Presented paper deals with adaptation procedure of a microturbine (MGT) for exploitation of refuse derived fuels (RDF). RDF often possess significantly different properties than conventional fuels and usually require at least some adaptations of internal combustion systems to obtain full functionality. With the methodology, developed in the paper it is possible to evaluate the extent of required adaptations by performing a thorough analysis of fuel combustion properties in a dedicated experimental rig suitable for testing of wide-variety of waste and biomass derived fuels. In the first part key turbine components are analyzed followed by cause and effect analysis of interaction between different fuel properties and design parameters of the components. The data are then used to build a dedicated test system where two fuels with diametric physical and chemical properties are tested – liquefied biomass waste (LW) and waste tire pyrolysis oil (TPO). The analysis suggests that exploitation of LW requires higher complexity of target MGT system as stable combustion can be achieved only with regenerative thermodynamic cycle, high fuel preheat temperatures and optimized fuel injection nozzle. Contrary, TPO requires less complex MGT design and sufficient operational stability is achieved already with simple cycle MGT and conventional fuel system. The presented approach of testing can significantly reduce the extent and cost of required adaptations of commercial system as pre-selection procedure of suitable MGT is done in developed test system. The obtained data can at the same time serve as an input for fine-tuning the processes for RDF production.

Keywords:
internal combustion engine, methodology, microturbine, thermodynamic cycle, waste derived fuel

Highlights:
• Dedicated experimental system was developed for reproducing MGT conditions.
• Combustion of liquefied biomass and tire oil was analyzed under MGT conditions.
• Steps to achieve stable, efficient and low emission operation were presented.
• Fuel limitations in terms of operating parameters and MGT design were presented.
1. INTRODUCTION

Advances in classification of refuse derived fuels and solid recovered fuels (RDF/SRF) with better quality management concepts (Rada and Andreottola, 2012) are paving the way towards utilization of a wide portfolio of different RDFs in micro gas turbines (MGT) by offering potential for rendering their characteristics through different processes for their conversion. By this it is possible to target also the systems in the very low power range where externally fired systems are not available. Thereby it is possible to obtain similar efficiencies as on a large scale (Lombardi et. al 2015). The most promising products for this purpose are pyrolysis oils and gasses from waste polymer materials (Williams, 2013) and waste lignocellulosics (Mohan, 2006), products of gasification (Arena, 2012) and products of direct liquefaction. Although these processes yield fuels with attractive physical and chemical properties, the utilization of the fuel in MGTs requires at least some research work to identify the main and barriers at MGT operation considering the main fuel properties, which are as follows:

- viscosity that influences mixture formation, increases CO emissions and dictates the complexity of fuel system (Sallevelt et. al 2014) and (Chiaramonti et. al 2005),
- heating value and stoichiometric ratio that influence energy density of the mixture and thus available power output or potential power de-rating (Hossain and Davies, 2013),
- density and heating value, which determines volumetric energy content and influences mixture formation as well as injection system performance.

Availability of research publications on utilization of waste-derived fuels in MGT setups is scarce and it is mainly focused on 1st generation biofuels. Thus, studies by (Prusi et al. 2012) and (Chiariello et. al 2014) used straight vegetable oil and their blends with diesel in a 30 kW unit, (Cavarzere et. al 2012) studied the combustion of straight vegetable oils from rapeseed, sunflower and soybean in a 50kW MGT, (Chiaramonti et al. 2013) tested refined rapeseed vegetable oil in 50 kW APU derived MGT whereas (Al-Shudeifat and Donaldson 2010) studied waste trap grease as a fuel in 150 kW MGT. Other authors focused mainly on methyl esters from various feedstock. (Habib et al. 2010) presented a study on 30 kW system firing methyl esters from soy, canola, rapeseed and hog-fat, (Chiaramonti et al. 2013) also tested methyl esters from mixed vegetable oils. Several studies on various methyl esters blended with diesel were also conducted, however they are at present time considered as an industrial standard, since recent EN590:2013 standard already allows up to 7% fatty acid methyl ester content. Some work was already performed also on externally fired systems (EFGT) (Al-attab and Zainal, 2010; Riccio and Chiaramonti, 2009), but due to significant differences to internal combustion engines, this area will be omitted here.

In some of upper cases, MGT systems required different levels of adaptations (e.g. fuel preheating, increased injection pressure) to obtain stable combustion process with manageable emission levels. When aiming at the usage of RDF it is thus important to select an MGT system being as suitable as possible for the use of target fuel in terms of its physical and chemical properties, which minimizes the required upgrades of the system. The main focus of the study is thus to establish an interaction between specific fuel properties, required system modifications and their impact on MGT performance and exhaust emissions, by:

- Designing a dedicated experimental microturbine engine being a versatile platform for exploring a wide test space with the aim of assessing impact of specific changes to identify suitable commercial MGT setups with intention to minimize the required changes for a target fuel.
- Developing holistic methodology for selection of appropriate microturbine system considering specific properties of RDFs, which is applicable also as guiding strategy for
fine-tuning of fuel production process. The presented methodology can significantly reduce the complexity and cost of commercial system adaptation by pin-pointing the critical components in the early stage of development.

- Identifying and analyzing design requirements on injection system suitable for utilizing wide variety of fuels by taking into account correct nozzle design, suitable turn-down ratio, proper atomization performance and sensitivity to thermal loading.
- Providing detailed insight into combustion chamber conditions with time-resolved thermodynamic and exhaust emission data in different operating regimes.
- Demonstrating the above items through the analysis on the overall system response with two different RDFs: diesel-like fraction of tire pyrolysis oil (TPO), obtained through vacuum pyrolysis of shredded waste tires which in certain properties resembles diesel fuel, and liquefied wood (LW), obtained through chemical liquefaction (solvolysis) by decomposing main constituents of lignocellulosic material in acidified multifunctional alcohols, with high viscosity and low energy content.

2. SYSTEM DEVELOPMENT

In Figure 1 a dedicated experimental system layout is presented with indicated key components and characteristic points for measuring the thermodynamic parameters of the working fluid and emission measurement devices. In the subsequent sections analyses will follow fuel and gas path in the downstream direction.

Figure 1. Experimental system layout with marked characteristic points.
Design of a fuel system is most profoundly influenced by the physical and chemical properties of the fuel that dictate the required performance of the system. Fuel properties that need to be considered when designing the fuel delivery system are summarized in Table 1.


<table>
<thead>
<tr>
<th></th>
<th>Tire pyrolysis oil (TPO)</th>
<th>Liquefied wood (LW)</th>
<th>Diesel fuel (D2)</th>
</tr>
</thead>
<tbody>
<tr>
<td>C [wt.%]</td>
<td>83.45 – 85.60</td>
<td>47.60</td>
<td>86.2</td>
</tr>
<tr>
<td>H [wt.%]</td>
<td>9.59 – 11.73</td>
<td>7.98</td>
<td>13.7</td>
</tr>
<tr>
<td>N [wt.%]</td>
<td>0.4 – 1.05</td>
<td>0.19</td>
<td>/</td>
</tr>
<tr>
<td>S [wt.%]</td>
<td>0.960</td>
<td>0.89</td>
<td>/</td>
</tr>
<tr>
<td>O [wt.%]</td>
<td>0.10 – 3.96</td>
<td>43.34</td>
<td>/</td>
</tr>
<tr>
<td>Density [kg/dm³]</td>
<td>0.923</td>
<td>1.3</td>
<td>0.83</td>
</tr>
<tr>
<td>LHV (lower heating value) [MJ/kg]</td>
<td>41.0</td>
<td>20.2</td>
<td>43.1</td>
</tr>
<tr>
<td>Stoichiometric ratio</td>
<td>~13.8</td>
<td>6.8</td>
<td>14.5</td>
</tr>
<tr>
<td>Viscosity</td>
<td>3.94 mm²/s @ 20°C</td>
<td>106 mm²/s @ 80°C</td>
<td>2 – 4.5 mm²/s @ 40°C</td>
</tr>
<tr>
<td>VHV (volumetric heating value) [MJ/L]</td>
<td>37.84</td>
<td>26.26</td>
<td>35.77</td>
</tr>
</tbody>
</table>

As presented in Table 1, TPO and LW feature significantly different composition and thus also fuel properties and can thus be considered as two fuels representing outmost (or near to outmost) envelope of liquid RDF. Tire pyrolysis oil has diesel-like physical properties, however due to high share of aromatic compounds it features low cetane number. It is almost fully characterized (Williams, 2013) and also tested several times in reciprocating engines (Murugan et al, 2008; Martinez et al, 2014; Koc, 2014; Vihar et al, 2015) and also in a MGT to explore the possibility for exploitation of polymer-derived waste (Seljak et al, 2014a). Liquefied wood represents a heavy fuel, with high viscosity and low heating value that was previously tested by (Seljak et al, 2012, 2014b, 2014c) to characterize combustion and emission formation mechanisms of LW in a microturbine.

2.1.1 Influence of viscosity on fuel system design

Viscosity influences pumping ability, pressure losses, cold start performance of the fuel system and determines the need for preheating which is also determined by injection system capability. In terms of viscosity, it is currently considered that 15 mm²/s (Lefebvre and Balal, 2010) is the maximum value suitable for gas turbines, although some authors propose even lower values of 12 mm²/s (Gupta et al, 2010) or 10 mm²/s (Chiaramonti et al, 2005) as this studies take into account original fuel systems. Therefore for fuels with viscosity below 10 mm²/s at room temperature, conventional fuel system, intended for diesel fuel, is applicable if volumetric heating values are similar to the one of the diesel fuel.

To achieve these viscosity limits for fuels featuring high viscosity values two approaches are available. First approach comprises various blending or refining methods, which can reduce benefits associated with the use of RDF. Second approach relies on fuel preheating system as the mean to reduce viscosity to an acceptable level.

The most important parameter for determining specifications of the fuel preheating system is temperature dependence of viscosity. This characteristic can often be calculated by the use of empirical regression functions or modified Andrade equation in case of bio-diesel or vegetable oils.
(Yilmaz, 2011) or it can be measured in the case of novel fuels as they often exhibit different temperature dependence than conventional fuels (Seljak, 2012; Chiaramonti et al., 2005) and do not obey semi-empirical equations. Measured dependence of LW viscosity on temperature is presented in Figure 2.

![Figure 2: Measured temperature dependence of viscosity for LW](image)

In preheating system, presented in Figure 3, maximum preheating temperature, maximum heat-up rate of the fuel and maximum residence time of the fuel in the heater are also constrained by the following fuel specific constraints:

- **Fuel aging and chemical stability at elevated temperatures.** Properties of certain heavy biomass-derived fuels exhibit strong dependence on retention at high temperatures. The viscosity can, in case of pyrolysis bio-oils, increase by 720% (Nolte and Liberatore, 2011) under elevated (90°C) temperature due to polymerization reactions. This dictates, that for unstable fuels, maximum residence time of the fuel in the heater must be short enough in order to avoid excessive viscosity increase during fuel conditioning phase.

- **Point of thermal decomposition.** Thermal decomposition occurs at high temperatures (above 350°C) (Altin and Esser, 2004) and begins with decomposition to elemental hydrogen, forming coke-like material. Especially in fuel system, care must be taken to avoid local temperatures beyond this point (i.e. on heating surfaces). Thus, sufficiently short residence time on heating elements should be achieved in order to avoid carbon buildup due to fuel decomposition. Carbon buildup is also affected by sulphur content in the fuel which can form metal sulphides which act as a catalyst at coke formation (Altin and Esser, 2004).

- **Fuel evaporation curve.** The overall temperature of the fuel should be kept below the point where loss of volatile fuel components occurs. As this point can be very low (often below 50°C), the maximum allowable temperature can be elevated by pressurizing the complete fuel system. As the fuel system pressure is always higher than combustion chamber pressure, explosive boiling may occur in the vicinity of fuel injector, which should be taken into account by closely monitoring combustion process through emission measurements and inspection of near-nozzle surfaces.
2.1.2 Influence of heating value, density and stoichiometric ratio on fuel system design

Heating value and density determine volumetric energy content of the fuel (Eq. 4) and thus flow capacity of the fuel supply system for particular engine. Volumetric fuel flow equals

\[ VHV \left[ \frac{MJ}{L} \right] = LHV \left[ \frac{MJ}{kg} \right] \rho \left[ \frac{kg}{L} \right] \]  

where \( VHV \) represents volumetric heating value, \( LHV \) lower heating value and \( \rho \) density of the fuel.

It is shown on the example of the analyzed fuel that density of the fuel can significantly deviate from the one of diesel fuel and thus \( VHV \) can be higher despite of lower \( LHV \) as is the case for TPO.

In terms of heating value and stoichiometric ratio, it has to be noted that the major parameter in internal combustion engines is stoichiometric mixture energy content. This parameter limits the power output of the engine and is far more important than heating value of the fuel. In case of low heating value and low stoichiometric ratio of the fuel, the mixture energy content can often be comparable to conventional fuels, as lower amount of air is required for stoichiometric combustion. However, this parameter is relatively unimportant in fuel system design and volumetric heating value is still the leading parameter when designing the fuel system.

2.2 Fuel Injection system

When designing a fuel injection system similar constraints apply as for the fuel supply system. Therefore, viscosity of the fuel is again one of the main drivers for selection of the system configuration. Generally, two types of injection nozzles are used in internal combustion engines, pressure and air-assisted atomizers.

2.2.1 Pressure atomizers

In pressure atomizers the energy required for break-up of continuous liquid jet is supplied by the liquid pressure, which is further converted into high fuel velocity at the nozzle discharge orifice. The relative velocity that occurs between the stagnant air and liquid jet, promotes the Kelvin-Helmholtz instabilities. These droplets then further break-up in accordance to Rayleigh-Taylor break-up due to aerodynamic forces acting in the opposite direction of their movement. The empirical correlation between fuel properties and Sauter mean diameter of droplets for most commonly used pressure swirl
atomizer can be described with equation 6 (Wang and Lefebvre, 1987).

\[ SMD = A \left( \frac{\sigma v^2}{\rho g \Delta p^2} \right)^{0.25} \left( t_l \cos \left( \frac{\theta}{2} \right) \right)^{0.25} + B \left( \frac{\sigma \rho_l}{\rho g \Delta p} \right)^{0.25} \left( t_l \cos \left( \frac{\theta}{2} \right) \right)^{0.25} \]  

(2)

where \( \theta \) is spray cone angle, \( t_l \) is liquid film thickness, \( \rho_l \) density of the fuel, \( \rho_g \) density of the medium in the combustion chamber, \( \sigma \) surface tension, \( \Delta P \) nozzle pressure difference and \( \mu_l \) dynamic viscosity. A and B are constants, describing nozzle geometry.

The drawback of pressure atomizers is the need for very high pressures and small fuel passages to convert the pressure to high velocity. Furthermore, the fuel flow control in pressure atomizers is strongly linked to the fuel pressure, thus the atomization quality is proportional to the fuel flow (Durdina et al, 2014). Therefore this type of injectors is most suitable for narrow range of mass flows and thus limits variations in fuel delivery rate.

### 2.2.2 Air-assisted atomizers

In this type of atomizers, energy for fuel jet break-up is additionally supplied by high pressure air-stream promoting the growth of instabilities. The spray break-up therefore depends on the detailed construction of the atomizer and is thus hard to find a unified correlation between fuel properties, atomizer geometry and droplet size spectra. For air-assisted atomizer this correlation was proposed by (Rizk and Lefebvre, 1984).

\[ \frac{SMD_{RLF}}{D} = 0.48 \left( \frac{\sigma_l}{\rho_A v^2 D} \right)^{0.4} \left( 1 + \frac{m_{lA}}{m_{l}} \right)^{0.4} + 0.15 \left( \frac{\mu^2}{\sigma_l r_l D} \right)^{0.5} \left( 1 + \frac{m_{lA}}{m_{l}} \right) \]  

(3)

where SMD is Sauter mean diameter, \( \mu \) is dynamic viscosity, \( \sigma \) surface tension, \( \rho \) density, \( v \) is relative velocity between air and liquid, \( A \) denotes air and \( L \) denotes liquid, \( D \) is liquid jet diameter.

The benefits of air-assisted atomizers are relatively low required fuel pressure and relatively large fuel delivery orifice cross section. This makes air-assisted atomizers more suitable for atomization in applications where variable fuel flows are required as atomization quality is relatively insensitive to fuel jet velocity within a large range of mass flows. Furthermore, twin-fluid atomizers are particularly suitable for fuels with high viscosity (Letho et. al, 2014), (Seljak et. al 2012) and (Sallevelt et. al 2014), as SMD is nearly independent of fluid viscosity up to 120 mPa·s and is dominated by the influence of air to fuel ratio and air pressure (Li et al, 2013).

### 2.2.3 Thermal protection

The thermal protection is important from the aspect of fuel thermal decomposition on the nozzle surfaces. Since over 90% of heat transfer to the nozzle comes from radiative component of gaseous bi-polar combustion products (CO₂, H₂O) (Lefebvre and Balal, 2010) it is essential to protect the frontal surface of the nozzle to reduce the in-nozzle temperature. The main factors determining the need for thermal protection are:

- Decomposition temperature: is the ultimate temperature that the fuel can reach before it exits the nozzle in order to avoid carbon buildup, which occurs due to thermal decomposition of the fuel on the nozzle surfaces. This can severely deteriorate spray pattern and lead to significant combustion related problems.
Viscosity and consequent preheating: higher fuel temperature reduces the ability of the fuel to act as a heat sink since it reaches the critical temperature faster due to the offset in nozzle intake temperature.

Oxygen content: oxygen content reduces the stoichiometric ratio of the fuel, whereas in addition high oxygen content promotes early mixture formation. Considering the additional air added to the fuel stream in the mixing chamber of airblast atomizers with internal mixing, the rich flammability limit could be reached in close proximity of the nozzle which can further increase heat radiation to the nozzle.

Primary air temperature in turbine engine: in case of high air temperature flowing into combustion chamber (i.e. in recuperated microturbines), the thermal loading on the nozzle is increased due to heat transfer to the frontal surfaces of the nozzle and results in higher in-nozzle temperatures (Seljak et al., 2014c).

To reduce the radiative heat transfer to the nozzle mixing chamber, thermal protection coating based on aluminum oxide was applied to the swirl-air nozzle in the applied experimental turbine. To assure that the applied coating has no influence on nozzle discharge flow field, the frontal surface of the original nozzle was modified through the iterative process. The 2 mm of frontal surface was then removed in order to make space for thin layer of coating which is then applied in two stages with conditioning between the application phases. Figure 4 and Figure 5 show original and altered nozzle frontal surface.

Figure 4. Swirl-air nozzle.

Figure 5. Thermal protection on swirl-air nozzle.
2.3 Combustion chamber

The experimental system includes a single can annular combustion chamber of an aircraft engine with a modified downstream part to yield an axis-symmetrical exhaust. The primary-to-secondary air ratio is around 30%–70%. The typical operating conditions for the reshaped can with reference diesel fuel are presented in Table 2. Combustion chamber with casing and intake/exit openings is shown in Figure 6.

Table 2. Typical operating conditions of the combustor.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Conditions at turbine inlet temperature (TIT) = 850°C</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air flow</td>
<td>0.207 kg/s</td>
</tr>
<tr>
<td>Fuel flow</td>
<td>0.003897 kg/s</td>
</tr>
<tr>
<td>Primary zone air flow</td>
<td>0.0621 kg/s</td>
</tr>
<tr>
<td>Air/Fuel ratio</td>
<td>53.12 kg/kg</td>
</tr>
<tr>
<td>Global excess air ratio</td>
<td>3.66</td>
</tr>
<tr>
<td>Primary zone excess air ratio</td>
<td>1.1</td>
</tr>
<tr>
<td>Stoichiometric ratio</td>
<td>14.5</td>
</tr>
</tbody>
</table>

It is evident from Table 2, that combustion in the primary combustor zone is roughly stoichiometric. When fuels with lower stoichiometric ratio are used in this design of combustion chamber, ideally, the primary-to-secondary air ratio should be adjusted, to maintain sufficiently high temperatures in primary zone. This can be helpful, as CO emissions are highly sensitive to peak combustion temperature and residence time (Kajita and Dalla Betta, 2003). Besides adjusting the air quantities, measures that omit wall-cooling air by blocking the wall-film intake ports were already tested and were also proved as a successful adaptation (Beran, 2011).

2.4 Exhaust heat regeneration
Aforementioned high primary air temperatures can efficiently be achieved by exhaust heat regeneration. The flexible design of the experimental test rig allows for on-line cycle change, i.e. switch from simple to regenerative Brayton cycle during operation of the engine. Thus, testing of different RDF is possible in two distinct environments within the same experimental apparatus. The flow path is presented in Figure 1.

The position of the continuously controlled diversion valve determines the amount of exhaust heat recuperated to primary air, thus the temperature can be varied within the interval between compressor discharge temperature (around 200 °C) and maximum achievable primary air temperature (in excess of 400°C).

2.5 Control strategy

Although, several options are possible for the lead control parameter, usually, turbine inlet temperature (TIT) is selected as a reference being the most important control parameter in turbine engines as it limits the thermodynamic potential of the working fluid and directly influences the efficiency of the cycle. The desired TIT is usually achieved by altering equivalence ratio (EQR), which is done by altering:

- fuel pressure,
- atomizing air pressure,
- mechanical load.

As the study focuses on the combustion and emission characterization of the fuel, it is beneficial to preserve simplicity of the system and high level of controllability or parameter variability. Therefore, the mechanical load is simulated by a throttle valve in the exhaust stream in order to obtain a more direct control over the operational parameters Therefore, two different approaches are possible to vary the power consumption at the throttle valve:

- by altering throttle valve opening angle and influencing the flow resistance coefficient (further denoted as TH strategy),
- by increasing fuel mass flow and consequently increasing turbine power resulting in higher air flow and generating larger pressure drop flow over throttle valve in a fixed position (further denoted as MF strategy).

3. OVERALL SYSTEM RESPONSE

Testing of the system was performed with aforementioned fuels, presented in Table 1. During operation, all relevant parameters were measured and post processed to obtain the thermodynamic data on the engine operation. Furthermore, during thermodynamic characterization, CO, THC and NOx emissions were measured to provide an insight into combustion process. The interpretation of emission trends was based on fuel properties and thermodynamic state of the working fluid. The overall results are presented in Table 3, where parameters and adaptations of the MGT which were defined as sufficient to comply with the objectives on stable and efficient operation are presented. The decisions were based on obtained data which is presented and discussed in the following sections.
Table 3. Test conditions.

<table>
<thead>
<tr>
<th>Minimum requirements of the system</th>
<th>Fuel</th>
<th>Liquefied wood</th>
<th>Tire pyrolysis oil</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fuel system</td>
<td>Preheating</td>
<td>Conventional</td>
<td></td>
</tr>
<tr>
<td>Injection system</td>
<td>Air-assisted</td>
<td>Pressure-swirl</td>
<td></td>
</tr>
<tr>
<td>Nozzle thermal protection</td>
<td>Required</td>
<td>Not necessary</td>
<td></td>
</tr>
<tr>
<td>Startup/shutdown</td>
<td>Dual fuel system (startup/shutdown with diesel fuel)</td>
<td>Single fuel system</td>
<td></td>
</tr>
<tr>
<td>Primary air temperature</td>
<td>Operation in regenerative cycle</td>
<td>Operation in simple cycle</td>
<td></td>
</tr>
<tr>
<td>Control strategy</td>
<td>Both possible</td>
<td>Both possible</td>
<td></td>
</tr>
</tbody>
</table>

Operational parameters

<table>
<thead>
<tr>
<th>Fuel temperature</th>
<th>100°C</th>
<th>20°C</th>
</tr>
</thead>
<tbody>
<tr>
<td>Turbine inlet temperature</td>
<td>750 , 800 and 850°C</td>
<td>750 , 800 and 850°C</td>
</tr>
<tr>
<td>Primary air temperature</td>
<td>450, 480 and 506 °C</td>
<td>182, 175 and 170°C</td>
</tr>
<tr>
<td>Control strategy used</td>
<td>Variation of fuel mass flow</td>
<td>Variation of throttle valve angle</td>
</tr>
</tbody>
</table>

| Denotation | LW-100-R-MF | TPO-20-S-TH |

The minimum system requirements and required operational parameters are strongly influenced by fuel type. Thus, more complex MGT is required to successfully obtain stable combustion of LW, while with TPO, a relatively simple system can be used with simple cycle, pressure atomization and no preheating. The exact choice of the MGT type might be application specific and for TPO, more complex system to obtain higher efficiencies can be used, however for LW, simple cycle setup with pressure atomization

3.1 Thermodynamic response

The system was equipped with pressure transducers and thermometers on characteristic points in a turbine engine in simple and regenerative cycle. In simple cycle, the characteristic points are presented in Table 4. The positions of sensors are indicated in Fig 1.

Table 4. Characteristic points for measuring the thermodynamic potential of the working fluid.

<table>
<thead>
<tr>
<th>Position</th>
<th>Measured quantities</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Compressor intake</td>
</tr>
<tr>
<td>2</td>
<td>Compressor discharge/ recuperator intake</td>
</tr>
<tr>
<td>3</td>
<td>Recuperator discharge/ combustion chamber intake</td>
</tr>
<tr>
<td>4</td>
<td>Combustion chamber discharge/ turbine intake</td>
</tr>
<tr>
<td>5</td>
<td>Turbine discharge/ throttle valve intake</td>
</tr>
<tr>
<td>6</td>
<td>Throttle valve discharge/ recuperator intake (in regenerative cycle) or exhaust to atmosphere (in simple cycle)</td>
</tr>
<tr>
<td>7</td>
<td>Recuperator discharge (in regenerative cycle)</td>
</tr>
</tbody>
</table>

In Figure 7, temperatures and pressures are presented for characteristic points along the air and fuel path for both test cases. With LW, preheated to 100°C and in regenerative cycle mode, the data was
measured for three different TITs. Measurements with TPO were performed at fuel temperature of 20°C. The absence of exhaust heat regeneration is visible in points 2-3, where only slight elevation of the temperature is visible. The minor temperature increase through the recuperator is a consequence of slight leakage of exhaust gas diversion valve. The working fluid at point 6 is discarded to the atmosphere, thus the temperature at the end of the cycle is significantly higher than in regenerative mode with LW.

**Figure 7.** Temperatures and pressures measured along the flowpath for different TIT values and a) tyre pyrolysis oil at 20°C in simple cycle mode and with throttle valve control and b) liquefied wood at 100°C in regenerative cycle with mass flow control.

It was also found out that, the control strategy has relatively little influence on the combustion process of the fuels, thus both approaches can be used, i.e. throttle valve angle variation and fuel flow variation as presented in Figure 8. Both strategies influence the overall equivalence ratio and are used here with intention to demonstrate the capability to control the system in either way. However a combination of both strategies might prove useful if detailed analysis of turbocharger would be required as the interrelation of throttle valve angle and fuel flow may allow to maintain constant turbine speed while at the same time varying turbine inlet temperature.

With TPO, the data was collected at a fixed fuel flow rate, while EQR was altered by reducing the air flow through the engine. This is visible through notably lower pressures in points 2-4 and lower temperature drop in points 4-5 which indicate lower available power for the turbine.

With LW, control of TIT was done through fuel mass flow variation, where air flow increased accordingly. This correlation is influenced by the larger pressure drop over the throttle valve and results in lower equivalence ratio at higher fuel flows, resulting in altered TIT. In case of TPO, the TIT control was done via throttle valve angle. The valve angles measured from fully open position for achieving TIT 750, 800 and 850 were 0°, 20° and 28° respectively. In this case, the fuel flow is maintained constant, and airflow reduces with increasing pressure drop over the throttle valve. It is also noticeable from Figure 8 that TPO requires roughly half of the flow of LW to maintain the same operating parameters at fully open throttle valve (TIT=750°C). This is in line with heating values of the fuels presented in Table 1.
3.2 Emission response

The emissions values of CO, THC, NOx are presented in Figure 9. Emission species were measured in NDIR, THC and CLD cell respectively. Values are corrected to 15% O2 to remove the influence of different EQR at different TIT (Seljak et al, 2014c). The measurements were performed with LW in regenerative cycle as this combination is necessary to obtain stable operation and with TPO in both thermodynamic cycles to demonstrate the influence of thermodynamic cycle while using the same fuel type. When comparing both fuels and their different properties, very similar emission values were measured with the LW and the TPO apart from the CO emissions at low temperatures, which are notably higher for the LW than for the TPO.

Different emission trends detected in Figure 9 between the TPO and the LW can be explained by further taking into account presented fuel properties and utilized thermodynamic cycle. For combustion of LW in regenerative cycle mode, the trends are exhibiting high sensitivity to TIT, which is in line with properties of the fuel, namely high viscosity and high content of aromatic hydrocarbons which originate from lignin decomposition during liquefaction. Increasing TIT results also in increasing pressures and temperatures of primary air. These high temperatures along the air-path, result in faster mixture formation. This helps to overcome the drawbacks of high viscosity and high aromatic content of LW by promoting the evaporation of droplets and ensures sufficient residence time of prepared combustible mixture on high temperatures to attain high conversion efficiency. Contrary, high CO and THC emissions at low TIT indicate impaired combustion process being mainly driven by too low temperatures in combustion chamber. Therefore it is recommended to lower TIT (below 750°C). By further reducing TIT, excessive amounts of CO are expected possibly accompanied by high THC emissions and fuel wall impingement which can lead to residue formation and damage on combustion chamber lining. When investigating the trends of NOx emissions, slight influence of TIT is also visible. This sensitivity of NOx emissions on TIT is a consequence of promoted thermal mechanism of NOx formation. Additionally, NOx emissions are also a consequence of fuel bound nitrogen (FBN) conversion to NOx. However, the influence of TIT and consequently EQR on quantity of NOx emissions, obtained from FBN is not visible here due to correction to 15% O2.

It is also worth noting that fuel preheating temperature plays a significant role in reduction of viscosity. This leads to smaller fuel droplets which shortens their heat-up and evaporation period. While this is beneficial for lowering CO emissions, while having almost no influence on emissions of NOx as presented in Figure 10.

Contrary to emission trends of the LW, TPO exhibits lower CO and THC values in simple cycle mode as well as in regenerative cycle mode as shown in Figure 9. This can be attributed to fuel properties,
notably lower viscosity. Significantly lower temperatures in primary zone of combustion zone when operating the experimental turbine in simple cycle mode still provide sufficient spray evaporation rate and sufficiently fast reaction rates even with the fuel at lower temperature (20°C in comparison to LW with 100°C). CO emissions are also almost insensitive to TIT which indicates that TPO can be used at lower TIT than LW without the risk of excessive emissions or fuel wall impingement. Lower TIT might as well reduce the efficiency of the engine, however this can increase durability due to lower thermal loading of components. THC emissions are almost unnoticeable due to almost complete dissociation of the fuel molecules in primary zone of combustion chamber. NO\textsubscript{x} emissions are higher than with LW which can be attributed to higher nitrogen content in the fuel and are insensitive to TIT which indicates that simple cycle mode slightly reduces the thermal component of NO\textsubscript{x}. In regenerative cycle, the impact of increased primary air temperature on emissions is mostly visible through minor increase in NO\textsubscript{x} concentrations which can be attributed to promoted thermal mechanism. Also, slight reduction in CO emissions when operating in regenerative mode can be observed with the reasons being the same as for LW – increased evaporation rate of the droplets and faster mixture formation allowing for longer residence time of the mixture on high temperatures.

![Figure 9. Emissions of CO, THC and NO\textsubscript{x}](image)

![Figure 10: Influence of LW preheating temperature on CO and NO\textsubscript{x} emissions.](image)

When trying to compare the upper CO and NO\textsubscript{x} with those obtained in externally fired energy recovery systems, significant influence of energy recovery system architecture and various operational parameters can be observed. However, the most suitable comparison with upper data is by comparing the combustion of RDF in fluidized bed (Duan et al, 2013), where peak CO emissions...
from RDF are below 50 ppm and NO\textsubscript{x} emissions are below 100 ppm for RDF. These values are comparable to emissions of LW and TPO in experimental microturbine engine, although to obtain this level of CO emissions with LW, high TIT and PAT are required.

### 3.3 Time-resolved thermodynamic and emission data

The possibility to control primary combustion air temperature (PAT) is of significant importance in experimental system as some fuels exhibit relatively poor combustion ability at certain air temperatures as already discussed above. The significance of PAT is once again presented in Figure 11 and Figure 12, where stability of operation with liquefied wood is assessed by the time resolved CO, THC and NO\textsubscript{x} emissions and thermodynamic data. In the following examples, the main control parameters, (i.e. fuel flow, throttle valve position, exhaust gasses diversion valve position and atomization air) flow were fixed to clearly expose impact of the primary air temperature. Variations of measured values are therefore solely a consequence of spontaneous mixture formation and combustion-related phenomena being feed-back into the system via the turbine-compressor coupling.

In Figure 11 operating parameters are presented at low PAT, which is obtained by operation at fully bypassed exhaust heat exchanger. PAT is in the range just below 200°C, fuel flow is maintained at 41 kg/h and fuel temperature at 105°C. These parameters result in TIT in the vicinity of 800°C. The time-resolved emission and air mass flow traces clearly indicate the instability of the combustion process. This is at first visible through CO and THC emissions, which are spontaneously changing for an order of magnitude. High concentrations of these two species are indicating severely impaired combustion of the LW, which is most likely a consequence of significant fuel impingement on the combustion chamber walls as indicated in (Seljak et al, 2014c), where it was shown that large amount of polymerized fuel deposits form on combustion chamber walls with low PAT and relatively clean surfaces are obtained with high PAT. The energy balance of the combustion process in the presented time interval also indicates large quantities of unburnt fuel, which results in lower pressure and air flow due to lower enthalpy flow of the exhaust gasses. Time traces of the addressed parameters are not fully aligned due fuel deposit build-ups and burn-offs from the walls of the combustion chamber. Thus, the peak CO values correspond to build-up formation and consequent incomplete combustion, followed by increase in chamber pressure and airflow, which corresponds to burn-off of deposits.
In Figure 12, the operation parameters are fixed at the same values as in the above case, apart from PAT, which is in this case fixed at 450°C and fuel flow, which is in this case fixed at 17 kg/h. To obtain a similar TIT (800°C), the energy balance dictates that the fuel flow is reduced as significant amount of energy is delivered in combustion chamber with primary air heat and thus higher PAT. Time-resolved traces in this case are significantly more stable. This is a consequence of high PAT which allows for higher evaporation rate and subsequent faster mixture formation, which is additionally promoted by higher velocities due to higher volumetric flows. Fuel impingement on the walls of the combustion chamber is thus inhibited preventing formation of the fuel deposits. Apart from these benefits, there regenerative cycle is characterized by an increase in NOx emissions which is a consequence of higher temperatures which promote the thermal mechanism of NOx formation.

Results in Figure 11 and Figure 12 indicate that PAT up to 200°C cannot support sufficiently fast mixture formation for fuels with similar properties as LW, but it is possible to obtain a stable combustion with the same setup if regeneration of primary air is employed where temperatures reach high values. This is a clear indication that for successful utilization of LW, system with regenerative cycle turbines should be selected.

Contrary to upper beneficial influence of regeneration on combustion stability, the negative influence of high PAT is manifested through high thermal loading of the fuel nozzle due to radiative and convective heat transfer from primary air. This results in carbon deposit formation as a consequence of fuel coking on high temperature surfaces of the nozzle. To avoid this, thermal protection, discussed in section 2.2.3 was developed. The efficiency of thermal protection is presented in Figure 13, where examples of nozzle deposits are shown for different operating conditions. Benefits of thermal protection are clearly visible as deposit formation is significantly reduced at high primary air temperature and the use of thermal protection. Efficiency of the thermal protection is sufficiently high.
to yield lower deposit rate compared to the case of low primary air temperature and absence of thermal protection. The brown colored deposits originate from impurities in the fuel, namely ash, similarly as observed in other studies (Prussi et al., 2012) and do not indicate thermal decomposition of the fuel on the nozzle surfaces. Contrary to black deposits which are a consequence of fuel coking, ash deposits originate from combustion process itself and do not affect the atomization process to such an extent as carbon deposits.

Figure 13: Benefits of thermally protected nozzle. Left – operation with low primary air temperature w/o ceramic coating, Center – operation with high primary air temperature w/o ceramic coating, Right – operation with high primary air temperature w/ ceramic coating.

5. CONCLUSIONS

A methodology to evaluate the extent of required adaptations to achieve stable, efficient, durable and low exhaust emitting operation of the microturbine engines while using RDFs was proposed in the paper. The methodology of testing provides the vital information on:

- preheating system with regards to viscosity, density, heating value and chemical stability of the fuel,
- injection system and nozzle suitability with regards to chemical stability, carbon residue and viscosity of the fuel,
- suitability of different thermodynamic cycles considering fuel combustion properties,
- assessment of combustion chamber conditions for supporting low heating value fuels
- assessment of combustion chamber conditions for supporting fuels with high viscosity.

The presented methodology is also suitable for finding the correct operating point for different fuels and pinpointing the weak spots in engine operation prior to demonstrator development. The defined design constraints for tested fuels dictate the required complexity of MGT system and can be summed into the following:

- LW requires fuel preheating system or other measures for viscosity reduction and twin-fluid nozzle while TPO can use a conventional fuel system and pressure-swirl nozzle.
- Volumetric energy density of LW in the fuel does not significantly influence the combustion process, since energy content of stoichiometric mixture is comparable to conventional fuels.
- Combustion of LW in simple Brayton cycle is unstable and regenerative Brayton cycle or
primary air preheating is required to avoid deposit build-up on combustion chamber walls. Stable combustion of TPO is obtained also in simple cycle and with low primary air temperatures.

- Thermal protection of the nozzle is required for LW due to fuel preheating and high temperature of primary air in regenerative cycle. This is not necessary for TPO.
- High TIT is beneficial for CO and THC emission reduction when using LW. With TPO, this effect is much less pronounced.

Analyses thus revealed that TPO could be used even in simple MGT setups without large adaptations of the system, whereas data obtained for LW indicates that this fuel requires highly complex MGT system with several adaptations to obtain stable combustion process.

AKNOWLEDGEMENTS

The authors acknowledge the support from:

- Ministry of Education, Science, Culture and Sport of the Republic of Slovenia through the contract no. 3211-10-000057 (Centre of excellence Polymer Materials and Technologies)
- University of Ljubljana through “Innovative scheme of co-funding doctoral studies” (no. 323)
- Slovenian Research Agency through contract L2-5468

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