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# Biomass Based Indirectly Heated Combined Cycle Plant: Energetic and Exergetic Performance Analyses

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**Abstract**: Biomass based power plant can be an excellent option for community scale power generation to overcome the fossils fuel shortage and environmental issues. Thermal performance and sizing of an indirectly heated combined cycle plant employing topping gas turbine (GT) block and bottoming steam turbine (ST) block integrated with biomass gasifier is analysed in this paper. The plant is capable of producing topping 20 kWe fixed output along with bottoming steam turbine output. A mix of Olive tree leaves and pruning is considered as fuel feed. Novelty of the scheme exists in the heating process of the topping GT cycle, which is done in an external combustor-heat exchanger (CHX) unit rather than the conventional combustor of GT cycle. The plant configuration is modelled and analysed using Cycle-Tempo software. Both energetic and exergetic performances are analysed over a range of topping cycle pressure ratio (4-20) and for different turbine inlet air temperatures (TIT=900, 1000 and 1100 deg C). In the base case scenario (topping cycle pressure ratio=4 and TIT= 900 deg C), the system efficiency is about 30.5%. The plant attains maximum efficiency at pressure ratio range 6-9 depending upon the TIT. Required specific air flow, which indicates the size of the ducting and CHX unit of the topping cycle decreases with increase in pressure ratio and at higher TIT's. From the Second Law analysis it is found that the major exergy losses occur at the gasifier and combustor-heat exchanger units, together account for 44% of the fuel exergy value.

**Keywords**: Energy, Exergy, Indirectly Heated, Biomass gasification, Combined cycle

## I. INTRODUCTION

The energy consumptions in the developing countries like India, China, and Bangladesh etc are increasing rapidly due to increase in population as well as rapid industrialization and urbanization [1]. The total installed power generation capacity of India through different sources is 211.8 GW (as on January 2013) with a peak energy shortage of 12.2 GW [2]. The Indian power sector is strongly dependent on the fossil fuels. But the limitation of fossil fuel reserves and the adverse environmental effect of the fossil fuel combustion are the primary challenges to meet the demand of electricity. The energy researchers are paying more attention towards the development of technologies that are more reliable, economic and environment friendly, as the burning of fossils fuels is a major source of greenhouse gas emissions [1, 3]. Biomass based power generation is attaining huge possibilities to overcome the energy scarcity problems and environmental issues due to its availability and $CO_2$ neutrality, especially in the rural and hilly part of India. The estimated annual power production capacity from biomass is about 17.5 GW while the installed capacity as on March 2012 is only 3.4 GW in India [4].

Biomass gasification is the process to convert the solid biomass into combustible producer gas mixture in oxygen starved condition. The major components of this gas are $CH_4$, $H_2$, CO, $CO_2$, $H_2O$ and $N_2$. The producer gas used in internal combustion engines to produce decentralized electricity. However, the main problem with these systems is the cost of maintenance and offers low efficiency. Since the internal combustion (IC) engines are sensitive to the presence of tar, particles and humidity in the producer gas, additional cleaning and drying systems are required after the gasifiers. For the rural areas with inadequate training, the system could fail due to poor maintenance and operational procedure [5]. The overall efficiency of power production from biomass can be increased to 35-40% using gas turbine-steam turbine (GT-ST) based combined cycle, integrating a gasifier in the system [6]. The worlds' first bio-gasification based combined cycle power plant was operated during 1996 in Varnamo, Sweden with an overall efficiency of 32%. The

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operation of the plant was stopped due to high operation and maintenance cost [7] as extensive gas cleaning and cooling is required for the combustor and gas turbine for continuous operation.

The biomass direct fired micro gas turbine power system has the maintenance difficulties mentioned above. The indirect heating eliminates the producer gas cleaning/cooling requirement as the gas is not directly combusted after the gasifier [8]. Moreover, clean air is used to operate the turbine engine that reduces the maintenance requirements and extends the turbine operation life.

This paper presents the energetic and exergetic performance of bio-gasification based indirectly heated combined cycle plant. The plant provides topping fixed electrical output of 20 kW along with bottoming steam turbine output. The exhaust from combustor-heat exchanger and GT unit is used to operate a bottoming steam cycle. A fixed bed downdraft gasifier is used in the present study. The proposed systems have been modelled and analysed using Cycle-Tempo software [9], which was developed by TU Delft (Delft University of Technology) for thermodynamic analysis of power and refrigeration cycle.

## II. MODEL DEVELOPMENT

The schematic diagram of an indirectly fired combined cycle plant using biomass derived producer gas as fuel is shown in Fig. 1. Solid Biomass (mixture of olive leaves and purnings) is fed to a downdraft gasifier (block 6) to convert it into gaseous fuel. This conversion process is take places in the presence of atmospheric air (commonly called air gasification) in sub-stoichiometric condition. The gaseous fuel after passing through the air heater (block7-which in turn preheats the inlet air to the gasifier) enters in the combustion chamber (block 8).

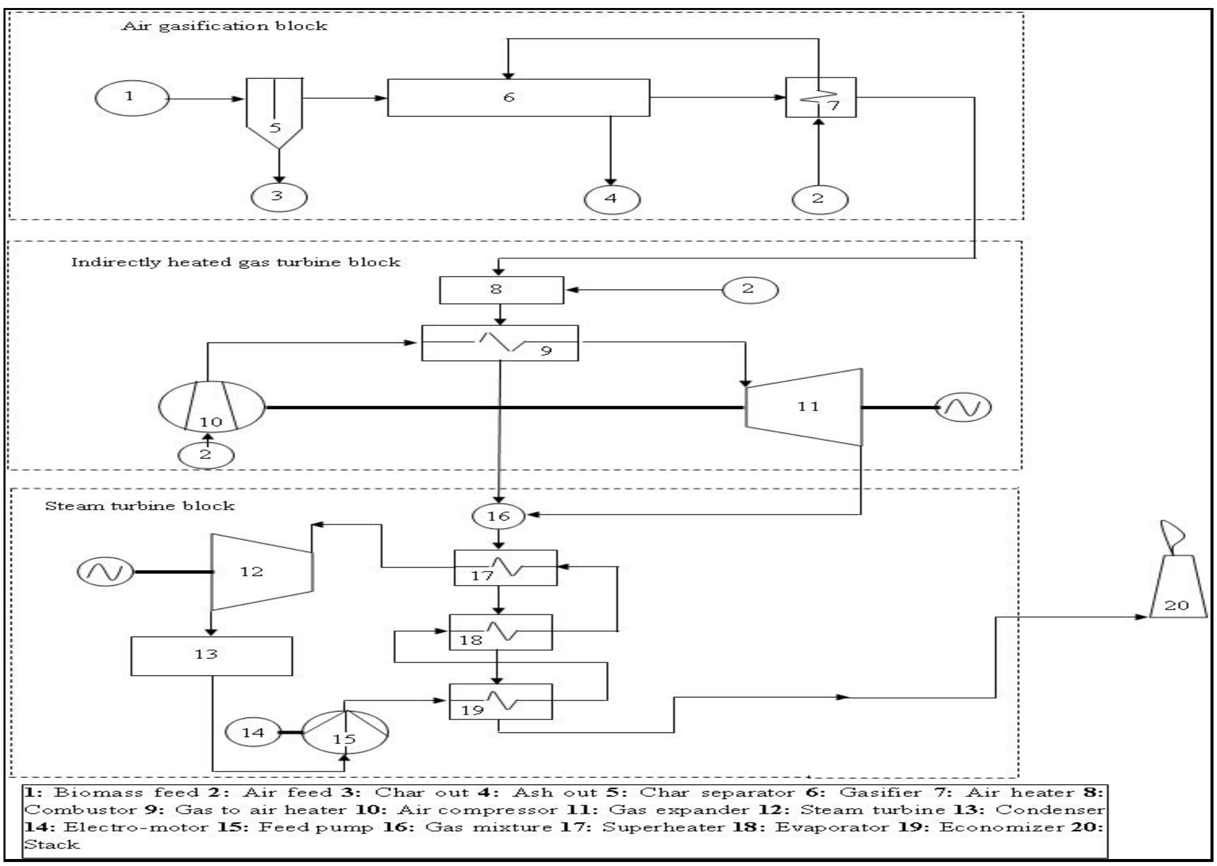


Fig. 1: Schematic diagram of the proposed plant.

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The producer gas gets combusted in the presence of atmospheric air (block 8) in the combustion chamber and flue gas is generated. The flue gas then enters into the heat exchanger (block 9) as hot fluid and flows through the shell side of it. Block 8 and block 9 together called as combustor-heat exchanger duplex (CHX) unit. The atmospheric air enters into the compressor (block 10) and after getting compressed enters into the heat exchanger and gets heated. The hot and compressed air then enters into the gas turbine (block 11). The hot air after expansion mixes with the exhaust from the heat exchanger at the gas mixture (block 16).

The bottoming steam cycle consisting of steam turbine (block 12), condenser (block 3) and feed pump (block 15) is used for power generation in combination with the heat recovery steam generator (HRSG) to recover exhaust heat of the topping cycle. Thus, topping GT cycle and bottoming ST cycle together operates as combined cycle. The HRSG consists of three sub-units of heat exchangers. The HRSG sub-units are superheater (SUP, block 17), evaporator (EVAP, block 18) and economizer (ECO, block 19) respectively. Minimum pinch point temperature (15 $^0$C) is maintained in the evaporator for better heat transfer between the water and the exhaust gas mixture. The exhaust gas mixture after passing through the HRSG unit is allowed to expose in the atmosphere through stack (block 22).

## III. MATHEMATICAL FORMULATION

### *A. Biomass Gasification*

Gasification is a thermo-chemical process to convert solid biomass feedstock (olive pit& purnings) into combustible synthetic gas in a sub-stoichiometric environment. The dry biomass fuel can be expressed by a generalized unified molecular formula $CH_aO_bN_cS_d$ where the subscripts a, b, c and d are determined from the ultimate analysis of biomass (Table I).

The global gasification reaction can be represented as follows:

$$CH_aO_bN_cS_d + m(O_2 + 3.76N_2) = X_1H_2 + X_2CO + X_3CO_2 + X_4H_2O + X_5CH_4 + X_6H_2S + X_7N_2 \quad (1)$$

Where $X_1$, $X_2$, $X_3$, $X_4$, $X_5$, $X_6$ and $X_7$ are the moles of $H_2$, CO, $CO_2$, $H_2O$, $CH_4$, $H_2S$ and $N_2$ respectively and m is air in moles.

TABLE I
CHARACTERISTICS OF OLIVE TREE LEAVES AND PURNINGS [10].

| **Parameter** | **Unit** | **Value** |
|---|---|---|
| *Ultimate Analysis* | *Mass percentage on wet basis* | |
| C | % | 47.1 |
| H | % | 6.18 |
| O | % | 41.66 |
| N | % | 0.55 |
| S | % | 0.1 |
| Ash | % | 4.2 |
| *LHV (MJ/kg)* | MJ/kg | 16.3 |
| Moisture | % | 7.2 |

The values of $X_1$ through $X_7$ are solved considering the carbon, hydrogen, oxygen, nitrogen and sulphur balance in the gasification reaction (Eq. (1)) as well as the chemical equilibrium of the water gas shift reaction and methane reaction [11] at the said temperature as follows:

$$\begin{aligned} CO + H_2O &= CO_2 + H_2 \\ C + 2H_2 &= CH_4 \end{aligned} \quad (2)$$

The assumptions made for gasification process are:

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- 5% (by mass) unconverted carbon (char) is assumed.
- Ashes are mainly consisting of $Al_2O_3$, $SiO_2$ and $Fe_2O_3$.
- Tar formation is not considered in this model.

Air blown fixed bed downdraft gasifier operating on atmospheric pressure is used in the present model. The bed temperature of the gasifier is $800^0$C. The gasification temperature is adjusted by controlling the oxidant (air)/biomass ratio $x_{OF}$ presented in Table III. To increase the gasification efficiency preheating of inlet air upto 275-$300^0$C is done (using gasifier cooling system). The efficiency of the gasifier is calculated as follows:

$$\eta_{gasi} = \frac{m_{p.g} LHV_{p.g}}{m_{biomass} LHV_{biomass}} \quad (3)$$

*B. Indirectly Heated Combined Cycle Modeling*

Although the configurations consider biomass based combined cycle plants, they differ from the conventional biomass integrated gasification combined cycle (BIGCC) plant in the process of combustion and heat transfer. The biomass-derived producer gas undergoes combustion in the combustor-heat exchanger duplex unit, which in turn heats the working medium of the GT cycle (air) through a high-temperature high-pressure (tube side) heat exchanger.
Assuming complete combustion and that of same for the major combustible gas in the producer gas mixture, the following chemical reaction takes place at the combustor:

$$\begin{aligned} X_1H_2 + X_2CO + X_3CO_2 + X_4H_2O + X_5CH_4 + X_6H_2S + X_7N_2 + m'(O_2 + 3.76N_2) &= X_8CO_2 + X_9H_2O \\ &+ (X_7 + 3.76m')N_2 + X_{10}O_2 + X_{11}SO_2 \end{aligned} \quad (4)$$

Where, the coefficients $X_1$ to $X_7$ are obtained from the producer gas composition and *m'* denotes the number of oxygen moles supplied for firing of producer gas. The coefficients $X_8$ to $X_{11}$ denote the moles of flue gas.
The post combustion temperature is calculated assuming an adiabatic combustion at the combustor:

$$\sum_j X_j(\overline{h_{fj}^o} + \Delta\overline{h})_{producergas} + \sum_j X_j(\overline{h_{fj}^o} + \Delta\overline{h})_{air} = \sum_j X_j(\overline{h_{fj}^o} + \Delta\overline{h})_{fluegas} \quad (5)$$

TABLE II
OPERATING PARAMETER OF THE PLANT

| Parameter | Unit | Value |
|---|---|---|
| Isentropic efficiency of compressor | % | 90 |
| Isentropic efficiency of GT | % | 90 |
| Oxidant-fuel ratio at gasifier | ---- | 1.8 |
| Equivalence ratio of combustors | ---- | 2.1 |
| Turbine inlet temperatures | $^0$C | 900, 1000, 1100 |
| Topping cycle power output | kW | 20 |
| ST cycle operating pressure | bar | 5 |
| ST cycle operating temperature | $^0$C | 350 |
| Isentropic efficiency of ST | % | 0.85 |
| Condenser pressure | bar | 0.1 |
| HRSG minimum pinch point temperature difference | $^0$C | 15 |

Heat exchange through the heat exchanger (block 9) is calculated as:

$$X_g = X_7 + X_8 + X_9 + X_{10} + X_{11} + 3.76m' \quad (6)$$

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The atmospheric air enters into the compressor and the specific work requirement is calculated as:

$$w_C = c_{p,a}(T_{c,o} - T_{c,i}) \qquad (7)$$

The specific work output from the GT is calculated as:

$$w_{GT} = c_{p,a}(T_{GT,i} - T_{GT,o}) \qquad (8)$$

The net power output from the topping cycles are considered to be 20 kW. The required air flow through the topping cycle is calculated as:

$$w_{net} = (w_{GT} - w_c)\eta_G \qquad (9)$$

The turbine outlet temperature is calculated using compressor pressure ratio (Π) and isentropic efficiency of the turbine.

The total exhaust gas flow rate to the HRSG and temperature after mixing process is calculated from the following mass and energy balance equations:

$$\begin{aligned} m_{f.g,m} &= m_f + m_a \\ m_f c_{p,f}\Delta T + m_a c_{p,a}\Delta T &= m_{f.g,m} c_{p,f.g,m}\Delta T \end{aligned} \qquad (10)$$

The steam generation rate at the HRSG is calculated as:

$$m_{f.g,m} C_{p.f.g,m}\Delta T = m_s \Delta h \qquad (11)$$

Where, $\Delta T$ represents the temperature difference of hot streams entering into HRSG and $\Delta h$ is the enthalpy difference ST inlet steam and pump outlet water.

The specific steam turbine power output is calculated as:

$$w_{ST} = (h_{ST,i} - h_{ST,o})\eta_G \qquad (12)$$

The specific work requirement for the pump is calculated as:

$$w_p = (h_{p,o} - h_{p,i})\eta_p \qquad (13)$$

Net power output from the combined cycle is calculated as:

$$w_{net} = (w_{GT} - w_c)\eta_G + (w_{ST} - w_p) \qquad (14)$$

The First Law efficiency of the combined cycle is calculated as:

$$\eta_{CC} = \frac{w_{net}}{m_{biomass} LHV_{biomass}} \qquad (15)$$

*C. Exergy Analysis*

Exergy values of all components in the plant are based on pressure, temperature and chemical compositions. The thermo-mechanical exergy is defined with respect to temperature and pressure of dead state. The specific thermo-mechanical flow of exergy at any state of the cycle is defined by a generalized equation as follows:

$$e_i = (h_i - h_o) - T_o(s_i - s_o) \qquad (16)$$

Where, *i* represent the state point at which exergy is evaluated and *o* at the exergy reference environment. Now,

(17)

$$h_i - h_o = \int_{T_o}^{T_i} c_p dT$$

$$s_i - s_o = \int_{T_o}^{T_i} c_p \frac{dT}{T} - \bar{R} \ln \frac{P_i}{P_o}$$

The chemical exergy is calculated separately via providing an environmental definition since power cycles involves gasifier and combustor. The fuel exergy is calculated from the following equation:

$$Ex_{fuel} = m_{biomass} LHV_{biomass} \beta \quad (18)$$

The multiplication factor ($\beta$) of the above equation [12] is calculated as:

$$\beta = \frac{1.044 + 0.0160\frac{H}{C} - 0.34493\frac{O}{C}(1 + 0.0531\frac{H}{C})}{1 - 0.4124\frac{O}{C}} \quad (19)$$

The specific chemical exergy of the producer gas is calculated as:

$$e_B^{ch} = \frac{X_1}{X_1 + X_2 + X_3 + X_4 + X_5 + X_8} e_{H_2}^{ch} + \frac{X_2}{X_1 + X_2 + X_3 + X_4 + X_5 + X_8} e_{CO}^{ch} + \frac{X_5}{X_1 + X_2 + X_3 + X_4 + X_5 + X_8} e_{CH_4}^{ch} \quad (20)$$

$e_{H2}^{ch}$ $e_{CO}^{ch}$ and $e_{CH_4}^{ch}$ represents the specific chemical exergy of $H_2$, CO and $CH_4$ respectively [12].
An exergy loss calculation of the plant with relative to the fuel exergy is being performed to identify the components where major exergy destruction occurs. Also the exergetic efficiency of the plant components is analyzed. The exergetic efficiency any component is calculated as:

$$n_{exergetic} = Ex_{out} / Ex_{in} \quad (21)$$

Where, $Ex_{in}$ is the sum of exergies of the streams entering into a component and the work inputs $W_{in}$, if any [13] and thus:

$$Ex_{in} = \Sigma(Ex_i)_{in} + \Sigma(W_i)_{in} \quad (22)$$

Subscript $i$ represent any incoming stream or work input. Similarly, exergy coming out $Ex_{out}$ from the control volume is calculated as:

$$Ex_{out} = \Sigma(Ex_i)_{out} + \Sigma(W_i)_{out} \quad (23)$$

The exergy loss is the difference between the incoming exergy and outgoing exergy of streams i.e. ($Ex_{in}$-$Ex_{out}$)

## IV. RESULTS AND DISCUSSION

The performance analysis of the conceptualized plant is carried out with the help Cycle-Tempo software. A gasifier reaction temperature of 800°C has been fixed and is regulated setting the oxidant–fuel ratio, $x_{OF}$. Table III presents the product gas composition obtained from Cycle-Tempo® software.

TABLE III
PERFORMANCE OF THE GASIFIER

| **Gas Composition (% mole fraction)** | |
|---|---|
| $H_2$ | 21.25 |
| CO | 24.91 |
| $CO_2$ | 7.10 |
| $N_2$ | 40.64 |
| $CH_4$ | 0.3 |
| $H_2O$ | 5.56 |
| LHV (MJ/kg) | 5.3 |

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|---|---|
| HHV (MJ/kg) | 5.7 |
| Oxidant-fuel ratio ($x_{OF}$) | 1.8 |
| Gasification efficiency (%) | 79.62 |

The biomass based IHCC plant is designed considering a base case (pressure ratio 4 and TIT=900$^0$C). The base case performance of the plant is tabulated bellow.

TABLE IV
BASE CASE PERFORMANCE OF THE PLANT

| **Parameter** | **Unit** | **Value** |
|---|---|---|
| Topping cycle pressure ratio | ---- | 4 |
| GT inlet temperature | $^0$C | 900 |
| GT cycle output | kW | 20 |
| Air flow rate | kg/s | 0.119 |
| ST cycle output | kW | 14.93 |
| Steam generation rate | kg/s | 0.025 |
| Energy efficiency | % | 30.5 |
| Exergy efficiency | % | 27.2 |
| Biomass flow rate | kg/hr | 25.2 |

The performance of the plant is found to be influenced greatly by variations in topping cycle pressure ratio and gas turbine inlet temperature. The plant overall efficiency initially increases with GT pressure ratio and then decreases for a fixed TIT as shown in Fig. 2. This is due to the fact that the steam turbine work output initially increases and then decreases with increase in pressure ratio (as shown in Fig. 3). Fig. 2 also shows that higher TIT results in higher efficiency. The optimal values of pressure ratio depend on TIT and are found to be 6.4, 7.2 and 8.1 for TITs 900, 1000 and 1100 deg C, respectively.

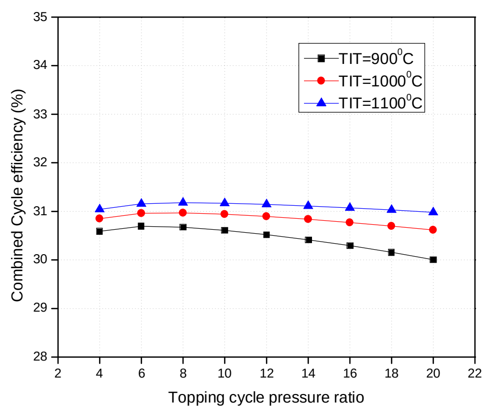


Fig. 2: Variation in plant efficiency with topping pressure ratio

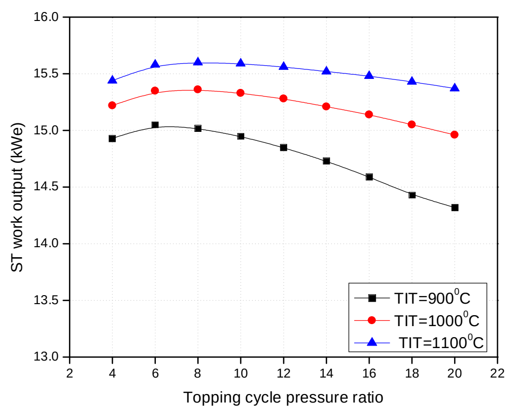


Fig. 3: Variation in ST work output with topping pressure ratio

The specific volume flow rate of air through the tubes of the CHX unit, and therefore the size of the CHX unit, is greatly influenced by the topping cycle pressure ratio. Higher topping cycle pressure ratio reduces the mass flow requirement of the GT block. This reduced mass flow rate, coupled with higher density at higher pressure, reduces the volume flow rate of air through the tubes of the CHX unit. Thus sizing of the topping cycle ducting and CHX unit decreases Figures 4 and 5 explain these variations.

The exergy analysis of the plant is carried out in this paper at the optimal operating pressure ratio (Π=8) and different

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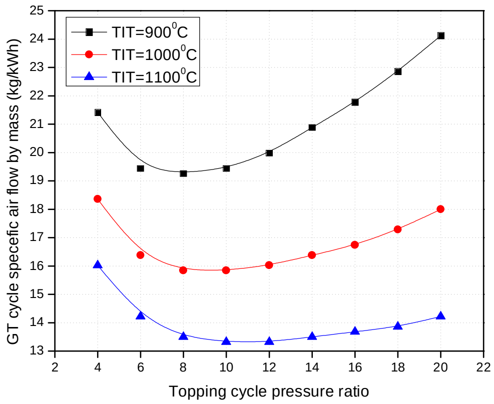
TIT's using similar methodology of Datta *et al.* [12]. Figure 4 shows the overall exergy of the plant with respect to fuel exergy at TIT=900$^0$C.



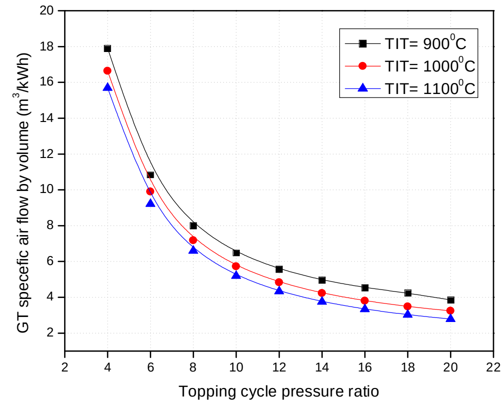
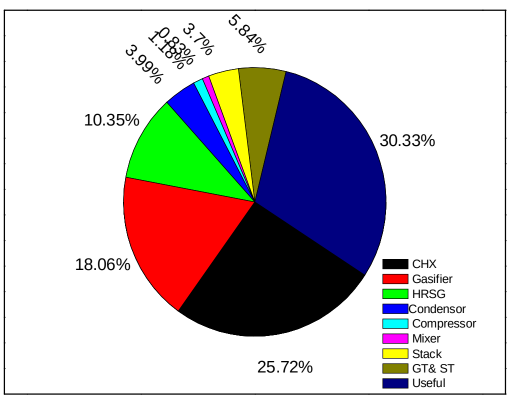




Fig. 4: Variation in specific air flow by mass with topping pressure ratio Fig. 5: Variation in specific air flow by volume with topping pressure ratio

It is seen from Fig. 6 that the major exergy destruction takes place in the gasifier and the CHX unit, together accounting for about 44% of the total input. Also some exergy losses occur in the HRSG and the turbines. The exergy destructions occurring in the other plant components are insignificant. The exergy destruction for the combustor-heat exchanger duplex unit is higher due to combustion and heat exchange across streams of high temperature. Higher amount of exergy loss also occurs at the HRSG unit due to heat transfer between gas and working fluid.



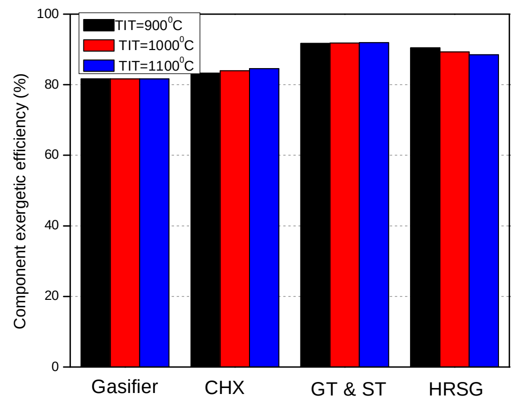




Fig. 6: Overall exergy of the plant at TIT=900$^0$C Fig. 7: Exergetic efficiency of the major exergy destructive components

The exergetic efficiency values of the major exergy destructive components at different TIT's are plotted in Fig. 7. It is seen that the exergetic efficiency for the turbine and the HRSG is about 90%, while that for the gasifier or the CHX unit is much lesser. The effect of TIT does not affect the gasifier performance.

## V. Conclusion

A configuration is proposed for an indirectly heated combined cycle incorporating a downdraft biomass gasifier using olive leave and purnings mixture as fuel. It is observed that the plant efficiency is maximized at some values of topping cycle pressure ratio, the value which depends upon the gas turbine inlet temperature. At a particular pressure ratio, the plant thermal efficiency increases with TIT. In addition, it is found that the gas turbine pressure ratio affects the size of the plant components, as it decreases the air flow rate across the gas turbine cycle. The maximum exergy destruction occurs at the gasifier, combustor-heat exchanger duplex unit and HRSG. It is seen form the analyses that the plant can be operated at the pressure ratio value of 8 and at higher turbine inlet temperatures.

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## NOMENCLATURE

| | |
|---|---|
| $X$: Mole fractions (%) | $h$: Enthalpy (kJ/kg) |
| $x_{OF}$: oxidant (air)- fuel ratio | $s$: Entropy (kJ/kgK) |
| $m$: Mass flow rate of air for gasification (kg/s) | **Greek Symbols** |
| $m_{p.g}$: Mass flow rate of producer gas (kg/s) | $\lambda$: Equivalence ratio |
| $m_{biomass}$: Mass flow rate of biomass (kg/s) | $\Pi$=: Topping cycle pressure ratio |
| $m_{air}$: Mass flow rate of air (kg/s) | $\beta$: Multiplication factor |
| $m_{f,g,m}$: Mass flow rate of flue gas mixture (kg/s) | $\eta_{gasi}$: Gasification efficiency |
| $m_f$: Mass flow rate of flue gas (kg/s) | $\eta_{cc}$: Combined cycle efficiency |
| $m_s$: Steam generation rate (kg/s) | **Abbreviation** |
| $C_{p,a}$: Specific heat capacity of air (kJ/kgK) | GT: Gas turbines |
| $C_{p,f}$: Specific heat capacity of flue gas (kJ/kgK) | ST: Steam turbine |
| $w_c$: Work requirement of compressor (kW) | HRSG: Heat recovery steam generator |
| $w_{GT}$: Work delivered by gas turbine (kW) | LHV: Lower heating value |
| $w_P$: Work requirement of pump (kW) | CHX: Combustor-heat exchanger duplex unit |
| $w_{ST}$: Work delivered by gas turbine (kW) | IHCC: Indirectly heated combined cycle |
| $w_{net}$: Net work output (kW) | p.g: Producer gas |

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**BIOGRAPHY**

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