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THERMAL PERFORMANCE OF AN INDIRECTLY HEATED BIOGASIFICATION BASED COMBINED CYCLE PLANT EMPLOYING RECIPROCATING COMPRESSOR

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ABSTRACT

Biomass based Indirectly Heated combined heat and power plant can be an excellent option for community scale power generation. The thermal performance of such a plant employing air-turbo generator block and bottoming steam turbine block has been analyzed over a range of pressure ratio (2-16) and for different turbine inlet air temperatures (1073, 1273 and 1373K). The conventional GT combustion chamber is replaced by a burner and heat exchanger. The conventional axial compressor-gas turbine assembly is replaced by double acting reciprocating compressor and a turbo generator assembly. The main objective of using reciprocating compressor is to reduce the work requirement. Double acting reciprocating compressor is used instead of single acting to avoid the pulsating flow. The simulation of thermodynamic model is done by Athena Visual Studio software. For base case configuration, (topping cycle pressure ratio 4, gas flow rate of 90 Nm3/hr and turbine inlet temperature (TIT)= 1273K) the turbocharger unit provides about 35.9 kW electrical output, while work requirement for compressor and pump unit is about 15.04 kW respectively. This work is supplies by the steam cycle which is about 17.05 kW. It is seen from the analysis that up to the pressure ratio 5 (cut-off point) the steam cycle can drive the work consuming components and beyond this pressure ratio an auxiliary power source is required to drive them. It is also seen from the analysis at TIT 1073 K this cut-off point is bellow pressure ratio 4 while at TIT 1373 K the cut-off point is above pressure ratio 5.

Keywords: indirectly heated, reciprocating compressor, turbo generator, CHP plant, community scale, cut-off point.

1. INTRODUCTION

Biomass as a renewable energy source is gaining increasing acceptance 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 [1]. Worldwide biomass ranked as fourth as energy resource, providing approximately 14% of the world's energy needs, particularly in rural areas for power generating as well as heating purpose in now days [2] and also having negligible adverse effect on environment [3]. Biomass fuelled cogeneration or combined heat and power generation (CHP) is been considered as a major alternative to traditional power generating systems in terms of significant energy saving and environmental conservation (carbon-dioxide neutral energy source) worldwide [4]. But, the 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. Many problems, however, arise with small scale power plants in maintaining efficient operation. Small scale downdraft and fluidized bed gasifiers with diesel or gas reciprocating engines have shown promising success in the rural areas. However, the main problem with these systems is the cost of maintenance. 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. Furthermore, the producer gas has to be cooled in order to increase the engine efficiency. Most of the current systems are using cold cleaning systems that suffer from higher maintenance cost compared to the hot cleaning systems. For the rural areas with inadequate training, the system could fail due to poor maintenance and operational procedure [5].

Design of very small-scale domestic or micro-CHP systems (10–50 kWe output) are presently difficult due to availability of a suitable prime mover. Very small scale prime movers are difficult to manufacturer in terms of promising reliable operation. Also the domestic micro-CHP system produces low thermal efficiency because of production of very small scale

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compressor and turbines with reasonable isentropic efficiencies. Also small roto-dynamic systems suffer from blade tip losses [6].

The biomass based directly 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. In this present model we have considered a burner-heat exchanger integrated unit instead of the conventional GT combustion chamber for heating of the working medium of topping cycle without affecting its composition, and therefore to protect the turbo generator's components from corrosion, erosion and deposition related problems. Moreover, clean air is used to operate the turbine engine that reduces the maintenance requirements and extends the turbine operation life. A double acting reciprocating compressor is also considered instead of axial compressor to reduce the work requirement and to avoid the pulsating flow that of single acting. To reduce the blade tip loss associated with micro-turbine, used in small scale power generation, an automotive turbocharger is used. Using vehicular turbocharger for the small scale biomass based indirectly fired combined system produces more efficiency the engine does not require any modifications unlike the micro turbine engines where the combustor chamber is totally removed thus adding to its high cost.

2. DESCRIPTION AND SCHEMATIC OF THE PROPOSED PLANT

The schematic diagram of an indirectly fired combined cycle plant using biomass derived producer gas as fuel is shown in fig 1.

Solid Biomass (e.g. wood chips/blocks, saw dust, coconut shell, rice husk etc.) is fed to a downdraft gasifier (G) to convert it into gaseous fuel (at **a**). This conversion process is take places in the presence of atmospheric air (commonly called air gasification). The gaseous fuel after passing through the scrubber (S), fine filter 1 (FF1) and safety filter (SF) enters in the combustion chamber (at **c**).

The producer gas gets combusted in the presence of atmospheric air in the combustion chamber. Excess amount of air is supplied for complete combustion and thus flue gas is generated. The flue gas enters into the heat exchanger (HEX) as hot fluid (at **d**) and flows through the shell side of it. The compressed air enters into the heat exchanger through the conical shaped portion fitted at the bottom side of the heat exchanger and flows through the tube banks. The atmospheric air enters into the compressor (at **1**) and after getting compressed enters into the receiver (R) for continuous supply of air to the expander. Compressed air after leaving the receiver enters into the tube banks of the heat exchanger (at **2**) and gets heated. The air compression process is done in a reciprocating compressor to reduce the work requirement associated with the compression process. Double acting air compressor (RC) is used instead of single acting to avoid the pulsating flow of air. The hot and compressed air then enters into the expander (at **3**), which is an automotive turbocharger (TG) considered in the present study.

A heat recovery unit (HRSG) is also used to recover the exhaust heat from the turbocharger unit (at **4**) and the exhaust from heat exchanger unit (at **e**). The exhaust from these two unit mixes (at **5**) and enters into the heat

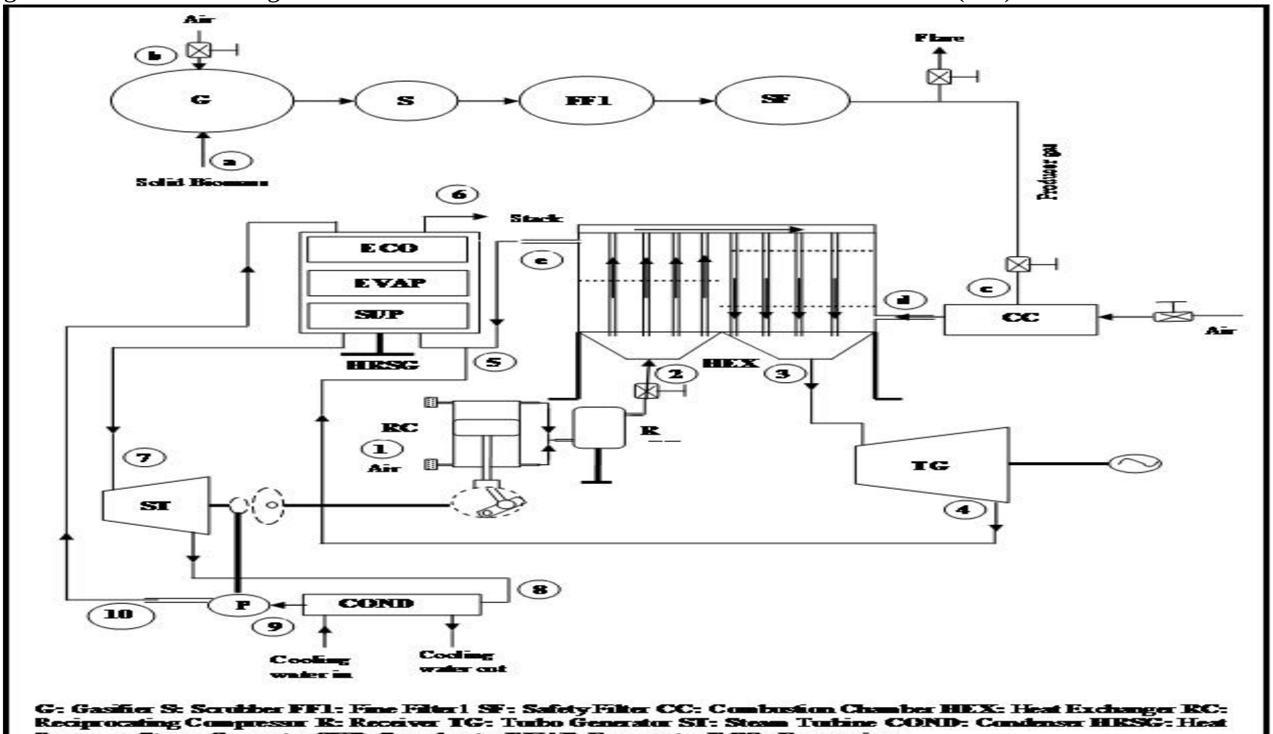

Fig 1: Schematic diagram of the proposed plant.

recovery steam generator (HRSG) unit to operate a bottoming steam cycle.

The purpose of integrating the above unit is to drive the work consuming components (which are the reciprocating compressor and the feed pump). The HRSG unit consists of three heat exchangers which are superheater (SUP), evaporator (EVAP) and economizer (ECO). Pinch point temperature (15 ^{0}C) is maintained in the evaporator for better heat transfer between the water and the flue gas mixture. The gas mixture after transferring heat to the working medium of the steam cycle gets exhausted through stack (at **6**) The high pressure water after being pumped by the feed pump (P) enters into the HRSG unit (at **10**) and converted into steam. The super heated steam enters into the steam turbine (at **7**) and gets expanded. The driving shafts of the reciprocating compressor and feed pump are connected to the shaft of the steam turbine (ST). After the expansion process, water enters (at **8**) into the condenser (COND). The condensed water enters into the (at **9**) feed pump and thus the cycle is completed.

3. METHODOLOGY

A fixed bed downdraft gasifier converts the woody biomass into gaseous fuel. The producer gas consists of CO, CO_2, H_2, CH_4, N_2 gases in the mixture (given in table 1). We have considered the gas composition derived from a particular gasifier model (Ankur Gasifier Model) and have varied the gas flow rate considering the same model.

The mass flow rate of producer gas (m_p) corresponding to the volume flow rate is calculated as

$$m_p = (MW_p / 22.4) N_p \quad (1)$$

Table 1: Gasification data [7]

Parameter	Unit	Value
Producer gas temperature	K	333
Gas flow rate	Nm^3/hr	94
Heating value	kJ/kg	4200
Cp	kJ/kg.K	1.168
Gas Composition		
CO	%vol	20.5
CO_2	%vol	11.5
H_2	%vol	19
CH_4	%vol	3
N_2	%vol	50

After calculating the mass of each constituent present in the producer gas (from vol. percentage) the thermal rating of producer gas calculated as

$$Q_p = m_p * HV_p \quad (2)$$

The producer gas gets combusted in the presence of atmospheric air. The stoichiometric oxygen requirement and finally, air requirement (considering 50% excess air) for combustion of producer gas is calculated as

$$m_{comba} = \{(100/23.2) * m_{combO_2}\}(1 + EA) \quad (3)$$

Where, m_{combO2} and EA represents the mass of stoichiometric oxygen requirement and excess air.
After complete combustion of producer gas flue gas is generated in the form of CO_2, H_2O, N_2 and O_2 (present because of excess air). Mass coefficient of each species is calculated from the combustion equation.
Specific heat capacity value for air as well as flue gas mixture is calculated using polynomial of temperature.
Temperature generated (T_{gcomb}) after the combustion of producer gas is calculated as

$$Q_p = m_p C_{pg}(T_{gcomb} - T_{gasiout}) + m_{comba} C_{pa}(T_{gcomb} - T_{ambianta}) \quad (4)$$

Total gas flow rate through heat exchanger is calculated as

$$m_{fluegasHEX} = m_{comba} + m_p \quad (5)$$

Now, Work required to drive the double acting reciprocating compressor (Wc) is calculated as

$$W_C = \frac{2n\gamma}{\gamma - 1}\left(\frac{m_{TGa}}{2}\right) R T_1 [(r)^{\frac{\gamma-1}{n.\gamma}} - 1] \quad (6)$$

Table 2: Assumptions for topping and bottoming cycle

Parameter	Unit	Value
Ambient air temperature	K	298
Air flow rate	kg/hr	0.1
Polytropic index of compression and expansion		1.4
Mechanical efficiency of compressor	%	85
Isentropic efficiency of turbocharger	%	90
Generator efficiency	%	95
Steam turbine inlet temperature	^{0}C	400
Steam turbine inlet pressure	Bar	5
Condenser inlet pressure	Bar	0.075
Min. Pinch point temp. Difference	^{0}C	15
Mechanical efficiency of steam turbine	%	95

Flue gas mixture enters into the heat exchanger for heating the working medium of the topping cycle. The effectiveness of the heat exchanger is 80%. Now, gas turbine inlet temperature (T_3) is calculated as

$$\varepsilon = \frac{C_{max}(T_3 - T_2)}{C_{min}(T_{gcomb} - T_2)} \quad (7)$$

And also heat exchanger outlet temperature (T_e) is calculated as

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$$\varepsilon = \frac{C_{\min}(T_{gcomb} - T_e)}{C_{\min}(T_{gcomb} - T_2)} \quad (8)$$

Where, T_2 is the compressor outlet temperature (calculated from adiabatic relations) and

$C_{\max} = m_{TGa} * C_{pa}$

$C_{\min} = m_{fluegasHEX} * C_{pfluegas}$

For, r_p <=4 the compressor acts as single stage compressor and r_p >4 the compressor is double stage. Based on turbocharger inlet temperature, ideal and actual outlet temperature (T_4) is calculated (considering the isentropic efficiency of the compressor).

Work done by the turbo-generator is calculated as

$$W_{TG} = \eta_G m_{TGa} C_{pa}(T_3 - T_4) \quad (9)$$

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.

Total gas flow rate through HRSG is given by

$$m_t = m_{fluegasHEX} + m_{TGa} \quad (10)$$

The temperature generated after mixing process is calculated as

$$m_{TGa} C_{pa} T_4 + m_{fluegasHEX} C_{pfluegas} T_e = m_t C_{pfluegas} T_5 \quad (11)$$

Approximate mass flow rate of water in HRSG can be calculated from the following equation

$$m_W (h_7 - h_{10}) = m_t C_{pfluegas}(T_5 - T_6) \quad (12)$$

Initially, T_6 is taken as 120 ^{0}C.

A pinch point temperature difference of 15^{0}C is maintained for better heat transfer. Work required to drive the pump is calculated as

$$W_p = 100 * V_{water}(P_7 - P_{10}) \quad (13)$$

Work done by the steam turbine is calculated as

$$W_{ST} = m_W (h_7 - h_8) \quad (14)$$

Net work output from combined cycle is calculated as

$$W_{NET} = (W_{TG} + W_{ST}) - (W_C + W_P) \quad (15)$$

Finally, efficiency of the plant is calculated as (based on producer gas composition)

$$\eta_{CC} = \frac{(W_{TG} + W_{ST}) - (W_C + W_P)}{Q_P} \quad (19)$$

4. RESULTS AND DISCUSSIONS

The performance analysis of the conceptualized plant is carried out with the help Athena Visual Studio. Codes were written in Athena Visual Studio in accordance with the thermodynamic model of the plant discussed in the previous section. The base case performance (at topping cycle pressure ratio 4) of the two plants is shown in the following table.

Table3: Base case performance of the plant

Parameter	Unit	Value
Topping cycle pr. Ratio	-----------	4
Rated gas flow	Nm^3/hr	90
Turbo-generator output	kW	35.97
Turbine inlet temperature	K	1273
Steam generation rate	kg/s	0.0255
Steam turbine output	kW	17.05
Compressor & pump work input	kW	15.04
Combined cycle efficiency	%	25.72

Now, the overall plant performances are illustrated in following graphs by varying the topping cycle pressure ratio and gas flow rate in a wide range.

Variation in combined cycle work output with the topping cycle pressure ratio is shown in fig 2. For different TITs, as pressure ratio increases the turbine inlet temperature increases and so turbo-generator work output increases (shown in figure 3), resulting the increased trend in combined cycle work output (except TIT=1073 K).

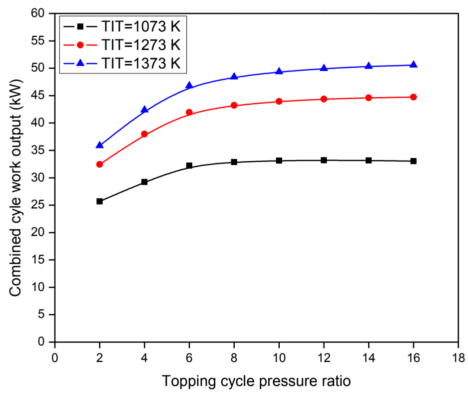

Fig.2. Variation of combined cycle work output with topping cycle pressure ratio.

The mixed stream from turbine and heat exchanger outlet is used for steam generation and so to drive the steam turbine. Again the turbine outlet temperature also goes on decreasing as the pressure ratio increases. Now, the turbine outlet temperature is higher at higher pressure ratios and at higher TITs but lower at lower TITs. The mixed stream temperature follows the same trends. Thus, the combined cycle work output goes on increasing at higher TITs and higher pressure ratios but the same initially increases and then decreases with increase in pressure ratio at lower TITs.

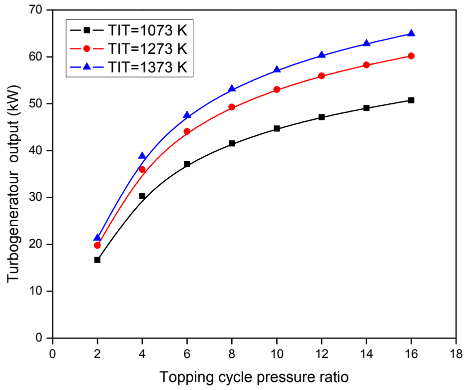

Fig.3. Variation of turbo generator work output with topping cycle pressure ratio.

The variation of turbo generator work output with topping cycle pressure ratio is shown in fig 3. As pressure ratio increases the TITs also increases and so turbo-generator work output increases. Thus the graph shows an increasing nature of turbo generator work output with increased pressure ratio.

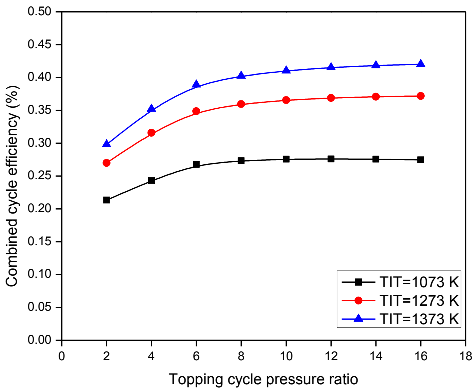

Fig.4. Variation of combined cycle efficiency with topping cycle pressure ratio.

The variation in combined cycle efficiency (based on producer gas composition) is shown figure 4.The graph follows the same trend as the combined cycle work output follows.

The effect of increased pressure ratio on steam turbine work output and combined work input to compressor and feed pump is shown in figure 5 to see the variations of cut-off points. As the TIT increases the steam generation rate also increases and so the steam turbine work output. From the graph it is clear that the plant may be operated at higher pressure ratios considering the fact that the TIT is also to be higher

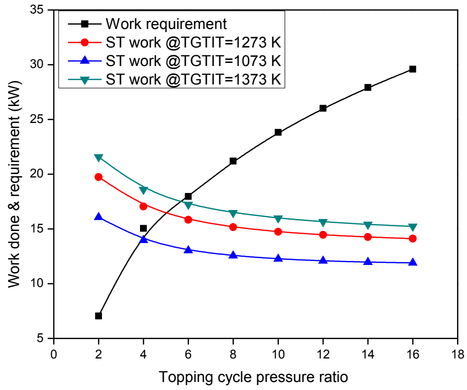

Fig.5. Variation of work done and work requirement of different components with topping cycle pressure ratio.

The variation in combined cycle work output and efficiency with topping cycle pressure ratio and variable producer gas flow rate are shown in the following figures. The TIT (=1273 K) is being kept constant. The producer gas flow rate is being varied at regular interval of 25 Nm^3/hr from base case value. Although three different turbine inlet temperatures are considered for our present study, but the graphs for TIT=1273K are presented to predict the performance.

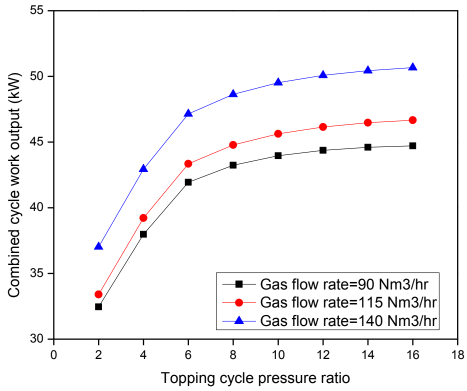

Fig.6. Variation of combined cycle work output with topping cycle pressure ratio (TIT=1273K).

The variation of combined cycle work output with topping cycle pressure ratio at turbine inlet temperature of 1273K is shown in figure 6. As the gas flow rate is increasing along with the pressure ratio the turbo generator work output and steam turbine work output also increasing. The increasing effect of work output from the both components leads to the increase in combined cycle work output.

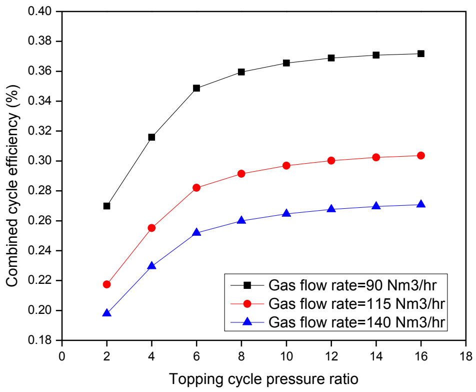

Fig.7. Variation of combined cycle efficiency with topping cycle pressure ratio (TIT=1273K).

Figure 7 shows the variation in combined cycle efficiency with topping cycle pressure ratio at turbine inlet temperature of 1273 K. From the figure 6 it is clear that as the gas flow rate increases the combined cycle work output also increase. This increase in combined cycle work output is not as much as the gas flow rate. Hence the efficiency of the plant is decreasing as the gas flow rate is increasing. The same trends of the graphs for TIT=1073K and TIT-1373K can be obtained.

5. CONCLUSIONS

The optimized thermodynamic performance of an indirectly heated BIGCC plant operating on reciprocating compressor and turbo generator is carried out in this paper using First Law of Thermodynamics. This analysis shows for base case configuration, the plant provides an efficiency of 25.72%. For a particular gas flow rate (=90Nm^3/hr), the turbo generator output increases with increase in pressure ratio at different turbine inlet temperatures. The turbo generator provides more output at higher TITs (shown in figure 3). The plant may be operated at higher pressure ratios considering the higher TIT (shown in figure 5). But the combined cycle work output and efficiency shows a flat nature at higher pressure ratios (shown in figure 2 and figure 4). Now from this part of analysis it is clear that the plant may be operated at turbine inlet temperature of **1273 K** and at pressure ratio **5** (Cut-off point at 1273 K). Performance of the combined cycle plant at TIT=1273 K and at different gas flow rate, with variation in topping cycle pressure ratio is presented in figure 6 and figure 7. It is clear from the above figures that as the gas flow rate increases the combined cycle work output increases but the efficiency decreases. It is viable to operate the plant at the gas flow rate of **115 Nm^3/hr** considering the fact of moderate combined cycle efficiency and work output.

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NOMENCLATURE

Symbols

c	Specific Heat
EA	Excess Air
HV	Heating Value (MJ/kg)
HEX	Heat Exchanger
H	Enthalpy
m	Mass Flow Rate
MW	Molecular Weight
n	Number of Stages
N	Producer Gas Flow Rate (Nm3/hr)
P	Pressure
R	Universal Gas Constant
r_p	Pressure ratio
Q	Thermal Rating
T	Temperature
V	Specific Volume
W	Work Done
η	Efficiency
ϵ	Effictiveness

Subscripts

C	Compressor
ambienta	Ambient Air
combO2	Oxygen Required for Combustion
comba	Air Required for Combustion
fluegasHEX	Flue Gas Heat Exchanger
g	Generator
gcomb	Generated After Combust
gasiout	Gasifier Outlet
p	Producer Gas
TG*a*	Turbo Generator Cycle Air
TG	Turbo Generator

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ST	Steam Turbine
w	Water

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