THE EFFECT OF DIESEL FUEL MIXTURE AND CAMELINA OIL ESTER ON THE PROCESS OF FUEL INJECTION IN TRACTION ENGINE

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Abstract

This paper presents the results of comparative research on basic physical and chemical properties of pure diesel fuel and two types' mixtures of Camelina oil ester (Camelina oil ester content of the mixture was 10% and 15%) and diesel fuel. Application of these mixtures has different physicochemical properties to power supply diesel engine, made it possible to detect existing differences in the injection process (pumping and spraying). The analysis of the injection process has been defined through experimental graphs of the fuel pressure in injection line and registered height of the needle injector. With the knowledge of these two fast-changing engine parameters and load–velocity conditions, there were calculated: fuel flow speed from nozzle, range and fuel spray disintegration time and the average critical diameter of the droplets.

In the final part of the article it was shown, that the fuel with different physicochemical properties has a significant impact on the injection process. There were also made a comparative assessment of the suitability test compounds for use in service.

Keywords: *fuel plant, process of injection, engine test bench diesel engine*

1. Introduction

A growing number of compression ignition engine enforces survey and placing on the market an increasing number of different alternative fuels that have a significant impact on energetic, economic, and mainly ecological indicators of engines work [1, 2].

When internal–combustion engine is powering with biofuel blends with diesel it should be aware that its use reduces power import and consumption of diesel oil. Liquid biofuels should also meet the criteria imposed hydrocarbon motor fuels.

The most important requirements for alternative fuels and liquid biofuels at the same time are [6]:

- occurrence in sufficiently large quantities,
- the cost of production and distribution comparable with conventional fuels,
- no need to implement the complex and costly design changes in engines,
- safety of use engines,
- easy storage and biodegradability,
- low toxicity of the fuel and its combustion products.

Injection and combustion processes are complex, periodically repeated, fast response processes taking place inside the cylinder piston of an engine. One of the quite easily obtained and reliable sources of information on processes of fuel injection and combustion are graphs: the fuel pressure in the injection, the injector needle lift and cylinder pressure [7, 8].

The use of fuels with different physico-chemical properties to the motor power supply ignition and study its impact on the rates of operation requires high accuracy to provide reliable results of the injection process, the disintegration spout of fuel and its combustion [9].

Solutions to this problem are sought, inter alia, the use of alternative fuels produced in a plant–based oil and their esters. Fuel plant has different physico–chemical properties as compared to hydrocarbon fuels. This causes differences in the process of stamping and fuel atomization as well as the process of combustion in the cylinder [7].

2. Research objective

The aim of the study was to assess the impact of diesel oil (ON) and two mixtures of the composition by volume: L–10 (90% ON and 10% camelina ester) and L–15 (85% ON and 15% ester of camelina) on the fuel injection process.

During the test engine has been operated at the maximum volume of fuel (and the two characteristic rotational speeds of the engine crankshaft) and the factory settings regulators.

3. The test stand, physical and chemical properties of fuel and a description of test methods

Perkins engine 1104C–E44T with direct fuel injection of Bosch VP–30 injection pump is used primarily as a source of drive of traction engines. Outside the EU and the U.S. is also used in cars.

The research was conducted on a typical built by BN–74/1340–12 and PN–88/S–02005 engine test bench [3].

The choice of fuel supply to the engine due to their availability on the market and taking into account the physical and chemical properties that have a significant impact on the process of fuel injection, ie its density, viscosity and surface tension.

Selected physical and chemical properties of the tested fuels affecting the fuel injection process is shown in Tab. 1. The Fig. 1 shows a block diagram of test stand.

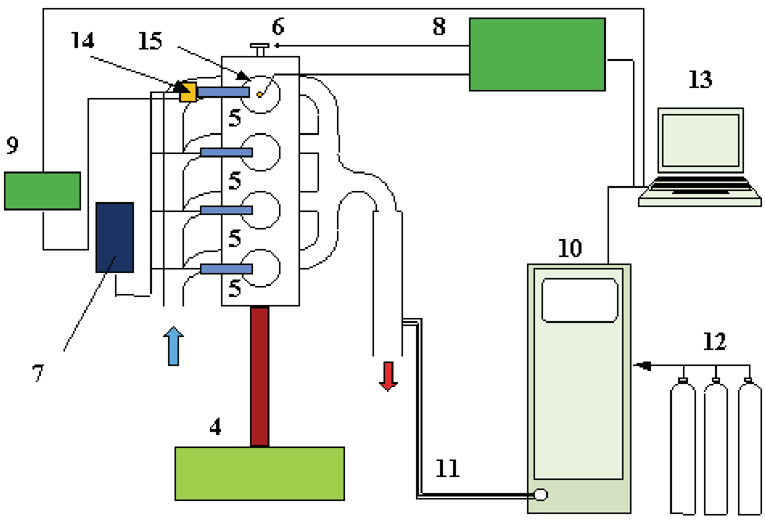

Fig. 1. Block diagram of test stand: 1 – Engine PERKINS 1104C–E44T, 2 – air inlet, 3 – exhaust; 4 – brake Schenck, 5 – Delphi fuel injectors, 6 – recorder crank angle; 7 – Bosch VP30 injection pump, 8 – complex system AVL IndiSmart; 9 – sensor signal amplifier, 10 – gas analyzer AVL CEB II, 11 – way heated, 12 – a set of reference gases, 13 – measuring computer

In the middle of preparing external characteristics of velocity for the engine speed n=1400 rpm (speed of the maximum torque) and n=2000 rpm (speed of the maximum effective power) were recorded: fuel consumption, fuel pressure the injection pipe, the pressure in the combustion chamber and the stroke of needle injector.

Tab. 1. Selected physical and chemical properties tested fuels affect fuel injection process [10]

PARAMETER	Diesel oil (ON–100%)	L10 (90% ON + 10% ester of camelina)	L15 (85% ON +15% ester of camelina)
Density at 15°C [kg/m^3]	835.4	841	848
Viscosity at 40°C [mm^2/s]	2.64	3.26	3.40
The calorific value [MJ/kg]	43.2	42.5	42.2

Indicator diagrams were removed using a measurement system with sensors piezoelectric AVL hallmarked statically and dynamically in conditions similar to the actual conditions of Perkins engine's work [5].

Using the measured values were determined:
- the pressure in the injection pipe,
- injector needle lift,
- the pressure in the combustion chamber.

After preliminary examination of indicator diagrams were determined: the pressure difference Δp between the pressure in the injector $p_{w.max}$ and the pressure in the cylinder $p_{c.max}$ [5]:

$$\Delta p = p_{w.max} - p_{c.max}, \text{ [MPa]}, \tag{1}$$

where:
$p_{w.max}$ –averaged maximum fuel injection pressure, [MPa],
$p_{c.max}$ –average maximum pressure in the combustion chamber, [MPa].

Range fuel stream depended on: the pressure difference in the well injector and cylinder pressure Δp, the density of the medium in the cylinder ρ_g and fuel density ρ_p and the diameter of the pinhole sprayer d_o. The outflow velocity of the fuel stream from the injector u_p (in the exhaust nozzle cross-section) was calculated from the equation [4]:

$$u_p = \mu \sqrt{\frac{2\Delta p}{\rho_p}} \text{ [m/s]}, \tag{2}$$

where:
Δp – difference fluid pressure before the nozzle holes and the pressure center, to which fuel is injected, [MPa],
ρ_p – density of the fuel, [kg/m^3],
μ – hydrodynamic influence coefficient depends on the design of the injector.

Calculating the average, critical droplet diameter d_{kr} for the examined fuel carried out according to the formula proposed by Sauter [4]:

$$d_{kr} = \frac{\sigma We_{kr}}{\rho_g u_p^2} \text{ [μm]}, \tag{3}$$

where:
σ – surface tension, [N/m],
ρ_g – the density of the gaseous medium, [kg/m^3],
We_{kr} – the critical Weber number, [m/s],
u_p – the flow rate of fuel flow from the injector, [m/s],

Time calculated from the flow of fuel from the nozzle to the jet break time t_r can be calculated from the equation [4]:

$$t_r = \frac{29\,\rho_p\,d}{(\rho_g \cdot \Delta p)^{1/2}}\ [\text{ms}], \tag{4}$$

where d – diameter nozzle – 0.32 mm [3].

By Hiroyasu penetration depth can be determined [4]:
S_1 – the range of primary fuel stream,
S_2 – secondary range jet fuel.
For $t \le t_r$:

$$S_1 = 0.39\left(\frac{2\Delta p}{\rho_p}\right)^{1/2} t\ [\text{mm}]. \tag{5}$$

For $t > t_r$:

$$S_2 = 2.95\left(\frac{2\Delta p}{\rho_g}\right)^{1/4} (d \cdot t)^{1/2}\ [\text{mm}]. \tag{6}$$

4. Graphical comparison of test results

Figure 2 shows a comparison of the maximum fuel pressure in the injector $P_{w.max}$. for used fuel and at the speed of the engine crankshaft n=1400 rpm and n=2000 rpm.

Figure 3 shows the initial flow rate, fuel flow from the injector U_p m/s for two rotational speeds of the engine, i.e. n=1400 rpm and n=2000 rpm.

Figure 4 shows the mean droplet diameters d_{kr} μm formed by the discharge of fuel from the nozzle hole of constant cross–through, designated by Sauter for two rotational speeds of the speeds of the engine crankshaft, i.e. n=1400 rpm and n=2000 rpm.

Figure 5 shows a comparison of fuel streams decay time t_r ms for two rotational speeds of the engine crankshaft, i.e. n=1400 rpm and n=2000 rpm.

Figure 6 shows a comparison of the instantaneous penetration stream S_1 mm for $t \le t_r$ at n=1400 rpm and at n=2000 rpm.

Figure 7 shows the instantaneous penetration of the jet (secondary) S_2 mm for the engine speed n=1400 rpm and for n=2000 rpm.

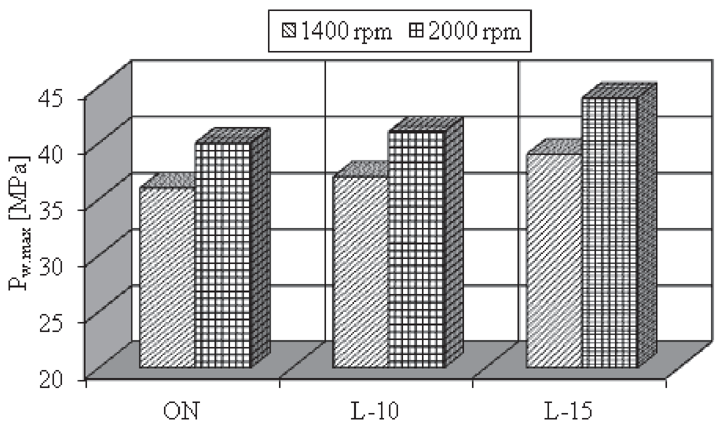

Fig. 2. Comparison of the maximum fuel pressure in the injector $P_{w.max.}$ for the rotational speed of the engine crankshaft n=1400 rpm and n=2000 rpm

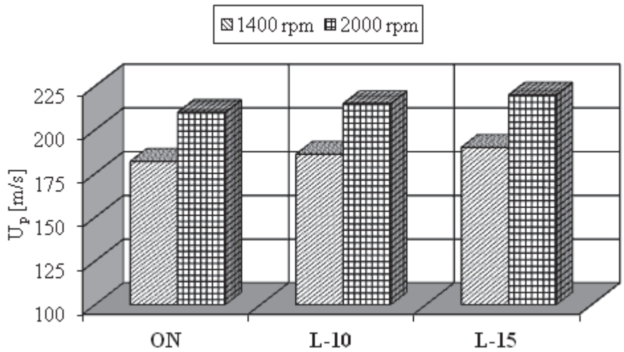

Fig. 3. Comparison of the initial flow rate from the fuel injector U_p for the rotational speed of the engine crankshaft n=1400 rpm and n=2000 rpm

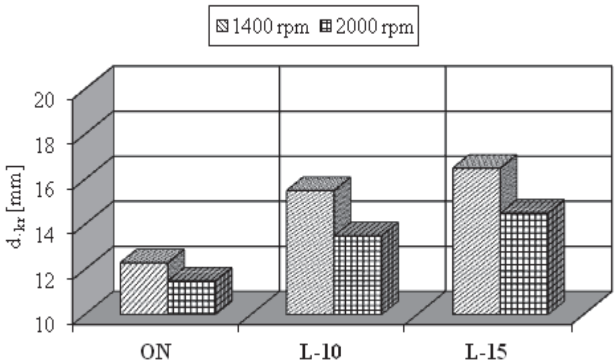

Fig. 4. Critical comparison of average droplet diameter d_{kr} for the rotational speed of the engine crankshaft n=1400 rpm and n= 2000 rpm

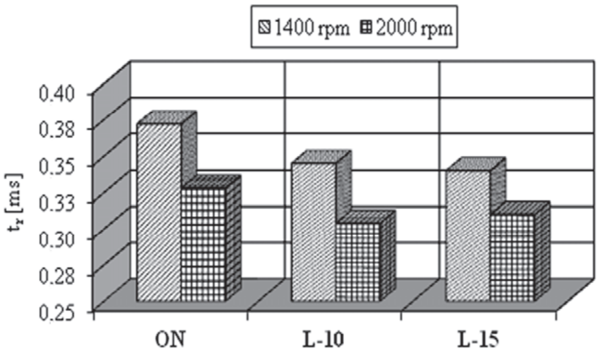

Fig. 5. Comparison of disintegration times of fuel streams t_r for the two engine speeds n=1400 rpm and n= 2000 rpm

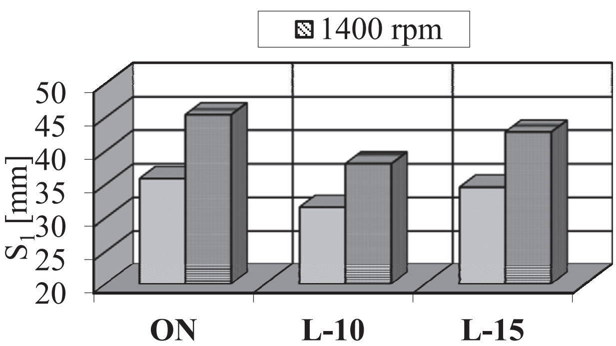

Fig. 6. Comparison of the initial instantaneous fuel stream penetration S_1 for $t \leq t_r$ and for two engine speeds n=1400 rpm and n=2000 rpm

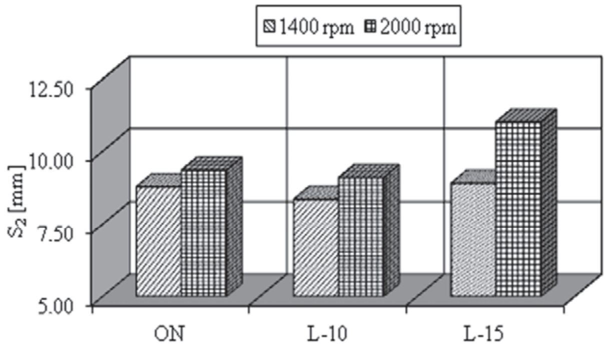

Fig. 7. Comparison of instantaneous penetration of secondary stream S_2 for $t>t_r$ and for two engine speeds: n=1400 rpm and n=2000 rpm

Conclusions

Based on the results obtained from the study can draw the following conclusions:

1. The highest average maximum fuel pressure in the injector $P_{w.max.}$ showed a mixture of L–15, and the lowest pressure occurred for hydrocarbon fuel ON (diesel oil) both the crankshaft rotational speed n=1400 rpm and n=2000 rpm. The absolute percentage difference between fuels, L–15 and ON is between 3-10%. This difference is due to the compressibility and variable density and viscosity of the tested fuels (Fig. 2).
2. Initial speed fuel injected U_p are the most fuel L–15, and the smallest of the fuel ON both the crankshaft rotational speed n=1400 rpm and n=2000 rpm. The absolute percentage difference between the two is, of 3-8%, which was due to the different densities and viscosities of fuels and compressibility (Fig. 3).
3. Research of average diameter of fuel droplets d_{kr}, have shown that smaller diameters and fuel ON at a speed of n=2000 rpm and n=1400 rpm are the most fuel L-15. The absolute percentage difference between the two is, of 21-28%. This is due to changes in engine load, which affects the fuel pressure in the injector and the density, viscosity and surface tension (Fig. 4).
4. Comparison of the stream disintegration times of fuel droplets t_r in the combustion chamber found that there are times for a less fuel L-10 and L-15 and are comparable both at speed n=1400 rpm and n=2000 rpm. Larger disintegration times are on the fuel supply to ON. Absolute percentage differences between the used fuel is about 10%. Due to the different size of the fuel pressure in the injector depending on the load and different density, viscosity and surface tension (Fig. 5).
5. The range of fuel streams S_1 ($t \le t_r$) for all subjects at the crankshaft rotational speed n=1400 rpm and n=2000 rpm fuel consumption is higher for ON and the smallest for fuel L–10 and L-15, which are almost comparable. Absolute percentage differences between the fuels used are from 6-8%. The reason for this is the different size of the counter in the cylinder and the fuel pressure in the injector depending on engine load and varying density, viscosity and surface tension (Fig. 6).
6. The range of fuel jets S_2 ($t>t_r$) for all subjects at the crankshaft rotational speed n=1400 rpm is comparable. However, when n=2000 rpm range jet fuel is the largest for the L-15 and the smallest for fuel L-10 and ON, which are almost comparable. Absolute percentage differences between the fuels used are from 20-25%. This is due to the different size of the counterpressure cylinder and the fuel pressure in the injector depending on the engine load, and a different density, viscosity and surface tension (Fig. 7)
7. It is appropriate to continue the research on the evaluation of the impact of the motor fuels both the mineral and vegetable on economic, energy and environmental indicators of the engine.

8. When assessing the economic aspect of the use of vegetable oil esters and their mixtures of hydrocarbon fuels is advisable to reduce the cost of production and distribution, so that prices of these fuels were comparable to those of hydrocarbon fuels.

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