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Simulation of Producer Gas Fired Power Plants with Inlet Fog Cooling and Steam Injection

Mun Roy Yap and Ting Wang
Energy Conversion and Conservation Center
University of New Orleans
New Orleans, Louisiana 70148-2220, USA

ABSTRACT

Biomass can be converted to energy via direct combustion or thermo-chemical conversion to liquid or gas fuels. This study focuses on burning producer gases derived from gasifying biomass wastes to produce power. Since the producer gases are usually low calorific values (LCV), the power plants performance under various operating conditions has not yet been proven. In this study, system performance calculations are conducted for 5MWe power plants. The power plants considered include simple gas turbine systems, steam turbine systems, combined cycle systems, and steam injection gas turbine systems (STIG) using the producer gas with low calorific values at approximately 30% and 15% of the natural gas heating value (on a mass basis). The LCV fuels are shown to impose high back compressor pressure and produces increased power output due to increased fuel flow. Turbine nozzle throat area is adjusted to accommodate additional fuel flows to allow compressor operate within safety margin. The best performance occurs when the designed pressure ratio is maintained by widening nozzle openings, even though the TIT is reduced under this adjustment.

Power augmentations under four different ambient conditions are calculated by employing gas turbine inlet fog cooling. Comparison between inlet fog cooling and steam injection using the same amount of water mass flow indicates that steam injection is less effective than inlet fog cooling in augmenting power output. Maximizing steam injection, at the expense of supplying the steam to the steam turbine, significantly reduces both the efficiency and the output power of the combined cycle. This study indicates that the performance of gas turbine and combined cycle systems fueled by the LCV fuels could be very different from the familiar behavior of natural gas fired systems. Care must be taken if on-shelf gas turbines are modified to burn LCV fuels.

NOMENCLATURE

BIGCC	Biomass Integrated Gasification Combined Cycle
CC	Combined Cycle
GT	Gas Turbine
HHV	High Heating Value (MJ/kg, Btu/lbm)
HP	High Pressure
HRSG	Heat Recovery Steam Generator
IGV	Inlet Guide Vane
IP	Intermediate Pressure
LCV	Low calorific value
LHV	Low Heating Value (MJ/kg, Btu/lbm)
LP	Low Pressure
m	Mass flow rate (kg/s)
MWe	Mega Watts Electricity
NG	Natural Gas
NOx	Nitrous Oxides
η	Thermal Efficiency
p	Pressure (bar, Pa, psi)
PG	Producer gas
RAM	Reliability, Availability and Maintainability
RH	Relative Humidity (%)
ST	Steam Turbine
T	Temperature (K, °C, °F)
TET	Turbine Exit Temperature (K, °C, °F)
TIT	Turbine Inlet Temperature (K, °C, °F)

INTRODUCTION

Since the discovery of fire, biomass has been used as a fuel and energy source. Approximately 14 % of the world's population still uses biomass as an energy source (Bedi, 2001) [1], and for developing countries, biomass accounts for 35% of their fuel needs (Bedi, 2001). In the U.S., there are more than 350 biomass power plants producing about 7500 Mega Watts (MWe) of electrical power, which is enough to power several million homes (Swanikemp, 2001) [2]. Utilizing proper management and advanced scientific methods of energy production, the modern society has made biomass, a sustainable and indigenous energy source.

Biomass is an attractive alternative fuel because it is renewable, sustainable, and indigenous. A common practice in agricultural, forest, and paper industries is the direct firing of biomass upon a boiler to create processed steam or for power generation purposes via steam turbines. This conventional method is subject to low efficiency and high emissions of NO_X, CO and other pollutants. A typical small steam-boiler power system has an electric energy conversion efficiency ranging from 20% to 30%. A method to resolve the aforementioned challenges is to install a gas turbine (GT) between the combustion flame and the steam turbine. A modern gas turbine can withstand inlet gas temperatures at approximately 1430 °C / 2600 °F. Exhaust gas exiting from the gas turbine can be utilized to boil and superheat steam via a Heat Recovery Steam Generator (HRSG). Combining the gas turbine and steam turbine systems forms the combined cycle system, which is used by many of the new natural gas fired power plants. A modern combined cycle efficiency ranges from 55 % to 58 %. A higher than 60% efficiency can be achieved by using an Advanced Turbine System (Layne, 2001) [3].

Since direct firing of biomass fuel is unclean, gasification becomes a means of providing the gaseous fuel used for combustion in a gas turbine. Gasification is a partial combustion process, producing a composition of synthetic gas (syngas) or producer gas. The dominant reactant composition in the syngas is hydrogen (H_2) and carbon monoxide (CO). Producer gas, which is typically not as clean as the syngas, also contains traces of ethane, ethylene, or benzene, etc. The heating value of producer gas is low, typically around 5.5 – 7.5 MJ/Nm^3, approximately 15 % - 20 % heating value of natural gas. The producer gas needs to be cleaned to remove impurities like particles, tars and other trace chemical elements before entering the gas turbine. In contrast to a conventional steam-boiler turbine system in which the exhaust gases are cleaned after the boiler or sometimes no cleaning is performed at all, in a gasification system the fuel is cleaned before entering the gas turbine system. This in turn results in a reduction in emissions, cost, and maintenance.

Biomass Integrated Combined Cycle (BIGCC) --- The combined cycle, flue-gas cleaning and gasification are currently existing technologies. The main challenge is integrating these three technologies together to produce an affordable, highly efficient, environmentally friendly, and reliable system. New technology for using biomass in an integrated gasification combined cycle (BIGCC) is in development and is subject to continuous improvement. Some of the technical challenges require in-depth understanding of the fundamental mechanisms involved in the process to overcome them. Other challenges require development efforts in pulling together existing technologies to make it work. The ultimate challenge is to build demonstration plants, accumulate operating experience, and continuously improve the performance.

Since the producer gases are usually low calorific values (LCV) with less than 15% of the natural gas heating values, six to ten times more of fuel mass flow is required to provide the adequate heating values to achieve rated power output. Therefore, additional fuel pump power will be needed, and the compressor backpressure will be increased accordingly. Under increased compressor pressure, the compressor stability margin will be reduced, and decision needs to be made to either (a) maintain the same stability margin and pressure ratio by opening up the turbine nozzle throat area or (b) compromise the compressor stability margin (i.e. allow a narrower stability margin) but achieve a higher efficiency by operating the compressor with a higher pressure ratio. In the meantime, if a GT is fired with LCV fuels with the equal energy as NG, LCV fuels will reduce the turbine inlet temperature (TIT) due to increased sensible heats absorbed by the non-fuel gases in the gas flow (eg. nitrogen and water vapor). Reduced TIT will result in reduced thermal efficiency. Furthermore, traditional approaches of using inlet air cooling or steam injection to augment GT output power could further strain the overloaded compressor when LCV fuels are used. It is not clear what would be the overall LCV-fired power plant performance under the influence of these factors at various operating conditions. This study is initiated to investigate potential issues and challenges of utilizing existing on-the-shelf gas turbines to burn the producer gases with the following **objectives**:

- Assess and compare the performances of power plants including single steam turbine systems (Ranking cycle), single gas turbine systems (Brayton cycle), and combined cycle systems.
- Study the effects of ambient temperature and humidity upon the overall system performance.
- Study the different HRSG sizing upon the performance of the overall system.
- Access and compare performance of gas turbine inlet air cooling and steam injection (STIG) into combustion chamber.

The comprehensive analysis and assessment are documented in the report by Yap and Wang [4]. This paper reports part of the results.

COMPUTATIONAL TOOL AND SIMULATED CASES

The commercial software Thermoflow (release 13, including THERMOFLEX, GT PRO and STEAM PRO) is employed in this study. The input information includes GT model, ambient conditions, general power plant requirements and GT setup (inlet cooling, stem injection, water injection, etc), fuel type, optional combined cycle with related components, and economic criteria. During the simulation, the fuel is continuously added until the exhaust temperature reaches the designed value for the natural gas fired system. The simulation performs iteration by first calculating the combustion process using the given air mass flow rate and producing the result of the exhaust temperature. From the exhaust temperature and the combustion reaction, the compressor performance margin is checked with the existing GT model to ensure that the compressor can perform the required duty. If the compressor is not able to compress the needed amount of air, the iteration of the combustion is undertaken again with reduced fuel mass flow rate, and hence, lowering the turbine inlet temperature (TIT) and the pressure ratio (for LCV cases). The iteration follows the same method as described earlier until there is a convergence, which would suggest a possible actual performance of the GT with the given inputs. The default approach described above will use the GT exhaust temperature as a guideline. TIT can replace the exhaust temperature and be specified as the operating criterion. The details can be found in the Thermoflow manual [5].

Instead of designing a new turbine to burn LCV fuels, this study focuses on selecting an existing on-shelf commercial GT and examine its performance when LCV fuels are used. In this study, Rolls Royce 501KH5 GT is selected for the 5 MWe plant simulations. A companion study is also conducted for 20MWe plants (Yap and Wang, 2004) [4]. In that case, the Siemens GT 10 is selected.

The simulations are conducted under a controlled condition using the ISO condition (59^{o}F, 60% relative humidity, and 1 atm) as the baseline case. Natural gas fired gas turbine is used as the reference case (Case1). Producer gases of various calorific values are fed into the GTs designed for natural gas. The simulations are conducted first by feeding the producer gases without modifying the GTs but fixing the turbine inlet temperature (TIT) at the maximum allowable temperature. Then modifications are made by increasing TIT (Case 2b) or opening the first stage nominal nozzle area (Case 2c) in the turbine to reduce the compression backpressure. Case 2d is followed by optimizing both TIT and nozzle area. Comparisons are made among different cases. Typical producer gases consist of hydrogen (H_2), carbon monoxide (CO), hydrocarbons, carbon dioxide (CO_2), nitrogen (N_2), water (H_2O), tar, alkali, and other volatiles. For this study, three representative producer gases derived from biomass are used as shown in Table 1.

The compositions of first two LCV fuels are obtained from the Biomass Gasifier in Hawaii (Ishimura, D. M. et. al, 1999) [6]. The lowest heating value 4.32 MJ/kg includes nitrogen while the medium heating value of 7.40 MJ/kg does not include nitrogen. The third producer gas, with a heating value of 10.31 MJ/kg, simulates a preheated dry syngas derived from an oxygen blown gasifier. The low heating value (LHV) of natural gas, 50.05 MJ/kg, serves as the reference. All of the fuels are assumed from the biomass gasifer delivered at a pressure of 10.34 bar and a temperature of 986 ºF (803 K) at inlet of the fuel compressor. The plant efficiency is calculated based on the fuel heating value at 298K. A summary of all the cases simulated is shown in Table 2.

Table 1: Fuel volume composition of the studied LCV producer gases

Volume composition	NG	PG 1	PG2	PG3
H_2	0 %	7.30 %	11.62 %	39.7 %
H_2O	0 %	23.00 %	36.62 %	0.00 %
N_2	0 %	37.20 %	0.00 %	0.00 %
CO	0 %	10.60 %	16.82 %	40.7 %
CO_2	0 %	14.60 %	23.26 %	19.6 %
C_xH_y	100%	7.30 %	11.68 %	0.00 %
HHV (MJ/kg) @803K	57.28	5.92	9.69	12.08
LHV (MJ/kg) @ 803K	51.62	5.02	8.23	11.16
HHV (MJ/kg) @ 298K	55.53	5.09	8.73	11.16
LHV (MJ/kg) @ 298K	50.05	4.32	7.40	10.31

RESULTS AND DISCUSSIONS

Under the ISO condition, 12 cases are simulated including: the simple cycle (gas turbine only), Rankine cycle (steam turbine only), and BIGCC (combined cycle with both gas and steam turbines). The three cycles are first simulated using natural gas followed by feeding different LCV fuels derived from biomass. The purpose of these simulations is to investigate the effects of different LHV fuels on the gas turbine and cycle performances.

(A) Simple Gas Turbine Cases 1, 2a, 2b, 2c 2d, 3 and 4

Case 1 --- Natural Gas, ISO

Case 1 is the baseline reference case burning natural gas. In this case, natural gas is preheated to 803K (986 °F) to match the temperature of the producer gases delivered from the gasifier. The LHV is 51,624 kJ/kg and HHV is 57,282 kJ/kg at 803K. One of the advantages of having a pre-heated fuel is to recover the low-grade waste heat to increase the overall thermal efficiency. In the simple cycle, the gas exhaust temperature is approximately 900K. Figure 1 shows the graphical output from Thermoflow. The states of each point is given with "p" representing pressure (bar), "T" for temperature (Kelvin), and "m" for mass flow rate (kg/s). The pressure ratio of the GT is 11.3 and the TIT is 2150 °F (1450 K). The net output power is 4.692 MWe and an efficiency of 30.32 %. This case is used as the benchmark with which the other cases are compared in Table 3.

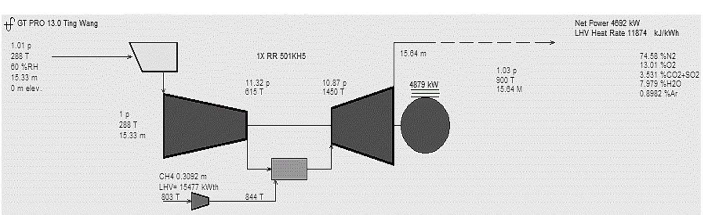

Figure 1 Thermoflow GT graphical output of Case 1, where T (K), p (bar) and m (kg/s.)

Table 2 Summary of all simulated cases

Case	Cycle	Fuel	LHV, MJ/kg	Ambient Temp, K	RH %	Notes
1	Simple	NG	51.62	288.71	60	ISO GT NG
2a	Simple	PG1	5.02	288.71	60	ISO GT PG1
2b	Simple	PG1	5.02	288.71	60	TIT Manipulation
2c	Simple	PG1	5.02	288.71	60	Nozzle Manipulation
2d	Simple	PG1	5.02	288.71	60	TIT and Nozzle Manipulation
3	Simple	PG2	8.23	288.71	60	ISO GT PG2
4	Simple	PG3	11.16	288.71	60	ISO GT PG3
5	Steam	NG	51.62	288.71	60	ST ISO NG
6	Steam	PG1	5.02	288.71	60	ISO ST PG1
7	Steam	PG2	8.23	288.71	60	ISO ST PG2
8	Steam	PG3	11.16	288.71	60	ISO ST PG3
9	Combined	NG	51.62	288.71	60	ISO GT and ST NG
10	Combined	PG1	5.02	288.71	60	ISO GT and ST PG1
11	Combined	PG2	8.23	288.71	60	ISO GT and ST PG2
12	Combined	PG3	11.16	288.71	60	ISO GT and ST PG3
13	Combined	PG1	5.02	298.15	30	Ambient variation
14	Combined	PG1	5.02	298.15	90	Ambient variation
15	Combined	PG1	5.02	305.35	30	Ambient variaton
16	Combined	PG1	5.02	305.35	90	Ambient variationn
17	Combined	PG1	5.02	298.15	30	Fog Cool Case 13 to 287.55K 100%RH
18	Combined	PG1	5.02	298.15	90	Fog Cool Case 14 to 295.93K, 100%RH
19	Combined	PG1	5.02	305.35	30	Fog Cool Case 15 to 293.13K, 100%RH
20	Combined	PG1	5.02	305.35	90	Fog Cool Case 16 303.15K 100%RH
21	Combined / STIG	PG1	5.02	298.15	90	Max Steam Case 14
22	Combined / STIG	PG1	5.02	298.15	90	Same Steam Mass Flow as Fogger Mass Flow of Case 18
23	Combined / STIG	PG1	5.02	305.35	30	Max Steam Case 16
24	Combined / STIG	PG1	5.02	305.35	30	Same Steam Mass Flow as Fogger Mass Flow of Case 19

Table 3 GT simple cycle summary of Cases 1, 2a, 2b, 2c, 2d, 3, and 4

Simple Cycle GT	ISO	ISO GT Manipulation				ISO	ISO
Case Number	1*	2a	2b	2c	2d	3	4
Ambient Temp, K (59F)	288.7	288.7	288.7	288.7	288.7	288.7	288.7
Rel. hum.	0.6	0.6	0.6	0.6	0.6	0.6	0.6
Pressure Ratio	11.3	13.4	14.0	11.3	11.0	12.7	12.0
TIT, K	1450	1383	1452	1421	1433	1451	1451
Air kg/s	15.3	15.3	15.3	15.3	15.3	15.3	15.3
Fuel LHV, MJ/kg	50.05	4.32	4.32	4.32	4.32	7.40	10.31
Fuel HHV, MJ/kg	55.54	5.10	5.10	5.10	5.10	8.73	11.16
Fuel kg/s	0.309	3.644	4.093	4.117	4.242	2.268	1.524
Fuel Compressor Power, MWe	0.080	0.605	0.680	0.684	0.705	0.400	0.324
GT Power, Mwe	4.879	6.050	6.848	7.006	7.171	5.913	5.291
GT LHV Effieicney, %	31.53	38.470	38.77	39.43	39.17	35.22	33.68
Net Power, Mwe	4.692	5.322	6.039	6.192	6.335	5.394	4.854
LHV Heat Rate, kJ / kW-hr	11874	10639	10529	10329	10403	11203	11651
LHV Plant Efficiency, %	30.32	33.84	34.19	34.85	34.60	32.13	30.90
Net Power Increase, %	0.00	13.43	28.71	31.97	35.02	14.96	3.45
Efficiency Increase, %	0.00	11.61	12.76	14.94	14.12	5.97	1.91
Heat Rate Increase, %	0.00	-10.40	-11.33	-13.01	-12.39	-5.65	-1.88
* Case 1 is the reference case for comparison with other cases. GT efficiency is based on fuel supplied at 10.34 bars and 803K (986F) from the gasifier. Net power = GT power - fuel compressor power- plant parasitic power (not shown in the table) Plant efficiency represents the entire power plant efficiency based on fuel supplied at 298K (77F).							

Case 2a --- Producer Gas 1, ISO, without modifying GT nozzle area.

This is the first case using producer gas as the fuel. The firing of producer gas in the GT combustion chamber results in a higher mass flow rate and higher back pressure. The increased backpressure is attributed to two mechanisms: (a) the irrecoverable total pressure loss due to friction in the combustor and the turbine, and (b) the recoverable increase of stagnation pressure to push more mass flow through the choked nozzle. The pressure ratio increases to 13.4, and the TIT is lowered by 120.6 °F (67 K) to 2029.73 °F (1383 K). The increased pressure ratio may result in the need for a modified compressor or adding additional stages to handle an 18.6 % increase of compression. The net power production is 5.322 MWe, which is 0.63 MWe (13.43 %) greater than a natural gas fired GT under similar ambient conditions. The additional 13.43 % of power increase is acceptable for the GT shaft and the generator as long as the increase is equal to or less than 25%. The reactants in the producer gas are composed primarily of CO and H_2. Combustion of these chemicals results in higher flame temperature, which may lead to more NOx emission and may require higher combustion liner cooling. Since the fuel is of lower calorific value, a higher mass flow rate of producer gas is needed to reach the designed TIT value. The thermal efficiency for this system is 33.84 %, about 3.52 percentage point (or 11.61%) higher than Case 1. The heat rate is correspondingly reduced by 10.4 % from Case 1.

The increased output power is contributed by increased gas flow rate and increased pressure ratio. The fuel compressor power is increased from 80 kW to 605 kW, which is consumed for compressing the producer gas to higher combustor pressure at higher fuel mass flow rate (3.644 kg/s producer gas compared to 0.309 kg/s of natural gas).

Case 2b --- Producer Gas 1, ISO, Maximum TIT

Case 2a's TIT is 120.6 °F (67 K) lower than the designed value. It is interesting to find out what the thermal efficiency would be if the TIT for producer gas fueled GT also reaches the designed value, 2150 °F (1450 K). Therefore, in Case 2b, the TIT value is specified as 2150 °F. When the option of the indirect control of TIT is selected, Thermoflow mimics the actual GT operation by using the turbine exhaust temperature to back calculate the TIT. The iteration process gives a TIT of 1452K instead of the assigned value of 1450K. The results give a net power of 6.039 MWe with a compression ratio of 14.0 and thermal efficiency of 34.19 %. In this case, the pressure ratio is increased, and the net power output is 28.7 % higher than the natural gas fired Case 1. Replacement of a stronger GT shaft and a modified compressor unit may be necessary. There is a 3.87 percentage point (or 12.76 %) increase in the efficiency while the heat rate decreases by 11.33%.

Case 2c --- Producer Gas 1, ISO, enlarged first-stage turbine nozzle area

Due to increased back pressure in both Cases 2a and 2b, the pressure ratios are higher than the designed value (11.3). With higher than the designed pressure ratio, the compressor is operated under a thin margin and instability is prone to be induced by surge or rolling stall. The partial-load performance can be questionable in Cases 2a and 2b, although the partial-load cases are not investigated in this study. To avoid the potential problems of operating the compressor too far away from the design point, in this case, the first stage nozzles openings are enlarged by 23 % to match the designed pressure ratio (11.3) as in Case 1. In addition, with the widening of the nozzle openings, the pressure ratio successfully reduces to 11.3, but the TIT is reduced to 2098°F (1421 K) while the net power increases 32% (6.192 MWe.) from Case 1. There is a need to replace the compressor, generator, and shaft due to the increased gross power. The plant thermal efficiency is rated at 34.85 %, which is 4.53 percentage points (or 14.94 %) higher than the natural gas fired Case 1.

Case 2d ---Producer Gas 1, ISO, high TIT and an enlarged first-stage turbine nozzle area

In this case, the TIT is assigned a value close to the maximum allowable value, and the first stage nozzle openings are enlarged to reduce the same pressure ratio as in Case 1. After opening the first stage turbine nozzles by 27.9% and increasing the TIT to 1433 K, the pressure ratio of 11.3 is maintained, and the net power output is 6.335 MWe with a thermal efficiency of 34.60%. Both the generator set and the turbine shaft may need to be replaced to accommodate the additional power produced.

Among the four Case 2's, enlarging the first stage turbine nozzle area without raising TIT (Case 2c) gives the highest thermal efficiency (34.85 %). However, opening the nozzle and raising the TIT to close to the maximum capacity (Case 2d) gives the highest output power (6.335 MWe), but the thermal efficiency drops 0.25 percentage point from Case 2c.

It must be noted that in the present study, although the overall plant thermal efficiency is calculated based on the fuel heating value at 298K (77F), the fuel compressor power is calculated based on the fuel supplied at 10.34bar and 803K. If the fuel compressor power is calculated based on the inlet condition at 298K and 1atm, it could require approximately 2.6MW. If this power is entirely treated as a parasitic power, the net power output and thermal efficiency for all the cases burning PG fuels will need to be re-evaluated. Most likely, a NG-fired GT would have a better thermal efficiency than a PG-fired GT. In this study, the high fuel line pressure and temperature are treated as free energy from the gasifier.

Case 3 --- Producer Gas 2, ISO

The effects upon a GT with medium calorific value producer gas as a fuel source are studied in this case. The producer gas is rated as 8,215 kJ/kg (LHV) and 9,689 kJ/kg (HHV) and fed at 986 °F (803 K). There is no adjustment to the TIT or the fist stage nozzle openings. The results give a pressure ratio of 12.7 (12.4 % higher than Case 1) with a TIT of 2152 °F (1451 K). The net power production is 5.394 MWe, which is 15 % higher than Case 1. Due to the higher fuel calorific value than in Case 2, the increased pressure ratio and net work output are acceptable to the selected GT. No specific modifications are needed. The thermal efficiency for this case is 32.13%, which is only 1.81 percentage point (or 5.97 %) higher than Case 1 and 1.71 percentage point (or 5.1 %) less than Case 2a. The fuel compressor power is 400 kW, which is lower than the fuel compressor consumption in Case 2a.

Case 4 --- Producer Gas 3, ISO

Producer gas 3 has the highest calorific value (11.156 kJ/kg LHV and 12.079 kJ/kg HHV) among the three producer gases selected in this study. The net power produced is 4.845 MWe, which is within the shaft power limits. This producer gas is actually a clean syngas with the composition purely of CO and H_2. The operating pressure ratio is 12, and the TIT is 2152 °F (1451K), both of which closely match the operating condition for natural gas fired Case 1. The plant thermal efficiency for this simulation is 30.90% and is marginally higher (1.9%) than Case 1.

In summary, the results in Table 1 show that fueling LCV producer gases to a simple gas turbine will require compressing more fuel mass flow to the combustor. The increased flue gas mass flow rate induces a higher back pressure and produces more output power. Even though the auxiliary power is required to compress a large amount of LCV fuel to a higher pressure, the overall net output power and net plant efficiency increase. The percentage of increased power and efficiency increases as the calorific value of the fuel reduces (see the increasing trend from Case 4, to Case 3 and to Case 2). More output power and higher plant efficiency can be further harnessed if the first stage nozzles openings are adjusted 23 - 28% wider to match the pressure ratio of the designed value. Typically, a higher pressure ratio and higher TIT increase the Brayton cycle efficiency. However, in the cases studied here, lower pressure ratio and lower TIT can achieve similar or slightest better performance (see Cases 2c and 2d vs. Cases 2a and 2b) because the higher pressure, induced by the LCV fuels, also increases more friction due to the significantly increased mass flow rate. In other words, the increased pressure ratio does not fully contribute to useful work.

(B) Steam Turbine Only Plant --- Cases 5, 6, 7, and 8

Case 5 --- Natural gas, steam turbine, ISO
Cases 6, 7, and 8 --- Producer gases, steam turbine, ISO

Using the ISO condition, a steam cycle is simulated with a boiler fired with natural gas. The steam turbine cycle is designed to produce roughly 5 MWe. The purpose of this case is to compare the performance differences between the standalone GT and the steam turbine (ST) with various fuels. The adiabatic flame temperature of natural gas combustion is greater than 2000 °F (1366.5 K) in the boiler with a 93.14 % thermal efficiency. The net power produced in Case 5 is 4.775 MWe with a steam cycle efficiency of 27.67 %, which is about 2.65 percentage points (8.74 %) lower than the GT performance in Case 1 (Table 4).

The same LCV producer gases used in Cases 2, 3 and 4, are used in Cases 5, 6, and 7, respectively in a steam power plant. In general, a sugarcane or wood mill needs steam for drying or for other manufacturing processes, so most plants produce steam in house by burning sugar cane bagasse or wood wastes in the boiler. In these simulated cases, no process steam is tapped, and all the steam is used for power generation. The inlet and exit condition of the steam turbine values are assigned as a standard design value regardless of the type of fuel used, and hence, the fuel is fed and fired until the designated superheated pressure (42.4 bar) and temperature (644K/700 °F) are reached.

Tables 4 and 5 summarize the results of stand-alone steam turbine performance for Cases 5, 6, 7, and 8 and their comparisons with the corresponding GT Cases 1, 2a, 3, and 4, respectively.

Table 4 Results of Cases 5 - 8 and comparisons of net electric power output and fuel mass flow rate with corresponding Cases 1, 2a, 3, and 4, respectively.

Case	Net Power MWe	Fuel Mass Flow Rate kg/s	% Power Difference	% Fuel Flow Rate Difference
5	4.775	0.340	1.8	10
6	4.782	3.63	10.1	0.9
7	4.778	2.17	11.4	4.3
8	4.784	1.60	1.4	3.7

In the GT plant, the parasitic power for compressing fuels is considered, but in the steam turbine plant, the fuel transport power is small and not considered because the furnace is operated at less than 2 atmospheric pressure. For the steam cycle cases, the effect of calorific value on the steam cycle performance is not as pronounced as in the single GT simple cycle cases. Producer gases of all three heating values render almost identical thermal efficiency at 30.54 %, 29.68 %, and 29.29 %, respectively, from lower heating value (4.32 MJ/kg) to higher heating value (10.31 MJ/kg). These efficiencies are all higher than the natural gas fired boiler in Case 5.

Table 5 Results of Cases 5 - 8 and comparisons of steam plant efficiency with corresponding gas turbine plant in Cases 1, 2a, 3, and 4, respectively.

Case	η, steam cycle %	η, boiler %	% Difference Cycle Efficiency
5	27.67	93.15	-8.7
6	30.54	91.12	-9.8
7	29.68	92.77	-7.6
8	29.29	93.81	-5.2

(C) Combined Cycles with Various fuels --- Cases 9 -12

Case 9 --- Natural gas, combined cycle, ISO
Cases 10, 11, and 12 --- combined cycle, ISO, Various producer gases

Case 9 simulates a combined cycle with the configuration of a GT, a heat recovery steam generator (HRSG), and a condensing non-reheat ST. Figures 2 shows the schematic output of Case 9, where each important state is shown next to the associated line of flow. Natural gas is fired in the same GT at ISO conditions as in Case1, which provides a benchmark reference for the combined cycle systems.

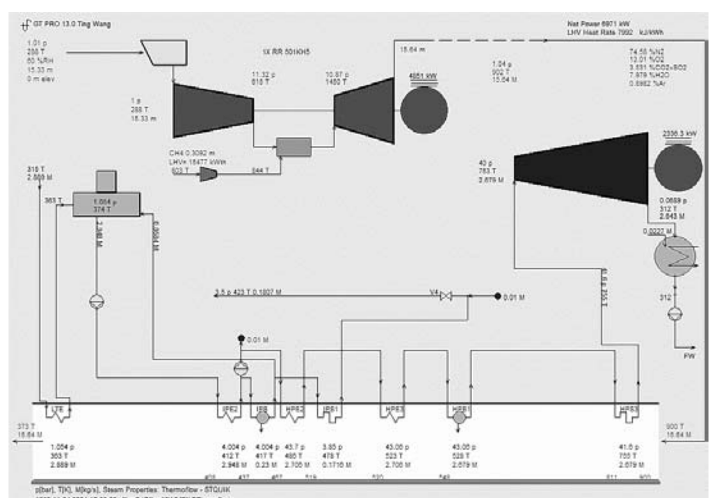

Figure 2 Thermoflow Case 9 schematic output

In Case 1, with the GT alone, thermal efficiency achieves 30.32 %. With the given combined cycle, heat is recovered via the HRSG. The superheated steam is fed into the ST to produce more power; hence the output power and the thermal efficiency are both significantly increased. The natural gas flows at 0.31 kg/s into the combustion chamber. In the combined cycle, the GT performance is identical to the cases where the GT is used as a simple cycle. The GT provides approximately 69.6 % of the total power output, while the ST provides the rest of the power. Compared with the simple GT cycle of Case 1, the net output power of the combined cycle (Case 9 in Table 6) increases 22.7%, and the plant efficiency increases 14.72 percentage points (or 48.55%).

The effects of LCV fuels to the combined cycle are similar to those on the simple GT system (efficiency increases 2 ~ 8 % and output power increases 2~13 %) because GT produces about 70% of the total plant power, and the effect of LCV fuels to the steam turbine is limited as shown in previous steam turbine Cases 6, 7, and 8. The percentage of increased power and efficiency decreases as the caloric value of the fuel increases (see the descending trend from Case 10, Case 11 to Case 12 in Table 6).

Table 6 Combined cycle performance for Cases 9, 10, 11, and 12 at ISO Condition.

Combined Cycle	**ISO**			
Case Number	9*	10	11	12
Ambient Temp, K (59 F)	288.2	288.2	288.2	288.2
Rel. hum.	0.6	0.6	0.6	0.6
Pressure Ratio	11.3	13.4	12.4	11.9
TIT, K	1450	1385	1408	1431
Air, kg/s	15.3	15.3	15.3	15.3
Fuel LHV, MJ/kg	50.05	4.32	7.40	10.31
Fuel HHV, MJ/kg	55.54	5.09	8.73	11.16
Fuel kg/s	0.309	3.659	2.123	1.483
Fuel Compressor Power, MWe	0.080	0.608	0.375	0.315
GT Power, MWe	4.851	6.067	5.508	5.133
ST Power, Mwe	2.336	2.555	2.526	2.422
Net Total Power, MWe	6.971	7.870	7.518	7.102
LHV GT Heat Rate, kJ / kW-hr	11486	9371	10278	10728
Total LHV Heat Rate, kJ / kW-hr	7992	7224	7530	7754
LHV GT Efficiency, %	31.34	38.42	35.03	33.56
Total LHV Thermal Efficiency, %	45.04	49.84	47.81	46.43
GT Power Increase, %	0.00	25.07	13.54	5.81
Total Power Increase, %	0.00	12.90	7.85	1.88
GT Efficiency Increase, %	0.00	22.59	11.77	7.08
Total LHV Efficiency Increase,%	0.00	10.66	6.15	3.09
Total LHV Heat Rate Increase,%	0.00	-9.61	-5.78	-2.98
*Case 9 is the reference case for comparisons with other cases GT Efficiency is based on fuel supplied at 10.34 bars and 803K (986F) Net power = (GT+ST) power - fuel compressor power- plant parasitic power (not shown in the table) Total efficiency is based on fuel supplied at 298K (77F).				

(D) Combined Cycles Under Different Ambient Conditions

Cases 13-16 ---Producer gas 1, combined cycle, various ambient conditions

The performance of a combined cycle burning producer gas at various ambient conditions is simulated in case 13 -16. Four representative ambient conditions are employed to examine how LCV fuels respond to ambient weather changes:

- Medium temperature and low humidity (77°F and 30 % RH)
- Medium temperature and high humidity (77°F and 90 % RH)
- High temperature and low humidity (90°F and 30 % RH)
- High temperature and high humidity (90°F and 90 % RH)

Cases 9 ~ 12 are simulated at ISO condition while cases 13 ~ 16 are simulated with four different ambient conditions as shown in Table 7. The fuel source is the producer gas 1 with 5.015 MJ/kg LHV supplied at the temperature of 986 °F (803 K). Comparing the results of Cases 13, 14, 15, and 16 with ISO Case 10, the output power of GT decreases by a range of 3.1 % to 5.7 %, while the plant total power is reduced by 2.2 % to 4.4 %. The combined cycle efficiency increases by 0.82 % to 1.38 %. There is a drop of 4.5 % to 6.7 % in compressor pressure ratio while the TIT increases by 0.9 % to 7.8 %. Although the steam turbine performance depends on the turbine exhaust temperature (TET) and the effectiveness of the HRSG, the effect of ambient condition is primarily manifested in the GT performance.

Table 7 Results of Different Ambient Conditions for Combined Cycle Cases 13 - 16 using Producer Gas1

Combined Cycle	ISO	Various Ambient conditions			
Case Number	10*	13	14	15	16
Ambient Temp, K (F)	288.1 (59)	298.1 (77)	298.1 (77)	305.3 (90)	305.3 (90)
Rel. hum.	0.6	0.3	0.9	0.3	0.9
Pressure Ratio	13.4	12.9	12.8	12.5	12.5
TIT, K	1385	1398	1397	1407	1404
Air, kg/s	15.3	14.6	14.5	14.2	14.0
Fuel LHV, MJ/kg	4.32	4.32	4.32	4.32	4.32
Fuel HHV, MJ/kg	5.09	5.09	5.09	5.09	5.09
Fuel kg/s	3.659	3.535	3.551	3.452	3.475
Fuel Compressor Power, MWe	0.608	0.587	0.590	0.573	0.577
GT Power, MWe	6.067	5.859	5.881	5.718	5.749
ST Power, Mwe	2.555	2.534	2.553	2.521	2.548
Net Total Power, MWe	7.870	7.664	7.701	7.524	7.577
LHV GT Heat Rate, kJ / kW-hr	9371	9372	9381	9377	9391
Total LHV Heat Rate, kJ / kW-hr	7224	7165	7163	7127	7125
LHV GT Efficiency, %	38.41	38.42	38.38	38.39	38.33
Total LHV Thermal Efficiency, %	49.84	50.25	50.26	50.51	50.53
GT Power Increase, %		-3.43	-3.07	-5.75	-5.24
Total Power Increase, %		-2.62	-2.15	-4.40	-3.72
GT Efficiency Increase, %		0.03	-0.08	-0.05	-0.21
Total LHV Efficiency Increase, %		0.82	0.84	1.34	1.38
Total LHV Heat Rate Increase, %		-0.82	-0.84	-1.34	-1.37
*Case 10 is the reference for comparisons with other cases. See other notes in Table 6.					

(E) Power Augmentation

To boost power production and efficiency, two power augmentation methods are considered: (a) Gas turbine fog inlet cooling and (b) Steam injected gas turbine (STIG).

Cases 17 - 20 --- Combined cycles with inlet fog cooling at four different ambient conditions, Producer Gas 1

GT fog inlet cooling has been considered as an economic and effective means to augment GT power output on hot or dry days (Nicholson, 2004) [7]. Mist or fog inlet can increase gas turbine output by cooling down the inlet air by evaporation of the fine water particles and hence reduces the compressor work. The added water increases the total mass flow rate, so it also slightly increases the output power. With increased fuel mass flow rate for using LCV producer gases, it is important to examine how the compressor is going to perform to allow more mass flow rate passing through the combustor and the turbine. The previous four cases (13 ~ 15), under different weather and humidity conditions, are simulated with GT inlet fog cooling as cases 17 ~ 20.

Figure 3 shows a representative row of foggers in front of the compressor in the diagram. The relative humidity for each Case of 17 ~ 20 is raised from 30% till 100% by assuming all the fine mist droplets are completely evaporated before entering the compressor. The water droplets are injected at 5 to 10 microns in diameter. The projected air temperature after fog cooling, but before entering the compressor, is assumed to reaching the wet bulb temperature. For example, for Case 17 after saturation fogging, the compressor inlet temperature is assumed to have reached the wet bulb temperature at 57.9 °F (287.5 K).

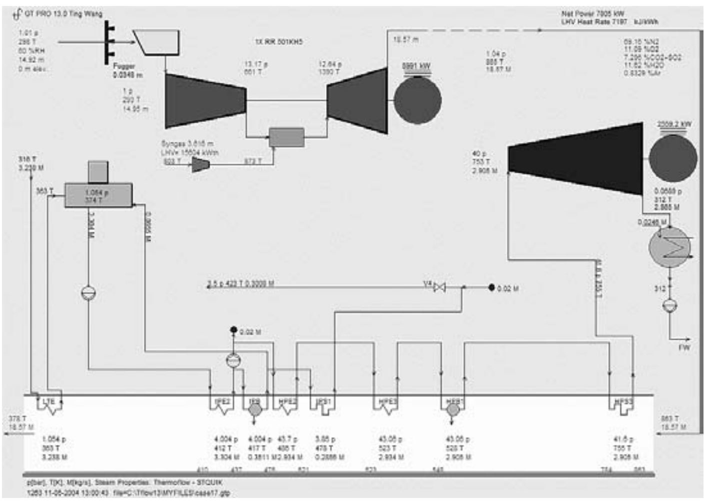

Figure 3 Thermoflow schematic output for Case 17 with inlet fog cooling using Producer Gas 1

Table 8 shows the comparison of the inlet fog cooling results with the ISO condition of Case 10. The results show inlet fog cooling will slightly increase the plant efficiency (1~2%) but not increase in the total output power when compared with the ISO condition. This is expected because the ISO condition is at relatively low ambient temperature, and at best, fog cooling can only reach the wet bulb temperature. However, the main purpose of employing the fog inlet cooling is not for improving the ISO condition; rather, it is for augmenting power for the hot or dry ambient conditions. Therefore, the interest is focused on comparing the Cases between 13 ~ 16 with 17 ~ 20 as shown in Table 9.

Table 9 shows that inlet fog cooling consistently provides power augmentation (0.4 ~ 3.7%) at dry or hot environment; however, the efficiency of each case slightly decreases (-0.11 ~ -0.97%). The power augmentation is more pronounced in the dry environment (30% RH for Cases 17 and 19) than in a hot environment (Case 20). Of course, the dry and hot environment harvests the most power augmentation out of fog cooling (Case 19) but with the worst degrading of efficiency. The result of the increased power output that accompanies the increased heat rate indicates more fuel is needed to heat up the saturated air for generating additional power at a less efficient way than the dry air.

Overall, power augmentation for a small gas turbine (5MWe) is not as good as a larger gas turbine. The use of LCV fuels further impedes the performance of fog cooling because the compressor is already heavily loaded by the increased back pressure due to excessive fuel mass flow rate. An additional increase of mass flow rate from the water vapor in a combined cycle does not augment as much power (in percentage) as the natural gas fired GT system.

Table 8 Combined Cycle Inlet Cooling Cases 17 - 20 using Producer Gas 1

Combined Cycle	ISO	Inlet Fog Cooling			
Case Number	10*	17	18	19	20
Ambient Temp, K (F)	288.1 (59)	298.1 (77)	298.1 (77)	305.3 (90)	305.3 (90)
Ambient RH	0.6	0.3	0.9	0.3	0.9
GT Inlet Temp after FogCooling , K (F)	288.1 (59)	287.5 (57.92)	295.9 (73)	293.1 (68)	303.1 (86)
RH after Fog Cooling	0.6	1.0	1.0	1.0	1.0
Pressure Ratio	13.4	13.1	12.9	13.1	12.5
TIT, K	1385	1390	1395	1390	1402
Air, kg/s	15.3	15.0	14.6	14.9	14.1
Fuel LHV, MJ/kg	4.32	4.32	4.32	4.32	4.32
Fuel HHV, MJ/kg	5.09	5.09	5.09	5.09	5.09
Fuel kg/s	3.659	3.616	3.569	3.615	3.496
Fuel Compressor Power, MWe	0.608	0.601	0.593	0.600	0.581
GT Power, MWe	6.067	5.991	5.911	5.990	5.783
ST Power, Mwe	2.555	2.559	2.557	2.559	2.554
Net Total Power, MWe	7.870	7.805	7.731	7.803	7.614
LHV GT Heat Rate, kJ / kW-hr	9371	9377	9380	9377	9390
Total LHV Heat Rate, kJ / kW-hr	7224	7197	7171	7197	7133
LHV GT Efficiency,%	38.42	38.39	38.38	38.39	38.34
Total LHV Thermal Efficiency, %	49.84	50.02	50.20	50.02	50.47
GT Power Increase, %		-1.25	-2.57	-1.27	-4.68
Total Power Increase, %		-0.83	-1.77	-0.85	-3.25
GT Efficiency Increase, %		-0.08	-0.10	-0.08	-0.21
Total LHV Efficiency Increase, %		0.36	0.72	0.36	1.26
Total LHV Heat Rate Increase, %		-0.37	-0.73	-0.37	-1.26
*Case 10 is the reference case for comparison with other cases. See other notes in Table 6.					

Table 9: Power Augmentation by Employing Inlet Fogging on Cases 13 ~16. The Results are Summarized as Cases 17~ 20

Combined Cycle	Effect of Fog Cooling at Various Ambient Conditions			
Case Number	13 -> 17	14 -> 18	15 -> 19	16 -> 20
Temperature, K & humidity, %	298, 30%--> 287, 100%	298, 90%--> 295, 100%	305,30%--> 293,100%	305, 90%-- > 303,100%
GT Power Increase, %	2.25	0.51	4.76	0.59
Total Power Increase, %	1.84	0.39	3.71	0.49
GT Efficiency Increase, %	-0.07	0.00	0.00	0.03
Total Efficiency Increase, %	-0.46	-0.12	-0.97	-0.12
Total Heat Rate Increase, %	0.45	0.11	0.98	0.11

Cases 21~ 24 (STIG) --- Combined cycles with combustor steam injection at various ambient conditions, Producer Gas 1

Steam injected into a combustor has been regularly employed in power industry to reduce NOx emissions and for augmenting output power as well. Steam injection was used in the 1940's to boost the power output of military airplane engines (Cohen, et. al., 1996) [8]. In the 1970's, steam injection was employed to lower the combustion flame temperature and reduce NOx formation (Petrotech, 2004) [9]. The performance of a steam injection cycle also depends on how the steam is produced through the heat recovery steam generator (HRSG). Four cases are simulated in STIG study with three considerations: varying ambient temperatures, employing maximum potential of steam injection, and using steam mass flow rate the same as the fogger water mass flow rate.

Cases 21 and 22 are at medium temperature (77°F) and 90% humidity. Cases 23 and 24 are at high (90°F) temperature and 60%

humidity. Case 21 and 23 are at a lower TIT where the firing temperature is kept low while maximizing the steam recovery and hence increasing the amount of steam being injected. The lowered TIT is roughly at 84 to 85 % of the designed maximum TIT of 1450 K (2150 °F / 946 °C). Cases 22 and 24 are simulated at a higher TIT with less steam recovered, and the steam injected mass flow rate is matched with similar mass flow rate of the fogger. Under the same ambient conditions, Cases 21 and 22 are compared to Case 14 and 18, respectively, while Cases 23 and 24 are compared to Cases 16 and 19 respectively.

Figure 4 shows the GT schematic with the HRSG, which supplies steam for injection into the combustion chamber via the high pressure sub-stream. The similar ambient condition in Case 14 (77°F, 90% RH) is now simulated with steam injection (Cases 21 and 22) into the combustion chamber to augment power under the condition of using LCV fuels. Case 21 investigates the cycle performance by maximizing the steam injection mass flow via the same HRSG. In Case 22, the amount of steam injection matches the same amount of water used for fog cooling in Case 18, which compares the effect of power augmentation between steam injection and fog inlet cooling. Cases 23 and 24 repeat the same simulation method as Case 21 (maximum steam injection mass flow) and 22 (matching water mass flow rate in Case 19) with the similar ambient condition of Case 16 (90 °F, 30 % RH).

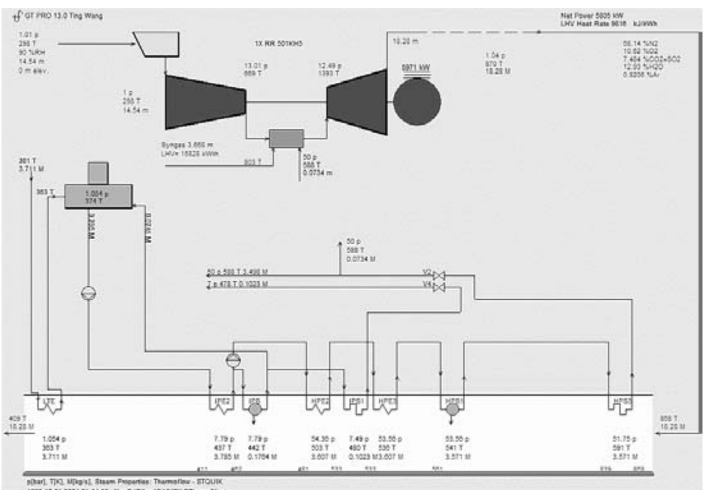

Figure 4 Thermoflow schematic output for combustor steam injection in Case 21

Table 10 summarizes the performance of steam injection Cases 21~ 24 and their comparisons with ISO Case 10. Both Cases 21 and 23 inject the maximum steam mass flow that can be generated by the HRSG at the pressure comparable to (or a bit higher than) the combustor pressure. When the steam is diverted into the combustor, the steam turbine operates at inefficient partial load. The results of Cases 21 and 23 show that the GT power increases 4~5%, and GT efficiency increases 15%. However, the steam turbine output drops 91%, and the total plant performance suffers a 25% reduction in net output power and 20 ~ 22% loss in efficiency. Basically, the increased performance in GT is not sufficient enough to make up the poor performance in the steam turbine, and subsequently, a loss to the entire plant occurs.

Steam injection does not help Cases 22 and 24 enough to augment power in comparison with the ISO condition; however, the thermal efficiency for Cases 22 and 24 is raised by 1.0 %. The slightly increased thermal efficiency of Cases 22 and 24 may be due to the use of the HRSG to recover waste heat to produce superheated steam.

Similar to the motivation of using fog inlet cooling, the purpose of using steam injection is not to augment power at the ISO condition; rather, it is for power augmentation under non-ISO conditions. The performance of Cases 21 and 23 are equally not as good when compared with Cases 14 and 16, respectively (Table 11). This indicates that maximize steam injection is not a good practice for a combined cycle designed with optimized load shares between the duties of GT and ST.

Table 10 Summary of combine cycle steam injection performance for Cases 21~24.

Case Number	10*	21	22	23	24
Ambient Temp, K (F)	288.1 (59)	298.1 (77)	298.1 (77)	305.3 (90)	305.3 (90)
Rel. hum.	0.6	0.9	0.9	0.3	0.3
Pressure Ratio	13.4	14.3	12.8	14.0	12.6
TIT, K	1385	1193	1396	1211	1404
Air, kg/s	15.3	14.5	14.5	14.2	14.2
Steam Injection (kg/s)	0	3.03	0.01	2.92	0.08
Fuel LHV, MJ/kg	4.32	4.32	4.32	4.32	4.32
Fuel HHV, MJ/kg	5.09	5.92	5.92	5.92	5.92
Fuel kg/s	3.659	3.324	3.553	3.302	3.470
Fuel Compressor Power, MWe	0.608	0.552	0.590	0.548	0.576
GT Power, MWe	6.067	6.370	5.886	6.314	5.774
ST Power, Mwe	2.555	0.236	2.565	0.230	2.484
Net Total Power, MWe	7.870	5.896	7.717	5.873	7.583
LHV GT Heat Rate, kJ / kW-hr	9371	8108	9376	8125	9337
Total LHV Heat Rate, kJ / kW-hr	7224	9321	7152	9077	7152
GT Thermal Efficiency, %	38.42	44.40	38.40	44.31	38.56
Total LHV Thermal Efficiency, %	49.84	38.62	50.34	39.66	50.34
GT Power Increase, %		4.99	-2.98	4.07	-4.83
Total Power Increase, %		-25.08	-1.94	-25.37	-3.65
GT Efficiency Increase, %		15.56	-0.05	15.33	0.36
Total LHV Efficiency Increase, %		-22.51	1.00	-20.43	1.00
Total LHV Heat Rate Increase, %		29.03	-1.00	25.65	-1.00

*Case 10 is the ISO reference case for comparisons with no steam injection. See other notes in Table 6.

Table 11 Comparison of combine cycle steam injection performance for Cases 21 – 24 vs. Cases 14, 18, 16, and 19.

Combined Cycle	Steam Injection Vs. Fog Cooling			
Case Number	21 vs.14	22 vs.18	23 vs.16	24 vs.19
Ambient Temp, K	298.1	298.1	305.3	305.3
Rel. hum.	0.9	0.9	0.3	0.3
GT Power Increase, %	8.31	-0.42	10.42	-3.61
Total Power Increase, %	-23.44	-0.18	-21.94	-2.82
GT Efficiency Increase, %	15.69	0.05	15.60	0.44
Total Efficiency Increase, %	-23.16	0.28	-21.48	0.64
Total Heat Rate Increase, %	30.13	-0.26	27.36	-0.63

Comparison of the inlet fog cooling and steam injection using the same amount of water mass flow (Cases 22 vs. 18 and Cases 24 vs. 19) indicates that steam injection provides a minor edge over fog inlet cooling in augmenting power under both dry and humid ambient conditions. Fog inlet cooling, however, shows a better efficiency than steam injection (Cases 19 vs. 24) under the dry ambient condition (30% RH). When relative humidity is high, fog cooling understandably underperforms the steam injection cases. In summary, steam performs better when expanded in the steam turbine than being injected in to the gas turbine combustor for a combined cycle system. This is especially valid for the system burning LCV fuels because the compressor is already burdened with higher back pressure. Brun, et. al. [10] specifically discussed a simplified method to evaluate the principal factors that affect the aerodynamic stability of a single shaft gas turbine's axial compressor.

Their analysis showed that when inlet and interstage water injection is combined with other factors such as LCV fuels and combustor steam injection, gas turbine compressor aerodynamic stability problems such as rotating stall and flutter will likely occur. These aerodynamic instabilities can be directly linked to blade high-cycle fatigue and possible catastrophic gas turbine failure. Furthermore, any water injection into a gas turbine (inlet, interstage, or combustor) will reduce the hot-section turbine parts life. Therefore, care must be taken to employ inlet fog cooling or combustor steam injection when LCV fuels are burned. A companion paper [11] specifically discusses the LCV fuels fired GT performance under inlet fog overspray.

(F) Limits and Uncertainty of the ThermoFlow Code

The uncertainty of the Thermoflow code consists of three major contributors:

(a) Compressor performance -- When inlet fogging is employed, the potential effect of water droplets on the compressor efficiency is not simulated. The compressor's performance is not calculated from the OEM's compressor performance map, which is usually not accessible, but estimated from a compressor performance map generated by Thermoflow.

(b) Combustor performance --- The combustor is assumed functional when burning producer gases. The combustor is treated as a black box, and the actual combustion mechanisms are not modeled. The pressure drop calculation is reasonable in the combustor

(c) Turbine performance --- To accommodate the significant increase of fuel flow, the turbine nozzle area is adjusted in the software to allow more gas to pass through. In the real engine, the turbine nozzle area cannot be adjusted by the operator; the OEM must redesign the turbine section if the same turbine efficiency is to be maintained. This could be done by enlarging the passage tip/hub ratio or redesigning the airfoils with a higher loading factor and low solidity (i.e. reducing the turbine blade number). If the airfoils are not to be changed, the turbine nozzle area can be opened up by altering the turbine blade stagger angle. By doing so, the turbine efficiency usually drops due to altered incidence angle and lift coefficient. In the Thermoflow simulation, the turbine is assumed to have the same isentropic efficiency when the turbine nozzle area is opened to accommodate more through-flow.

The heat balance errors of all the cases are less than 0.05%. For NG cases, the uncertainties of plant net power and thermal efficiency are estimated to be within 3%. For cases fired with producers' gases, the uncertainty is estimated within 5%, assuming the assumptions made in the software hold valid and the software limits are understood.

CONCLUSIONS

This study conducts performance analyses of small power plants fueled with LCV produces gases derived from biomass gasification process. The analyses are performed using the proven commercial code Thermoflow (release 13).

Three biomass derived producer gases with low heating values (LHV) of 4.32MJ/kg, 7.4MJ/kg, and 10.31MJ/kg respectively, are used. To achieve the rated power, the LCV fuels must be supplied at a flow rate approximately 4.6 to 10 times more than a natural gas fired gas turbine. These increased mass flow rates resulted in higher back pressures and increased loads on the compressor blades. To accommodate the backpressure problem, the first stage turbine nozzle openings are widened to achieve the design compression ratio of the gas turbine engine. Analysis is conducted to compare the gas turbine system performance with higher pressure ratios to those cases with the designed pressure ratio. When compared with the natural gas fueled system, the results show that typically the producer gas fueled system result in higher net output power and increased plant efficiency for 5 MWe plants based on fuel supplied from gasifier at 803K. The best performance occurs when the designed pressure ratio with widened nozzle openings are achieved, even though the TIT are relatively low (Cases 2c, and 2d). The increased power and efficiency are attributed to increased total gas mass flow rate. Without widening the nozzle openings, the pressure ratio will be higher than the designed value, and the increase of output power and the efficiency (Cases 2a) are less than the cases with widened nozzles, irrespective of the increased TIT value without widened nozzle openings (Case 2b).

When the heating value increases from 4.32MJ/kg to 7.4MJ/kg and 10.31MJ/kg, the GT net output power and efficiency are all higher than the natural gas fueled system in a subsequently reduced amount -- increase of net power is 28% ~3% and increase of efficiency is 14% ~ 2%. The similar trend repeats in the combined cycle cases: CC power increases 13% ~ 2% and efficiency increases 11% ~ 3%.

The ambient temperature and relative humidity plays a significant role in the performance of a gas turbine system. The result of 5MWe inlet fog cooling increases GT output power from 4.76 % to 0.5% (Cases 17, 18, 19, and 20) with negligible GT efficiency changes. Inlet fog cooling provides lower power augmentation for the combined cycle than for a simple GT cycle, raging from 3.7% ~ 0.39%, and adversely affects the CC efficiency ranging from -0.1% ~ -1%. The reduced benefit of inlet fog cooling for a combined cycle could be explained by the heavily loaded compressor condition, attributed to using the LCV fuels. An additional burden from the fog cooling with the increased mass flow rate results in an undesirable compressor performance and reduced TIT, both factors contribute to a marginal GT power augmentation and an adverse impact to total plant efficiency when LCV fuels are used.

Comparison between inlet fog cooling and steam injection using the same amount of water mass flow indicates that steam injection is doing worse than inlet fog cooling in augmenting power output when LCV fuels are used. The results show that the steam performs better when the steam is to be expanded in the steam turbine than being injected into the gas turbine combustor for a combined cycle system. This is especially valid for the system burned with LCV fuels because the compressor is already burdened with increased backpressure. Maximizing steam injection, at the expense of supplying the steam to the steam turbine, significantly reduces both the efficiency and the output power of the combined cycle. Therefore, maximizing steam injection is not a good practice for a combined cycle because the combined cycle is usually designed with optimized load shares between the duties of GT and ST.

The above conclusions are based on the fuel compressor power being calculated on the inlet condition at 803K and 10.34 bars. If the fuel compressor power is calculated based on 298Ka and 1 atm, and is entirely treated as a parasitic power, the net power output and thermal efficiency will need to be re-evaluated.

This study indicates that the performance of GT and CC systems fueled by the LCV fuels could be very different from the familiar behavior of natural gas fired systems. Care must be taken if on-shelf GTs are used to burn LCV fuels.

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REFERENCES

1. Bedi, Emil, 2001, "Biomass Energy," http://www.geocities.com/dieret/re/Biomass/biomass.html
2. Swanikemp, R., April 1, 2001, "Project developers consider new solid fuels, new technologies," Power; Vol. 145, Issue: 2
3. Layne, Abby W., 2001, "Advanced Turbine Systems Program," National Energy Technology Laboratory http://www.netl.doe.gov/publications/factsheets/program/prog002.pdf
4. Yap, M. R. and Wang, T., "Simulation of Producer Gas Fueled Power Plants," ECCC Report 2004-07, Energy Conversion and Conservation Center, University of New Orleans.
5. Thermoflow manual, release 13, 2004, Thermoflow Inc
6. Ishimura, D. M, Kinoshita, C. M., Masutani, S. M., Turn, S. Q., 1999, "Cycle Analyses of 5 and 20 MWe Biomass Gasifier-Based Electric Power Stations in Hawaii," Journal of Engineering for Gas Turbines and Power, Vol. 121, pp. 25-30.
7. Nicholson, Alex, 2004, "Cooling Potential Across The US," MEE Industries Inc., http://www.meefog.com/turbine/gtps-cp.html
8. Cohen, C, Rogers, GFC, Saravanamauttoo, HIH, 1999, Gas Turbine Theory, 4th ed., Longman Group Limited, Essex, England
9. Petrotech, July 1999, "Gas Turbine Steam Injection System For NOx Reduction," <www.petrotechinc.com/pdf/94023.pdf>, Product Bulletin: 94023.
10. Brun, Klaus, Kurz, R., and Simmons, H. R., 5005 "Aerodynamic Instability and Life Limiting Effects of Inlet And Interstage Water Injection Into Gas Turbines," Proceedings of ASME Turbo Expo 2005, paper GT2005-68007, Reno.
11. Khan, J. and Wang, T., " Fog and Overspray Cooling for Gas Turbine Systems Using Low Calorific Value Fuels" ASME Paper GT2006-90396, to be presented at the ASME Turbo Expo2006, Barcelona, Spain, May 8-11, 2006.