**Extended operability of a commercial air-staged burner using a synthetic mixture of biomass
derived gas for application in an externally fired micro gas turbine**

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20 *Abstract*

21 Biomass gasification converts solid biomass into a gaseous fuel that is more versatile and can be
22 used in many applications. However, biomass gasification gas contains some contaminants and
23 inert compounds. The contaminants can cause several problems in the downstream equipment
24 and undesirable emissions while the inert compounds can affect the lower heating value of the
25 gas. Because of these characteristics, there have been difficulties in finding a conversion
26 technology using biomass gasification gas for heat and power generation. In this regard,
27 externally fired gas turbines open a possibility for this combustible gas since due to its
28 configuration, combustion takes place outside the conventional gas turbine cycle. For this
29 reason, combustion studies of biomass derived gas are important. In this work the operability
30 of a commercial air-staged natural gas burner is shown in terms of CO, UHC, and NO_x emissions
31 using a synthetic mixture of biomass gasification gas. Two fuel gas mixtures simulating the
32 composition of biomass gasification gas are injected in the combustor. Each fuel gas contains
33 different injection rates of benzene in order to represent tars and to understand their effect on
34 the combustion performance. Additionally, the equivalence ratio is varied in a range of lean
35 conditions in order to find an optimum operation point for the burner studied. The results
36 showed that the presence of polyaromatic hydrocarbons such as benzene reduced the CO
37 concentrations in the exhaust gas while it increased the concentrations of unburned
38 hydrocarbons (UHC) at equivalence ratios lower than 0.68. Additionally, NO_x emissions showed
39 a relatively constant trend over the range of equivalence ratios studied for both fuels. It was
40 also observed that NO_x emissions increase with the addition of benzene in the fuel gas. An
41 optimum point with regards CO and UHC concentrations was found for the fuels tested.

42 *Keywords:* Simulated biomass gasification gas, two stage combustion, benzene injection, NO_x
43 emissions, carbon monoxide, unburned hydrocarbons.

44 *Nomenclature*

45 LHV: Lower heating value, MJ/Nm³

46 $\dot{m}_{air}$: Air mass flow rate, kg/h

47 $\dot{m}_{fuel}$: Fuel mass flow rate, kg/h

48 T_{ad}: Adiabatic temperature, K

49 UHC: Unburned hydrocarbons, ppm

50 *Greek*

51 Φ : Overall equivalence ratio

52 *1. Introduction*

53 Biomass gasification is a thermochemical conversion process through which solid biomass is
54 converted into a combustible gaseous fuel. This gas is composed mainly of CO, H₂, N₂, CO₂, CH₄,
55 H₂O (vapor) and light hydrocarbons. In addition, it contains trace amounts of NH₃, HCN, small
56 char particles, tars, and ash known as contaminants [1, 2].

57 Compared to solid biomass, biomass gasification gas presents several advantages in application
58 and emissions reduction [3]. The gas phase allows a simpler control and regulation of the

combustion process. The use of biomass also reduces CO₂ emissions and dependence on fossil fuels [2, 4].

The main disadvantages of biomass gasification gas are the contaminants and the lower heating value of the gas. The contaminants in the producer gas can cause erosion, corrosion, difficulties when burning, internal depositions and emissions. The low heating value embodies challenges in the use of already existing natural gas burners as well as in their design. These disadvantages represent a barrier to find a conversion technology capable to use producer gas for heat and electricity generation [1, 2].

At small scales biomass gasification gas can be used in combination with internal combustion engines and gas turbines to generate heat and power. However, there are several challenges in these systems. The application of syngas with internal combustion engines and gas turbines shows technical difficulties due to the contaminants present in the combustible gas. Cleaning the producer gas can overcome these difficulties [1]. However, the system becomes complex and costly with the addition of cleaning steps [5]. An externally fired gas turbine is another technology that offers an option with regard to the contaminants in the combustible gas. The configuration of externally fired gas turbines is such that combustion takes place outside the cycle at atmospheric pressure. Fig. 1 shows a schematic of an externally fired gas turbine. The flue gas heats the compressed air that later expands in the turbine. In this way, the flue gas is not in direct contact with the turbine. This characteristic opens the possibility of using biomass gasification gas without the strict requirements of high quality fuels as it is the case for internal combustion engines and conventional gas turbines [6].

The system shown in Fig. 1 corresponds to an externally fired micro gas turbine prototype designed for a target of 5kW electrical and 17kW thermal power. The prototype was built by Compower AB. This system was initially designed and tested for natural gas. However, in order to apply biomass gasification gas in such a system, investigation regarding its combustion is necessary. Syngas combustion has been studied for premixed flames, for diffusion flames and for a mix of both. Lean premixed flames are used to lower the emissions and it is especially interesting when using natural gas. However, for syngas, flashback is likely to occur. Flashback is caused by the high diffusivity of hydrogen. Diffusion flames on the other hand, offer combustion stability compared to lean premixed flames. However, diffusion flames are also known to produce higher amounts of thermal NO_x compared to lean premixed flames [7, 8]. This characteristic of diffusion flames can be handled by wet and dry NO_x control methods [7]. Among these methods, air stage combustion is one of the effective methods to be used in diffusion flames for NO_x control [9].

Studies using syngas in diffusion flames with and without air staging are summarized as follows. Adouane et al. [10] studied the conversion of ammonia to NO using an air-staged combustor named Winnox-TUD. Ammonia was used to represent fuel-bound nitrogen from biomass derived gas. Additionally, the effect of primary air-fuel equivalence ratio and methane on ammonia conversion was evaluated. A minimum conversion of NH_3 to NO was reported at an air-fuel equivalence ratio of 0.7. The addition of methane increased the conversion of NH_3 to NO while no substantial effect was reported when varying the secondary air [10]. Similar results were reported by Hasegawa et al. [11] with regards to the effect of methane addition to the conversion of NH_3 to NO [11]. The reaction mechanisms of syngas combustion were simulated

by Chacartegui et al. [12] based on a diffusion combustor with special emphasis in injection of steam in the combustion chamber. The simulation was able to predict gaseous emissions in three different heavy duty gas turbines. The results showed that steam injection reduced NO_x formation in all the cases. Also, syngas fuels containing higher amounts of CO were reported to show higher NO_x emissions. It was also concluded that CO, unburned hydrocarbons and NO_x showed to be dependent on the residence time [12]. Huynh and Kong [13] studied NO_x emissions of biomass syngas derived from three different types of biomass and three different oxygen enriched gasification conditions. A linear decreasing trend was reported for NO_x emissions as the combustion became leaner for two of the biomass feedstock tested (pine wood and maple-oak wood) while a peak NO_x was shown for seed corn, the third biomass feedstock studied. Seed corn was also reported to produce the highest ammonia concentration in the fuel gas giving the highest NO_x emissions [13]. Giles et al. [14] studied the dilution effect of N_2 , H_2O , and CO_2 in NO_x emissions of syngas diffusion flames. The results showed that CO_2 and H_2O are more effective diluents to reduce NO_x than N_2 . The effectiveness of H_2O was addressed to its high specific heat reducing the concentration of CH radicals and hence prompt NO [14].

As can be seen from previous studies, the main concerns in combustion of biomass gasification gas are related to the stability of combustion and emission levels. Diffusion flames are a current option for stability problems of syngas combustion until lean-premixed combustors are able to overcome the stability issues of syngas. Regarding combustion performance of producer gas, most of the studies considering a synthetic mixture of biomass gasification gas did not deal with a tar representative component. Additionally, studying the effect of tars is suggested by Huynh

and Kong [13]. The present work studies the combustion of a surrogate mixture of biomass derived gas using benzene as tar representative.

This work is focused on experimental tests of combustion of a synthetic mixture of biomass derived gas using a commercial air-staged natural gas burner. The objective is to show the operability of the burner in terms of CO, UHC and NO_x emissions taking into consideration the effect of the injection rate of benzene in the combustion performance. Studying combustion of biomass gasification gas is important since incomplete combustion can generate undesirable emissions. Additionally, soot and fouling materials can later deposit on heat transfer surfaces. This can affect the performance of the complete externally fired gas turbine cycle.

2. Methodology

Experiments were carried out in an air-staged diffusion burner at atmospheric pressure. Fig. 2 shows a schematic of the experimental setup. A mixture of H₂, CO, CO₂, and CH₄ was stored in a gas bottle while N₂ was stored in a separated gas bottle. A mixing panel was used to completely mix the flows from these bottles. Water was injected to this mixture to complete the composition of simulated biomass gasification gas. In order to represent tar in the most simplified way, benzene was also injected. The injection point for both was heated to ensure their vaporization. A heating tape was used with the purpose to provide the necessary heat to keep both at a temperature of about 110°C. Benzene was added at two different injection rates. The line containing the final fuel mixture was constantly heated up using an electric heating tape and was insulated. The final fuel temperature was about 90°C.

144 The flow of combustion air varied in a range of lean equivalence ratios for each fuel with
145 different benzene injection rate. The air inlet temperature was about 20°C.

146 The burner was firstly warmed up using natural gas for 30 minutes. Initially, the fuel was ignited
147 using a spark rod. Then, the fuel was switched to the simulated biomass gasification gas
148 gradually. When the synthetic mixture was feeding the burner alone there was an interval of 10
149 minutes before starting with the measurements for the first equivalence ratio. After that, the
150 change was made only in the air mass flow rate while keeping the fuel mass flow rate constant.
151 The time interval between the changes of the air mass flow rates was about 5 minutes. Finally,
152 during the next five minutes the emissions data reported in this paper was collected. The
153 results presented in this work are the average of 5 minutes data. During this time, it was
154 controlled that the concentrations of CO, UHC, and NO_x did not vary greatly. The time for the
155 tests was adapted to the fuel consumption.

156 Three thermocouples were placed to measure the temperature closer the flame of the first
157 stage (T₁), the temperature closer to the flame of the second stage (T₂), and the temperature
158 furthest from the flame (T₃). Samples of the flue gases were taken continuously at an
159 approximate distance of 90 cm from the flame. This implied residence times varying from 4 to
160 14 seconds for the highest to the lowest mass flow rates respectively. No special coating was
161 applied to the thermocouples closer to the flames. This can cause an error in the reading of the
162 temperatures due to radiation effects. The flue gases pass through a gas treatment unit to
163 remove the water and cool down the samples before entering the gas analyzers. CO (ppm) was
164 measured using a nondispersive infrared equipment (NDIR). A chemiluminescent detector was

used to measure NO, NO₂, and NO_x (ppm). The unburned hydrocarbons were detected using a Flame Ionization Detector (FID). The results of the measurements are on a dry basis.

For each fuel flow, the overall equivalence ratio (ϕ) was varied in a range of lean conditions.

The equivalence ratio was calculated using the equation below [15]:

$$\Phi = \frac{(\dot{m}_{air}/\dot{m}_{fuel})_{stoichiometry}}{(\dot{m}_{air}/\dot{m}_{fuel})_{actual}} \quad (1)$$

where $\dot{m}_{air}$ and $\dot{m}_{fuel}$ correspond to the air and the fuel mixture mass flow rates respectively.

The air and fuel mass flow rates were measured and controlled using Bronkhorst mass flow controllers. Table 1 shows the air and fuel mass flow rates and the corresponding equivalence ratios.

Since all the experiments run at lean conditions, the effect of dilution was corrected at 3%

Oxygen using the following formula [15]:

$$Corrected\ PPM_{3\% O_2 level} = Measured\ PPM * \frac{20.9 - 3}{20.9 - (\%O_2\ measured)} \quad (2)$$

2.1 Description of the setup

As seen in Fig. 2, the main components of the experimental rig were a burner, gas mixing panel, thermocouples, syringe pumps, and gas analyzers. The measurement of the mass flow rate of the stream coming from the gas bottle containing a gas mixture of H₂, CO, CH₄, and CO₂ had an

uncertainty of ± 0.05 kg/h. The nitrogen mass flow rate had an uncertainty of ± 0.14 kg/h. The air mass flow rate had an expanded uncertainty varying from ± 0.35 to ± 0.56 kg/h. Benzene and water were injected using syringe pumps with a pumping rate accuracy of ± 0.048 ml/h. N type thermocouples were placed at different distances from the burner. The thermocouples had an expanded uncertainty of $\pm 3.3^{\circ}\text{C}$ with a coverage factor of 2.

All the gas analyzers were calibrated before each experiment with zero and span gases with a blend tolerance of ± 2 vol%. The mean standard deviation for CO was in the range of 0.2 – 0.8%. The mean standard deviation for NO_x and UHC were in the ranges of 0 – 0.9% and 0.3 – 8.3% respectively. The highest mean standard deviations for UHC correspond to the readings for an equivalence ratio of 0.68 for the fuel with highest injection rate of benzene. There was a variation in the readings ranging from 0 to 2 ppm with most of the readings being close to zero. The largest source of uncertainty for all the readings was due to the calibration gas.

2.2 Description of the burner

The burner used in the tests was a v-shaped two stage diffusion burner from Midco International Inc. [16]. A schematic of the burner together with the geometrical description is shown in Fig. 3. It was originally designed for natural gas, but in these experiments it was tested in off-design conditions for biomass gasification gas. The burner was 22 cm long (L), 21 cm high (H), and 15 cm wide (W). The original width of the burner was 30 cm, but it was reduced by half for the current experiments due to the low heat rate used in the tests. The heat rate of the burner was kept constant at 5.8 kW for all the tests. This is about 5% of the burner capacity and about 20% of loading for the externally fired micro gas turbine prototype system from

202 Compower AB. This low range was used in order to keep the lab exhaust system at low
203 temperatures.

204 The geometry of the burner separates the air in two stages. Stage combustion allows the
205 formation of two separated flames in the combustion zone. The air is distributed according to
206 the diameter of the orifices in each plate of the v-shaped burner. The first stage in the burner
207 contains five rows of small orifices varying from 2.3 to 3 mm distributed as shown in Fig. 3.
208 These orifices allow fewer amounts of air to enter. This region constitutes a fuel rich diffusion
209 zone due to the limited amount of air for oxidation of the fuel. The second stage of the burner
210 is composed of five rows with 6 columns. Each of them contains orifices of 8 mm permitting
211 more air to enter. This region is a fuel lean reaction zone where a larger amount of air is
212 provided to complete oxidation of the fuel. The geometry of the burner only allowed the
213 manipulation of the overall equivalence ratio.

214 The positions of the more important thermocouples in the combustor are shown in Fig. 4. The
215 thermocouple closer to the flame (T_1) was positioned at 14 cm after the fuel injection zone. A
216 second thermocouple (T_2) was placed closer to the second stage combustion zone at 12 cm
217 from the first thermocouple. Another thermocouple (T_3) was placed closer to the sampling
218 point. This thermocouple was named the thermocouple furthest from the flame. In order for
219 the sample line to be at an adequate temperature, this region of the burner was not covered
220 with insulating material.

221

222

223 2.3 Fuel

224 Simulated biomass gasification gas was used in the tests with the following components CO,
225 H₂, CH₄, CO₂, and N₂ as its main constituents. The composition of the simulated biomass
226 gasification gas has been chosen basically to represent typical product gas composition of air
227 gasification of wood through a downdraft gasification process. Depending on how much wood
228 has been dried and the type of gasification technique applied, the amount of water vapor and
229 the amount of tar present in the fuel mixture varies. The fuel composition used in this work has
230 been reported by Erlich and Fransson [17] as a typical product gas of air gasification of wood in
231 a downdraft gasifier. For these experiments, water and benzene were injected to complete the
232 simulated mixture. Benzene was used as tar model compound. Tar in biomass gasification gas
233 has been defined as all organic contaminants with a molecular weight larger than that of
234 benzene [18]. However, in these studies benzene was chosen in order to calibrate this system
235 and to allow further comparison with other components present in tar from biomass
236 gasification gas. Additionally, benzene allowed an easier control when injected in the fuel
237 especially when the fuel line required to be continuously heated. The total composition of the
238 simulated biomass gas is shown in Table 2. The adiabatic flame temperature and lower heating
239 value provided in this table are calculated based on the composition of the fuels and
240 correspond to stoichiometric combustion.

241 With the objective to represent two different tar concentrations of biomass gasification gas,
242 benzene was injected at two different flow rates. The injection rates were 100 and 1 g/Nm³ of
243 the fuel respectively. These concentrations are commonly found in biomass gasification gas

derived from an updraft and a downdraft gasifier according to Milne et al. [19]. The lower heating value of the mixtures is influenced by the amount of tars added as can be seen in Table 2.

3. Results and discussion

3.1 Effect of equivalence ratio

The concentrations of CO and unburned hydrocarbon (UHC) versus the equivalence ratio are shown in Figs. 5 and 6. As can be seen, a similar tendency of CO and UHC concentrations is observed. At leaner equivalence ratios, there is an adverse effect on CO and UHC emission levels. Although there are larger amounts of air available, the temperature in the combustor is reduced as can be seen in Fig 8. This reduction of temperature slows down the oxidation reactions of CO and UHC increasing their final concentrations. A minimum concentration is detected at an equivalence ratio of 0.68 for both CO and UHC emissions for the fuel containing the largest injection rate of benzene while equivalence ratios between 0.8 – 0.68 show a minimum concentration of CO and UHC for the fuel with lowest injection rate of benzene.

Regarding NO_x emissions, they can be formed through three main pathways. Thermal NO_x is formed from the reaction of nitrogen and oxygen from the air. It becomes significant at temperatures around 1800 K. Prompt NO_x is formed from the reactions of hydrocarbon radicals (CHs) with nitrogen. It often occurs in fuel rich conditions. Fuel NO_x is formed from reactions involving fuel bound nitrogen at relatively low temperatures.

Fig. 7 shows the effect of the equivalence ratio on NO_x emissions. Following the composition of the fuels in this study, NO_x is produced from the thermal and prompt NO_x mechanisms. Simulation studied from Giles et al. [14] have shown that prompt mechanism also contributes to NO in syngas (H_2 and CO) mixtures containing methane. The low NO_x concentrations as shown in Fig. 7 for the fuels studied are due to dilution and low temperatures caused by components such as H_2O , N_2 , and CO_2 in the fuel mixture composition. Studies have shown that these components are effective in reducing NO_x in flames containing H_2 and CO [14].

As can be seen in Fig. 7 for the fuel mixture containing the lowest injection rate of benzene, NO_x emissions behave almost constant as the equivalence ratio becomes leaner. This can be due to the non-premixed nature of the flame used in this study. In non-premixed flames there can be regions where the reactions take place at stoichiometric conditions depending on the diffusion of the reactants [20]. The dependence of thermal NO_x formation on the overall equivalence ratio in diffusion flames is less sensitive than in premixed flames.

For the fuel with larger injection rate of benzene, there is a slightly low NO_x emission at an equivalence ratio of 0.8 (Fig.7). This is because at this equivalence ratio there is slightly richer conditions in the first stage of the burner due to fewer amounts of air flow than at equivalence ratios between 0.68 – 0.32. This gives lower NO_x emissions for an equivalence ratio of 0.8. As the air mass flow rate increases, there is more oxygen available in the first stage causing an increase of NO_x . Although there is a slightly decreasing trend of NO_x for the range of equivalence ratios between 0.68-0.32, the results are relatively linear. The difference is within

the range of uncertainty level of the instrument. As in the previous case this is due to a lower dependence of thermal NO_x formation on the overall equivalence ratio in diffusion flames.

At equivalence ratios between 0.4-0.2 and 0.32-0.2 for low and high benzene injection rate respectively, NO_x emissions slightly increase. At these equivalence ratios, there is a higher air velocity in the first stage and the combustion in this region will be closer to stoichiometric or even become leaner. Major part of the oxidation takes place in the first stage. The principle of stage combustion (i.e. rich region followed by a lean region) is no longer taking effect. This causes a slight increase in NO_x emissions. This can also be seen in the temperature measurements. Figs. 9 and 10 show the temperature measurements closer to the first and second stage of the flame respectively. There is a slight increase in temperature as the equivalence ratio becomes leaner. This increase is more noticeable for the fuel with lower injection rate of benzene and for equivalence ratios of 0.32-0.2. The temperature closer to the second stage (T_2) (See Fig. 10) also shows an increase in the temperature at an equivalence ratio of 0.2 and 0.32-0.2 for the fuel with the highest and lowest injection rate of benzene respectively. Since most of the combustion is taking place in the first stage, the heat released is detected by the second thermocouple showing an increase in temperature. The flame is no longer stabilized in two regions. It is rather one flame in the first stage producing the increase in NO_x emissions.

3.2 Effect of injection of Benzene

Figs 5, 6, and 7 also show the effect of injection of benzene in the fuel gases. At very lean conditions, (overall equivalence ratios in the range of 0.56 to 0.20) lower CO concentrations are

detected for the fuel gas with higher injection rate of benzene. This is because as previously explained more heat is released by the fuel containing 100 g/Nm^3 of benzene during combustion. From Table 2 it can be seen that the adiabatic flame temperature of the fuel with larger injection rate of benzene is higher than the adiabatic flame temperature of the fuel with lower injection rate of benzene. The addition of benzene increases the heating value of the fuel and correspondingly the flame temperature. Moreover, the flame temperatures (Figs. 9 and 10) for the fuel with 100 g/Nm^3 of benzene are higher than the flame temperatures for the fuel with lower injection rate of benzene. These higher temperatures allow more CO to complete oxidation. A minimum is observed at an equivalence ratio of 0.68 for both cases. For an equivalence ratio of 0.8, the CO emissions for the largest benzene injection are higher than the CO emissions for 1 g/Nm^3 of benzene. This is because of the larger carbon content in the fuel mixture with 100 g/Nm^3 of benzene. This in combination with the rich conditions in the first stage of the burner generates more CO. Once CO enters the second stage, the oxidation is not completed because large amounts of CO are competing with each other for oxygen.

Fig. 6 shows the UHC concentrations with the different injection rates of benzene. Higher concentrations of UHC are shown for the fuel gas with the highest injection rate of benzene. This is expected since the injection of benzene implies a higher amount of hydrocarbons in the fuel gas. Fig. 6 also shows a minimum point where the UHC concentrations are very similar for both cases. This minimum point is registered at an equivalence ratio of 0.68.

As can be seen in Fig. 7, NO_x concentrations are higher in total for the fuel gas containing larger rate of injection of benzene for all the range of equivalence ratios. The fuel with higher benzene

injection rate releases more heat as can be seen in the temperature measurements in Figs. 9 and 10. The addition of benzene increases the temperature of the flame therefore the thermal NO_x contribution for the fuel with the largest benzene concentration is higher than for the fuel with the lowest injection rate of benzene.

From these observations, it can be seen that for the operation of the two-stage burner in this study the presence of polyaromatic hydrocarbons such as benzene, which is present in real tar in biomass derived gas, increase the formation of NO_x emissions compared with a fuel gas with low concentrations of polyaromatic hydrocarbons.

4. Conclusions

A commercial air staged natural gas diffusion burner was tested in off-design conditions for combustion of a synthetic mixture of biomass gasification gas. The operability of the burner with regards combustion emissions was shown for a fuel power of 5.8 kW. An optimum point for low CO and UHC concentrations was found. The effect of the overall equivalence ratio was reported as well as the effect of benzene concentrations in the fuel gas. The main conclusions can be summarized as follows:

- An equivalence ratio of 0.68 shows to be an optimum point for this burner with regards CO and UHC concentrations for combustion of a synthetic mixture of biomass derived gas containing 1 and 100 g/Nm³ of benzene in the fuel.
- A relatively linear constant trend is observed for NO_x emissions over the range of equivalence ratios studied for both injection rates of benzene. This is due to a lower sensitivity of thermal NO_x formation in a diffusion flame.

- At equivalence ratios lower than 0.68, the presence of polyaromatic hydrocarbons such as benzene reduces the concentrations of CO in the flue gases. Instead, the concentrations of UHC are increased with the increase of polyaromatic hydrocarbons such as benzene in the whole range of equivalence ratios. Benzene, which is present in real tar in biomass derived gas, increases the formation of NO_x emissions compared to a fuel gas with low content of polyaromatic hydrocarbons.

Tar in biomass gasification gas contains several other components that can affect the combustion process in different ways. Future tests using real biomass gasification gas are recommended. Understanding the contributions of thermal and fuel NO_x mechanisms in combustion of biomass gasification gas is also important. Additionally, the velocity of the fuel and air in the burner can change at higher power outputs, so further tests are suggested as future work. A CFD model of the combustor is also recommended in order to suggest improvements in the burner since the distribution of the fuel and the air can vary according to the geometry of the burner. These results are important to evaluate the performance of the burner studied to be used later in an externally fired gas turbine and understand the effect of polyaromatic hydrocarbons such as benzene in combustion of biomass derived gas. These studies are important when less strict quality fuels are to be used in more flexible systems such as the externally fired gas turbine.

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Table 1

Table 1 Fuel and air mass flow rates and corresponding equivalence ratios

Fuel with 1 g/Nm ³ of benzene			Fuel with 100 g/Nm ³ of benzene		
Fuel	Air	Equivalence	Fuel	Air	Equivalence
mass flow	mass flow	ratio	mass flow	mass flow	ratio
rate	rate		rate	rate	
kg/h	kg/h	Φ	kg/h	kg/h	Φ
4.63 ± 0.15	7.03 ± 0.38	0.80	2.90 ± 0.14	7.68 ± 0.35	0.80
	8.27 ± 0.39	0.68		9.04 ± 0.39	0.68
	10.04 ± 0.40	0.56		10.98 ± 0.40	0.56
	12.78 ± 0.41	0.44		13.97 ± 0.42	0.44
	17.57 ± 0.44	0.32		19.21 ± 0.46	0.32
	28.12 ± 0.54	0.20		30.69 ± 0.56	0.20

Table 2

Table 2 Fuel mixtures and their corresponding lower heating value.

Fuel type	Biomass gasification gas composition (%volume)							LHV	T _{ad}
	CO	H ₂	CH ₄	CO ₂	N ₂	H ₂ O	C ₆ H ₆	MJ/Kg	K
Fuel with [1g/Nm ³]	23.98	11.64	2.38	9.50	47.49	5.00	0.03	4.51	1928
Fuel with [100g/Nm ³]	23.35	11.33	2.31	9.25	46.24	4.87	2.65	7.21	2132

FIG1

Fig. 1 Schematic of an externally fired gas turbine.

FIG2

Fig. 2 Schematic of the experimental setup.

FIG3

Fig. 3 Schematic of the burner together with the main geometrical characteristics [16].

FIG4

Fig. 4 Position of more important thermocouples in the combustor.

FIG5

Fig. 5 Concentrations of CO with increasing the equivalence ratio

FIG6

Fig. 6 Concentrations of UHC with increasing the equivalence ratio

FIG7

Fig. 7 Concentrations of NOX with increasing the equivalence ratio

FIG8

Fig. 8 Temperature measurements of the thermocouple furthest from the flame

FIG9

Fig. 9 Temperature measurements of the thermocouple closer to the first stage

FIG10

Fig. 10 Temperature measurements of the thermocouple closer to the second stage