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REMOVAL OF TECHNICAL AND ECONOMIC INDICATORS OF THE D-240 ENGINE WHEN USING BIOFUELS BY APPLYING THE DIESEL-RK SOFTWARE COMPLEX

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Ukraine is one of the energy-deficient countries because it covers its needs in fuel and energy resources from domestic reserves by only 53% (imports 75% of the required natural gas and 85% of crude oil and petroleum products). Dependence on oil imports is seen by most developed countries as a matter of national and energy security, and the use of petroleum products as energy sources poses a significant environmental risk.

According to the analysis of the state of the world engine industry, the most effective measures to improve the design of the engine are: development and implementation of a new workflow with effective methods of mixing and combustion; development and creation of the design of the system of automatic regulation of diesel rotation. frequency to improve fuel economy And to reduce toxicity in partial load mode, some cylinders are switched off at idle.

To strengthen the requirements for fuel consumption standards and the level of toxicity of exhaust gases, as well as to increase the reliability and efficiency of agricultural power engines, it is necessary to formulate measures to improve its design.

The choice of a fuel should be determined by the optimal combination of environmental and economic performance of the engine. Prospects for the use of a particular fuel of plant origin are noted. If for fuels of petroleum origin the improvement of its properties is determined by special processing, then oils with the set characteristics can be received already in "field" by selection of the corresponding grades of plants, use of fertilizers, agronomic actions, etc.

The problem of reducing the consumption of diesel fuel at idle and low load can be solved by excluding from its operation part of the cylinders (this method is widely used) and closing the cycle of one cylinder.

A comparative analysis of the main technical and environmental performance of the D-240 diesel engine when working on traditional and alternative fuels using the computer program Diesel-RK.

Keywords:. *diagnostics, engine, indicators, biofuel, D-240, indicator indicators, technical and economic characteristics*

F. 2. Fig. 9. Table 4. Ref. 9.

1. Formulation of the problem

The development and commissioning of new fuels requires theoretical and experimental research to be able to record the status and environmental performance of a diesel engine when using different fuels.

Experience of practical use of liquid biofuels, such as rapeseed oil (RA), in internal combustion engines (ICE) shows that the design of ICE does not provide for the use of this type of hydrocarbon fuel, as biofuels are not designed for this type of fuel combustion plant, and its use causes a number serious operational problems, such as loss of engine power; inability to start the engine from a cold state; failure of exhaust valves; reduction of service life of the fuel equipment (fuel pumps (PN) and atomizer sprays); increase of injection pressure to 25%; violation of the conditions of the organization of the fuel combustion process in the combustion chamber [1].

Biofuels are aggressive against certain sealing materials, in particular rubber products, paints and varnishes, as well as certain non-ferrous metals (aluminum, zinc, copper and their alloys). In most cases, this phenomenon is caused by imperfection or simplification of the technological process of production of esters (insufficient removal of catalyst, methanol residues, lack of neutralization of oil, etc.).

In terms of heat of combustion, biofuel is inferior to mineral diesel fuel (37.2 MJ/kg in biofuels against 42.5 MJ/kg in mineral diesel fuel). Therefore, the power of a biofuel engine is reduced by an average of 7%, and fuel consumption increases by about 5-8%. In addition, biofuels have less resistance to oxidation compared to diesel fuel, which is especially important for long-term storage of ethers in pure form. Oxidation can lead to an increase in acid number and viscosity, as well as the formation of harmful compounds (resins) that can block fuel filters. Excessive acidity may be due to inconsistent quality characteristics of raw materials. Therefore, esters should not be stored for more than 6 months, otherwise antioxidants should be added to them [2].

The identified operational problems are tried to be solved by giving the biofuel a set of properties that correspond to the properties of traditional petroleum fuel, adding to its composition various additives and additives, such as alcohols (methyl, ethyl) and esters (methyl-tert-butyl).

Another way to solve problems with the use of biofuels, namely operational, is to change the design of heat engines and fuel plants. Design and development of internal combustion engines, gas turbines should be carried out for a specific type of biofuel, such as rapeseed, colza, burdock or flax. The implementation of measures in this area has prospects, but their practical implementation requires a long time and significant financial investment [3].

Another area is the combustion of biofuels as part of fuel hydrocarbon mixtures, which include as the main components 50-70% of petroleum fuels and 30-50% of biofuels; adding to biofuel properties of molecular structure and structure similar to regular petroleum fuel (jet-cavitation and (or) rotor-pulsation processing).

To calculate the performance and determine the impact of biofuels and their mixtures on the D-240 engine, the program "Diesel-RK" is used.

The D-240 engine (4Ch11/12,5) is installed on the MTZ-80 general-purpose tractor (82) and self-propelled agricultural machinery [4].

2. Analysis of recent research and publications

All types of promising alternative fuels currently used in internal combustion engines, by type of raw material can be divided into large groups: petroleum, biomass and even gases of various origins [5].

To date, there are many technologies for producing diesel fuel from biomass as a renewable raw material.

This issue was addressed by: G.M. Kaletnik, M.G. Sandomirsky, L.P. Wednesday, W.F. Anisimov, Ya.Yu. White and many others.

Recently, scientific research in the field of rapeseed oil and its products - methyl or ethyl esters - has become widespread. For example, in Germany, a tractor diesel "Steur WD 408/43" was tested on a mixture of RO and diesel fuel in equal proportions. After 287 hours of operation, there was ringing of the rings, tarring of the exhaust channel and a large deposit on the exhaust valves, which means that part of the oil did not participate in the combustion process, and getting on the parts of the cylinder-piston group, formed scale [6].

3. The aim of the study

Carry out computer simulation of the D-240 engine using biofuels and its mixtures by using the DIESEL-RK software package.

4. Presenting main material

According to research and analysis in the country and abroad, the transition of diesel engines with a traditional design scheme to alternative fuels, especially biofuels based on vegetable oil, has caused a certain degree of deterioration of their normal operation. Nozzle holes and their coking. When the engine is running on oil instead of fuel for a long time, a large amount of soot in the combustion chamber can cause the engine to malfunction [7].

The calculation program provides for the determination of technical and environmental indicators during engine operation at the crankshaft speed n = 2200 min^{-1} (rated power mode) and the maximum torque mode (n = 1600 min^{-1}, μ_r = 298 N · m). The type of fuel is determined: diesel fuel (DP) -100%, biofuel B100 and mixtures B5 and B20.

Some calculation dependencies used by the software package are given below.

Residual gas ratio [8,9]:

$$\gamma_r = \frac{P_r(T_O + \Delta T)}{(\varepsilon \cdot P_a - P_r) \cdot T_r}, \qquad (1)$$

where $T_O = 288^0K$ – ambient air temperature, ε =16,0 – the degree of compression according to the technical characteristics of the engine, P_r = 0,108 MPa – exhaust gas pressure, Tr = 750^0 K – the temperature of the exhaust gases, P_0 = 0,1 MPa – atmospheric pressure, P_a – pressure at the end of the intake stroke, MPa, ΔT = 20 ^{0}K – heating temperature of the charge at the inlet.

$$P_a = \cdot P_0 - \Delta P.$$

The pressure loss at the inlet ΔP_a is calculated by the formula:

$$\Delta P_a = (\beta^2 + \xi_{ВП}) \cdot \frac{\omega^2{}_{ВП}}{2} \cdot \rho_o \cdot 10^{-6}, \text{ MPa} \qquad (2)$$

where β – the damping coefficient of the charge velocity in the considered cross section of the cylinder, $\xi_{ВП}$ – the coefficient of resistance of the intake system, $\omega_{ВП}$ – the average speed of the charge in the smallest cross section of the intake system [9].

The calculated dependences of the D-240 engine when working on different types of fuel modeled in the software package DIESEL-RK are given in table. 1-4.

It is also worth paying attention to the comparative physico-chemical composition of diesel fuel (biofuel), which determines the specifics of the diesel engine D-240 when using biofuel, which is presented in Fig. 1.

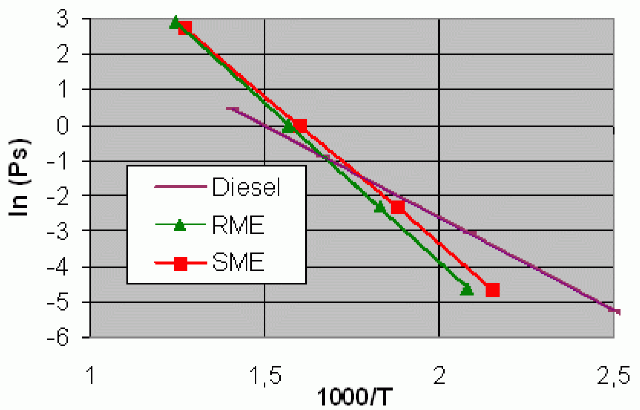

Fig. 1. Couple vapor pressure of pure fuels depending on temperature

Table 1.

Effective performance of D-240 when working on Biofuel SME B5

Fuel: Biofuel SME B5	1859.4 - T_max - Maximum Cylinder Temperature, K
--- PARAMETERS OF EFFICIENCY AND POWER ---	6.0000 - CA_p.max - Angle of Max. Cylinder Pressure, deg. A.TDC
2200.0 - RPM - Engine Speed, rev/min	19.000 - CA_t.max - Angle of Max. Cylinder Temperature, deg. A.TDC
56.990 - P_eng - Piston Engine Power, kW	3.3170 - dp/dTheta- Max. Rate of Pressure Rise, bar/deg.
6.5420 - BMEP - Brake Mean Effective Pressure, bar	1.8829 - Ring_Intn- Ringing / Knock Intensity, MW/m^2
247.39 - Torque - Brake Torque, N m	7645.9 - F_max - Max. Gas Force acting on the piston, kg
0.05468 - m_f - Mass of Fuel Supplied per cycle, g	System: Custom Fuel Injection System
0.25328 - SFC - Specific Fuel Consumption, kg/kWh	728.76 - p_inj.max- Max. Sac Injection Pres. (before nozzles), bar
0.23834 - SFC_ISO - Specific Fuel Consumption in ISO, kg/kWh	511.59 - p_inj.avr- Mean Sac Press. for Total Fuel Portion, bar
0.35632 - Eta_f - Efficiency of piston engine	19.022 - d_32 - Sauter Mean Diameter of Drops, microns
8.4212 - IMEP - Indicated Mean Effective Pressure, bar	20.000 - SOI - Start Of Injection or Ignition Timing, deg. B.TDC
0.45867 - Eta_i - Indicated Efficiency	30.474 - Phi_inj - Duration of Injection, CA deg.
9.1667 - Sp - Mean Piston Speed, m/s	10.708 - Phi_ign - Ignition Delay Period, deg.
1.5798 - FMEP - Friction Mean Effective Pressure, bar (Intern.Exp)	- ... - calculated by modified Tolstov method : 10.7
0.80549 - Eta_m - Mechanical Efficiency of Piston Engine	9.2921 - SOC - Start of Combustion, deg. B.TDC
--------- ENVIRONMENTAL PARAMETERS -----------	0.12256 - x_e.id - Fuel Mass Fraction Evaporated during Ignit. Delay
1.0000 - po_amb - Total Ambient Pressure, bar	85.000 - Phi_z - Combustion duration, deg.
288.00 - To_amb - Total Ambient Temperature, K	Phi_z 5%= 4.2; Phi_z 50%= 18.6; Phi_z 95%= 47.4
1.0000 - p_Te - Exhaust Back Pressure, bar (after turbine)	3.3546 - Rs_tdc - Swirl Ratio in the Combustion Chamber at TDC

0.98000 - po_afltr - Total Pressure after Induction Air Filter, bar

----- TURBOCHARGING AND GAS EXCHANGE -----

0.98000 - p_C - Pressure before Inlet Manifold, bar
288.00 - T_C - Temperature before Inlet Manifold, K
0.09480 - m_air - Total Mass Airflow (+EGR) of Piston Engine, kg/s
0.0000 - Eta_TC - Turbocharger Efficiency
1.0554 - po_T - Average Total Turbine Inlet Pressure, bar
762.89 - To_T - Average Total Turbine Inlet Temperature, K
0.09727 - m_gas - Mass Exhaust Gasflow of Pison Engine, kg/s
1.7219 - A/F_eq.t - Total Air Fuel Equivalence Ratio (Lambda)
0.58077 - F/A_eq.t - Total Fuel Air Equivalence Ratio
-0.29939 - PMEP - Pumping Mean Effective Pressure, bar
0.93294 - Eta_v - Volumetric Efficiency
0.91428 - Eta_vo - Volumetric Efficiency defined by Ambient Parameters
0.03602 - x_r - Residual Gas Mass Fraction
0.98413 - Phi - Coeff. of Scavenging (Delivery Ratio / Eta_v)
0.53920 - BF_int - Burnt Gas Fraction Backflowed into the Intake, %
1.5682 - %Blow-by - % of Blow-by through piston rings

---------------- INTAKE SYSTEM --------------------

0.97136 - p_int - Average Intake Manifold Pressure, bar
291.92 - T_int - Average Intake Manifold Temperature, K
35.388 - v_int - Average Gas Velocity in intake manifold, m/s
298.68 - Tw_int - Average Intake Manifold Wall Temperature, K
110.47 - hc_int - Heat Transfer Coeff. in Intake Manifold, $W/(m^2*K)$
183.61 - hc_int.p - Heat Transfer Coeff. in Intake Port, $W/(m^2*K)$
73.157 - v_int.p - Max Velocity in a Middle Section of Int. Port, m/s
10.035 - A_v.thrt - Total Effective Valve Port Throat Area, cm^2
Valve Dim. Estim.: Num=1 Dv= 51.6 Dt= 44.8 Ds=13.4 Lv= 8.1 Lv_max= 12.9 mm

----------------- EXHAUST SYSTEM -----------------------

1.0404 - p_exh - Average Exhaust Manifold Gas Pressure, bar
760.12 - T_exh - Average Exhaust Manifold Gas Temperature, K
88.626 - v_exh - Average Gas Velocity in exhaust manifold, m/s
16.398 - Sh - Strouhal number: Sh=a*Tau/L (has to be: Sh > 8)
697.40 - Tw_exh - Average Exhaust Manifold Wall Temperature, K
216.77 - hc_exh - Heat Transfer Coeff. in Exhaust Manifold, $W/(m^2*K)$
555.11 - hc_exh.p - Heat Transfer Coeff. in Exhaust Port, $W/(m^2*K)$
236.56 - v_exh.p - Max Velocity in a Middle Section of Exh. Port, m/s
10.472 - A_v.thrt - Total Effective Valve Port Throat Area, cm^2
Valve Dim. Estim.: Num=1 Dv= 51.2 Dt= 45.6 Ds=13.3 Lv= 6.7 Lv_max= 12.8 mm

-------------------- COMBUSTION ---------------------

1.7500 - A/F_eq - Air Fiel Equival. Ratio (Lambda) in the Cylinder
0.57143 - F/A_eq - Fuel Air Equivalence Ratio in the Cylinder
79.422 - p_max - Maximum Cylinder Pressure, bar
1.5000 - Rs_ivc - Swirl Ratio in the Cylinder at IVC
24.870 - W_swirl - Max. Air Swirl Velocity, m/s at cylinder R= 33

----------- ECOLOGICAL PARAMETERS -------------

16.900 - Hartridge- Hartridge Smoke Level
1.7311 - Bosch - Bosch Smoke Number
0.43585 - K,m^{-1} - Factor of Absolute Light Absorption, 1/m
0.43166 - PM - Specific Particulate Matter emission, g/kWh
778.32 - CO_2 - Specific Carbon dioxide emission, g/kWh
1441.2 - NOx.w,ppm- Fraction of wet NOx in exh. gas, ppm
14.757 - NO_2 - Specif. NOx emis. reduc. to NO_2, g/kWh (Zeldovich)
3.5471 - SE - Summary emission of PM and NOx
0.01054 - SO_2 - Specific SO_2 emission, g/kWh

-------------- CYLINDER PARAMETERS ----------------

1.2238 - p_ivc - Pressure at IVC, bar
356.01 - T_ivc - Temperature at IVC, K
45.504 - p_tdc - Compression Pressure (at TDC), bar
940.73 - T_tdc - Compression Temperature (at TDC), K
4.4622 - p_evo - Pressure at EVO, bar
1086.1 - T_evo - Temperaure at EVO, K

--------- HEAT EXCHANGE IN THE CYLINDER ------

1121.5 - T_eq - Average Equivalent Temperature of Cycle, K
343.86 - hc_c - Aver. Factor of Heat Transfer in Cyl., $W/m^2/K$
512.26 - Tw_pist - Average Piston Crown Temperature, K
420.00 - Tw_liner - Average Cylinder Liner Temperature, K
465.64 - Tw_head - Average Head Wall Temperature, K
381.30 - Tw_cool - Average Temperature of Cooled Surface head of Cylinder Head, K
398.16 - Tboil - Boiling Temp. in Liquid Cooling System, K
10707 - hc_cool - Average Factor of Heat Transfer, $W/(m^2*K)$ from head cooled surface to coolant
2143.4 - q_head - Heat Flow in a Cylinder Head, J/s
1991.0 - q_pist - Heat Flow in a Piston Crown, J/s
1480.0 - q_liner - Heat Flow in a Cylinder Liner, J/s

-- MAIN ENGINE CONSTRUCTION PARAMETERS --

16.000 - CR - Compression Ratio
4.0000 - n_inj - Number of Injector Nozzles
0.19900 - d_inj - Injector Nozzles Bore, mm
30.000 - Phi_inj - Injection Duration for specif. Inj. Profile, deg.
0.0000 - m_f_ip - Fuel Mass for specified Injection Profile, g
66.000 - EVO - Exhaust Valve Opening, deg. before BDC
16.000 - EVC - Exhaust Valve Closing, deg. after DC
16.000 - IVO - Intake Valve Opening, deg. before DC
46.000 - IVC - Intake Valve Closing, deg. after BDC

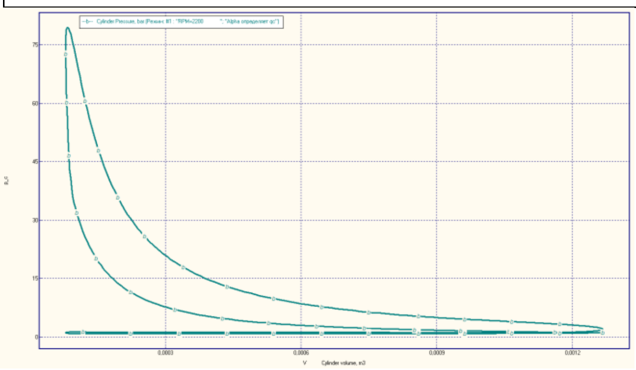

Fig. 2. The schedule of indicator indicators of working process D-240 at work on Biofuel SME B5

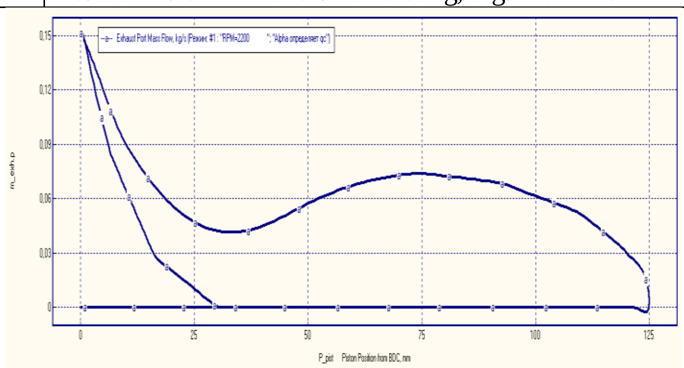

Fig. 3. Environmental performance of D-240 when working on Biofuel SME B5

Table 2.

Effective performance of D-240 when working on Biofuel SME B20

Fuel: Biofuel SME B20
-- PARAMETERS OF EFFICIENCY AND POWER --
2200.0 - RPM - Engine Speed, rev/min
57.435 - P_eng - Piston Engine Power, kW
6.5931 - BMEP - Brake Mean Effective Pressure, bar
249.32 - Torque - Brake Torque, N m
0.05320 - m_f - Mass of Fuel Supplied per cycle, g
0.24451 - SFC - Specific Fuel Consumption, kg/kWh
0.23753 - SFC_ISO - Specific Fuel Consumption in ISO, kg/kWh
0.35753 - Eta_f - Efficiency of piston engine
8.4691 - IMEP - Indicated Mean Effective Pressure, bar
0.45926 - Eta_i - Indicated Efficiency
9.1667 - Sp - Mean Piston Speed, m/s
1.5781 - FMEP - Friction Mean Effective Pressure, bar (Intern.Exp)
0.80687 - Eta_m - Mechanical Efficiency of Piston Engine
------- ENVIRONMENTAL PARAMETERS ---------
1.0000 - po_amb - Total Ambient Pressure, bar
288.00 - To_amb - Total Ambient Temperature, K
1.0000 - p_Te - Exhaust Back Pressure, bar (after turbine)
0.98000 - po_afltr - Total Pressure after Induction Air Filter, bar
----- TURBOCHARGING AND GAS EXCHANGE -----
0.98000 - p_C - Pressure before Inlet Manifold, bar
288.00 - T_C - Temperature before Inlet Manifold, K
0.09480 - m_air - Total Mass Airflow (+EGR) of Piston Engine, kg/s
0.0000 - Eta_TC - Turbocharger Efficiency
1.0557 - po_T - Average Total Turbine Inlet Pressure, bar
776.13 - To_T - Average Total Turbine Inlet Temperature, K
0.09630 - m_gas - Mass Exhaust Gasflow of Pison Engine, kg/s
1.7225 - A/F_eq.t - Total Air Fuel Equivalence Ratio (Lambda)
0.58054 - F/A_eq.t - Total Fuel Air Equivalence Ratio
-0.29788 - PMEP - Pumping Mean Effective Pressure, bar
0.93259 - Eta_v - Volumetric Efficiency
0.91393 - Eta_vo - Volumetric Efficiency defined by Ambient Parameters
0.03556 - x_r - Residual Gas Mass Fraction
0.98454 - Phi - Coeff. of Scavenging (Delivery Ratio / Eta_v)
0.53008 - BF_int - Burnt Gas Fraction Backflowed into the Intake, %
1.5612 - %Blow-by - % of Blow-by through piston rings
----------------- INTAKE SYSTEM -----------------
0.97133 - p_int - Average Intake Manifold Pressure, bar
291.94 - T_int - Average Intake Manifold Temperature, K
35.392 - v_int - Average Gas Velocity in intake manifold, m/s
298.69 - Tw_int - Average Intake Manifold Wall Temperature, K
110.47 - hc_int - Heat Transfer Coeff. in Intake Manifold, W/(m^2*K)
183.57 - hc_int.p - Heat Transfer Coeff. in Intake Port, W/(m^2*K)
73.158 - v_int.p - Max Velocity in a Middle Section of Int. Port, m/s
10.035 - A_v.thrt - Total Effective Valve Port Throat Area, cm^2
Valve Dim. Estim.: Num=1 Dv= 51.6 Dt= 44.8 Ds=13.4 Lv= 8.1 Lv_max= 12.9 mm
---------------- EXHAUST SYSTEM -----------------
1.0404 - p_exh - Average Exhaust Manifold Gas Pressure, bar
73.32 - T_exh - Average Exhaust Manifold Gas Temperature, K
89.091 - v_exh - Average Gas Velocity in exhaust manifold, m/s
16.540 - Sh - Strouhal number: Sh=a*Tau/L (has to be: Sh > 8)
707.74 - Tw_exh - Average Exhaust Manifold Wall Temperature, K
218.15 - hc_exh - Heat Transfer Coeff. in Exhaust Manifold, W/(m^2*K)
558.66 - hc_exh.p - Heat Transfer Coeff. in Exhaust Port, W/(m^2*K)
238.69 - v_exh.p - Max Velocity in a Middle Section of Exh. Port, m/s

1869.1 - T_max - Maximum Cylinder Temperature, K
6.0000 - CA_p.max - Angle of Max. Cylinder Pressure, deg. A.TDC
19.000 - CA_t.max - Angle of Max. Cylinder Temperature, deg. A.TDC
3.2342 - dp/dTheta- Max. Rate of Pressure Rise, bar/deg.
1.7901 - Ring_Intn- Ringing / Knock Intensity, MW/m2
7665.8 - F_max - Max. Gas Force acting on the piston, kg
System: Custom Fuel Injection System
715.97 - p_inj.max- Max. Sac Injection Pres. (before nozzles), bar
502.84 - p_inj.avr- Mean Sac Press. for Total Fuel Portion, bar
18.467 - d_32 - Sauter Mean Diameter of Drops, microns
20.000 - SOI - Start Of Injection or Ignition Timing, deg. B.TDC
30.136 - Phi_inj - Duration of Injection, CA deg.
12.282 - Phi_ign - Ignition Delay Period, deg.
- ... - calculated by modified Tolstov method : 12.3
7.7178 - SOC - Start of Combustion, deg. B.TDC
0.18651 - x_e.id - Fuel Mass Fraction Evaporated during Ignit. Delay
90.400 - Phi_z - Combustion duration, deg.
Phi_z 5%= 3.4; Phi_z 50%= 17.6; Phi_z 95%= 48.0
3.3561 - Rs_tdc - Swirl Ratio in the Combustion Chamber at TDC
1.5000 - Rs_ivc - Swirl Ratio in the Cylinder at IVC
24.880 - W_swirl - Max. Air Swirl Velocity, m/s at cylinder R= 33
--------- ECOLOGICAL PARAMETERS ----------
17.133 - Hartridge- Hartridge Smoke Level
1.7516 - Bosch - Bosch Smoke Number
0.44245 - K,m^{-1} - Factor of Absolute Light Absorption, 1/m
0.43072 - PM - Specific Particulate Matter emission, g/kWh
769.40 - CO_2 - Specific Carbon dioxide emission, g/kWh
1289.5 - NOx.w,ppm- Fraction of wet NOx in exh. gas, ppm
13.087 - NO_2 - Specif. NOx emis. reduc. to NO_2, g/kWh (Zeldovich)
3.3053 - SE - Summary emission of PM and NOx
0.00513 - SO_2 - Specific SO_2 emission, g/kWh

----------- CYLINDER PARAMETERS -------------
1.2237 - p_ivc - Pressure at IVC, bar
356.14 - T_ivc - Temperature at IVC, K
45.498 - p_tdc - Compression Pressure (at TDC), bar
940.99 - T_tdc - Compression Temperature (at TDC), K
4.4978 - p_evo - Pressure at EVO, bar
1105.8 - T_evo - Temperaure at EVO, K
------ HEAT EXCHANGE IN THE CYLINDER -------
1125.9 - T_eq - Average Equivalent Temperature of Cycle, K
341.49 - hc_c - Aver. Factor of Heat Transfer in Cyl., W/m^2/K
512.19 - Tw_pist - Average Piston Crown Temperature, K
420.00 - Tw_liner - Average Cylinder Liner Temperature, K
465.63 - Tw_head - Average Head Wall Temperature, K
381.28 - Tw_cool - Average Temperature of Cooled Surface head of Cylinder Head, K
398.16 - Tboil - Boiling Temp. in Liquid Cooling System, K
10702 - hc_cool - Average Factor of Heat Transfer, W/(m^2*K) from head cooled surface to coolant
2142.9 - q_head - Heat Flow in a Cylinder Head, J/s
1991.7 - q_pist - Heat Flow in a Piston Crown, J/s
1515.2 - q_liner - Heat Flow in a Cylinder Liner, J/s
-- MAIN ENGINE CONSTRUCTION PARAMETERS --
16.000 - CR - Compression Ratio
4.0000 - n_inj - Number of Injector Nozzles

10.472 - A_v.thrt - Total Effective Valve Port Throat Area, cm^2	0.19900 - d_inj - Injector Nozzles Bore, mm
Valve Dim. Estim.: Num=1 Dv= 51.2 Dt= 45.6 Ds=13.3 Lv= 6.7 Lv_max= 12.8 mm	30.000 - Phi_inj - Injection Duration for specif. Inj. Profile, deg.
--------------- **COMBUSTION** --------------	0.0000 - m_f_ip - Fuel Mass for specified Injection Profile, g
1.7500 - A/F_eq - Air Fiel Equival. Ratio (Lambda) in the Cylinder	66.000 - EVO - Exhaust Valve Opening, deg. before BDC
0.57143 - F/A_eq - Fuel Air Equivalence Ratio in the Cylinder	16.000 - EVC - Exhaust Valve Closing, deg. after DC
79.629 - p_max - Maximum Cylinder Pressure, bar	16.000 - IVO - Intake Valve Opening, deg. before DC
	46.000 - IVC - Intake Valve Closing, deg. after BDC

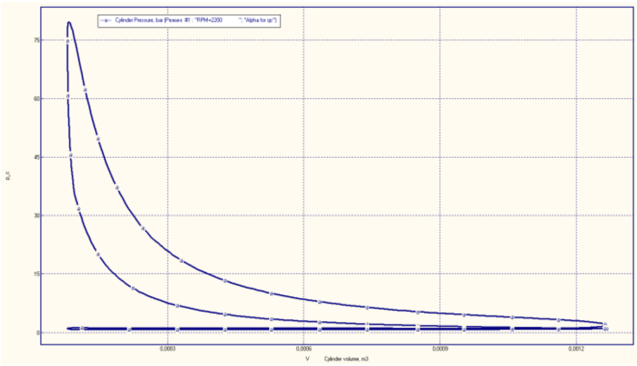

Fig. 4. The schedule of indicator indicators of working process D-240 at work on Biofuel SME B20

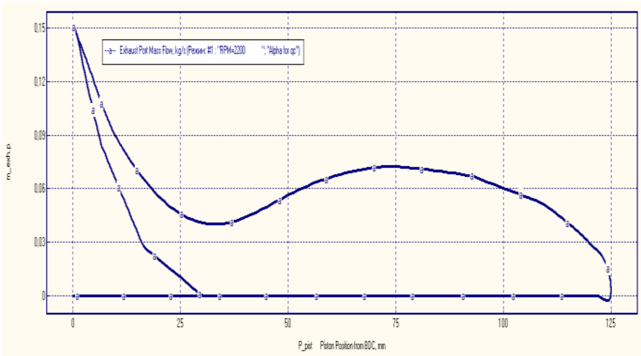

Fig. 5. Environmental performance of the D-240 workflow when working on Biofuel SME B20

Table 3.

Effective performance of D-240 when working on Diesel No. 2

Fuel: Diesel No. 2	1889.7 - T_max - Maximum Cylinder Temperature, K
--- **PARAMETERS OF EFFICIENCY AND POWER** ---	5.0000 - CA_p.max - Angle of Max. Cylinder Pressure, deg. A.TDC
2200.0 - RPM - Engine Speed, rev/min	20.000 - CA_t.max - Angle of Max. Cylinder Temperature, deg. A.TDC
58.336 - P_eng - Piston Engine Power, kW	3.3297 - dp/dTheta- Max. Rate of Pressure Rise, bar/deg.
6.6965 - BMEP - Brake Mean Effective Pressure, bar	1.8742 - Ring_Intn- Ringing / Knock Intensity, MW/m^2
253.23 - Torque - Brake Torque, N m	7802.9 - F_max - Max. Gas Force acting on the piston, kg
0.05178 - m_f - Mass of Fuel Supplied per cycle, g	System: Custom Fuel Injection System
0.23433 - SFC - Specific Fuel Consumption, kg/kWh	702.62 - p_inj.max- Max. Sac Injection Pres. (before nozzles), bar
0.23493 - SFC_ISO - Specific Fuel Consumption in ISO, kg/kWh	492.07 - p_inj.avr- Mean Sac Press. for Total Fuel Portion, bar
0.36148 - Eta_f - Efficiency of piston engine	17.842 - d_32 - Sauter Mean Diameter of Drops, microns
8.5729 - IMEP - Indicated Mean Effective Pressure, bar	20.000 - SOI - Start Of Injection or Ignition Timing, deg. B.TDC
0.46277 - Eta_i - Indicated Efficiency	29.813 - Phi_inj - Duration of Injection, CA deg.
9.1667 - Sp - Mean Piston Speed, m/s	13.882 - Phi_ign - Ignition Delay Period, deg.
1.5783 – FMEP - Friction Mean Effective Pressure, bar (Intern.Exp)	- ... - calculated by modified Tolstov method : 13.9
0.80926 - Eta_m - Mechanical Efficiency of Piston Engine	6.1176 - SOC - Start of Combustion, deg. B.TDC
-------- **ENVIRONMENTAL PARAMETERS** ---------	0.26612 - x_e.id - Fuel Mass Fraction Evaporated during Ignit. Delay
1.0000 - po_amb - Total Ambient Pressure, bar	83.400 - Phi_z - Combustion duration, deg.
288.00 - To_amb - Total Ambient Temperature, K	Phi_z 5%= 3.2; Phi_z 50%= 16.0; Phi_z 95%= 44.2
1.0000 - p_Te - Exhaust Back Pressure, bar (after turbine)	3.3558 - Rs_tdc - Swirl Ratio in the Combustion Chamber at TDC
0.98000 - po_afltr - Total Pressure after Induction Air Filter, bar	1.5000 - Rs_ivc - Swirl Ratio in the Cylinder at IVC
----- **TURBOCHARGING AND GAS EXCHANGE** ----	24.879 - W_swirl - Max. Air Swirl Velocity, m/s at cylinder R= 33
0.98000 - p_C - Pressure before Inlet Manifold, bar	----------- **ECOLOGICAL PARAMETERS** -----------
288.00 - T_C - Temperature before Inlet Manifold, K	16.469 - Hartridge- Hartridge Smoke Level
0.09479 - m_air - Total Mass Airflow (+EGR) of Piston Engine, kg/s	1.6933 - Bosch - Bosch Smoke Number
0.0000 - Eta_TC - Turbocharger Efficiency	0.42366 - K,m^{-1} - Factor of Absolute Light Absorption, 1/m
1.0553 - po_T - Average Total Turbine Inlet Pressure, bar	0.40604 - PM - Specific Particulate Matter emission, g/kWh
774.28 - To_T - Average Total Turbine Inlet Temperature, K	755.06 - CO_2 - Specific Carbon dioxide emission, g/kWh
0.09644 - m_gas - Mass Exhaust Gasflow of Pison Engine, kg/s	1203.0 - NOx.w,ppm- Fraction of wet NOx in exh. gas, ppm
1.7224 - A/F_eq.t - Total Air Fuel Equivalence Ratio (Lambda)	12.019 - NO_2 - Specif. NOx emis. reduc. to NO_2, g/kWh (Zeldovich)
0.58057 - F/A_eq.t - Total Fuel Air Equivalence Ratio	3.0705 - SE - Summary emission of PM and NOx
-0.29806 - PMEP - Pumping Mean Effective Pressure, bar	0.00937 - SO_2 - Specific SO_2 emission, g/kWh
0.93254 - Eta_v - Volumetric Efficiency	----------- **CYLINDER PARAMETERS** --------------
0.91389 - Eta_vo - Volumetric Efficiency defined by Ambient Parameters	1.2238 - p_ivc - Pressure at IVC, bar

0.03562 - x_r - Residual Gas Mass Fraction	356.16 - T_ivc - Temperature at IVC, K
0.98448 - Phi - Coeff. of Scavenging (Delivery Ratio / Eta_v)	45.501 - p_tdc - Compression Pressure (at TDC), bar
0.53147 - BF_int - Burnt Gas Fraction Backflowed into the Intake, %	941.10 - T_tdc - Compression Temperature (at TDC), K
1.5619 - %Blow-by - % of Blow-by through piston rings	4.4915 - p_evo - Pressure at EVO, bar
------------- **INTAKE SYSTEM** ------------------	1102.6 - T_evo - Temperaure at EVO, K
0.97134 - p_int - Average Intake Manifold Pressure, bar	----- **HEAT EXCHANGE IN THE CYLINDER** ---------
291.94 - T_int - Average Intake Manifold Temperature, K	1129.6 - T_eq - Average Equivalent Temperature of Cycle, K
35.384 - v_int - Average Gas Velocity in intake manifold, m/s	341.58 - hc_c - Aver. Factor of Heat Transfer in Cyl., W/m^2/K
298.69 - Tw_int - Average Intake Manifold Wall Temperature, K	512.87 - Tw_pist - Average Piston Crown Temperature, K
110.46 - hc_int - Heat Transfer Coeff. in Intake Manifold, W/(m^2*K)	420.00 - Tw_liner - Average Cylinder Liner Temperature, K
183.56 - hc_int.p - Heat Transfer Coeff. in Intake Port, W/(m^2*K)	466.12 - Tw_head - Average Head Wall Temperature, K
73.152 - v_int.p - Max Velocity in a Middle Section of Int. Port, m/s	381.36 - Tw_cool - Average Temperature of Cooled Surface
10.035 - A_v.thrt - Total Effective Valve Port Throat Area, cm^2	head of Cylinder Head, K
Valve Dim. Estim.: Num=1 Dv= 51.6 Dt= 44.8 Ds=13.4 Lv= 8.1 Lv_max= 12.9 mm	398.16 - Tboil - Boiling Temp. in Liquid Cooling System, K
--------------- **EXHAUST SYSTEM** -----------------	10722 - hc_cool - Average Factor of Heat Transfer, W/(m^2*K)
1.0404 - p_exh - Average Exhaust Manifold Gas Pressure, bar	from head cooled surface to coolant
771.48 - T_exh - Average Exhaust Manifold Gas Temperature, K	2153.8 - q_head - Heat Flow in a Cylinder Head, J/s
89.011 - v_exh - Average Gas Velocity in exhaust manifold, m/s	2002.0 - q_pist - Heat Flow in a Piston Crown, J/s
16.520 - Sh - Strouhal number: Sh=a*Tau/L (has to be: Sh > 8)	1517.3 - q_liner - Heat Flow in a Cylinder Liner, J/s
706.11 - Tw_exh - Average Exhaust Manifold Wall Temperature, K	--**MAIN ENGINE CONSTRUCTION PARAMETERS**--
217.73 - hc_exh - Heat Transfer Coeff. in Exhaust Manifold, W/(m^2*K)	16.000 - CR - Compression Ratio
557.59 - hc_exh.p - Heat Transfer Coeff. in Exhaust Port, W/(m^2*K)	4.0000 - n_inj - Number of Injector Nozzles
238.34 - v_exh.p - Max Velocity in a Middle Section of Exh. Port, m/s	0.19900 - d_inj - Injector Nozzles Bore, mm
10.472 - A_v.thrt - Total Effective Valve Port Throat Area, cm^2	30.000 - Phi_inj - Injection Duration for specif. Inj. Profile, deg.
Valve Dim. Estim.: Num=1 Dv= 51.2 Dt= 45.6 Ds=13.3 Lv= 6.7 Lv_max= 12.8 mm	0.0000 - m_f_ip - Fuel Mass for specified Injection Profile, g
-------------- **COMBUSTION** ----------------	66.000 - EVO - Exhaust Valve Opening, deg. before BDC
1.7500 - A/F_eq - Air Fiel Equival. Ratio (Lambda) in the Cylinder	16.000 - EVC - Exhaust Valve Closing, deg. after DC
0.57143 - F/A_eq - Fuel Air Equivalence Ratio in the Cylinder	16.000 - IVO - Intake Valve Opening, deg. before DC
81.054 - p_max - Maximum Cylinder Pressure, bar	46.000 - IVC - Intake Valve Closing, deg. after BDC

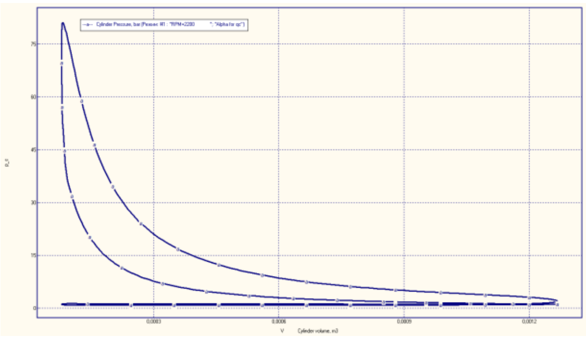

Fig. 6. The schedule of indicator indicators of working process D-240 at work on Diesel No. 2

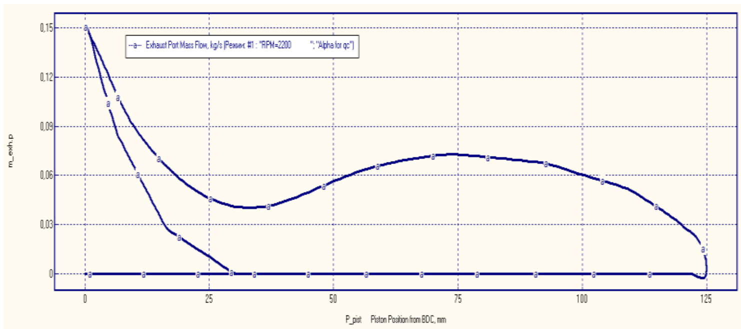

Fig. 7. Environmental performance of the workflow D-240 when working on Diesel No. 2

Table 4.

Effective performance of D-240 when working on Biofuel SME B100

Fuel: Biofuel SME B100	1463.1 - T_max - Maximum Cylinder Temperature, K
---- **PARAMETERS OF EFFICIENCY AND POWER** ---	6.0000 - CA_p.max - Angle of Max. Cylinder Pressure, deg. A.TDC
2200.0 - RPM - Engine Speed, rev/min	24.000 - CA_t.max - Angle of Max. Cylinder Temperature, deg. A.TDC
43.804 - P_eng - Piston Engine Power, kW	1.7904 - dp/dTheta- Max. Rate of Pressure Rise, bar/deg.
5.0284 - BMEP - Brake Mean Effective Pressure, bar	0.62077 - Ring_Intn- Ringing / Knock Intensity, MW/m2
190.15 - Torque - Brake Torque, N m	5993.3 - F_max - Max. Gas Force acting on the piston, kg
0.05964 - m_f - Mass of Fuel Supplied per cycle, g	System: Custom Fuel Injection System
0.35941 - SFC - Specific Fuel Consumption, kg/kWh	781.27 - p_inj.max- Max. Sac Injection Pres. (before nozzles), bar
0.30718 - SFC_ISO - Specific Fuel Consumption in ISO, kg/kWh	544.44 - p_inj.avr- Mean Sac Press. for Total Fuel Portion, bar
0.27654 - Eta_f - Efficiency of piston engine	20.367 - d_32 - Sauter Mean Diameter of Drops, microns
6.8487 - IMEP - Indicated Mean Effective Pressure, bar	20.000 - SOI - Start Of Injection or Ignition Timing, deg. B.TDC
0.37665 - Eta_i - Indicated Efficiency	31.605 - Phi_inj - Duration of Injection, CA deg.
9.1667 - Sp - Mean Piston Speed, m/s	3.8835 - Phi_ign - Ignition Delay Period, deg.

1.5221 - FMEP - Friction Mean Effective Pressure, bar (Intern.Exp)
0.76764 - Eta_m - Mechanical Efficiency of Piston Engine
--------- **ENVIRONMENTAL PARAMETERS** -------------
1.0000 - po_amb - Total Ambient Pressure, bar
288.00 - To_amb - Total Ambient Temperature, K
1.0000 - p_Te - Exhaust Back Pressure, bar (after turbine)
0.98000 - po_afltr - Total Pressure after Induction Air Filter, bar
------ **TURBOCHARGING AND GAS EXCHANGE** -----
0.98000 - p_C - Pressure before Inlet Manifold, bar
288.00 - T_C - Temperature before Inlet Manifold, K
0.09540 - m_air - Total Mass Airflow (+EGR) of Piston Engine, kg/s
0.0000 - Eta_TC - Turbocharger Efficiency
1.0564 - po_T - Average Total Turbine Inlet Pressure, bar
839.10 - To_T - Average Total Turbine Inlet Temperature, K
0.09606 - m_gas - Mass Exhaust Gasflow of Pison Engine, kg/s
1.7228 - A/F_eq.t - Total Air Fuel Equivalence Ratio (Lambda)
0.58044 - F/A_eq.t - Total Fuel Air Equivalence Ratio
-0.29821 - PMEP - Pumping Mean Effective Pressure, bar
0.93858 - Eta_v - Volumetric Efficiency
0.91981 - Eta_vo - Volumetric Efficiency defined by Ambient Parameters
0.03374 - x_r - Residual Gas Mass Fraction
0.98440 - Phi - Coeff. of Scavenging (Delivery Ratio / Eta_v)
0.49142 - BF_int - Burnt Gas Fraction Backflowed into the Intake, %
1.4722 - %Blow-by - % of Blow-by through piston rings
-------------------- **INTAKE SYSTEM** ----------------------
0.97118 - p_int - Average Intake Manifold Pressure, bar
291.93 - T_int - Average Intake Manifold Temperature, K
35.664 - v_int - Average Gas Velocity in intake manifold, m/s
298.71 - Tw_int - Average Intake Ma'nifold Wall Temperature, K
110.91 - hc_int - Heat Transfer Coeff. in Intake Manifold, W/(m^2*K)
184.41 - hc_int.p - Heat Transfer Coeff. in Intake Port, W/(m^2*K)
73.448 - v_int.p - Max Velocity in a Middle Section of Int. Port, m/s
10.035 - A_v.thrt - Total Effective Valve Port Throat Area, cm2
Valve Dim. Estim.: Num=1 Dv= 51.6 Dt= 44.8 Ds=13.4 Lv= 8.1 Lv_max= 12.9 mm
------------------ **EXHAUST SYSTEM** ----------------------
1.0399 - p_exh - Average Exhaust Manifold Gas Pressure, bar
835.80 - T_exh - Average Exhaust Manifold Gas Temperature, K
95.700 - v_exh - Average Gas Velocity in exhaust manifold, m/s
17.195 - Sh - Strouhal number: Sh=a*Tau/L (has to be: Sh > 8)
767.69 - Tw_exh - Average Exhaust Manifold Wall Temperature, K
226.27 - hc_exh - Heat Transfer Coeff. in Exhaust Manifold, W/(m^2*K)
579.44 - hc_exh.p - Heat Transfer Coeff. in Exhaust Port, W/(m^2*K)
251.43 - v_exh.p - Max Velocity in a Middle Section of Exh. Port, m/s
10.472 - A_v.thrt - Total Effective Valve Port Throat Area, cm^2
Valve Dim. Estim.: Num=1 Dv= 51.2 Dt= 45.6 Ds=13.3 Lv= 6.7 Lv_max= 12.8 mm
-------------------- **COMBUSTION** ----------------------
1.7500 - A/F_eq - Air Fiel Equival. Ratio (Lambda) in the Cylinder
0.57143 - F/A_eq - Fuel Air Equivalence Ratio in the Cylinder
62.256 - p_max - Maximum Cylinder Pressure, bar
- ... - calculated by modified Tolstov method : 3.9
16.117 - SOC - Start of Combustion, deg. B.TDC
0.00427 - x_e.id - Fuel Mass Fraction Evaporated during Ignit. Delay
229.40 - Phi_z - Combustion duration, deg.
Phi_z 5%= 6.8; Phi_z 50%= 36.4; Phi_z 95%=141.4
3.3619 - Rs_tdc - Swirl Ratio in the Combustion Chamber at TDC
1.5000 - Rs_ivc - Swirl Ratio in the Cylinder at IVC
24.924 - W_swirl - Max. Air Swirl Velocity, m/s at cylinder R= 33
----------- **ECOLOGICAL PARAMETERS** -------------
60.291 - Hartridge- Hartridge Smoke Level
4.4388 - Bosch - Bosch Smoke Number
2.1485 - K,m^{-1} - Factor of Absolute Light Absorption, 1/m
2.1593 - PM - Specific Particulate Matter emission, g/kWh
1029.1 - CO_2 - Specific Carbon dioxide emission, g/kWh
663.50 - NOx.w,ppm- Fraction of wet NOx in exh. gas, ppm
8.8930 - NO_2 - Specif. NOx emis. reduc. to NO_2, g/kWh (Zeldovich)
8.4681 - SE - Summary emission of PM and NOx
0.03594 - SO_2 - Specific SO_2 emission, g/kWh
-------------- **CYLINDER PARAMETERS** ------------------
1.2229 - p_ivc - Pressure at IVC, bar
354.45 - T_ivc - Temperature at IVC, K
45.399 - p_tdc - Compression Pressure (at TDC), bar
935.15 - T_tdc - Compression Temperature (at TDC), K
4.8959 - p_evo - Pressure at EVO, bar
1209.1 - T_evo - Temperaure at EVO, K
------- **HEAT EXCHANGE IN THE CYLINDER** --------
960.27 - T_eq - Average Equivalent Temperature of Cycle, K
306.33 - hc_c - Aver. Factor of Heat Transfer in Cyl., W/m^2/K
480.69 - Tw_pist - Average Piston Crown Temperature, K
20.00 - Tw_liner - Average Cylinder Liner Temperature, K
437.00 - Tw_head - Average Head Wall Temperature, K
377.11 - Tw_cool - Average Temperature of Cooled Surface head of Cylinder Head, K
398.16 - Tboil - Boiling Temp. in Liquid Cooling System, K
9491.6 - hc_cool - Average Factor of Heat Transfer, W/(m^2*K) from head cooled surface to coolant
1523.3 - q_head - Heat Flow in a Cylinder Head, J/s
1396.1 - q_pist - Heat Flow in a Piston Crown, J/s
1535.3 - q_liner - Heat Flow in a Cylinder Liner, J/s
--**MAIN ENGINE CONSTRUCTION PARAMETERS**--
16.000 - CR - Compression Ratio
4.0000 - n_inj - Number of Injector Nozzles
0.19900 - d_inj - Injector Nozzles Bore, mm
30.000 - Phi_inj - Injection Duration for specif. Inj. Profile, deg.
0.0000 - m_f_ip - Fuel Mass for specified Injection Profile, g
66.000 - EVO - Exhaust Valve Opening, deg. before BDC
16.000 - EVC - Exhaust Valve Closing, deg. after DC
16.000 - IVO - Intake Valve Opening, deg. before DC
46.000 - IVC - Intake Valve Closing, deg. after BDC

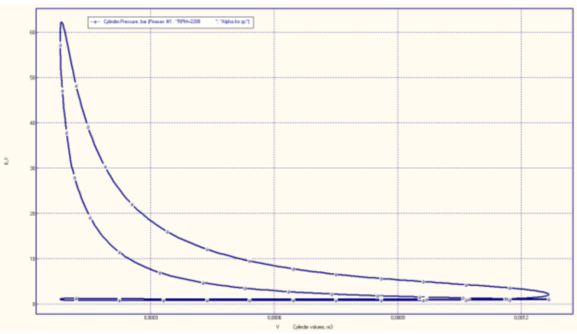

Fig. 8. The schedule of indicator indicators of working process D-240 when working on Biofuel SME B100

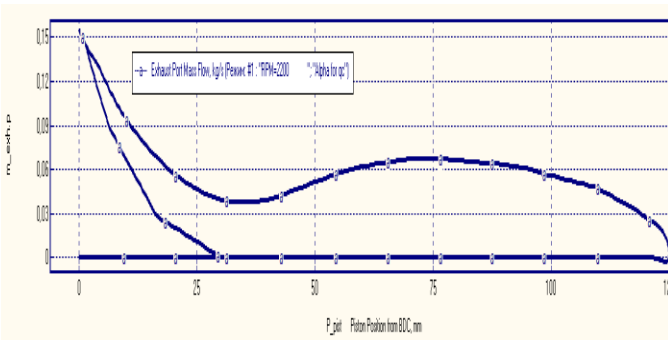

Fig. 9. Environmental performance of D-240 when working on Biofuel SME B100

The shortest way to the widespread use of liquid biofuels as fuel in tractor engines without changing their design is to improve their fuel systems through the introduction and use of new processes in the technological cycle of fuel preparation. use of two-fuel systems, including systems of supply of oil and biofuels, use of regular oil fuel as starting and reserve fuel

As for the economic indicators of work, according to preliminary estimates (excluding the cost of restoring soil fertility), the benefits of using biofuels are obvious.

Environmental indicators that determine the safety of human life and associated with the use of agricultural machinery that consumes biofuels by 8-10% are better than the use of mineral diesel fuel.

5. Conclusions

1. Because biofuels contain oxygen-containing substances (O = 10.9%), its heat of combustion is lower (Q_n = 39.45 MJ/kg) than that of diesel fuel (Q_n = 42), 5 MJ/kg, Figure 13) (O = 0.4%). This fact leads to a decrease in diesel power (up to 25% in nominal mode).

2. Compared with diesel fuel, the specific consumption of biofuel in the nominal mode of operation of the engine increased by 37%.

3. Studies of fuel injection and mixing show that the average droplet diameter in the MERO study increased to 20%, which increases the range of the jet compared to diesel fuel and adversely affects the mixing process. And it burns.

4. The use of vegetable alternative fuels with a reduction of carbon content by 10% can reduce CO_2 emissions. The rate of formation of soot particles during the combustion of biofuels is 8.8 times higher than the rate of combustion of diesel fuel.

Recommendations. In order for the technical and economic performance of MERO to reach the optimal level, it is necessary to strengthen such processes as cleaning, injection, mixing and incineration. Heating of the injected fuel (~ 70 °C) can positively influence these processes, which will lead to the improvement of physicochemical parameters of the fuel, increase of the fuel injection pressure (~ 80 MPa) will also reduce the diameter of sprayed fuel droplets, air turbulence. evaporation and stirring.

Список використаних джерел

1. Eberhart S. A., Russel W. A. Stability parameters for comparing varieties. *Crop Sci.* 1986. V. 6. № 1. P. 34–40.
2. Калетнік Г. М. Біопалива: ефективність їх виробництва та споживання в АПК України: навч. посібник. К: Аграрна наука, 2010. 327 с.
3. Окоча А. І., Антипенко А. М. Паливно-мастильні та інші експлуатаційні матеріали. К.: Урожай, 1996. 336 с.
4. Murugesan A., Subramanian R., Nedunchezhian N. Biodiesel as an alternative fuel for diesel engines. *Renew sust energy rev*. 2009. P. 653–662.
5. Семенов В. Біодизельне паливо для України. Вісник Національної академії наук України. 2007. № 4. С. 18–22.
6. Tokarchuk D. M., Pryshliak N. V., Tokarchuk O. A., Mazur K. V. Technical and economic aspects of biogas production at a small agricultural enterprise with modeling of the optimal distribution of energy. *INMATEH – Agricultural Engineering*. 2020. Vol. 61. №. 2. P. 339–349.
7. Анісімов В. Ф., Яцковський В. І., П’ясецький А. А. Рябошапка В. Б. Напрямки створення багатопаливних двигунів на базі дизельного циклу. *Промислова гідравліка і пневматика*. 2011. №2 (32). С. 100–105.
8. Бурлака С. А., Явдик В. В., Єленич А. П. Методи досліджень та способи оцінки впливу палив з відновлюваних ресурсів на роботу дизельного двигуна. *Вісник Хмельницького національного університету*. 2019. №2 (271). С. 212–220.
9. Малаков О. І., Бурлака С. А., Михальова Ю. О. Математичне моделювання та основи конструювання вібраційних змішувачів. *Вісник Хмельницького національного університету*. 2019. № 5 (277). С. 30–33.

Reference

[1] Eberhart, S. A., Russel, W. A. (1986). Stability parameters for comparing varieties. *Crop Sci. 6(1).* 34–40. [in English].
[2] Kaletnik, H.M. (2010). *Biopalyva: efektyvnist' yikh vyrobnytstva ta spozhyvannya v APK Ukrayiny*: *navch. posibnyk*. K: Ahrarna nauka. [In Ukrainian].
[3] Okocha, A. I., Antypenko, A. M. (1996). *Palyvno-mastyl'ni ta inshi ekspluatatsiyni materialy*. K.: Urozhay. [in Ukrainian].
[4] Murugesan, A., Subramanian, R., Nedunchezhian, N. (2009). Biodiesel as an alternative fuel for diesel engines. *Renew sust energy rev*. 653–662. [in English].
[5] Semenov, V. (2007). Biodyzel′ne palyvo dlya Ukrayiny. *Visnyk Natsional'noyi akademiyi nauk Ukrayiny*, *4*. 18–22. [in Ukrainian].
[6] Tokarchuk, D. M., Pryshliak, N. V., Tokarchuk, O. A., Mazur, K. V. (2020). Technical and economic aspects of biogas production at a small agricultural enterprise with modeling of the optimal distribution of energy. *INMATEH – Agricultural Engineering*. *61(2)*. 339–349. [in English].
[7] Anisimov, V. F., Yatskovs′kyy, V. I., P′yasets′kyy, A. A., Ryaboshapka, V. B. (2011). Napryamky stvorennya bahatopalyvnykh dvyhuniv na bazi dyzel′noho tsyklu. *Promyslova hidravlika i pnevmatyka*. *2 (32)*. 100–105. [in Ukrainian].
[8] Burlaka, S.A., Yavdyk, V.V., Yelenych, A.P. (2019). Metody doslidzhen′ ta sposoby otsinky vplyvu palyv z vidnovlyuvanykh resursiv na robotu dyzel′noho dvyhuna. *Visnyk Khmel'nyts'koho natsional'noho universytetu*. *2 (271)*. 212–220. [in Ukrainian].
[9] Malakov, O.I., Burlaka, S.A., Mykhal′ova, Y.O. (2019). Matematychne modelyuvannya ta osnovy konstruyuvannya vibratsiynykh zmishuvachiv. *Visnyk Khmel'nyts'koho natsional'noho universytetu*. *5*(*277*). 30–33. [in Ukrainian].

ЗНЯТТЯ ТЕХНІКО-ЕКОНОМІЧНИХ ПОКАЗНИКІВ ДВИГУНА Д-240 ПРИ ВИКОРИСТАННЯ БІОПАЛИВА ШЛЯХОМ ЗАСТОСУВАННЯ ПРОГРАМНОГО КОМПЛЕКСУ ДИЗЕЛЬ-РК

Україна є однією з енергетично дефіцитних країн, оскільки покриває свої потреби в паливно-енергетичних ресурсах за рахунок внутрішніх запасів лише на 53% (імпортує 75% необхідного природного газу та 85% сирої нафти та нафтопродуктів). Залежність від імпорту нафти більшість розвинених країн розглядає як питання національної та енергетичної безпеки, а використання нафтопродуктів як джерел енергії створює значний екологічний ризик.

Згідно з аналізом стану світової машинобудівної промисловості, найбільш ефективними заходами щодо вдосконалення конструкції двигуна є: розробка та впровадження нового робочого процесу з ефективними методами змішування та згоряння; розробка та створення конструкції системи автоматичного регулювання обертання дизеля. частота для покращення економії палива, а для зниження токсичності в режимі часткового навантаження деякі циліндри вимикаються на холостому ходу.

Для посилення вимог до нормативів витрат палива та рівня токсичності відпрацьованих газів, а також підвищення надійності та ефективності сільськогосподарських силових двигунів необхідно сформулювати заходи щодо вдосконалення його конструкції.

Вибір палива має визначатися оптимальним поєднанням екологічних та економічних показників двигуна. Відзначаються перспективи використання того чи іншого палива рослинного походження. Якщо для палива нафтового походження поліпшення його властивостей визначається спеціальною обробкою, то олії із заданими характеристиками можна отримати вже в «польові» шляхом підбору відповідних сортів рослин, застосування добрив, агротехнічних заходів тощо.

Проблему зниження витрати дизельного палива на холостому ходу та малому навантаженні можна вирішити, виключивши з його роботи частину циліндрів (цей метод широко використовується) і замикаючи цикл одного циліндра.Проведено порівняльний аналіз основних технічних та екологічних показників дизельного двигуна Д-240 при роботі на традиційному та альтернативному паливі з використанням комп'ютерної програми Дизель-РК.

Ключові слова:. *діагностика, двигун, індикатори, біопаливо, Д-240, індикаторні показники, техніко-економічні характеристики*

Ф. 2. Рис. 9. Табл. 4. Літ. 9.

СНЯТИЕ ТЕХНИКО-ЭКОНОМИЧЕСКИХ ПОКАЗАТЕЛЕЙ ДВИГАТЕЛЯ Д-240 ПРИ ИСПОЛЬЗОВАНИИ БИОТОПЛИВА ПУТЕМ ПРИМЕНЕНИЯ ПРОГРАММНОГО КОМПЛЕКСА ДИЗЕЛЬ-РК

Украина является одной из энергетически дефицитных стран, поскольку покрывает свои потребности в топливно-энергетических ресурсах за счет внутренних запасов всего на 53% (импортирует 75% необходимого природного газа и 85% сырой нефти и нефтепродуктов). В зависимости от импорта нефти большинство развитых стран рассматривают как вопросы национальной и энергетической безопасности, а использование нефтепродуктов как источников энергии создает значительный экологический риск.

Согласно анализу состояния мировой машиностроительной промышленности, наиболее эффективными мерами по усовершенствованию конструкции двигателя являются: разработка и внедрение нового рабочего процесса с эффективными методами смешивания и сгорания; разработка и создание конструкции системы автоматической регулировки вращения дизеля. частота для улучшения экономии топлива, а для снижения токсичности в частичной нагрузке некоторые цилиндры выключаются на холостом ходу.

Для усиления требований к нормативам расхода топлива и уровня токсичности отработавших газов, а также повышения надежности и эффективности сельскохозяйственных силовых двигателей необходимо сформулировать меры по усовершенствованию его конструкции.

Выбор топлива должен определяться оптимальным сочетанием экологических и экономичных показателей двигателя. Отмечаются перспективы использования того или иного топлива растительного происхождения. Если для топлива нефтяного происхождения улучшение его свойств определяется специальной обработкой, то масла с заданными характеристиками можно получить уже в полеве путем подбора соответствующих сортов растений, применения удобрений, агротехнических мероприятий и т.п.

Проблему снижения расхода дизельного топлива на холостом ходу и малой нагрузке можно решить, исключив из его работы часть цилиндров (этот метод широко используется) и замыкая цикл одного цилиндра.Проведен сравнительный анализ основных технических и экологических показателей дизельного двигателя Д-240 при работе на традиционном и альтернативном топливе с использованием компьютерной программы Дизель-РК.

Ключевые слова:. *диагностика, двигатель, индикаторы, биотопливо, Д-240, индикаторные показатели, технико-экономические характеристики*

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