data [Chapter:3, Chapter:4 and Chapter:5]. The biomass gasifier was considered operating at best gasification efficiency with average calorific value producer gas 4.5 MJ m\(^{-3}\). The maximum thermal output of the gasifier was 30 kW. Average dry biomass (moisture about 10 %) consumption rate was 8.5 kg h\(^{-1}\). Only maximum 5 kW thermal output of producer gas was used for tea drying because of the size limitation of the tray dryer. The improved solar air heater was considered with maximum thermo-hydraulic efficiency of 74% at average radiation of 790 W m\(^{-2}\) in a sunny day in the month of May 2012 at Tezpur University campus.

5.8.1 Dryer efficiency

Fig.5.16 shows variation of tea drying efficiency and average dryer temperature with time. It is clear that at the beginning of black tea drying, the
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efficiency gradually increased to a maximum value (60%) then sharply decreased until completion (10%) of drying process. This is because initially some amount of thermal energy was utilized for preheating the fermented tea before start of moisture diffusion process. The dryer temperature gradually increased because of difficulty of removing internal moisture from the tea particles. At this stage, maximum amount of thermal energy was consumed if drying fluid temperature remained at initial temperature. Near completion of tea drying process, reduction of drying fluid temperature near (70-80) °C and longer drying completion time, would reduce specific energy consumption of tea dryer. Therefore, an average tea drying efficiency 40 % may be considered.

Fig.5.16 Variation of drying efficiency with drying air temperature with drying time

5.8.2 Specific energy consumption in hybrid drying

Fig.5.17 shows variation of tea drying energy from 9.00 a.m. to 2.40 p.m. considering a sunny day with available solar radiation similar to that had been presented in Fig.5.8. It is clear that specific energy consumption for per kilogram of
made tea was estimated 25.50 MJ in batch drying of black tea while using both producer gas and improved solar air heater.

![Bar chart showing variation of producer gas and solar thermal energy from 9.00 a.m. to 14.40 p.m.](image)

Fig. 5.17 Variation of producer gas and solar thermal energy from 9.00 a.m. to 14.40 p.m

It is clear that minimum contribution of solar energy was 12.9% at (9.00 - 9.40) a.m., while maximum contribution was 27.23% at (12.00 - 12.40) p.m. By using five 1.65 m² improved solar air heater average 20% saving in biomass energy is possible.

### 5.8.3 Economic analysis of biomass gasification and improved solar air heater hybrid system for tea drying

Annual black tea manufactured by an average-size tea estate in Sonitpur district (Assam) was 990 t in the year 2011-2012 as computed by geographical information system mapping. Reported tea plantation areas in a FCC image (band 2, 3 and 4) were seen in dark-red to red tone depending on whether they are directly planted or appearing below shaded trees in different sizes with regular sharp edges indicating the presence of a fence around it. The average yield of black tea
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manufactured was assumed as $0.2 \text{ t ha}^{-1}$ to compute annual black tea production for a representative average size tea estate. The total back tea production was calculated by multiplying yield of black tea by total mapped area of the tea estate [239].

Since, annual black tea manufactured by an average size tea estate in Sonitpur district (Assam) was 990 t (GIS mapping). The corresponding coal requirement is 825 t (Coal 1.35 kg kg$^{-1}$ of black tea manufactured). It was also observed from this study that thermal efficiency was 20% for a conventional coal fired furnace and (80-90) % for proposed producer gas fired furnace used for tea drying. Therefore, producer gas fired furnace is a better option over coal-fired system. Coal international prices was 95 $ \text{t}^{-1}$, Birol et al., [240] and that of woody biomass was 11 $ \text{t}^{-1}$ in the year 2011-2012. A 454 kW thermal woody biomass gasifier was considered that could substitute 28% of this thermal load [241]. It is estimated that by plantation of 22.5 ha *Bambusa tulda*, this thermal load of said biomass gasifier may be met [242]. If 400 m$^2$ of tea factory galvanized roof were converted by using black painting, plywood insulation and tempered glass enclosure to convert into solar air heater then average 20 % of biomass energy may be saved. The annual carbon-dioxide reduction 2189 t is achievable [243]. The payback period of the hybrid renewable thermal energy based system is less than fifteen month and benefit to cost ratio is 1:1 (Appendix: A8).

The limitation of improved solar air heater is that its performance may deteriorate with variation of solar radiation for an industrial scale black tea drying unit. In this case, thermal energy storage or oversized solar air heater will be useful. The solar and biomass gasifier based black tea drying is for partial substitution of conventional energy not for 100% replacement. This will reduce CO$_2$ emission from tea factories in Assam (India). The economic analysis was performed with data available from literature for a scaled up tea industry traditionally using inefficient coal fired furnace. However, actual payback period may be more than the computed one due to presence of real industrial tea drying system that is limitation of the study.