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# Study on performance of compression engine operated by biodiesel fuel

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**Abstract.** The analysis of the performance of biofuel is aimed at evaluating the energy efficiency of operating the engine with the use of biodiesel fuel as function of the fuel's composition and other physical-and-chemical parameters. The mathematical models and analysis techniques known to the authors do not take into account the effect that the use of different bio-diesel fuels has on the operation of the engine and, therefore, need refinement in terms of the mathematical expressions and empirical formulae that describe the physical processes taking place in the engine's cylinders. The aim of the study is to improve the mathematical relations taking into consideration the physical-and-chemical parameters of different types of fuel. The research methods proposed in the article are based on step-by-step consideration of the mathematical models of processes that follow each other, with due account for their possible overlapping, which jointly have an effect on the engine's output indices. The boundary conditions and parameter increments are pre-set in electronic work sheets. Thus, it becomes possible, using the refined mathematical model, to calculate the main performance indices of the diesel engine with due account for the changes in the physical-and-chemical parameters of the fuel. The novelty of the described approach is in the possibility, through the use of the refined model and taking into account the data on the composition of the fuel and the design and operation parameters of the engine, to calculate the indices that allow evaluating the efficiency of use of specific fuels in the internal combustion engine under consideration. In results, formulas for the calculation of the effective power of the engine, fresh air charge density, excess air factor, effective specific fuel consumption and combustion pressure have been developed. Combustion pressure modelling and experimental data is presented.

**Key words:** Biofuel, Efficiency, Composition, Physical-and-chemical parameters of the fuel, Internal combustion engine.

## INTRODUCTION

In the process of analysing the efficiency of use of a biofuel, the determining factor is the operating efficiency of the engine running on that fuel and many researches are made on this topic (Raheman et al., 2004; Ulusoy et al., 2004; Choi et al., 2006; Kaplan et al., 2006, Reyes et al., 2006, Özgünay et al., 2007; Utlu, 2008; Hazar, 2009, Ozsezen et al., 2009, Xue et al., 2011). Thus, the consideration is given to the main factors that have effect on the energy performance of the engine (Apostolakou et al., 2009; Baranauskas et al, 2015). The other important issue is the fuel consumption rate per unit of energy output.

Hence, the output parameters of the problem under study are the external characteristics of the engine running on different types of fuel. There is a great number of models describing the work processes that take place in the engine cylinder (Naitoh et al., 1992; Mikita et al., 2012; Shabir et al., 2014; Olt et al., 2015; Zhou et al., 2015). However, such models not always take into account the effect that the fuel has on the performance indices of the engine, while those taking this effect into consideration are complicated and require carrying out a number of additional experimental studies in order to determine the coefficients and characteristics of the related processes used as input data (Osetrov, 2005).

Accordingly, it is a task of current concern to develop a mathematical model capable of describing with the use of semi-empirical relations the work processes in the diesel engine cylinder and, on the basis of their analysis, modelling the engine's external characteristics with due account for the physical-and-chemical properties of different types of fuel.

The mathematical models used for the calculation of external characteristics of engine performance can be divided into two types. The models of the first type are based on the heat calculation of the engine (Merker et al., 2012a). They are rather complicated, since they include quite a number of various parameters specifying the involved processes: gas exchange (intake-scavenging-exhaust), air and fuel mixing and combustion in the engine cylinder, heat transfer, internal energy losses and other. While such a mathematical model is suitable for calculating the engine operation at its rated duty, it cannot be used in case of the maximum torque duty in view of the absent coefficients that can be determined only experimentally. The second type of models is used more extensively for practical calculations and is based on the use of semi-empirical equations of the external characteristics with experimentally obtained coefficients (Osetrov, 2005; Ghobadian et al., 2013), and it is represented by the cubic trinomial formula:

$$\overline{N} = A \cdot \overline{n} + B \cdot \overline{n}^2 - C \cdot \overline{n}^3, \tag{1}$$

where $\overline{N} = N/N_{max}$ – relative (normed) power equal to the ratio between the current power N and the maximum (rated) power $N_{max}$; A, B and C – coefficients depending on the type of engine; $\bar{n} = n/n_{max}$ – relative rate of rotation equal to the ratio between the current rate of rotation n and the maximum rate of rotation $n_{max}$ of the engine's crankshaft.

In the simplified case, the formula, in which only the relation with the torque is taken into account, can be used. The empirical model (1) for the calculation of the engine's external characteristic based on the statistical generalization of the external characteristics of engines of the same type.

The task of investigating the engine performance is important in case of changing the fuel type. When the engine is operated with the use of an alternative fuel, a need arises to adjust the control parameters (Osetrov, 2005; Küüt et al., 2015, Ilves et al.,

2019) in view of the fact that the fuel of another type has different physical-and-chemical parameters and, in order to optimise the operation of the engine with another type of fuel, it is necessary to take into account the changed parameters of the work process.

To make allowance for the effect of the fuel's physical-and-chemical parameters on the engine's performance indices, it will not suffice just to make use of one empirical relation, because the above-mentioned properties of the fuel can have a considerable impact on the working conditions of the engine.

For that reason, it is suggested to use in the formula (1) not the engine's rated power specified in the engine's specification, but rather the power rating obtained by calculation or experimentally at a rated duty under the pre-set operating conditions, taking into account the type of fuel, ambient conditions etc. In recent years, a number of scientific studies have mainly focused on the kinetics of the combustion of biofuels, including biodiesel (Ra & Reitz, 2011, Maawa et al., 2020, Ismail et al., 2013, Lejre, et al., 2020). In addition, models for calculating engine torque have been developed (Alcan et al., 2020). All research deals with models one by one. For example, reactions of sulfur compounds, optimization of torque models, etc. are discussed. However, a model system that takes into account the characteristics of the engine being developed and the fuels used in it is important for engine development.

In that case, one more mathematical model is required, which would provide for carrying out the calculation on the basis of the physical-and-chemical properties of the bio-diesel fuel. In other words, it is necessary to find relations determining the engine's power at its rated duty.

The aim of the study is the development of a mathematical model for the practical analysis of the efficiency of use of biofuel in diesel engines, which would take into account the fuel's composition. The developed model in the article is designed to develop the 4Ч11.0 / 12.5 diesel engine.

## MATERIAL AND METHODS

### Mathematical model

The methods of research are based on mathematical modelling of the physical and physical-and-mechanical processes taking place in the engine cylinder, which are taken into consideration one after another and related with each other in a cause-effect relationship. Microsoft Excel electronic work sheets have been used for the mathematical modelling of the processes that take place in the diesel engine cylinder, when bio-diesel fuels of different compositions are used. Developed model and non-referred equations based on the theory of the internal combustion engine of the Heywood (1988).

The analysis and calculation of the effective power of the engine can be carried out with the use of the following known analytical dependence (Heywood, 1988):

$$N_e = \frac{p_e \cdot V_h \cdot n \cdot i}{30 \cdot \tau_{en}}, \tag{2}$$

where $p_e$ – mean effective pressure in the cylinder [MPa]; $V_h$ – cylinder displacement; $n$ – rate of rotation of the crankshaft [min$^{-1}$]; $i$ – number of cylinders; $\tau_{en}$ – number of strokes in the cycle of the engine.

Pressure $p_e$ can be expressed as the following difference:

$$p_e = p_i - p_m, \tag{3}$$

where $p_i$ – actual mean indicated pressure, with due account for the smoothing out of the indicator diagram [MPa]; $p_m$ – mean pressure of the mechanical losses [MPa].
Pressure $p_i$ is determined by the formula:

$$p_i = p_i' \cdot \upsilon, \tag{4}$$

where $p_i'$ – theoretical mean indicated pressure [MPa]; $\upsilon$ – coefficient of smoothing of the indicator diagram;$p_i'$

The theoretical mean indicated pressure can be found using the Grinevetsky – Mazing formula for the composite Diesel cycle (Merker et al., 2012a, 2012b):

$$p_i' = \frac{p_c}{\varepsilon - 1} \cdot \left[ r_p \cdot (\rho - 1) + \frac{r_p \cdot \rho}{n_2 - 1} \cdot \left( 1 - \frac{1}{\delta^{n_2 - 1}} \right) - \frac{1}{n_1 - 1} \cdot \left( 1 - \frac{1}{\varepsilon^{n_1 - 1}} \right) \right], \tag{5}$$

where $p_c$ – pressure at the end of the compression stroke [MPa]; $\varepsilon$ – compression ratio; $r_p$ – pressure ratio; $\rho$ – preliminary expansion ratio; $n_1$ – polytropic index of compression; $n_2$ – polytropic index of expansion; $\delta$ – subsequent expansion ratio.
The $r_p$ index is determined as follows:

$$r_p = \frac{p_z}{p_c}. \tag{6}$$

where $p_z$ – maximum pressure of the engine operating cycle [MPa].

Preliminary expansion ratio:

$$\rho = \frac{V_z}{V_c}, \tag{7}$$

where $V_z$ and $V_c$ – volumes of the cylinder at the piston's position that corresponds to the maximum pressure in the cylinder during the cycle (point $z$ on the indicator diagram) and the combustion chamber, respectively by Mikita et al. (2012).

The established formula of the total cylinder volume can be used to determine the volume $V_z$ (Merker et al., 2012a, 2012b):

$$V_a = V_h + V_c. \tag{8}$$

The next step is to use the cylinder displacement formula with due account for the number of dimensions:

$$V_h = \frac{\pi \cdot D^2}{4} \cdot S \cdot 10^3, \tag{9}$$

where $D$ – cylinder diameter [m]; $S$ – piston stroke [m].

The formulae (8, 9) can be used to find the current volume, when the piston is at a random point (for example, at the point z of the indicator diagram):

$$V_x = \frac{\pi \cdot D^2}{4} \cdot S_x \cdot 10^3 + V_c \tag{10}$$

where $S_x$ – current stroke position of the piston [m], which depends on the angular displacement of the crankshaft and is determined by the formula (Merker et al., 2012a):

$$S_x = r + L_{cr} - r \cdot \cos\varphi - L_{cr} \cdot \cos\beta, \qquad (11)$$

where $r$ – crank throw [m]; $L_{cr}$ – connecting rod length [m]; $\varphi$ and $\beta$ – angles of the position of the crank and the deflection of the connecting rod from the vertical axis, respectively.

The variable $L_{cr}$ can be eliminated with the use of the parameter $\alpha$ – crank-and-rod mechanism constant (ratio between the crank throw and the connecting rod length):

$$\alpha = \frac{r}{L_{cr}} \qquad (12)$$

hence,

$$L_{cr} = \frac{r}{\alpha}. \qquad (13)$$

Taking into account the centred type of the crank-and-rod mechanism (for example, in case of the 4Ч11,0 / 12,5 diesel engine), the piston stroke is equal to two throws of the crank:

$$S = 2 \cdot r \qquad (14)$$

hence,

$$r = S/2. \qquad (15)$$

After substituting the relations (11, 13, 15) and the values of the angles $\varphi$ and $\beta$ corresponding to the position of the crank at the point $z$ of the indicator diagram with the respective index into the Eq. (10), the following formula for the volume $V_z$ is obtained:

$$V_z = \frac{\pi \cdot D^2 \cdot S}{8} \cdot \left[1 + \frac{1}{\alpha} \cdot (1 - \cos\beta_z) - \cos\varphi_z\right] \cdot 10^3 + V_c \qquad (16)$$

Subsequent expansion ratio:

$$\delta = \frac{\varepsilon}{\rho}. \qquad (17)$$

Mean pressure of the mechanical losses (Merker et al., 2012a):

$$p_m = a + b \cdot W_{p.av}, \qquad (18)$$

where $a$ and $b$ – empirical coefficients [MPa and MPa s m$^{-1}$ respectively]; $W_{p.av}$ – mean piston speed [m s$^{-1}$], which for practical calculations can be found from the formula:

$$W_{p.av} = \frac{S \cdot n}{30}. \qquad (19)$$

**Calculations princapples**

In order to verify the adequacy of the obtained mathematical model of the work processes in the engine cylinder, it is necessary to set the initial data. The 4Ч11,0/12,5 diesel engine has been chosen by the authors as the subject of research. The initial data are represented in the Table 1.

After substituting the initial data into the developed mathematical model represented by the formulae (31–35), (42), (43), (47–50), (53–55), it is possible to calculate the part of the indicator diagram that includes the compression polytropy curve and the combustion curve of the 4Ч11,0/12,5 engine.

In order to obtain the value $T$ of the gas temperature in the cylinder, it is necessary to carry out experimental research. The main point of such research is in the indication of the engine's performance.

**Table 1.** Initial data for modelling work processes in 4Ч11,0/12,5 engine at its rated duty

| Pos. | Description | Designation [unit] | Numerical value | | |
|---|---|---|---|---|---|
| | | | At rated duty | At experimental duty with diesel fuel | At experimental duty with biofuel |
| 1 | 2 | 3 | 4 | 5 | 6 |
| 1 | Rotation rate of crankshaft | $n$ [rpm] | 2,200 | 2,200 | 2,200 |
| 2 | Compression ratio | $\varepsilon$ | 16 | 16 | 16 |
| 3 | Atmospheric pressure | $p_0$ [MPa] | 0.101325 | 0.09799167 | 0.09799167 |
| 4 | Combined effect of air intake system coefficients on fresh air charge parameters | $\beta_{in}^2+\xi_{in}$ | 2.5 | 2.5 | 2.5 |
| 5 | Piston stroke | $S$ [m] | 0.125 | 0.125 | 0.125 |
| 6 | Cylinder diameter | $D$ [m] | 0.11 | 0.11 | 0.11 |
| 7 | Crank-and-rod mechanism constant | $\alpha$ | 0.272 | 0.272 | 0.272 |
| 8 | Ambient air temperature | $T_0$ [K] | 293.15 | 303.15 | 303.15 |
| 10 | Angle of bevel of valve couple (seat angle) in gas distribution mechanism | $\alpha_v$ [deg] | 45 | 45 | 45 |
| 11 | Stroke of intake valve in: first cylinder | $h_{v1}$ [m] | 0.01215 | 0.01215 | 0.01215 |
| 12 | Stroke of intake valve in: second cylinder | $h_{v2}$ [m] | 0.01215 | 0.01215 | 0.01215 |
| 13 | Stroke of intake valve in: third cylinder | $h_{v3}$ [m] | 0.0122 | 0.0122 | 0.0122 |
| 14 | Stroke of intake valve in: fourth cylinder | $h_{v4}$ [m] | 0.0124 | 0.0124 | 0.0124 |
| 15 | Valve throat diameter on machined face in intake port | $d_n$ [m] | 0.0462 | 0.0462 | 0.0462 |
| 16 | Cyclic fuel injection rate | $V_{fc}$ [mm$^3$ cycle$^{-1}$] | 62.7365 | 56.5148 | 56.5148 |
| 17 | Density of fuel: diesel | $\rho_{df}$ (at $T$ = 323 °K) [kg m$^{-3}$] | 825 | - | - |
| 18 | Density of fuel: bio-diesel | $\rho_{bf}$ (at $T$ = 323 °K) [kg m$^{-3}$] | 856 | - | - |
| 19 | Molar mass of air | $\mu_a$ [kg kmol$^{-1}$] | 28.96 | 28.96 | 28.96 |
| 20 | Cylinder purging coefficient | $\varphi_c$ | 1 | 1 | 1 |
| 21 | Additional cylinder charging coefficient | $\varphi_{c.ad}$ | 1.0996 | 1.022 | 1.022 |
| 22 | Residual gas temperature | $T_r$, K | 825 | 825 | 825 |

*Table 1 (continued)*

| 1 | | 2 | 3 | 4 | 5 | 6 |
|---|---|---|---|---|---|---|
| 23 | Mass fractions of main elements in composition of diesel fuel: | carbon | $C_{df}$ | 0.85665 | 0.85665 | - |
| 24 | | hydrogen | $H_{df}$ | 0.133 | 0.133 | - |
| 25 | | sulphur | $S_{df}$ | 0.005 | 0.005 | - |
| 26 | | oxygen | $O_{df}$ | 0.01 | 0.01 | - |
| 27 | Mass fractions of main elements in composition of bio-diesel fuel: | carbon | $C_{bf}$ | 0.772 | - | - |
| 28 | | hydrogen | $H_{bf}$ | 0.119 | - | - |
| 29 | | sulphur | $S_{bf}$ | 0 | - | - |
| 30 | | oxygen | $O_{bf}$ | 0.109 | - | - |
| 31 | Fresh air charge preheating temperature | | $\Delta T$ , [K] | 20–40 | 20 | 22 |

Brief technical description of subject of research. The engine under research is a naturally aspirated four-cylinder four-stroke Diesel engine with an open combustion chamber and a liquid cooling system. The model is D-240, produced by the Minsk Engine Works. The fuel system is of a closed type. The split-type fuel-feed equipment comprises the UTN-5 inline fuel-injection pump with an inbuilt all-speed governor and closed-type FD-22 fuel nozzles with 5-orifice spray atomizers as well as thick-walled steel high pressure fuel lines. The diameter of an injection orifice in the spray atomizer is 0.32 mm. The pear-shaped combustion chamber in the piston has been developed by the Central Research and Development Diesel Institute (TsNIDI). The engine power rating is 58.8 kW.

The following grades of fuel have been used, when carrying out the experimental studies: mineral diesel fuel EN 950 grade C and biofuel RSOME (methyl ethers of rapeseed and sunflower oils).

In the next paragraphs, the engine indication procedure is described.

The indication of the engine's performance implies plotting its indicator diagram that shows the relation between the pressure $p$ in the cylinder and the crankshaft angular displacement $\varphi$, i.e. finding the function $p = f(\varphi)$.

The the following equipment is required to indicate an engine: an indicator instrument, the tested engine, a braking device, instruments for measuring the fuel and air consumption and other metering equipment.

**Experimental tests**

**Equipment for experiment.** The indicator instruments used for tests can be of different types. In the study under consideration an electronic indicator has been used, as that is the most accurate and compact type of design. The principle of operation of the electronic indicator is in the conversion of the gas pressure into an electronic signal. The conversion is performed by the piezoelectric transducer 1 (Fig. 1, a), screwed into the cylinder head 2 of the engine in the area of the fourth cylinder.

However, the installation of the pressure transducer alone will result in obtaining only the ordinates for the indicator diagram. In order to find the abscissae, an engine crankshaft position sensor (CSPS), i.e. the induction pick-up (Fig. 1, b) has to be installed.

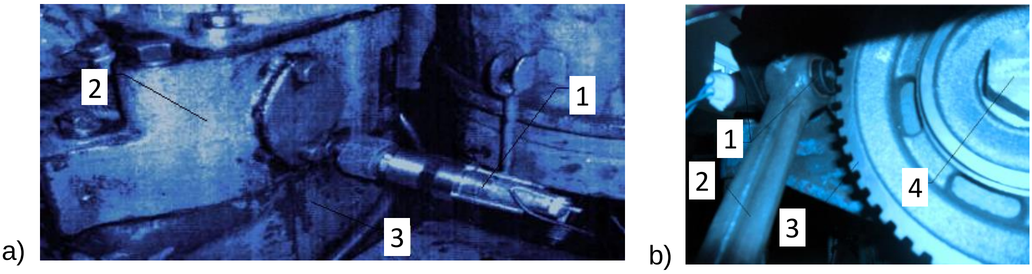


**Figure 1.** Placement of transducers in engine: a) 1 – piezoelectric gas pressure transducer DW-150, 2 – engine cylinder head, 3 – engine crankcase; b) 1 – induction pick-up 191.38472Д, 2 – sensor mounting bracket, 3 – crankshaft balancer pulley 21214-1005060-4, 4 – pulley hold-down bolt.

The reception and processing of the signals sent by the pressure transducer and the CSPS have been performed by the measurement module ZET 210, an analogue-to-digital and digital-to-analogue converter (ADC DAC), connected with the personal computer (PC) via a USB cable (Fig. 2, a). The special ZETLab interface included in the measurement module supply package has been used for displaying the signals on the PC monitor. In essence, the measurement module is an electronic oscillograph capable of simultaneously receiving and processing up to 16 signals.

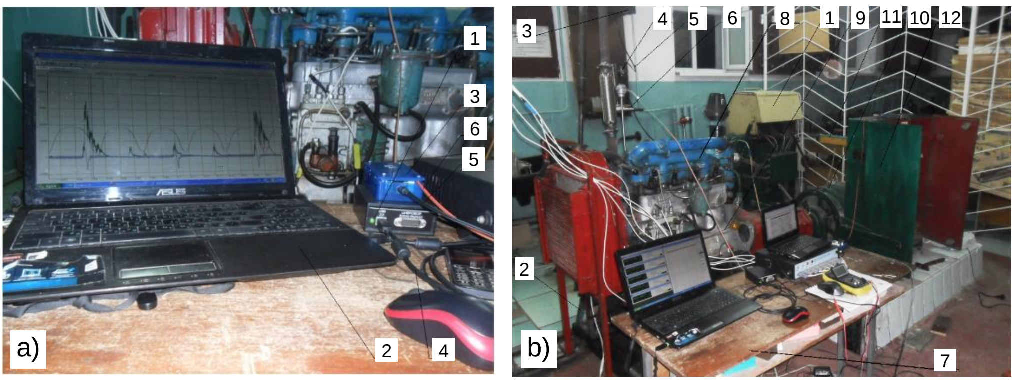


**Figure 2.** Equipment for laboratory experiments: a) ZETLab measurement module connected to 4Ч11,0/12,5 diesel engine: 1 – ADC DAC; 2 – PC; 3 – switching unit (SB); 4 – cable HS USB 2.0; 5 – power cable of SB (12 V); 6 – cable of input signals from transducers. b) КИ-5542 run-in check-out test bed for internal combustion engines: 1 – rheostat; 2 – stand; 3 – fuel tank; 4 – three-way valve; 5 – fuel supply line; 6 – weighing unit; 7 – measuring and recording equipment; 8 – diesel engine; 9 – linkage (cable) for driving governor lever of fuel injection pump; 10 – balancing electric machine; 11 – control panel; 12 – weighing system.

In order to support the supply of power and reception of signals to/from the transducers, the special switching unit 3 (Fig. 2, a) has been developed. Power is supplied to the switchboard (SB) from the standard accumulator battery of the engine, the unit, in its turn, transforms the voltage as necessary to supply power to each of the sensors (if needed) and receives from the latter their respective signals. Further, the signals are transmitted to the ADC DAC and displayed on the PC monitor in the form of oscillograms. In the study under consideration, two oscillograms are enough to generate the indicator diagram – the oscillograms of the signals from the pressure transducer and the CSPS with the same axis of abscissae (shows the duration of the experiment in time). The data from the oscillograph can be recorded both in the graphical and tabular formats, which is convenient for their further processing.

In the capacity of the braking unit, the КИ-5542 commercial run-in check-out test bed for internal combustion engines (Fig. 2, b) has been used. The test bed comprises the balancing asynchronous electric machine 10 with a phase-wound rotor, model 4АНК, coupled with the diesel engine 8 via a universal joint. The electric machine is used for starting the diesel engine, when it acts as an electric motor and rotates the engine at the starting frequency, and for putting load on the diesel engine during the test, when it acts as an electric brake and imparts to the diesel engine the loading magnetic moment that arises due to the rotating magnetic field. The stator of the electric machine is mounted so as to allow it to rotate about its axis and is balanced via a system of cable and lever linkages by the pendulum of the weighing system 12, which, in its turn, is attached to the gauging pointer that swings on the dial gauged in kg. Thus, when the electric machine switches to the electric brake mode of operation putting load on the engine, a rotational torque arises, which turns the stator through some angle until it is balanced by the weight of the weighing system’s pendulum. The amount of deflection of the pendulum is what identifies the rotational torque delivered by the engine.

The load on the diesel engine is changed by changing the strength of current on the rotor windings of the electric machine with the use of the liquid rheostat 1, which is a 300 L tank with electrolyte (water solution of caustic soda) and three steel electrodes immersed in the electrolyte. The area of contact between the electrodes and electrolyte changes together with the electrode immersion depth, which implies varying the resistance on the rotor windings and that results in the alteration of the electromagnetic braking torque that loads the diesel engine 8.

During the operation of the test bed, when the electric machine rotor windings are energized, the electrolyte is continuously mixed by the agitator, which is switched on from the push-button station on the control panel 11. The other push-button station also installed in the control panel is for switching on the motor reducer that sinks or hoists the electrodes. Apart from them, the control panel features the standard engine metering equipment, in particular, the engine tachometer, which has been used for measuring the crankshaft rotation frequency in order to control the speed condition of the engine.

The values of the gas temperature $T$ in the cylinder for the formula (43) are taken from the experimental indicator diagram of the 4Ч11,0/12,5 engine presented in the paper by Romanov (2010).

In order to supply the engine with fuel and measure its consumption rate, the test bed is equipped with the tank 3, the three-way valve 4 connected with the tank via the low pressure fuel supply line 5 and the electronic weighing unit 6 for measuring the mass consumption, on which a graduated cylinder is placed for measuring the volume consumption of fuel.

Thus, the КИ-5542 test bed provides for establishing various load and speed duties of the tested engine as well as measuring its effective parameters under these duties.

Besides, the following sensors have been used for measuring auxiliary parameters in the process of engine indicator diagram plotting: a mass air flow (MAF) sensor BOSCH 0280218116, two thermoelectric couples ТЭРА ТХА 1-24 К-2-И 500Ф10 (0…+ 1,060 °C) 10/2008 and ТХА – 2088 ХА(К)/21 (– 40…+ 600 °C) 90.06.04600 placed in the exhaust manifold for measuring the exhaust gas temperature in the middle of the flow, VIKA sensors for recording the fuel-feed equipment operation parameters and a pressure transducer from the Delfin-1M test bench. The VIKA pressure transducer is installed in the fuel injection line for measuring the pressure of fuel before the fuel nozzle, while the Delfin-1M pressure transducer is put as a clip on the fuel supply line after the pump outlet for measuring the deformation of the fuel supply line. These sensors have also been connected to the ZETLab measurement module by means of the SB.

Also, the ambient air temperature and pressure have been measured with the use of mercury-filled thermometer and barometer gauge as well as the relative air humidity with the use of an aspirated psychrometer.

**Preparation of experiment.** The laboratory experiment was carried out in the laboratory of internal combustion engines. In order to install the DW-150 gas pressure transducer, it was necessary to prepare the engine cylinder head. For that purpose, the engine was partially disassembled, the engine head was dismantled. After a thorough examination of the head, two canals were bored – one horizontal and one vertical – for connecting the transducer with the internal chamber of the cylinder. In the horizontal bore, thread was tapped for screwing the transducer into the cylinder head. Before mounting the cylinder head back into its position, the valve system was checked with regard to the tightness and seating of the valve disks, also the seating of the injector nozzle needles in the wells of spray tips as well as the evenness of the sealing surfaces on the cylinder head and the cylinder block were checked.

The design of this model of engine did not provide for the installation of a CSPS sensor, therefore, a 191.38472Д sensor and a 21214-1005060-40 crankshaft balancer pulley supplied for standard VAZ-2105 motor cars were used. In order to equip the diesel engine with the sensor, the bracket 2 (see Fig. 2, b) and the pulley hold-down bolt 4 were used. As the 21214-1005060-40 pulley and the tested engine's standard crankshaft pulley were of different sizes, two pulleys had to be used together, which implied manufacturing an increased length bolt. Copper washers were foreseen for the adjustment of the clearance between the sensor 1 and the teeth of the balancer pulley 3.

In order to connect the sensor to the oscillograph, the switching circuit connecting the sensor and the ZETLab measurement module was designed and assembled by soldering in the SB. To ensure a reliable signal, the maximum possible clearance between the sensor and the teeth of the balancer pulley that provided a dependable signal was established experimentally.

Prior to fixing the balancer pulley, it was necessary to ensure that its missing tooth was against the sensor at the moment, when the pistons of the first and fourth cylinders were at their top dead centres (TDC). The TDC was determined with the use of a dial indicator with the cylinder head removed from the engine, while the correct position of the balancer pulley with its missing tooth against the CSPS was found by simultaneously rotating the crankshaft and watching the oscillogram on the PC. As soon as the signal oscillogram crossed the X-axis in the area of the missing tooth in correspondence with the TDC mark, the balancer pulley 3 was fixed with the hold-down bolt 4 (see Fig. 2, b).

In order to ensure the correct operation of the weighing system of the КИ-5542 test bed, it was necessary to carry out its calibration. For the calibration of the weighing system, a gauging arm and a set of standard scale weights were used. A scale weight was put on the free end of the gauging arm and the deflection of the weighing system’s pointer by one scale division value was fixed. Then greater scale weights were put on and the further pointer deflections were fixed. The scale weight that weighed 1 kg had to deflect the system’s pointer through the one kilogramme mark on the dial. If the pointer deflections did not coincide with the marks corresponding to the weights put on the end of the gauging arm, the system’s dial had to be re-graduated. After gauging the dial clockwise, it was gauged counter-clockwise as well.

After calibrating the measuring mechanisms, the standard tachometer and the electronic scales, the installed transducers were calibrated.

The gas pressure transducer was gauged with the use of a compressor with a pressure gauge. As a result of the gauging, the relation between the measured voltage and the gas pressure had been established.

The signal received from the CSPS could be converted into the crankshaft revolutions. For that purpose, it was sufficient to start the command ‘Tachometer’ under the tab ‘Measurement’ of the ZETLab interface, then, in the opened dialog box ‘Revolutions’, select the channel that corresponded to the CSPS signal. After that, the rpm values obtained by the electronic means were verified with the use of a hand tachometer. In case the readings disagreed, correction factors had to be input into the electronic tachometer settings.

The calibration of the MAF sensor was carried out in the heat engineering laboratory in the ICE and AFR department of the VNAU with the use of the unit for measurement of the isochoric heat capacity of air that comprised a differential pressure gauge and a fan. After connecting the MAF sensor with the fan in sequence, so as to make the whole flow of the air taken into the unit to pass through the sensor, the sensor could be gauged. As a result of the calibration of the sensor, a gauging characteristic of the relation between the measured voltage and the mass of air passing through the sensor in a unit time was obtained.

The calibration of the thermoelectric couples was done by immersing them into a container with vegetable oil. The oil temperature was changed with the use of a heat source. A mercury-filled thermometer with a dial that had its upper limit at a level of at least 300 °C was used for measuring the oil temperature variation. As a result, a gauging characteristic of the measured voltage variation as a function of the temperature was obtained.

The transducers for recording the operation parameters of the fuel supply system were calibrated with the use of the КИ-3333 fuel nozzle check-out test bed. In order to place the VIKA transducer on the centre line of the fuel flow, a T-joint was manufactured, which connected the fuel injection line, the nozzle and the sensor. As a result, a gauging characteristic of the relation between the measured voltage and the fuel pressure was obtained. The pressure transducer from the Delfin-1M was not calibrated. It was used for correlation of the signal from the VIKA sensor.

In the process of calibration, all the sensors were connected to the ZETLab measurement module and their signals were displayed on the electronic oscillograph. For that purpose, a circuit was developed for connecting the sensors with due consideration of the need to supply power to them. All the transducer power supply and switching circuits were soldered together on one motherboard installed in the SB.

Prior to starting the engine indication, it was also necessary to synchronize the signals received from the gas pressure transducers and the CSPS. For the synchronization of the mentioned signals, the 'Multichannel Oscillograph' tab was selected in the ZETLab interface, then the respective channels were selected, the required adjustments were made and the oscillograms of the said signals were started. In order to generate the signals, the crankshaft was rotated by the electric machine, while the supply of fuel to the diesel engine was stopped. The maximum value of the signal from the gas pressure transducer had to coincide with the moment, when the signal from the CSPS sensor crossed the X-axis in the area of the missing tooth, which corresponded to the TDC. In case the maximum pressure did not coincide with TDC, it was necessary to loosen the bolt 4 (Fig. 1, b)) and adjust the position of the balancer pulley 3 with regard to the position of the crankshaft, then tighten the bolt.

**Procedure of experiment.** For obtaining the indicator diagram, the electronic oscillograph is set to operate with seven measurement channels in order to be able to receive signals from all the above-mentioned sensors. The engine is started and warmed up until the temperature of the cooling fluid in the cooling system reaches 50 °C, then it is set to its rated operating duty. For that purpose, the fuel injection pump governor control arm is brought to the position of the maximum fuel supply rate. After the engine crankshaft reaches the maximum revolution rate, the electric machine 10 (Fig. 2, b) is switched on and the electrodes of the rheostat 1 are immersed into the electrolyte, thus increasing the load on the diesel engine. Under these conditions, the rotation frequency of the crankshaft will go down. After reaching a revolution rate of $n$ = 2,200 min$^{-1}$ and the maximum possible load for that frequency, the immersion of the electrodes is stopped. At this stage, the engine is under its rated duty. After reaching the rated duty, it is necessary to check whether the operating conditions have stabilised. The conclusion about the stabilization of the reached conditions can be made on the basis of the indications of the thermoelectric couples. The stabilised indications of the thermocouples represent the stabilised exhaust gas temperature, which implies the stabilised engine load conditions. In case the load conditions change during the stabilization of the thermal conditions, that can be adjusted by manually turning the electrode immersion motor reducer with the use of the special hand wheel. The stable rated operating duty is considered to be established, when the load and speed and thermal parameters become constant.

After the engine has reached its rated operating duty, the fuel consumption rate is to be measured. For that purpose, the time, in which the mass of the fuel is reduced by 100 g according to the readings of the electronic scales, is measured with the use of a stop watch. Also, the readings of the weighing system pointer in kg are registered. When the fuel consumption rate is measured, the start and the end of the experiment are recorded from the time scale of the oscillograph.

After the experiment is completed, the marked interval of the oscillogram is saved in the memory.

## RESULTS AND DISCUSSION

### Mathematical model

Substituting the formulae (3–7, 16–19) into the expression (2), the following equation is obtained:

$$N_e = \frac{\pi \cdot D^2 \cdot S \cdot n \cdot i}{120 \cdot \tau_{en}} \times \left( \frac{p_c \cdot \upsilon}{\varepsilon - 1} \cdot \left\langle \frac{p_z}{p_c} \cdot \left\{ \frac{\varepsilon - 1}{2} \cdot \left[ 1 - \cos\varphi_z + \frac{1}{\alpha} \cdot (1 - \cos\beta_z) \right] + \right.\right.\right.$$
$$+ \frac{1 + \frac{\varepsilon - 1}{2} \cdot \left[ 1 - \cos\varphi_z + \frac{1}{\alpha} \cdot (1 - \cos\beta_z) \right]}{n_2 - 1} \times \tag{20}$$
$$\left. \times \left[ 1 - 1 \cdot \left( \varepsilon \cdot \left( 1 + \frac{\varepsilon - 1}{2} \cdot \left[ 1 - \cos\varphi_z + \frac{1}{\alpha} \cdot (1 - \cos\beta_z) \right] \right)^{-1} \right)^{n_2 - 1} \right] \right\} -$$
$$\left.\left. - \frac{1}{n_1 - 1} \cdot \left( 1 - \frac{1}{\varepsilon^{n_1}} \right) \right\rangle - \left[ a + \frac{b \cdot n \cdot S}{30} \right] \right).$$

In order to simplify the notation, the following coefficients are introduced in the formula (20):

$$k_1 = \frac{\varepsilon - 1}{2} \cdot \left[ 1 - \cos\varphi_z + \frac{1}{\alpha} \cdot (1 - \cos\beta_z) \right], \tag{21}$$

$$k_2 = n_2 - 1. \tag{22}$$

Thereby, the following is obtained:

$$N_e = \frac{\pi \cdot D^2 \cdot S \cdot n \cdot i}{120 \cdot \tau_{en}} \cdot \left[ \frac{p_c \cdot \upsilon}{\varepsilon - 1} \cdot \left\langle \frac{p_z}{p_c} \cdot \left\{ k_1 + \frac{1 + k_1}{k_2} \cdot \left[ 1 - \left( \frac{\varepsilon}{1 + k_1} \right)^{-k_2} \right] \right\} - \frac{1 - \varepsilon^{-n_1}}{n_1 - 1} \right\rangle - \right.$$
$$\left. - \left( a + \frac{b \cdot n \cdot S}{30} \right) \right], \tag{23}$$

where $k_1$, $k_2$ – simplification coefficients representing the position of the piston at the point z of the indicator diagram and the polytropic index of expansion less unity respectively.

With the use of the formula (23), the rated power can be determined for the two typical duties of the engine – the rated duty and the maximum torque duty. As regards the other duties of the engine, they require applying the method of calculation, in which the empirical cubic trinomial equation (1) is used. The equation can be written out in the following form (Komakha & Ryaboshapka, 2016):

$$N_{ex} = N_{enom} \cdot \frac{n_x}{n_{nom}} \cdot \left[ A + B \cdot \frac{n_x}{n_{nom}} - C \cdot \left( \frac{n_x}{n_{nom}} \right)^2 \right] \tag{24}$$

where $A$, $B$, $C$ – empirical coefficients depending on the types of engine and fuel; $N_{enom}$ – engine power rating [kW]; $n_x$ – current value of the rate of revolution of the engine crankshaft [min$^{-1}$]; $n_{nom}$ – rated revolutions of the crankshaft [min$^{-1}$].

The relation between the engine's torque and effective power output is determined by the known expression:

$$M_e = \frac{9{,}550 \cdot N_e}{n}. \tag{25}$$

he effective specific fuel consumption is determined by the formula:

$$g_e = \frac{120 \cdot G_{fc} \cdot n \cdot i}{N_e \cdot \tau_{en}}, \tag{26}$$

where $G_{fc}$ – cyclic fuel injection rate [g cycle$^{-1}$].

The following relation exists between the volume and mass rates of the cyclic injection:

$$G_{fc} = V_{fc} \cdot \rho_f \cdot 10^{-9}, \tag{27}$$

where $V_{fc}$ – cyclic fuel injection rate [mm$^3$ cycle$^{-1}$]; $\rho_f$ – density of the fuel [kg m$^{-3}$].

Thus, the relation (27) defines the cyclic fuel injection rate depending on the fuel density. The relation can be used in practical calculations in case of switching to another biofuel with a different density as compared to the standard diesel fuel (Osetrov, 2005).

After substituting the formula (23) into the equation (26), while taking into account (27), the following equation for determining the specific fuel consumption is obtained:

$$g_e = \frac{1{,}44 \cdot 10^{-5} \cdot V_{fc} \cdot \rho_f \cdot D^{-2} \cdot (S \cdot \pi)^{-1}}{\left[ \frac{p_c \cdot \upsilon}{\varepsilon - 1} \cdot \left\langle \frac{p_z}{p_c} \cdot \left\{ k_1 + \frac{1 + k_1}{k_2} \cdot \left[ 1 - \left( \frac{\varepsilon}{1 + k_1} \right)^{-k_2} \right] \right\} - \frac{1 - \varepsilon^{-n_1}}{n_1 - 1} \right\rangle - \left( a + \frac{b \cdot n \cdot S}{30} \right) \right]}, \tag{28}$$

The formula (28) allows determining the specific fuel consumption at the two typical engine duties – the rated duty and the maximum torque duty. For calculating the specific fuel consumption at other load and revolution duties of the engine, the method of calculation based on the empirical equation can be used:

$$g_{ex} = g_{enom} \left[ D - E \frac{n_x}{n_{nom}} + G \left( \frac{n_x}{n_{nom}} \right)^2 \right], \tag{29}$$

where $D$, $E$, $G$ – empirical coefficients that depend on the type of engine and the fuel type.

The hourly fuel flow rate (Heywood, 1988; Merker et al, 2012a,b):

$$G_f = g_e \cdot N_e \cdot 10^{-3} \tag{30}$$

Thereby, the equations (21–25, 28–30) represent a mathematical model for the calculation of the external control characteristic, the analysis of which allows evaluating the operating efficiency of the engine, while the indices $\rho_f$, $p_c$, $p_z$, $\varphi_z$, $\beta_z$, and the coefficients $A$, $B$, $C$, $D$, $E$, $G$ vary depending on the type of fuel and have an effect on the performance parameters $N_e$, $M_e$, $g_e$, $G_f$.

In order to determine the parameters $p_c$, $p_z$, $\varphi_z$, $\beta_z$, it is necessary to take under consideration the work processes in the engine cylinder, in particular, the combustion process.

The pressure in the engine cylinder during the compression process changes according to the polytropy relation:

$$p = p_a \cdot \left(\frac{V_a}{V_x}\right)^{n_1}, \tag{31}$$

where $p_a$ – pressure at the end of the intake [MPa], which in case of the 4Ч11,0/12,5 engine, i.e. a naturally aspirated engine, is determined as the following difference:

$$p_a = p_0 - \Delta p_a, \tag{32}$$

where $p_0$ – atmosphere pressure [MPa]; $\Delta p_a$ – loss of pressure at intake due to intake system resistance and decrease of the charge velocity in the cylinder [MPa], which with some allowance can be determined from the Bernoulli equation:

$$\Delta p_a = \left(\beta_{in}^2 + \xi_{in}\right) \cdot \left(\frac{\omega_{in}^2}{2}\right) \cdot \rho_0 \cdot 10^{-6} \tag{33}$$

where $\beta_{in}$ – coefficient of charge velocity decrease in the cylinder cross-section under consideration; $\xi_{in}$ – coefficient of the intake system's resistance referred to its narrowest cross-section; $\omega_{in}$ – mean charge velocity at the smallest cross-section of the intake system [m s$^{-1}$]; $p_0$ – density of the charge at the inlet (in case of a naturally aspirated engine – density of the atmospheric air) [kg m$^{-3}$].

The $\omega_{in}$ velocity can be found using the formula:

$$\omega_{in} = n \cdot \frac{r \cdot \pi^2 \cdot D^2}{120 \cdot f_{in}} \cdot \sqrt{1 + \alpha^2} \cdot \tag{34}$$

where $f_{in}$ – area of the smallest cross-section in the intake system, which corresponds to the area of the passage section between the valve's head and seat and is determined with the use of the formula:

$$f_{in} = \pi \cdot h_v \cdot (d_n \cdot \cos\alpha_v + h_v \cdot \alpha_v \cdot \cos^2\alpha_v), \tag{35}$$

where $h_v$ – intake valve stroke [m]; $d_n$ – intake port throat diameter [m]; $\alpha_v$ – valve seat angle [rad].

The density of the atmospheric air depends on the absolute temperature $T_0$ [K] and pressure $\rho_0$ [MPa] of the ambient air and is found from the known formula (Heywood, 1988):

$$\rho_0 = p_0 \cdot 10^6 \big/ R_a \cdot T_o \tag{36}$$

where $R_a$ – specific gas constant of air [J (kg·K)$^{-1}$].

On the other hand, there is a formula for the density of the air charge fed into the cylinder obtained in the technique by Pyadichev (1989):

$$\rho_a = \frac{1.29}{1 + 3.665 \cdot 10^{-3} \cdot (T_a - 273)} \tag{37}$$

where $\rho_a$– air density [kg m$^{-3}$] and $T_a$ – air temperature [K] at the end of the intake stroke (point $a$ of the indicator diagram of the engine).

The formula (37) is of practical interest, since it requires changing only one parameter, i.e. the temperature $T_a$, to determine the density of the charge.

In the authors' opinion, the formula (37) can be used for determining the charge density at the beginning of the intake, i.e. the atmospheric air density, if the temperature $T_a$ is replaced with the temperature $T_0$. In order to verify the suggested replacement, it is necessary to check the relation between the formulae (36) and (37).

First, the temperature $T_0$ in the formula (36) is substituted by the following expression:

$$T_0 = 273.15 + \Delta T_0, \tag{38}$$

where 273.15 – Kelvin scale temperature equal to 0 centigrade; $\Delta T_0$ – difference between the ambient temperature and zero centigrade [°C].

After the values of the pressure and the gas constant of the air corresponding to normal atmospheric conditions, i.e. $p_0 = 0.101325$ MPa, $R_a = 287$ J (kg·K)$^{-1}$, are substituted into the formula (36) and the expression (38) is taken into account, the following is obtained:

$$\rho_0 = \frac{p_0 \cdot 10^6}{R_a \cdot T_o} = \frac{0.101325 \cdot 10^6}{287 \cdot (273.15 + \Delta T_0)} = \frac{353.05}{273.15 + \Delta T_0}. \tag{39}$$

After factoring the coefficient 273.15 out from the denumerator of the formula (39), the following expression is arrived at:

$$\rho_0 = \frac{p_0 \cdot 10^6}{R_a \cdot T_o} = \frac{1.2925}{1 + \dfrac{\Delta T_0}{273.15}} = \frac{1.2925}{1 + 3.661 \cdot 10^{-3} \cdot \Delta T_0}. \tag{40}$$

In the formula (40), the coefficient 1.2925 represents the density value $\rho_0$ [kg m$^{-3}$] at 0 °C, the coefficient $3.661 \cdot 10^{-3}$ K$^{-1}$ is a quantity per degree of absolute temperature (the difference that takes account of the effect the temperature variation by one degree has on the air density).

Hence, the known formula (36) and the formula (37) proposed by Pyadichev (1989) can be equated and written in the following form:

$$\rho_0 = \frac{p_0 \cdot 10^6}{R_a \cdot T_o} = \frac{1.2925}{1 + 3.661 \cdot 10^{-3} \cdot (T_0 - 273.15)}. \tag{41}$$

The obtained formula takes into account the effect of the ambient temperature on the air density, but it leaves out of account the effect of the atmospheric air pressure on it. For that reason, the authors suggest improving the formula (41) by means of introducing the coefficients that take into consideration the deviation of the atmospheric pressure from its value (101,325 Pa) that corresponds to the normal (standard) atmospheric conditions:

$$\rho_0 = \frac{1.2925 \cdot \left[1 + 9.869233 \cdot \left(p_0 - 1.01325 \cdot 10^{-1}\right)\right]}{1 + 3.661 \cdot 10^{-3} \cdot (T_0 - 273.15)}, \tag{42}$$

where 9.869233 MPa$^{-1}$ – quantity per MPa of the atmospheric pressure at normal conditions; $\rho_0$ – 1.01325·10-1 = Δp (the pressure difference that takes account of the effect the deviation of the pressure from its normal value has on the air density).

Thus, the formula (42) can be used for the practical calculation of the intake charge density for a naturally aspirated diesel engine taking into account the effect of the ambient air pressure and temperature on it, which in its turn has effect on the performance parameters of the engine.

Hence, through the use of the formulae (31–35), (42), it becomes possible to model the polytropic curve of compression of the engine, which allows analysing the quality of the work process in the engine cylinder. Also, that enables assessing the adequacy of the mathematical model by comparing the modelled and experimental polytropic curves of compression.

The next important process, the parameters of which have effect on the performance indices of the engine, is the combustion process.

The authors suggest modelling the variation of the pressure during the working medium expansion as a result of combustion with the use of the relation proposed by Pyadichev (1989):

$$p = \frac{V_\beta \cdot R \cdot T \cdot 10^{-3}}{V_x}, \tag{43}$$

where $V_\beta$ – current volume of the working medium [mol]; $R$ – absolute gas constant, $R$ = 8.314 kJ (kmol·K)$^{-1}$; $T$ – temperature of the gases in the cylinder [K].

The current volume of the working medium has to be determined taking into account the quantity of moles of the intake charge and the molecular change of the working medium (Merker et al., 2012b):

$$V_\beta = M_a \cdot \left( 1 + \frac{m_H \cdot x}{19.04 \cdot \lambda \cdot \left( n_C + \frac{m_H}{4} \right)} \right) \tag{44}$$

where $M_a$ – quantity of air charge [mol] at the end of the intake stroke; $m_H$, $n_C$ – quantities of the hydrogen and carbon molecules in the fuel, respectively; $x$ – fraction of the fuel burnt out in the combustion process; $\lambda$ – excess air factor.
The quantity of air charge $M_a$ is determined with the use of the formula:

$$M_a = M_1 \cdot (1 + \gamma_r) \tag{45}$$

where $M_1$ – quantity of intake air charge [mol] fed into the cylinder; $\gamma_r$ – coefficient of residual gas.

The quantity of air intake charge fed into the cylinder can be found with the use of the formula:

$$M_1 = \frac{V_h \cdot \eta_v \cdot \rho_o}{\mu_{mix}}, \tag{46}$$

where $V_h$ – cylinder displacement; $\eta_v$ – volumetric efficiency factor; $\mu_{mix}$ – current molecular mass of the gas mixture, which changes as a result of combustion of the fuel.

The formulae (44–46) can be used to refine the formula by Pyadichev (1989) for determining the current volume of the working medium used in the operating duty cycle:

$$V_\beta = \frac{V_h \cdot \eta_v \cdot \rho_o}{\mu_{mix}} \cdot (1 + \gamma_r) \cdot \left( 1 + \frac{m_H \cdot x}{19.04 \cdot \lambda \cdot \left( n_C + \frac{m_H}{4} \right)} \right) \tag{47}$$

The volumetric efficiency factor is found with the use of the formula (Merker et al., 2012b):

$$\eta_v = \varphi_{c.ad} \cdot \frac{\varepsilon}{\varepsilon - 1} \cdot \frac{p_a}{p_0} \cdot \left(1 - \frac{\varphi_c}{\varepsilon \cdot \varphi_{c.ad}} \cdot \frac{p_r}{p_a}\right) \cdot \frac{T_0}{T_0 + \Delta T} \tag{48}$$

where $\varphi_{c.ad}$ – coefficient of the additional charging of the cylinder due to the valve overlapping; $\varphi_c$ – coefficient of cylinder purging; $p_r$ – residual gas pressure [MPa]; $\Delta T$ – temperature of air charge preheating at the intake [K]. Coefficient of residual gas (Merker et al., 2012a):

$$\gamma_r = \frac{\varphi_c}{\eta_v \cdot (\varepsilon - 1)} \cdot \frac{p_r}{p_0} \cdot \frac{T_0}{T_r} \tag{49}$$

The current molecular mass of the gas mixture (working medium) in the cylinder can be found using the formula (Pyadichev, 1989):

$$\mu_{см} = \left[44.01 \cdot n_C \cdot x + 9.01 \cdot x \cdot m_H + 32 \cdot \left(n_C + \frac{m_H}{4}\right) \cdot (\lambda - x) + \right. \\ \left. +105.844 \cdot \lambda \cdot \left(n_C + \frac{m_H}{4}\right)\right] \cdot \left[4.76 \cdot \lambda \cdot \left(n_C + \frac{m_H}{4}\right) + \frac{m_H \cdot x}{4}\right]^{-1} \tag{50}$$

The excess air factor is determined with the use of the formula:

$$\lambda = \frac{M_1}{G_{fc} \cdot L_0}, \tag{51}$$

where $L_0$ – theoretical air quantity needed for the combustion of one kilogramme of fuel.

The quantity of moles of intake air charge can be determined using the following formula:

$$M_1 = \frac{p_0 \cdot \eta_v \cdot V_h \cdot 10^{-3}}{8.314 \cdot T_0}. \tag{52}$$

The next step is to substitute formulae (9, 27, 52,) into (51) with due account for the known stoichiometric relations (Deviatin et al., 2007). Then, introducing the simplification coefficients, the following refined formula for determining the excess air factor is obtained:

$$\lambda = \frac{\pi \cdot D^2 \cdot S \cdot k_3 \cdot (1 - k_4) \cdot \left(1 - \frac{\varphi_c}{k_3} \cdot \frac{p_r}{p_0 - k_4}\right) \cdot \frac{p_0}{T_0 + \Delta T}}{6.315 \cdot 10^{-9} \cdot V_{fc} \cdot \rho_f \cdot \left(\frac{C}{12} + \frac{H}{4} + \frac{S}{32} - \frac{O}{32}\right) \cdot (\varepsilon - 1)} \tag{53}$$

where $k_3$, $k_4$ – simplification coefficients representing the product of the coefficient of additional charging by the compression ratio and the effect of the main parameters of the crank-and-rod mechanism, gas distribution mechanism and the ambient air parameters on the process of the cylinder filling, respectively; C, H, S, O – composition of the fuel in unit fractions representing carbon, hydrogen, sulphur and oxygen, respectively.

The coefficients $k_3$ and $k_4$ can be determined using the formulae:

$$k_3 = \varphi_{c.ad} \cdot \varepsilon, \tag{54}$$

$$k_4 = \frac{(\beta_{in}^2 + \xi_{in}) \cdot n^2 \cdot S^2 \cdot \pi^2 \cdot D^4 \cdot (1 + \lambda_{cr}^2) \cdot (12,755982 \cdot p_0 + 2,925 \cdot 10^{-1})}{2,4 \cdot 10^5 \cdot h_v^2 \cdot (d_n \cdot \cos\alpha_v + h_v \cdot \alpha_v \cdot \cos^2\alpha_v)^2 \cdot (7,33 \cdot p_0 \cdot T_0 - 2,19)} \tag{55}$$

Thus, the formulae (43), (47–50), (53–55) make it possible to model the part of the indicator diagram that represents the process of combustion in the engine cylinder.

**Results of experimental and mathematical simulation study**

The duration of the laboratory experiment, in which the fuel consumption rate under the rated duty is measured, is usually equal to $t$ = 20 s and can be expressed in cycles with the use of the following formula:

$$n_c = \frac{n}{120} \cdot t \tag{56}$$

After the values at a revolution rate of $n$ = 2,200 min$^{-1}$ are substituted into the formula (56), the following is obtained:

$$n_c = \frac{2{,}000}{120} \cdot 20 = 366.7\ cycles.$$

Within that time, the oscillograph records 366 full cycles, i.e. the data for 366 indicator diagrams.

Further, the data obtained from the gas pressure gauge and the CSPS are saved for the consequent processing in the tabular format. The data are processed in the Microsoft Excel electronic spreadsheet environment.

First, the values on the $x$ scale are converted from the time $\tau$ into the degrees of turn of the crankshaft $\varphi$ with the use of the formula:

$$\varphi = 6 \cdot n \cdot \tau. \tag{57}$$

It should be noted that $\varphi$ varies from 0 to 720°, starting from the zero mark of the crankshaft (the piston of the fourth cylinder is at its TDC, at the end of the exhaust stroke). The zero mark corresponds to the moment when the signal of the CSPS sensor crosses the X-axis in the area of the missing tooth on the balancer pulley. Hence, the variation of the indicator on the axis of abscissae is of the cyclic type.

Thereafter, the signal received from the gas pressure gauge is converted from mV into MPa, using for that purpose the gauging characteristic that was obtained earlier.

The further processing of the experimental data is carried out in accordance with the procedure described in the paper (Johnson & Leone, 1977; Slavinskas et al., 2018). For all the 366 cycles the arithmetical mean values are found and, consequently, a single indicator diagram is obtained, which is cropped leaving only the 300°–430° range, because only the processes of compression and expansion as well as the combustion process are of greater importance for the analysis of the operating duty cycle efficiency.

The next step is to estimate the random error value using for that purpose the root-mean-square deviation $\sigma_i$, the variance $\sigma_i^2$. Further, the gross errors are excluded from the experimental data by using the method of the maximum and minimum possible measurement values in the sequence under consideration. In case two or more blunders are revealed, the experiment has to be repeated.

As a result, the following indicator diagram for diesel fuel has been obtained (Fig. 3, the solid line).

A similar indicator diagram is presented for the type of engine under consideration in the paper (Johnson & Leone, 1977) which confirms the validity of the authors' experimental study.

After the experiment with diesel fuel is completed, the fuel in the test bed tank is replaced by biodiesel fuel, then the engine is run idle for some time in order to consume the diesel fuel remaining in the fuel supply system and switch to the operation of the engine completely on RSOME. The signs of the complete transition are a certain reduction of the engine operation loudness and the emergence of the specific aroma of the exhaust gas. It has been experimentally established that the time for the complete changeover to the operation with biofuel for the 4Ч11,0/12,5 engine idling at a crankshaft revolution rate of $n$ = 1,500 min$^{-1}$ is equal to 20 min (Ilves et al., 2017).

Then the similar experiment is carried out with the use of RSOME biofuel.

The results of the experiment with the use of biofuel have been processed following the procedure described earlier in the paper.

Having completed the processing procedure, the indicator diagram for biofuel operation has been obtained (Fig. 3, dot line).

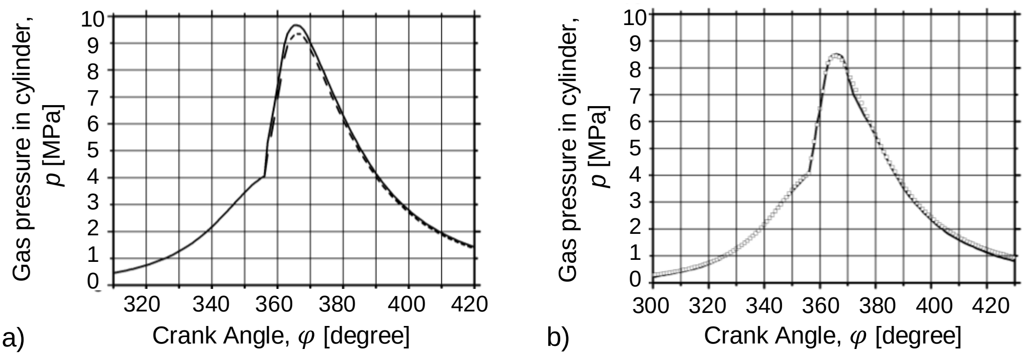


**Figure 3.** Relation between gas pressure in cylinder $p$ and angular displacement of engine crankshaft $\varphi$: a) experimental data for operation with: ——— – diesel fuel; ▬ ▬ ▬ – biodiesel fuel; b) for operation with diesel fuel: ——— – experimental □□□□□ – analytical.

Engine performance parameters are present in Table 2. Differences between the biodiesel power data of model and experimental test is approximately 2.7%. The accuracy is a similar for the all calculated data in Table 2.

**Table 2.** Analysis results of the engine Ч11.0/12.5 pressure diagrams

| Indicator | Diesel fuel, model | Diesel fuel, experiment | | Biodiesel fuel, model | Biodiesel fuel, experiment |
|---|---|---|---|---|---|
| | | Special condition ($p_0$ = 0.098 MPa, $T_0$ = 273 $K$) | Standard condition ($p_0$ = 0.101 MPa, $T_0$ = 273 $K$) | | |
| $N_e$, kW | 49.62 | 50.61 | 55.61 | 50.60 | 52.01 |
| $p_e$, MPa | 0.57 | 0.581 | 0.639 | 0.581 | 0.597 |
| $M_e$, Nm | 215.4 | 219.7 | 241.4 | 219.6 | 225.8 |
| $g_e$, g (kWh)$^{-1}$ | 248.0 | 243.1 | 245.6 | 253.3 | 245.4 |
| $\lambda$ | 1.61 | 1.61 | 1.67 | 1.79 | 1.79 |

The value of the gas temperature in the cylinder is found on the basis of the formula (43):

$$T = \frac{p \cdot V_x \cdot 10^3}{V_\beta \cdot R}. \tag{58}$$

The values of the gas temperature in the cylinder $T$ obtained by analysing the indicator diagram are substituted into the formula (43) of the mathematical model.

In Fig. 4, the temperatuure change with diesel and biodiesel fuel is calcualted. The temperature of the biodiesel combustion in cylinder is higher as it is with diesel fuel. The results are realistics because the biodiesel combustion is carry out on leaner mixture (relative air/fuel ratio is 1.79).

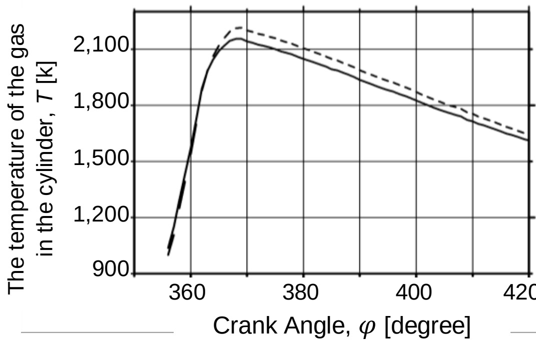


**Figure 4.** Temperature change in engine cylinder depending on crankshaft rotation angle.
Temperatuuri $T$ muutumise dünaamika mootori silindris olenevalt väntvõlli pöördenurgast $\varphi$: ——— – diesel fuel; ■ ■ ■ – biodiesel fuel.

**Opportunities for synthesis of indicator diagram.**

In order to determine the amount of the burnt out fuel $x$, it is necessary to synthesise the indicator diagram. After synthesising the indicator diagram, the combustion curve $x = f(\varphi)$ can be obtained.

For synthesising the indicator diagram, the equation (Merker et al., 2012a) can be used:

$$dQ = G_{fc} \cdot Q_H \cdot \frac{dx}{d\varphi} d\varphi - \delta Q_w \tag{59}$$

where $dQ$ – elementary quantity of heat supplied to the working medium in the cylinder [J]; $Q_H$ – lower calorific value [J kg$^{-1}$]; $\frac{dx}{d\varphi}$ – combustion rate [deg$^{-1}$]; $d\varphi = 6 \cdot n \cdot d\tau$ – increment of the angular displacement of the crankshaft in a time of $d\tau$; $\delta Q_w$ – elementary quantity of heat transferred from the working medium into the cylinder wall [J]:

$$\delta Q_w = \alpha_T \cdot F \cdot (T - T_{cm}) \cdot d\tau, \tag{60}$$

where $\alpha_T$ – heat-transfer coefficient; $F$ – cylinder bore area; $T_{cm}$ – cylinder wall temperature.

The obtained equation represents the differential of the quantity of heat removed into the cylinder wall with respect to time, i.e. the rate of the energy loss due to the transfer of heat from the gas to the wall.

The energy loss rate can be transformed into the function of the crankshaft rotation angle (Heywood, 1988):

$$dQ_w = \frac{\alpha_T \cdot F \cdot (T - T_{cm})}{6 \cdot n} \cdot d\varphi, \tag{61}$$

Taking into account the (61), the equation (59) can be written in the differential form:

$$dQ = G_{fc} \cdot Q_H \cdot dx - dQ_w, \tag{62}$$

On the basis of the obtained equation (62), the burnt out fuel amount differential $dx$ can be expressed as follows:

$$dx = \frac{dQ + dQ_w}{G_{fc} \cdot Q_H}. \tag{63}$$

$dQ$ can be found using the first law of thermodynamics (Merker et al., 2012a):

$$dQ = p \cdot dV + dU, \tag{64}$$

where $p \cdot dV$ – work for changing the volume [J]; $c$ [J].

In order to determine the elementary volume $dV$ in the formula (64), the following property of differentials in approximate analysis can be applied:

$$dV = \Delta V = V - V_0, \tag{65}$$

where $V - V_0$ – increment of the cylinder volume function, which is determined by the formula (16) for an arbitrary point on the indicator diagram in relation to the crankshaft's angular displacement $\varphi$.

The change of internal energy $dU$ can be determined by the formula (Romanov, 2010; Semyonov & Ryaboshapka, 2013):

$$dU = \frac{d(mC_{Vm} \cdot p \cdot V)}{8314}, \tag{66}$$

where $mC_{Vm}$ – average heat capacity of the working medium at a constant volume [kJ kg$^{-1}$·K$^{-1}$], which is represented by the following linear function in the paper (Romanov, 2010):

$$(mC_{Vm})_{t_0}^{t} = a_1'' + b'' \cdot (t - t_1), \tag{67}$$

where $t$ and $t_1$ – current and minimum values of the final temperature ºC of the combustion gases in the work process under consideration; the coefficients $\alpha_1''$ and $b''$ for an interval of the final temperatures ($t_1 - t_2$) of the combustion gases in the internal combustion engine are determined with the use of the following expressions for $\lambda \geq 1$:

$$\begin{cases} a_1'' = (mC_{Vm})_{t_0,(\lambda=1)}^{t} - \left[ \dfrac{(mC_{Vm})_{t_0,(\lambda=1)}^{t_1} - (mC_{Vm})_{t_0,(\lambda_{\max})}^{t_1}}{\lambda_{\max} - 1} \right] \cdot (\lambda - 1); \\ a_2'' = (mC_{Vm})_{t_0,(\lambda=1)}^{t} - \left[ \dfrac{(mC_{Vm})_{t_0,(\lambda=1)}^{t_2} - (mC_{Vm})_{t_0,(\lambda_{\max})}^{t_2}}{\lambda_{\max} - 1} \right] \cdot (\lambda - 1); \end{cases} \tag{68}$$

The values for the fraction of burnt out fuel $x$ in the formulae (47), (50) for the engine under consideration have been determined with the use of the formulae

(59) – (68) and are presented in the papers (Romanov, 2010; Semyonov & Ryaboshapka, 2013) in the form of fuel combustion (heat generation) characteristic curves.

In Fig. 3, two indicator diagrams are shown: 1) experimentally obtained indicator diagram of the 4Ч11,0/12,5 engine presented in the paper (Romanov, 2010) (solid line) and 2) indicator diagram calculated with the use of the mathematical model under consideration (marker line).

As is seen from the figure, the mathematical model provides a close approximation of the experimental indicator diagram by the calculated one. Therefore, the described model can be considered adequate and used for modelling the work process in the 4Ч11,0 / 12,5 engine cylinder.

By analysing the indicator diagram (Fig. 3), it is possible to obtain the parameters $p_c$, $p_z$, $\varphi_z$, $\beta_z$, which specify the course of the work process in the engine cylinder and determine the work of the working medium done in one cycle. Hence, these parameters have effect also on the operating efficiency of the engine. As the operating efficiency of the engine is laid down already at the stage of its design calculation and engineering, all the parameters involved in the discussed mathematical model (parameters of design, duty etc.) can be assumed as not depending on the technical condition of the engine, but corresponding to the performance data specified by the manufacturer. Subsequently, it would be reasonable to suggest that in case of using bio-diesel fuel, the parameters that depend on the fuel properties change and, accordingly, the operating efficiency of the engine also changes.

In view of the aforesaid, it is reasonable to consider the problem of determining the efficiency of the engine from the point of view of determining the efficiency of the fuel utilisation, that is, the statement of problem becomes as follows: how does the operating efficiency of the engine change, when the type of fuel changes?

The formula (28) of the mathematical model refers to the efficient performance indices, the formula (53) refers to the performance of the engine's operating duty cycle, while the parameters *C, H, S, O* and $\rho_f$ vary depending on the type of fuel.

## CONCLUSIONS

1. A mathematical model that allows to determine the engine's efficient performance indices and operating duty cycle performance and to estimate the efficiency of its operation and the efficiency of fuel utilisation, including the use of bio-diesel fuel, has been developed.

2. The formula for the calculation of fresh air charge density introduced by Pyadichev (1989) has been refined and applied for determining the density of the atmospheric air fed into the engine cylinder. The refined formula (42) is suitable for practical calculations, since it takes into account the temperature and pressure of the ambient air and their effect on the air density and, consequently, on the engine's power output in case of deviation from standard conditions.

3. The formula for the calculation of the excess air factor has been refined. The refined formula (53) takes into account the design parameters of the crank-and-rod mechanism and the gas distribution mechanism, the ambient air parameters etc., which have effect on the process of the cylinder filling, the composition of the fuel and its density and cyclic injection rate.

4. The formula for the calculation of the effective power of the engine at its rated duty based on the well-known Grinevetsky – Mazing equation for determining the work in the composite Diesel cycle has been refined. The refined formula (23) takes into account the kinematic parameters of the crank-and-rod mechanism as well as the parameters of the diesel engine operating duty cycle.

5. The formula for the calculation of the effective specific fuel consumption, which allows to evaluate the operating efficiency of the engine taking into account the type of the fuel, has been refined. The refined formula (28) takes into account the density and cyclic injection rate of the fuel, which provides for estimating the efficiency of operating the engine on biomass fuel.

6. The analysis of the indicator diagrams (Fig. 3) has proved that the mathematical model provides close agreement between the calculated data and the results of the experiments. The difference between the experimentally obtained and calculated values of the parameters is equal to: for $p_c$ – 0.1%, for $p_z$ – 0.77%, which makes reasonable using the mathematical model under consideration for the comparative calculation of the operating duty cycle parameters of the engine run on diesel and bio-diesel fuels.

7. Thus, the obtained mathematical model allows obtaining the values of the effective parameters of operation of the engine with a sufficient degree of accuracy and evaluating the efficiency of use of bio-diesel fuel by means of calculation and analysis.

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