Article citation info:
CHWIST, M., PYRC, M., GRUCA, M. et al. By-products from thermal processing of rubber waste as fuel for the internal combustion piston engine. *Combustion Engines*. 2020, **181**(2), 11-18. https://doi.org/10.19206/CE-2020-202

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CE-2020-202

By-products from thermal processing of rubber waste as fuel for the internal combustion piston engine

The article presents results of investigation on the combustion of a mixture of pyrolysis oil from tires and regular fuel in the internal combustion reciprocating piston engine. The tested fuel consisted of: diesel fuel and pyrolysis oil at amount of 10% by volume. The tests were carried out on a single-cylinder naturally aspirated compression-ignition engine. The engine was equipped with a common rail fuel injection system and an electronic control unit that allowed changing injection timing. A comparative analysis of pressure-volume charts for the reference fuel, which was diese fuel, and for a mixture of diesel with 10% addition of pyrolysis oil was carried out. Injector characteristics for the reference fuel and the mixture were determined. Engine efficiency for both fuels was also determined. Unrepeatability of the engine work cycles for the diesel fuel and the tested mixture was calculated. Finally, exhaust toxic emission was analyzed. It was found that the pyrolysis oil can be used as valuable additive to regular diesel fuel at amount up to 10%, however, toxic exhaust gases emission was increased.

Key words: *pyrolysis oil, thermal conversion, biocrude, diesel engine, alternative fuels*

1. Introduction

Thermal processing of organic matter is an alternative method of obtaining fuel for internal combustion piston engines. Pyrolysis is one of the methods of organic thermal processing. Pyrolysis is a process in which organic material is heated up to temperature of approximately 500°C in oxygen-free atmosphere. The composition and quantity of individual fractions of pyrolysis products depends on the following [5, 15]:

- physical-chemical properties of the input material,
- temperature,
- pyrolysis reactor type,
- process speed (heating up time, retention time).

During pyrolysis, organic material is converted into the following [12, 13]:

- the gas fraction, also called pyrolysis gas, which apart from methane, ethane, carbon monoxide and dioxide and water vapors also contains vapors of hydrocarbon compounds,
- solid fraction consisting of carbon, metals and other inert substances,
- liquid fraction called pyrolysis oil, that is formed after cooling the pyrolysis gas to ambient temperature. The composition of pyrolysis oil includes: condensed hydrocarbon compounds, tars, water, alcohols and organic acids. Raw pyrolytic oil is a dark-brown substance with a strong characteristic odor [2, 8].

Exemplary fractions produced from the pilot installation at the Czestochowa University of Technology are shown in Fig. 1. The pyrolysis installation is equipped with a screw feeder for moving the input material. Heating is carried out by electric heaters. The system is equipped with a controller to control the set point temperature and a frequency converter that allows changing the retention time of keeping the charge material in the heating zone. The largest share is a solid fraction, which contains significant amount of energy up to 90% in comparison to the input material. The gas fraction constitutes approximately 30% and contains nearly 10% input energy of the material [10].

The most desirable product of pyrolysis is the oil fraction, which can be used as liquid fuel that can be origin for further processing to obtain various chemical compounds. The gas fraction is most often used for process itself. The solid fraction is managed as a final product – char or activated carbon [4, 11, 15].

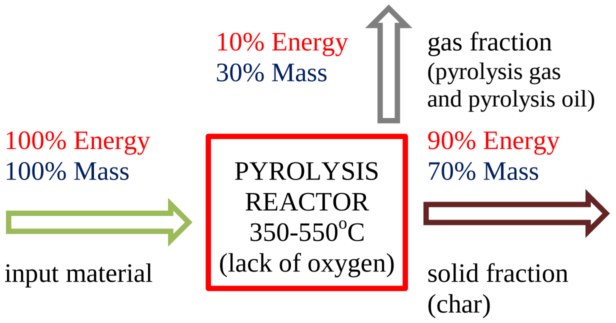


Fig. 1. Mass and energy distribution in pyrolysis process

Pyrolysis reactors have various construction. The most common installations are as follows: fixed bed reactor, fluidized bed reactor, screw conveyor reactors, rotary cone reactor, ablation reactor. Depending on the type of reactor, the worn out tires can load the reactor directly (fixed bed reactor) or require fragmentation (e.g. fluidized bed reactor). The pyrolysis process can be carried out with the addition of a catalyst, which aim is to remove unwanted compounds from the gas [7, 12].

As the input material for the pyrolysis process, various kinds of organic waste: from the food industry, waste from the agricultural industry, waste from the fish industry, waste from the wood industry, municipal waste, special purpose crops can be used. The problem of managing worn car tires has been growing in recent years. Tires do not degrade in the natural environment for up to 100 years.

Received 5 March 2020; Received in revised form 30 April 2020; Accepted 30 April 2020
Available online 30 June 2020

They are a nuisance waste and have been classified as waste that should be professionally utilized [6, 9].

The amount of tires waste increases year by year. All rubber waste constitute about 80%. It is estimated that 1.5 billion tires are produced annually in the world. Until now, the main technology of tires recycling is to burn them in cement plants. Another option is to use them for the pyrolysis process [14].

The literature presents results of research on applying pyrolysis oil (PO) from tires as the additive to regular fuel (diesel fuel – DF) and also as a stand-alone fuel. For example, in [8] an analysis of engine operation was carried out on a mixture of diesel fuel mixed with biodiesel and pyrolysis oil from tires. The addition of individual components was as follows: 80% diesel fuel, 10% biodiesel and 10% pyrolysis oil from tires. As a result of the experiment, the authors found that the addition of this pyrolysis oil increases the engine torque and power. The addition of the pyrolysis oil reduces NO_x and CO emissions. According to the authors, the disadvantage of tire pyrolysis oil is the need to filter and desulphurize it before feeding it to the engine.

In article [11], the authors used tire pyrolysis oil as fuel to a 4-cylinder diesel engine. They concluded that the tire pyrolysis oil has a lower cetane rating but is still a promising alternative fuel. They observed that the engine can only work on pyrolysis oil without addition of improvers to get higher cetane number. The use of oil from the pyrolysis of tires allows to maintain the emission of toxic gas components at a level comparable to those where the engine is running on diesel fuel.

In [6] Authors used oil from tire pyrolysis as the additive to diesel fuel. The authors created mixtures with pyrolysis oil of 10%, 20% and 50% by volume. They concluded that the addition of 10% pyrolysis oil may occur as best alternative fuel due to satisfactory good engine performance and environmental aspects as well.

Based on the literature analysis, combustion tests of the mixture of diesel fuel and oil from pyrolysis of tires in a single-cylinder diesel engine were taken. Hence, the main target of the work was to investigate the pyrolysis oil as the additive at 10% to diesel fuel.

2. Description of the research stand

The test stand is shown in Fig. 2. It is the engine coupled with a generator connected to power grid. To measure electrical power three-phase electricity meter (1) was used. The three-phase asynchronous generator/motor (2), with a power of 15 kW at 400 V rated voltage is used first to start up the engine and then it works as a dynameter. The asynchronous motor has two operating speeds: 1465 rpm and 975 rpm. A lower rotation speed of 975 rpm was applied to investigation.

The technical specifications of the compression ignition engine S320 (3) are shown in Table 1.

The engine underwent a number of modifications to be adapted to research. The most important change was modification of the cylinder head. It has additional mounting sockets needed to mount an additional injector and a pressure sensor. Currently, two common rail injectors and a piezoelectric pressure sensor are installed in the cylinder head. The second important change of the engine was replacement of the original evaporation based cooling system by an radiator cooler. The cooling system works at ambient pressure. The cooling system and all elements of test stand are shown in Fig. 3.

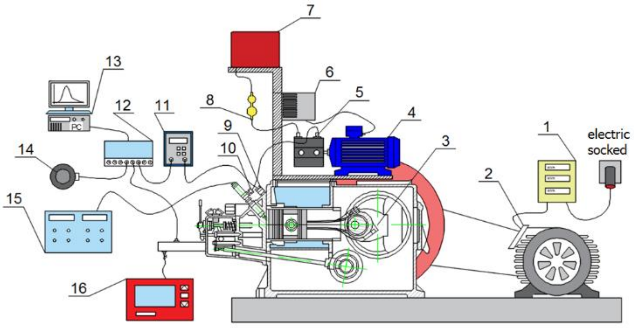


Fig. 2. Diagram of the test stand

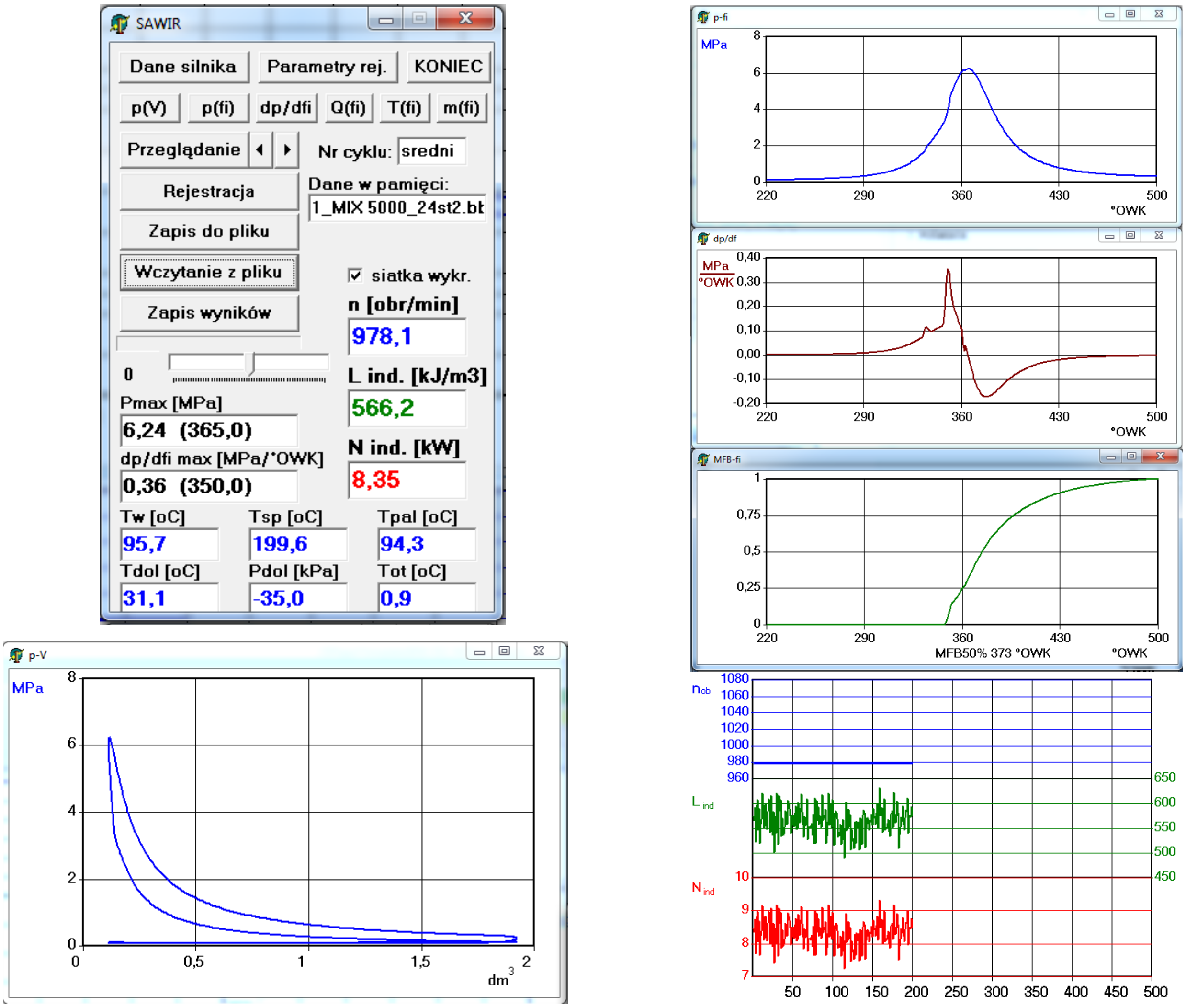
Table 1. Technical data of the engine S320

Type of engine	Four-stroke
Fuel	Diesel Fuel
Number of cylinders	Single
Direct of cylinder	Horizontal
Number of valves	2
Cylinder bore	120 mm
Piston stroke	160 mm
Engine displacement	1810 cm^3
Compression ratio	17
Nominal power(1)	13.2 kW for 1500 RPM
Maximum torque(1)	93 Nm for 1300 RPM
Specific fuel consumption	234 g/kWh 172 g/HPh
Capacity lubrication system	10.5 dm^3
Oil consumption	4.1 g/kWh
Beginning of fuel delivery	variable
Injection pressure	to 200 MPa
Suction valve timing	Opening: 23 CA deg before TDC Closing: 40 CA deg after BDC
Exhaust valve timing	Opening: 46 CA deg before BDC Closing: 17 CA deg after TDC
Valve clearance for cold engine	suction 0.4 mm exhaust 0.4 mm
Oil pressure in the warm engine	150-300 kPa
Cooling	open water system
Engine weight	315 kg
(1) Parameters for temperature +15°C, pressure 100 kPa, relative humidity 60%	

The original Andoria S320 engine fuel system was replaced by a modern common rail fuel injection system. The three-section twin-piston pump (5) is driven by an asynchronous engine (4) with a power of 2.2 kW. The injection pump contains three pressing sections arranged radially every 120°. The pump is equipped with a pressure regulator. The fuel injection pressure is controlled with a controller. During the tests, fuel is injected at a pressure of 100 MPa. The asynchronous motor (4) is powered by a frequency converter (6), which allows to control fuel flowrate. A fuel tank (7) is mounted next to the test stand. Fuel consumption is measured by the fuel flow meter (8). The fuel system is equipped with a fuel cooler and an additional preliminary, supporting low pressure a fuel pump. The cooler is needed at the stand, because with long time continuous work of the engine the fuel temperature increases significantly. For fuel injection, the BOSCH 0445110076 (10) injector is used. An electronic system (15) is used to control the injector timing. The controller makes it possible to adjust the injection time with resolution of 10μs. In addition, it allows shifting start of injection. The electronic system reads pulses from the encoder which is installed on the camshaft (14). The encoder generates 360 pulses per revolution. As the encoder is installed on the camshaft, the change of injection start is at resolution of 2 deg crankshaft revolution. For measuring in-cylinder pressure the Kistler piezoelectric sensor type 6061B (9) is used. This sensor is mounted into the cylinder head. The sensor sensitivity is –25.8 pC/bar, measuring range up to 25 MPa. The signal from the piezoelectric sensor needs amplification, therefore another device in the measurement pathway is the charge amplifier (11). The signal after amplification is registered by the National Instrument measurement data acquisition card (12). The encoder signal (14) is used to determine p-V diagram in real time working conditions. SAWIR program by Dr M. Gruca (13) is used to register and process engine parameters in real time working conditions. The view of the program window with real-time charts is shown in Fig. 4.

Fig. 3. The test stand

The exhaust gas analyzer Radiotechnika model AI9600 (16) is used to measure the concentration of toxic exhaust compounds.



Nr cyklu: sredni

Dane w pamięci: 1_MIX 5000_24st2.bt

☑ siatka wykr.

n [obr/min] 978,1

L ind. [kJ/m3] 566,2

Pmax [MPa] 6,24 (365,0)

dp/dfi max [MPa/°OWK] 0,36 (350,0)

N ind. [kW] 8,35

Tw [oC] 95,7

Tsp [oC] 199,6

Tpal [oC] 94,3

Tdol [oC] 31,1

Pdol [kPa] -35,0

Tot [oC] 0,9

Fig. 4. View of the SAWIR program graphical interface and results

3. Methodology

Before tests, correct determination of TDC (top dead center) location was done. Next, several engine geometrical data were input to the software: crank radius, connecting rod length, piston diameter, compression ratio. The engine worked on reference fuel to reach operating temperature, i.e. 363 K. At the beginning, the injector characteristics for reference fuel and mixture of diesel fuel+10%PO were determined, the injection time was changed in the range of: 2–6 ms with resolution of 10 µs. Fuel consumption time was measured from a 41 cm^3 meter. Fuel consumption time was measured with accuracy of 0.01 s. Each test series consisted of 200 engine consecutive combustion events. During the experiment, the influence of the beginning of the fuel injection start on the p-v plots was also examined. At the same time, the concentrations of CO, NO_x, HC in the exhaust gas were measured. The tests were conducted for regular diesel fuel as the reference fuel and the mixture of diesel fuel and 10%PO. The proposed addition of 10% pyrolysis oil to diesel fuel resulted from the research conducted so far on fuels. During the study, the authors measured: the concentration of hazardous compounds in the exhaust gas using an exhaust gas analyzer that was connected to the exhaust gas outlet duct. During the fuel test, a fixed injection time of 5 ms was maintained.

4. Analysis of results

The time-pressure signals obtained from tests were used to determine the following: indicated mean effective pressure (IMEP), injector characteristics, thermal efficiency of the engine, toxic exhaust emissions.

The tests consisted of 2 stages. In the first stage, engine tests were carried out with reference fuel, i.e. diesel fuel. The fuel injection timing was changed in the range of 24–32 CA deg bTDC. The injection time was 5 ms at 100 MPa injection pressure. Start of injection was limited by diesel knock occurring at timings advanced over 32 CA deg bTDC. Graphs with exemplary in-cylinder pressure curves are shown in Fig. 5.

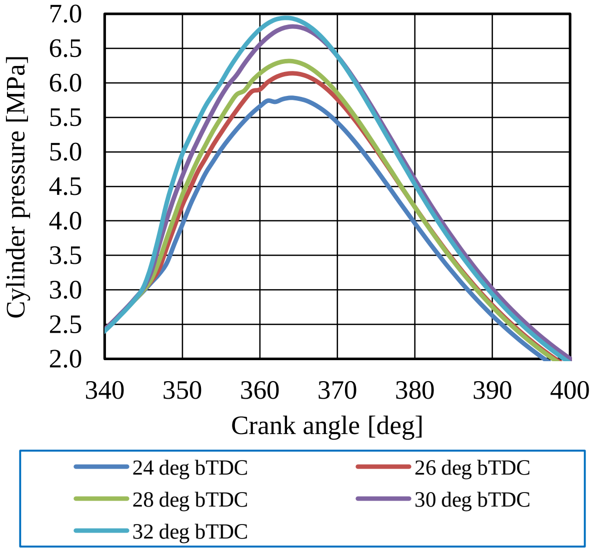


Fig. 5. Cylinder pressure vs. crank angle range 340–400 deg for a variable start of fuel injection, fuel: DF

In next stage of tests, as seen in Fig. 6, the fuel applied to tests was the mixture containing diesel fuel and oil from tire pyrolysis (PO) at 10% by volume. Amount of PO was limited to 10% only with risk of damaging the high pressure fuel pump. As previously, the fuel injection timing was changed in the range of 24–34 CA deg bTDC. The duration of injection was also 5 ms. The injection pressure was 100 MPa.

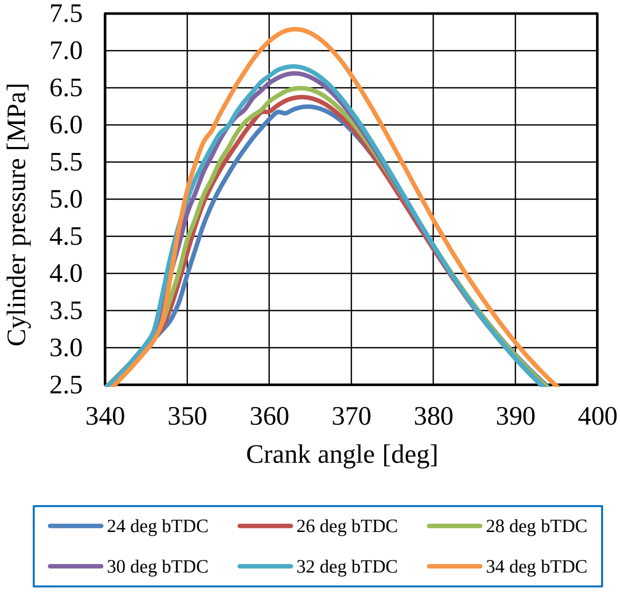


Fig. 6. Cylinder pressure vs. crank angle range 340–390 deg for a variable start of fuel injection, fuel: DF+10%PO

On the basis of the data obtained, a comparative analysis of the combustion pressure traces inside the engine cylinder was carried out maintaining fixed injection timings. Both in-cylinder pressure and pressure rise rates for reference fuel and mixture: DF+10%PO are presented in Figs 7 and 9.

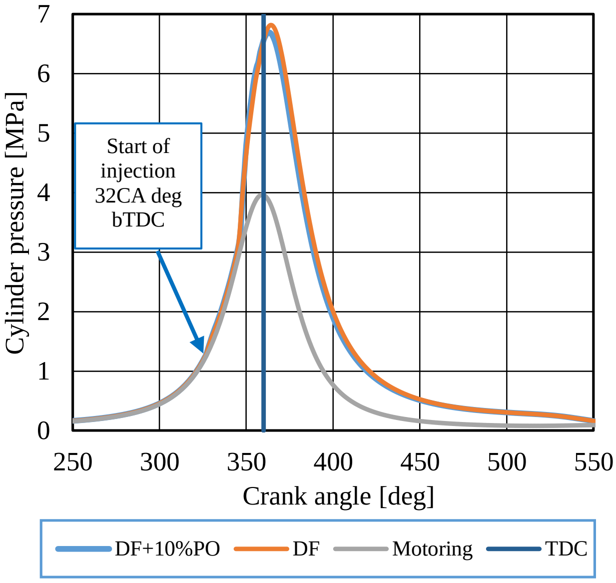


Fig. 7. Cylinder pressure vs. crank angle range 250–550 deg for a variable fuel

The graph (Fig. 7) shows a slight decrease in in-cylinder pressure after adding 10% PO. It affected lower combustion pressure and decrease in the indicated work (Lind the same as IMEP) and engine power. Pressure rise rate slightly increased with PO addition as shown in Fig. 8. The mean indicated work (IMEP) from 200 measurements was 578 kJ/m^3. The unrepeatability of the engine's cycles was 4.18%. In the case of an engine fueled with a mixture of diesel fuel and PO, the mean indicated work is 530 kJ/m^3 (decrease by 8.3%) and the engine work cycles unrepeatability unremarkably increased to 4.93%.

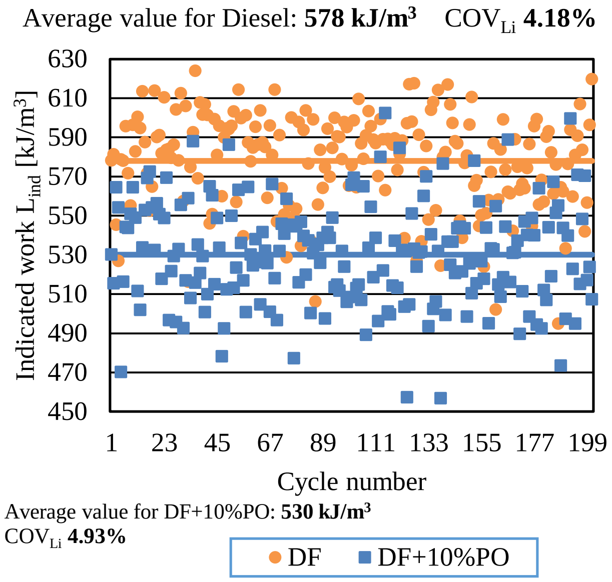


Fig. 8. Unrepeatability of the engine work for reference fuel and for research fuel

Figure 9 shows the rate of pressure rise. It can be seen that in the initial phase there is a faster pressure increase for DF+10%PO than for reference fuel (DF). Visible peaks, deviations from mileage caused by the start and end of fuel injection.

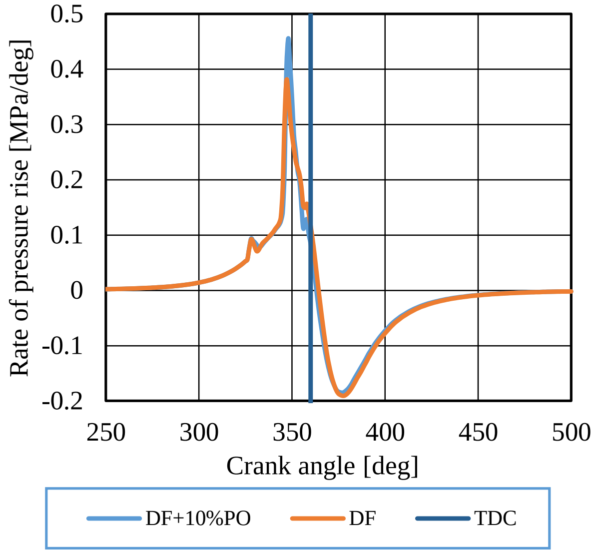


Fig. 9. Rate of pressure rise vs. crank angle

Figure 10 shows the pressure increase due to fuel combustion in the engine cylinder. The pressure plots present positive pressure over the pressure from motored events. The ignition timing in the studied case was 17 CA deg. The injection time was 5 ms, i.e. fuel injection in crank angle was 29.25 deg bTDC. Fuel injection ended approximately 1 CA deg bTDC.

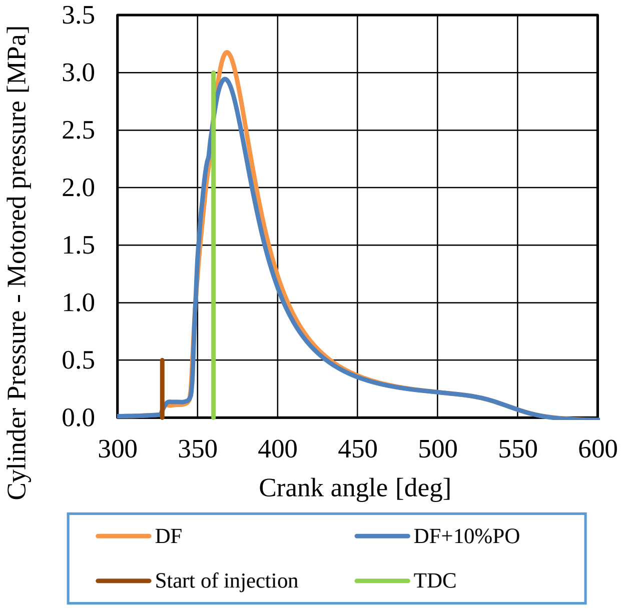


Fig. 10. Combustion pressure increase

The injector characteristics were determined for the reference fuel (DF) and for the tested mixture: DF+10%PO. The results are shown in Fig. 11.

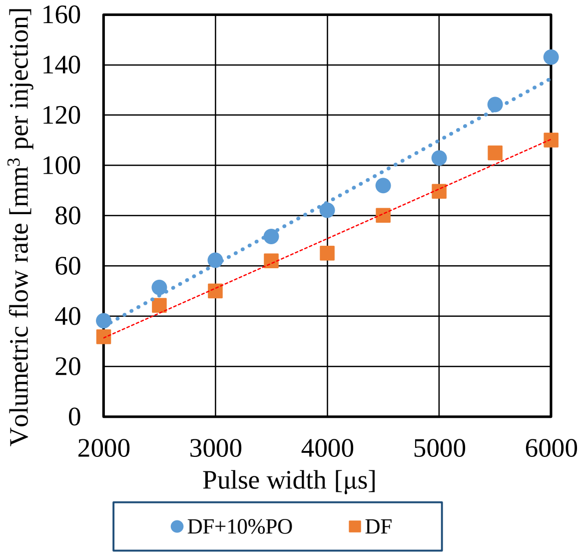


Fig. 11. Characteristic curves of the injection rate for both fuel

From the injector characteristics it can be seen that the addition of 10%PO increases volumetric flow. Based on the measurement of the amount of fuel burned in a given time and indicated work, the engine efficiency could be calculated according to the equation:

$$\eta_T = \frac{L_i}{Q_F} \qquad (1)$$

where: L_i – indicated work per cycle [kJ], Q_F – fuel energy in a dose injected to the engine cylinder [kJ].

Table 2. Net heating value of fuels used [1, 3]

	DF	PO	DF+10%PO
Net heating value [J/mg]	42.5	39	42.15
Density [g/cm^3]	0.82	0.95	0.833
Net heating value [J/mm^3]	34.85	37.05	35.07

The results of the indicated efficiency for the tested fuels with various injection timings are shown in Fig. 12. The efficiency of the engine slightly decreased while supplying the engine with DF+10%PO mixture. The engine achieved its highest efficiency for the most advanced fuel injection timing (32 CA deg bTDC). The efficiency was over 36%. The efficiency decreased with delayed fuel injection. The lowest calculated efficiency was 26% at injection timing of 24 CA deg bTDC.

In addition, in Table 3, the IMEP as average from 200 combustion events is presented. For diesel fuel IMEP reaches the highest value for 30 CA deg bTDC. For DF+10%PO the highest IMEP value is for start of injection timing 32 CA deg bTDC.

Table 3. Average IMEP for reference fuel and Diesel Fuel+10%PO

Start of injection timing	IMEP Fuel DF	IMEP Fuel DF+10%PO
[CA deg bTDC]	[kPa]	[kPa]
24	462	449
26	514	522
28	507	501
30	578	530
32	552	546

The lowest IMEP achieved for the retarded fuel injection i.e. 24 CA deg bTDC.

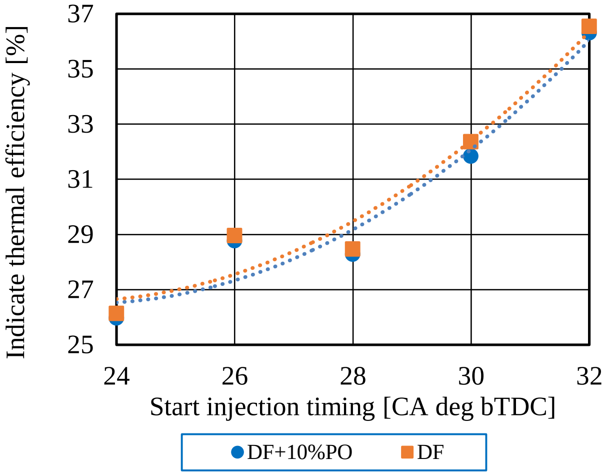


Fig. 12. Indicated thermal efficiency curves

The concentration of toxic exhaust compounds was also measured during the tests. The results of the CO concentration in exhaust gases are shown in Fig. 13.

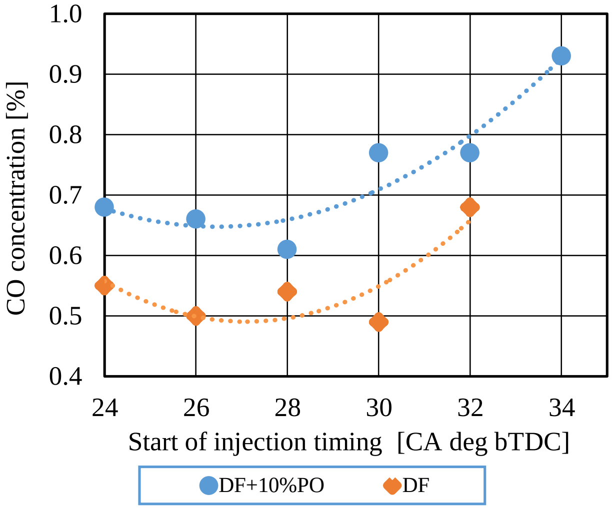


Fig. 13. CO concentration in exhaust gases

The CO concentration increased significantly after adding 10%PO. The minimum CO was observed at 28 CA deg. Retarding or advancing the injection increased the CO content in the exhaust gas. It probably was caused by heavy hydrocarbons in the pyrolysis oil that required longer time for breaking up chemical bonds and complete combustion that caused combustion process of PO slowed, that resulted in slowing the overall combustion.

The NO_x content is shown in Fig. 14. Regardless the fuel fed, the earlier the fuel injection into the cylinder, the higher the NO_x content in the exhaust gas. For all cases, the NO_x concentration was higher when the engine was fueled with DF+10%PO. Higher NO_x emission is difficult for explanation when both CO and unburnt HC also increased. The NO_x emission probably was caused by locally higher temperatures that promoted thermal mechanism for NO_x formation.

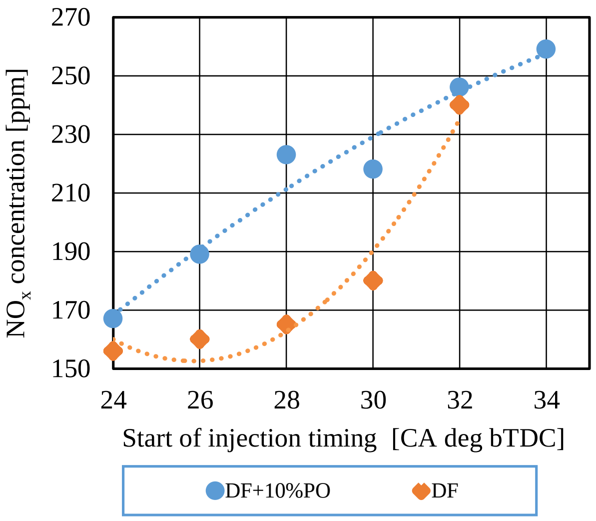


Fig. 14. NO_x concentration in flue gases

The toxic unburnt hydrocarbons (HC) in exhaust gases is shown in Fig. 15. The HC was also significantly higher in the case of engine operation on the mixture: DF+10%PO rather than on reference fuel (DF). The reason was probably the same as for CO, so presence of heavy hydrocarbons in the pyrolysis oil taken for investigation.

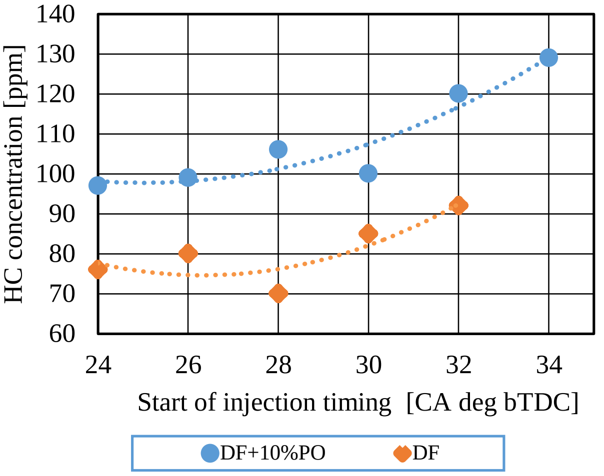


Fig. 15. HC concentration in exhaust gases

5. Conclusion

Utilization of liquid products from the thermal treatment of waste rubber (car tires) through combusting them in the IC engine is one of the methods that might be less harmful to the natural environment.

Oil from tire pyrolysis (PO) can be mixed with diesel fuel (DF) at amount of 10% by volume and used to power diesel engines.

As observed, addition of this pyrolysis oil (PO) causes decrease in-cylinder combustion peak pressure, that causes decrease in the mean indicated work (IMEP) and the engine power as well as its efficiency.

After adding PO the unrepeatability of consecutive engine work cycles increased slightly.

PO added to DF causes increase in injector flowrate by 15–25%.

The addition of pyrolysis oil from tires to diesel fuel causes increase in toxic engine exhaust emissions: carbon monoxide, nitrogen oxides and unburnt hydrocarbons.

Acknowledgements

European Union's Horizon 2020 research and innovation programme under grant agreement No 691232 – Knocky – H2020-MSCA-RISE-2015/H2020-MSCA-RISE-2015 and the research was additionally co-financed by Polish Ministry of Science within the frame of science support funds for international co-funded projects in 2016-2019.

Nomenclature

BDC	bottom dead center	HPh	horse power-hour
bTDC	before top dead center	IMEP	indicated mean effective pressure
CA	crank angle	Li	indicated work
CI	compression ignition	NO_x	nitrogen oxides
CO	carbon monoxide	PO	pyrolysis oil
COV	coefficient of variation	QF	fuel injected energy
DF	diesel fuel	RPM	revolutions per minute
HC	unburnt hydrocarbons	TDC	top dead center

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