Manuscript to be sent to the Journal Applied Thermal Engineering

Effect of the fuel type on the performance of an externally fired micro gas turbine cycle

Fabiola Baina^{a,b,*}, Anders Malmquist^{a,2}, Lucio Alejo^{b,1}, Torsten H. Fransson^{a,3}

^a Department of Energy Technology, School of Industrial Technology and Management (ITM),
Royal Institute of Technology (KTH), 100 44 Stockholm Sweden

^b Facultad de Ciencias y Tecnología (FCyT), Universidad Mayor de San Simón (UMSS),
Cochabamba, Bolivia

*Corresponding author at: Department of Energy Technology, School of Industrial
Technology and Management (ITM), Royal Institute of Technology (KTH), 100 44 Stockholm
Sweden. Tel.: +46 8 790 74 42; fax: +46 8 20 41 61

E-mail addresses: fabiola@kth.se (F. Baina), andmal@kth.se (A. Malmquist)
lalejo@fcyt.umss.edu.bo (L. Alejo), fransson@energy.kth.se (T.H. Fransson).

¹ Tel.: +591 4 23 36 48; fax: +591 4 23 25 48.

² Tel.: +46 8 790 74 38; fax: +46 8 20 41 61.

³ Tel.: +46 8 790 74 75; fax: +46 8 20 41 61.

Abstract

Externally fired gas turbines open the possibility of using fuels of lower quality than conventional gas turbines and internal combustion engines. This is because in externally fired gas turbines, the flue gases heat the compressed air in a high temperature heat exchanger. This heat exchanger can more easily deal with contaminants present in the flue gases. In this regard, the configuration of externally fired gas turbines represents an interesting option for biomass gasification gas. The contaminants and low heating value (LHV) of this fuel have made it difficult to find a conversion technology for heat and power generation. For this reason, it is important to study the influence of biomass derived gas as fuel on the performance of this system and consider the effects of the contaminants in the

high temperature heat exchanger. This is studied in this work through simulations using Aspen Plus and Matlab. The test data of an externally fired micro gas turbine prototype was used to validate the simulation. The fuel considered was biomass gasification gas with varying concentrations of benzene 100, 10 and 1 g/Nm³ (hereafter named m100, m10, and m1 respectively). Additionally, mixtures of biomass derived gas and methane were studied for 10 and 50% of the thermal power of the combustor. The fuel inlet temperature to the combustor varied from 150°C to 750°C in order to represent the fuel gas after removal of particles by a cyclone and a filter. The results showed that the electrical power output increases when high fuel inlet temperatures to the combustor are used. Additionally, although it would be expected that fuels with higher LHV (lower heating value) show higher temperatures and higher output power, this does not always occur because of the composition of the fuels and their respective flue gas temperatures. The addition of methane does not have a large effect on the electrical power output. For a fixed temperature limit in the heat exchanger, the composition of the fuels does not play an important role. However, high fuel inlet temperatures to the combustor show slightly higher efficiencies. Additionally, the effect on the electrical power output of increasing the pressure drop as a result of increased thickness of deposit materials in the heat exchanger was analyzed.

Keywords: Externally fired gas turbine; biomass derived gas; electrical power output; fuel inlet temperature; lower heating value; flue gas temperature; pressure drop.

Nomenclature

$AF_{stoichiometric}$	Stoichiometric air to fuel ratio, dimensionless
$Cp_{air\ 1-2}$	Average specific heat capacity, J/kgK
$Cp_{air\ 3-4}$	Average specific heat capacity, J/kgK
Cp_{water}	Average specific heat capacity, J/kgK
k	Thermal conductivity, W/mK
$m_{air\ combustor}$	Air mass flow rate to the combustor, kg/s
$m_{air\ stoichiometric}$	Theoretical air mass flow rate to the combustor, kg/s

56	$m_{air\ to\ compressor}$	Air mass flow rate to the compressor, kg/s
57	m_{water}	Water mass flow rate, kg/s
58	m_{fuel}	Fuel mass flow rate, kg/s
59	$\eta_{generator}$	Generator efficiency, dimensionless
60	η_{mec}	Mechanical efficiency, dimensionless
61	η_{SC}	Isentropic compressor efficiency, dimensionless
62	η_{ST}	Isentropic turbine efficiency, dimensionless
63	η_{CHP}	Overall efficiency, dimensionless
64	P	Pressure, Pa
65	ΔP	Pressure drop, Pa
66	P_C	Compressor power, Watts
67	P_{el}	Electrical power output, Watts
68	P_T	Turbine power output, Watts
69	P_{th}	Thermal power output, Watts
70	T	Temperature, K
71	$T_{w,in}$	Water inlet temperature, K
72	$T_{w,out}$	Water outlet temperature, K
73	U	Overall heat transfer coefficient, W/m ² K
74	Greek letters	
75	γ	Heat capacity ratio, dimensionless
76	Subscripts	
77	1	Air inlet to the compressor

78	2	Air inlet to the heat exchanger
79	3	Air outlet from the heat exchanger
80	4	Exhaust air from the turbine
81	5	Flue gases
82	6	Hot gas inlet to the heat exchanger
83	7	Hot gas outlet from the heat exchanger

84 **Abbreviations**

85	CETD	Cold end temperature difference
86	EFGT	Externally fired gas turbine
87	IGCC	Integrated gasification combined cycle
88	LHV	Lower heating value, MJ/kg
89	TIT	Turbine inlet temperature, K

90

91

92 **1. Introduction**

93 Biomass gasification gas is a combustible gas mainly composed of H₂, CO, CH₄, CO₂, N₂, H₂O,
 94 and light hydrocarbons. It also contains tar, dust, particulates, alkali metals, ammonia and
 95 other compounds that are known as contaminants [1]. The composition of biomass
 96 gasification gas varies depending on the biomass feedstock, the type of gasifier, and the
 97 operating conditions [2].

98 There are advantages and disadvantages in the use of biomass gasification gas. The main
 99 advantages are its versatility and easy control of the combustion process. For instance,
 100 emissions from biomass derived gas can be controlled more easily than combustion of solid
 101 biomass [3]. The main disadvantages are the contaminants in the gasification gas and the

102 lower heating value (LHV). The contaminants in biomass derived gas can cause erosion,
103 corrosion, and fouling in the surfaces in contact with the gas [4]. Additionally, the gas
104 produced from gasification has a lower heating value (LHV) that can be one tenth of that of
105 natural gas [2].

106 The characteristics of biomass derived gas have made it difficult to find a conversion
107 technology capable to deal with this fuel. Steam turbines commonly use low quality fuels
108 such as biomass derived gas. These systems are suitable for large scales (MW ranges).
109 However, for small scales the water-steam technology is complex and costly [5]. Moreover,
110 at small scales, other technologies such as internal combustion engines and gas turbines
111 offer higher efficiencies [6]. Gas turbines and internal combustion engines are known to
112 require fuels with very low or no concentrations of tar, alkali metals and particulates. These
113 components can cause abrasion and fouling in the blades of gas turbines and nozzle of
114 internal combustion engines [2]. In order to fulfill these requirements, cleaning methods are
115 needed after gasification. However, the addition of these steps make these systems costly
116 and complex [7].

117 At small scales, energy conversion technologies with less strict fuel quality requirements
118 than conventional gas turbines and internal combustion engines are interesting for biomass
119 gasification gas. In this regard, externally fired gas turbines integrated with a biomass
120 gasifier can be an option since in such systems the flue gases are not in contact with the
121 turbine [7]. This makes externally fired gas turbines a fuel flexible system [5].

122 Externally fired gas turbines are mainly composed of a compressor, a turbine, a high
123 temperature heat exchanger, and a combustor. The configuration is such that compressed
124 air enters the high temperature heat exchanger where heat from the flue gases is
125 transferred to the air. Later, this hot compressed air expands in the turbine.

126 The heat exchanger is then the more critical component of the cycle. The heat exchanger is
127 expected to be able to deal with dirty fuels more easily than the turbine itself [5]. It should
128 be able to stand high temperatures for long periods of time and handle the flue gases from
129 biomass gasification gas [8]. Additionally, several parameters can affect its performance such
130 as the geometry, material and the hot and cold side fluid characteristics.

The integration of externally fired gas turbines with biomass gasification gas has been studied by some authors. Al-attab and Zainal [9] developed a small scale externally fired micro gas turbine (about 5 kWe) coupled to a biomass gasifier. The overall efficiency of the system was about 30%. Recovery of heat from the hot flue gases was suggested to increase the overall efficiency.

Vera et al. [10] modeled a 70kW_e and 160 kW_{th} externally fired gas turbine coupled to a downdraft gasifier (gasifier-EFGT). An overall and an electrical efficiency of 64% and 20% were reported respectively. A trade-off between the heat exchanger area and the electrical efficiency was carried out. Additionally, the gasifier-EFGT cycle was also compared with an integrated gasification combined cycle (IGCC) plant. Both with an electrical power output of 70kW_e. It was found that the electrical and overall efficiency of the gasifier-EFGT was higher than that of the IGCC system.

Datta et al. [11] evaluated the influence of the pressure ratio, turbine inlet temperature (TIT), and cold end temperature difference (CETD) of the heat exchanger on the thermal efficiency of a 100kW externally fired gas turbine [11]. Optimum operating conditions were found for high thermal efficiencies and for a specific air flow through the turbine. Based on the effects of these parameters, three different sets of operating conditions were studied for this system. A trade-off in the operating parameters was carried out for a 100 kW externally fired gas turbine [11].

Soltani et al. [12] studied an externally fired gas turbine system in the range of 1 MW integrated with a gasifier using an advanced exergy analysis. The results showed that the interaction between the constituents was not strong for the system proposed. Additionally, it was suggested that improving the efficiency of the heat exchanger can improve the performance of the system although higher exergy destruction was reported for the combustor and gasifier [12].

Arena et al. [7] compared technically and economically a gasifier-internal combustion engine and a gasifier-externally fired gas turbine both for a plant capacity of 200 kWe. Higher reliability and internal rate of return was offered by the first option. However, the authors indicated that decrease in costs due to increased installations of EFGT systems could make this system competitive in the future.

Cordiner and Mulone [13] evaluated the performance of an externally fired gas turbine based demonstration plant of about 70 kWe using wood as feedstock. A model of this plant was developed in order to investigate the effect of the fuel characteristics on the performance of the plant. The results showed an electrical efficiency of 13% when using dry pinecone. An almost linear increase of power output and electrical efficiency was found with increasing the higher heating value of biomass and the turbine inlet temperature [13]. A dual combustion system was proposed by Riccio and Chiaramonti [14] based on an externally fired gas turbine using solid biomass and topping combustion with natural gas to increase the efficiency. Pantaleo et al. [15] studied a techno-economic analysis of a similar system. A comprehensive review of externally fired gas turbines is presented Al-attab and Zainal [16].

Some studies considering the integration of biomass gasification gas with an externally fired gas turbine was summarized previously. As can be seen, externally fired gas turbines offer a potential option to use biomass gasification gas in a more simple way. In this work, the experimental data of a prototype plant from Compower AB [17] is used in a simulation in order to evaluate the effect of different fuels based on biomass gasification gas and mixtures with methane on the performance of the externally fired micro gas turbine prototype. The effect of materials that can deposit in heat transfer surfaces as a result of using biomass gasification gas with less strict cleaning requirements is also analyzed.

2. Methodology

The study is divided in two parts. First, a short description of the plant layout is given. Then, the simulation of the system is explained.

2.1 Description of the externally fired micro gas turbine unit

The microturbine prototype was manufactured by Compower AB [17]. This system consists of a compressor, a turbine, a high temperature heat exchanger, a natural gas burner with the possibility to operate with biogas, and a water heat exchanger. The layout of the system is shown in Fig. 1. In this prototype, the air exiting the turbine is mixed with the flue gases from the combustor in order to limit the hot side inlet temperature to the heat exchanger to

650°C [18]. This is due to limitations of the material. The high temperature heat exchanger is a corrugated counter flow plate heat exchanger [19].

The fuel used to test the operation of this system was a mixture containing 35% CO₂ and 65% CH₄ volume/volume (v/v). This fuel mixture is named biogas hereafter. The measurements from the tests at steady state conditions are shown in Table 1. Compower AB estimated an error for the measurements of around ±5%. The system was designed to produce 5 kWe and 17 kWth energy. However, during the test the power did not exceed 2 kWe. This was due to an unsatisfactory design of the compressor and turbine to reach the power of 5kWe.

In this work the integration of this prototype with a biomass gasifier is considered. The layout of the system studied is shown in Fig. 2. Removal of particles and dust is considered to some extent. For this reason, depending on the cyclone and filter to be used, the fuel inlet temperature is considered to be in a range of 150°C to 750°C. It is important to note that for this study, the gasifier, cyclone and filter did not form part of the simulation. Instead, typical composition of biomass gasification gas was taken into account.

2.2 Simulation of the system

The simulation package Aspen Plus [20] was used to simulate the complete system and Matlab [21] was used to develop a detailed model of the heat exchanger. Information regarding the detailed model can be found in [22]. The following considerations have been assumed for the study:

- The system is operating at steady state conditions.
- Complete conversion of the fuel to carbon dioxide and water is assumed.
- Thermal losses were considered negligible.
- Tar components are represented by benzene at three different concentration rates.

2.2.1 Description of the model

The components modeled in Aspen Plus [20] were the combustor, the compressor, the turbine, and the water heat exchanger. The main heat exchanger was also simulated in Aspen plus using the results obtained from the detailed model in Matlab [21].

2.2.1.1 Combustor

The combustor is designed for natural gas with the possibility to operate with biogas. The power given to the system is about 30 kW. According to the manufacturer an excess air of 20% was used in the tests. The fuel mass flow rate was then calculated by:

$$\% \text{ Excess air} = \frac{m_{air \text{ combustor}} - m_{air \text{ stoic hiometric}}}{m_{air \text{ stoic hiometric}}} \quad (1)$$

The stoichiometric air to fuel ratio is:

$$AF_{stoic \text{ hiometric}} = \left(\frac{m_{air}}{m_{fuel}} \right)_{stoic \text{ hiometric}} \quad (2)$$

For the validation of the system, biogas with a composition of CH₄ 65% and CO₂ 35 % volume/volume (v/v) was used in the model. The stoichiometric air to fuel ratio for this mix is $AF_{stoichiometric} = 6.21$.

In order to investigate the effect of biomass gasification gas and mixtures of biomass derived gas with methane on the performance of the cycle, simulations were carried out changing the composition of the fuel and the fuel inlet temperature to the combustor. The power provided by the combustor was kept constant at 30 kW for all the fuels.

In these studies, the gasifier itself was not included in the simulation. Instead, a typical composition of the product gas was used directly. This composition corresponds to wood as feedstock from a downdraft gasifier [23]. In order to represent tar, benzene was added to this composition at three different concentrations 1, 10, and 100 g/Nm³ (hereafter named m1, m10, and m100 respectively). Additionally, in order to study the effect of mixtures of biomass gasification gas with methane, two more fuel compositions were added to the analysis. These two fuel gas mixtures contained methane for 10 and 50% of the burner power. The compositions of the fuels used in the simulations are shown in Tables 2 and 3.

The fuel inlet temperature was varied for all the fuel mixtures presented in Tables 2 and 3 from 150 to 750°C [23]. In order to have comparable results, all fuels were simulated at the same temperature.

242 The chemical reactions considered for the combustion process were:

243

244 2.2.1.2 Compressor and turbine

245 For the compressor and turbine, the pressure ratios and isentropic efficiencies were
246 calculated from the measured values (Table 1). Equations (7) and (8) were used to calculate
247 the isentropic efficiencies for the compressor and turbine.

$$\eta_{sc} = \frac{T_1}{T_2 - T_1} * \left[\left(\frac{P_2}{P_1} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right] \quad (7)$$

$$\eta_{ST} = \frac{T_3 - T_4}{T_3 * \left[1 - \frac{1}{\left(\frac{P_3}{P_4} \right)^{\frac{\gamma-1}{\gamma}}} \right]} \quad (8)$$

248 2.2.1.3 Heat exchanger

249 A detailed model of the heat exchanger has been developed in Matlab [21]. The model is
250 based on a correlation that was adjusted to the experimental results. Further details of the
251 model can be found in [22]. The overall heat transfer coefficient calculated with Matlab [21]
252 has been used in the Aspen Plus [20] simulations.

253 2.2.1.4 Water heat exchanger

254 For this component, a water inlet temperature of 20°C was considered. The heat transfer
255 area corresponding to this device is about 0.83 m². The water outlet temperature was 60°C
256 or slightly above depending on the fuel type and hot side outlet temperature from the main
257 heat exchanger. The water mass flow rate was assumed to be 0.097 kg/s.

258 2.2.1.5 Electrical and thermal efficiencies

259 The electrical power output was calculated from:

$$P_{el} = (P_T * \eta_{mec} - P_C) * \eta_{generator} \quad (9)$$

From the manufacturer data, the mechanical efficiency is assumed to be 0.99 while the generator efficiency is 0.93.

Equations (10) and (11) were used to calculate the compressor and turbine power (P_C and P_T).

$$P_C = m_{air\ to\ compressor} * C_{p_{air\ 1-2}} * (T_2 - T_1) \quad (10)$$

$$P_T = m_{air\ to\ compressor} * C_{p_{air\ 3-4}} * (T_3 - T_4) \quad (11)$$

Equation (12) was used to calculate the thermal power output (P_{th}).

$$P_{th} = m_{water} C_{p_{water}} (T_{w,in} - T_{w,out}) \quad (12)$$

The electrical and overall efficiencies were calculated using Equations (13) and (14).

$$\eta_{el} = \frac{P_{el}}{m_{fuel} LHV} \quad (13)$$

$$\eta_{CHP} = \frac{P_{el} + P_{th}}{m_{fuel} LHV} \quad (14)$$

The electrical and overall efficiencies were calculated based on the fuel gas mass flow rate. The calculation was not based on the fuel mass rate of the solid biomass entering the gasifier. Additionally, the lower heating value (LHV) of the gas was used.

2.2.2 Parameter variations

The parameters that have been varied in the analysis can be divided in three parts:

First, the effect of the fuel type and fuel inlet temperature to the combustor on the performance of the cycle was analyzed. In this analysis, the composition of the fuels varied together with their corresponding lower heating value and fuel inlet temperature to the combustor.

Second, the amount of air excess air for each type of fuel was varied between 20% and 43% in order to meet the operation temperature limit for the heat exchanger (650°C).

277 Additionally, the cycle performance was studied when considering an operation of the heat
278 exchanger up to 850°C. For this case, the burner power was the manipulated parameter.

279 Third, the effect on the cycle performance of deposit materials placed in the heat transfer
280 surface was analyzed. In this analysis a range of thicknesses of deposit materials was
281 considered as such. The growth of fouling over time was not considered in this study. The
282 deposit materials from combustion of biomass gasification gas can have different
283 composition depending on the contaminants present in the fuel gas. The conductivity of
284 these deposit materials were varied in a range of values corresponding to soot with 0.07
285 W/m*K [24] to components such as alkali metals like KCl and NaCl with 2.08-3.78 W/m*K
286 [25].

287 **3. Results and discussion**

288 *3.1 Validation of the model*

289 In order to predict the behavior of the externally fired gas turbine unit with different fuels,
290 the model was validated with experimental data provided by Compower AB [17]. The
291 measured parameters against the parameters calculated by the model are shown in Tables 4
292 and 5. The measured parameters were obtained from average values extracted from the
293 tests. All the parameters are measured at steady state conditions and at a generator speed
294 of about 130,000 rpm. The measured temperatures are: outlet temperature from the
295 compressor (T_2), temperature from the hot side inlet to the heat exchanger (T_6), turbine inlet
296 temperature (T_3), and the turbine outlet temperature (T_4). The measured temperature from
297 the hot side inlet to the heat exchanger (T_6) was calculated from measured parameters such
298 as the excess air, the air to combustion mass flow rate, the compressor air mass flow rate,
299 and the turbine outlet temperature (T_4). This was due to uncertainty in the measurement of
300 T_6 . In order to compare T_6 with the calculated value from the simulation, it will be included
301 in the validation as one of the measured parameters since it is calculated from measured
302 temperatures and mass flow rates. Table 5 presents the overall heat transfer coefficient (U)
303 and the power output (P_{el}) calculated from the experimental data and the values calculated
304 by the model. The deviations of the model range from $\pm 0.2\%$ to $\pm 4.3\%$. The smallest
305 deviations are detected for the outlet temperatures from the compressor while the largest
306 deviations correspond to the electrical power output and the overall heat transfer

coefficient. These differences are due to the implications of the assumptions of isentropic efficiencies, complete combustion, and neglected heat losses in the simulation of the cycle.

3.2 Effect of the fuel type on the performance of the cycle

The fuel compositions considered for this section have been shown earlier in the text in Tables 2 and 3 with their corresponding lower heating value (LHV).

3.2.1 Effect of fuel type and fuel inlet temperature to the combustor

Fig. 3 shows the electrical power output (P_{el}) versus the fuel inlet temperature to the combustor for all the fuel types studied. Fig. 3a shows the fuels with no addition of methane while Fig.3c shows the fuels with natural gas addition for 50% of the burner power. The electrical power output for all the fuels studied is compared to that of methane and biogas. As can be seen in all cases, the electrical power output increases as the fuel inlet temperature to the combustor increases. The largest difference is seen for the fuels with no addition of methane where the electrical power output of m100, m10, and m1 is higher than that of the same fuels at 150°C by 12%, 20%, 21% respectively (Fig. 3a). Methane and biogas show the lowest increase in electrical power output.

The electrical power output (P_{el}) for the fuel mixtures m100, m100-10, m100-50, m10-50, and m1-50 exceed that of methane and biogas at a certain fuel inlet temperature to the combustor. As marked in Fig. 3, for the fuel mixture m100 (with no addition of methane) such a point is given at a fuel inlet temperature of about 660°C and 220°C for methane and biogas respectively. For the fuel mixture m100-10, this point is given at around 650°C and 150°C with respect to methane and biogas. Finally, for the fuels with the largest addition of methane, this change occurs at 580°C for m100-50 with respect of methane and at 480°C for m10-50 and m1-50 with respect of biogas. These points are marked in Fig. 3. The fuel mixtures m1, m10, m1-10, and m10-10 reach about the same power output of biogas at high fuel inlet temperatures to the combustor. As the addition of methane increases in the fuel mixture, the fuels based on biomass gasification gas resemble more the behavior of methane.

The observations previously described are due to the flue gas temperatures as indicated in Fig. 4. The flue gas temperatures for all the fuels entering the combustor at temperatures of

150°C and 750°C are shown in Fig. 4 for 20% excess air. As can be seen there is an increase of the flue gas temperature for all the fuels at 750°C compared to that at 150°C. The lowest increase corresponds to methane with 3% and the highest increase corresponds to the fuel mixture m1 with 13%.

In Fig. 4 it can also be seen that while there is a larger discrepancy of the flue gas temperatures of the fuels entering the combustor at low temperatures, this discrepancy reduces at high fuel inlet temperatures. For instance, the flue gas temperature of m1 is lower than that of methane by 12% at 150°C. However, at 750°C the flue gas temperature of m1 is lower than that of methane by 4%.

The differences mentioned previously in the flue gas temperatures are due to the different composition of the fuels and larger volume of biomass derived fuels compared to that of methane. The fuels with larger volume store more sensible heat that is later released during combustion.

Fig. 5 shows the electrical power output (P_{el}) against the fuel type and the flue gas temperature. As can be seen at 150°C (Fig.5a) the highest electrical output power corresponds to the fuels with higher flue gas temperatures. For instance, the power output of methane is higher than that of the fuel mixture with the lowest flue gas temperature (m1) by 18%. The flue gas temperature is correspondingly higher by 12%. Additionally, it can also be seen that the electrical power output of m100-50 slightly exceeds that of biogas by 3%. However, the lower heating value of biogas (LHV) is higher than that of m100-50 by 37%. For a fuel inlet temperature to the combustor of 400°C, the electrical power output of the fuel mixtures with higher concentration of benzene and higher addition of methane show about the same electrical power output of biogas. For fuels entering the combustor at 750°C, the electrical power output and flue gas temperature of methane are still higher than those of m1 by 4%. However, the difference is smaller than at 150°C. The electrical power output of m100, m100-10 and m100-50 is about the same of methane. However, the LHV of methane is six and four times larger than that of the same fuels. The electrical power output of the fuels without addition of methane is about the same of the fuels with an addition of methane for 10% and 50% of the burner power. For fuels entering the combustor at high fuel inlet temperatures, the addition of methane does not have a significant effect.

Fig. 6 shows the electrical power output against the fuel type and the turbine inlet temperature (TIT). As can be seen the highest turbine inlet temperature corresponds to the fuels with highest electrical output power for all the fuels. Comparing Figs. 5 and 6, it can also be seen that the turbine inlet temperature conducts in a similar manner as the flue gas temperatures.

For a specific heat exchanger, the temperature and mass flow rates for the cold and hot side affect the turbine inlet temperature of the externally fired gas turbine. While the parameters for the cold side are kept constant for all the fuels, the hot side temperature depends on the flue gas temperature and exhaust temperature from the turbine.

The exhaust temperature from the turbine and hot side inlet temperature to the heat exchanger are shown in Fig. 7. The exhaust temperature from the turbine and the hot side inlet temperature behave in a similar manner as the turbine inlet temperature. This relation is expected since the higher the hot side inlet temperature the higher the turbine inlet temperature will be and as a result the higher the exhaust temperature.

It would be expected that fuels with higher LHV (lower heating value) show higher temperatures and produce higher electrical output power. This is true to some extent for some fuels however the composition of the fuels, the fuel inlet temperature to the combustor, the flue gas temperature and the turbine inlet temperature also affect the system studied. It is important to note that in this work, the lower heating value (LHV) between gases is compared. For solid biomass the effect can be different as studied by Cordiner and Mulone [13].

The electrical and overall efficiencies for this system and all the fuel types are shown in Fig. 8. Both are higher for the fuels with higher concentration of benzene and higher fuel inlet temperature. The electrical efficiency varies from 5 to 7% while the overall efficiency is about 61% for all the fuels.

3.2.2 Performance of the cycle for a temperature limit in the heat exchanger

The operation temperature for the heat exchanger in this system is limited to 923K (650°C). For this system to run with the fuel compositions and inlet temperatures to the combustor previously indicated, the excess air would require to increase. This is to lower the hot side

inlet temperature to the heat exchanger to the limit for its operation. Fig. 7 shows the hot side inlet temperature of the heat exchanger for an excess air of 20%. As can be seen, most of the fuels show temperatures above 923K. Fig. 9 shows the hot side inlet temperature to the heat exchanger and electrical power output for fuel inlet temperatures to the combustor of 150°C and 400°C with 43% excess air. As can be seen, for the temperature limit of the heat exchanger, the electrical power output is about the same regardless of the fuel composition. However, the fuel inlet temperature to the combustor is lower for the fuels with higher content of benzene and higher addition of methane. For instance, the fuel mixtures m10 and m1-10 show a hot side inlet temperature of about 923K and an electrical power output of about 1.6 kW for a fuel inlet temperature to the combustor of 400°C. The fuel mixture m100 shows about the same hot side inlet temperature and electrical power output but for a fuel entering the combustor at 150°C. It can be said that for a temperature limit in the heat exchanger, the composition of the fuels does not have a significant effect on the electrical power output. The electrical and overall efficiencies of the system with an excess air of 43% are shown in Fig. 10. The electrical efficiency varies in the range of 4.8 to 6% while the overall efficiency is about 60.5 to 61%. The electrical efficiency is low due to the low operation temperature of the heat exchanger.

For a heat exchanger material able to work at temperatures of about 850°C, a higher burner power would be required in this system. Fig. 11a shows the hot side inlet temperature to the heat exchanger against the burner power for the fuel mixtures m100, m10 and m1 at different fuel inlet temperatures to the combustor. As can be seen, for a hot side inlet temperature to the heat exchanger of 1123K (850°C) a burner power of about 40 kW is required for the fuel mixture m100 at 750°C of fuel inlet temperature to the combustor. For the fuel mixtures m10 and m1 the burner power required is larger. This is due to the increase in energy content as m100 contains higher concentration of benzene. The electrical power output for all the fuels against the burner power is shown in Fig. 11b. As can be seen, the electrical power output is not greatly affected by the composition of the fuels for a burner power output corresponding to a fixed hot side inlet temperature in the heat exchanger. The operation temperature of the heat exchanger and the turbine inlet temperature have a greater effect than the composition of the fuels. The small difference in the electrical power output between fuels m100 and m1 is due to larger mass flow rates

involved for the fuel mixture m1 than for the fuel mixture m100. The electrical and overall efficiencies for this case are shown in Fig. 12. The electrical efficiency varies from 10% to 11% while the overall efficiency varies from 68 to 70%. The lowest electrical efficiency corresponds to the fuel mixture m1 at a fuel inlet temperature to the combustor of 150°C while the highest electrical efficiency corresponds to m100 at a fuel inlet temperature of 750°C. The corresponding overall efficiency for m1 is slightly larger than that of m100. This is again due to the slightly larger mass flow rate of the fuel mixture m1. It can also be seen that high fuel inlet temperatures to the combustor can result in slightly higher electrical efficiencies.

For a fixed hot side inlet temperature, the electrical power output is not greatly affected by the composition of the fuels. The hot side inlet temperature to the heat exchanger is an important parameter since it influences the turbine inlet temperature. Increasing the excess air reduces the efficiencies in the system.

3.3 Effect of deposit materials in the heat exchanger

If the use of biomass gasification gas with less cleaning requirements is considered for externally fired gas turbines, it is important to study the effect on the electrical power output of increase thicknesses of deposit materials in the heat transfer surfaces. In the system studied, the exhaust air from the turbine is mixed with the flue gases and represents the hot side mass flow rate of the heat exchanger as can be seen in Figure 1. As a result, the pressure drop in the hot side of the heat exchanger can affect the performance of the turbine. The pressure ratio and the turbine inlet temperature are the most important parameters that affect the performance of a gas turbine.

In a previous work [22] it was observed that the effectiveness of the heat exchanger depends on the characteristics of the deposit material when fouling is considered. The increase of thickness of deposit materials in this work is considered as such and it is not representing an increase over time. The buildup of fouling over time is dependent of several parameters such as the velocity of the fluids, the geometry of the heat exchangers, the fluids themselves and temperatures. In this work, a simple approach to consider the effect deposit materials was used. This approach is explained in more detail in a previous work [22]. The thermal conductivities of the deposit materials and their thickness over the heat transfer

surface are considered. Fig. 13 shows the effect on the electrical power output of different thicknesses of deposit materials and the corresponding pressure drop (ΔP) in the hot side of the heat exchanger for a range of thermal conductivities varying from 0.07 to 3.78 W/m*K. As can be seen, the electrical power output reduces for high thicknesses of deposit materials and pressure drop for low thermal conductivities. This is due to increased resistance due to fouling resulting in a decrease of effectiveness of the heat exchanger and thus a decrease in the electrical power output. For deposit materials with high thermal conductivities, the electrical power output shows an increase with the increment in the thickness of deposit materials and pressure drop. This effect is due to reduced hydraulic diameters when the thickness of deposit materials increases increasing the velocity of the fluids and hence increasing the heat transfer coefficient [22]. The reduced hydraulic diameters also increase the pressure drop and this reduces the turbine pressure ratio.

For high thermal conductivities, the effect of reduced turbine pressure ratios is not clear although the pressure drop increases. This occurs until a point in which the power output starts decreasing. This is not shown as clearly for the case of low thermal conductivity since the effect of lower working temperatures covers the effect of the pressure drop. At a thickness of deposit material of about 0.15 mm (marked with a vertical line in Fig.13) corresponding to a pressure drop of almost 3000 Pa (11% higher than the pressure drop of the heat exchanger under clean conditions) there is a variation of around $\pm 10\%$ of the electrical power output when no fouling is considered depending on the thermal conductivity of the deposit material.

4. Conclusions

Simulations of an externally fired gas turbine prototype were carried out with different fuels. The fuels considered were methane, biogas, biomass gasification gas varying the concentrations of benzene representing tar as 100, 10, and 1 g/Nm³ and the same fuel mixtures of biomass gasification gas varying the concentration of methane at levels of 10 and 50% of the burner power. Methane was used as base line to compare the fuel types. Additionally, the effect of varying the fuel inlet temperature to the combustor from 150°C to 750°C was analyzed. The base line for this analysis was the operating conditions of the fuels at 150°C. Moreover, the effect on the electrical power output of different thicknesses of

deposit materials and pressure drop in the heat exchanger was analyzed. The main conclusions are as follows:

- In general, the electrical power output increases as the fuel inlet temperature to the combustor increases. However, this increase is different depending on the fuel composition. This affects the flue gas temperatures and thus the turbine inlet temperature.
- The electrical power output (P_{GTel}) for all the fuel mixtures with higher benzene content exceeds that of methane and biogas at a certain fuel inlet temperature.
- The addition of methane to the fuel mixtures does not have a significant effect on the electrical power output.
- It would be expected that fuels with higher LHV show higher temperatures and higher electrical output power. However, the composition of the fuels affects the flue gas temperature and thus the turbine inlet temperature. Both play an important role if the hot side inlet temperature in the heat exchanger is not restricted.
- For a fixed working temperature of the heat exchanger the effect of the composition of the fuels is not significant on the electrical power output. This is because the operation temperature of the material limits the turbine inlet temperature for a specific heat exchanger. The electrical efficiency can be slightly higher for fuels entering the combustor at high temperatures.
- High pressure drops due to increased thickness of deposit materials can affect the electrical power output of the system. However, this effect is much smaller than the effect of the reduced working temperatures as the thickness of deposit materials increase.

Although the composition of the fuels does not have a large effect on the electrical power output for a specific heat exchanger, it is important to consider the fuel contents with regards fouling particles for the heat exchanger operation. High temperature operation of the heat exchanger in this system allows a better use of the sensible heat in biomass gasification gas. There is potential in this system when operating at high temperatures. The results presented in this work are important when deciding the conditions of the fuel to be used in externally fired gas turbines. These results also contribute towards further development of externally fired gas turbines using biomass derived gas as fuel.

Acknowledgements

The financial support of the Swedish International Development Cooperation Agency (SIDA), the division of Heat and Power Technology, Department of Energy Technology at Royal Institute of Technology (KTH) in Sweden, and Universidad Mayor de San Simon (UMSS) in Bolivia are acknowledged for this work. InnoEnergy “STandUP for Energy” and Compower AB are acknowledged for providing data of the externally fired gas turbine unit.

References

- [1] J. A. Ruiz, M. C. Juárez, M. P. Morales, P. Muñoz, and M. A. Mendivil, “Biomass gasification for electricity generation: Review of current technology barriers,” *Renew. Sustain. Energy Rev.*, vol. 18, pp. 174–183, 2013.
- [2] M. Asadullah, “Barriers of commercial power generation using biomass gasification gas: A review,” *Renew. Sustain. Energy Rev.*, vol. 29, pp. 201–215, Jan. 2014.
- [3] J. Held, “Gasification - Status and technology,” Malmö, Sweden, 2012.
- [4] A. V Bridgwater, “The technical and economic feasibility of biomass gasification for power generation,” *Fuel*, vol. 74, no. 5, pp. 631–653, 1995.
- [5] M. Kautz and U. Hansen, “The externally-fired gas-turbine (EFGT-Cycle) for decentralized use of biomass,” *Appl. Energy*, vol. 84, no. 7–8, pp. 795–805, Jul. 2007.
- [6] L. Dong, H. Liu, and S. Riffat, “Development of small-scale and micro-scale biomass-fuelled CHP systems - A literature review,” *Appl. Therm. Eng.*, vol. 29, no. 11–12, pp. 2119–2126, 2009.
- [7] U. Arena, F. Di Gregorio, and M. Santonastasi, “A techno-economic comparison between two design configurations for a small scale, biomass-to-energy gasification based system,” *Chem. Eng. J.*, vol. 162, no. 2, pp. 580–890, 2010.
- [8] J. Schulte-Fischedick, V. Dreißigacker, and R. Tamme, “An innovative ceramic high temperature plate-fin heat exchanger for EFCC processes,” *Appl. Therm. Eng.*, vol. 27, no. 8–9, pp. 1285–1294, Jun. 2007.

- 543 [9] K. A. Al-attab and Z. A. Zainal, "Turbine startup methods for externally fired micro gas
544 turbine (EFMGT) system using biomass fuels," *Appl. Energy*, vol. 87, no. 4, pp. 1336–
545 1341, 2010.
- 546 [10] D. Vera, F. Jurado, and J. Carpio, "Study of a downdraft gasifier and externally fired
547 gas turbine for olive industry wastes," *Fuel Process. Technol.*, vol. 92, no. 10, pp.
548 1970–1979, Oct. 2011.
- 549 [11] A. Datta, R. Ganguly, and L. Sarkar, "Energy and exergy analyses of an externally fired
550 gas turbine (EFGT) cycle integrated with biomass gasifier for distributed power
551 generation," *Energy*, vol. 35, no. 1, pp. 341–350, Jan. 2010.
- 552 [12] S. Soltani, M. Yari, S. M. S. Mahmoudi, T. Morosuk, and M. A. Rosen, "Advance exergy
553 analysis applied to an externally-fired combined-cycle power plant integrated with a
554 biomass gasification unit," *Energy*, vol. 59, pp. 775–780, 2013.
- 555 [13] S. Cordiner and V. Mulone, "Experimental-numerical analysis of a biomass fueled
556 microgeneration power-plant based on microturbine," *Appl. Therm. Eng.*, 2014.
- 557 [14] G. Riccio and D. Chiaramonti, "Design and simulation of a small polygeneration plant
558 using biomass and natural gas in a dual combustion micro gas turbine (BIO_MGT)," *Biomass and Bioenergy*, vol. 33, no. 11, pp. 1520–1531, 2009.
- 560 [15] A. Pantaleo, S. Camporeale, and N. Shah, "Natural gas-biomass dual fuelled
561 microturbines: Comparison of operating strategies in the Italian residential sector," *Appl. Therm. Eng.*, vol. 71, no. 2, pp. 686–696, 2014.
- 563 [16] K. A. Al-attab and Z. A. Zainal, "Externally fired gas turbine technology: A review," *Appl. Energy*, vol. 138, pp. 474–487, 2015.
- 565 [17] Compower AB, "Compower - Heat and power generated at home," *Compower*, 2014.
566 [Online]. Available: www.compowers.se/. [Accessed: 02-Feb-2014].
- 567 [18] CompowerAB, "Operating instruction manual - Micro gas turbine ET10," Stockholm,
568 Sweden, 2010.

- 569 [19] A. Malmquist, "Analysis of a gas turbine driven hybrid drive system for heavy
570 vehicles," Royal Institute of Technology, Stockholm, Sweden, 1999.
- 571 [20] Aspen Plus v 8.0, "Aspen Plus®." Aspen Technolog, Inc, USA, 2012.
- 572 [21] Matlab R2013b, "Matlab." The MathWorks, Inc, Natick, Massachusetts, 2013.
- 573 [22] F. Baina, A. Malmquist, L. Alejo, B. Palm, and T. Fransson, "Analysis of a high-
574 temperature heat exchanger for an externally-fired micro gas turbine," *Appl. Therm.*
575 *Eng.*, vol. 75, pp. 410–420, 2014.
- 576 [23] C. Erlich and T. H. Fransson, "Downdraft gasification of pellets made of wood, palm-oil
577 residues respective bagasse: Experimental study," *Appl. Energy*, vol. 88, no. 3, pp.
578 899–908, 2011.
- 579 [24] J. E. Hesselgreaves, "An approach to fouling allowances in the design of compact heat
580 exchangers," *Appl. Therm. Eng.*, vol. 22, no. 7, pp. 755–762, May 2002.
- 581 [25] Design Institute for Physical Properties and AIChE, *Knovel DIPPR Project 801 - Full*
582 *Version*. Design Institute for Physical Property Research/AIChE, 2012.
- 583

Table 1 Measured parameters of the prototype. Data was provided by Compower AB.

Parameters	Unit	Data 1	Data 2	Data 3
Generator speed	rpm	130,821	131,148	131,257
Ambient temperature	K	290.8	291.2	291.4
Ambient pressure	kPa	102.9	102.9	102.9
Combustor				
ΔP combustor	Pa	998.8	985.5	995.4
$\dot{m}_{\text{air combustor}}$	kg/s	0.0117	0.0119	0.0120
$\dot{m}^*_{\text{fuel}}$	kg/s	0.0014	0.0014	0.0014
Compressor				
Isentropic efficiency	-	0.70	0.70	0.70
Pressure ratio _{compressor}	-	2.39	2.40	2.40
$P_{\text{out compressor}}$	kPa	246.3	246.9	247.1
$T_{\text{out compressor}}$ ($T_{\text{in air side to HX}}$)	K	409.1	409.6	409.9
$\dot{m}_{\text{air to the compressor}}$	kg/s	0,0902	0.0916	0.0908
Turbine				
Isentropic efficiency	-	0.77	0.76	0.76
Pressure ratio _{turbine}	-	2.26	2.27	2.27
$P_{\text{in turbine}}$	kPa	241.6	242.3	242.5
$P_{\text{out turbine}}$	kPa	106.7	106.8	106.8
$T_{\text{in turbine}}$ (TIT) ($T_{\text{out air side to HX}}$)	K	857.7	862.3	864.5
$T_{\text{out turbine}}$	K	730.5	735.0	737.0
High temperature heat exchanger (HX)				
$T^*_{\text{in gas side}}$	K	918.6	927.3	930.9
$T^*_{\text{out gas side}}$	K	535.2	538.9	536.1
$\Delta P_{\text{gas side}}$	Pa	2,644	2,679	2,694
$\Delta P_{\text{air side}}$	Pa	4,690	4,632	4,599

*Data is based on calculations using mass and energy balances

Table 2

Table 2 Fuel compositions and their corresponding lower heating value

Fuel type		Biomass gasification gas composition (%volume)							LHV
		CO	H ₂	CH ₄	CO ₂	N ₂	H ₂ O	C ₆ H ₆	MJ/kg
Fuel [1g/Nm ³]	m1*	23.98	11.64	2.38	9.50	47.49	5.00	0.03	4.66
Fuel [10g/Nm ³]	m10*	23.92	11.61	2.37	9.47	47.37	4.99	0.27	4.92
Fuel [100g/Nm ³]	m100*	23.35	11.33	2.31	9.25	46.24	4.87	2.65	7.39

* Reference name

Table 3 Fuel compositions with methane addition

Fuels mixed with natural gas		Biomass gasification gas composition (%volume)							LHV
		CO	H ₂	CH ₄	CO ₂	N ₂	H ₂ O	C ₆ H ₆	MJ/kg
Fuel**[1g/Nm ³]-10%	m1-10*	23.60	11.45	3.94	9.35	46.73	4.92	0.03	5.11
Fuel**[1g/Nm ³]-50%	m1-50*	20.92	10.15	14.83	8.29	41.43	4.36	0.02	8.47
Fuel**[10g/Nm ³]-10%	m10-10*	23.52	11.41	4.03	9.31	46.57	4.90	0.27	5.40
Fuel**[10g/Nm ³]-50 %	m10-50*	20.70	10.04	15.51	8.20	41.00	4.32	0.24	8.92
Fuel**[100g/Nm ³]-10%	m100-10*	22.73	11.03	4.91	9.00	45.01	4.91	2.58	8.07
Fuel**[100g/Nm ³]-50%	m100-50*	18.74	9.09	21.61	7.42	37.10	3.91	2.13	12.85

* Reference name

** 10 and 50% is a share based on the thermal power output of the combustor

Table 4 Measured temperatures against temperatures calculated by the model

Parameters	Data 1	Data 2	Data 3
T ₂ measured, K	409.1	409.6	409.9
T ₂ simulated, K	408.3	408.8	409.1
Relative error, %	0.2	0.2	0.2
T ₃ measured, K	857.7	862.3	864.5
T ₃ simulated, K	870.7	872.9	880.2
Relative error, %	1.5	1.2	1.8
T ₄ measured, K	730.5	735.0	737.0
T ₄ simulated, K	742.4	744.8	751.4
Relative error, %	1.6	1.3	2.0
T ₆ measured, K	918.6	923.3	926.8
T ₆ simulated, K	928.5	931.2	938.7
Relative error, %	1.1	0.9	1.3
T ₇ measured, K	535.2	536.7	539.0
T ₇ simulated, K	537.7	539.2	541.6
Relative error, %	0.5	0.5	0.5

Table 5

Table 5 Measured and calculated overall heat transfer coefficient (U) and electrical power output (P_{el}).

Parameters	Data 1	Data 2	Data 3
$U_{\text{measured}}, \text{ W/m}^2\text{K}$	47.7	48.5	47.0
$U_{\text{calculated}}, \text{ W/m}^2\text{K}$	48.6	49.2	49.0
Relative Error, %	1.9	1.4	4.3
$P_{el \text{ measured}}, \text{ W}$	1724.0	1778.1	1763.0
$P_{el \text{ calculated}}, \text{ W}$	1699.3	1707.6	1757.8
Relative Error, %	1.4	4.0	0.3

Fig. 1 Layout of the Compower externally fired micro turbine prototype

Fig. 2 Layout of the system

Fig. 3 Electrical power output (P_{el}) versus fuel inlet temperature. a) Fuels without methane b) Fuels with methane addition for 10% of the burner power c) Fuels with methane addition for 50% of the burner power

Fig. 4 Flue gas temperatures of the fuels at 20% excess air

Fig. 5 Electrical power output (P_{el}) against the fuel type and flue gas temperature. a) Fuels inlet temperature 150°C. b) Fuels inlet temperature 400°C. c) Fuels inlet temperature 750°C

Fig.6 Electrical power output (P_{el}) against the fuel type and turbine inlet temperature. a) Fuels inlet temperature 150°C. b) Fuels inlet temperature 400°C. c) Fuels inlet temperature 750°C.

Fig. 7 Exhaust temperatures from the turbine and hot side inlet temperature to the heat exchanger for all the fuel types. a) Fuel inlet temperature 150°C b) Fuel inlet temperature 400°C c) Fuel inlet temperature 750°C

Fig. 8 Electrical and overall efficiency for 20% excess air

Fig. 9 Hot side inlet temperature to the heat exchanger and electrical power output for fuel inlet temperatures to the combustor of 150°C and 400°C with 43% excess air

Fig. 10 Electrical and overall efficiencies for 43% excess air and operation of the heat exchanger up to 650°C

Fig. 11 Hot side inlet temperature to the heat exchanger and electrical power output against the burner power

Fig. 12 Electrical and overall efficiency for 20% excess air and operation of the heat exchanger up to 850°C

Fig. 13 Electrical power output versus thickness of deposit materials and pressure drop in the hot side of the heat exchanger