Experimental study of diesel engine running on petroleum diesel fuel and rapeseed oil

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Abstract. The article discusses the problem of replacing traditional diesel fuel with an alternative one. Rapeseed oil is considered as an alternative motor fuel. A deterioration in the quality of injection and atomization processes was noted when converting the engine to rapeseed oil. Experimental studies of the MD-6 diesel engine have shown that it can operate effectively on both petroleum diesel fuel and rapeseed oil. At the same time, the transfer of this engine from petroleum diesel fuel to rapeseed oil was accompanied by an increase in brake-specific fuel consumption, an increase in exhaust gas opacity, and an increase in emissions of nitrogen oxides and carbon monoxide. These negative phenomena are partly eliminated by adjusting the fuel injection advance angle. Improved engine performance is also achieved by optimizing the characteristics of the fuel supply system and two-phase fuel supply. The implementation of these measures makes it possible not only to achieve the performance of a diesel engine characteristic of running on diesel fuel, but also to meet the requirements for fuel efficiency and exhaust gas toxicity that are promising for small-sized high-speed diesel engines.

1 Introduction

At the current stage of development of transport power plants with internal combustion engines, one of the main problems is achieving the required environmental characteristics of these plants [1, 2, 3]. Effective means of achieving the required toxicity indicators of exhaust gases (EG) of diesel engines include the use of various alternative fuels [4, 5, 6]. Promising alternative fuels for diesel engines include various biofuels. Among the biofuels that have found the greatest application in diesel engines, vegetable oils and their derivatives - methyl, ethyl and butyl ethers - should be highlighted [6, 7, 8]. Research continues on the operation of diesel engines using vegetable oils and their mixtures with diesel fuel, including the search for solutions to injection and atomization problems [9-16].

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Currently, the share of vegetable oil use in the fuel sector is so small that its increase will not have any impact on food security. Especially considering the possibility of using technical (not used for food purposes) plant varieties. Expired, used (for example, frying oils) or low-quality oils can be used as fuel.

There are two possible approaches to the production of alternative fuels, depending on the complexity of their production. When it is high labour intensive, requires high pressure, expensive catalysts and others (when processing into methyl and butyl ethers), centralized production is used. Due to their properties, such as high cetane number, viscosity and density, esters can be used both as an independent fuel and as a mixed fuel. Decentralized production, carried out directly at the point of consumption, does not involve a large degree of processing and consists, as a rule, of purification from water and impurities. This is usually used in agro-industrial complexes where there is proximity to raw materials.

When converting diesel engines to run on vegetable oils, the operating process of diesel engines is transformed. This is caused by the following factors: vegetable oils have a heavier fractional composition, due to the presence of a large number of fatty acids, increased density, viscosity and compressibility, and worse volatility. As a result, most of the indicators of injection, atomization and mixture formation worsen, namely, the average diameter of fuel droplets increases, the opening angle of fuel jets decreases, and the range of fuel jets increases; the proportion of film mixture formation increases, the structure of the fuel stream in the cross section becomes more uneven. At the same time, diesel engines with divided and semi-divided combustion chambers, in which the requirements for the quality of the fuel atomization process are less stringent, are more suitable for operating on vegetable oils. The purpose of the study is a comparative assessment of the parameters of a diesel engine with a semi-separated combustion chamber when running on petroleum diesel fuel and rapeseed oil.

2 Research on the parameters of a diesel engine running on petroleum diesel fuel and rapeseed oil

Small-sized high-speed diesel engines with direct fuel injection are increasingly used as power units for small-scale mechanization, mobile generators, compressors, pumping units, etc. When adapting these diesel engines to operate on vegetable oils and fuels based on them, the problem arises of matching the geo-metric characteristics of the jets of atomized fuel with the shape of the combustion chamber (CC), which is associated with differences in the properties of vegetable oils from the properties of petroleum diesel fuel. The study confirmed the differences in the physical and chemical properties of petroleum diesel fuel (DF) and rapeseed oil. In particular, under normal atmospheric conditions, the viscosity of oil diesel fuel of average composition is $\nu=3-4$ mm$^2$/s, and that of rapeseed oil of average composition is $\nu=80$ mm$^2$/s. The density of these is equal to $\rho=830$ and 916 kg/m$^3$, respectively. Under the same conditions, the instantaneous (true) compressibility coefficient of petroleum diesel fuel is about $\alpha=70 \cdot 10^{-11}$ Pa$^{-1}$, and for rapeseed oil it is equal to $\alpha=52 \cdot 10^{-11}$ Pa$^{-1}$. Under normal atmospheric conditions, the surface tension coefficient of petroleum diesel fuel is $\sigma=27.1$ mN/m, and that of rapeseed oil is $\sigma=33.2$ mN/m.

To assess the influence of the properties of rapeseed oil on the flow of work processes, experimental studies were carried out on a small-sized high-speed diesel engine type MD-6 (1h 8.0/7.5) without supercharging, which has a combustion chamber in the piston (Figure 1) and volume-film mixture formation.
The standard fuel supply system of the diesel engine under study contains a single-plunger high-pressure fuel pump (HPF) with a spool-type plunger pair with a plunger diameter $d_{pl}=6$ mm, plunger stroke $h_{pl}=6$ mm and a mushroom-type injection valve. The fuel pump is connected through a high-pressure injection pipeline with a length of $L=400$ mm to a closed-type multi-nozzle injector with an injection start pressure $p_{inj}=21$ MPa. Some technical characteristics of the MD-6 diesel engine are presented in Table 1. The test bench for conducting experimental research was created on the basis of the MD-6 diesel engine as part of the AD4-T400-V mini-power plant, i.e. The operation of a diesel engine was studied directly as part of an electrical unit that generates electric current. The engine was loaded by a generator included in the power plant, when electrical consumers of various powers were connected to it. The adopted engine loading scheme made it possible to determine the electrical power (as the product of current and voltage) and the effective engine power, taking into account the generator efficiency. The MD-6 diesel engine as part of the power plant operated in regulatory characteristic modes at almost a constant crankshaft rotation speed of $n=3000$ min$^{-1}$.

Table 1. Technical characteristics of MD-6 diesel engine.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Meaning</th>
</tr>
</thead>
<tbody>
<tr>
<td>engine's type</td>
<td>four-stroke, diesel</td>
</tr>
<tr>
<td>number of cylinders</td>
<td>1</td>
</tr>
<tr>
<td>cylinder diameter/piston stroke, mm</td>
<td>80/75</td>
</tr>
<tr>
<td>cylinder displacement, l</td>
<td>0.377</td>
</tr>
<tr>
<td>compression ratio</td>
<td>19.3</td>
</tr>
<tr>
<td>mixture formation method</td>
<td>direct injection</td>
</tr>
<tr>
<td>timing mechanism</td>
<td>valve type with top valve arrangement</td>
</tr>
<tr>
<td>valve overlap angle, degrees of crankshaft rotation, degrees c.c.</td>
<td>38-40</td>
</tr>
<tr>
<td>supply system</td>
<td>high pressure fuel pump with all-mode centrifugal regulator</td>
</tr>
<tr>
<td>rated speed, min$^{-1}$</td>
<td>3000 ± 20</td>
</tr>
<tr>
<td>liter power, kW/l, not less</td>
<td>11</td>
</tr>
<tr>
<td>engine operating power at $n=3000$ min$^{-1}$, kW, not less</td>
<td>4.0</td>
</tr>
</tbody>
</table>
Brake-specific fuel consumption at maximum power mode, g/(kW·h), no more | 280
---|---
degree of unevenness of the speed controller, %, no more | 5
average piston speed, m/s | 7.5
crankshaft rotation speed at idle speed, min⁻¹:
- minimum | 1200 ± 100
- maximum, no more | 3200
maximum torque value, N·m, not less | 13

The test bench contained a complex of control and measuring equipment. To measure the soot content in the exhaust gas, a Bosch soot meter type EFAW-65 was used. To measure the content of toxic components of exhaust gas - nitrogen oxides NOx and carbon monoxide CO, a TESTA 350 gas analyzer was used. The concentration of these toxic components in dry exhaust gas was determined in ppm (parts per million by volume). To indicate the diesel operating process, its cylinder head was modified to install a piezoquartz cooled pressure sensor type 8QP505C, which has high temperature stability and small size. The installation diagram of the pressure sensor in the MD-6 diesel cylinder head is shown in Figure 1.

To evaluate the type of fuel on the performance of an MD-6 diesel engine, its experimental studies were carried out. The engine's experimental research program included testing its load characteristics when running on diesel fuel and rapeseed oil. The characteristics were measured by successively increasing the load from minimum to maximum at a constant crankshaft rotation speed n=3000 min⁻¹ at setting fuel injection advance angles (FIAA) $\theta=25$ and 33° crankshaft rotation (c.c.) to TDC. FIAA equal to $\theta=33^\circ$ c.c. before TDC, was optimal from the point of view of fuel efficiency indicators. FIAA equal to $\theta=25^\circ$ c.c. before TDC, ensured a decrease in the dynamics of the fuel combustion process - the ignition delay period $\tau_i$ and the maximum combustion pressure $p_z$ and the rate of pressure increase during combustion $dp/d\varphi$. These indicators were determined by taking indicator diagrams of the pressure of the working charge in the engine cylinder (the change in pressure in the cylinder along the angle of c.c. $\varphi$). The ignition delay period was determined when indicating the working process directly from the indicator diagram (Figure 2), as the segment from the beginning of fuel injection into the combustor until the curve separates from the compression line.

![Fig. 2. Indicator diagrams of the MD-6 diesel engine operating in a mode with a rotation speed of $n=3000$ min⁻¹ and an average indicator pressure $p_i=0.51$ MPa on various fuels at different UOVT $\theta$: 1 - diesel fuel, $\theta=25^\circ$ p.c.v. to TDC; 2 - rapeseed oil, $\theta=25^\circ$ c.c. to TDC; 3 - rapeseed oil, $\theta=33^\circ$ c.c. to TDC.](image-url)
Shown in Figure 3 characteristics of the indicators of the dynamics of the fuel combustion process indicate that at a constant FIAA θ=25° c.c. before TDC, the transfer of the MD-6 diesel engine from DT to RM in the studied modes is accompanied by an increase in the ignition delay period $\tau_i$ by 0.3-0.4 ms. This is explained by the lower cetane number of RM compared to DT (their cetane numbers are 36 and 45 units, respectively). However, in this case, the maximum combustion pressure $p_z$ of rapeseed oil turned out to be somewhat lower compared to work on diesel fuel, and the difference in the rate of pressure increase during combustion $dp/d\phi$ depends on the load mode. In modes with low loads ($p_e<0.2$ MPa) this figure is higher when using DT, and in modes with medium and high loads ($p_e>0.2$ MPa) - when using PM.

![Figure 3: Dependence of the indicator period of ignition delay $\tau_i$, the rate of pressure rise $dp/d\phi$ and the maximum combustion pressure $p_z$ of the MD-6 diesel engine on the load (average effective pressure $p_e$) when the engine operates on various fuels at different UOVT $\theta$: 1 - diesel fuel, $\theta=25°$ c.c. to TDC; 2 - rapeseed oil, $\theta=25°$ c.c. to TDC; 3 - rapeseed oil, $\theta=33°$ c.c. to TDC.](image)

Research has confirmed that the dynamics of the combustion process strongly depend on the FIAA. When the MD-6 diesel engine operates at PM and increases the UOVT $\theta$ from 25 to 33° c.c. before TDC, the ignition delay period $\tau_i$ increases by 2.5-3.0 ms (see Figure 3). In particular, in full load mode ($p_e=0.42$ MPa) the ignition delay period was $\tau_i=1.47$ ms. As $\tau_i$ increases, the amount of fuel in the mixture with air, prepared for the combustion process and burned in its first stage, increases. This leads to an increase in the maximum cycle pressure $p_z$, the rate of pressure increase $dp/d\phi$ and a decrease in the exhaust gas temperature $T_G$. At the specified load mode ($p_e=0.42$ MPa) these parameters are respectively $p_z=0.84$ MPa, $dp/d\phi=0.75$ MPa/deg and $T_G=460°$ C. As a result of such an increase in the FIAA, the effective engine efficiency $\eta_e$ increases, and the Brake-specific fuel consumption (BSFC) $g_e$ decreases (Figure 4).
Fig. 4. Dependence of the hourly fuel consumption $G_t$ of the MD-6 diesel engine, its effective efficiency $\eta_e$ and brake-specific fuel consumption $g_e$ on the load (average effective pressure $p_e$) when the engine operates on various fuels at different UOVT $\theta$: 1 - diesel fuel, $\theta=25^\circ$ c.c. to TDC; 2 - rapeseed oil, $\theta=25^\circ$ c.c. to TDC; 3 - rapeseed oil, $\theta=33^\circ$ c.c. to TDC.

When the MD-6 diesel engine operated at full load ($p_e=0.42$ MPa) on oil diesel fuel, its effective efficiency was $\eta_e=0.30$, and the minimum BSFC $g_e=290$ g/(kW·h) (Figure 4), which corresponds to hourly fuel consumption $G_t=1.1$ kg/h and air consumption $G_v=31$ kg/h. In this case, the exhaust gas temperature was $T_G=380^\circ$C, and the exhaust smoke was equal to $K_X=2.3$ units on the Bosch scale (Figure 5). When converting a diesel engine to PM, its BSFC in this mode increased to $g_e=440$ g/(kW·h), and the effective efficiency decreased to $\eta_e=0.22$. Such a significant deterioration in the fuel efficiency of a diesel engine when operating on PM can be prevented by increasing the FIAA to $\theta=33^\circ$ c.c. to TDC. When the MD-6 diesel engine was operating at full load ($p_e=0.42$ MPa) on a RM with such a UOVT, its BSFC turned out to be equal to $g_e=350$ g/(kW·h), and the effective efficiency $\eta_e=0.26$.

Some change in the operating process parameters of the MD-6 diesel engine when operating on a PM also leads to a change in exhaust gas toxicity indicators. When using this type of fuel, an increase in the proportion of fuel burned in the first stage leads to an increase in local temperatures in the combustion chamber, which increases the rate of formation of nitrogen oxide. So, in a diesel engine operating on PM at rated load mode ($p_e=0.42$ MPa) at $\theta=33^\circ$ c.c. before TDC, the concentration of nitrogen oxides in the exhaust gas $C_{NOx}$ is 1550 ppm (Figure 5). However, at $\theta=25^\circ$ c.c. Before TDC, the NOx content in the exhaust gas decreases to $C_{NOx}=1080$ ppm, which is approximately equal to the concentration of this component in the exhaust gas when operating on diesel fuel.
Dependence of exhaust smoke $K_x$ the content of nitrogen oxides $CNO_x$ and carbon monoxide $CCO$ in the exhaust gas of MD-6 diesel engine on the load (average effective pressure $p_e$) when operating on various fuels at different UOVT $\theta$: 1 - diesel fuel, $\theta=25^\circ$ c.c. to TDC; 2 - rapeseed oil, $\theta=25^\circ$ c.c. to TDC; 3 - rapeseed oil, $\theta=33^\circ$ p.k.v before BMT.

Converting the MD-6 engine to rapeseed oil leads to a significant increase in carbon monoxide emissions. When the diesel engine is operating at full load $(p_e=0.42 \text{ MPa})$ on the RM with UOVT $\theta=25^\circ$ c.c. before TDC, the content of carbon monoxide in the exhaust gas is $CCO=9000 \text{ ppm}$, which is an order of magnitude higher than that of a diesel engine running on diesel fuel (see Figure 5). The emission of this toxic exhaust gas component can be significantly reduced by increasing the FIAA. So, when the diesel engine is operating at full load on the PM at $\theta=33^\circ$ c.c. before TDC, the carbon monoxide content decreases significantly and is equal to $CCO=2600 \text{ ppm}$. This decrease in the concentration of carbon monoxide $CO$ is explained by a decrease in fuel consumption $G_T$ (Figure 4) and a corresponding increase in the amount of oxygen in the diesel combustion chamber, as evidenced by the experimentally obtained values of the excess air coefficient $\alpha$ and the oxygen content $C_{O_2}$ in the exhaust gas. When operating an MD-6 diesel engine with FIAAs $\theta=25^\circ$ and $33^\circ$ c.c. before TDC, the excess air coefficient turned out to be 1.2 and 1.6, respectively, and the oxygen content in the exhaust gas was 6.0 and 10.0%. In the same mode at $\theta=33^\circ$ c.c. before TDC, exhaust smoke decreases by 13% and amounts to $K_x=4$ units on the Bosch scale (see Figure 5). A decrease in exhaust opacity with an increase in the UOVT is also associated with an increase in the oxygen content in the diesel combustion chamber.

3 Conclusions

As shown in the study, efficient diesel operation is possible not only on petroleum diesel fuel, but also on rapeseed oil. At the same time, converting this engine from diesel to PM leads to a deterioration in both BSFC and exhaust gas toxicity indicators. The reason for this is that in small-sized engines running on diesel fuel, when changing the type of fuel, the fuel supply processes change - fuel injection and atomization, its self-ignition and subsequent
combustion. As a result, there is a discrepancy between the shape of the combustion chamber and the fuel supply parameters, and the quality of the fuel injection and atomization processes, mixture formation and combustion are disrupted. Therefore, to achieve the required indicators of fuel efficiency and exhaust toxicity, it is necessary to implement measures that improve the quality of the diesel operating process.

The analysis showed that in addition to regulating the FIAA, such measures include optimizing the characteristics of the fuel supply system (in particular, the geometry of the flow part of the injector nozzle), two-phase fuel supply, and supplying part of the fuel to the engine intake manifold. A significant reserve for improving diesel performance is the transition from “pure” PM to mixed biofuels and optimization of their composition. The implementation of these measures will make it possible not only to achieve the performance of the diesel engine under study, characteristic of operation on diesel fuel, but also to meet the requirements for fuel efficiency and exhaust toxicity that are promising for small-sized high-speed diesel engines.

References

5. B. Kegl, M. Kelg, S. Pehan, *Green Diesel Engines Biodiesel Usage in Diesel Engines* (Spring, 2013)