Combustion and emission formation phenomena of tire pyrolysis oil in a common rail Diesel engine

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ABSTRACT

A pure tire pyrolysis oil produced from waste tires was utilized in a modern 4-cylinder, turbocharged and intercooled, automotive Diesel engine. Due to its low cetane number, cetane improvers, external energy addition or increased compression ratios are generally required for its use in Diesel engines. Successful utilization of pure tire pyrolysis oil is also achievable with the addition of pilot injection but limited to mid- to high-load operating range. The first objective of the present study is therefore focused on further extension of the operating range towards lower loads by novel combined application of the exhaust gas recirculation and tailored main injection strategy. As the second objective, the article provides for the first time an in-depth analysis of the particulate emissions of the tire pyrolysis oil measured with two different methods. In this area it identifies and reasons challenges related to determination of the particulate emissions for alternative fuels. The original contribution of the presented approach thus arises from the holistic assessment of interactions between the exhaust gas

recirculation ratios, injection parameters and combustion as well as gaseous and particulate emissions formation phenomena.

KEYWORDS

Tire pyrolysis oil, Compression ignition engines, Exhaust gas recirculation, Injection strategy, Gaseous emissions, Particulate matter.

1. INTRODUCTION

Increasing fuel demand on one side and a growing problem of waste disposal on the other drive the attentions of researchers to seek for new, clean and efficient waste-to-energy technologies. About one billion waste tires are produced every year over the globe [1] and many of them are landfilled. Since they are non-degradable material and their thermo-mechanical properties prevents their direct reuse, their high calorific value (35-40 MJ/kg) [2] as well as considerable amount of carbon black makes them a good feedstock for fuel production.

Pyrolysis represents an environmental friendly and efficient way to transform solid wastes into fuels with acceptable chemical and physical properties for the use in the internal combustion engines [3]. The most suitable pyrolysis subtype for conversion of waste vehicle tires, is vacuum pyrolysis [4], by which also tire pyrolysis oil (TPO) used in this study was produced. Since the main distillation properties of the TPO are similar to commercial Diesel fuel (D2) [5], compression ignition engines stand out as being potential power units for the use of the TPO.

Most of the existing research with TPO was done on non-automotive engines. Authors in [1] and [3] were running tests on a single cylinder 0,4 l, air cooled engine, in [6] tests were performed on a single cylinder water cooled engine and in [7] 1.5 l single cylinder engine was

used. Since analysed engines have limited possibilities of control strategy adaptations those results can, in general, not be properly transferred to modern engines. In addition, there is a limited number of studies found in the literature on utilizing TPO in modern multi-cylinder automotive compression ignition (CI) engines. Authors in [8] were running tests on a 4-cylinder, 3.3 l, marine engine with mechanically controller direct injection, authors in [9] were using 6-cylinder, 6.9 l, heavy duty engine with the same direct injection system and in [2], tests were performed in a 4-cylinder, 2 l, light duty engine with common-rail fuel system. However, none of listed studies successfully utilize pure TPO on modern multi-cylinder automotive CI engine.

Selected studies indicate that because of its low cetane number, TPO must be mixed with the Diesel fuel, Jatropha methyl ester or cetane improver in order to be used in Diesel engines. Authors in [1] were using several D2-TPO blends from 5% to 100% TPO, where diesel like combustion was not achieved at higher TPO concentrations. In [2] authors were running tests with only 5% TPO blend, while in [3] authors have stated that over 40% TPO, added in the D2, makes engine behave extremely irregular. In [8] only up to 10% blend was used in the study and in [10], up to 75% low sulphur TPO fuel were used. In both studies, where TPO was blended with Jatropha methyl ester [11,12], maximum concentration of the TPO fuel in the tested blend was limited to 80% in [11] and up to 50% in [12]. In [6] authors were using diethyl ether as a cetane improver to increase the cetane number of TPO fuel.

Other approaches to overcome issue of low cetane number suggest increasing the temperature at start of fuel injection to achieve sufficient temperature required for auto ignition of the fuel. This was achieved either through increasing the intake air temperature by external heater [13] or by increase of engine compression ratio to 22-24 [7]. The first approach deteriorates total

energy efficiency and thus applicability of the system, while the second approach cannot be realized with highly boosted engines, which yield high power densities and high effective efficiencies. Both methods also results in higher in-cylinder temperatures and thus higher nitric oxides (NO_x) emissions. In the foremost study [14] a pure TPO was utilized in a turbocharged multi-cylinder engine without any of the aforementioned aids. Results indicate that TPO can be efficiently used in turbocharged non-intercooled CI engines at high loads, which opens its use in power generation. Although above mentioned results are promising, it is suboptimal to utilize the engine without intercooler as it decreases engine efficiency and increases temperature loading of the components and NO_x emissions. In another study [15], it was elaborated that addition of pilot injection (PI) and tailoring of the PI strategy can optimize thermodynamic parameters of turbocharged intercooled Diesel engine, which allows for successful mid- to high-load operation under the use of pure TPO.

To further enlarge the operation range while utilizing the pure TPO without the addition of cetane improvers and without any external heat addition, the first objective of this study is focused on further extending operation of a modern turbocharged and intercooled Diesel engine to low loads. To achieve this objective an in depth analyses of impacts of different exhaust gas recirculation (EGR) ratios and different injection parameters on the combustion and emission formation phenomena of the pure TPO are performed for the first time. These results are benchmarked against the data obtained for the Diesel fuel. This holistic assessment provides guidelines and pinpoints suitable control strategies for efficient utilization of the pure TPO. The novelty of the proposed approach is schematically presented in the Figure 1, which summarizes on the left side existing methods for TPO utilization in Diesel engines and on the right side the proposed novel approach.

The second objective of this paper is focused on analyses of particulate emissions. In this area, the article provides, to the best of authors' knowledge, a foremost in-depth analysis of the particulate emissions of the TPO measured with different methods: the Micro Soot Sensor and the Gravimetric Filter Module. This analysis identifies and reasons significant differences between both methods when determining particulate mass for the alternative TPO, which opens many challenges that are discussed in the Results.

Figure 1: A schematic diagram of the TPO utilization methods in Diesel engines.

2. FUEL PROPERTIES

The TPO used in this study was produced by vacuum pyrolysis method, which has the potential to produce a low sulphur fuel with reasonably high yield from tires [16]. Waste tires were cut into pieces (mean size 100 mm x 100 mm) where steel wires and fabric fibres were previously removed. Pyrolysis process was performed between 600°C and 700°C and the retention time was 60 min. Fraction between 190°C and 350°C was used for further studies.

Properties of the TPO and Diesel fuel were analysed with standard methods and results are presented in Table 1. Utilized commercial Diesel fuel complies with the specifications of the

SIST EN-590 standard. TPO features higher density and lower heating value (LHV) on mass basis, which results in slightly higher volumetric energy density. Major properties (Table 1) as calorific value, viscosity and density are comparable to those of D2 while sulphur content of the tested TPO is significantly higher, which restricts its use in road vehicles. If required, this could be solved by reduction of the sulphur content in the fuel or with desulfurization of exhaust gases as described in [14].

Another important difference between the TPO and the D2 arises from significantly lower cetane number of the TPO, which is reflected in its poor ignition properties thus making its use in CI engines a lot more challenging. It seems generally accepted that the cetane number of TPO is certainly below 30. In [7] authors claim that it was estimated that the cetane number values of TPO are typically between 5 and 25, while in [11] it was reported that cetane number is in the range 25-30. However, the authors in [17] claim that cetane number of 44 could be achieved for the TPO before entering desulfurization process. On the other hand conventional Diesel fuels have much higher cetane number of at least 51, as set in the European standard SIST EN-590 [5]. Difference between D2 and TPO cetane number was confirmed by the calculated value of cetane index (Table 1), which may be used to estimate the cetane number of the fuel [18]. Although there is discrepancy between cetane number and calculated cetane index, it can be concluded that TPO features lower reactivity compared to Diesel fuel.

Table 1: Main properties of TPO and Diesel fuels

Property\Fuel	TPO	Diesel
Density [kg/L]	0.92	0.83
LHV on mass basis [MJ/kg]	42.7	42.95
LHV on volume basis [MJ/L]	39.3	35.6
Energy content of the stoichiometric mixture [MJ/kg]	2.89	2.74
Aromatic content [% m/m]	39.3	26.0 [19]

Viscosity [mm ² /s]	3.22 @ 20°C	2.54 @ 40°C
C [% m/m]	83.45-85.60 [4]	87.0
H [% m/m]	9.59-11.73 [4]	13.0
N [% m/m]	0.40-1.05 [4]	/
S [% m/m]	0.96	<0.001 [5]
O [% m/m]	0.10-3.96	/
H/C ratio [/]	0.112-0.140	0.149
PAH [% m/m]	9.2	2.7
Ash [% m/m]	>0,18	<0,01[5]
FAME content [% m/m]	0	7
Calculated cetane index ¹	28.6	53.2

¹ Following SIST EN ISO 4264

3. EXPERIMENTAL SETUP

The experimental work was performed on a 4-cylinder, 4-stroke, turbocharged, intercooled 1.6 litre PSA light-duty Diesel engine (DV6 ATED4), equipped with a common-rail injection system and uncooled EGR, whose main characteristics are given in Table 2. For the purpose of this study the engine was reworked in a way to gain full control over injection timing, quantity and injection pressure as well as to allow for efficient switching between the fuels being prerequisites for the anticipated combustion optimisation study. The main characteristics of fuel injection system are given in Table 3. The engine was coupled with a Zöllner B-350AC eddy-current dynamometer controlled by Kristel, Seibt & Co control system KS ADAC. A schematic of the experimental system is presented in Figure 2.

Table 2: Engine characteristics

Engine	PSA DV6ATED4
Cylinders	4, inline
Displacement	1560 cm ³
Bore × stroke	75 mm × 88.3 mm
Compression ratio	18:1
Maximum power	66 kW @ 4000 rpm
Maximum torque	215 Nm @ 1750 rpm
Air intake	Turbocharged
Cooling system	Water cooled

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Table 3: Fuel injector characteristics

Fuel injection system	Common rail
Injector type	BOSCH 0445 110 239
Number of holes	6
Hole diameter	0.115 mm
Spray angle	149 °
Maximum IP	1600 bar

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165 For the indicated cycle analysis, calibrated piezo-electric pressure transducer (AVL GH14D)
 166 was used in combination with charge amplifier AVL MICROIFEM, connected to 16 bit, 4
 167 channel National Instruments data-acquisition system with maximum sampling frequency of 1
 168 MS/s/ch. An optical shaft encoder Kistler CAM UNIT Type 2613B provided an external
 169 trigger and an external clock at 0.1 deg crank angle (CA) for data acquisition and injection
 170 control system. Top dead centre (TDC) was determined by capacitive sensor COM Type
 171 2653. The maximum uncertainty of pressure measurement, which combines uncertainties of
 172 pressure transducer, charge amplifier and data-acquisition system, is 0.31% and maximum
 173 uncertainty of pressure measurement corresponding to crank angle was therefore 0.96% [20].

174

175 Data acquisition and injection control embedded system (denoted as NI cRIO system in Fig.
 176 1) was based on National Instruments cRIO 9024 processing unit and 9114 chassis. The same
 177 system was used for indication of in-cylinder pressure traces and engine control. The
 178 information of position and duration of individual injection was transferred through digital
 179 communication to Drivven system. Drivven system generates inlet metering valve signal
 180 (IMV signal) and injector energizing signal (IE signal) to control common rail pressure and
 181 fuel injection timings. Drivven system was connected to the PC and energizing characteristic
 182 for the injectors along with IMV parameters were set in CalView software.

183

Fuel supply systems of both fuels were completely separated in order to avoid mixing of fuels in the fuel system and allow for shorter transition times during fuel changes. Switching between the fuels was done right before the common rail pump. Fuels were placed in separate tanks with separate fuel filter and separate measuring devices. D2 mass flow was measured with AVL 730 gravimetric balance, with measuring accuracy of $\pm 0.12\%$, while TPO mass flow was measured with Micro Motion ELITE CMFS, with measuring accuracy of 0.05% . Intake air flow was measured with Meriam laminar flow meter 50MC2-6F, with measuring accuracy of 0.86% . Additionally, the engine was fitted with instrumentation for measurement and monitoring several temperatures and pressures.

The exhaust emissions were analysed and recorded using a portable exhaust monitoring equipment Horiba OBS-2200, which provides accuracy of $\pm 2.5\%$ of full scale and AVL M.O.V.E. PM, with mass flow controller measuring accuracy of $\pm 0.3\%$ and dilution accuracy of $\leq 3\%$. Exhaust samples were extracted from the tailpipe with the help of sharp-edged open-ended probe installed near the centreline of the exhaust pipe. Heated sampling pipe was used to guide wet exhaust gases from sampling attachment on the exhaust pipe to the measuring device. Horiba emission measurement device comprise non-dispersive infrared analyser (NDIR) for detection of carbon monoxide (CO) and carbon dioxide (CO₂), flame ionization detector (FID) for detection of total hydrocarbons (THC) and chemiluminescent detector (CLD) for detection of NO_x emissions. AVL M.O.V.E. PM comprise combination of two measurement principles: AVL Micro Soot Sensor (MSS) and Gravimetric Filter Module (GFM) for measurement of particulate matter (PM) mass.

Figure 2: Schematic of the experimental setup.

4. TEST PROCEDURE

Engine was always started with the D2 and after several minutes the transition to the TPO was performed. During the transition period the fuel system was flushed with the desired fuel in order to avoid mixing of the fuels in the fuel system. The fuel system has been modified in order to allow bleeding through the fuel return line during the transition period. Experiments were performed in thermally stabilized steady-state engine operating points at two different engine speeds and two different loads. The list of the analysed operation points is provided in Table 4.

In every operating point full matrix of three different main injection (MI) timings and three different EGR valve positions were tested, which gives us total of 36 variations of operating conditions operation points. The table of all operating parameters, including injection duration for PI and MI, for the 36 cases is due to space limitation presented in Appendix A. Modification of injection parameters were simultaneously performed on all four cylinders, while the injection pressure was determined on the basis of speed/load parameters and was equal as in the original engine control unit (ECU) injection strategy for selected speed/load

for hot engine at ambient conditions of standard atmosphere. Original MI timings of the ECU for selected operating points presented in Table 4 were -6, -4, -17 and -19 consequently (cases number 1, 10, 19 and 28 in Appendix A). The MI timing variation was selected on the basis of preliminary screening where it was found that larger variations lead to suboptimal combustion. Due to the difference in density between the analysed fuels, the injection duration for MI was changed to keep constant torque within desired operating point (Table 7 and 8 in Appendix A).

Table 4: Operating points with corresponding engine operating parameters

Engine speed [min^{-1}]	Torque [Nm]	Main injection timing [$^{\circ}\text{CA}$]	EGR rate [pos. %]
1500	60	-4/-6/-8	0%/25%/60%
1500	100	-2/-4/-6	0%/25%/60%
3000	35	-15/-17/-19	0%/25%/60%
3000	100	-17/-19/-21	0%/25%/60%

4.1. Thermodynamic analysis

In-cylinder pressure measurements were performed over 100 consecutive cycles for each variation of injection parameters at a sampling resolution of 0.1°CA . Representative in-cylinder pressure trace was generated via a two steps approach. First, 100 consecutive cycles of the individual operating point were averaged to eliminate Cycle-to-Cycle Variations (CCV) due to signal noise [21], whereas Chauvenet's criterion [22] was taken into account to eliminate the points that significantly deviate from the others. Second, pressure oscillations in the combustion chamber that occur as a result of partial auto-ignition of the fuel [20] were eliminated by applying low pass finite impulse response (FIR) filter. The representative pressure trace was then used as input for the Rate-Of-Heat-Release (ROHR) analysis that was performed with the AVL Burn tool [23].

4.2. Emission analysis

Emissions were measured continuously at sampling resolution of 1 s. The final result for emission values in the selected operation point was obtained by averaging over a 60 s window

after thermal stabilisation. Emission analyses were performed for basic exhaust gas species – THC, CO and NO_x. CO₂ and water (H₂O) will not be discussed here as they directly correlate to fuel consumption and carbon balance. Since the majority of fuel sulphur is converted into sulphur dioxide (SO₂) during combustion process, the emissions of SO₂ can be very well predicted by taking into account linear correlation between fuel consumption and SO₂ concentration in exhaust gasses [24].

The exhaust PM emissions were measured with a portable exhaust gas monitoring equipment AVL M.O.V.E. PM using two different methods: AVL MSS and GFM. MSS consist of photoacoustic soot sensor where light absorbing particles in the aerosol samples are periodically heated by absorption of amplitude-modulated light [25]. The heat related expansion of particle generates acoustic pressure waves that are detected by a microphone [25]. The most widely used term for light-absorbing carbonaceous aerosols is Black carbon (BC) [26]. On the other hand, GFM provides information on sum PM mass, collected on a filter over the entire acquisition period with that particular filter and is usually used to calibrate the real-time information from MSS. In this case, integral MSS real-time values in a certain acquisition interval are compared with GFM data obtained in the same acquisition interval to determine a weight factor, as

$$weight\ factor = \frac{GFM\ mass}{\int_t MSS\ signal \cdot dt}. \quad (1)$$

MSS real-time values are then multiplied with the weight factor in order to calibrate the MSS measurements.

5. RESULTS AND DISCUSSION

In the following section, combustion parameters and emissions for cases, listed in the Table 4 are presented and commented. Influence of the EGR and the main injection variation on the

combustion characteristics of the TPO and the Diesel fuel are analysed and compared in order to determine guideline for TPO utilization in the modern turbocharged and intercooled Diesel engines. In the first part of this section, CCVs are analysed, in the second part combustion parameters are presented and the third part represents emission results.

One of the tested operating points (at 1500 min⁻¹ and 100 Nm) was selected for detailed thermodynamic analysis, which was then further used for the explanation of emission formation.

5.1. Cycle-to-cycle variations

CCV of combustion parameters play important role to determine engine combustion process stability and repeatability, which influences engine performance and emissions, however higher CCV can influence also the wear of the engine. In order to analyse CCV, statistical analysis of maximum pressure for 100 consecutive cycles was performed for D2 and TPO. Operating point at 1500 rpm and 60 Nm was chosen due to its low stability in baseline case, which resembles worst case scenario for all analysed cases. Qualitative results of pressure variations are presented in the Figures 3 and 4 and quantitative results of pressure peaks in the Table 5. Operating points are denoted as: “fuel type_start of MI_EGR valve opening (i.e. D2_-4_25) where fuel indicates D2 or TPO, start of MI denotes its value related to the TDC with negative values indicating values before the TDC and EGR valve position is given in percent of its opening.

From the Figure 3, it is discernible that combustion is more stable in the case of the D2, which can be ascribed to its higher cetane number. Lower cetane number of the TPO results in longer ignition delay, which results in higher CCV due to stochastic nature of particular physiochemical phenomena influencing the ignition delay. This indicates that it is not suitable

to continuously operate the engine fuelled with the TPO at low load (i.e. 60 Nm) due to low combustion stability and high CCV. In order to overcome this issue and to enhance combustion stability the EGR was introduced.

Figure 3: Cycle-to-cycle pressure variations of 100 consecutive cycles in operating point at 1500rpm and 60Nm for cases a.) D2_-6_0 and b.) TPO_-6_0.

In the cases, presented in the Figure 4, EGR valve was 60% opened, which resulted in higher in-cylinder temperature at the start of compression phase. Higher in-cylinder temperature allows for shorter ignition delay and therefore significantly more stable combustion for the TPO case is achieved.

Figure 4: Cycle-to-cycle pressure variations of 100 consecutive cycles in operating point at 1500rpm and 60Nm for cases a.) D2_-6_60 and b.) TPO_-6_60.

To evaluate CCV variations quantitatively, Coefficient Of Variation (COV) for CCV maximum pressure was determined with the Equation (2) and presented in the Table 5. It is calculated as the ratio between standard deviation (σ) of the maximum pressure values and mean maximum pressure for 100 consecutive cycles.

$$COV = \frac{\sigma}{Mean\ Max\ Pressure} \cdot 100 \quad (2)$$

In the TPO case it can be seen that COV is 40% lower for the measurement TPO_-6_60, compared to the measurement TPO_-6_0, whereas for the D2 the difference is in range of 6%. Furthermore, the Table 5 shows that for the measurement D2_-6_0, COV is 50% lower than the one of TPO_-6_0. From the acquired results, it can be concluded that TPO combustion CCV are higher than D2 combustion CCV but it is possible to mitigate them with increasing the EGR rate. This already partly confirms the hypothesis that EGR can as well be used to increase temperature at start of injection (SOI) with similar effectiveness as avoiding the use of the intercooler. All analysed steady-state operating points from Table 5 were measured three times to analyse reproducibility of the results. Maximum pressure deviation of the mean pressure cycle at different repetitions of the same operating point is slightly higher when running on TPO (Table 5), which is also anticipated from the analyses of the COV.

Table 5: Cycle-to-cycle maximum pressure variations

Measurement	Mean max pressure [bar]	Max pressure std. Deviation [bar]	COV [%]	Max. pressure deviation of the mean cycle at different repetitions of the same op. point [%]
D2_-6_0	90.68	0.46	0.50	1.21
D2_-6_60	80.43	0.38	0.47	1.17
TPO_-6_0	90.00	0.91	1.02	2.45
TPO_-6_60	80.00	0.48	0.60	1.92

Presented results clearly indicate that introduction of the EGR additionally extends operating range with the TPO to lower loads while preserving diesel-like combustion. These results are consistent with the results presented in the study [15] that were measured on the same engine.

In [15] the lowest reported load, which was already characterized by relatively high CCV, was 80 Nm at 1500 rpm. In this study, addition of the EGR thus extended operation range to 60 Nm at 1500 rpm with much smaller CCV and thus higher combustion stability, while at 3000 rpm addition of the EGR enables diesel-like combustion at loads as low as 35 Nm.

5.2. Combustion analysis

In this section, combustion analysis is presented for the TPO and the D2 under variations of the start of MI and EGR valve position to interrelate fuel properties, control parameters and ROHR. Detailed analysis will be performed for the operating point 1500 rpm and 100 Nm. The baseline case (D2_-4_0) features the same injection parameters and injection pressure as the original series production ECU, whereas all other cases feature adapted main injection duration in order to reach constant torque (Table 7 and 8 in Appendix A). In this operating point the temperature of the D2 measured on the fuel return line right after the injector of the observed cylinder was 77°C whereas temperature of the TPO was 1°C higher.

5.2.1. Influences of the variation of the start of main injection

In this analysis EGR valve was kept closed and start of PI was kept constant at -33°CA for all presented cases in order to isolate the influences of variations of start of MI on the combustion characteristics. Injection parameters (e.g. start of energizing and duration) as well as total injected mass are presented in Appendix A. It is discernible from the Figure 5 that the variations have strong influence on pressure traces for the D2 as well as for the TPO, which is directly reflected in ROHR values.

Figure 5: a.) Pressure trace, b.) ROHR and c.) Temperature for operating point at 1500 rpm and 100 Nm at various main injection crank angles.

It can be seen in Figure 5b that in the D2 case, PI ignites at around -15°CA , whereas shifts of the MI ignition directly reflect MI timing variation. It can also be observed that the shape of the ROHR related to the MI is very similar irrespective of the MI timing. Obviously, earlier combustion results in higher in-cylinder pressures.

In the TPO case, PI ignites considerably later compared to the D2 being the consequence of the lower cetane number of the TPO, causing longer ignition delay. PI ignition thus occurs very close to the TDC where higher temperature is present and consequently pressure traces feature lower values up to the TDC for the TPO. Figure 5b also indicates that TPO features longer ignition delay period of the MI. This is besides fuel properties related also to the fact that in cylinder temperature is lower at the start of MI injection (Figure 5c) as the fuel injected

during the PI did not yet burn out. Longer ignition delay results in more intense premixed combustion and consequently higher ROHR peak values in comparison to D2 cases. Despite the fact that PI did not burn out completely at the time of MI for the TPO, it should be emphasised that introduction of PI represents a crucial mean for at least similar phasing of the ROHR of the TPO compared to the one of the D2 in a turbocharged and intercooled engine at part loads. Introduction of the PI thus represents a significant improvement compared to our previous study [14], as it allows for the use of the TPO in an intercooled engine and also at part load operation. Comparing the TPO and the D2, it can be concluded that start of MI variations influence TPO cases similarly to the D2 cases. Advancing MI results in advanced ignition of the MI and thus in higher absolute pressure values. MI timing does also not significantly affect ignition delay of the TPO because the temperature at the start of injection does not change significantly in analysed cases. Likewise the shape of the ROHR related to the MI is very similar irrespective of the MI timing for the TPO.

5.2.2. Influences of the EGR rate variations

In the previous section it was presented that shifts in MI directly transfer to the shifts of the ROHR related to the MI for the D2 and the TPO and thus it does not significantly affect ignition delay for either of the utilized fuels in the analysed engine. To gain additional control over the combustion process, it is beneficial to control also the ignition delay. This is of particular importance for the TPO because, as analysed in the previous sections, longer ignition delays render approaching Diesel like combustion with the TPO thus contributing to larger COV at lower loads. As the ignition delay of the TPO is mainly driven by chemical kinetics due to its low cetane number, it can be concluded that temperature at start of injection stands out as the most influential parameters for its control.

Higher EGR rate results in lower exhaust mass flow through the turbine and compressor consequently due to shift of both devices to lower efficiency area in lower charge pressure in the intake manifold. This is directly reflected also in the pressure traces during the compression phase, where the difference is most pronounced for the D2 (Figure 6a).

It is discernible from Figure 6b that the analysed range of EGRs, EGR rate variation does not notably render shapes of the ROHRs curves for the D2. This can be related to the fact that due to high reactivity of the D2 and due to sufficiently low EGR concentration, either increase of temperature (Figure 6c) and EGR concentration do not significantly render chemical kinetics (Figure 6b) or that in the analysed range these two effects compensate each other.

Unlike the D2, a significantly different trend is observed for the TPO. As temperature is the limiting factor due to lower reactivity (low cetane number) of the TPO, shorter ignition delays are observed for the TPO with increasing EGR rate. This indicates that increase in EGR rate has a significantly higher impact on enhancing chemical kinetics as altering of the MI timing. Despite that, shortening of ignition delay periods of the PI due to increasing EGR rate increases the temperature at the start of MI (Figure 6c), which results in reduced intensity of the premixed combustion. It can thus be concluded that introduction of the PI and the EGR allow for approaching the Diesel like combustion with the use of the TPO in turbocharged and intercooled engines at part loads, whereas timing variation of the MI allow for ROHR shifting. Temperature traces also reveal two competing effects. One is the temperature increase at start of MI due to increase in charge temperature as a consequence of introduction of hot EGR gasses. The other is increased specific heat of charge which increases with increasing EGR and causes lower temperature increase during combustion. Therefore, the maximum in-cylinder temperature is obtained at 25% EGR.

Figure 6: a.) Pressure trace, b.) ROHR and c.) Temperature for operating point at 1500 rpm and 100 Nm at various EGR valve positions.

5.3. Emissions

In this section, the exhaust gaseous species emissions and particulate matter emissions are analysed for varying load at 1500 rpm and 3000 rpm.

5.3.1. Gaseous emissions

In this section THC, CO and NO_x emission are analysed. The results of gaseous emissions are grouped depending on the engine speed, whereas in each group emissions for low and medium loads, for start of MI variations and EGR valve opening are presented for both utilized fuels. Each curve is denoted as “engine speed in rpm_engine load in Nm_start of MI in °CA after TDC”.

Operating points at 1500 rpm

In this section, the exhaust gaseous species emissions of THC, CO and NO_x are analysed for various loads at 1500 rpm.

THC emissions

Comparing THC emissions for the TPO and the D2 at various engine loads, it can be observed in Figures 7a and 7b that THC emissions are lower at higher engine loads for both fuels. This can be attributed to higher in-cylinder temperatures resulting in shorter ignition delay and therefore less time for migration of the mixture into colder parts of the combustion chamber as well as in higher conversion rate of hydrocarbons.

It can also be seen that THC emissions are higher for the TPO, which can be reasoned by chemical characteristics of the fuel. The TPO is composed of higher share of ignition resistant cyclic hydrocarbons [27] which are known by their high auto ignition temperature resulting and longer ignition delay. Delayed ignition of PI for TPO (as indicated in Figures 5b and 6b) is prolonging the premixing process, leading to migration of the injected fuel into colder parts of the combustion chamber and crevices, while at the same time increasing the mixture air-fuel ratio results in increasing the volume of the air-fuel mixture outside the flammability limits.

Variations of MI timing and EGR rate have negligible effect on THC emissions for the D2. However in the case of TPO the effect on THC emissions is non-negligible. It can be generally seen in Figure 7b that the THC emissions of the TPO at high load slightly reduce with higher EGR rate, which can be attributed to higher in-cylinder temperature at the start of MI (as indicated in Figure 6c) which confirms the effectiveness of proposed EGR introduction. Increased temperature reduces MI ignition delay and prevents the migration of

fuel vapours to colder parts of combustion chamber and to crevices. At low engine load, THC emissions for the TPO first increase when increasing EGR valve opening up to 25%, which can be explained with lower reaction rates due to the lower in-cylinder oxygen concentration, but at the same time still relatively low temperature levels. However, at 60% EGR valve opening, higher in-cylinder temperature at start of MI offsets (as indicated in Figure 6c) this effect and reaction rate is again increased due to higher temperature. In addition, with increasing temperature ignition delay of the MI becomes shorter and less fuel vapours migrate into colder parts of the combustion chamber, which results in lower THC emissions. The advantage of EGR in THC emissions reduction is quite clear, however precise fitting of EGR rate might be necessary in order to avoid negative effects of reduced charge oxygen content. Results suggest that with tailored MI timing, also the negative effect of low EGR rates on THC emissions could be reduced.

CO emissions

The CO emissions generally originate from combustion chamber areas where already reacting air-fuel mixture is diluted and cooled, which leads to local quenching of combustion reactions, and areas with low air-fuel ratio as well as from the local lean flameouts. In the case of the D2, presented in the Figure 7c, it is discernible that CO concentration is increasing with increasing EGR rate due to lower oxygen concentration in the combustion chamber. Lower global and local air-fuel ratios result in increased time needed for complete carbon oxidation as well as in larger oxygen deprived regions, giving way to generation of CO. This effect is less pronounced at higher engine loads, since in-cylinder temperature further increases, offsetting the effect of lower oxygen concentrations and leading to slightly lower CO emissions due to higher reaction rates.

Figure 7: a.) D2 THC emissions, b.) TPO THC emissions c.) D2 CO emissions d.) TPO CO emissions e.) D2 NO_x emissions and f.) TPO NO_x emissions for operating points at 1500 rpm, 60 and 100 Nm with original and shifted main injections and various EGR valve positions.

In the case of the TPO, CO values are fairly constant with different EGR variations. Here longer ignition delay (as indicated in Figure 5b) results in higher peak in-cylinder temperatures which increase the reaction rate in a certain interval of combustion process, promoting oxidation of CO. Another reason may also lie in compensating influences of higher temperature due to addition of EGR and lower reaction rate due to lower in-cylinder oxygen concentration. Furthermore, increasing EGR rate results in higher temperature at start of MI, which shortens ignition delay and reduces CO emissions, but at the same time lower air-fuel ratios result in longer combustion duration, which might again increase CO emissions, contributing to another compensating influence, which can lead to relatively steady CO concentrations over the EGR variation range.

NO_x emissions

The combustion of the TPO is characterized with slightly higher NO_x emissions, compared to the D2, which can be attributed to higher percentage of nitrogen content in the TPO (Table 1) being efficiently transformed into NO_x. Nitrogen content was later in the text denoted by fuel bound nitrogen (FBN). Generally, the conversion of FBN to NO_x has relatively low sensitivity to operating point of the engine, thus the differences between different operating points here can be mostly attributed to thermal mechanism. From the Figures 7e (D2) and 7f (TPO) it is discernible that advanced start of MI result in higher NO_x values for both fuels due to higher peak pressure rise during the initial portion of the combustion process, resulting in higher peak in-cylinder temperature. Contrary, increasing EGR rate results in lower oxygen content in the combustion zone, resulting in reduced formation rate of NO_x [28]. However, the TPO exhibits similar sensitivity to MI timing and EGR rate as the D2. Relatively constant increase in NO_x emissions characteristic for the TPO is therefore mostly a consequence of the aforementioned FBN effect. Introduction of EGR is therefore also an effective measure for reduction of NO_x with TPO which can successfully offset the negative effect of FBN in TPO.

This approach could be considered as even more appropriate as with D2, because it does not affect the CO emissions.

Operating points at 3000 rpm

At 3000 rpm (Figure 8), emissions feature similar trends to those measured at 1500 rpm. Therefore only the most pronounced differences will be discussed in this section to preserve brevity of the paper.

THC emissions

The most pronounced difference is visible for the THC emissions of the TPO at 3000 rpm and 35 Nm (Figure 8b). Here, lower in-cylinder temperatures due to lower load lead to longer ignition delays, which are in the case of late MI timing also combined with the insufficient residence time of the droplets and mixture at high temperatures, causing notably increased THC emissions. In this operating point there were also indications that non-firing cycles might be occurring as discussed in section 5.1, again leading to elevated THC emissions. In the case of low load, the use of EGR does not present an advantage in terms of THC emissions, since EGR lowers in-cylinder oxygen concentration resulting in lower reaction rate which might increase THC emissions. However relatively low effect on THC at 25% EGR indicates that this strategy could be used to influence NO_x emissions. Results suggest that for THC emissions at low load, the most important measure is to avoid late MI timing (-17) which causes high THC emissions. However, advancing the MI timing too far might again result in an increase in THC emissions due to too low temperature levels at the time of injection. With current density of the experimental matrix the results suggest that the minimum THC emissions could be obtained in the vicinity of the MI timing of -19. At high load condition (3000 rpm and 100Nm), this trend is less pronounced and THC emissions are almost insensitive to MI timing, however they do reduce slightly at highest EGR rates due to aforementioned influences of charge temperature at start of injection

Figure 8: a.) D2 THC emissions, b.) TPO THC emissions c.) D2 CO emissions d.) TPO CO emissions e.) D2 NOx emissions and f.) TPO NOx emissions for operating points at 3000 rpm, 35 and 100 Nm with original and shifted main injections and various EGR valve position.

548 *CO emissions*

549 At 3000 rpm, the CO emissions of the D2 are higher particularly at mid-load conditions
550 (Figure 8c) due to higher piston velocity and consequently reduced reaction time as well as
551 lower air-fuel ratio. At low load, higher air-fuel ratio is the main driver for reduced CO
552 emissions, which is also the reason for lower sensitivity of CO on high EGR rates.

553 At 3000 rpm, the sensitivity of CO emissions of the TPO on EGR rate is notably lower which
554 is in line with results at 1500 rpm. Only a slight increase is observed at higher load and high
555 EGR rates which is a consequence of competing influences of oxygen content and
556 temperature at start of injection. This is in line with the cetane number of the TPO, requiring
557 higher in-cylinder temperature to reduce the CO emissions.

558 *NO_x emissions*

559 The NO_x emissions are slightly higher for the operating points at 3000 rpm and 100 Nm for
560 both fuels, compared to the operating points at 1500 rpm and 100 Nm due to the higher
561 temperatures in the combustion zone, caused by higher boost pressure and lower heat loss
562 during compression and consequently higher NO_x production rate during combustion of the
563 fuel injected during the MI. On the other hand, higher air-fuel ratio at 3000 rpm and 35 Nm in
564 comparison to the one at 1500 rpm and 60 Nm contributes to notably lower NO_x emissions for
565 both fuels as it reduces the in-cylinder temperature. Correlation of NO_x emissions with EGR
566 and MI timing is unchanged in comparison to the operating points at 1500 rpm. Also, the
567 FBN content in the fuel is reflected in steady increase in NO_x emissions over the entire
568 operating range with TPO. This again confirms that introduction of EGR, accompanied with
569 tailored MI timing could visibly expand the operating range of the engine.

5.3.2. Particle mass emissions

In this section, two sets of results are presented. First the values of MSS for both fuels are shown in Figure 9a, 9b and Figure 10a, 10b at 1500 rpm and 3000 rpm. As MSS measuring principle relies mainly on highly absorptive material which is ideally composed of black carbon [10], the emissions measured by MSS are considered to be consisting mostly of soot and will be also denoted as “soot” in this section. The second set of results comprises data, measured by MSS and multiplied with the weight factor as was presented in section 4.2. The end calibration of MSS measurements is done on the basis of data from the GFM module which besides soot collects also a wide spectra of other particulate matter, including inorganic species and heterogeneous particles. Hence, these results, obtained with combined methods (MSS+GFM) will be denoted as “particulate matter” (PM) in this section. They are shown in Figure 9c, 9d and Figure 10c, 10d with the intention to highlight the fuel specific discrepancy between the two methods.

Soot emissions, measured by the MSS

As shown in Figure 9a and 9b in all operating regimes the soot emissions measured with MSS are increased at higher EGR rates for both fuels. This is caused by correspondingly lower oxygen concentrations inside the combustion chamber promoting generation of soot particles. The necessary conditions for soot generation are generally considered to be: sub-stoichiometric conditions (λ values below 0.25-0.5) [29] within an appropriate temperature window and with appropriate residence time. While the general trend of soot emissions is similar for the D2 and the TPO, absolute values are roughly two-fold higher for the TPO. There are two main reasons that support this behaviour of the TPO:

- Relatively high content of aromatic and polyaromatic hydrocarbons (PAH) in the TPO (Table 1), which are considered as soot precursors [2].

- In-cylinder conditions, supporting generation of soot by occurrence of areas with lower temperature, particularly at low engine load which supports partly reacted mixture escape to colder parts of combustion chamber which impairs soot oxidation. Soot oxidation is also impaired by low overall temperatures during expansion stroke leading to higher soot emissions.
- Possible detection of other PM beside soot, which has significantly lower, but non-negligible coefficient of absorption and thermal expansion and could contribute to soot signal, giving a false positive values.

Figure 9: Comparison of PM micro soot and PM weighted at 1500 1/min for D2 and TPO.

Focusing on variation in the MI timing (Figure 9a) a uniform influence on soot emissions is visible for the D2. Here, advanced MI timings result in higher in-cylinder temperatures (Figure 5c) due to the larger amount of the fuel that burns close to the TDC which lowers soot emissions. In the case of the TPO, the trend is not that uniform and late injection timing at low load results in low in-cylinder temperatures (Figure 9b) where TPO is subjected to larger CCV. Albeit the expected lower air-fuel ratio, which could increase soot emissions at higher load, TPO exhibits lower soot emissions at higher load due to beneficial effect of temperature, which aids at soot oxidation.

Figure 10: Comparison of PM micro soot and PM weighted at 3000 1/min for D2 and TPO

Soot emissions, measured with the MSS at 3000 rpm for both fuels (Figure 10) demonstrate similar trends as characteristic for those at 1500 rpm. While the trends are similar, the absolute values of soot mass emissions are higher at higher engine speed (Figure 10a, 10b) for all variations of operating conditions. This can be attributed to higher piston velocity and consequently shorter combustion period, causing insufficient residence time of reacting

mixture which prevents soot oxidation. The effect of higher piston velocity and lower air-fuel ratio is also pronounced when combined with further lowered oxygen concentration at highest EGR rate (60%) and higher load where soot emissions for both fuels were almost one order of magnitude higher than at 1500 rpm.

Soot emissions are again notably higher for the TPO, which is more prone to soot formation than D2 due to previously mentioned reasons. The influence of MI is visible particularly for the TPO, where advancing the timing leads to reduced soot emissions. Combined with high piston velocity at low loads it leads to insufficient residence time and possible fuel impingement on combustion chamber walls. Strong influence of EGR rate is also visible, which is in line with reducing oxygen content when increasing EGR rate.

At both engine speeds, the EGR therefore does not aid at reduction of soot emissions, however by employing advanced MI timing, the negative effect of EGR rate could be reduced and beneficial effects on NO_x and THC emissions could still be achieved.

PM emissions, measured by the MSS and calibrated by the GFM

In case of the D2, the PM results, obtained with the combined MSS and GFM methods (Figures 9c, 10c) are very close to soot emissions measured by the MSS (Figures 9a, 10a). This indicates that the MSS and the GFM either detect similar particles or that both methods are calibrated in a way to yield similar results for the D2. Unlike for the D2, for the TPO, the results for PM, measured with MSS+GFM (Figures 9d, 10d) are more than one order of magnitude larger than the MSS data alone (Figures 9b, 10b) at both engine speeds. This indicates that weight factor between MSS and GFM data for the TPO is far higher than for the D2.

644

645 This is confirmed by the analysis of the weight factors that are for the D2 between 1.2 and
646 3.09, whereas they reach up to 49.85 for the TPO. This means that PM mass of the TPO is
647 consisting of large amount of PM with low absorbance, which is not detected by the MSS but
648 rather only captured by the GFM on filter papers, indicating low absorbance of this PM and/or
649 its different correlation between absorbance and acoustic response of the MSS. The first
650 might occur when particles contain significant amount of non-carbon material (inorganics,
651 sulphates, etc...), while the other could occur already due to different morphologies [26].

652

653 **Comparison of both methods**

654

655 Reasons for such a large discrepancy between MSS and MSS+GFM data for the TPO can be
656 attributed to several different phenomena which are mainly linked to the principles of the
657 MSS measurements and determination of weight factor with the aid of the GFM. Analysing
658 the photo-acoustic and gravimetric method, the transferability of the used measurement
659 procedure to PM of other fuels is subject to the following parameters:

- 660 • Homogeneity of PM
- 661 • Chemical composition of PM
- 662 • Morphology of PM
- 663 • Absorbance of PM
- 664 • Size distribution of PM
- 665 • Thermal expansion of PM

666

667 All of the above listed properties of PM can significantly influence the acoustic response of
668 the MSS, which can easily lead to under and overestimation of measured PM emissions.
669 Generally, overestimation of the PM by the MSS method is unlikely to occur as it is unlikely

for average particle to exceed the light absorbance of black carbon, so most of the errors with alternative fuels would happen in the direction of underestimation of PM.

Specific reasons for discrepancy between MSS and MSS+GFM with TPO can be related to the fuel properties (Table 1) and their effects on emitted PM properties. They can be summarized as:

- Sulphur content of the TPO: Sulphur species consist in part of polycyclic aromatic sulphur species, which may survive the combustion process and be deposited as an unburned fuel fraction on the particles [7] or be included in the structure of the particles, which are in the case of soot consisting of PAH. Giving particles a different density would, due to sulphur presence, lead to underestimation of mass measured with the MSS, as its response is most likely fitted to mass-to-weight ratio of PM from the D2. As the D2 does not contain sulfureted PAH, it may be that its influence is not taken into account in the MSS calibration procedure.
- Ash content of the TPO: According to SIST EN-590, the D2 should have very low ash content, while tested TPO could contain over 0.18 wt.% of ash (Table 1). Ideally, the ash related compounds in combustion products of the TPO (sulphates and oxides) would most likely exhibit very low absorption coefficients in comparison to soot, thus, the MSS would be unable to fully detect them, whereas they would be fully reflected by the GFM data. Furthermore, ash related compounds in PM could be present with several different morphologies [26]. They can be either pure ash particles, coated (encapsulated) with soot or vice versa, or mixed with soot [30] – this significantly influences the absorptive properties and hence the acoustic response of MSS as it

depends on how much of the surface is representing black carbon properties and how much of the particle mass is indeed composed of carbon.

- Size distribution of the particles: Ash particles could act as nucleation cores early in the combustion process, giving way to generation of larger particles than expected with the D2, which would again influence the mass-to-surface ratio of PM to an extent that is not characteristic for the D2. In such a case, the fitted response of the MSS would again underestimate the realistic quantity of the PM.
- Morphology of the particles: Morphology is not only influenced by the ash content as described above, but also by the soot growth and by the soot oxidation. Considering that significantly different soot generation and oxidation kinetics could be characteristic for the TPO due to its chemical composition, also the morphology of the particles could be significantly different compared to those characteristic for the D2. Complex shapes, with significant portions of surface placed in the shadows, not visible to the MSS light source could influence acoustic response of particles, which may exhibit different correlation to the particle mass.
- Adsorption of other species: SO_2 , NO_2 and several hydrocarbons can be adsorbed on PM, similarly as deposition and absorption of sulfureted hydrocarbons on soot. That would increase the particle mass, but would not be detected with MSS, rather only with GFM as the surface of the particle would only marginally increase.
- Measuring device specific parameters: The dilution parameters such as temperature, humidity, residence time and dilution ratio have also significant effect on particle

formation, especially in nucleation mode size range [30]. Therefore, one of the reasons for significant difference of the weight factor between the TPO and the D2 could also be attributed to the change in the dilution ratios which is occurring constantly during the measurements to accommodate the sensitivity of sensors.

- Thermal expansion coefficient of PM: With heterogeneous composition of PM, different vibrational responses of different PM components to thermal excitation can lead to different estimates of otherwise similar particle mass. Thus in case of TPO, heterogeneous composition of particles and their different thermal expansion properties can lead to underestimation of PM, measured with MSS.

Discussion

The relatively large discrepancy and significant underestimation of PM emissions with the MSS is indicating that PM, generated with TPO and possibly also other alternative fuels might exhibit significantly different composition than PM generated with D2. Thus, the methods for measurement of PM emissions of conventional fuels might be questionable to evaluate the combustion performance of complex fuels, obtained through different waste-to-energy processes. Furthermore, a critical assessment of measured results with several methods is mandatory to provide trustful data, which could pose a problem in the case of spreading of alternative fuels. Nevertheless, different chemical composition of PM originating from combustion of different fuels has most probably also different environmental impact and different impact on human health, thus further research in this area is necessary to minimize the potential damage to the environment and impose plausible limits for the alternative fuels that will gain increased market share.

CONCLUSION

It can be concluded that TPO represents a promising waste-derived fuel for Diesel engines despite its lower cetane number compared to the Diesel fuel. It was shown in the article that Diesel like combustion of pure TPO without the use of cetane improvers and any external energy addition can be achieved in a modern turbocharged and intercooled engine equipped with the common rail and the EGR. The influence of key engine control parameters that allow for achieving this objective can be summarized as:

- Introduction of EGR increases temperature at start of injection and by this reduces CCV thereby allowing for extension of the operating range towards lower loads while preserving diesel-like combustion.
- MI timing does not significantly influence ROHR, however maximum in-cylinder pressure increases with advanced MI timing, but stays on similar level as with D2.

It was also shown that with the use of TPO it is possible to achieve comparable gaseous emissions as those measured with the use of the D2. With variation of key engine control parameters, the following influences on TPO combustion were revealed:

- Increasing EGR rate successfully contributes to reduction of NO_x emissions while THC emissions first increase and then decrease at high EGR rates, low engine loads and high engine speed. The effect of EGR on CO emissions is marginal.
- Advancing MI timing is steadily increasing NO_x emissions, which also reflect the effect of FBN, giving a steady rise over entire operating range of the engine in comparison to D2, however this effect can be countered with introduction of EGR.
- Late MI timing, particularly at low loads and high engine speed is increasing THC emissions as a result of short combustion duration. In this case, CO emissions are marginally affected.

772

773 The influence of key engine control parameters on PM emissions was evaluated with two
774 methods; MSS (measuring mainly soot with photoacoustic method) and MSS+GFM
775 (measuring also other particulates with gravimetric method). The findings can be summarized
776 as follows:

- 777 • Soot emissions (measured with MSS) were two-fold higher than for D2. This reflects
778 high PAH content of TPO.
- 779 • Introduction of EGR substantially contributed to soot emissions indicating the
780 importance of sufficient in-cylinder oxygen concentration at the time of MI.
- 781 • Large discrepancies were identified when comparing soot emission (measured with
782 MSS) to PM emissions (measured with MSS+GFM). Based on known composition of
783 TPO, the reason for discrepancy was identified to be different chemical composition,
784 morphology and homogeneity of the PM of TPO in comparison to the PM of D2
785 where MSS and MSS+GFM values coincided very well. These differences indicate
786 that different environmental impact of PM of TPO is likely to be expected which
787 suggests that a critical assessment of the PM effect on environment and health is
788 necessary to impose plausible emission limits on alternative fuels.

789

790 The significance of this study can be recognized in elaborating the key guidelines on injection
791 and EGR rate strategy that enable successfully use of the pure TPO in modern turbocharged
792 and intercooled Diesel engine over a very broad operating range, which makes it very
793 interesting for example for the stationary power applications. Furthermore, the proposed
794 paper opens challenges of PM emissions of alternative fuels with significantly different fuel
795 properties as derivatives of the crude oil that go far beyond the presented analysis.

796

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798

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800 equipment.

801

802 **NOMENCLATURE**

803 BC black carbon

804 CA crank angle

805 CCV cycle-to-cycle variations

806 CI compression ignition

807 CLD chemiluminescent detector

808 CO carbon monoxide

809 CO₂ carbon dioxide

810 COV coefficient of variation

811 D2 commercial Diesel fuel

812 ECU electronic control unit

813 EGR exhaust gas recirculation

814 FBN fuel bound nitrogen

815 FID flame ionization detector

816 FIR finite impulse response

817 GFM gravimetric filter module

818 H₂O water

819 IE injector energizing

820 IMV inlet metering valve

821 LHV lower heating value

822 MI main injection

823 MSS Micro Soot Sensor

824	% m/m	mass/mass percentage
825	NDIR	non-dispersive infrared analyser
826	NO _x	nitrous oxides
827	PAH	polyaromatic hydrocarbons
828	PI	pilot injection
829	PM	particulate matter
830	ROHR	rate of heat release
831	SO ₂	sulphur dioxide
832	SOI	start-of-injection
833	TDC	top dead centre
834	THC	total hydrocarbons
835	TPO	tire pyrolysis oil
836		
837		

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923 **Appendix A**

924 Table 6: D2 operating parameters

Measurement		EGR valve position	EGR mass flow	MI		IP	PI		Total injected mass of fuel	PM micro soot	PM weighted	Dilution ratio	Weight factor
Nr.	rpm_torque	[%]	[%]	[°CA]	[μs]	[bar]	[°CA]	[μs]	[mg/cycle]	[mg/m³]	[mg/m³]	[-]	[-]
1	1500_60	0	0.0	-6	545	700	-25	240	12.464	3.610	11.170	6	3.09
2	1500_60	0	0.0	-8	595	700	-25	240	11.996	3.979	12.313	6	3.09
3	1500_60	0	0.0	-4	535	700	-25	240	12.352	3.079	9.527	6	3.09
4	1500_60	25	10.1	-6	540	700	-25	240	12.201	5.330	16.493	6	3.09
5	1500_60	25	10.3	-8	595	700	-25	240	12.070	5.780	17.884	6	3.09
6	1500_60	25	10.1	-4	535	700	-25	240	12.088	4.770	14.761	6	3.09
7	1500_60	60	25.1	-6	545	700	-25	240	12.317	29.632	91.689	6	3.09
8	1500_60	60	25.7	-8	600	700	-25	240	12.014	27.291	84.447	6	3.09
9	1500_60	60	25.1	-4	540	700	-25	240	12.147	30.590	94.655	6	3.09

10	1500_100	0	0.0	-4	710	840	-33	280	19.803	5.131	6.198	10	1.2
11	1500_100	0	0.0	-6	670	840	-33	280	19.643	4.433	5.355	10	1.2
12	1500_100	0	0.0	-2	700	840	-33	280	19.727	6.632	8.011	10	1.2
13	1500_100	25	7.0	-4	710	840	-33	280	19.813	6.810	8.226	10	1.2
14	1500_100	25	7.6	-6	670	840	-33	280	19.933	6.242	7.540	10	1.2
15	1500_100	25	6.7	-2	700	840	-33	280	19.849	8.326	10.057	10	1.2
16	1500_100	60	12.0	-4	720	840	-33	280	20.186	24.958	30.146	10	1.2
17	1500_100	60	12.7	-6	670	840	-33	280	19.877	19.777	23.888	20	1.2
18	1500_100	60	11.3	-2	710	840	-33	280	20.553	24.649	29.772	20	1.2

19	3000_35	0	0.0	-19	650	550	-46	250	9.787	4.295	5.188	10	1.2
20	3000_35	0	0.0	-21	585	550	-46	250	9.873	2.628	3.174	10	1.2
21	3000_35	0	0.0	-17	615	550	-46	250	9.981	5.371	6.487	10	1.2
22	3000_35	25	6.5	-19	630	550	-46	250	9.407	5.807	7.014	10	1.2
23	3000_35	25	6.3	-21	565	550	-46	250	9.449	3.437	4.151	10	1.2
24	3000_35	25	5.9	-17	595	550	-46	250	9.467	7.835	9.464	10	1.2
25	3000_35	60	19.3	-19	610	550	-46	250	9.075	25.330	30.595	10	1.2
26	3000_35	60	19.3	-21	535	550	-46	250	9.022	13.157	15.892	10	1.2
27	3000_35	60	19.2	-17	580	550	-46	250	9.128	29.725	35.904	10	1.2

28	3000_100	0	0.0	-17	670	950	-50	250	20.844	2.554	7.904	6	3.09
29	3000_100	0	0.0	-19	650	950	-50	250	20.966	1.952	6.041	6	3.09
30	3000_100	0	0.0	-15	665	950	-50	250	21.071	3.217	9.954	6	3.09
31	3000_100	25	5.9	-17	660	950	-50	250	20.246	4.894	15.145	6	3.09
32	3000_100	25	5.8	-19	635	950	-50	250	20.449	3.733	11.550	6	3.09
33	3000_100	25	5.8	-15	650	950	-50	250	20.533	6.113	18.915	6	3.09
34	3000_100	60	19.5	-17	665	950	-50	250	20.878	118.011	365.158	40	3.09
35	3000_100	60	19.5	-19	645	950	-50	250	20.956	116.490	360.452	40	3.09
36	3000_100	60	19.2	-15	660	950	-50	250	20.959	130.908	405.065	40	3.09

925 Table 7: TPO operating parameters

Measurement		EGR valve position	EGR mass flow	MI		IP	PI		Total injected mass of fuel	PM micro soot	PM weighted	Dilution ratio	Weight factor
Nr.	rpm_torque	[%]	[%]	[°CA]	[μs]	[bar]	[°CA]	[μs]	[mg/cycle]	[mg/m³]	[mg/m³]	[-]	[-]
1	1500_60	0	0.0	-6	555	700	-25	240	13.180	3.5	111.8	40	32.16
2	1500_60	0	0.0	-8	615	700	-25	240	12.932	6.3	202.9	40	32.16
3	1500_60	0	0.0	-4	550	700	-25	240	13.045	3.2	103.3	40	32.16
4	1500_60	25	7.2	-6	550	700	-25	240	12.890	16.4	529.0	40	32.16
5	1500_60	25	7.5	-8	615	700	-25	240	12.889	8.0	257.5	40	32.16
6	1500_60	25	7.0	-4	550	700	-25	240	13.005	12.2	393.3	40	32.16
7	1500_60	60	22.1	-6	555	700	-25	240	13.080	60.8	1956.9	40	32.16
8	1500_60	60	23.3	-8	615	700	-25	240	12.788	40.2	1291.3	40	32.16
9	1500_60	60	22.7	-4	550	700	-25	240	12.992	54.8	1763.2	40	32.16
10	1500_100	0	0.0	-4	705	840	-33	280	21.139	5.0	249.7	10	49.85
11	1500_100	0	0.0	-6	670	840	-33	280	20.872	4.5	222.8	10	49.85
12	1500_100	0	0.0	-2	700	840	-33	280	20.895	5.0	247.8	10	49.85
13	1500_100	25	5.3	-4	705	840	-33	280	20.551	6.6	328.8	10	49.85
14	1500_100	25	5.8	-6	670	840	-33	280	20.610	7.2	358.0	10	49.85
15	1500_100	25	4.4	-2	705	840	-33	280	21.091	8.1	402.5	10	49.85
16	1500_100	60	10.2	-4	720	840	-33	280	21.128	23.4	1164.1	10	49.85
17	1500_100	60	11.1	-6	680	840	-33	280	20.988	23.1	1153.2	10	49.85
18	1500_100	60	9.6	-2	715	840	-33	280	21.506	21.5	1071.8	10	49.85
19	3000_35	0	0.0	-19	680	550	-46	250	10.889	5.6	278.6	10	49.85
20	3000_35	0	0.0	-21	625	550	-46	250	10.813	15.6	779.8	10	49.85
21	3000_35	0	0.0	-17	685	550	-46	250	12.221	33.8	1684.7	10	49.85
22	3000_35	25	7.7	-19	660	550	-46	250	10.402	23.1	1176.7	20	49.85
23	3000_35	25	6.9	-21	600	550	-46	250	10.376	43.6	2175.5	20	49.85
24	3000_35	25	4.4	-17	670	550	-46	250	11.689	183.9	9163.3	20	49.85
25	3000_35	60	24.0	-19	690	550	-46	250	10.909	158.6	2492.7	50	15.71
26	3000_35	60	26.2	-21	585	550	-46	250	10.025	115.4	1813.7	50	15.71
27	3000_35	60	25.4	-17	640	550	-46	250	10.669	337.9	5309.5	50	15.71
28	3000_100	0	0.0	-17	685	950	-50	250	22.377	3.7	58.6	20	15.71
29	3000_100	0	0.0	-19	645	950	-50	250	22.136	2.7	42.7	20	15.71
30	3000_100	0	0.0	-15	675	950	-50	250	22.326	4.0	62.8	20	15.71
31	3000_100	25	5.9	-17	665	950	-50	250	21.749	6.0	94.9	20	15.71
32	3000_100	25	5.7	-19	625	950	-50	250	21.697	4.1	64.4	20	15.71
33	3000_100	25	5.7	-15	660	950	-50	250	21.726	6.9	108.5	20	15.71
34	3000_100	60	18.0	-17	670	950	-50	250	21.715	133.6	2098.6	30	15.71
35	3000_100	60	17.8	-19	650	950	-50	250	21.350	109.5	1720.5	30	15.71
36	3000_100	60	17.2	-15	700	950	-50	250	22.140	170.7	2681.5	30	15.71