Proceedings of the
International Conference on Mechanical Engineering 2011
(ICME2011) 18-20 December 2011, Dhaka, Bangladesh

ICME11-RE-009

SIMULATED PERFORMANCE OF AN INTEGRATED BIOMASS GASIFICATION COMBINED CYCLE EMPLOYING INDIRECTLY-HEATED GAS TURBINE

Pradip Mondal and Sudip Ghosh

Department of Mechanical Engineering, Bengal Engineering & Science University, Shibpur, India.

ABSTRACT
This paper presents a conceptual configuration of a biomass derived producer gas fuelled combined cycle plant with a topping gas turbine block and a bottoming steam turbine plant. The conventional GT combustion chamber is replaced by a combustor heat exchanger duplex (CHX) unit. A downstream heat recovery steam generator (HRSG) with a reheat steam cycle has been considered for integration with the topping GT plant. A part of steam is extracted from the steam cycle, after reheating, which in turn is utilized for steam gasification of biomass in the gasifier unit. The CHX unit is used for heating the working medium of GT cycle without affecting its composition. The power output from the GT cycle, in the base case (GT cycle pressure ratio 5) configuration, is considered to be 100kW, while the corresponding steam turbine output is 132 kW. The power cycle is analyzed using energy methods for a range of pressure ratio across the GT block. Athena visual studio was used to develop the program for the process simulation and the results were compared with the results of Aspen Plus simulation for an identical cycle. The detailed calculation of the CHX unit has also been done with Aspen plus. From the thermodynamic analysis it is found that the integrated system is found to give a maximum overall thermal efficiency (based on the lower heating value of the producer gas) of 42% at a pressure ratio of 8.

Keywords: Indirectly Heated, Combined Cycle

1. INTRODUCTION
Growing environmental concern has encouraged the development of energy conversion systems with negligible adverse effect on environment [1]. Biomass is gaining increasing acceptance as a renewable energy source worldwide, making possible the perspective of reducing both fossil fuel depletion and greenhouse gas (mainly CO_2) and NO_X emissions due to fossil fuel utilization [2]. Worldwide biomass ranks fourth as energy resource, providing approximately 14% of the world's energy needs, particularly in rural areas [3].

Combined heat and power generation (CHP) or cogeneration has been considered worldwide as a major alternative to traditional systems in terms of significant energy saving and environmental conservation (carbon-dioxide neutral energy source). Existing biomass gas turbine power plants operate on directly fired (combustion of fuel in combustion chamber of a gas turbine cycle) technology. Thus it requires complex gas clean-up systems with sophisticated gas turbine technology, to minimize the damage of the gas turbine [4]. However, the main problem with directly fired systems is the cost of maintenance. Since the gas turbines are very sensitive to presence of tar, particles and humidity in the producer gas.

The direct use of biomass derived producer gas in the combustion chamber of GT cycle can cause corrosion, erosion and deposition, which damages the downstream components. Particulates can damage turbine blades and volatilized/melted and solid compounds can cause deposition [5]. Indirect heating of the working medium of the power system may be used with the help of a heat exchanger to avoid the above mentioned problems.

2. PLANT CONFIGURATION AND COMPONENT DESCRIPTION
Figure 1 shows the schematic of a biomass derived producer gas fuelled BIGCC plant, considered in the present study. The power output from the Brayton cycle is considered to be 100kW for base case. Ambient air at point 1 enters in the compressor and leaves at point 2. Compressed air then enters the combustor heat exchanger (CHX) duplex unit which raises the air temperature to 1000^0C. But heating of the compressed air is done indirectly with the help of the heat exchanger. Hot air then enters the gas turbine (GT) at point 3 and expands to point 4.

In the gasifier solid biomass is converted into gaseous fuel in the presence of steam known as steam gasification. The producer gas enters the CHX unit and gets combusted in the presence of ambient air to raise the temperature of the gas stream

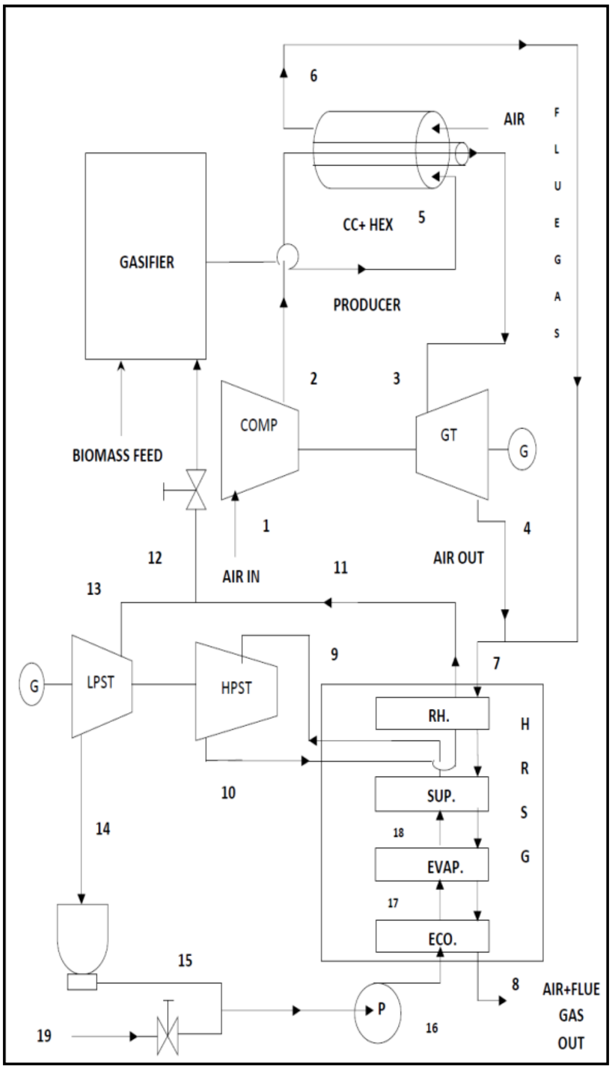

Fig1. schematic of an indirectly heated BIGCC plant.

Exhaust from GT cycle and flue gas from heat exchanger are mixed adiabatically at point 7 and the mixed stream enters the HRSG. Part of the steam produced in the HRSG is utilized for steam gasification in the gasifier. In the HRSG, steam is reheated in the reheater (RH) (after expansion in HP Steam Turbine) from point 10 to the point 11 and the superheater (SUP) superheats steam from point 18 to 9. The hot gas mixture is used to evaporate the water from point 17 to 18 in the evaporator (EVAP) and finally it is utilized to heat the feed water in the economiser (ECO) from point 16 to 17. The minimum allowable exit temperature (at point 8) for the HRSG exhaust is set to 120^0 C (393 K). The superheated steam at point 9 (65 bar pressure and 500^0 C) enters the high pressure (H.P.) steam turbine and expands till 2 bar. Then some amount of steam is extracted at point 12 for steam gasification purpose. The steam requirement for steam gasification of biomass is regulated by a flow control valve. Then the remaining steam at point 13 enters in the low pressure (L.P.) steam turbine, after doing work in turbine, steam is exhausted in the condenser at point 14 at 0.07 bar pressure. After being condensed in the condenser, water is pumped from point 15 to point 16 i.e. at the boiler pressure. Make up water is supplied at point 19 at condenser pressure which mix with condensed water, and then total water is pumped to boiler pressure.

3. METHODOLOGY ADOPTED IN PRESENT STUDY

Table1: Assumptions in the present study

PARAMETER	UNIT	VALUE
Ambient air temperature	K	300
Gas turbine inlet temperature	K	1273
Isentropic efficiency of compressor	-----	0.85
Isentropic efficiency of turbine	-----	0.88
Steam temperature from superheater outlet	K	773
Exhaust gas temperature from HRSG	K	393
GT cycle output	kW	100

Table2: Gasification data [6]

PARAMETER	UNIT	VALUE
Gasification temperature	K	750
Gas Composition		
CO	%vol	20.6
CO_2	%vol	35.3
H_2	%vol	31.4
CH_4	%vol	8.4
C_2H_2	%vol	2.0
C_2H_4	%vol	0.7
C_2H_6	%vol	1.6

The biomass composition in the present model is considered to be $CH_{1.5}O_{0.7}$ [6]. Fixed bed gasification of biomass in the presence of steam occurs in the gasifier (known as steam gasification) and producer gas is generated. The gas composition is shown in Table 2.

3.1 Gas Turbine Cycle

Initially the air flow rate ($m_{a,B}$ kg/s) through topping GT cycle is calculated by equating net work of the cycle to 100 kW for base.

$$(W_{net})_{GT} = W_T - W_C \qquad (1)$$

Again heat required for the cycle is calculated as

$$Q = m_{a.B} C_{p.a}(T_3 - T_2) \quad (2)$$

According to the requirement of the heat in the topping GT cycle required mass flow rate of producer gas is calculated.
Thermal efficiency of gas turbine cycle is the ratio of net work done in the gas turbine cycle and the required heat input.

$$(\eta_{th})_{GT} = (W_{net})_{GT}/Q \quad (3)$$

3.2 Combustor Heat Exchanger Duplex Unit

Producer gas enters in the Combustor-Heat exchanger duplex unit and gets combusted in the presence of atmospheric air.

Now required mass flow rate of producer gas to the combustor heat exchanger duplex unit is

$$m_p = (m_b + m_s) \quad (4)$$

The required biomass flow rate (m_b) and steam flow rate (m_s) are 0.02 kg/s and 0.028 kg/s.

Now theoretical oxygen required (in Molar basis) for combustion of each mole the producer gas is given by

$$N_{o2} = (\frac{v_{Co}}{2} + \frac{v_{H2}}{2} + v_{CH_4} * 2 + v_{C_2H_2} * 2.5 + v_{C_2H_4} * 3 + v_{C_2H_6} * 3.5) N_p \quad (5)$$

N_p represents the total moles of producer gas flowing in the CHX unit.
Now for complete combustion of the producer gas excess amount of air is needed. Total mass flow rate of air (m_a) required (considering 25% excess air) for complete combustion is given as

$$m_a = M_a N_a \quad (6)$$

Ma and *Na* represent the molecular weight and required moles of air. The value of *Na* is calculated by considering the excess air and equation (5).

After combustion of the producer gas mixture in the CHX unit flue gas such as CO_2, H_2O, N_2 and excess O_2 are generated in the same unit.
Specific heat capacity for flue gas mixture including air in the GT cycle is considered as a polynomial of temperature [7].

The flue gas (m_f) flow rate through CHX unit is

$$m_f = (m_a + m_p) \quad (7)$$

Finally, Temperature generated (T_g) after the combustion of producer gas is calculated as follows

$$m_p LHV_p = m_f C_{p.f} \int_{273}^{Tg} dT \quad (8)$$

Now flue gas temperature T_6 (Heat exchanger outlet temperature) is calculated as follows

$$m_f C_{p.f}(T_6 - T_8) = m_p LHV_p - Q \quad (9)$$

Effectiveness of the CHX unit is calculated as

$$\varepsilon = \frac{m_{a.B} C_{p.a}(T_3 - T_2)}{m_f C_{p.f}(T_5 - T_6)} \quad (10)$$

3.3 HRSG Unit

Hot air stream after expanding in gas turbine mixes with hot flue gas stream coming out from the Combustor Heat exchanger duplex unit and this mixed stream is used in HRSG to produce steam. The temperature generated after mixing process is calculated with the help of general equation of mixing.

Total gas flow rate (m_t) through HRSG is given by

$$m_t = (m_{a.B} + m_f) \quad (11)$$

Approximate mass flow rate of water (m_w) in the bottoming cogeneration plant can be found out from the overall energy balance in the heat recovery steam generator (HRSG) and the enthalpy (h) at different point can be found out from the steam table. Approximate mass flow rate of water in HRSG can be calculated from the following equation

$$m_W\{(h_9 - h_{16}) + (h_{13} - h_{10})\} = m_t C_{p.f}(T_7 - T_8) \quad (12)$$

Pinch point analysis was made for better heat transfer in the HRSG uint.

Work done in high pressure steam turbine is calculated from the following equation

$$W_{HPST} = m_W(h_9 - h_{10}) \quad (13)$$

Work done in low pressure steam turbine is also calculated from the following equation

$$W_{LPST} = (m_W - m_s)(h_{13} - h_{14}) \quad (14)$$

Now, efficiency of the combined cycle on producer gas composition basis is given as

$$(\eta_{th})_{overall} = \frac{(W_{net})_{GT} + W_{HPST} + W_{LPST}}{m_p LHV_p} \quad (15)$$

4. RESULTS AND DISCUSSION

The performance analysis of the conceptualized BIGCC plant is carried out with the help of two software; which are Athena Visual Studio and Aspen Plus. Codes were written in Athena Visual Studio in accordance with the thermodynamic model of the plant discussed in the previous section. Athena Visual Studio uses a FORTRAN compiler to generate executable program and produce results. Subsequently, Aspen Plus interface is used to produce a process flow chart of the integrated plant and simulate the performance of the plant components in further detail.

4.1 Results of Athena Simulation

Table3: Base case performance of the plant

PARAMETER	UNIT	VALUE
Brayton Cycle Output	kW	100
Air flow rate for GT block	Kg/s	0.441
Lower heating value of fuel	MJ/kg	8.47
Air flow required for combustion	Kg/s	0.263
Temperature after combustion of product gas	K	1746
Water mass flow rate for HRSG	Kg/s	0.129
Steam Turbine output	kW	131.6
Overall plant efficiency	%	40.16

Now the overall plant performances are illustrated by varying the topping cycle pressure ratio in a wide range.

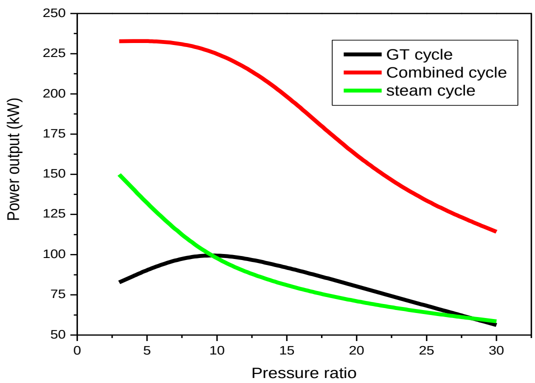

Fig 2. Variation of power output with pressure ratio.

Variation of power output with topping cycle pressure ratio is shown in **Figure2**. Combined cycle output initially increases but then goes on decreasing with the increase of pressure ratio. Increase of pressure ratio, leads in lowering of steam generation in HRSG (shown in **Figure 3**) and so steam turbine output is decreasing as shown in this figure. Therefore, initially combined cycle output takes an upward trend with increase of pressure ratio due to initial increase in gas turbine output and goes on decreasing due to decrease in steam turbine output.

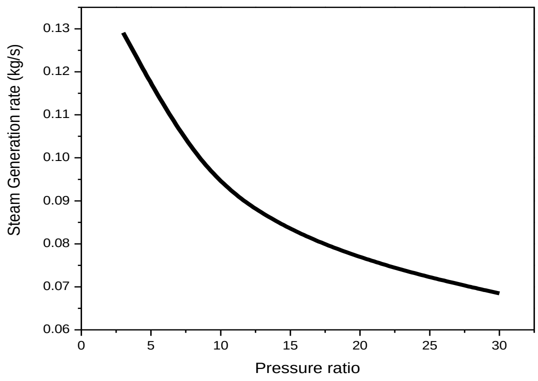

Fig 3. Variation of steam generation rate in HRSG with pressure ratio.

Figure3 shows the variation of mass flow rate of water in the HRSG with pressure ratio. At higher pressure ratio GT exhaust temperature decreases, which leads to reduced evaporation in HRSG.

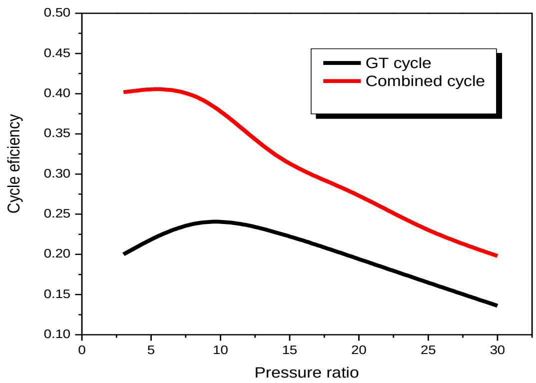

Fig 4. Variation of cycle efficiency with GT cycle pressure ratio.

Figure 4 shows the variation of overall efficiency with pressure ratio. Overall efficiency of the combined cycle initially increases and goes on decreasing with the increase of pressure ratio.

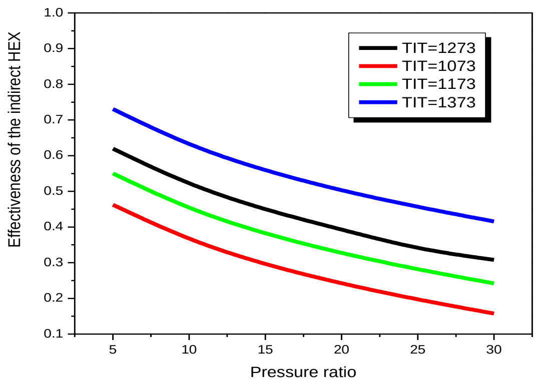

Fig 5. variations of required effectiveness of the indirect heat exchanger (CHX) with pressure ratio at various turbine inlet temperatures (TIT).

Figure5 shows the variation of the required effectiveness of the heat exchanger with compressor pressure ratio at various turbine inlet temperatures. For a fixed turbine inlet temperature effectiveness of the indirect heat exchanger is decreasing with increase in pressure ratio. This implies that a lower amount of heat transfer and heat exchange area is needed in the duplex heat exchanger.

The same Athena model was analyzed with the help of Aspen Plus software by providing a set of initial input, especially to determine the performance of the heat exchangers used in the model. For base case, the input data (detailed of input data are presented in table) for the Aspen model are taken from the output data from Athena software. The results of Aspen model are as similar as Athena model.

4.1 Model Simulation with Aspen Plus.

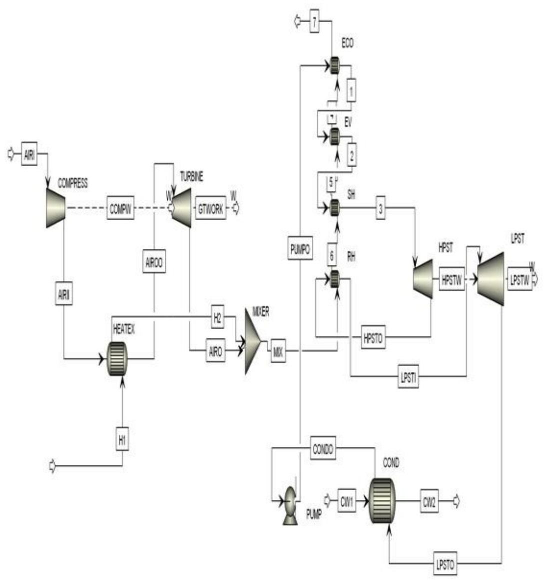

Fig 6. Schematic diagram of the indirectly fired BIGCC plant using Aspen Plus.

Table 4: Aspen input parameter

PARAMETER	UNIT	VALUE
Hot gas flow rate	Kg/s	0.331
Hot gas inlet temperature	K	1746
Air flow rate through GT cycle	Kg/s	0.441
Gas turbine inlet temperature	K	1273
Compressor pressure ratio	-----	5
Steam flow rate through steam cycle	Kg/s	0.1291
Inlet temperature of steam to HPST	K	773
Inlet pressure of steam to HPST	bar	65
Inlet temperature of steam to LPST	K	773
Inlet pressure of steam to LPST	bar	2
Inlet pressure to condenser	bar	0.075

The overall plant output data obtained from Aspen model is tabulated bellow.

Table 5: Aspen output data

PARAMETER	UNIT	VALUE
Gas compressor input	kW	88.25
Gas turbine output	kW	184.61
HPST output	kW	78.18
LPST output	kW	60.72
Plant Efficiency	%	40.81

The power output from low pressure steam turbine using Aspen plus is slightly higher than power output from Athena programming due to the fact that steam required for gasification purpose (0.028 kg/s) is not considered in Aspen Plus and so overall plant efficiency is higher than the Athena result for base case configuration.

The output data for the heat exchangers used in Aspen plus model is presented in the following table.

Table 6: Aspen output data for the heat exchangers used in the model

	CHX	COND	ECO	EVAP	SH	RH
Min. temp of approach [k]	100	10	10	10	10	50
Inlet hot stream temp [k]	1746	362	584	614	814	915
Inlet cold stream temp[k]	498	288	305	553.8	553	393
Outlet cold stream temp [k]	1273	296	554	554	692	687
Required heat exchanger area [sqm]	1.02	11.2	3.71	0.90	2.71	0.33
Product of UA [J/sec-K]	875	9593	3159	771	2304	288
LMTD [k]	433	34.37	56.2	43.62	87.6	314

5. CONCLUSIONS

Energy analysis for the conceptualized BIGCC plant indicates that for base case configuration, GT cycle provides thermal efficiency of 21.62% while combined cycle provides thermal efficiency of 40.16%. But GT cycle gives a maximum efficiency of 24.52% at a pressure ratio of 10.2 and for combined cycle the efficiency value being 42% at a pressure ratio of 8. Therefore the combined cycle may be operated in the range of pressure ratio 8-10.5 to get very close to cycle efficiency. From the Aspen analysis of the same plant it is found that, the efficiency is quite similar to the efficiency of the Athena model for base case configuration.

6. REFERENCES

1. Ghosh S., De S., 2006, "Energy analysis of a cogeneration plant using coal gasification and solid oxide fuel cell", Energy 31: 345–363.
2. Fracnco A., Giannini N., 2005, "Perspective for the use of biomass as a fuel in combined cycle power plants", International journal of thermal sciences, 44: 163-177.
3. Bain L. Richard, Overend P. Ralph, Craig R. Kevin, 1998, "Biomass-fired power generation", Fuel processing technology, 54: 1-16.
4. Syred C., Fick w., Griffiths A.J., Syred N., 2000, "Cyclone gasifier and cycle combustor for the use of biomass derived gas in the operation of a small gas turbine in co-generation plant", Fuel, 83: 2381-2392.
5. Al-attab K.A., Zanial Z.A, 2010, "Turbine startup methods for externally fired micro gas turbine (EFMGT) system using biomass fuels", applied energy, 87: 1336-1341.
6. Turn S., Kinohita C., Zhangt Z., Ishimura D., Zhous J., 1998, "An experimental investigation of hydrogen production from biomass gasification" Int J. Hydrogen Energy 23:641-644.
7. Sontag, Borgrakke, Wylen Van, "Fundamentals of Thermodynamics", Tata McGraw Hill, New Delhi.

7. MAILING ADDRESS

SUDIP GHOSH
Department of Mechanical Engineering,
Bengal Engineering & Science University, Shibpur
Howrah -711103, West Bengal, India
Phone **:** +91 33 2668 4561 (ext 279)
E-mail **:** *sudipghosh.becollege@gmail.com*