NOVEL BIOMASS BASED ENERGY CONVERSION SYSTEMS FOR SUSTAINABLE RURAL DEVELOPMENT

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 ISSN 1923-6654 (print)
ISSN 1923-6662 (online). Available at http://www.ssrn.com/link/OIDA-Intl-Journal-Sustainable-Dev.html

Abstract*:* Recent years have witnessed rapid development in renewable energy technologies. Several renewable energy technologies have found wide acceptance and deployment in the developed as well as the developing nations. Apart from solar PV, biomass based energy systems have gained much popularity, particularly in the developing economies, where unavailability of grid electricity in remote and rural areas paved the way for wide deployment of biomass based energy systems. This paper focuses on two novel and high-efficiency biomass gasification based conceptualized combined cycle plants that offer promising performance. Biomass gasification, when integrated to combined cycles, offers high efficiency compared to the conventional biomass based gas engines. An indirectly heated GT cycle has been considered in the topping cycle, while in the bottoming cycle both Steam Rankine cycle (SRC) has been considered in one configuration and Organic Rankine cycle (ORC) has been considered for the other. The study reveals that high overall efficiency is achievable in such combined cycle systems. ORC is found to give much better energetic performance, giving conversion efficiency as high as 46% while similar configuration with SRC yields about 39% overall efficiency.

Keywords*:* Biomass gasification; Combined cycle, Organic Rankine cycle, efficiency, cogeneration.

INTRODUCTION

Biomass based power generation is getting increased importance worldwide in overcoming the energy scarcity problems and environmental issues [1]. Wide ranges of biomass gasifier-gas engine assembly are commercially available for decentralized power generation. However, these systems suffer from low overall efficiency (~20-25 %) and extensive gas cleaning and cooling requirement [2]. The internal combustion (IC) engines are very sensitive to the presence of tar, particles and moisture in the producer gas and so additional gas cleaning and drying systems are required after the gasifiers which ultimately affect the operation and maintenance cost [3]. The energy researchers are paying attention into the integration of bio-gasification with gas turbine (GT)-steam turbine (ST) combined cycle plant, which can improve the overall electrical efficiency substantially [4]. The Worlds' first bio-gasification based combined cycle power plant was operated during 1996 in Varnamo, Sweden with an overall efficiency of 32% [5]. The operation of the plant was stopped due to high operation and maintenance cost as extensive gas cleaning and cooling were required for the combustor and gas turbine.

Solid biomass is thermo-chemically converted into gaseous products through its gasification in oxygen deficient environment. The main components of the gas are CH_4, H_2, CO, CO_2, H_2O and N_2. Different types of tars are also produced during gasification as by-product [6]. The producer gas needs to be extremely clean to avoid erosion, corrosion of and particulate deposition on the gas turbine blades and blockage of the fuel injectors. Also, since the calorific value of the product gas is low compared to natural gas, modification of the combustor is also essential for the usage of producer gas in conventional GT [7].

The gas cleaning and cooling complexities along with the modification requirement of combustor and rotor balding can be avoided by implementing a combustor-heat exchanger duplex (CHX) unit instead of conventional gas combustor used in a GT cycle [8, 9]. The CHX unit burns the producer gas and heats up the air for GT. The exhaust from the GT is utilized in generating superheated steam for the bottoming Rankine cycle. Datta et al [7] carried out an energetic and exergetic performance analyses of an externally fired gas turbine cycle, wherein it is seen that the cycle reaches a maximum efficiency at particular value of topping cycle pressure ratio, depending on turbine inlet temperature (TIT) and cold end temperature difference (CETD) of the heat exchanger. Zanial and Al-attab [10] had experimentally studied the performance of the heat exchanger used in an EGFT cycle. It is observed that the heat exchanger is capable of producing about 700^0C TIT at 63% average effectiveness. Soltani et al. [11] had also carried out the similar analysis of an externally fired combined cycle (EFCC) plant. Most of the studies suggest that the design of the heat exchanger, used in the CHX unit is the most critical parameter from operational point of view.

In this paper, thermodynamic modeling and performance assessment of two bio-gasification based indirectly-heated combined cycle plants are reported. In both the schemes an indirectly heated GT cycle has been considered in the topping cycle, while in the bottoming cycle Rankine cycle (SRC) has been consider for one scheme and Organic Rankine cycle (ORC) for the other. The issue of the sizing of the major plant components has also been discussed. The gas turbine block has a fixed output of 30 kWe. Saw dust is used as fuel feed and the gasifier is considered to be of downdraft type. Variation in overall efficiency, work output, electrical specific biomass consumption (ESBC) etc are analyzed over a range of the topping cycle pressure ratio and gas turbine inlet temperature. The model development and thermal performance assessment have been carried out using Cycle-Tempo software [12].

PLANT CONFIGURATIONS AND DESCRIPTIONS

Figure 1 shows the schematic diagram of the conceptualized biomass based indirectly heated combined cogeneration plant with bottoming steam Rankine cycle (SRC). Solid biomass (saw dust) is fed to a downdraft gasifier (block 6) to convert it into producer gas, in the presence of atmospheric air, in sub-stoichiometric condition. The hot producer gas is fed into the combustor (block 7), where it gets combusted in the presence of recirculated gas turbine (GT) exhaust air. Flue gas, generated during the combustion process, enters the shell side of a high pressure high temperature (tube side) air heater (block 8). Block 7 and block 8 together called as combustor-heat exchanger duplex (CHX) unit, which heats up the working medium (air) of the topping GT cycle.

Atmospheric air after passing through the compressor (block 9) enters the CHX unit and gets heated up. The hot and compressed air then expands in the gas turbine (block 10) to atmospheric pressure and enters the superheater box (block 11) and produces superheated steam for the ST . An electric generator is coupled with the gas turbine rotor to produce electricity. The heat exhausted from the CHX unit is recovered through a heat recovery steam generator (HRSG), consisting of three sub-units; HP evaporator (block 16), economizer (block 17) and LP evaporator (block 18). The HP evaporator, economizer, steam turbine (ST-block 12), condenser (block 13) and feed pump (block 15) constitute the bottoming steam power cycle. The pump is driven by an electric motor (block 14). The LP evaporator produces utility steam for heating purpose. The final exhaust from the plant is exposed to the atmosphere through stack (block 19).

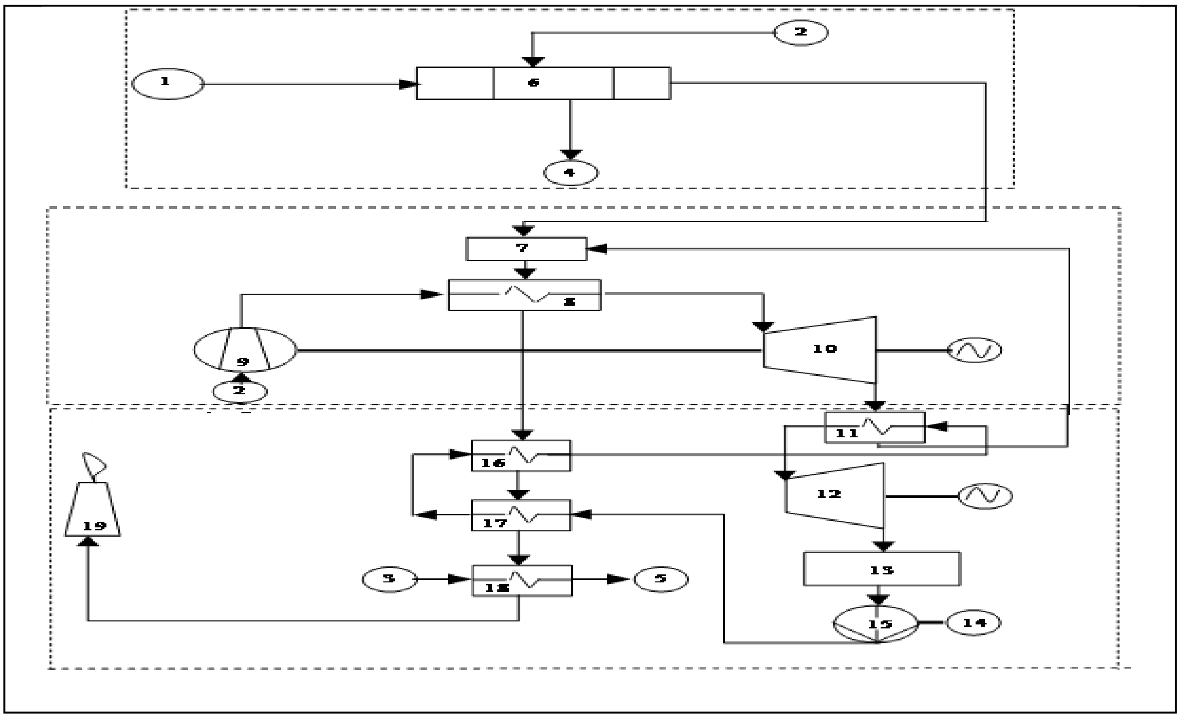


Figure1: Combined Cycle Scheme with Bottoming Steam Rankine Cycle (SRC)

Figure 2 shows the schematic diagram of the plant with similar topping cycle configuration but with bottoming organic Rankine cycle (ORC). The ORC-superheater along with ORC-turbine, condenser, feed pump and internal recuperator constitutes the bottoming organic rankine cycle where toluene is the working fluid. The cycle operates on supercritical mode. The exhaust from the gas turbine (block 8) enters the ORC-superheater (block 9) and produces supercritical organic vapor for the organic vapor turbine (ORC-VT, block 14). The exhaust of ORC-VT enters an internal recuperator (block-10) which preheats the organic fluid entering the heat recovery unit (block 9). Finally, the exhaust enters the condenser (block 13), followed by the feed pump (block-12). The pumped fluid enters the internal recuperator, gets preheated and enters the ORC-superheater and thus the bottoming cycle continues. The pump is driven by an electric motor (block 11). The final exhaust from the plant is exposed to atmosphere through stack (block 15).

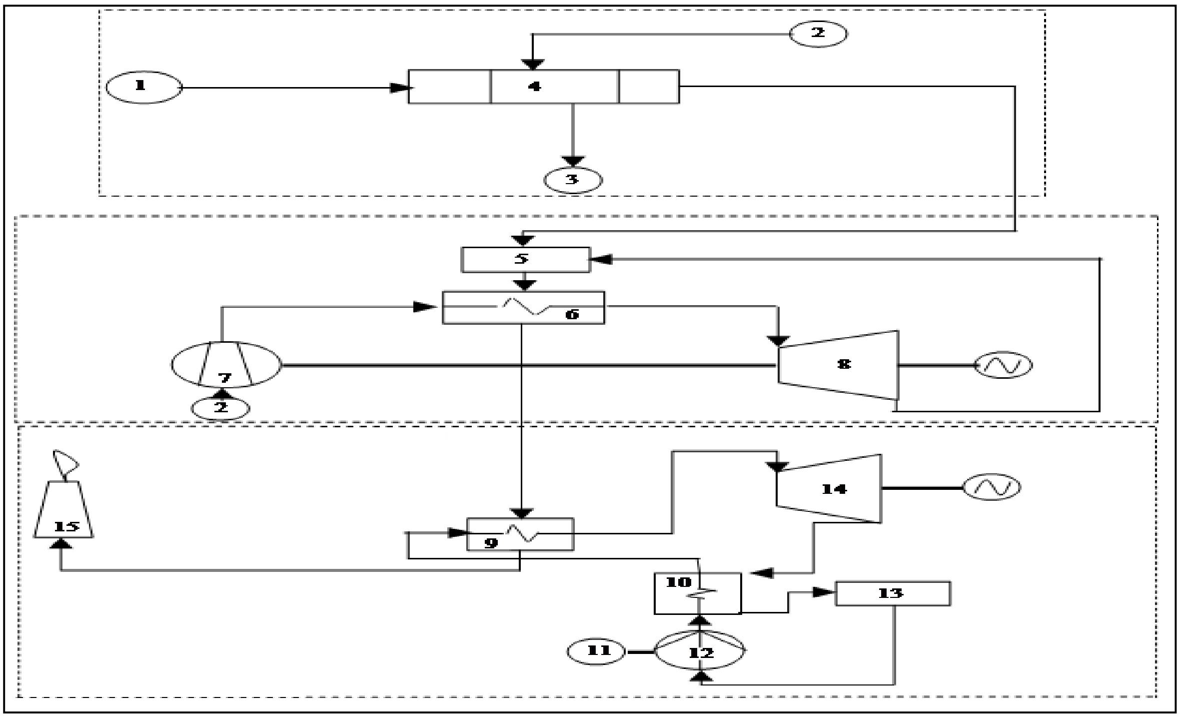


Figure2: Combined cycle scheme with bottoming organic Rankine cycle (ORC)

The simulated performances of the two configurations are assessed over a wide range topping cycle pressure ratio (rp=4 to 16) and turbine inlet temperature (TIT=900, 1000, 1100 deg C) for the GT block. The study also includes discussion on the sizing of the major plant components.

MODEL DEVELOPMENT AND FORMULATION

Thermodynamic First Law analyses have been carried out for the plant configurations, varying certain design and performance parameters. The model development and thermal performance assessment have been carried out using Cycle-Tempo software. In Cycle-Tempo, different components models are available in the component library from which necessary components can be picked and connected to represent a process or a cycle. The following sections describe the assumptions and applicable thermodynamic relations for the relevant processes and components.

Assumptions

The following assumptions are made for the analyses:

- No extraneous heat loss occurs in the plant components and in the ducts.
- No pressure loss occurs in the gas path, air path and in the steam path.
- Hot end temperature difference between gas and air in the CHX unit is 20^0C.

- The bottoming steam cycle consists of non reheat Rankine cycle operating at 25 bar and 400^0C. The condenser pressure is 0.1 bar. For the ORC, the maximum vapor pressure and temperature are 46 bar and 340^oC and the condenser pressure is 0.035 bar.
- The isentropic efficiencies of air compressor and GT are 90%, while the same for bottoming ST is 85%.
- The process steam block of the first scheme operates at 1.5 bar.
- For the HRSG, minimum pinch point temperature difference is set to10^0C. The stack temperature is 120^0C.

Gasification Unit

The gasifier is a fixed bed downdraft type and chemical equilibrium model is considered by Cycle Tempo for model calculations. The ultimate analysis of biomass is presented in Table 1. Moisture content is 16 %. The equivalence ratio for the gasification is 0.35 and gasification temperature is 680^0C. Tar formation is considered to be negligible and ash is represented by SiO_2 for this model.

Table 1: Ultimate Analysis of the Biomass Feed [1].

Composition	Mass Percentage on Dry Basis (%)
C	52.28
H	5.2
N	0.47
O	40.85
Ash	1.2

Compressor and Gas/air Turbine Unit

Compressor (block 9) and the Gas turbine (block 10) units are modeled following standard thermodynamic relations and accordingly work input/output is calculated.

Combustor-Heat Exchanger Duplex (CHX) Unit

Hot producer gas gets combusted in the combustion chamber of the CHX unit in the presence of recirculated GT exhaust air. Complete combustion of all combustible elements is considered and accordingly heat evolved as well as gas temperature are calculated by model. The products of combustion (that occurs in the combustor section of CHX unit), exchange heat with the compressed air within the tubular heat exchanger section of the CHX unit, thus, transferring heat to the topping cycle working fluid, i.e. air.

Bottoming Cycle

The bottoming cycles differ in configuration and working fluid. While the SRC in the first configuration gives power output, its exhaust is further recovered in a low pressure evaporator that produces process steam. The ORC in the second configuration has no process steam evaporator. The turbine outputs, pump inputs as well as the heat exchange rates are calculated based on usual thermal procedures. Table 2 shows properties of Tuluene.

Table 2: Properties of Toluene [13]

Parameter	Unit	Value
Molecular Formulae	----	C_7H_8
Molecular Weight	Kg/kmol	92
Critical Temperature	^{o}C	318.65
Critical Pressure	bar	41.06
Boiling Point	^{o}C	110.7

Work and Heat Outputs

Net work output from combined cycle in each case is the sum of work outputs from gas turbine and from steam or vapor turbine, as is the case for bottoming cycle.

$$W_{CC} = W_{net.GT} + W_{ST} \quad (1)$$

The overall electrical efficiency of the combined cycle plant is expressed as:

$$\eta_{e.CC} = \frac{W_{CC}}{m_b\, LHV_b} \quad (2)$$

Where, m_b represents the biomass consumption rate equivalent to one formula mol of biomass feed to the plant.

Electrical specific biomass consumption-ESBC (kg/kWh) is expressed as:

$$ESBC = \frac{3600 m_b}{W_{CC}} \quad (3)$$

Although several approaches have been proposed, considering first law to evaluate the performance of a cogeneration plant, the fuel energy savings ratio (FESR) calculation method is significant for indicating the savings in fuel of a combined power and heating plant instead of separate power and heating plants[13, 14].

The fuel saving for the first plant is expressed considering a pair of bio-gasification based separate heating and power plants as:

$$\Delta F = \left(\frac{W_{CC}}{\eta_{e,ref}} + \frac{Q_U}{\eta_{b,ref}}\right) - m_b\, LHV_b \quad (4)$$

The reference efficiency values are set to be 25 percent and 80 percent for the individual power and heat plants respectively. The FESR is given by:

$$FESR = \frac{\Delta F}{\left(\frac{W_{CC}}{\eta_{e,ref}} + \frac{Q_U}{\eta_{b,ref}}\right)} \quad (5)$$

RESULTS AND DISCUSSIONS

Energetic performance of the conceptualized plants along with some discussion on the sizing of the major plant components are reported in this section. The performance of the gasifier, considering saw dust as biomass feed is presented in Table 3. The producer gas composition, obtained from the model is compared with the commercially available gasifier (Ankur Gasifier [2]). For both the configurations, a base case is considered where the topping cycle pressure ration is 4 and the turbine inlet temperature is 1000°C (r_p=4 and TIT=1000^{0}C). The performances of the plants at the base case are shown in Table 4. It is seen form "Table 3" that the plant with ORC bottoming cycle gives far more electrical efficiency, 46% as against 39% for the plant with bottoming SRC (for base case configurations). Configuration with SRC is also capable of producing process steam at arate of 160kg/hr. This corresponds to a FESR of about 45. The ESBC of the configurations are 0.56 kg/kWh and 0.47 kg/kWh respectively, the ORC-integrated plant consuming less fuel for every unit of generated power.

Table 3: Comparative of the Gasifier Model Output and Experimental Result of Ankur Gasifier

Gas Composition (% mole fraction)	Present Model	Ankur Gasifier
H_2	21.44	18±3
CO	22.14	19±3
CO_2	10.57	10±3
CH_4	0.54	Upto 3
N_2	39.09	45-50
H_2O	5.76	------
Air-fuel Ratio	1.6	1.5-1.8
LHV (MJ/kg)	5.04	4.40-5.40
Gasifier efficiency (%)	82	------

Table 4: Base Case Performance of the Two Configurations

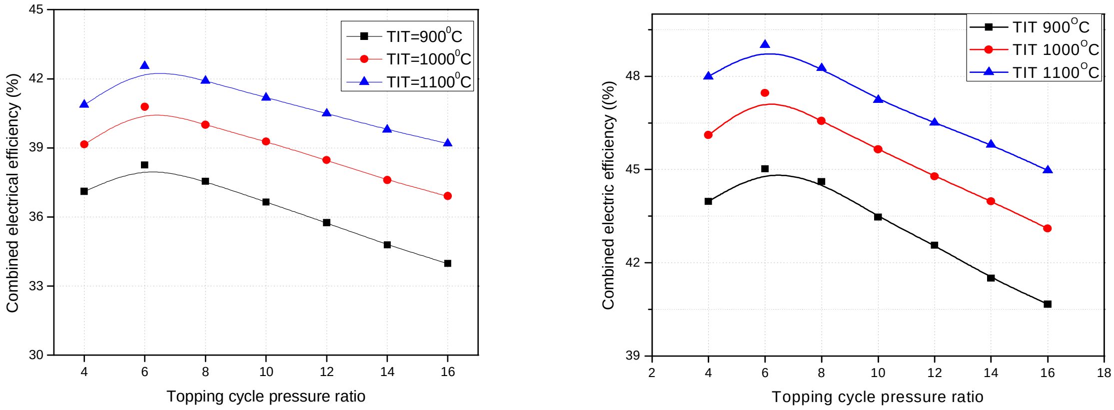
(a) Configuration with Bottoming *SRC*

Parameter	Unit	Value
GT Output	kWe	30
SRC-ST Output	kWe	8.63
Overall Electrical Efficiency	%	39.15
ESBC	kg/kWh	0.56
Utility Steam Generation Rate	kg/h	159
FESR	%	45.10

(b) Configuration with Bottoming ORC

Parameter	Unit	Value
GT Output	kWe	30.0
ORC-VT Output	kWe	17.8
Overall Electrical Efficiency	%	46.11
ESBC	kg/kWh	0.47

The variation in overall electrical efficiency with topping cycle pressure ratio at different gas turbine inlet temperatures is shown in Fig. 3. The performance of the plants is found to be influenced greatly by the variations in topping cycle pressure ratio and gas turbine inlet temperature. Overall electrical efficiency of the plant initially increases with increase in topping cycle pressure ratio and then decreases for a fixed TIT as shown in Fig. 3. This is due to fact that the value of specific work output from the topping cycle is maximized at certain values of pressure ratio and then decreases with further increase in pressure ratio.



(a) Configuration with Bottoming SRC (b) Configuration with Bottoming ORC

Figure 3: Variation in Overall Electrical Efficiency with Topping Cycle Pressure Ratio.

Now, the net work output from the topping cycle is considered to be fixed for the present study. Hence, with increase in pressure ratio, the required air consumption (by mass) of the topping cycle initially decreases, gets minimized at certain pressure ratio and then increases with increase in pressure ratio, as shown in Fig. 4. This ultimately results in the required ESBC to decrease initially and then to increase with increase in pressure ratio as shown in Fig. 5. Hence the overall electrical efficiency of the plant initially increases, gets maximized and then decreases with increase in pressure ratio. Fig. 3 also indicates that higher TIT results in higher efficiency. The maximum efficiency point shifts slightly towards the right end of the graph as the TIT increases.

It is also evident from Fig. 4 that the air consumption rate decreases with increase in TIT at a particular pressure ratio as the specific work output from the topping cycle is higher at higher TITs. Also, for a fixed value of topping cycle pressure ratio, the required ESBC decreases as the TIT increases. This is because of the required air flow through topping GT cycle decreases with increase in TITs thus ultimately affecting in the required heat input to decrease with increase in the same.

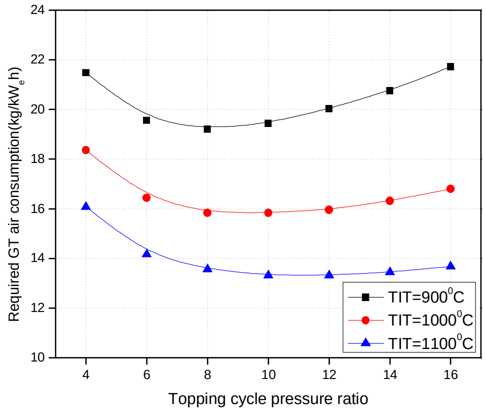
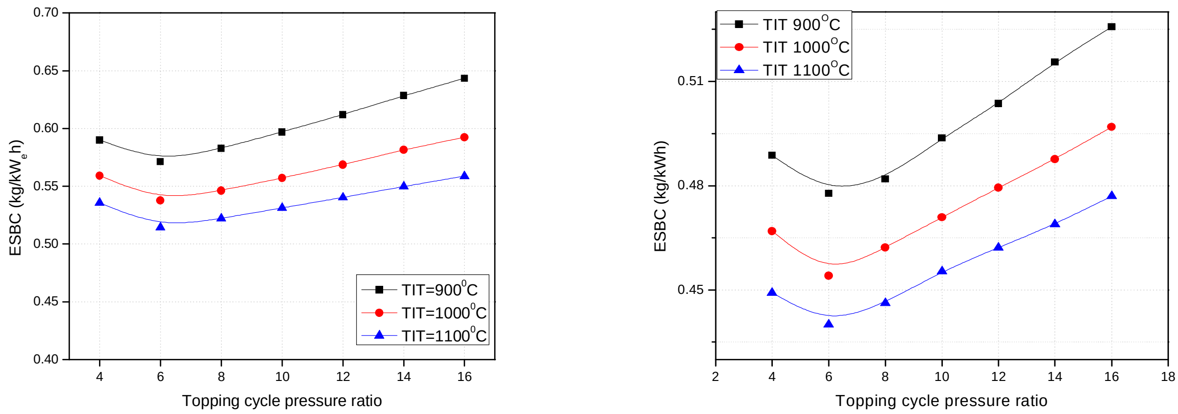


Figure 4: Variation in Required Air Flow by Mass, through Topping Cycle with Pressure Ratio.



(a) Configuration with Bottoming SRC

(b) Configuration with Bottoming ORC

Figure 5: Variation in Electrical Specific Biomass Consumption (ESBC) with Topping Cycle Pressure Ratio.

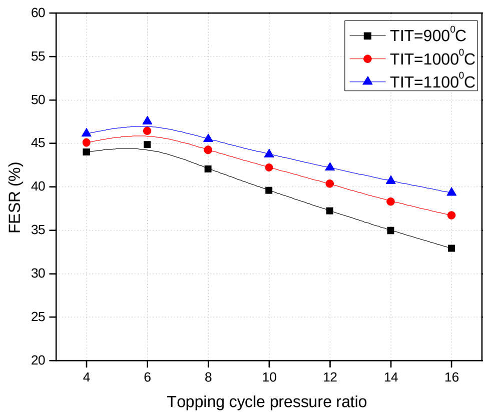


Figure 6: Variation in Fuel Energy Savings Ratio (FESR)

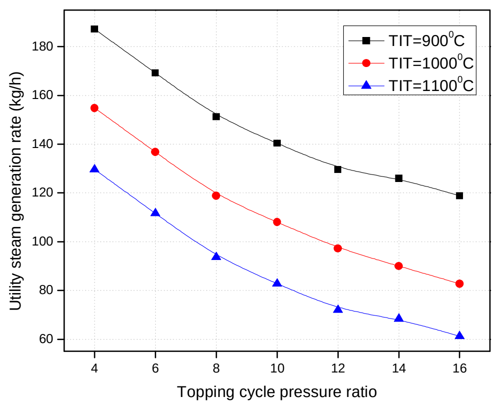


Figure 7: Variation in Utility Steam Generation Rate

The variation in fuel energy savings ratio (FESR) with topping GT cycle pressure ratio is shown in Fig. 6. The graph follows a trend opposite to that of Fig. 5 and this is obvious and can be clearly understood. Fig. 7 shows the variation in utility steam generation rate with GT cycle pressure ratio. The value decreases with increase in pressure ratio. Also the steam generation rate is higher at lower TITs due to the fact that at lower TITs the required air consumption as well as biomass consumption increases, leading to more heat release in the LP evaporator. This results in increased process steam generation rate.

The required topping cycle air flow rate also influences the size of the gas/air turbine. For the low pressure end of the turbine the size is usually determined by the specific air consumption by mass while that for high pressure end is determined by the specific air consumption by volume [7]. Fig. 8a and Fig. 8b show the variation in required air consumption by volume with topping cycle pressure ratio for both units to predict the sizes. It is seen from both figures that sizes of the said units decrease with increase in pressure ratio as well as with higher TITs. However, the increased metal thickness at higher TITs may limit the economic advantages owing to reduced sizes.
The CHX unit is one of the most important and critical components used in the plant. A high pressure high temperature heat exchanger is required for this purpose and the design needs to be optimized from the sizing and cost point of view.

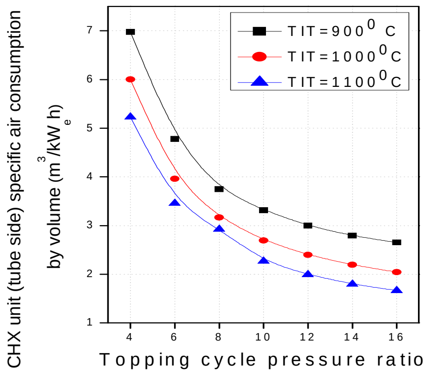


Figure 8a: Variation in CHX unit (tube side) specific air consumption by volume with pressure ratio.

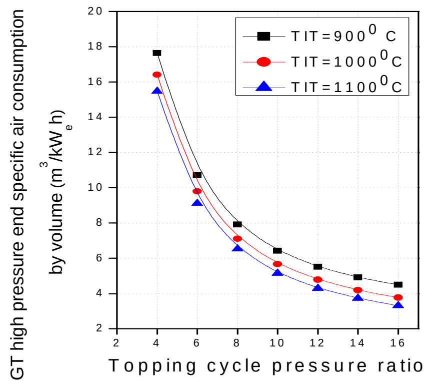


Figure 8b: Variation in GT high pressure end specific air consumption by volume with pressure ratio.

Table 5: Performances of the Plant and Sizing of the Heat Exchangers at Different Operating Conditions for Configuration with Bottoming SRC

	Scenario 1: r_p=6, TIT=900^{0}C				Scenario 2: r_p=6, TIT=1000^{0}C				Scenario 3: r_p=6, TIT=1100^{0}C			
Performance of the Plant												
Required Biomass Flow rate (kg/s)	0.006				0.0057				0.0055			
Required Air Flow Rate (kg/s)	0.163				0.137				0.118			
Electrical Efficiency (%)	38.25				40.78				42.56			
FESR (%)	44.85				46.42				47.51			
Utility Steam Generation (kg/h)	169.2				136.8				68.4			
Sizing of the Heat Exchangers	ΔT_H (K)	ΔT_L (K)	Q_{tran}	UA	ΔT_H (K)	ΔT_L (K)	Q_{tran}	UA	ΔT_H (K)	ΔT_L (K)	Q_{tran}	UA
Gas to Air Heater (block 8)	21.1	99.1	119	2.35	22.3	123	116	1.96	21.6	147	114	1.73
Superheater (block 11)	115	268	4.20	0.03	187	334	4.35	0.02	259	401	4.53	0.015
Evaporator (block 16)	118	10.0	20.6	0.47	142	10.0	21.5	0.43	167	10.0	22.4	0.40
Economizer (block 17)	83.9	165	4.20	0.04	83.9	160	4.36	0.04	83.9	155	4.54	0.039
Gas to Process Steam Generator (block 18)	100	95	17.0	0.17	95.3	95.0	13.6	0.14	90.1	95.0	11.2	0.12

Table 6: Performances of the Plant and Sizing of the Heat Exchangers at Different Operating Conditions for Configuration with Bottoming ORC

	Scenario 1: r_p=6, TIT=900^{0}C				Scenario 2: r_p=6, TIT=1000^{0}C				Scenario 3: r_p=6, TIT=1100^{0}C			
Performance of the Plant												
Required Biomass Flow rate (kg/s)	0.0065				0.0058				0.0055			
Electrical Efficiency (%)	45.019				47.467				49.009			
Sizing of the Heat Exchangers	ΔT_H (K)	ΔT_L (K)	Q_{tran}	UA	ΔT_H (K)	ΔT_L (K)	Q_{tran}	UA	ΔT_H (K)	ΔT_L (K)	Q_{tran}	UA
Gas to Air Heater (block 6)	73.5	164	118	1.04	54.9	162	114.7	1.16	51.2	182.6	113	1.09
ORC Superheater (block 8)	67.0	3.94	54.2	2.44	65.4	3.94	45.7	2.09	85.8	3.94	42.9	1.61
Internal recuperator (block 13)	35.7	9.43	13.5	0.68	35.7	9.43	11.3	0.57	35.7	9.43	10.6	0.53

It is seen from Fig. 4 that both the plants are most efficient at about of r_p=6. The performance of the plants along with sizing of the heat exchangers used in the plant is shown in Table 5 and Table 6, at different TITs. Table 5 shows the size of the heat exchanger (block 8) used in the CHX unit decreases with increase in TIT.

Table 6 shows the size of the heat exchanger (block 6) used in the CHX unit is lower at TIT= 900^0C due to higher high end temperature difference of the said unit. However, the efficiency value is lower compared to the efficiency value at TIT=1100^0C, although the size of the heat exchanger is almost same at the both TITs.

CONCLUSIONS

This paper focuses on two novel and high-efficiency biomass gasification based conceptualized combined cycle plants. An indirectly heated GT cycle has been considered in the topping cycle, while in the bottoming cycle both Steam Rankine cycle (SRC) has been considered in one configuration and Organic Rankine cycle (ORC) has been considered for the other. The study reveals that high overall efficiency is achievable in such combined cycle systems. ORC is found to give much better energetic performance, giving conversion efficiency as high as 46% while similar configuration with SRC yields about 39% overall efficiency, in the base case configurations. It is further found that the plants give highest efficiency at a topping cycle pressure ratio of about 6. The size of the tubing of the CHX unit is also found to be influenced greatly by the design and operating parameters.

ACKNOWLEDGEMENT

The authors acknowledge the cooperation of TU Delft (Delft University of Technology), The Netherlands, for providing the Cycle-Tempo software to the Thermal Simulation and Computation (TSC) lab at the Mechanical Engineering Department of IIEST, Shibpur, which has been used in the present study.

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