A Novel Syngas-fired Hybrid Heating Source for Solar-Thermal Applications: Energy and Exergy Analysis

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Abstract

A hybrid heating source using biomass-derived syngas is proposed to enable continuous operation of standalone solar thermal power generation plants. A novel, two-stage, low temperature combustion system is proposed that has the potential to provide stable combustion of syngas with near-zero NO_x emissions. The hybrid heating system consists of a downdraft gasifier, a two-stage combustion system, and other auxiliaries. When integrated with a solar cycle, the entire system can be referred to as the integrated gasification solar combined cycle (IGSCC). The supercritical CO_2 Brayton cycle (SCO_2) is selected for the solar cycle due to its high efficiency. The thermodynamic performance evaluation of the individual unit and the combined system has been conducted from both energy and exergy considerations. The effect of parameters such as gasification temperature, biomass moisture content, equivalence ratio, and pressure ratio is studied. The efficiency of the IGSCC exhibited a non-monotonic behavior. A maximum thermal efficiency of 36.5% was achieved at an overall equivalence ratio of 0.22 and pressure ratio of 2.75 when the gasifier was operating at $T_g = 1073$ K with biomass containing 20% moisture. The efficiency increased to 40.8% when dry biomass was gasified at a temperature of 973 K. The exergy analysis revealed that the maximum exergy destruction occurred in the gasification system, followed by the combustion system, SCO_2 cycle, and regenerator. The exergy analysis also showed that 8.72% of the total exergy is lost in the exhaust; however, this can be utilized for drying of the biomass.

Keywords: Solar biomass; supercritical CO_2 Brayton cycle; gasification; syngas; catalytic/MILD combustion; exergy

Nomenclature

CSP	Concentrating solar power
SCO_2	Supercritical CO_2
ISCC	Integrated solar combined cycle
IGSCC	Integrated gasification solar combined cycle

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b-IGCC	Biomass integrated gasification combined cycle
RCL	Rich catalytic lean burn
MILD	Moderate or intense low oxygen dilution
NO _x	Oxides of nitrogen
SO _x	Oxides of sulfur
HE _x	Heat exchanger
Y	Mass fraction
X	Mole fraction
R _u	Universal gas constant
W _{net}	Total work output (kW)
h	Enthalpy (kJ/kmol)
g	Gibb's function (kJ/kmol)
T	Temperature (K)
s	Entropy (kJ/kmol K)
C _p	Specific heat at constant pressure (kJ/kmol K)
LHV	Lower heating value (kJ/kmol)
P	Pressure (MPa)
n	Quantity (kmol)
ex	Exergy (kJ/kmol)
<i>Greek symbols</i>	
Φ	Overall equivalence ratio
η	Efficiency
<i>Subscript</i>	
o	Standard state (298.15 K, 0.1 MPa)
i	Species
j	Species or component
g	gasification
s	solar
isen	isentropic
w	water
org	Organic
cat	Catalytic
p	Product
r	Reactant
f	Formation
ov	Overall

ph	Physical
ch	Chemical
t	Total
I	First Law
II	Second Law
max	Maximum
<i>Superscript</i>	
o	Standard state (298.15 K, 0.1 MPa)
s	solar

1. Introduction

Recently, there has been tremendous interest in solar energy and associated technologies such as concentrating solar power (CSP), photovoltaics, energy storage materials, organic Rankine cycles and supercritical CO₂ Brayton cycle. The interest has been propelled primarily by the growing demand for clean, sustainable and decentralized power that solar energy can supply. But solar-thermal plants, unlike fossil fuel power plants, presently encounter two major challenges: (a) low plant capacity factor due to the varying solar insolation, and (b) reduced plant efficiency due to low operating temperatures.

The capacity factor of solar plants has improved over the years due to the advancements in thermal energy storage technology [1, 2]. However, these systems offer limited duration of backup (Figure 1) and the problem becomes severe during long periods of cloud cover. To this day, continuous operation of standalone solar plants remains a challenge and is a key bottleneck in its large scale implementation. The efficiency of solar plants is also low; around 20% for parabolic trough concentrators and less than 10% for organic Rankine cycles [3, 4]. High-temperature CSP using solar towers can reach efficiencies up to 23% by heating the working fluid in the range of 800-1000 K [5]. In this temperature range, the use of supercritical technology with CO₂ as the working fluid can increase the cycle efficiency up to 30% [6]. Higher temperatures in the range of 1000-1300 K have been achieved in demonstration plants using pressurized air and multistage receivers that can improve the cycle efficiency further [7, 8]. However, conventional molten salt storage systems cannot operate at such high temperatures (800-1300 K). Potential storage materials such as fire bricks, lithium liquid salt, MgCl₂, Al, etc. are still in the research and development phase.

To overcome these challenges, a number of hybrid-CSP configurations have been proposed in the literature. Until recently, the hybridization schemes mostly concentrated on the usage of solar energy for preheating, steam generation, chemical reforming, etc., to reduce consumption of fossil fuels such as coal and natural gas [5, 10]. This form of hybridization is known as the integrated solar combined

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cycle (ISCC) and is currently operational at power plants in Egypt, Morocco and Algeria [9]. However, the solar share in the generated power is usually less than 15% (Table 1) and fossil fuels remain the main source of energy. A renewable alternative to ISCC that is gaining interest for CSP hybridization is biomass.

Biomass is one of the most important sources of renewable energy due to its availability from multiple sources and carbon neutrality. It can be used through direct combustion, or via gasification, in which the chemical energy of the solid biomass is converted into the chemical energy of a gaseous fuel termed as producer gas or syngas. The conversion efficiency of biomass to electricity through gasification and direct combustion are comparable, with a maximum of 43% and an average of 27% [11]. The capacity of such technologies reported in the literature may vary from 1-10 MW for pyrolysis [12] to about 100-300 MW for gasification and direct combustion [13, 14]. Biomass generated electricity with an average price of 6.9 c/kWh is not cost effective when compared to electricity from fossil fuels such as natural gas (3.1-5 c/kWh), pulverized coal (4.3-5.2 c/kWh), and coal gasification (4.1-6.1 c/kWh) [15]. However, biomass is preferred over coal when other factors such as environmental impact, human health, soil erosion, etc. are considered [16]. In terms of CO₂ emission, the highest reported CO₂ emission (132 gCO₂eq/ kWh) by a biomass power production unit [17] is less than one-third of the lowest-emitting natural gas plant and one-fifth of the lowest emitting coal power plant [11].

Given the renewable nature of both biomass and CSP, a number of feasibility studies have been conducted for its deployment in countries such as India, Brazil and Australia [18 - 22]. The studies suggest that CSP-biomass hybridization is promising in the range of 2-25 MW and can be used for tri-generation, electricity production and process heat generation. A study by Fonseca et al. [23] has reported that the greenhouse gas (GHG) emissions from such a hybrid would be 10 times lower than those from a CSP plant with non-renewable extensions. According to another study, the CSP-biomass hybrid plant can be commercially viable in Australia as the investment is 69% lesser than that of standalone CSP without thermal storage [24]. Very few studies are available on the thermodynamic analysis of different CSP-biomass hybridization schemes. One such work has reported an optimal cycle thermal efficiency of 27% when biomass is used to accommodate the fluctuation in solar intensity [25]. Another study has recently reported a maximum thermal efficiency of 36.7% when CSP is hybridized with biomass integrated gasification combined cycle (b-IGCC) [26]. Thermodynamic analysis of CSP-biomass for tri-generation in the 2-10 MW range has also been investigated [25]. On the implementation side, the first and the only commercial CSP-biomass plant has been operational in Spain since 2012 [9]. It has a capacity of 22.5 MW and is supplied by 2 x 22 MW thermal biomass units operating on waste forest biomass and energy crops. The operational experience obtained from this plant will be valuable in the development of newer CSP-biomass hybridization strategies.

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The focus of the present work is to investigate the thermodynamic performance of a syngas-fired hybrid heating source that can substitute supplementary fossil fuel firing in standalone CSP plants. When such a hybrid heating source is integrated with a CSP plant, it can be referred to as the integrated gasification solar combined-cycle (IGSCC). The hybrid heating source is a self-sustaining unit and consists of a downdraft biomass gasifier, a two-stage combustion system, and other auxiliaries. It can be retrofitted to existing CSP plants to improve their capacity factor. The performance of the heating source has been assessed by combining it with a supercritical CO₂ (SCO₂) Brayton cycle [6]. The SCO₂ cycle has been selected as it promises higher efficiency compared to the steam-Rankine cycle. The SCO₂ technology is currently under research and development and shows tremendous potential for high-temperature CSP. In the present concept, the SCO₂ cycle can operate independently as a standalone unit during the day, while the biomass syngas can be used for supplementary firing at night. The power output from such a hybridization scheme also matches with the daily load demand curve that peaks between 18:00 hours and 24:00 hours. Regarding the novelty of the present work, the first aspect concerns the proposed hybrid combustor concept using rich catalytic combustor as a first-stage and a moderate or intense low oxygen dilution (MILD) combustor as a second stage. Moreover, the application of biomass-derived syngas as a hybrid energy source for solar thermal power plants as proposed in the present work is also novel. Furthermore, a comprehensive energetic and exergetic analysis for such solar-biomass hybrid systems such as the one conducted in the present work, is not reported in the literature. Finally, the application of this hybrid heating source to a supercritical CO₂ Brayton cycle is also novel. The next few sections present the detailed thermodynamic modeling and performance assessment of the biomass gasifier, the two-stage combustion system, the SCO₂ cycle and the combined system (IGSCC).

2. Cycle description

2.1. Hybrid heating source

The hybrid heating source with all the components is shown in Fig. 2. Atmospheric air entering the cycle (1) is divided into two streams (4 and 5) for the two-stage combustion system. Stream 4 is mixed with the syngas from the biomass gasifier (3) to form a rich fuel-air mixture (6). It is then compressed (7) and partially oxidized in the first stage of the combustion system to form a high-temperature stream required by the second stage of the combustion system (8). The second air stream (5) is also compressed (9) and preheated in a regenerator and then supplied to the second stage of the combustion system (14) to complete the oxidation of the fuel. The high-temperature output (10) from the combustion system is expanded in a turbine to power the two compressors. In the process, it also generates additional power. The exhaust from the turbine (11) then supplies thermal energy to the SCO₂ cycle through a heat exchanger. The output from the heat exchanger (12) preheats the incoming air in a regenerator (9 – 14) before it is vented to the atmosphere (13).

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The two-stage combustion technology proposed in this study offers multiple advancements over conventional combustion systems. The system is inspired by the rich catalytic lean burn (RCL) concept [27-29]. In the RCL technology, a fraction of the reactants is oxidized under fuel-rich conditions in a catalytic reactor. The exhaust is subsequently mixed with additional air to oxidize the remaining fuel in a second gas-phase reactor under fuel-lean conditions. Here, we propose the use of a low-NO_x technology known as moderate or intense low-oxygen dilution (MILD) combustion [30 - 34] as a substitute for the homogeneous reactor in the RCL technology. The MILD combustion concept is proven to achieve near-zero NO_x emissions due to its low-temperature operation (< 1500 K) [35]. The internal recirculation of exhaust gasses in MILD combustion increases the residence time of the reactants and ensures complete combustion with zero particulate emissions. The rich catalytic reactor generates the high-temperature reactants essential for MILD combustion and provides flame stability under overall fuel-lean conditions. Such a hybrid combustion technology offering low-temperature exhaust (< 1500 K), near-zero pollutant emissions and flame stability is ideal for the IGSCC concept.

The key parameters governing the performance of the hybrid heating source are the fuel-air equivalence ratio, type of biomass feedstock and its moisture content and pressure ratio. While the type of biomass feedstock is region-specific, the other parameters can be varied independently to control the various state points. The details of the parameters for the hybrid heating source are shown in Table 2.

2.2. Solar cycle

The schematic of the supercritical CO₂ Brayton cycle used for the present analysis has been shown in Figure 3 [6]. The boiler in the figure represents the heat exchanger used to supply energy to the working fluid (CO₂) from the exhaust. The summary of the key operating parameters used in the analysis is shown in Table 3. More details of the cycle can be found elsewhere [6].

3. Mathematical modeling

3.1 Assumptions

The major assumptions for the hybrid heating source are as follows:

- a) The gases behave as an ideal gas mixture.
- b) All the reactions in the gasifier are in thermodynamic equilibrium [37, 38].
- c) Dry syngas at 298.15 K and 0.1 MPa is produced by the gasification system after gas cleaning (3) [39].
- d) The turbine and compressor isentropic efficiencies are 89% and 85%, respectively [3].
- e) Pressure drop in heat exchangers and combustors are 2% and 3% of inlet pressure, respectively [3].

- f) O_2 is completely consumed in the catalytic reactor [40]. This assumption was later validated using equilibrium calculations using STANJAN [41].
- g) Complete oxidation of H_2 , CO and CH_4 takes place during the MILD combustion phase [35].
- h) The temperature difference in the regenerator $T_{12} - T_{14} = 10$ K.

The assumptions for the SCO_2 cycle are the same as enumerated in [6] and the key parameters have been summarized in Table 3.

3.2 Numerical formulation

3.2.1 Gasification

The gasification process is modeled assuming chemical equilibrium of the species. The chemical formula for dry woody biomass used as feedstock is assumed to be $CH_{1.44}O_{0.66}$ [38]. The global reaction for the gasification of this feedstock can be expressed as:

where, w is the water present per kmol of dry biomass, m is the amount of O_2 supplied per kmol of dry biomass, and n_1 , n_2 , n_3 , n_4 , and n_5 are the coefficients of the product species. The moisture in the gasifier (in kmol) per kmol of dry biomass is given by

$$w = \frac{M_f Y_w}{M_w (1 - Y_w)} = \frac{24 Y_w}{18 (1 - Y_w)} \quad (2)$$

M_f and M_w are the molecular weights of the dry biomass and water; Y_w is the mass fraction of water in the biomass. The 6 unknowns, n_1 to n_5 and m are solved using one energy balance equation, two chemical equilibrium equations, and three elemental balances for C, H and O in Eq. (1). Chemical equilibrium is assumed for the methane formation and water-gas shift reactions [36, 38];

The equilibrium constants for the above reactions can be written as:

$$K_1 = \frac{P_{CH_4}/P_o}{(P_{H_2}/P_o)^2} = \frac{X_{CH_4} P_o}{(X_{H_2})^2 P_g} \quad (5)$$

$$K_2 = \frac{(P_{CO_2}/P_o)(P_{H_2}/P_o)}{(P_{CO}/P_o)(P_{H_2O}/P_o)} = \frac{X_{H_2} X_{CO_2}}{X_{H_2O} X_{CO}} \quad (6)$$

where, P_j and X_j are the partial pressures and mole fractions of the j^{th} species, respectively in the product gas. P_g is the gasification pressure and P_o is the reference pressure (0.1 MPa). The value of the equilibrium constants has been determined using polynomial functions of the gasification

temperature [38]. The gasifier is assumed to be adiabatic and an energy balance equation is formed equating the enthalpy of the reactants with that of the products where the products are assumed to be at the gasification temperature [38].

The chemical exergy of biomass is calculated as follows [36]:

$$\beta = \frac{1.044 + 0.016(H/C) - 0.34493(O/C)[1 + 0.0531(H/C)]}{1 - 0.4124(O/C)} \quad (7)$$

$$ex_{ch,biomass} = \beta LHV_{biomass} + w \cdot ex_{ch,w} \text{ (/kmol dry biomass)} \quad (8)$$

where, $LHV_{biomass}$ is the lower heating value of dry biomass and $ex_{ch,w}$ is the exergy of water. H/C and O/C represent the atomic ratios in the dry biomass. The value of β in the present case is 1.1294 for the composition of biomass selected (Table 2). The energy and exergy efficiencies of the gasification process are defined as:

$$\eta_{I,g} = \frac{LHV_{gas}}{LHV_{biomass}} = \frac{\sum n_i LHV_i}{LHV_{org}} \quad (9)$$

$$\eta_{II,g} = \frac{ex_{ch,gas}}{ex_{ch,in}} = \frac{\sum n_i (ex_{ch,i}^o)}{ex_{ch,biomass} + m \sum (ex_{ch,O_2}^o + 3.76 ex_{ch,N_2}^o)} \quad (10)$$

where, n_i is the number of kmol, LHV_i is the lower heating value (kJ/kmol) and $ex_{ch,i}^o$ is the standard chemical exergy of the i^{th} species in the product gas. Only chemical exergy has been considered in calculating the exergy efficiency as the reactants and products are assumed to be at standard conditions. The standard chemical exergies at 298.15 K and 0.1 MPa pressure were taken from Szargut et al. [43] and are shown in Table 6.

3.2.2 Catalytic combustion

A methane reforming model [44] has been used to simulate fuel-rich combustion of syngas in the catalytic reactor. The global reaction can be expressed as;

The coefficients on the reactor side, $n_{i,7}$ are known and depend on the syngas composition from the gasifier and equivalence ratio in the catalytic reactor Φ_{cat} . The seven unknowns on the product side that include six $n_{i,8}$ values and T_8 can be solved using four elemental balances (C, H, O and N), one energy balance, and two equilibrium reactions. The equilibrium reactions used such as the methane formation reaction and water-gas shift reaction [44] are listed below:

The water-gas shift reaction has previously been used in the gasification model. The equilibrium constant for the methane formation reaction is given by:

$$K_3 = \exp[(g_{CO}(T) + 3g_{H_2}(T) - g_{CH_4}(T) - g_{H_2O}(T))/R_u T] = \frac{(P_{CH_4}/P_o)(P_{H_2O}/P_o)}{(P_{CO}/P_o)(P_{H_2}/P_o)^3} = \frac{X_{CH_4}X_{H_2O}}{X_{CO}(X_{H_2})^3} \left(\frac{P_o}{P}\right)^2 \quad (14)$$

Here, P is the system pressure and P_o is the standard reference pressure (0.1 MPa). The Gibb's function g_i (kJ/kmol) as a function of temperature has been evaluated by fitting polynomial curves to the data obtained from NIST JANAF tables [45]. For energy balance, the equation for adiabatic flame temperature calculation has been used [46]:

$$H_r(T_{in}) = H_p(T_{ad}) \quad (15)$$

where, H_r and H_p represent the enthalpy of the reactants and the products, respectively, T_{in} is the initial temperature of the reactants and T_{ad} is the adiabatic flame temperature. H_p and H_r have been calculated as

$$H_p(T) = \sum_i n_i [h_{f,p,i}^o + h_{p,i}(T) - h_{p,i}(T_o)] \quad (16)$$

$$H_r(T) = \sum_i n_i [h_{f,r,i}^o + h_{r,i}(T) - h_{r,i}(T_o)] \quad (17)$$

where n_i , $h_{f,i}^o$ and h_i represent the number of moles, enthalpy of formation and sensible enthalpy of species i , respectively. The enthalpy at the various state points for each species is given by:

$$h_i(T) - h_i^o(T_o) = \int_{T_o}^T C_{p,i} dT \quad (18)$$

The specific heat as a function of temperature has been obtained from NIST JANAF tables [45].

3.2.3 MILD combustion

The composition after MILD combustion (state 10) is determined using a global reaction for the complete combustion of syngas with air at an overall equivalence ratio Φ_{ov} :

The four unknowns, $n_{CO_2,10}$, $n_{H_2O,10}$, $n_{O_2,10}$ and $n_{N_2,10}$, can be obtained from the elemental balances of C, H, O and N. The overall equivalence ratio, Φ_{ov} , is an input parameter and determines the number of moles of O_2 at state 1 based on the composition of the fuel at state 3. The temperature T_{10} can be

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obtained by an energy balance (adiabatic flame temperature) between states 8, 14 and 10. The temperature and composition at state 8 are known from the model for catalytic combustion. The temperature at state 14 is solved iteratively with the assumption that $T_{12} - T_{14} = 10$ K.

3.2.4 Turbine and compressor

The temperature at the exit of the turbine and compressors is determined using isentropic efficiency of the devices. The pressure ratio across the compressor P_7/P_6 is an input parameter. The pressure ratio across the turbine is determined after considering pressure drop in the combustors and heat exchangers.

$$H_{\text{isen,comp}} = \frac{(H_{7,\text{isen}} - H_6)}{(H_{7,\text{actual}} - H_6)} \quad (20)$$

$$\eta_{\text{isen,turb}} = \frac{(H_{10} - H_{11,\text{actual}})}{(H_{10} - H_{11,\text{isen}})} \quad (21)$$

3.2.5 Heat exchangers

The temperature at the exit of the heat exchangers is calculated using an energy balance between the incoming and outgoing streams neglecting heat losses. The pressure drop in each heat exchanger is assumed to be 2% of the inlet pressure. The minimum temperature difference between the hot and cold streams has been assumed to be 10 K. For the numerical modelling of the SCO_2 cycle, an approach similar to [6] has been followed.

3.3 Performance metrics

The performance of the IGSCC has been assessed in terms of the energy and exergy efficiency.

3.3.1 Energy efficiency

The cycle thermal efficiency denotes the total work produced as a fraction of the input energy. The total work produced by the cycle is

$$W_{\text{net}} = W_{\text{SCO}_2} + \{(H_{10} - H_{11}) - (H_7 - H_6) - (H_9 - H_5)\} \quad (22)$$

$$W_{\text{SCO}_2} = \eta_{I,\text{SCO}_2} (H_{11} - H_{12}) \quad (23)$$

where, W_{SCO_2} and η_{I,SCO_2} are the total work produced and the thermal efficiency of the SCO_2 cycle, respectively. The thermal energy input to the cycle is equal to the LHV of biomass. The overall thermal efficiency of the cycle is given by:

$$\eta_I = \frac{W_{\text{net}}}{\text{LHV}_{\text{biomass}}} \quad (24)$$

3.3.2 Exergy efficiency

The exergy at every state point was calculated as the sum of physical and chemical exergy of each species. The physical exergy is defined as:

$$ex_{ph,i} = [h(T) - h(T_o)] - T_o [s(T) - s(T_o)] \quad (25)$$

The total exergy of the individual species is calculated as:

$$ex_{t,i} = ex_{ph,i} + ex_{ch,i}^o \quad (26)$$

The summation over all the species gives the total exergy as shown below:

$$ex_t = \sum_i n_i ex_{t,i} \quad (27)$$

The kinetic and potential energies were neglected. The entropy at the various state points for each species is calculated as:

$$s_i(T) - s_i^o(T_o) = \int_{T_o}^T C_{p,i} \left(\frac{dT}{T} \right) - R \ln \left(\frac{P_i}{P_o} \right) \quad (28)$$

The molar specific absolute entropies at 298.15 K and 0.1 MPa have been adopted from NIST JANAF tables [45]. The overall exergetic efficiency of the cycle is given by:

$$\eta_{II} = \frac{W_{net}}{ex_{ch,biomass} + m \sum (ex_{ch,O_2}^o + 3.76 ex_{ch,N_2}^o)} \quad (29)$$

The exergetic efficiency of the j^{th} component in the cycle is calculated as:

$$\eta_{II,j} = \frac{\sum (ex_t)_{j,out}}{\sum (ex_t)_{j,in}} \quad (30)$$

The subscript j refers to the various components of the cycle such as the compressor, turbine, combustion system, gasification system, SCO_2 cycle, etc. The exergy destroyed in the j^{th} component as a fraction of the total input exergy is expressed as:

$$X_{des,j} = \frac{\sum (ex_t)_{j,in} - \sum (ex_t)_{j,out}}{ex_{ch,biomass} + m \sum (ex_{ch,O_2}^o + 3.76 ex_{ch,N_2}^o)} \quad (31)$$

All calculations were performed using MATLAB R2014b (version 8.4). For the SCO_2 cycle, the properties were calculated using REFPROP 9.0 [47].

4. Results and discussion

This section presents the results on operational characteristics and performance of the individual components as well as the IGSCC. The analysis of the individual components is presented

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first and the operating conditions have been determined based on this analysis. This is followed by the overall performance evaluation of the IGSCC. Finally, the results are compared with the performance of other CSP-biomass hybrid cycles reported in the literature.

4.1 Gasification system

The gasification system significantly influences the overall performance of the IGSCC. The key parameters influencing the performance of the gasifier is the type of biomass, moisture content in the feedstock and the gasification temperature. The type of biomass selected in the present study is wood ($\text{CH}_{1.44}\text{O}_{0.66}$), with moisture content varying between (0 – 30) %. The gasification temperature has been varied between 973 – 1323 K, and can be controlled by changing the air-fuel ratio.

4.1.1 Validation

The gasification model used is validated with the experimental results of Jayah et al. [48] and the equilibrium model of Datta et al. [36] and the comparison is summarized in Table 7. Except for methane species concentration, the model prediction matches reasonably well with the reported data. The lack of agreement in methane concentration can be attributed to the non-attainment of equilibrium at the exit of the gasifier because of its finite length [48]. However, as the methane percentage in air-blown gasification is low (generally < 2%), the underprediction does not have a significant influence on the fuel calorific value [39].

4.1.2 Performance

The output from the gasifier after gas cleaning and removal of moisture (state 3) contains H_2 , CO, CH_4 , CO_2 and N_2 . For air-blown gasification, typical composition of syngas is 20% CO, 20% H_2 , 2% CH_4 , 12% CO_2 and 46% N_2 [39]. However, the composition can vary significantly with the operating conditions. The H_2/CO ratio and CH_4 volume percentage in the product gas as a function of gasification temperature at constant moisture content are shown in Figs. 4 and 5, respectively. The H_2/CO ratio varies between 0.61 and 1.52. A higher moisture content in the feedstock increases the H_2 content in the product gas through the water-gas shift reaction (WGSR). Since the WGSR is exothermic, an increase in gasification temperature decreases the H_2/CO ratio. This corroborates with the fact that the equilibrium constant K_2 decreases with an increase in temperature, thereby shifting the equilibrium of the reaction $\text{CO} + \text{H}_2\text{O} \rightarrow \text{CO}_2 + \text{H}_2$ towards the reactant side. The increase in gasification temperature also decreases the methane percentage in the product gas (Figure 5) and approaches zero at high temperatures. The presence of moisture in the biomass increases the methane percentage in the output.

Figures 6 and 7 represent the effect of gasification temperature and moisture content on the energy and exergy efficiency of the gasification process, respectively. The energy efficiency decreases with

an increase in the gasification temperature. This is because a large portion of the sensible enthalpy obtained from the chemical energy of the biomass is lost during the gas cleaning process (state 2G to 3). The energy required to vaporize the moisture in the biomass is also lost during the drying process. Hence, the gasification efficiency decreases with increase in the moisture content of the feedstock. For similar reasons, the exergy efficiency of the biomass also decreases with increase in the gasification temperature and moisture content of the feedstock. The results suggest that dry biomass gasified at the minimum possible temperature would result in maximum energy and exergy efficiency. The minimum possible temperature at which all the solid carbon in the biomass is converted to the gas phase is known as the carbon boundary point (CBP). However practical limitations such as tar condensation and slow kinetics lead to the operation of gasifiers above the CBP [49].

4.1.3. Operating conditions

For all remaining studies in this paper, the gasification temperature has been kept constant at 1073 K [38] that corresponds to an air-fuel ratio of 1.84. The moisture content in the biomass is assumed to be 20% by mass. These values are typical of downdraft gasifiers and can be assumed to be constant unless otherwise stated. The output from the gasifier under the stated operating conditions after gas cleanup (state 3) is shown in Table 8.

4.2 Catalytic combustion

The syngas obtained from the gasifier with properties corresponding to those mentioned in Table 8 is mixed with air and partially oxidized in the catalytic reactor. Figure 8 shows the variation of the outlet temperature of the reactor (T_8) as a function of the catalytic equivalence ratio (Φ_{cat}) for a constant pressure ratio (P_7/P_6). The increase in pressure ratio increases the inlet temperature (T_7) to the reactor and consequently increases the outlet temperature. The increase in the equivalence ratio decreases the outlet temperature due to a reduction in the O_2 content. However, the temperature decrease is more gradual at higher equivalence ratios. At lower equivalence ratios, the temperature in the reactor can exceed the thermal limit of ceramic substrates, which is around 1200 K. As observed from Fig. 8, an equivalence ratio below 3 is undesirable for continuous operation of the reactor. The suitable operating range would be $1000\text{ K} \leq T_8 \leq 1200\text{ K}$. The lower limit of 1000 K has been set considering that the autoignition temperature of H_2 , CO and CH_4 is usually in the range of 800 – 1000 K and hence suitable for MILD combustion.

The conversion percentage of the reacting components in the catalytic reactor as a function of the equivalence ratio (Φ_{cat}) at $P_7/P_6 = 3$ is shown in Figure 9. The conversion percentage has been defined as

$$\text{Conversion \%} = \left[\frac{(n_{i,in} - n_{i,out})}{n_{i,in}} \right] \times 100 \quad (32)$$

where n refers to the number of moles and i refers to species such as CO, H₂, and CH₄. At low Φ_{cat} (high temperature and high O₂), the conversion of methane to CO, H₂, CO₂, and H₂O is 100%. However, methane is produced at higher Φ_{cat} from the methane forming reaction $\text{CO} + 3\text{H}_2 \leftrightarrow \text{CH}_4 + \text{H}_2\text{O (g)}$ (R₃, equation 12). This can also be observed from the equilibrium constant K_3 (inset in Figure 9) that increases with reduction in temperature and signifies the dominance of the forward reaction. Hydrogen is consumed more as compared to CO because of the presence of CO₂ in the syngas. The output composition from the catalytic reactor influences the combustion characteristics in the MILD combustor. A higher H₂ content favours auto-ignition of the mixture in the MILD combustor.

4.3 MILD Combustion

The variation of the temperature at the outlet of MILD combustion (T_{10}) as a function of pressure ratio (P_7/P_6) at a constant overall equivalence ratio (Φ_{ov}) is shown in Figure 10. The temperature increases with increase in Φ_{ov} due to lesser amount of excess air in the exhaust. However, the decrease in temperature with increase in pressure ratio is counter-intuitive as the inlet temperature T_9 increases with the pressure ratio. This is observed due to the regeneration of thermal energy in the cycle. At low-pressure ratio, the temperature drop in the turbine ($T_{10} - T_{11}$) is reduced. The excess enthalpy after the SCO₂ cycle (T_{12}) is used for preheating the air to higher temperatures (T_{14}) that leads to an increase in T_{10} . At higher pressure ratios, the enthalpy drop in the turbine is higher and causes a reduction in T_{12} and T_{14} . This reduction in the air preheat temperature causes a reduction in the outlet temperature T_{10} . However in all the cases, the temperature is less than 1800 K, beyond which thermal NO_x emissions increase significantly [46]. This low temperature is also suitable from turbine blade material considerations. A micro gas turbine without turbine blade cooling can sustain a maximum temperature of around 1223 K, whereas turbines with ceramic heat exchanger materials may withstand inlet temperatures up to 1573 K [36].

4.4 Supercritical CO₂ Brayton cycle

The performance of the supercritical CO₂ Brayton cycle has been analyzed in this section. The important parameter from the point of this study is the inlet temperature to the heat exchanger T_{11} (or $T_{3s} = T_{11} - 10$). While the authors in an earlier work [6] restricted the maximum cycle temperature to 1000 K, it has been extended to 1300 K in the present analysis for the purpose of conducting parametric studies. The model for the SCO₂ cycle has been validated with the cycle thermal efficiency data reported by Garg et al. [6] as shown in Fig. 11. The results match reasonably well with the reported data.

4.4.1 Performance

The optimum expansion ratio (P_{3s}/P_{4s}) for the SCO_2 cycle is shown as a function of low side pressure (P_{1s}) for constant inlet temperature (T_{11}) in Fig. 12. The pressure ratio is said to be optimum when the cycle thermal efficiency is the maximum for a given low side pressure. The optimum expansion ratio increases with low side pressure when the maximum cycle pressure is not constrained (denoted by markers). However, when the maximum cycle pressure is restricted to 300 bar, the optimum expansion ratio curves follow the locus of $P_{max} = 300$ bar beyond a certain value of low side pressure. This is because the optimum value is greater than P_{max}/P_{1s} , as observed in Fig. 12 (markers), and the maximum allowed expansion ratio (that also happens to be the optimum expansion ratio) is $300/P_{1s}$, when P_{1s} is expressed in bar. The variation of cycle thermal efficiency with low side pressure is shown in Fig. 13. There is no significant difference between the efficiency with $P_{max} = 300$ bar (denoted by lines) and the cases in which P_{max} is unrestricted (denoted by markers). The deviation can be observed only at higher values of low side pressure. This also implies that the gain associated with higher expansion ratio is not justified as it causes marginal increase in the cycle efficiency. The cycle efficiency can also be observed to increase with inlet temperature.

4.5 Performance of IGSCC

The results obtained from the performance assessment of the major components of the cycle are now combined to evaluate the overall performance of the IGSCC in terms of energy and exergy efficiencies. The performance of the IGSCC is dependent on the performance of the individual units. The variation of the energy and exergy efficiency of the IGSCC as a function of the pressure ratio for a constant overall equivalence ratio is shown in Figs. 14 and 15, respectively. Two significant characteristics can be observed in the efficiency curves. First, the efficiency increases with increase in the overall equivalence ratio at a constant pressure ratio. This can be attributed to the higher operating temperature of the hybrid heating source that increases the inlet temperature to the SCO_2 cycle. The SCO_2 cycle efficiency increases with an increase in the inlet temperature, as shown earlier in Fig. 13. Secondly, the efficiency reaches a maximum at an intermediate pressure ratio that appears to be optimum. In this case, a maximum thermal efficiency of 36.5% and maximum exergy efficiency of 32.1% have been obtained at a pressure ratio of 2.75 and $\Phi_{ov} = 0.22$. The exergy efficiency curves follow a similar trend as that of the energy efficiency.

The point of maximum efficiency implies a balance between the power generated by the hybrid heating source and the SCO_2 cycle. These two together contribute to the total power output of the IGSCC. An analysis of the work output from the turbine and SCO_2 cycle revealed that the turbine produced lesser power than the SCO_2 cycle at low pressure ratio. This was due to the small temperature drop in the turbine and higher temperature of state 11 that increased the thermal efficiency of the SCO_2 cycle. The reverse trend was observed at higher pressure ratios where the SCO_2 cycle efficiency decreased, as shown in Fig. 16. The maximum cycle efficiency was reached at

an intermediate pressure ratio where the output from both the turbine and the SCO₂ cycle was high. It must be mentioned here that the temperature of the exhaust (T_{13}) increased monotonically with an increase in the pressure ratio and overall equivalence ratio. The exhaust temperature was 445.9 K for pressure ratio 2, $\Phi_{ov} = 0.16$ and 539.1 K for pressure ratio 5, $\Phi_{ov} = 0.22$. This excess enthalpy can be utilized for drying of biomass and improvement of the cycle performance.

4.6 Effect of gasification on the IGSCC performance

Until now, the cycle performance has been analyzed considering the gasifier to operate at a fixed temperature of 1073 K using biomass with 20% moisture content. Figure 17 shows the variation of IGSCC thermal efficiency and SCO₂ cycle efficiency with gasification temperature for two different values of moisture content in the biomass. The pressure ratio has been fixed at 2.75 and $\Phi_{ov} = 0.22$. The overall thermal efficiency decreases with increase in moisture content and gasification temperature due to the decrease in gasification efficiency. The decrease in SCO₂ cycle efficiency is less sensitive to the gasification parameters as compared to the IGSCC. The reduction in the calorific value of the fuel at higher gasification temperature and moisture content decreases the peak operating temperature in the SCO₂ cycle. For a gasification temperature of 973 K and using biomass with 0% moisture, a maximum overall efficiency of 40.8% can be achieved. The corresponding exergy efficiency obtained is 36%.

4.6 Exergy analysis of individual components

Improvement in the cycle performance can be achieved by determining the sources of exergy destruction. The total exergy budget of the cycle as a fraction of the input exergy (state 1G + state 1) at a pressure ratio of 2.75 and $\Phi_{ov} = 0.22$ is shown in Figure 18. The gasifier is operated at $T_g = 1073$ K with biomass containing 20% moisture. It is observed that the gasification system involves the maximum exergy destruction, followed by MILD combustion, SCO₂ cycle, the catalytic reactor, and the regenerator. It must be mentioned here that 8.72% of the total input exergy is lost in the exhaust (state 13). The gasifier performance can be improved by utilizing the exhaust to reduce the biomass moisture content. The gasifier exergy efficiency improves by 5% and cycle exergy efficiency improves by 2.15% when biomass moisture content is reduced from 20% to 0%. The utilization of the exhaust has reasonable influence on the cycle performance as it not only improves gasification efficiency but also reduces the exergy wasted through the exhaust.

The irreversibility in MILD combustion is due to the conversion of chemical exergy of the fuel into the physical exergy of the products. The high exergy destruction (12.6%) can be attributed to the low equivalence ratio that causes a reduction in the outlet temperature T_{10} . Although the exergy destruction can be reduced by increasing the overall equivalence ratio, the value is restricted due to constraints on the maximum turbine inlet temperature. Use of turbines with ceramic blades and

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internal cooling arrangement can improve the performance at the cost of higher investment and operation complexities. The exergy destruction in the SCO_2 cycle (10.5%) is due to the irreversibility in the individual components such as the turbine, compressor, regenerator and gas cooler. The turbine and compressor isentropic efficiencies have been assumed to be 75% and 80%, respectively. The improvement in these parameters will increase the exergy efficiency of the SCO_2 cycle and the overall cycle as well. The catalytic combustion process has relatively low exergy destruction because of the high equivalence ratio that causes fractional conversion of the fuel into products. All other components have relatively low exergy destruction ($\leq 2.5\%$). The exergy of the biomass is recovered as work output from both the turbine and the SCO_2 cycle. The ratio of exergy recovery varies with the pressure ratio. At pressure ratio of 2, the ratio of the exergy recovered by the SCO_2 cycle and the turbine is 1.16, whereas, at a pressure ratio of 5, the ratio is 0.45. At the point of maximum efficiency, this value is equal to 0.75.

4.7 Performance comparison with other cycles

Since the main source of energy in the IGSCC is biomass, its performance is compared with those of b-IGCC and CSP-biomass hybrid cycles. Table 9 shows the performance comparison of IGSCC cycle with a few other cycles from the literature. It is observed that the IGSCC with a maximum thermal efficiency of 40.8% outperforms other cycles by a significant margin. Also, it is close to the maximum biomass to electricity conversion efficiency of 43%. It must be mentioned that the performance of the IGSCC will vary depending on the selection of the modeling parameters such as the turbine and compressor efficiencies, pressure drop and heat loss. The high efficiency of the IGSCC can be attributed to the SCO_2 cycle that allows it to operate at higher temperatures. This is also favorable from the exergy utilization perspective as the combustion systems can supply energy at a temperature higher than the operating temperature of CSP technologies. In comparison, the parabolic trough supplying to a steam-Rankine cycle can operate at a maximum temperature of around 700 K and thus offers a lower efficiency. The variety of gasification and CSP technologies available at present offers numerous possible configurations of CSP-biomass hybridization.

5. Conclusions

A syngas-fired hybrid heating source has been proposed for CSP-biomass hybridization to enable continuous operation of solar thermal power plants. This concept essentially involves the use of biomass-derived syngas in a novel two-stage low temperature combustion system with potential to provide stable combustion of syngas with near-zero NO_x emissions. A thermodynamic analysis has been conducted to study the performance of the IGSCC from the first and second law considerations to determine the optimum use of the syngas derived from biomass gasification. The gasification temperature, biomass moisture content, equivalence ratio, and pressure ratio are the main factors considered for parametric investigations. It was observed that a decrease in the gasification

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temperature and biomass moisture content increases the cycle efficiency by improving the gasification efficiency. The temperature of catalytic combustion and MILD combustion were restricted due to material constraints of the catalyst substrate and turbine blade, respectively. The efficiency of the SCO_2 cycle increased with increase in the inlet temperature to the heat exchanger and reached a maximum value of 44.4% at 1300 K. Based on the constraints imposed on the IGSCC, a maximum thermal efficiency of 36.5% was achieved at $\Phi_{\text{ov}} = 0.22$ and pressure ratio of 2.75 when the gasifier was operating at $T_g = 1073$ K with biomass containing 20% moisture. The efficiency increased to 40.8% when dry biomass was gasified at a temperature of 973 K. The exergy analysis revealed that the maximum exergy destruction occurred in the gasifier, followed by MILD combustion, SCO_2 cycle, catalytic combustion, and regenerator. The exergy analysis also predicted that 8.72% of the total exergy was lost in the exhaust and could be used for drying of biomass. The higher efficiency of the present IGSCC concept in comparison with other CSP-biomass hybrids can be attributed to the high efficiency associated with the SCO_2 cycle.

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7. References

- [1] S. Kuravi, J. Trahan, D.Y. Goswami, M.M. Rahman, E.K. Stefanakos, Thermal energy storage technologies and systems for concentrating solar power plants, *Prog. Energy Combust. Sci.* 39 (2013) 285–319. doi:10.1016/j.pecs.2013.02.001.
- [2] B. Xu, P. Li, C. Chan, Application of phase change materials for thermal energy storage in concentrated solar thermal power plants: A review to recent developments, *Appl. Energy*. 160 (2015) 286–307. doi:10.1016/j.apenergy.2015.09.016.
- [3] Y. Li, Y. Yang, Thermodynamic analysis of a novel integrated solar combined cycle, *Appl. Energy*. 122 (2014) 133–142. doi:10.1016/j.apenergy.2014.02.017.
- [4] M. Liu, N.H. Steven Tay, S. Bell, M. Belusko, R. Jacob, G. Will, et al., Review on concentrating solar power plants and new developments in high temperature thermal energy storage technologies, *Renew. Sustain. Energy Rev.* 53 (2016) 1411–1432. doi:10.1016/j.rser.2015.09.026.

- [5] E.J. Sheu, A. Mitsos, A. a. Eter, E.M. a. Mokheimer, M. a. Habib, A. Al-Qutub, A Review of Hybrid Solar–Fossil Fuel Power Generation Systems and Performance Metrics, *J. Sol. Energy Eng.* 134 (2012) 041006–041006. doi:10.1115/1.4006973.
- [6] P. Garg, P. Kumar, K. Srinivasan, Supercritical carbon dioxide Brayton cycle for concentrated solar power, *J. Supercrit. Fluids.* 76 (2013) 54–60. doi:10.1016/j.supflu.2013.01.010.
- [7] A. Kribus, R. Zaibel, D. Carey, A. Segal, J. Karni, A solar-driven combined cycle power plant, *Sol. Energy.* 62 (1998) 121–129. doi:10.1016/S0038-092X(97)00107-2.
- [8] A. Kribus, P. Doron, R. Rubin, J. Karni, R. Reuven, S. Duchan, et al., A MULTISTAGE SOLAR RECEIVER: THE ROUTE TO HIGH TEMPERATURE, *Sol. Energy.* 67 (2000) 3–11. doi:10.1016/S0140-6701(01)80433-1.
- [9] N.R.E.L. Website, http://www.nrel.gov/csp/solarpaces/by_country.cfm accessed on 17/02/2016.
- [10] J. Dersch, M. Geyer, U. Herrmann, S.A. Jones, B. Kelly, R. Kistner, et al., Trough integration into power plants-a study on the performance and economy of integrated solar combined cycle systems, *Energy.* 29 (2004) 947–959. doi:10.1016/S0360-5442(03)00199-3.
- [11] A. Evans, V. Strezov, T.J. Evans, Sustainability considerations for electricity generation from biomass, *Renew. Sustain. Energy Rev.* 14 (2010) 1419–1427. doi:10.1016/j.rser.2010.01.010.
- [12] A. V Bridgwater, A.J. Toft, J.G. Brammer, A techno-economic comparison of power production by biomass fast pyrolysis with gasification and combustion, *Renew. Sustain. Energy Rev.* A.V. Bridg. Al. / *Renew. Sustain. Energy Rev.* 6 (2002) 181–248. doi:10.1016/S1364-0321(01)00010-7.
- [13] K.R. Craig, M.K. Mann, Cost and Performance Analysis of Biomass-Based Integrated Gasification Combined-Cycle (BIGCC) Power Systems, *Biomass.* (1996). <http://www.nrel.gov/docs/legosti/fy97/21657.pdf>
- [14] W. Alonso-Pippo, C.A. Luengo, J. Koehlinger, P. Garzone, G. Cornacchia, Sugarcane energy use: The Cuban case, *Energy Policy.* 36 (2008) 2163–2181. doi:10.1016/j.enpol.2008.02.025.
- [15] E.S. Rubin, C. Chen, A.B. Rao, Cost and performance of fossil fuel power plants with CO₂ capture and storage, *Energy Policy.* 35 (2007) 4444–4454. doi:10.1016/j.enpol.2007.03.009.
- [16] R.M. Sáez, P. Linares, J. Leal, Assessment of the externalities of biomass energy, and a comparison of its full costs with coal, *Biomass and Bioenergy.* 14 (1998) 469–478. doi:10.1016/S0961-9534(98)00016-6.

- [17] D. Styles, M.B. Jones, Energy crops in Ireland: Quantifying the potential life-cycle greenhouse gas reductions of energy-crop electricity, *Biomass and Bioenergy*. 31 (2007) 759–772. doi:10.1016/j.biombioe.2007.05.003.
- [18] J.H. Peterseim, A. Herr, S. Miller, S. White, D.A. O’Connell, Concentrating solar power/alternative fuel hybrid plants: Annual electricity potential and ideal areas in Australia, *Energy*. 68 (2014) 698–711. doi:10.1016/j.energy.2014.02.068.
- [19] R. Soria, J. Portugal-Pereira, A. Szklo, R. Milani, R. Schaeffer, Hybrid concentrated solar power (CSP)-biomass plants in a semiarid region: A strategy for CSP deployment in Brazil, *Energy Policy*. 86 (2015) 57–72. doi:10.1016/j.enpol.2015.06.028.
- [20] J.H. Peterseim, A. Tadros, S. White, U. Hellwig, J. Landler, K. Galang, Solar tower-biomass hybrid plants - Maximizing plant performance, in: *Energy Procedia*, 2013: pp. 1197–1206. doi:10.1016/j.egypro.2014.03.129.
- [21] J.D. Nixon, P.K. Dey, P.A. Davies, The feasibility of hybrid solar-biomass power plants in India, *Energy*. 46 (2012) 541–554. doi:10.1016/j.energy.2012.07.058.
- [22] J.H. Peterseim, S. White, A. Tadros, U. Hellwig, Concentrating solar power hybrid plants - Enabling cost effective synergies, *Renew. Energy*. 67 (2014) 178–185. doi:10.1016/j.renene.2013.11.037.
- [23] M.B. da Fonseca, W.-R. Poganietz, H.-J. Gehrmann, Environmental and economic analysis of SolComBio concept for sustainable energy supply in remote regions, *Appl. Energy*. 135 (2014) 666–674. doi:10.1016/j.apenergy.2014.07.057.
- [24] J.H. Peterseim, U. Hellwig, A. Tadros, S. White, Hybridisation optimization of concentrating solar thermal and biomass power generation facilities, *Sol. Energy*. 99 (2014) 203–214. doi:10.1016/j.solener.2013.10.041.
- [25] T. Srinivas, B. V. Reddy, Hybrid solar-biomass power plant without energy storage, *Case Stud. Therm. Eng.* 2 (2014) 75–81. doi:10.1016/j.csite.2013.12.004.
- [26] Y. Tanaka, S. Mesfun, K. Umeki, A. Toffolo, Y. Tamaura, K. Yoshikawa, Thermodynamic performance of a hybrid power generation system using biomass gasification and concentrated solar thermal processes, *Appl. Energy*. 160 (2015) 664–672. doi:10.1016/j.apenergy.2015.05.084.
- [27] S. Etemad, L. L. Smith, K. Burns, System study of rich catalytic/lean burn (RCL) catalytic combustion for natural gas and coal-derived syngas combustion turbines. DOE final report DE-FG26-02NT41521, DOE, Precision Combustion (2004).

- [28] L.L. Smith, H. Karim, M.J. Castaldi, S. Etemad, W.C. Pfefferle, Rich-Catalytic Lean-burn combustion for fuel-flexible operation with ultra low emissions, *Catal. Today*. 117 (2006) 438–446. doi:10.1016/j.cattod.2006.06.021.
- [29] L.L. Smith, H. Karim, M.J. Castaldi, S. Etemad, W.C. Pfefferle, V. Khanna, et al., Rich-Catalytic Lean-Burn Combustion for Low-Single-Digit NO_x Gas Turbines, *J. Eng. Gas Turbines Power*. 127 (2005) 27. doi:10.1115/1.1787510.
- [30] J. Wüning, Flameless oxidation to reduce thermal NO_x-formation, *Prog. Energy Combust. Sci.* 23 (1997) 81–94. doi:10.1016/S0360-1285(97)00006-3.
- [31] A. Cavaliere, M. De Joannon, Mild combustion, *Prog. Energy Combust. Sci.* 30 (2004) 329–366. doi:10.1016/j.peecs.2004.02.003.
- [32] S. Kruse, B. Kerschgens, L. Berger, E. Varea, H. Pitsch, Experimental and numerical study of MILD combustion for gas turbine applications, *Appl. Energy*. 148 (2015) 456–465. doi:10.1016/j.apenergy.2015.03.054.
- [33] T. Zornek, T. Monz, M. Aigner, Performance analysis of the micro gas turbine Turbec T100 with a new FLOX-combustion system for low calorific fuels, *Appl. Energy*. 159 (2015) 276–284. doi:10.1016/j.apenergy.2015.08.075.
- [34] P.F. Li, J.C. Mi, B.B. Dally, F.F. Wang, L. Wang, Z.H. Liu, et al., Progress and recent trend in MILD combustion, *Sci. China Technol. Sci.* 54 (2011) 255–269. doi:10.1007/s11431-010-4257-0.
- [35] V.K. Arghode, A.K. Gupta, K.M. Bryden, High intensity colorless distributed combustion for ultra low emissions and enhanced performance, *Appl. Energy*. 92 (2012) 822–830. doi:10.1016/j.apenergy.2011.08.039.
- [36] A. Datta, R. Ganguly, L. Sarkar, Energy and exergy analyses of an externally fired gas turbine (EFGT) cycle integrated with biomass gasifier for distributed power generation, *Energy*. 35 (2010) 341–350. doi:10.1016/j.energy.2009.09.031.
- [37] A. Bhattacharya, D. Manna, B. Paul, A. Datta, Biomass integrated gasification combined cycle power generation with supplementary biomass firing: Energy and exergy based performance analysis, *Energy*. 36 (2011) 2599–2610. doi:10.1016/j.energy.2011.01.054.
- [38] Z.A. Zainal, R. Ali, C.H. Lean, K.N. Seetharamu, Prediction of performance of a downdraft gasifier using equilibrium modeling for different biomass materials, *Energy Convers. Manag.* 42 (2001) 1499–1515. doi:10.1016/S0196-8904(00)00078-9.

- [39] S. Dasappa, D.N. Subbukrishna, K.C. Suresh, P.J. Paul, G.S. Prabhu, Operational experience on a grid connected 100kWe biomass gasification power plant in Karnataka, India, *Energy Sustain. Dev.* 15 (2011) 231–239. doi:10.1016/j.esd.2011.03.004.
- [40] S. Cimino, G. Landi, L. Lisi, G. Russo, Rh-La(Mn,Co)O₃ monolithic catalysts for the combustion of methane under fuel-rich conditions, *Catal. Today.* 117 (2006) 454–461. doi:10.1016/j.cattod.2006.06.016.
- [41] <http://navier.engr.colostate.edu/~dandy/code/code-4/> accessed on 17/02/2016.
- [42] Y.A. Cengel, M.A. Boles, *THERMODYNAMICS: AN ENGINEERING APPROACH*, 2015. doi:10.1017/CBO9781107415324.004.
- [43] J. Szargut, D. R. Morris, F. R. Steward, *Exergy analysis of thermal, chemical, and metallurgical processes* (1987).
- [44] A.P. Simpson, A.E. Lutz, Exergy analysis of hydrogen production via steam methane reforming, *Int. J. Hydrogen Energy.* 32 (2007) 4811–4820. doi:10.1016/j.ijhydene.2007.08.025.
- [45] M.W. Chase, *NIST-JANAF Thermochemical Tables*, 1998.
- [46] S.R. Turns, *An introduction to combustion: concepts and applications*, System. 499 (2000) 411. doi:10.1016/j.ijhydene.2008.07.121.
- [47] E.W. Lemmon, M.L. Huber, M.O. McLinden, *NIST Standard Reference Database 23: Reference Fluid Thermodynamic and Transport Properties (REFPROP), Version 9.0*, Phys. Chem. Prop. (2010). <http://www.mendeley.com/research/nist-standard-reference-database-23-reference-fluid-thermodynamic-transport-properties-refprop-versi/\nhttp://www.nist.gov/srd/upload/REFPROP9.pdf>.
- [48] T.H. Jayah, L. Aye, R.J. Fuller, D.F. Stewart, Computer simulation of a downdraft wood gasifier for tea drying, *Biomass and Bioenergy.* 25 (2003) 459–469. doi:10.1016/S0961-9534(03)00037-0.
- [49] R. Karamarkovic, V. Karamarkovic, Energy and exergy analysis of biomass gasification at different temperatures, *Energy.* 35 (2010) 537–549. doi:10.1016/j.energy.2009.10.022.

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Figure 2: Schematic diagram of the IGSCC including the SCO₂ cycle. The portion of the cycle without the SCO₂ cycle forms the hybrid heating source. *Abbreviations:* Comp. – compressor, SCO₂ – supercritical CO₂, MILD – moderate or intense low-oxygen dilution

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Project	Location	Technology	MWe	Solar MWe	Solar %	Status
Kuraymat	Egypt	Parabolic trough	140	20	14.29	Operational
Hassi R'Mel	Algeria	Parabolic trough	150	20	13.33	Operational
Ain Beni M.	Morocco	Parabolic trough	470	20	4.25	Operational
Agua Prieta	Mexico	Parabolic trough	478	14	2.92	Planned

Table 1. Solar energy component in some existing and planned ISCC plants [9]. The solar percentage has been defined as (Solar MWe/ Total MWe) x 100%.

GASIFICATION SYSTEM [36 – 38]	
Gasifier type	Downdraft
Gasification temperature	973 – 1323 K
Biomass	Wood
Formula	$\text{CH}_{1.44}\text{O}_{0.66}$
Moisture content	0 – 30%
LHV of biomass (dry)	417888 kJ/kmol
Ultimate analysis (dry basis, weight percentage)	50% C, 6% H, 44% O
Exit from gas cleaner (T_3 , P_3)	298.15 K, 0.1 MPa
Moisture content at state 3	0%
COMBUSTION SYSTEM	
Equivalence ratio (catalytic comb.)	2 – 6.5
Overall equivalence ratio	0.16 – 0.22
Pressure drop (each combustion chamber)	3% of inlet pressure [3]
Heat loss	Adiabatic
AUXILIARIES	
Turbine isentropic efficiency	89% [3]
Compressor isentropic efficiency	85% [3]
Pressure ratio ($P_7/P_6 = P_9/P_5$)	2 – 5

Table 2. Parameters of the hybrid heating source.

Supercritical CO₂ Brayton cycle [6]	
Critical point properties of CO ₂	304.2 K, 7.39 MPa
Minimum cycle temperature, T _{1s}	308 K
Maximum cycle temperature, T _{3s}	700 – 1300 K
Expansion ratio, P _{3s} /P _{4s}	2 – 4
Low side pressure, P _{1s}	7.5 – 8.5 MPa
Turbine isentropic efficiency, $\eta_{t,isen}^S$	75%
Compressor isentropic efficiency, $\eta_{c,isen}^S$	80%
Pressure drop in heat exchanger	3.5% of inlet pressure
Leakage loss	$0.0003 * \sqrt{P_{3s} - P_{4s}}$
Min. temp. difference in heat exchanger (T _{6s} – T _{2s})	12 K

Table 3. Operating parameters of SCO₂ cycle.

Chemical species	Formula	Phase	Enthalpy of formation (kJ/kmol)
Water	H ₂ O	g	-241820
Water	H ₂ O	l	-285830
Methane	CH ₄	g	-74850
Hydrogen	H ₂	g	0
Oxygen	O ₂	g	0
Carbon monoxide	CO	g	-110530
Carbon dioxide	CO ₂	g	-393520
Nitrogen	N ₂	g	0

Table 4. Enthalpy of formation of various species [42].

Chemical species	Formula	Lower heating value (kJ/kmol)
Methane	CH ₄	800800
Hydrogen	H ₂	240000
Carbon monoxide	CO	282800

Table 5. Lower heating values of various fuel species [42].

Chemical species	Formula	Phase	Standard chemical exergy (kJ/kmol)
Water	H ₂ O	g	9500
Water	H ₂ O	l	900
Methane	CH ₄	g	831650
Hydrogen	H ₂	g	236100
Oxygen	O ₂	g	3970
Carbon monoxide	CO	g	275100
Carbon dioxide	CO ₂	g	19870
Nitrogen	N ₂	g	720

Table 6. Standard chemical exergy of various species [43].

Dry gas composition	Jayah et al. [48] Experiment	Datta et al. [36] Equilibrium analysis	Present model
Moisture content = 16%, A-F ratio = 2.2			
H ₂	18.3	19.0	19.2
CO	20.2	24.8	22.9
CO ₂	9.7	8.0	10.0
CH ₄	1.1	0.4	0.1
N ₂	50.7	47.9	47.8
Moisture content = 18.5%, A-F ratio = 2.03			
H ₂	17.2	20.9	21.6
CO	19.6	23.8	22.6
CO ₂	9.9	9.3	10.7
CH ₄	1.4	1.0	0.3
N ₂	51.9	45.1	44.8

Table 7: Comparison of the gasification model with the experimental results of Jayah et al. [48] and equilibrium model of Datta et al. [36].

Temperature	298.15 K
Pressure	0.1 MPa
Composition (vol. %)	23.72% H ₂ , 22.20% CO, 0.74% CH ₄ , 11.57% CO ₂ , 41.77% N ₂
H ₂ /CO ratio	1.07
LHV	125635.52 kJ/kmol (5.61 MJ/m ³)
$\eta_{I,g}$	87.10%
$\eta_{II,g}$	75.15%

Table 8. Properties of syngas produced from the gasifier (state 3).

Table 9: Comparison of thermal efficiency of IGSCC with other hybrid cycles.

Cycle	Reference	Thermal Efficiency (%)
Biomass to electricity	Evans et al. [11]	27%*
CSP + biomass	Srinivas et al. [25]	27%
b-IGCC + CSP	Tanaka et al. [26]	36.7%
IGSCC	Present study	40.8%

* Average of a range of efficiencies as reported by Evans et al. [11].