Effects of primary air temperature on emissions of a gas turbine fired by liquefied spruce wood

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Abstract

Innovative lignocellulosic biofuel, obtained through solvolysis of spruce wood in multifunctional alcohols was tested in an experimental turbine engine with different primary air temperatures. Variation of primary air temperature was used to emulate two types of micro gas turbine generators – fully recuperated and simple cycle setups resulting in different temperatures of combustion chamber intake air. Results indicate that different temperatures, velocities, and flow conditions in primary zone of combustion chamber strongly influence droplet penetration depth and rate of mixture formation as well as emissions formation. For the innovative lignocellulosic biofuel emission trends of CO and THC were found to be significantly higher in simple cycle mode, whereas no influence of operation mode was observed for baseline diesel fuel. NOx emissions of innovative biofuel generally increased in regenerative cycle mode and also became sensitive on turbine inlet temperature, with similar trends being observed with diesel fuel. In the case of innovative fuel, deposits of char and polymerized fuel were observed on the combustor walls after operation in simple cycle mode, whereas in regenerative cycle mode, only small amounts of ash deposits were found on hot path surfaces, indicating beneficial influence of high primary air temperatures on combustion efficiency of innovative fuel.

Abbreviations: \( w \) – mass fraction, \( v \) - volume fraction, \( M \) - molar mass, \( EQR \) – equivalence ratio, \( sth \) - stoichiometric ratio, \( CO \) – carbon monoxide, \( THC \) – total hydrocarbons, \( NO_x \) – nitrogen oxides, \( TIT \) - turbine inlet temperature, \( FID \) – flame ionization detector, \( CLD \) – chemiluminescence detector, \( NDIR \) – non-dispersive infrared, \( LW \) – liquefied wood, \( D2 \) – no.2 diesel oil, \( REG \) – regenerative cycle, \( SPL \) – simple cycle, \( PM \) – particulate matter, \( pTSA \) – para-toluensulfonic acid.
1. Introduction

In the search of suitable alternative energy solutions making possible more dispersed and stable future energy production, biofuels are considered as an important long-term pillar of sustainable exploitation of primary resources. 1st generation biofuels are already available on the market for several years now, but they revealed several weaknesses such as competition with food supply and high initial feedstock price [1]. With this in mind, development of 2nd generation biofuels is still under way and great effort is being put into advancement of fuels from lignocellulosic material. These materials do not interfere with food feedstock and are widely available through wood and leftovers from wood industry. Several procedures were developed to make lignocelluloses suitable for use in current power generating technology and the most penetrating solutions are presented below to provide that basis for analyses of the investigated fuel.

2nd generation biofuels, namely ethanol, could be obtained through hydrolysis followed by fermentation of lignocellulosic material [1]. Considering that ethanol can only be produced from cellulose fibers (cellulosic ethanol), processes which are able to convert also lignin fraction of the input material are more attractive – these are mainly gasification processes, pyrolysis and different types of liquefaction. One of the options is solvolysis of lignocellulosic materials which requires organic solvent and moderate temperatures (120 – 250°C) at atmospheric pressure. Solvents used can be as phenol, acetone and polyhydric alcohols or their mixtures. To optimize the process, several techniques were already investigated to shorten the duration of liquefaction reactions. They include assistance of microwaves [2] or ultrasound [3] and addition of different catalysts i.e. para-toluensulfonic acid (pTSA) [4] and sulfuric acid [5]. With correctly tuned reaction parameters, liquefaction times could already be as low as 15 minutes [3].

Fuel, investigated in this article, originates from ultrasonically assisted liquefaction of Norwegian spruce (Picea abies) in polyhydric alcohols in the presence of acidic catalyst. In literature, the product is usually named “liquefied wood” (LW) [6], [7]. This process was chosen based on the analysis, which considered feedstock availability, reliability as well as costs of the production equipment, which is very important for end-users like industry and local communities. Conversion of lignocellulosic material is very efficient, as 990 g kg^-1 of initial material is converted into liquid products when exposed to temperature of 160°C and atmospheric pressure. High conversion efficiency can also reduce aftertreatment costs, especially on filtration procedure due to low solid residue that requires removal. Polyhydroxy alcohols used in the liquefaction process, were glycerol and di-ethylene glycol. To lower the costs of input material, properly treated residual glycerol from biodiesel production could also be used. Partial substitution of glycerol with di-ethylene glycol can be performed to reduce the viscosity of the product [8].

In contrast to other thermochemically obtained fuels, i.e. pyrolysis oil or bio-crude (product of hydrothermal liquefaction), the energy demand for this process is fairly low and amounts to 400 J kg^-1 [3] with ultrasonic assistance and 7.29 MJ kg^-1 without ultrasonic assistance [3]. With ultrasonic assistance liquefaction process is completed in 15 min, whereas without ultrasonic assistance this time increases up to 90 min. Lower calorific value of the product is 20.2 MJ kg^-1 and thus only 20 KJ MJ^-1 of energy value is consumed for fuel production when ultrasonic assistance is applied. For comparison, production of bio-oil, which has lower calorific value, uses 72 kJ MJ^-1 of fuel [9], which equals 1.3 MJ kg^-1 for bio-oil with calorific value of 18 MJ kg^-1.

Proposal of wood, liquefied in ethylene glycol, upgraded with hydrogenation, for use as a fuel was first time assessed in [10] and [11]. Further studies on hydrogenation for fuel purposes were conducted in [12] with the basic fuel being similar to the one used in the first experimental evaluation of non-upgraded liquefied wood in experimental recuperated micro gas turbine (MG T) [13]. This study proved the first evidence on stable combustion of non-upgraded LW. It was also shown that exhaust emissions are strongly dependent on turbine inlet temperature (TIT) and fuel preheat temperature [13]. CO concentration was ranging from 159 – 341 μL L^-1 and NOx from 50 – 76 μL L^-1, whereas emissions trends showed different TIT trend compared to those when utilizing diesel fuel (D2) as a reference fuel.
Furthermore, two different formulations of LW were tested where it was shown that strong influence of wood content in LW on exhaust emissions is present [14].

Current technology of heat engines on small and medium scale is primarily focused on internal combustion reciprocating engines and gas turbines. Application of LW in piston engines was not considered in this study due to unfavorable physical and chemical characteristics of the LW, i.e., high viscosity, low volatility, and high autoignition temperature (Section 2.2 and Ref. [13]). Although fuels with characteristics similar to LW can be utilized in gas turbines featuring external combustion (EFGT), the objectives of this research is aimed to utilization of the LW in directly fired gas turbines with internal combustion. This objective is mainly reasoned by the following facts:

1.) utilization of LW in directly fired gas turbines enables very large scalability of the power units as applicable gas turbines cover the range from few tenths kilowatts upward,

2.) directly fired gas turbines with internal combustion enable attaining high efficiencies, whereas they can also be coupled with an additional bottoming cycle if production of electric energy is to be maximized,

3.) lower specific investment cost in comparison to steam turbine plants.

However, even if EFGT systems utilize high temperature heat exchanger to transfer the heat from flue gas to working medium, their efficiency can still be higher than directly fired systems. This happens in the case of gaseous fuels, provided at atmospheric pressures that require compression before pressurized combustion in closed cycle of EFGT [15]. To obtain sufficiently high temperatures in working fluid to further elevate the efficiency, high temperatures are required in the furnace, which poses a problem with high temperature heat exchanger materials and heat losses to environment [16].

Due to the fact that exhaust emission in directly fired gas turbines greatly depend on TIT and additionally on the combustion chamber air intake temperature, the cross relation of these two parameters needs to be evaluated to further evaluate the potentials of LW as a fuel in commercially available gas turbines. This research was thus motivated by large diversity of MGT setups available on the market which greatly differ in recuperation intensity due to differences in their design and thus in their applicability. Generally, two different types of MGTs are available. One type is designed to maximize shaft power production and usually uses regenerative cycle with exhaust heat recovery to optimize the efficiency. Recovered heat is recycled to the working fluid thereby increasing its temperature, which can reach as high as 320°C with increasing trend for upcoming designs [17]. On the other hand, MGTs which are designed to supply larger quantities of heat to support e.g., industrial process or district heating network usually employ lower primary air temperature. These turbines operate in simple cycle mode as residual heat is effectively streamlined to water or steam. Primary air temperature of these designs equals compressor discharge temperature and it is therefore dependent on its pressure ratio. Results presented in this paper thus also include the first evaluation of the combustion performance of non-upgraded LW in non-recuperated, low pressure ratio MGT.

To determine most suitable turbine configuration, which enables low emission operation while utilizing LW, it is therefore necessary to investigate the influence of air temperature in the primary zone of the combustion chamber. This is of particular importance due to relatively unfavorable properties of the fuel that can be identified through liquefaction process characteristics and through measured values presented below in Table 2, which significantly influence mixture preparation and combustion through reduced atomization quality, aggravated evaporation and dissociation of fuel molecules.

In presented work, different degrees of recuperation intensity and therefore different primary air temperatures will be analyzed against the following criteria:

- exhaust emissions
- thermodynamic parameters,
For all operating points, comparative measurements were performed with diesel fuel (D2) to provide benchmark results. These results serve as the basis for analyzing the underlying phenomena governed by the combination of the operating conditions and fuel properties.

2. Materials and methods

2.1 Turbine engine

Experimental gas turbine was developed with the objective to offer a generic platform for investigations of different alternative fuels (Fig. 1). An automotive-type turbocharger is used for air supply and enthalpy extraction from hot gases. Combustion process takes place in one segment of can-annular combustor, originating from aircraft jet-engine. This setup, on one hand resembles professional gas turbines, whereas on the other hand it offers the possibility to easily modify the system, which is essential when different types of fuels are investigated. To avoid unnecessary construction and maintenance complications, power withdrawal is simulated through the throttle valve, which is located downstream of the main turbine rotor. Such fictive load is manifested through increased backpressure of the turbine, which reduces the enthalpy drop across the turbine and thus the power available for compressor. Throttle valve is therefore a very effective device for controlling TIT via two strategies: a) increasing fictive load by closing the throttle valve and thus reducing air mass flow or b) increasing pressure drop over throttle valve in fixed position by simultaneously increasing fuel and air flow. This setup allowed for an effective and simple load simulation which closely resembled conditions encountered in a commercial MGT in terms of thermodynamic parameters.

The measurement of TIT as a main control parameter is performed via two K-type shielded thermocouples. To avoid excessive radiative heat transfer from the flame, excessive heat transfer from thermocouple shields to atmosphere, and possibility the influence of poorly mixed dilution and combustion air, the following measures were taken:

- The duct from combustion chamber to turbine entry was elongated to 500 mm to provide sufficient length for flow development in order to avoid anomalies in temperature distribution over the flow cross section.
- The same elongation provided significant reduction in radiative heat transfer from the flame to thermocouples (with the flame being roughly 610 mm from the thermocouple shields).
- The thermocouples were slightly overlapping to reduce the conduction of heat to the outside walls of the duct.
- Both measured temperatures were then averaged to provide a representative mean of the TIT.

The layout of TIT measuring thermocouples is presented in Fig. 1.
To control primary air temperature, compressor discharge air is drawn through a heat exchanger (Fig. 2). Regenerative heating with exhaust gases can be either active or it can be bypassed by diverting exhaust gases away from heat exchanger, thus allowing to simulate simple or regenerative Brayton cycle. It is possible to change the rate of regenerative heating during experiments without the need to turn-off the system.

Experiments were performed using Swirl-air nozzle with internal mixing, using pressurized air at 700 kPa to atomize the relatively low-pressure (200 kPa) fuel stream. Indications on atomization quality and possible anomalies in spray formation or flame position are obtained through optical access on three distinctive positions: primary zone area being in line with the fuel nozzle, dilution zone of the combustor and on turbine entry. Location of other sensory equipment for supervision of thermodynamic data is shown in Fig. 2. Emissions were measured using a constant flow device, with FID, CLD and NDIR analyzer, while soot emission was measured with opacity meter. Measuring ranges of separate gas analyzers are presented in Table 1. Turbines usually operate with high excess air ratios which makes detection of some components in exhaust gases difficult due to their low concentration. This was the main issue when measuring smoke emissions with D2 operation via the opacity meter. However, due to unknown combustion behavior at low primary air temperatures, benchmark measurements were made to detect possibly excessive soot formation which was expected in the case of LW combustion in simple cycle mode.
2.2 Fuel preparation and conditioning

Precise description of fuel production and liquefaction procedure is explained in Refs. [3], [6], [13], therefore only brief overview is given here. Dried particles from debarked Norwegian spruce (Picea abies), harvested near Postojna, Slovenia (45°41' N, 14°12' E), representing a 244 g kg⁻¹ of initial reactant mixture, 367 g kg⁻¹ of glycerol and 367 g kg⁻¹ of di-ethylene glycol and 22 g kg⁻¹ of para-toluensulfonic acid were used as input material. Spruce dust was added to acidified glycols, preheated to 160°C. Liquefaction reactions were additionally accelerated by the use of 2 kW ultrasound probe.

Time of liquefaction has a significant influence on containment of the solid residue. If liquefaction time is shortened, wood is not completely decomposed and solid residue originates directly from wood particles, however when liquefaction time is prolonged, residue is reformed from decomposition products, mainly from reactions between degraded lignin and cellulose molecules [18]. Previous research work in [13], [6] and [3], revealed the optimum time with minimum yield of solid residue to be around 15 minutes. After liquefaction, the fuel is filtered with 100 μm mesh and cooled to the room temperature. Production and utilization procedure of liquefied product is presented in Fig. 3.

Fig 3. Fuel production process and utilization

Elemental composition and heating value of the obtained product, which are given in Table 2, were measured with Preggl-Dumas method. High oxygen content originates from the elemental composition of reactants:

- Picea abies wood contains almost 500 g kg⁻¹ oxygen, although this value varies accordingly to the content of lignin, cellulose and extractives,
- glycerol contains 520 g kg⁻¹ oxygen,
- diethylene glycol contains 450 kg kg⁻¹ oxygen.
Due to high oxygen content calorific value is almost halved in comparison to D2. Considering the composition of combustible mixture, formed from LW, its H/C ratio is close to that of D2 and energy density of the stoichiometric mixture is also almost the same due to its lower stoichiometric ratio (6.8:1).

Table 2: Properties of the Liquefied wood [13].

<table>
<thead>
<tr>
<th>Product</th>
<th>Liquefied wood (1.5:1.5:1 – glycerol:di-ethylene glycol:wood)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density</td>
<td>1.3 kg L⁻¹</td>
</tr>
<tr>
<td>C</td>
<td>0.4760 kg kg⁻¹</td>
</tr>
<tr>
<td>H</td>
<td>0.0798 kg kg⁻¹</td>
</tr>
<tr>
<td>N</td>
<td>0.0019 kg kg⁻¹</td>
</tr>
<tr>
<td>S</td>
<td>0.0089 kg kg⁻¹</td>
</tr>
<tr>
<td>O</td>
<td>0.4334 kg kg⁻¹</td>
</tr>
<tr>
<td>Stoichiometric ratio</td>
<td>6.8</td>
</tr>
<tr>
<td>HHV</td>
<td>21.9 MJ kg⁻¹</td>
</tr>
<tr>
<td>LHV</td>
<td>20.2 MJ kg⁻¹</td>
</tr>
<tr>
<td>pH</td>
<td>3.5</td>
</tr>
<tr>
<td>Water content</td>
<td>&lt; 0.02 kg kg⁻¹</td>
</tr>
</tbody>
</table>

It is also important to expose relatively high viscosity of the LW because it directly influences atomization properties and is, besides surface tension, frequently the limiting factor in utilization of alternative fuels. Temperature dependence of kinematic viscosity is presented in Fig. 4.

![Kinematic viscosity of liquefied spruce wood](image)

**Fig. 4**: Kinematic viscosity of liquefied spruce wood.

High viscosity of the LW at low temperatures is a major difficulty that has to be overcome before acceptable mixture formation and thus high combustion efficiency can be achieved. Fig. 4 and previous researches [13] indicate that temperature of the fuel at the fuel injector should be at least 80°C to ensure sufficient atomization. Preheating system was therefore used to heat the LW to 90°C. At this temperature, the viscosity of LW is 100 mm² s⁻¹ which is nearly 20 times higher compared to viscosity of D2 at 20°C.

Design of the fuel system was previously presented in [13], therefore only the most important characteristics are stated here:
brass or 316 stainless steel should be used for piping and process equipment,
surface temperature of heating elements should be lower than 160°C,
residence time of the fuel on high temperatures should be as short as possible,
areas with stagnating flow should be avoided.

2.3 Experimental procedure

During experiments, gas turbine was started using compressed air with bypassed heat exchanger to reduce the pressure drop along the air path. After pilot flame was lit, D2 was introduced through the main nozzle promoting self-sustained operation. Switch of fueling from D2 to LW was done manually when the temperatures throughout the system stabilized to their expected values. Experiments were performed at three different TITs: 750°C, 800°C and 850°C for recuperated and 700°C, 800°C and 900°C for simple cycle mode. The TIT interval was narrower for the recuperated system mainly due to two reasons. On the lower end of the interval, fuel demand for sustaining desired TIT was very low, resulting in too lean combustion and unstable operation close to lean blow off limit. On the upper part of the interval, combination of high TIT and high primary air temperature resulted in a very high thermal loading of the experimental system also on the cold side of the air path, so maximum TIT was reduced in accordance with research equipment restrictions. Second series of tests were performed with unrecuperated turbine and thus the compressor discharge temperature was equal to the combustion chamber primary air temperature. Other parameters varied according to TIT, which was set to 700°C, 800°C and 900°C by altering fuel mass flow. By altering fuel mass flow, airflow and pressure ratio increased accordingly although this increase was lower in areas with high TIT, thus providing suitable EQR to attain desired TIT. This resulted also in slightly higher turbocharger (TC) speed as a function of expansion ratio, TIT and sum mass flow. TC speed was estimated on the basis of compressor operating map and is presented in Table 3. Key parameters for both modes of operation together with ambient conditions are presented in Table 3.

Table 3: Operating parameters of experimental turbine.

<table>
<thead>
<tr>
<th></th>
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<th></th>
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<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>LW @ 750 Reg</td>
<td>743.2</td>
<td>414.0</td>
<td>2.13</td>
<td>0.137</td>
<td>14.16</td>
<td>640</td>
<td>17.7</td>
<td>32.2</td>
<td>98467</td>
</tr>
<tr>
<td>LW @ 800 Reg</td>
<td>800.2</td>
<td>436.0</td>
<td>2.37</td>
<td>0.154</td>
<td>17.59</td>
<td>713</td>
<td>17.0</td>
<td>33.5</td>
<td>98497</td>
</tr>
<tr>
<td>LW @ 850 Reg</td>
<td>849.5</td>
<td>466.9</td>
<td>2.55</td>
<td>0.167</td>
<td>20.12</td>
<td>755</td>
<td>19.2</td>
<td>30.4</td>
<td>98453</td>
</tr>
<tr>
<td>D2 @ 750 Reg</td>
<td>747.9</td>
<td>412.9</td>
<td>2.06</td>
<td>0.133</td>
<td>6.25</td>
<td>621</td>
<td>20.1</td>
<td>29.3</td>
<td>98709</td>
</tr>
<tr>
<td>D2 @ 800 Reg</td>
<td>805.8</td>
<td>441.7</td>
<td>2.28</td>
<td>0.147</td>
<td>7.63</td>
<td>689</td>
<td>21.7</td>
<td>28.1</td>
<td>98684</td>
</tr>
<tr>
<td>D2 @ 850 Reg</td>
<td>850.5</td>
<td>465.9</td>
<td>2.44</td>
<td>0.159</td>
<td>8.68</td>
<td>731</td>
<td>23.0</td>
<td>27.1</td>
<td>98661</td>
</tr>
<tr>
<td>LW @ 700 Spl</td>
<td>692.2</td>
<td>101.6</td>
<td>1.81</td>
<td>0.219</td>
<td>17.78</td>
<td>491</td>
<td>21.3</td>
<td>51.1</td>
<td>97610</td>
</tr>
</tbody>
</table>
Data acquisition was made after initial stabilization of the operating parameters for each point of the matrix. Tests were followed by disassembly of the engine and careful visual inspection of the injection nozzle, combustion chamber and turbine rotor.

3. Results and discussion

Experimental data, gathered through aforementioned variation of several parameters closely represents the actual conditions present in a micro gas turbine (MGT) and can be easily transferred to other small turbine systems. Measured values of different operating points were not directly compared in this study due to their different excess air ratios. The influence of different equivalence ratios was eliminated by correction of the emissions concentrations to oxygen concentration of $\text{v}_{\text{O}_2} = 0.15$ in exhaust gases as proposed in [19].

Since direct $\text{O}_2$ concentration was not measured, the corresponding correction factor had to be calculated from the excess air ratio that was evaluated considering fuel composition and its stoichiometric ratio as well as measured fuel and air flow (compressor air flow plus atomizing air flow). From this data and from the calculated exhaust gas composition it is possible to determine the amount of oxygen from air that reacted with fuel under assumption of complete combustion. Eq. (1) gives theoretic mass of excess oxygen in exhaust gases:

$$w_{O_2 \text{exh}} = w_{O_2 \text{air}} \left(1 - EQR_{15\% \text{O}_2 \text{exh}} \right), \quad (1)$$

Eq. (2) gives the equivalence ratio at exhaust gas oxygen concentration $\text{v}_{\text{O}_2} = 0.15$:

$$EQR_{15\% O_2 \text{exh}} = \frac{sth(w_{O_2 \text{air}} - w_{O_2 \text{exh}})}{sth w_{O_2 \text{air}} + w_{O_2 \text{exh}}} \quad (2)$$

and Eq. (3) gives the correction applied to measured emission concentrations:

$$v_{i \text{ exh}} = v_{i} \frac{EQR_{15\% O_2 \text{exh}}}{EQR_{i}} \quad (3).$$

The following species are considered when calculating composition of exhaust gases: $\text{CO}_2$, $\text{H}_2\text{O}$, $\text{SO}_2$, $\text{NO}$, $\text{O}_2$ and $\text{N}_2$. According to the above procedure, the emissions concentrations were increased if exhaust gases were diluted to more than $\text{v}_{\text{O}_2} = 0.15$ and vice versa. This correction is commonly applied to measured data when interpreting gas turbine emissions with a purpose to remove ambiguity when comparing different sets of experimental data [19].
The results are plotted versus TIT to expose the influence of different temperature states in combustion chamber on thermodynamic parameters and emissions concentrations. Additionally, measured values of emissions are also presented in comparison to equivalence ratios with intention to distinguish the influence of temperature state and of global excess air rate on emissions.

### 3.1 Variation of operating parameters

Fig. 5 shows interdependence of primary air temperature and TIT for both modes of operation – simple and regenerative cycle mode and for both fuels – LW and D2. It is discernible from the figure that primary air temperature increases with increasing TIT for both modes of operation. In the simple mode, primary air temperature increases due to increased pressure ratio across the compressor, whereas in the regenerative mode it additionally increases due to higher heat flux in the heat exchanger.

Fig. 5: Primary air temperature versus turbine inlet temperature.
Fig. 6 quantifies an intuitive interrelation between fuel consumption, TIT, calorific value of the fuel and selection of the operational mode. Here, compensation of lower calorific of LW with increased fuel flow in comparison to D2 to obtain similar heat release for both fuels is visible. This is also the reason for EOR reduction (visible in Fig. 7) when operating with high primary air temperature. However, even if EQR is different for different fuels at the same TIT, this does not influence other thermodynamic parameters (pressure ratio, air mass flow, turbine outlet temperature due to the same global conditions that are present upstream of the turbine. This can be partly seen in Fig. 8, where almost no change in pressure ratio is detected when performing measurements with either D2 or LW in both simple and regenerative cycle mode.

When considering simple and regenerative cycle turbine engines, it should be taken into account that pressure ratio is selected either with the focus on maximum efficiency or maximum specific power, whereas the pressure ratio values also differ based on the selected cycles. In the case of regenerative cycle the efficiency can be increased by regeneration up to the pressure ratio where compressor discharge temperature reaches the turbine discharge temperature, whereas until this pressure ratio efficiency trend is significantly influenced by the efficiency of the regeneration with high regeneration efficiencies allowing for high efficiencies at low pressure ratios. Specific power exhibits a maximum where compressor discharge temperature equals turbine discharge temperature. Without additional measures (intercooling and/or reheating), the most suitable regenerative cycle engines for achieving high efficiency are therefore micro gas turbines with low pressure ratios being in-line with their low price and low complexity.

In the case of simple cycle mode, the maximum efficiency is achieved at much higher pressure ratios than in regenerative cycle and even surpasses the pressure ratio corresponding to peak specific power, which is generally reached at slightly lower pressure ratios. Similarly as for the regenerative cycle turbine engines, the efficiency drops with further elevation of pressure ratio also for the simple cycle ones. This can be reasoned by the difference in compressor and turbine polytropic efficiencies, where reduction of compressor efficiency with increasing pressure ratio is more pronounced than increase in the turbine efficiency increases at the same increase in expansion ratio. Reduction in compressor efficiency is often called “preheat effect” [20] as the reason for reduced efficiency of compressor occurs due to higher temperatures at higher pressure ratios and thus larger frictional losses. The maximum
specific work is thus achieved at pressure ratio, usually slightly lower than maximum efficiency, where energy input can be maximized. Usually, the design point for aircraft turbines is somewhere in-between the two maximums where efficiency and specific power curves are relatively flat and it is thus possible to obtain high efficiency together with high specific power [20].

Fig. 7: Equivalence ratio for target operating points.

Fig. 8: Pressure ratio for target operating points.
3.2 Effects on carbon monoxide emissions.

Mechanisms leading to reduction of CO emissions at elevated TIT were already explained in [13]. Therefore, this paper focuses on the effects of different primary air temperatures on CO emissions. First, in depth analysis of the CO emissions of the LW is presented, which is followed by the comparison of the results for the D2.

Fig. 9 demonstrates the beneficial effect of the higher primary air temperature to obtain lower CO emissions. This was somehow expected as LW, due to its physical and chemical properties, represents a relatively heavy fuel. High viscosity results in a formation of larger droplets and this mechanism is further enhanced by high density (1300 kg m$^{-3}$) and low volatility of the fuel. Higher primary air temperature is therefore beneficial to either increase evaporation rate and to promote mixing of fuel and air to reduce the penetration depth of the spray.

CO as an intermediate species in combustion process, is formed due to insufficient residence time of the combustible mixture at high temperatures provided that sufficient oxygen is available as in the analyzed case. Trends presented in Fig. 9 are the consequence of several interacting mechanisms. Since the temperature of the LW was equal in both cases, it can be assumed that initial droplet sizes at the combustion chamber entry were the same. Considering the fact that the applied nozzle featured internal mixing with double surface impingement it is very unlikely that spray is subject of any further break-up mechanism after exiting the nozzle. Therefore primary air temperature does not influence initial droplet size, however, higher primary air temperature in the first zone of combustion chamber still shortens the first phase of the evaporation process by shortening heat-up period of the droplet through increased radiative and mostly convective heat transfer. This mechanism reduces penetration depth of the liquid phase of LW. Shorter evaporation period gives more time for diffusion and advection of fuel vapors and formation of combustible mixture. Increase of convective heat transfer is further supported by the fact that higher velocities are present in the primary zone due to lower density of primary air. Higher velocities are additionally related to more intensive turbulence which further promotes mixing of the fuel vapor and air.

![Fig. 9: Carbon monoxide emissions corrected to exhaust gas oxygen concentration $\nu_{O_2} = 0.15$.](image)
Operation with LW in the regenerative cycle mode thus promotes faster formation of the combustible mixture and assures that combustion takes place sufficiently fast, before excessive amounts of intermediate species pass the primary zone of the combustor, which is characterized by the highest temperatures. If latter occurs, partly burnt mixture enters the dilution zone, which is characterized by rapid quenching of the reactions. It can therefore be concluded that this phenomenon is less likely to occur for high primary air temperatures. This conclusion is also supported by the statement presented in [19], where it was shown that entrapment of partly burnt mixture in liner cooling air adds a notable amount of CO to exhaust gases. The effect of this entrapment is similar as aforementioned survival of intermediate species through primary zone and is thereby reduced by higher air temperature.

Reaction quenching is more intense for cooling air temperatures just above 100°C (simple cycle engine) than at 400°C as in case of regenerative cycle engine. Thereby the lowest CO emissions when firing LW were achieved at highest TIT and highest air temperature. Absolute reduction of CO over the measuring range is almost constant at roughly 220 μL L⁻¹ when comparing regenerative or simple cycle system. However, it is evident from the trend of CO emissions over TIT that further elevation of TIT, would reduce the concentration of this species even more. This could indicate that commercial MGT systems using LW as a fuel could possibly have lower flexibility in terms of loading, as operation with partial load (and reduced TIT) could lead to excessive amounts of CO. Similar trend is also shown in [21], where various methyl esters and hog fat were used as a fuel, i.e. in [22] esterified jatropha oil-diesel blend and in [23] waste trap grease oil was used as a fuel.

More pronounced dependence of CO emission and thus of the combustion efficiency on TIT can be observed for LW in comparison to D2. This difference most likely originates from the specific molecular composition of wood decomposition products which contain numerous cyclic hydrocarbons that possess high autoignition temperature. Therefore CO emissions of the D2 do not follow the same trend as is characteristic for the LW. With D2, CO concentration is relatively insensitive to primary air temperature and TIT. This is clearly in line with higher volatility of D2 in comparison to LW and its molecular composition with high content of species with low autoignition temperature. Therefore, when using D2 in simple cycle mode, evaporation rate is already sufficiently high, mixture formation is sufficiently fast and thus beneficial effect of high primary air temperature is not visible through emissions readings. This phenomenon can partly be explained by the high excess air ratio (visible through EOR) which makes detection of small differences in CO concentration difficult.

### 3.4 Effects on total hydrocarbons emissions

Total hydrocarbons include fuel vapors and unreacted radicals (partly thermally decomposed fuel molecules) that escape from the combustion chamber. Although their formation mechanisms are more complex compared to those of CO [19] reasons for generation of both species feature similar basis. Therefore trends of THC emissions over TIT and primary air temperature follow those of the CO emissions for both fuels. The effect of elevated primary air temperature is visible through lowered THC concentrations where intense reduction of emissions concentrations is noticed especially for LW (Fig. 10). With primary air temperature sufficiently high, increased evaporation rate and faster mixture formation contributes to lower THC emissions mainly through extension of combustible mixture residence time. A sharp reduction in THC emissions is visible at 800°C TIT in simple cycle operation. Although it is difficult to precisely define the responsible mechanism due to relatively complex molecular composition of LW fractions, the reasons for this could be in variation of primary air temperature along the tested TIT interval, visible in Fig. 5. From 800°C to 900°C TIT in simple cycle mode, the primary air temperature is between 150°C and 170°C which is in the range of glycerol flash point. Considering the large amount of glycerol used as a solvent, the reason for THC reduction might lie in high enough temperature even in the coldest spots of combustion chamber for glycerol to ignite and thus reduce the THC emissions originating from glycerol in LW.

Unlike the LW, nearly negligible reduction of THC concentration can be observed for D2 when increasing TIT, whereas D2 is already characterized by very low THC concentrations throughout the complete TIT
range. These two trends are in accordance with fuel compositions, where LW is characterized by the high content of cyclic hydrocarbons which require high temperatures to dissociate. The content of these molecules is notably lower in D2 than in LW and thus beneficial effects of increased primary zone temperature are less pronounced for the D2 as the majority of the molecules dissociates at lower temperatures.

![Graph](image)

Fig. 10: Unburned hydrocarbon emissions corrected to exhaust gas oxygen concentration $\nu_{O2} = 0.15$.

### 3.5 Effects on emissions of nitrous oxides

Contrary to beneficial effects of elevated air temperature on THC and CO emissions, increased concentration of NO$_x$ species was measured at high primary air temperatures for both fuels. It was expected that concentration will rise with TIT due to higher temperatures which promote thermal mechanism. However, when employing low primary air temperature, NO$_x$ emissions were almost independent from TIT for both fuels, whereas with high air temperature, stronger dependence of NO$_x$ species concentration on TIT is visible in Fig. 11 especially for LW. Several authors propose that prompt mechanism of NO$_x$ formation is the prevailing one in gas turbines [21, [24 and according to this mechanism formation of NO$_x$ should not be significantly influenced by the primary air temperature and TIT. In the case of low air temperature this is clearly supported by the measured data as NO$_x$ emissions are nearly constant over TIT. At high primary air temperatures, formation through thermal mechanism most likely intensifies as concentrations of NO$_x$ are increasing with increasing TIT. This trend is also more pronounced when using LW.

Prompt mechanism occurs mainly in rich flames through CH radicals even at low temperatures due to low activation energies of governing reactions. Conditions when employing low primary air temperature are similar when using either LW or D2. Primary air temperature varies from 100°C to 170°C and equivalence ratio is higher than for recuperated turbine. In comparison to recuperated turbine this means richer combustion, which clearly promotes prompt mechanism in the simple cycle turbine regardless of the fuel type. As this is presumably the prevailing mechanism, overall NO$_x$ concentrations are relatively insensitive to TIT. A different trend is observed for the recuperated turbine with more pronounced sensitivity of NO$_x$ formation mechanisms on TIT and air temperature as shown in Fig. 10. With elevated
air temperature, equivalence ratio drops and prompt mechanism is suppressed, however, thermal mechanism obviously exceeds its temperature threshold (above 1700°C [24]) in certain areas of the combustion chamber and additional component of NO\(_x\) starts to form. As a consequence of low EQR, large amounts of oxygen are available which then further increases the formation rate of the thermal NO\(_x\). This trend is a bit more pronounced when firing LW with high primary air temperature. This was attributed to high quantity of bonded oxygen which is available directly in the flame zone where temperatures are the highest and thus formation rate of this NO\(_x\) component increases. As D2 has almost negligible oxygen content, the described increasing trend is more gently sloping.

Formation of NO\(_x\) is also possible through fuel bound nitrogen or via nitrous oxide. The latter is highly unlikely to cause detectable difference in concentrations between low and high air temperature or between fuel types because this mechanism requires three body reaction and is therefore preconditioned by high pressures, which are usually not encountered in MGTs and also not in the applied experimental turbine. Additionally, pressure ratios in both, simple and regenerative mode, are equal and therefore the amount of NO\(_x\) formed through nitrous oxide (N\(_2\)O) should be the same, as governing reactions have low activation energy and should not be influenced by different primary air temperatures, again regardless of the fuel type.

However, formation of NO\(_x\) from fuel bound nitrogen is more likely to contribute to increased NO\(_x\) emissions when firing LW in both modes of operation. In this case, organic NO\(_x\) originates from 1.9 g kg\(^{-1}\) of nitrogen that LW contains. In conventional fuels, conversion of FBN (fuel bound nitrogen) to NO\(_x\) is usually very efficient. If at given experimental conversion efficiency of FBN would be unity, the total contribution to the emitted NO\(_x\) would be around 150 μL L\(^{-1}\). Thereby it is evident, that this conversion is far from unity. Additionally, a study in [14] indicates that under similar conditions (with neutralized type of LW), roughly 330 g kg\(^{-1}\) of FBN was converted to NO\(_x\). Considering this fact, higher NO\(_x\) emissions of the LW in comparison with D2 in simple cycle mode could be partly attributed to FBN. FBN undergoes conversion to NO\(_x\) also at high primary air temperatures. Considering measured concentrations with LW and D2, it could be concluded that contribution of thermal and prompt mechanism is actually lower when firing LW than D2 in regenerative cycle mode. However, with this set of measured data it is not possible to determine the ratio between separate NO\(_x\) formation mechanisms (thermal, prompt and FBN borne NO\(_x\)) for LW or D2. On the basis of discovered trends it is only possible to identify the intensities of separate mechanisms which are very important when developing NO\(_x\) reduction strategies when using LW.

It could be expected that differences in evaporation rate and atomization quality between D2 and LW could lead to differences in local EQR and temperature distribution. As described above, temperature and concentration field in combustion chamber mainly influence formation rates of thermal and prompt NO\(_x\). As initial droplet sizes of the LW are larger than those of the D2 and evaporation rate of LW is lower, it is possible that a form of single droplet combustion could develop which promotes thermal NO\(_x\) formation. In contrast, richer mixture in the mid-space between droplets, which is characteristic for the D2, could promote formation of prompt NO\(_x\) and in the same time suppress thermal NO\(_x\) due to larger rate of premixing. However, this theory would have to be further investigated to accurately evaluate the effects of initial droplet sizes and consequent EQR and temperature distribution.

With LW as a novel fuel, limited conclusions on detailed chemical kinetics of NO\(_x\) formation through interpretation of emissions measurements can be made. The major observation was that NO\(_x\) emissions in regenerative cycle mode exhibit steeper trend over TIT for LW in comparison to D2. This was attributed to two competing mechanisms of NO\(_x\) formation. It seems that at lower TIT, oxygen content in the LW helps to suppress prompt NO\(_x\) formation through elimination of CH radicals in the flame zone, while at higher temperatures the same oxygen promotes formation of NO\(_x\) through thermal mechanism.
The collective influence of primary air temperature and EQR on emissions is presented in Fig. 12. As EQR is directly influenced by TIT (and vice versa), the trends here are similar as in Fig. 9, Fig. 10 and Fig. 11. However, it is once again more clearly visible that EQR is not the only influencing factor in emissions formation and that CO concentrations are also highly dependent on primary air temperature as described above. Even at very low EQR, where high CO emissions would be expected according to trend observed in simple cycle operation, where CO emissions are elevating with lower EQR values, the CO emissions are low. This indicates that high primary air temperature offsets the high CO emissions that would otherwise be expected at low values of EQR, by maintaining sufficiently high temperature in primary combustion zone and thus avoiding local blowouts due to insufficient fuel concentration.

The trends of THC emissions are similar indicating that EQR is not the sole leading parameter influencing combustion ability of LW. D2 exhibits insensitivity to both, EQR and primary air temperature due to its superior physical and chemical properties over LW for THC as well as for aforementioned CO emissions.

The collective influence of EQR and primary air temperature on NO\textsubscript{x} emissions is visible. Tests with low primary air temperature revealed steady NO\textsubscript{x} emissions over higher end of EQR, indicating that EQR is not the main influencing factor on NO\textsubscript{x} emissions in operation with low primary air temperatures. At lower end of EQR and with high primary air temperatures, the NO\textsubscript{x} emissions become dependent also on EQR and are notably higher due to higher overall temperature in primary zone of combustion chamber, providing suitable environment for thermal NO\textsubscript{x} formation.
3.7 Smoke emissions

Measurements of smoke emissions were primarily performed due to hypothesis that lower primary air temperature would significantly reduce evaporation rate of LW and cause notable contribution to soot formation through reduced homogeneity of the mixture. However, measurements revealed low quantity of soot in all cases even in simple cycle mode operation. Due to evident low formation of soot particles and at the same time low EQR it was not possible to distinguish between opacity of exhaust gases when operating on diesel fuel or LW. Considering reduced atomization ability of the LW, low degree of soot formation is believed to be the consequence of high content of bonded oxygen.

Although the literature in the area of how high (433.4 g kg\(^{-1}\)) oxygen content in LW influences the particulate matter (PM) and soot emissions in combustion engines is yet unavailable, numerous studies are available on other oxygenized fuels, i.e. biodiesels [25], diglyme [26], diethyl adipate, dimethyl carbonate, ethyl carbonate, dimethoxymethane and ethanol [27] which represent a wide interval of oxygen contents (up to 421 g kg\(^{-1}\)). It is widely agreed in upper and also several other studies that oxygen content in the fuel is beneficial for reduction of PM emissions.

In certain cases, this is a consequence of the C-C bond absence, i.e. ethers which contain C-O-C bond and thus during combustion favor the formation of CO, leaving only one C atom to contribute to the pool of soot-forming species [28]. Even if LW is not fully characterized and does not necessarily contain ether-like structures to support this mechanism, it still contains several OH groups which are present in several components of LW (as functional groups in p-coumaryl alcohol, coniferyl alcohol, glycerol, diethylene glycol). OH radicals limit aromatic ring growth and intercept soot particles [29], thus strongly contributing to reduction in PM emissions. Furthermore, oxygen content shifts the pyrolysis process in fuel rich areas of the spray, which further reduces soot formation. Additional benefit of oxygenated fuels is identified in [30] where biodiesel was observed to produce much more reactive soot particles in diffusive combustion. This results in acceleration of soot burnout in the final stages of combustion process, leading to higher opacity of exhaust gasses.
When looking specifically on LW it can be noted that through fuel-bound oxygen, one third of stoichiometric oxygen is already delivered in the combustible mixture which offsets the richest possible EQR towards stoichiometric, thus lower amount of diffusive oxygen is needed further in mixture formation process.

### 3.8 Visual inspection of hot path

Even if the emissions concentrations are corrected to exhaust gas oxygen concentration $v_{O2} = 0.15$ and the effect of different rates of dilution is eliminated, some characteristics of operation with different EQR have to be taken into account when comparing different primary air temperatures. The required fuel flow for target TIT varies greatly for different degrees of recuperation which directly influences the overall equivalence ratio as well as local A/F ratios in different parts of the combustion chamber. Temperature of the air significantly influences flow dynamics and velocity of the air-stream which is further responsible for formation of the mixture and local A/F ratios. After careful inspection of atomizer data it was concluded, that during operation with single fuel (LW or D2), mean droplet size did not change significantly with variation of fuel flow as this type of atomizers (Swirl-air) employ large quantities of atomizing air and its atomization ability is insensitive to fuel flow rate [31], which differed for a factor of 2.5 between two utmost cases (low primary air temperature and high TIT versus high primary air temperature and low TIT).

Visual inspection of hot path components after operation with LW and with different degrees of recuperation revealed heavy fuel deposits in case of low primary air temperature and surprisingly clean hot path when employing high primary air temperature as can be seen in Fig. 13. Black deposits resembled a gummy substance which was most likely partly polymerized unburnt LW. Polymerization occurred only on the walls of the combustion chamber where low temperature cooling air enabled prolonged exposure of the wall-impinged fuel to temperatures over 160°C. As was described in [13], mechanisms of repolymerization start to take place above this temperature. Fuel impingement is more pronounced in simple cycle turbine and it is mainly related to the increased flow of LW, required to maintain target TIT. Larger heat capacity of the injected LW (due to larger mass) accompanied by low air temperature extended the evaporation rate to such an extent that droplets penetrated the primary zone and impinged on the combustion chamber walls. This effect was further stimulated by reduced turbulence due to lower velocities in the air path. Contrary, at high air temperatures, the combustion chamber remained free of any fuel traces in polymerized or coke form. This was also the case when utilizing D2 either with high or low primary air temperatures.

![Fig. 13: Deposits on combustion chamber with low (left) and high (right) primary air temperature.](image)
Surfaces in the hot path downstream of the fuel nozzle when operating with LW were covered with resistive deposits of light brown color, regardless of operating mode of the turbine. These deposits did not affect stability of operation during the experiments. Brown deposits most likely originate from ash composition in the LW, although the quantity of the ash should be low as technical grade glycerin and technical grade di-ethylene glycol was used in fuel processing. Considering the type of wood used, the overall ash content in the fuel was estimated to be roughly $1.1 \text{ g kg}^{-1}$.

During operation and disassembly of experimental turbine, formation of carbon deposits was also observed on the discharge surfaces of the nozzle (Fig. 14) when operating with LW. High primary air temperature induced heavy formation of carbon deposits, whereas at low air temperatures deposits were mainly composed of ash and were substantially less developed. The main reason for this is high fuel preheat temperature ($90°C$) in the case of LW that reduces the temperature margin towards the temperature where thermal degradation occurs. In [13], where influence of liquefied wood preheat temperature on combustion performance was tested, same fuel nozzle upgraded with thermal protection coating to avoid coke formation was used. In this study optimal preheat temperature of the liquefied wood which presents an important parameter to obtain lowest emissions possible was defined.

Fig. 14: Deposits on fuel injection nozzle surfaces for low (left) and high (right) primary air temperature.

In both, i.e. simple and regenerative mode, luminous radiation from soot particles in the flame and nonluminous radiation from heteropolar gases [19] represent heat sources that contribute to the polymerization of the fuel at the nozzle. Moreover, convective heating with hot primary air in the regenerative cycle mode represents an additional heat source. Therefore, deposits were formed only at high air temperatures, whereas at low air temperatures, i.e. simple cycle mode, nozzle discharge surfaces were clean due to convective cooling of the nozzle with primary air. Because radiative heat flux grows with fourth power of temperature, any elevation of primary air temperature increases nozzle surface temperature. Data given in [19] shows that at certain conditions, radiative heat transfer can be as much as 20 times higher than convective heat transfer.

4. Conclusions

The scope of this work was to evaluate the impact of combustion chamber air temperature in the MGTs when firing innovative biofuel, produced by liquefaction of spruce wood in acidified polyhydroxy alcohols.
Different air temperatures were achieved by regenerating exhaust gas heat through heat exchanger. The results of the study suggest significant influence of operating mode on exhaust emissions of innovative fuel and its ability to support stable operation in MGT.

The main driver for this is indeed the change in primary combustion air temperature, which affects the parameters in the primary zone. Besides increased velocity and turbulence intensity in regenerative cycle mode, the change in operational cycle influenced also EQR, resulting in a much lower air/fuel ratio. Variation of EQR of the simple or of the regenerative cycle revealed that the contribution of reducing EQR is perceivable through elevation in CO and THC emissions in both modes of operation. However, the change in operational mode from simple to regenerative cycle results in overall lower measured values of CO and THC. In fact, emission values of the regenerative cycle are even lower than the values that would correspond to extrapolating the data of the simple cycle mode. Thus, when operating MGT with LW, adequate regeneration rate should be selected with regard to emission trends as it seems that at certain EQR values, achieved in-between simple and fully regenerative mode of operation, emissions exhibit a maximum value of CO and THC.

The selection of adequate primary air temperature by incorporating appropriate degree of regeneration also has the potential to shape the NOx emissions as they are strongly dependent on operating mode and as in the regenerative cycle they also exhibit a strong TIT dependence. Thus a high degree of regeneration might elevate NOx emissions to the point where exhaust gas after-treatment would be required. Additionally, primary air temperature also influences the deposit formation on combustion chamber walls and fuel injection nozzle, with the first being prone to deposit build-up in operation with simple cycle mode, and the other one prevailing in regenerative cycle mode, once more indicating that suitable degree of regeneration has a potential to successfully support stable combustion of LW in MGTs.

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5. References


