FUEL QUALITY IMPACT ANALYSIS FOR IMPLEMENTATION OF CORN COB GASIFICATION GAS IN CONVENTIONAL GAS TURBINE POWER PLANTS

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Abstract

Implementation of alternative fuels in gas turbine facilities is a challenging step towards cleaner and more responsible energy production. Despite numerous technical, economical and legal obstacles, possibilities for partial or complete substitution of fossil fuels are still subject of profound research. From all possible solutions, one with high acceptance is the symbiosis of existing gas turbine technologies and new ways of waste biomass energy utilization through firing or co – firing of biomass gasification gas. Therefore, the implementation of corn cob gasification gas with CO_2 recirculation in gas turbines is analyzed in this paper. The followed methodology approaches this solution through two different scenarios each with 5 different cases. In the first scenario fuel mass flows are kept constant regardless of the fuel quality change consequence of the corn cob gas share, while in the second scenario fuel volume flows are assumed constant. Impact of fuel composition changes on combustion product characteristics was analyzed using CHEMKIN PRO with GRI–Mech 3.0. Finally, fuel quality impacts on a gas turbine power plant performance are analyzed using a mathematical model that enables the simulation of a 3.9 MW

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experimentally correlated gas turbine. The results show that utilization of corn cob gasification gas is possible through co-firing with natural gas with acceptable values without modification of the fuel system or gas turbine.

Key words: gas turbine, gasification, bioenergy, simulation, biomass

1. Introduction

Biomass for energy production is a promising alternative source that is underused in many developing agricultural regions [1]. In most cases, it is obtained from industrial and/or agricultural waste, thus providing an energy source at acceptable costs. On the other hand, biomass usage for energy production is an effective way of reducing both fossil fuels consumption and energy dependence, especially in energy struggling countries.

Implementation of biomass gasification gas for large power generation, i.e. gas turbine facilities, can provide a useful end of biomass as a secondary energy source that can be implemented in different energy consumption sectors [1, 2]. In addition, biomass based electricity production has considerable impacts on the environment by allowing the reduction of atmospheric emissions of greenhouse gases (GHGs) when compared to conventional electricity production. According to all mentioned advantages, gasification of biomass and implementation of produced gas represents a promising solution for sustainable development while ensuring local environment protection improvements.

However, there are significant technical, economical and legal disadvantages of biomass gasification gas implementation in developing regions. Technological disadvantages are reflected in the great variety of calorific values of the produced gases, different to those calorific values of natural gas for which gas turbine power plants are designed. Biomass gasification gas, with low methane content, represents a low calorific gas that is not suitable for direct utilization in gas turbine plants. Thus, a promising solution is to introduce biomass gasification gas through co-firing with natural gas. In addition, this solution has positive effects on CO_2 reduction, better than facilities for CO_2 capture and storage [3].

Moreover, variation of fuel heating values by dilution of natural gas with CO_2 and/or N_2 shows a decrease of NO_x concentrations in the combustion products [4], setting a technology with low emissions and some energy recovery. Biomass and biomass – natural gas combined cycles have been investigated with various working fluids and various working cycles [5 - 7] showing different, highly potential possibilities of biomass application in gas turbine power plants.

Nevertheless, this solution also presents great challenges for researchers as well as for turbine designers. Implementation of a gas with different calorific value than natural gas in gas turbine equipment could cause operational irregularities. These irregularities could be eliminated by applying appropriate gas turbine modifications which may cause certain energy transformation effects [8]. Effects of fuel calorific value changes could also be decreased by using partial fuel substitution through co – firing of low calorific gas with natural gas [9 - 11]. However, the most common effect of fuel substitution is a decrease of gas turbine plant efficiency due to decreased amounts of methane in the low calorific gas compared to natural gas [12], therefore the turbine is operating in off – designed regimes. Herein, careful analysis is required before implementing the use of these gases with natural gas. Promising research shows that the use of biomass gas with 6 MJ/Nm^3 calorific value in mixtures of 35 – 50% (vol) with natural gas do not need significant modifications of the gas turbine system [13]. Therefore, progression and implementation of this concept can be pursued as long as good understanding of the potential thermodynamic and operational impacts of a new blend in a power cycle intended for the conversion are achieved. For such an aim, performance of gas turbine power plants when co – firing low calorific gas with natural gas can be analyzed through mathematical modeling [14] due to the high flexibility and wide stability range of gas turbines.

Regarding biomass feedstock, the alternatives are vast particularly in farming areas located in developing countries. Special interest can be found in the large quantities of waste biomass obtained from the production processes of products harvested for human consumption. Amongst them, corn is one of the largest agricultural products worldwide, thus the use of corn cob has a great potential for gasification processes.

Therefore, this work seeks to establish the parameters for the study and implementation of corn cob gasification gas as additive of methane for gas turbine power applications, providing valuable information for the use of the resource, thus ensuring higher environmental and economic benefits are achieved by future users of this concept.

2. Material and methods

2.1. Mathematical Model

A new mathematical model was developed to understand the impacts of different corn cob gasification gas blends. The model was initially calibrated and used in other works [15, 16]. Model development and validation are presented in detail [17].

An applied mathematical model that considers the processes of non-adiabatic expansion and cooling in the turbine as a whole is used in this work. The basic assumption of the method is the continual distribution of the cooling air along the gas turbine, with the computation of the expansion process of the combustion gases and cooling air separately. This method was selected as the 'reference method' [18].

Verification of the mathematical model was based on the comparison of model predictions against the manufacturer's data for a reference gas turbine [19]. For the model, it was necessary to make some adjustments for correspondence to the actual gas turbine plant [19]. The first adjustment of the base model is the introduction of water vapor impacts through the ratio of water vapor and fuel mass flow rates at the combustion chamber inlet, α, as well as the water vapor enthalpy difference, applied in equations (3), (4) and (5).

The enhanced version of the mathematical model considers the variation of the turbine polytropic efficiency due to operational regime changes. Variation of the polytropic efficiency due to operational regime changes is not analyzed sufficiently to date. Drawing upon preliminary analyses and comparison with reference data, it was concluded that the assumption of a constant polytropic efficiency regardless operating regime change needed to be replaced. Mass flow rates changes of combustion products through the gas turbine results in 'off–design' operation, and hence variation of the turbine polytropic efficiency. Therefore, the second adjustment of the base model was the introduction of the variable polytropic gas turbine efficiency, as a function of the combustion products mass flow rate, consistent with the gas turbine plant reference data, equation (7). Equation (7) shows acceptable accuracy for the reference gas turbine [19] using the following coefficient values $k_1 = -1.0372$, $k_2 = 32.179$ and $k_3 = -248.68$. It is recognized that application of this polytropic efficiency equation to different gas turbine types should be further analyzed.

From previous studies, opinion is divided concerning the introduction of fuel enthalpy in the combustion chamber energy balance. For instance, the referenced mathematical method [18] does not consider fuel enthalpy, while other authors [20] consider fuel enthalpy as an important part of the combustion chamber energy balance. Therefore, to better understand the influence of fuel enthalpy, it was introduced into the mathematical model for simulation of the flow behavior, heat transfer and energy transformation as a third adjustment, applied in equations (3), (4) and (5).

Specific work of compression is calculated by following equation:

$$L_C = c_{p_{air}}\Big|_1^2 \cdot T_1 \cdot \left(\Pi_C^{\frac{1}{\eta_{pC}} \cdot \frac{R_{air}}{c_{p_{air}}\big|_1^2}} - 1 \right) \tag{1}$$

where $c_{p_{air}}\Big|_1^2$ is the averaged value of specific heat of air through the compression [kJ/kgK], T_1 is the air temperature at the compressor inlet [K], Π_C is the compressor pressure ratio [-], η_{pC} is the polytropic efficiency of the compressor [-], R_{air} the universal gas constant for air [kJ/kgK].

Temperature of air at the compressor outlet:

$$T_{2t} = T_1 \cdot \Pi_C^{\frac{1}{\eta_{pC}} \frac{R_{air}}{c_{p_{air}}\big|_1^2}} \tag{2}$$

Specific heats of air and combustion products are calculated by [21].

Coefficient b is fuel mass flow relative to air mass flow at the combustion chamber inlet [-], and it is calculated by the following equation:

$$b = \frac{\dot{m}_{fuel}}{\dot{m}_2} = \frac{c_{p_{cp}}\Big|_1^3 \cdot (T_{3t}-T_1) - c_{p_{air}}\Big|_1^2 \cdot (T_{2t}-T_1)}{\eta_{CC}\cdot(LHV+h_{fuel}) - c_{p_{cp}}\Big|_1^3 \cdot (T_{3t}-T_1)\cdot(1+\alpha) - \alpha\cdot(h_{CC2}-h_{CC1})} \tag{3}$$

where $\dot{m}_{fuel}$ is a fuel mass flow [kg/s], $\dot{m}_2$ is the air mass flow at the compressor outlet [kg/s], $c_{p_{cp}}\Big|_1^3$ is the averaged value of combustion products specific heat [kJ/kgK], $\mathrm{T_{3t}}$ is the combustion products temperature at the turbine inlet [K], η_{CC} is the efficiency of a combustion chamber [-], LHV is the lower heating value [kJ/kg], h_{fuel} is the specific enthalpy of fuel at combustion chamber inlet [kJ/kg], h_{CC1} is the enthalpy of water vapor at the combustion chamber inlet [kJ/kg], h_{CC2} is the enthalpy of water vapor at the combustion chamber outlet [kJ/kg].

The specific work of the expansion in a gas turbine is calculated as:

$$L_T = c_{p_{cp-air}}\Big|_3^4 \cdot \frac{(1-z-r_{air})\cdot(1+b\cdot(1+\alpha))\cdot T_{3t} + r_{air}\cdot M \cdot T_{2t}}{(1-z-r_{air})\cdot(1+b\cdot(1+\alpha)) + r_{air}} \cdot \left(1 - \Pi_T^{\;\eta_{pT}\cdot\frac{R_{cp-air\,(3-4)}}{c_{p_{cp-air}}\big|_3^4}}\right) \tag{4}$$

where $c_{p_{cp-air}}\Big|_3^4$ is the averaged value of gas specific heat through the expansion [kJ/kgK], z is air mass flow for sealing relative to air mass flow at the compressor inlet [-], r_{air} is cooling air mass flow specified to compressor inlet mass flow [-], M is the cooling air distribution factor [-], Π_T is the turbine pressure ratio [-], η_{pT} is the polytropic efficiency of a turbine [-], $R_{cp-air\,(3-4)}$ is the universal gas constant through expansion [kJ/kgK].

Supplied heat equation is formed from combustion chamber energy balance:

$$q_{sup} = \frac{1}{\eta_{CC}} \cdot \Big[(1 - z - r_{air}).\big(1 + b \cdot (1 + \alpha)\big).c_{p_{cp}}\Big|_0^3 . (T_{3t} - T_0) - (1 - z - r_{air}).c_{p_{air}}\Big|_0^2 \cdot (T_{2t} - T_0) - \alpha \cdot b \cdot (1 - z - r_{air}) \cdot (h_{CC2} - h_{CC1})\Big] - b \cdot (1 - r_{air}) \cdot h_{fuel} \tag{5}$$

Finally, the efficiency of the entire gas turbine plant is defined as:

$$\eta_{GTP} = \frac{(L_T - L_C) \cdot \eta_m}{q_{sup}} \tag{6}$$

where η_m is the mechanical efficiency [-].

The variation of the polytropic gas turbine efficiency is calculated by the equation:

$$\eta_{pT} = k_1 \cdot \dot{m}_{cp}{}^2 + k_2 \cdot \dot{m}_{cp} - k_3 \tag{7}$$

In the case when low calorific gases are introduced in the gas turbine combustion chamber, which is designed for natural gas combustion, it is necessary to introduce higher amounts of fuel to provide the required temperature of the combustion products at the turbine inlet. A limiting criterion for the increase of the fuel mass flow is the gas turbine geometry. When the gas turbine mass flow rate reaches its maximum value, choking occurs. In this case, the turbine mass flow rate cannot be increased regardless of the decrease of the pressure behind the turbine stage, or at the turbine outlet. Thus, the maximum flow rate through a singular turbine stage is defined as a function of the gas state at the beginning of the expansion [12, 22-25]:

$$\dot{M}_{max} = A_0 \cdot \sqrt{\kappa \cdot \left(\frac{2}{\kappa + 1}\right)^{\frac{\kappa+1}{\kappa-1}} \cdot p_{in} \cdot \rho_{in}} \tag{9}$$

Where A_0represents the nozzle cross - sectional area in the stationary vane [m^2], κ is the isentropic exponent [-], p_{in} is pressure [Pa] and ρ_{in} the combustion products density at the turbine stage inlet [kg/m^3].

Finally, the fuel system propulsion is defined by two criteria, Wobbe index and fuel velocity. Gases with Wobbe Indices different to natural gas need modified configurations of the fuel system including necessary nozzle pressure drops. WI is defined by the eq. 10 [26], where ±5% is quoted as the possible variation in WI that can be handled by standard fuel gas control systems without adjustments; however, in reference [27], that range is ±10%.

$$WI = LHV/\sqrt{(SG)} \tag{10}$$

where SG is a specific gravity [-].

If the fuel mass flow is a known value, the the fuel velocity is calculated as:

$$c_{fuel} = \frac{\dot{m}_{fuel}}{\rho_{fuel} \cdot A_{fuel}} \tag{11}$$

where ρ_{fuel} is a density of analyzed fuel [kg/m^3] and A_{fuel} is a fuel pipe cross - sectional area [m^2].

2.2. Model calibration

Calculations undertaken using the mathematical model were based on analyses of the main parameters of a gas turbine plant running on 100% natural gas. The design operation regime was assessed with a 100% load, while the off-design operation regime was evaluated over a range of operating conditions (from 90% to 10% load). The model sensitivity to the following parameters was analysed: supplied heat, power, heat rate, gas turbine plant efficiency and temperature of the combustion products at the gas turbine outlet. The parameters of the package - CX501E KB5

Turbine with natural gas combustion were used for verification purposes. The engine employed as the prime mover in the Centrax Type CX501E-KB5 generator set is an aeroderivative engine based on a single-shaft design. An industrial Rolls-Royce Allison 501-KB5 gas turbine, used in the study, delivers more than 3,900 kWe with an exhaust temperature of 550°C. Measurements were undertaken with natural gas (i.e. LHV = 47,497 kJ/kg at 15°C). The water vapor mass flowrate and fuel mass flowrate ratio at the entrance of the combustion chamber was set at 0.4 kg/kg.

Results showed good correlation between the numerical cycle and the manufacturer results, Fig. 1, with acceptable relative error for the analyzed parameters: supplied heat 0.26%, generated power 0.27%, heat rate 0.22%, gas turbine outlet temperature 1.74% and gas turbine plant efficiency of 0.18%.

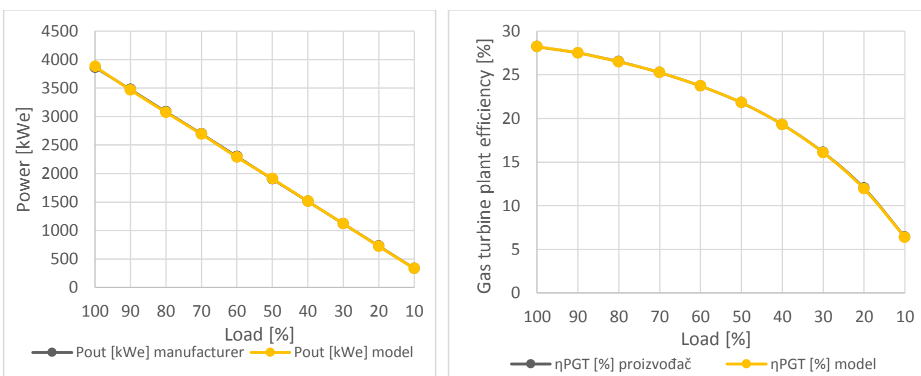

Fig 1. Correlation between the numerical model and an actual turbine using NG/air, [15]

This model has been also used successfully to simulate extravagant conditions using OXYFUEL combustion with methane as a fuel (OF) and Argon/CO2/Water vapor as working fluids (CARSOXY) [15, 16]. It must be emphasized that the rationale behind the development of this model was to evaluate the potential use of more complex blends, such as corn cob gasification gas combined with methane.

2.3. *Corn cob gasification gas combustion modelling*

In order to determine the combustion products from the use of corn cob gasification gas, initial analyses using CHEMKIN – PRO were conducted using the reaction chemical model GRI–Mech 3.0 [28 – 30]. The results were then used for the numerical model of a gas turbine running with corn cob gasification gas.

For all conditions, a hybrid Perfectly Stirred Reactor-Plug Flow Reactor (PSR-PFR) network was employed, which is commonly used to simulate mixing and flow characteristics in gas turbine combustor networks [31]. The reactor configuration consists of two clusters; the first cluster is a Perfectly Stirred Reactor (PSR), in which three distinct zones are applied; a mixing zone where fuel is partly premixed, a flame region directly connected to the former and the central recirculation zone (CRZ) where the products are recirculated [15,16]. Recirculation in the first cluster is set at 20% of the product gases, which is approximated from previous experimental campaigns using similar burners [32 - 34]. A Plug Flow Reactor (PFR) represents the second cluster used for post-flame operation along a 0.1m duct [35]. This hybrid PSR- PFR network was used to simulate the combustion of determined blends in a gas turbine combustor, Fig. 2.

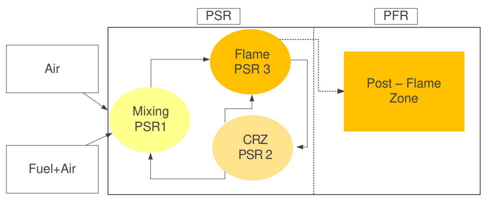

Fig 2. PSR-PFR Schematic

The combustion numerical simulation was performed for two scenarios; 1) the fuel mass flow was maintained constant for all cases; 2) fuel volume flow is maintained constant for all cases. Due to blend composition changes, the density of the blends also changes causing an increase of mass flow if the volume flow is maintained constant. Therefore, the second scenario requires higher amounts of fuel in the combustion chamber, thus causing changes in the combustion products composition and energy parameters.

2.4. Fuel Selection

Corn cob gasification gas obtained experimentally [1] was considered for this work. The analyzed gas has a low methane share (about 2%) and low calorific value; therefore it is defined as a low calorific gas, Table 1.

Table 1. *Gas composition obtained by simulation of downdraft corn cob gasification, [1]*

	N_2 [vol%]	**CO_2 [vol%]**	**CO [vol%]**	**H_2 [vol%]**	**CH_4* [vol%]**	**H_2O [vol%]**	**LHV [MJ/m^3]**	**LHV [MJ/kg]**
Dry gas	43.01	10.42	19.4	16.67	1.83	8.7	4.90	5.26
Wet gas	47.09	11.41	21.24	18.26	2.00	-	5.37	5.77

*fixed methane share in the gas

Analyses of the pure corn cob gasification gas characteristics showed that the Wobbe Index (WI) difference with methane is above 80%. According to Rowen [26], for WI differences above 50%, implementation of alternative gases in conventional gas turbine plants would be possible only through co-firing with natural gas. Thus, for co-firing analyses, five fuel blends with different corn cob gas ratios were defined: from Case 1 (pure natural gas) to Case 5 (with 40% of corn cob gasification gas), with increments of 10% between cases. Two complementary analyses were carried out, as initial CHEMKIN-PRO results would be fed into the gas turbine model to determine the thermodynamic parameters of the use of corn cob gasification gas in an industrially correlated gas turbine. Matrix of the corn cob combustion tests is given in table 2.

Tabel 2. *Parameters at the combustion chamber inlet*

Case	Scenario 1			Scenario 2			Pressure			$m_{air\ stoch.}$ [kg/s]
	m_{fuel} [kg/s]	m_{air} [kg/s]	ER [-]	m_{fuel} [kg/s]	m_{air} [kg/s]	ER [-]	p_{fuel} [bar]	p_{air} [bar]	$p_{combustion}$ [bar]	
1	0.29	14.47	0.89	0.29	14.47	0.89	11.22	9.69	9.69	16.2
2	0.29	14.47	0.78	0.30	14.47	0.81	11.22	9.69	9.69	14.2
3	0.29	14.47	0.68	0.31	14.47	0.73	11.22	9.69	9.69	12.3
4	0.29	14.47	0.58	0.32	14.47	0.65	11.22	9.69	9.69	10.6
5	0.29	14.47	0.49	0.33	14.47	0.57	11.22	9.69	9.69	8.9

2.5. Types of Manifold Injection

There are various types of manifold injection systems in these gas turbines. Each of them has different purposes for various flow rate applications. As the Wobbe Index of corn cob gasification gas is so different to natural gas, the rational of using a different manifold was evident. This is a point that will be evaluated through the following sections. The configuration of a multiple fuel system related to the calorific value of gas fuel is analyzed in [26], where the following configurations are analyzed: single manifold (Fig. 3), dual manifold (Fig. 4), and separate gas systems (Fig. 5). A single manifold fuel system is a standard fuel system in gas turbines designed for one type of fuel with a Wobbe Index difference ranging ~5% [26] when compared to natural gas, or 10% according to [27].

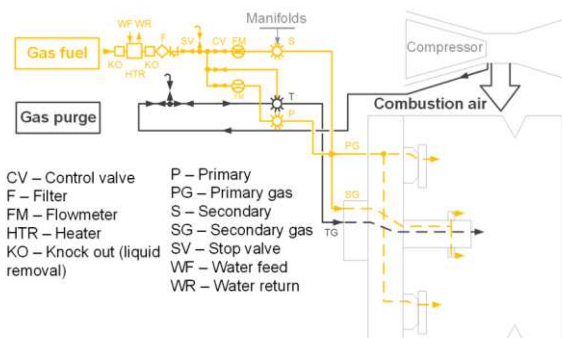

Fig 3. Single manifold fuel system for application of one type of fuel [36]

If the Wobbe Index difference is in a range from 5% (or 10%) to 25% it is necessary to apply dual manifold fuel systems [26]. In dual manifold fuel systems, each nozzle has two passageways for the gaseous fuels, Fig. 4 [36]. The first passageway leads to the primary nozzle area, through which gas will pass all the time. The second passageway leads to a secondary nozzle, which is separated from the primary nozzle area by a transfer valve, which can be opened in case of reaching the upper limit value of the pressure ratio in the primary nozzle area; allowing additional fuel flows to pass through the secondary nozzle. The primary nozzle orifice is usually on the face of the nozzle, while the secondary nozzle is set in the swirler [26].

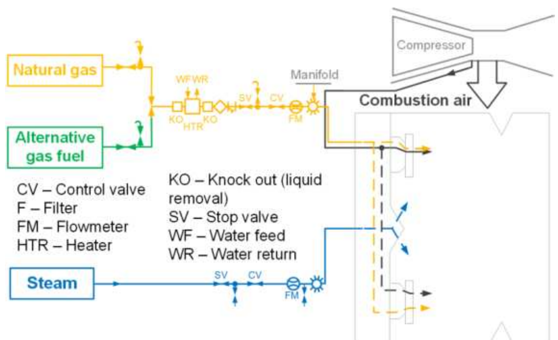

Fig 4. Dual manifold fuel system with steam injection, [36]

If the Wobbe Index difference is in a range from 25% to 50% it is possible to employ dual manifold fuel systems or separate gas systems [26]. To determinate which one, it is necessary to analyze other limitation criteria such as the fuel system propulsion. The maximum permissible value of fuel velocity is 20 m/s [37].

Finally, If the Wobbe Index difference is higher than 50% it is necessary to use separate gas systems, Fig. 5 [36]. According to Rowen [26], the need to separate the fuel systems arises from the variation of the Wobbe Indices which exceeds the turn down ratio capability of a single manifold fuel system. The transfer valve which connects two separate manifolds is necessary to provide combinations of fuel nozzle orifice areas with great Wobbe Index variances.

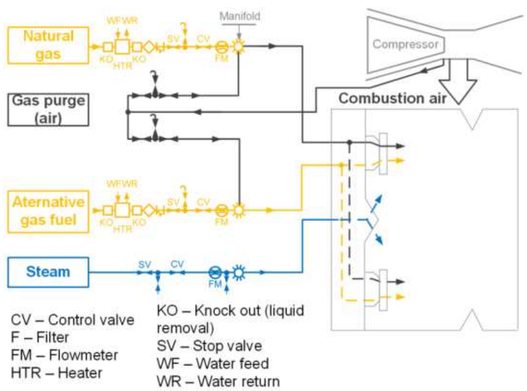

Fig 5. Separate gas fuel systems with steam injection, [36]

Thus, this work seeks to define the thermodynamic parameters of these blends and determine the hardware changes needed to run corn cob gasification gases at this great variety of conditions, thus giving users enough information to consider the use of these blends.

3. Results and Discussion

3.1. Reaction modelling

Increase of corn cob gas share in the fuel mixture decreases the adiabatic temperature ~7% for 10% of corn cob gas share in the first scenario, while in the second scenario the decrease is ~5,6%, Fig. 6.

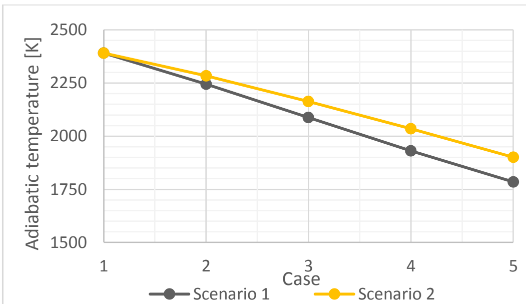

Fig 6. Variation of adiabatic combustion temperature as function of fuel blend composition for the considered scenarios with constant mass and volume fuel flows, respectively.

For the analyses of the combustion products composition, impacts on the specific heating values and the parameters of energy transformation in the gas turbine power plant, changes of the following gases in the combustion products were analyzed: oxygen, water vapor, carbon dioxide and nitrogen, Fig. 7.-10. This is in accordance with applied calculation of the combustion products specific heats [17].

The oxygen share in the combustion products has increased with increase of the corn cob gasification gas share due to the large amounts of oxygen in the gasification gas, Fig. 7. On the other hand, water vapor share has decreased with increase of the corn cob gasification gas share, Fig. 8, consequence of the decrease of hydrocarbons in the reactants.

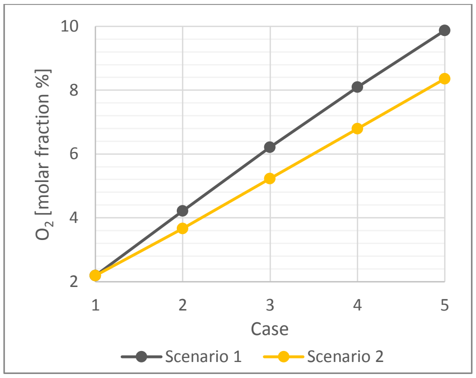

Fig 7. Variation of oxygen share in the combustion products as function of fuel blend composition for the considered scenarios with constant mass and volume fuel flows, respectively.

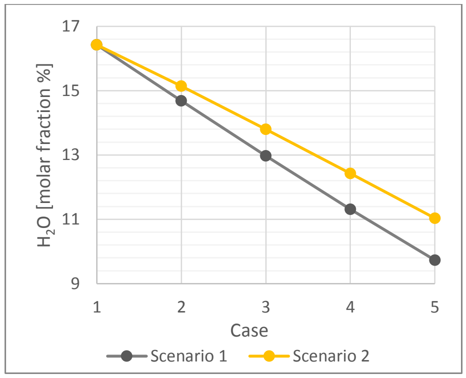

Fig 8. Variation of steam share in the combustion products as function of fuel blend composition for the considered scenarios with constant mass and volume fuel flows, respectively.

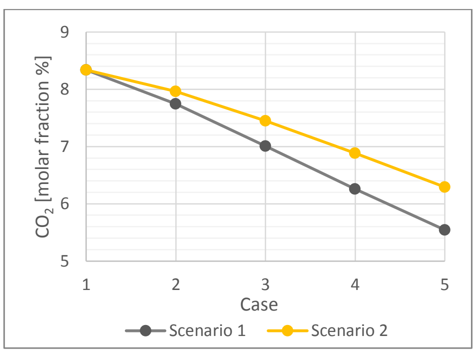

Fig 9. Variation of carbon dioxide share in the combustion products as function of fuel blend composition for the considered

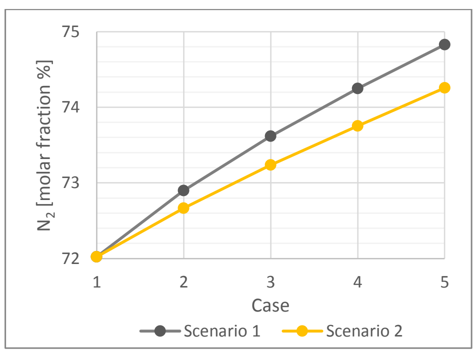

Fig 10. Variation of nitrogen share in the combustion products as function of fuel blend composition for the considered scenarios with

scenarios with constant mass and volume fuel flows, respectively.

constant mass and volume fuel flows, respectively.

Carbon dioxide in the combustion products has decreased with an increase of the corn cob gasification gas share, due to large amounts of oxygen and low levels of methane in the gasification gas, Fig. 9. The nitrogen share in the combustion products has also increased, as expected, with the increase of the corn cob gasification gas share due to the large amount of nitrogen in the gasification gas, Fig. 10.

3.2. Gas turbine simulation

To define possibilities for the use of corn cob gasification gas in gas turbines, the two previously discussed scenarios with 5 different cases were analyzed. In both scenarios fuel flows are defined according to the first analyzed case where pure natural gas is combusted.

Gas turbine plant parameters when co – firing of corn cob gas and natural gas is applied are calculated using the previously discussed mathematical simulation model [17]. The calculated indicators of energy transformation in the gas turbine plant are: supplied heat, power, gas turbine plant efficiency and turbine outlet temperature.

In both analyzed scenarios the supplied heat decreases with an increase of corn cob gasification gas in the fuel blend due to a decrease of low heating value of the fuel mixture, Fig. 11. Decrease of supplied heat is lower in the second scenario, when constant volume flows of fuel and air are applied, especially for cases with higher corn cob gas share in the fuel blend. For example, in case 5, supplied heat values in scenario 2 are about 13% higher than in scenario 1.

Power production is also decreased by increasing corn cob gasification gas share, effect caused by the decrease of the low heating value of the applied fuel, Fig. 12. Decrease of power is lower in the second scenario, especially for cases with higher corn cob gas share in the fuel blend. For example, in case 4, the produced power in scenario 2 is 15,4% higher than in scenario 1, while for case 5 the difference between both scenarios is ~22%.

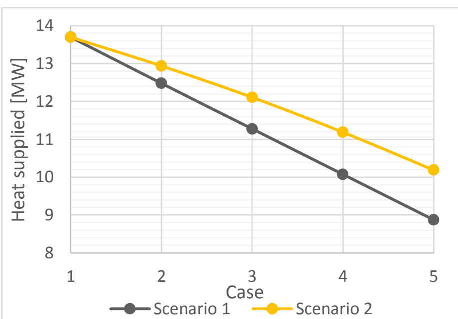

Fig 11. Variation of supplied heat as function of fuel blend composition for the considered scenarios with constant mass and volume fuel flows, respectively.

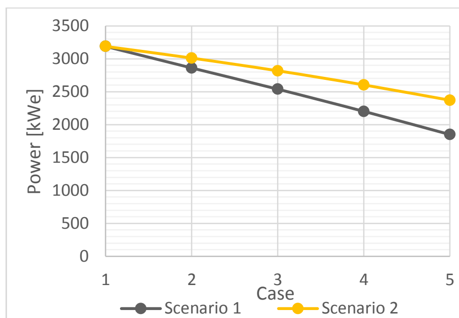

Fig 12. Variation of produced power as function of fuel blend composition for the considered scenarios with constant mass and volume fuel flows, respectively.

The difference in gas turbine plant efficiency values for analyzed scenarios is significant. As shown in Fig. 13, if mass flows of fuel and air are kept constant, efficiency is significantly decreased due to an increase of corn cob gas in the fuel blend. The efficiency decrease is mainly caused by the decrease of fuel energy content and the change of combustion products density, while in the second scenario, when fuel and air volume flows are kept constant the efficiency of the gas turbine plant is almost constant. Higher values of plant efficiency in scenario 2 are caused by higher amounts of fed fuel, compared to scenario 1.

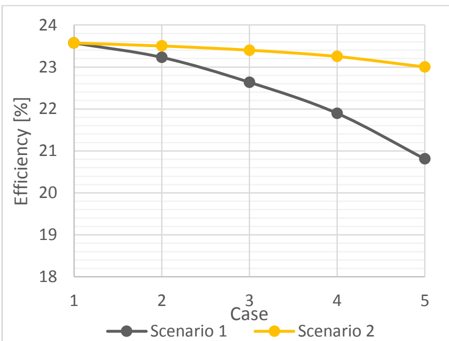

Fig 13. Variation of gas turbine plant efficiency as function of fuel blend composition for the considered scenarios with constant mass and volume fuel flows, respectively.

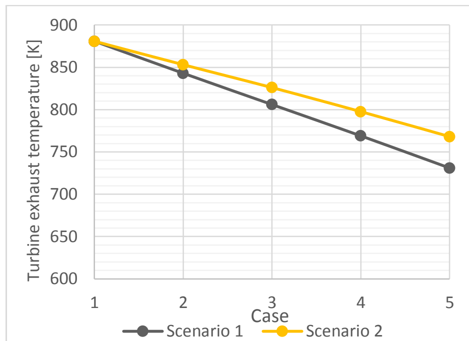

Fig 14. Variation of produced power as function of fuel blend composition for the considered scenarios with constant mass and volume fuel flows, respectively.

The decrease of combustion products temperature at the turbine exhaust with an increase of corn cob gas in the fuel blend is caused by lower heating value of the applied fuel and decreased turbine inlet temperature. Temperature decrease due to changes of fuel composition is lower for the second scenario, as shown in Fig. 14. When volumetric flows are kept constant, the exhaust turbine temperature is 5.2% higher for case 5 in the second scenario compared to the first scenario.

Results of the gas turbine show that actual flows of combustion products at the analysed turbine stage inlet are lower than the maximum possible flow rate for each analysed case, calculations obtained using equation (9), Fig. 15.

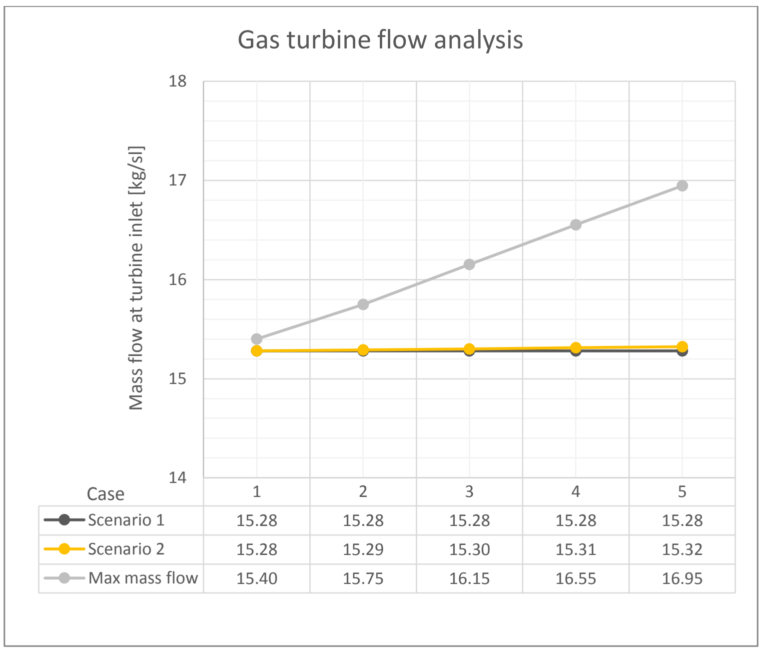

Fig 15. Results of gas turbine flow analyses for the considered scenarios with constant mass and volume fuel flows, respectively, compared to the maximum mass flows through the gas turbine

Wobbe Index difference analyses, Fig. 16, show that a standard single manifold fuel system can be applied for cases 1 and 2 without modifications, since WI difference values are lower than 10% [27]. For cases 3, 4 and 5 WI the difference is higher than 10% and lower than 50%; therefore, it is necessary to modify the fuel system to dual manifold system [26]. Results of the fuel system propulsion analysis show that fuel velocity for all cases is lower than the maximum permissible value of 20 m/s [37], Fig. 14; therefore, application of separate fuel systems is not necessary [25].

Therefore, fuel system modifications are not necessary using this fuel at the analyzed conditions for cases 1 and 2, in both considered scenarios. On the other hand, for cases 3, 4 and 5 it is recommended to upgrade the fuel system to dual manifold system regarding its pressure at the fuel nozzles [26]. Fuel flows used in the calculations for scenario 1 and 2 for all 5 analyzed cases could be applied via single manifold and dual manifold system according to Rowen [26], considering that a scenario 2 with volume flow is kept constant, thus increasing the thermodynamic parameters of the gas turbine system. According to the results of the gas turbine propulsion analysis, modifications are not necessary.

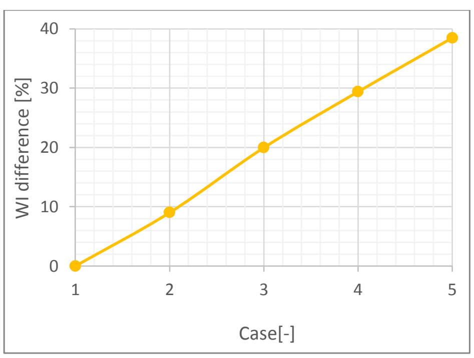

Fig 16. Variation of WI difference for the considered cases

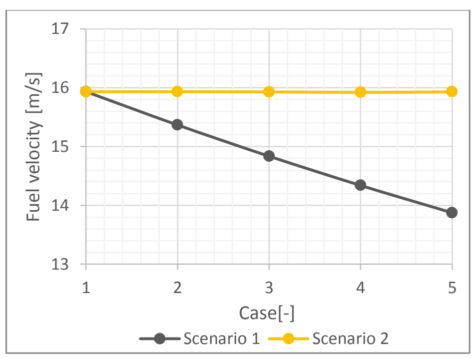

Fig 17. Variation of fuel velocity as a function of fuel blend composition for the considered scenarios with constant mass and volume fuel flows, respectively.

3.3. NO_x emissions

The emission analysis has been performed to determine the affect of fuel change to NOx values. Both considered scenarios, constant mass and volume fuel flow, have been analysed for all five cases, from pure natural gas to 40% corn cob gasification gas in the fuel blend. For the analysis NO and NO_2 were considered. Chemkin PRO results of the combustion products, Fig. 18. and 19. respectively, show decrease of both NO and NO_2 values with increase of corn cob gas in the fuel blend. For every increase of 10% of corn cob gas in the fuel blend, NO values decrease for about 75% in scenario 1, and about 66% in scenario 2. NO_2 values decrease for about 60% in scenario 1, and about 52% in the scenario 2, for every 10% increase of corn cob gas share in the fuel blend.

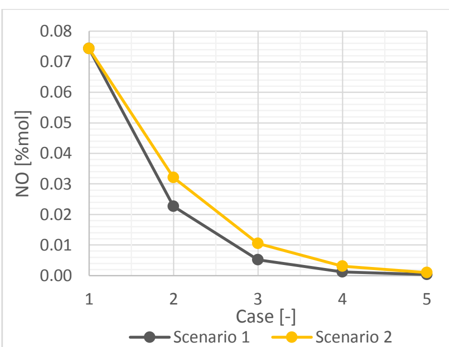

Fig 18. Variation of NO as a function of fuel blend composition for the considered scenarios with constant mass and volume fuel flows, respectively.

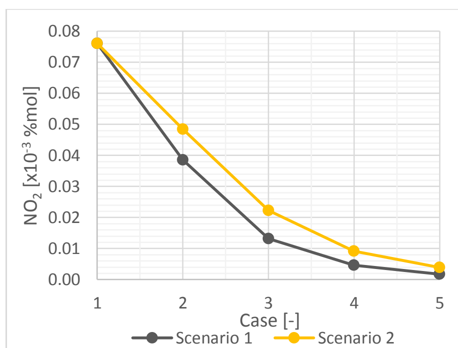

Fig 19. Variation of fuel velocity as a function of fuel blend composition for the considered scenarios with constant mass and volume fuel flows, respectively.

4. Conclusions

This study focuses on the implementation of biomass gasification gas into conventional gas turbine power plants, with the purpose of reducing fossil fuels consumption and increase of alternative fuels share for energy balance. A fuel quality impact analysis has been done for corn cob gasification gas. The analyzed gas is defined as low calorific gas, without potential of being implemented in gas turbine plant by itself. Implementation requires co-firing with natural gas. Numerical simulations were performed for two scenarios, for fuel blends with different corn cob gas ratios from pure natural gas to fuel blend with 40% of corn cob gasification gas, with increments of 10% between cases, with constant fuel mass flow and constant fuel volume flow, respectively. Results of reaction modelling show decrease of the adiabatic temperature ~7% for 10% of corn cob gas share in the first scenario, while in the second scenario the decrease is ~5,6%. The oxygen and nitrogen share in the combustion products has increased, while water vapor and carbon dioxide has decreased with an increase of the corn cob gasification gas share. Gas turbine simulation results show significant decrease in the analyzed parameters with the increase of corn cob gas share in the fuel mixture. On the other hand, parameter values increase in the second scenario with an increase of the amount of fuel introduced into the gas chamber. Thus, this scenario is the one recommended for implementation of this gas. Moreover, fuel system modifications are not necessary using this fuel at the analyzed conditions for cases 1 and 2, in both considered scenarios, with maximal corn cob gasification share of 10%. On the other hand, for cases 3, 4 and 5,with corn cob gasification gas share 20%, 30% and 40% respectively, it is recommended to upgrade the fuel system to dual manifold system regarding Wobbe Index differences, which would allow co-firing up to 40% of corn cob gasification gas in the fuel blend.

Results of gas turbine propulsion analyses show that actual flows of combustion products at the entrance of the analysed turbine stage are lower than the maximum permissible flow rate for each analysed case, therefore modification of gas turbine are not necessary. Further analysis should be performed in direction of increasing the supplied heat by increasing the fuel mass flow in order to achieve maximal efficiency with fuel blends with lower heating values (with 30% and 40% corn cob gas share).

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