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Diesel engine and fuel-supply system characteristics for testing ethanol as additive fuel

J. Olt and V. Mikita

Institute of Technology, Estonian University of Life Sciences, Kreutzwaldi 56, EE51014 Tartu, Estonia; e-mail: villu.mikita@emu.ee

Abstract. The use of ethanol as additive fuel requires special preparation of the internal combustion engine. In research work it is important to follow the technical rules for the test engine provided by the manufacturer. In this article, for the purposes of testing the local ethanol fuel made of lignocellulose raw materials, a dual fuel-supplying system test engine D–120 was used. During the tests diesel fuel pilot injection was controlled by a standard fuel-supply system. Ethanol fuel for the operation of the engine was supplied through an inlet manifold with the help of a carburettor. In this article the technical specification for the test engine D–120, for engine and fuel-supply system adjustment characteristics and formulas for calculating the output parameters will be presented. The paper includes also the comparison of the value of output parameters of the engine depending on the ratio of the fuel supply equipages. The results of testing the use of the fuel mixture of ethanol and diesel fuel together with the suitable ratio of fuel mixture for diesel engine operation will be presented as well.

Key words: Engine preparing for additive fuel testing, quantitative and qualitative fuel mixing methods, adjustment characteristics and output parameters of the fuel-supply system and diesel engine, ethanol and diesel fuel mixture ratio.

INTRODUCTION

Today in the European Union one of the research priorities is to work out new alternative fuels and to implement their use in internal combustion engines. At present there is a wide classification of alternative fuels used in internal combustion engines (Bosch, 2007). The most common additive fuel for piston engine is ethanol, which is used as a standalone fuel, but also as an additive in other types of fuel (Harndorf, H., 2008; Steinbach, N., 2006). The wide use of ethanol in diesel engines is hindered by its physical-chemical properties and the nature of the Diesel cycle (Merker et al., 2004). It is also possible to evaluate the use of ethanol as additive fuel by engine testing methods. This can be done by comparing the output parameters of the combustion process of the engine and indicator factors by using different fuel mixtures. For that purpose a research engine must be equipped with an additional device for the delivery of ethanol; in addition, also high quality measuring instruments and non-standard test methods are necessary. The novelty of the solution suggested by the authors of this article lies in the two fuel-supply systems of the test engine: the main fuel-supply system which ensures the pilot injection of base fuel and supplementary fuel-supply system delivering ethanol fuel mixtures with varying composition and volume. This solution ensures that the engine starts up easily and will work in a wide range of operating conditions. Ethanol fuel is directed to the work process via the inlet manifold using carburettor type K–22Г with a special coiled tube.

The research is done with the minimum ratio of fuel mixture for the test engine at which the engine operates satisfactorily in a wide range of load modes. The fuel mixture consists of standard diesel fuel and 96% pure ethanol. The loss of power due to the low level of diesel fuel injection is compensated by adding ethanol. The engine testing was carried out in laboratory conditions.

MATERIALS AND METHODS

Bringing the technical conditions of the diesel fuel-supplying system into conformity

The test object obtained for the lab, a diesel engine D–120, had initially several technical faults from the manufacturer. The most significant of those was a high vibration level at all operating modes. In order to reduce this shortcoming, the high-pressure pipes of the diesel fuel-supplying system (hereinafter DFS) and the injector nozzles from the manufacturer were replaced, the in-line injection pump fuel delivery stroke and the variations between its sections were brought into conformity with the rated value and the injection pressure of the injectors was adjusted.

Bringing the technical conditions of the engine into conformity

With the help of an indicator device the top dead centre (TDC) of the first cylinder of the engine was specified, the position of its dial and the numerical values of the static fuel delivery angle (hereinafter SFDA) on the crankshaft pulley were determined. The expansion gaps of the timing gear were brought into conformity with the technical specifications of the engine. In order to determine the optimal SFDA, the according adjustment characteristic was performed. The test data were issued in the form of tables and spread indicator diagrams. The SFDA was chosen based on the values of the parameters of maximum engine power and minimum specific fuel consumption (Fig. 1) and specified (Taylor C. F. V1 and V2, 1998) based on the regularity of fluctuation Indicator pressure (Fig. 2) ($p_{z.max}$ = 52 bar; α_{ca} = 9° after top dead center (ATDC)).

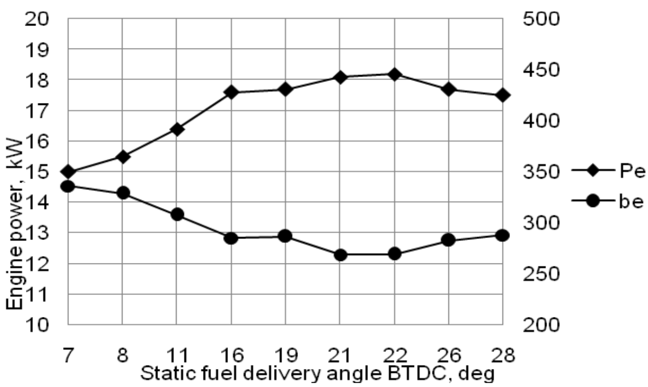

Figure 1. Dependence of engine power and specific fuel consumption on the SFDA in crank angle degrees.

The selected optimal SFDA was 21 crank angle degrees before top dead center (BTDC) of the first cylinder. The data of the test engine have been presented in Table 1.

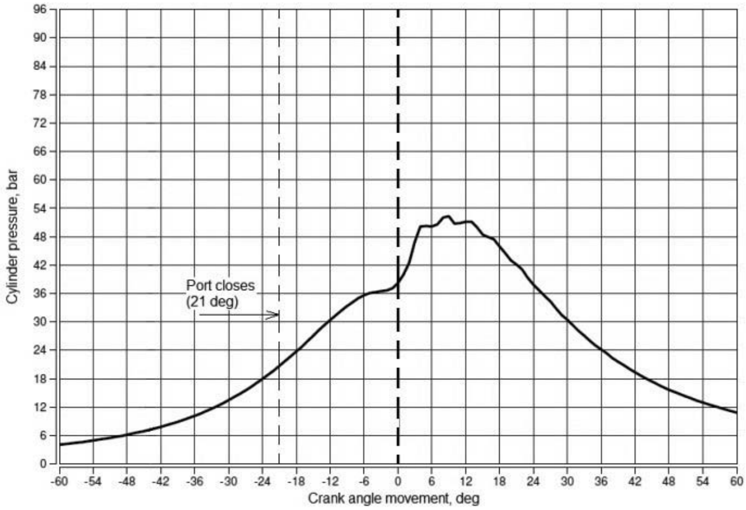

Figure 2. Indicator diagram of diesel engine D–120 – the optimal static fuel delivery angle.

Table 1. Technical specifications of diesel engine D–120**.**

Number of cylinders	2
Cylinder diameter	105mm
Piston stroke	120mm
Volume	2.08 liter
Pump fuel delivery	59 ± 2mm^3 stroke^{-1}
Power	18.4kW
Maximum torque	99.5Nm
Pressure ratio	16.5
Nominal rotational speed	1,800 ± 27 rpm
Maximum torque achieved between	1,260 – 1,400 rpm
Maximum rotational speed	1,950 rpm
Minimal rotational speed	800 – 1,050 rpm
Specific fuel consumption	245g kWh^{-1}
Fuel consumption nominal	6.37kg h^{-1}
Fuel consumption on maximum idle	1.9kg h^{-1}
Cooling system	air-cooled

Tuning the DFS and determining the governor characteristics

In order to control the DFS pump fuel delivery, the in-line injection pump was equipped with an auxiliary device, which enabled to control the position of the rack to the accuracy of one millimeter. The adjustment characteristics of the injection pump based on the amount of fuel delivered were performed in the following modes: n_{fp} = 400; 500; 600; 700; 800; 900 and 970 rpm. The regularities of pump fuel delivery have been brought out in Fig. 3. Based on the achieved test results the pump fuel delivery stroke volumes necessary for the characteristics of the mixture formation of the engine were found. The graph of Fig. 3 shows that the in-line injection pump fuel delivery fluctuates linearly depending on the position of the rack.

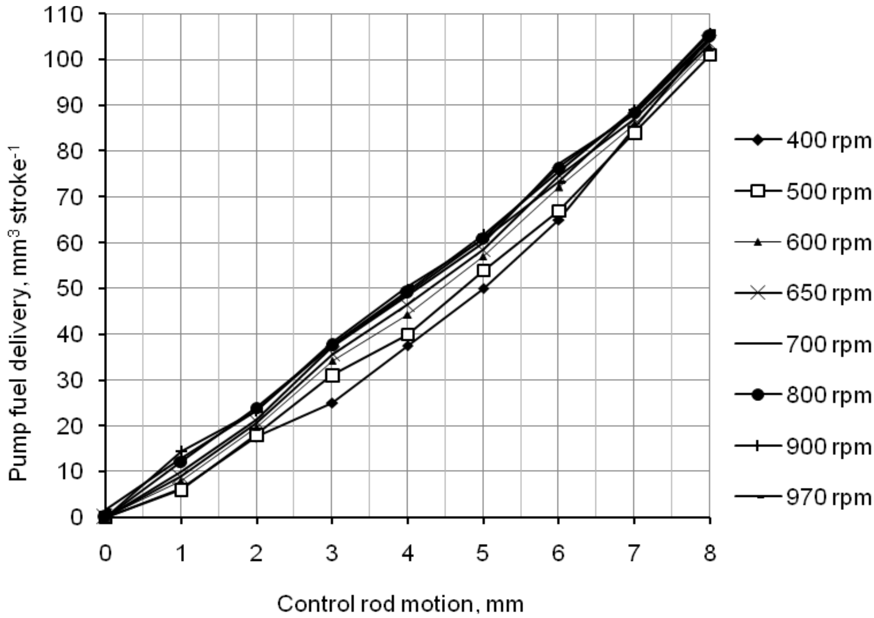

Figure 3. Dependence of in-line injection pump fuel delivery on the position of the rack.

The presented characteristic reveals the technical condition of the in-line injection pump 2УТНИ 11100515 as follows:

a) injection pump has been completed with high quality precision pairs;
b) pump fuel delivery varies at operating modes with varying rack position as a linear function.

The test results show that the changes in the volumes of pump fuel delivery are within maximum permissible error starting from the rotational speed of the camshaft n_{fp} = 650 rpm. Based on that, the determined minimum camshaft rotational speed for the following engine tests will be $n_e = 2 \cdot n_{fp}$, which is n_e = 1,300 rpm. In order to determine the output parameters of the injection pump its full and part governor characteristics were performed, which have been presented in Fig. 4. Based on the presented graphs the more typical speed modes will be determined for the diesel engine D–120. The following formulas were used for calculating the output parameters of the fuel-supplying equipment.

Formula for calculating pump fuel delivery (mm^3 stroke^{-1}):

$$V_f = \frac{10^3 \cdot \sum_1^4 V_i}{n_c}, \tag{1}$$

where V_i is capacity of section i (cm^3 min^{-1}) and n_c is total number of cycles.

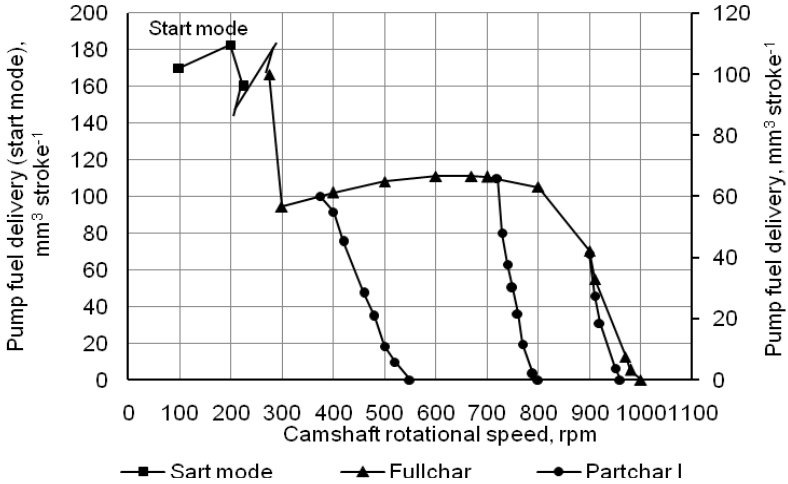

Figure 4. Full and part characteristics of the in-line injection pump 2УТНИ 11100515.

The relative variation factor (%) of the pump fuel delivery was determined by the following formula:

$$\sigma_k = \frac{2(V_{i.\max} - V_{i.\min})}{V_{i.\max} + V_{i.\min}} \cdot 100\,\%, \tag{2}$$

where $V_{i.min/max}$ is the minimum and maximum productivity of sections (cm^3 min^{-1}).

The fuel consumption (kg h^{-1}) was calculated by the following formula:

$$B_f = (6 \times V_f \times i_c \times n_{fp} \times \rho_f) / 10^5, \tag{3}$$

where ρ_f is the density of fuel (g cm^{-3}), i_c is the number of cylinders in the engine and n_{fp} is the rotational speed of the pump's camshaft, rpm.

Preparation of the diesel engine for research

For the purposes of the research the diesel engine was equipped with the necessary measuring instruments and the carburettor K-22Г for the delivery of ethanol, presented in Fig. 5. Measurements performed on the test engine D–120: a) exhaust gas temperature by shielded thermocouple, to the accuracy of 1°C; b) the SFDA to the accuracy of one degree by the crank angle degree; c) indicator pressure in the second cylinder of the engine with a Kissler 701A type sensor, to the accuracy of 1 bar; d) crank angle degree with the device Indimodul, to the accuracy of one degree by the crank angle degree; e) ingredients and smoke opacity of the engine's exhaust gas with the gas analyzer Bosch BEA 150, respectively, to the accuracy of 0.001 percent by volume and 0.01 m^{-1}; f) exposure conditions: barometric pressure, air temperature and air humidity, respectively, to the accuracy of one mm Hg; 0.1°C and 1%.

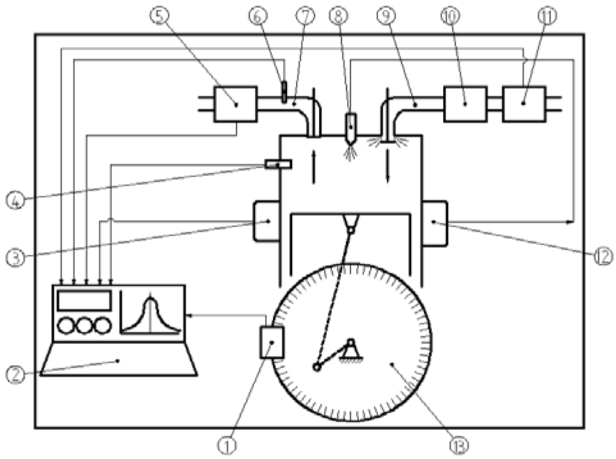

Figure 5. Diesel engine D-120 and its measurement and auxiliary devices: 1 – sensor of crankshaft movement angle and sensor of rotational speed; 2 – engine control remote with AVL Indimodul device; 3 – engine unit for pressure and temperature sensors; 4 – pressure sensor Kissler 701A; 5 – opacimeter BAE 350-FIN; 6 – exhaust gases temperature sensor; 7 – exhaust manifold; 8 – injector; 9 – inlet manifold; 10 – carburettor K-22 Г; 11 – air consumption meter SuperFlow 6-1490; 12 – diesel supply equipment with control system Horiba ATS LFM 2003; 13 – graduated scale.

The output parameters of the engine were calculated using the following formulas (Martyr & Plint, 2007):

Engine power (kW):

$$P_e = \frac{T_e \cdot n_e}{9{,}550}, \tag{4}$$

where T_e is the engine's torque (Nm) and n_e is the engine's rotational speed (rpm).

Specific fuel consumption (g kWh^{-1}):

$$b_e = \frac{1{,}000 \cdot B_f}{P_e}, \tag{5}$$

where B_f is the fuel consumption (kg h^{-1}).

Excess air ratio:

$$\lambda_a = \frac{B_a}{(14.3 \cdot B_f)}, \tag{6}$$

where B_a is the actual air consumption (kg h^{-1}).

Mean effective pressure (MPa):

$$p_e = \frac{3.14 \cdot 10^{-3} \cdot \tau_t \cdot T_e}{\sum V_h}, \tag{7}$$

where τ_t is the number of strokes of the engine and $\sum V_h$ is the engine-swept volume of cylinders (l).

Engine efficiency:

$$\eta_e = \frac{3{,}600}{(Q_a \cdot b_e)}, \tag{8}$$

where Q_a is the lower calorific value of diesel fuel (MJ kg^{-1}) and b_e is the specific fuel consumption (g kWh^{-1}).

Mechanical efficiency:

$$\eta_m = \frac{P_e}{P_e + P_{mk}}, \tag{9}$$

where P_{mk} is the friction losses power (kW).

Indicated thermal efficiency:

$$\eta_i = \frac{3{,}600}{(Q_a \cdot b_i)}, \tag{10}$$

where b_i is the indicated specific fuel consumption (g kWh^{-1}).

Volumetric efficiency:

$$\eta_v = \frac{33.3 \cdot B_a}{(\sum V_h \cdot n_e \cdot \rho_{env})}, \tag{11}$$

where ρ_{env} is environmental density (kg m^{-3}).

The comparison results of injection delivery characteristics over operating speed for different engine setups are presented in Table 2.

Table 2. Comparison data of injection delivery characteristics over operating speed for the engine D–120.

Parameter	unit	1,200	1,400	1,500	1,600	1,700	1,800	1,807	1,826	1,850
$b_{e.s}$	g kWh^{-1}	2704	282.8	284.8	284.8	280.3	278.1	281.4	286.4	495.6
$b_{e.w}$	g kWh^{-1}	275.3	271.7	284	273.5	277.6	274.8	275.7	277.6	340
$\lambda_{a.s}$	-	2.23	2.17	2.13	2.24	2.39	2.78	2.85	3.55	5.14
$\lambda_{a.w}$	-	2.49	2.59	2.40	2.24	2.45	2.63	2.70	3.19	4.82
$p_{e.s}$	MPa	0.68	0.66	0.66	0.63	0.59	0.50	0.48	0.38	0.15
$p_{e.w}$	MPa	0.67	0.68	0.68	0.66	0.62	0.57	0.56	0.47	0.25
$\eta_{v.s}$	-	0.708	0.702	0.694	0.698	0.683	0.668	0.668	0.667	0.662
$\eta_{v.w}$	-	0.791	0.829	0.801	0.702	0.721	0.715	0.713	0.718	0.699
$\eta_{e.s}$	-	0.312	0.298	0.296	0.296	0.301	0.303	0.300	0.294	0.170
$\eta_{e.w}$	-	0.306	0.310	0.297	0.308	0.304	0.307	0.306	0.304	0.248

Note: As an example, $\eta_{e.s}$ and $\eta_{e.w}$ are respectively engine efficiency with the carburettor K-22 Γ and without it.

The purpose of performing the injection delivery characteristics over operating speed with differently equipped engine D–120 was to determine how much the resistance of the carburettor type K-22Γ with coiled tube affects the engine's output parameters. Analysis of the test data shows that the carburettor does not significantly disturb the operation of the diesel engine. The engine's air consumption and hence also the engine's volumetric parameters change to some extent, 8.5 and 9.0%, respectively. The remaining efficiency and economic parameters are within the limits of 3–4%. The reduction of air consumption and efficiency is due to the carbureting device which is necessary for creating negative air pressure in the device, in order to move the fuel-air mixture in the calibrated jets and to call forth the atomization of the formed fuel-air mixture which will then be directed to the engine's cylinder.

RESULTS AND DISCUSSION

The diesel engine's adjustment characteristics regarding the pump fuel delivery

The research tests of the fuel mixture containing ethanol mixture and diesel fuel were performed based on customized adjustment characteristics and load characteristics and the indicated output parameters. Using the above mentioned characteristics there were determined the relative regularity between the in-line injection pump regulator lever and the position of the rack, and also the corresponding engine output parameters. The adjustment characteristics were performed at load modes n_e = 1,840, 1,820, and 1,800 rpm. The engine output parameters were measured and calculated. The following regularities were found: a) at the engine's low, average and high loads one millimeter fluctuation in the position of the rack changes power 5kW, 3kW, and 2kW, respectively; b) the engine's pump fuel delivery fluctuation's dependence on the position of the rack is linear, which shows the high quality of the manufacturing of the injection pump section elements; c) the regularities in the change of the hourly consumption of fuel received on the DFS test bench and engine test bench are analogous and practically the same at low load modes; d) at average and higher load modes the regularities in the change of the fuel consumption vary up to 7%, which is caused by the raise of the indicator pressure in the cylinder of the engine and its effect on the injector's injection pressure, and by the reduction of the in-line injection pump volumetric efficiency due to the decreased rotational speed of the engine's crankshaft.

Determining the load characteristics of the test engine D–120

In order to determine the regularity in the fluctuation between the test engine's input and output parameters, a number of load characteristics were performed using standard diesel fuel with the following speed modes: n_e = 1,200; 1,300; 1,400; 1,500; 1,600; 1,700; and 1,800 rpm. During the tests the fluctuation in the parameters of the diesel engine's vibration, noise, and capacity of exhaust gas and the standard parameters were measured. Based on the engine's output parameters, the following test modes were selected: n_e = 1,300; 1,500; and 1,800 rpm. Load mode n_e = 1,300 rpm was selected as the maximum torque; load mode n_e = 1,500 rpm was in conformity with the best effectiveness and economic output parameters and load mode n_e = 1,800 rpm became the nominal mode. The test results show that specific fuel consumption at various speed modes significantly differs at low loads where the value of pump fuel delivery changes within the limits of V_f = 5–20mm^3 stroke^{-1}. As the mean effective pressure rises to 0.20 MPa and above, which corresponds to the pump fuel delivery value fluctuation within the limits of V_f = 20–50mm^3 stroke^{-1}, the values of special fuel consumption in the wide load fluctuation mode remain uniformly stable. The specific fuel consumption fluctuation graph shows that the minimum values of load modes are n_e = 1,300–1,500 rpm. The efficiency parameters of load characteristics are presented in Figs. 6 and 7.

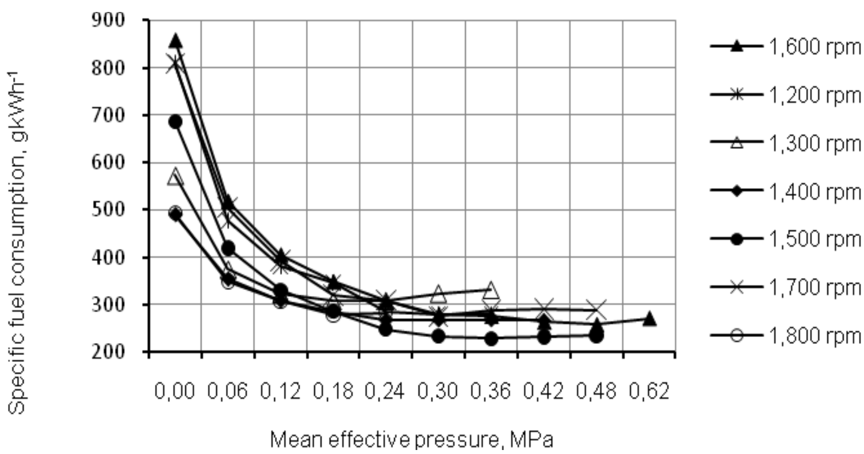

Figure 6. Regularity of specific fuel consumption fluctuation dependent on the mean effective pressure at varying load modes.

The analysis of the engine efficiency fluctuation regularity in Fig. 7 shows that $\eta_{e.max}$ achieves the maximum value at the mean effective pressure of 0.30 MPa and above, which also corresponds to the pump fuel delivery fluctuation within the limits of 25–50mm^3 stroke^{-1}. Engine efficiency achieves its maximum value $\eta_{e.max}$ = 0.37 with effective pressure 0.30–0.40 MPa. Engine efficiencies are at maximum with load modes n_e = 1,400 and 1,500 rpm. Based on the test result of load characteristics we selected the load modes n_e = 1,300; 1,500; and 1,800 rpm for the purpose of future researches of ethanol as an additive fuel. The fuel consumption of the engine (3) was compared to that determined by the position of the rack (Fig. 3). The regularity occurred where the fuel consumption during the engine test ($B_{f.etb}$) was lower than the fuel consumption received during the injection pump bench test ($B_{f.ptb}$):

$$B_{f.ptb} = k \times B_{f.etb}, \tag{12}$$

where k is a factor which takes into account the following parameters during the engine tests:

a) variance in the rotation speed of the engine's crankshaft and the resulting reduction of the high pressure line's volumetric efficiency;
b) interaction of the indicator pressure formed in the cylinder;
c) error in the measurement line, k = 1.15–1.35.

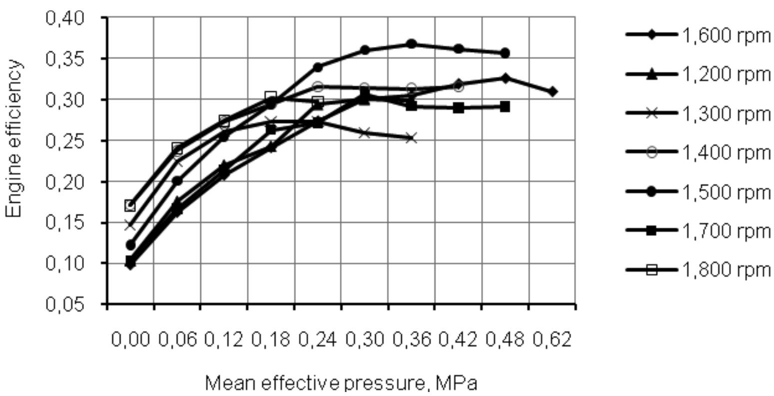

Figure 7. Regularity of engine efficiency fluctuation dependent on the mean effective pressure at varying load modes.

Diesel engine standard ethanol load characteristics

The ethanol-load characteristic for the engine D–120 was performed in the mode n_e = 1,800 rpm = const and load was increased according to the amount of added ethanol. The fuel mixture was formed of 96% ethanol and diesel fuel. The amount of pilot injection was kept at minimum and as stable as possible. The amount of ethanol was adjusted by changing the cross-section of the main metering jet. The test results are presented in Table 3. As the data show, by adding 96% ethanol to the fuel mixture the diesel engine operates up to 4kg h^{-1} without the engine's output parameters deteriorating and the temperature of the exhaust gas increasing by a significant amount.

The engine worked steadily up to the ethanol consumption $B_{f.et} = 5.5$kg h^{-1}. As the amount of ethanol increases, indicator pressure grows in the engine along with the fluctuation sound of its pressure waves, which has become known as *rumble* in diesel engines. With the increase of ethanol by 0.5kg h^{-1}, the main combustion phase of the combustion process becomes longer on the average by three crank angle degrees. In the indicator diagram's fast combustion phase, the pressure increases with the speed of 2 bar deg^{-1}, when using 96% purity ethanol.

Table 3. Engine D–120 test data when using 96% ethanol and diesel fuel mixture with standard load characteristics.

Parameter	Test 1	Test 2	Test 3	Test 4	Test 5	Test 6	Test 7
$B_{f.df}$, kg h^{-1}	2.1	2.1	2.5	2.5	2.5	2.4	2.4
$B_{f.et}$, kg h^{-1}	3.3	3.3	3.6	4.1	4.2	4.8	5.2
$\sum B_f$, kg h^{-1}	5.4	5.4	6.1	6.6	6.7	7.2	7.6
P_e, kW	13.5	14.0	17.0	19.0	19.0	21.0	21.0
	2.26	2.34	2.01	1.89	1.79	1.76	1.62
t_{egt}, ^{0}C	<200	<200	220	230	260	280	280

The data of test 1 show the initial situation of the load characteristic, where: 1) the test engine runs on the mode $n_e = 1,800$ rpm; 2) the main metering jet of the carburettor is closed and ethanol flows are passing via auxiliary jet of main metering jet; 3) the minimum ethanol consumption is $B_{f.\ et} = 3.3$kg h^{-1}; 4) the minimum diesel fuel consumption is $B_{f.df} = 2.1$kg h^{-1}; 4) the total amount of fuel consumption is $\sum B_f = 5.4$kg h^{-1}. At the present load mode, where the mixture ratio of $B_{f.df} : B_{f.et}$ = 1:1,6, the test engine gained the power to P_e = 13.5kW. The data of test 7 characterize the final test situation, where: 1) the diesel fuel consumption was kept approximately the same, $B_{f.df}$ = 2.4kg h^{-2}; 2) the ethanol consumption was maximum, $B_{f.\ et}$ = 5.2kg h^{-2}. Respectively, the mixture ratio of $B_{f.df} : B_{f.\ et}$ = 1:2.2 and the test engine gained power to P_e = 21.0kW.

Diesel engine customized ethanol load characteristics

Subsequently load characteristics were performed with the diesel engine D–120 operating at nominal rotation speed, while the output parameters were kept unchanged and the increase of torque due to the added ethanol was decreased by lowering the volume of pilot injection. The test results are presented in Table 4.

Table 4. Diesel engine customized ethanol load characteristics.

Parameter	Test 1	Test 2	Test 3	Test 4	Test 5	Test 6	Test 7	Test 8	Test 9
$B_{f.df}$,kg h^{-1}	5.0	2.4	2.4	2.3	2.2	2.0	1.8	1.2	1.2
$B_{f.et}$, kg h^{-1}	0	3.3	3.3	3.6	4.2	4.5	4.8	5.7	6.3
$\sum B_f$, kg h^{-1}	5.0	5.7	5.7	5.9	6.4	6.5	6.6	6.9	7.5
P_e, kW	17.0	17.0	17.0	17.0	17.0	17.0	17.0	17.0	17.0
$\dot{e}_a$	2.40	2.20	2.20	2.14	1.93	1.82	1.80	1.81	1.59
RH, %	100	88.4	87.9	87.9	87.4	86.4	86.2	84.7	84.7

The introduction and implementation of the above mentioned test technology proved that the testing methods are reliable and the test data are repeatable. For the purposes of subsequent tests only that type of load characteristics was used. The test data show that by decreasing the position of the regulating lever 100.0% to 84.7%, the pilot injection decreased two times, its volume being up to 1.2kg h^{-1}. The low excess-air ratio shows the high quality of the combustion process and the low amount of carbon black in the exhaust gases.

CONCLUSIONS

1. At load mode n_e = 1,800 rpm the engine D−120 nominal power, the specific fuel consumption, and fuel consumption are accordingly $P_{e.nom}$ = 18.3kW; $b_{e.nom}$ = 268.5g kWh^{-1} and $B_{f.nom}$ = 4.9kg h^{-1}.
2. The optimum static fuel delivery angle of the engine $\alpha_{st.opt}$ = 21° BTDC.
3. The combustion process maximum indicator pressure $p_{z.max}$ = 52 bar, and is located at the angle of rotation of the crankshaft ATDC $\alpha_{pz.max}$ = 9°.
4. The carburettor preparing and directing ethanol fuel into the cylinder reduces the engine air consumption and volumetric efficiency 8.5–9.0%, respectively.
5. According to the experimental results of the diesel engine the following modes were selected: the mode of maximum torque n_e = 1,300 rpm; the mode of the best efficiency and economy output parameters n_e = 1,500 rpm, and the nominal mode n_e = 1,800 rpm.
6. The specific fuel consumption at different speed modes has significant fluctuations with low loads where the value of the pump fuel delivery is within the limits of V_f = 5–20mm^3 stroke^{-1}. Within the changes of high loads, the specific fuel consumption values are uniformly stable.
7. The engine efficiency reaches its maximum value from p_e = 0.30 MPa, which is in accordance with the changes of the cycle delivery within 25–50mm^3 stroke^{-1}. $\eta_{e.max}$ = 0.37 when p_e = 0.39 MPa.
8. The fuel consumption varies when measured in the engine tests and high pressure pump stand tests. The consumption is significantly dependent on the average efficiency pressure of the engine.

The analysis of the engine combustion process reveals that directing qualitatively and quantitatively different fuels to the combustion process is technically acceptable. This was proved also by the stand tests of the test engine D–120. After the midterm testing cycle with ethanol fuel there was no deterioration in the engine output parameters and final pressure of the compression process. The tests revealed that ethanol purity 96% is well functioning as an additive engine fuel and does not have any negative effect on the values of the engine output parameters.

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