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THE INDICATOR DIAGRAMS IN P-V COORDINATES CALCULATED AND MEASURED ON THE ENGINE STAND

Vasile Blaga [1], Mihai Blaga [2]*

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RESEARCH ARTICLE

ABSTRACT: This paper presents the numerical results of a simulation of the thermodynamic cycle for in spark ignition engine (SI) in normal external conditions (T_0=293 K; $p_0=10^5$ Pa) and revolution n=5500 rpm, at full charge. The simulation took in account the constraint for λ=1, for air pollution reduction whit combustible gases This paper presents the numerical results of a simulation of the thermodynamic cycle for in spark ignition engine (SI) in normal external conditions (T_0=293 K; $p_0=10^5$ Pa) and revolution n=5500 rpm, at full charge. The simulation took in account the constraint for λ=1, for air pollution reduction whit combustible gases [1]. The indicator diagram was calculated in p-V coordinates, and in p-α coordinates. Equations for thermal processes that take place in the engine were determined and then entered into a computer program. Using these equations, the state parameters were calculated at the characteristic points of the engine cycle. The research was conducted in order to obtain information that could be used improving the characteristics of an engine SI. The operating mode of the engine is defined by the speed of rotation and load, but it involves knowledge, in addition to these, of the thermal regime (parts temperature, cooling fluid temperature, intake air temperature, exhaust temperature, altitude corrections and etc.). The logic diagram of the calculation program is presented. The calculation program for determining the indicator diagram is done in Mathcad, both for the low pressure area and for the high pressure. The engine stand and the pressure transducer with their characteristics are presented. The diagrams are shown in p-V and p-φ coordinates. The p-V indicator diagrams are calculated by the presented program based on data measured on the engine stand.. The indicator diagram was calculated in p-V coordinates, and in p-α coordinates. Equations for thermal processes that take place in the engine were determined and then entered into a computer program. Using these equations, the state parameters were calculated at the characteristic points of the engine cycle. The research was conducted in order to obtain information that could be used improving the characteristics of an engine SI.

[1]*Vasile Blaga, University of Oradea, Mechanical Engineering and Automotive Department, 1 University Street, Bihor County, Romania, vblaga.ar@gmail.com,* ORCID ID: 0000-0002-6843-9832 *(*Corresponding author)*
[2]*Mihai Blaga, University of Oradea, Mechanical Engineering and Automotive Department, 1 University Street, Bihor County, Romania, mihaiblaga332@gmail.com,* ORCID ID: 0000-0003-1806-2691

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KEY WORDS: *thermodynamic cycle, indicator diagram, engine stand, characteristic points, the engine speed, low pressure, high pressure*

INDIKATORSKI DIJAGRAMI U P-V KOORDINATAMA ODREĐENI PRORAČUNOM I MERENJEM NA PROBNOM STOLU

REZIME: U ovom radu su prikazani numerički rezultati simulacije termodinamičkog ciklusa za benzinski motor u normalnim spoljašnjim uslovima (T_0=293 K; p_0=105 Pa) pri broju obrtaja n=5500 o/min i pri punom punjenju. U simulaciji je uzeto u obzir ograničenje za λ=1, kako bi se smanjilo zagađenje produktima sagorevanja [1]. Indikatorski dijagram je izračunat u p-V koordinatama, i u p-α koordinatama. Određene su jednačine za termičke procese koji se odvijaju u motoru, a zatim unete računski program. Koristeći ove jednačine, izračunati su parametri stanja u karakterističnim tačkama ciklusa motora. Istraživanje je sprovedeno u cilju dobijanja informacija koje bi mogle da se koriste za poboljšanje karakteristika motora SI. Režim rada motora je definisan brojem obrtaja i opterećenjem, ali podrazumeva poznavanje, pored ovih, i toplotnog režima (temperatura delova, temperatura rashladne tečnosti, temperatura usisnog vazduha, temperatura izduvnog gasa, korekcije nadmorske visine itd.) [1]. Prikazan je algoritam programa za proračun indikatorskog dijagrama. Programski paket je razvijen u Mathcad okruženju i namenjen je za određivanje indikatorskog dijagrama kako u području niskog pritiska, tako i u području visokog pritiska. Predstavljen je probni sto sa motorom i davačima pritiska odgovarajućih karakteristika. Proračunati dijagrami su prikazan u p-V i p-φ koordinatama. Dijagrami prikazani u p-V koordinatama se, računaju pomoću razvijenog programa i na bazi izmerenih veličinama na motoru određena su odstupanja proračuna i merenja.

KLJUČNE REČI: *termodinamički ciklus, indikatorski dijagram, optini motor, karakteristične tačke, broj obrtaja, nizak pritisak, visok pritisak*

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INTRODUCTION

1. THE MODEL CALCULATION FOR CYCLE THERMODYNAMIC DIAGRAM P – V

The mathematical equations of transformation that make up the thermodynamic cycle after which spark-ignition engines run are [2], [6]:

The polytrophic compression a-c:

$$p = p_a \cdot \left(\frac{V_a}{V}\right)^{n_c};\ \mathrm{V} \in [V_a; V_c] \tag{1}$$

$$V_a = V_c + V_s = \frac{V_s}{\varepsilon - 1} + V_s \tag{2}$$

The isochors burn c-z:

$$p = \alpha \cdot p_c\ ; V = V_c = ct. \tag{3}$$

The polytrophic relaxation z-u:

$$p = p_z \cdot \left(\frac{V_z}{V}\right)^{n_u};\ \mathrm{V} \in [V_c; V_u];\ \mathrm{V}_u = \delta \cdot V_c \tag{4}$$

The polytrophic relaxation u-d:

$$p = p_u \cdot \left(\frac{V_u}{V}\right)^{n_d};\ \mathrm{V} \in [V_u; V_d];\ \mathrm{V}_d = V_{d_1} = V_a \tag{5}$$

The freely evacuation d-d$_1$:

$$p = p_{d1}\ ; V = V_a \tag{6}$$

The forced evacuation d$_1$-r:

$$p = p_{d1}\ ;\ V = V_r = V_c \tag{7}$$

The adiabatic extend to evacuation r-r$_1$:

$$p = \left(\frac{V_r}{V_{r_1}}\right)^{k_e};\ V_{r_1} = V_r \cdot \left(\frac{p_r}{p_a}\right)^{\frac{1}{k_e}} \tag{8}$$

Admissible at constant pressure r$_1$-a:

$$p = p_a\ ; V = V_a \tag{9}$$

2. THE SCHEME OF THE FUEL SUPPLY SYSTEM

It is proposed to use the calculation model of the engine SI with gasoline injection, when modeling Renix multipoint electronic injection systems. In Figure 1. The fuel supply scheme of the Renix system used to model the system proposed by the author is presented.

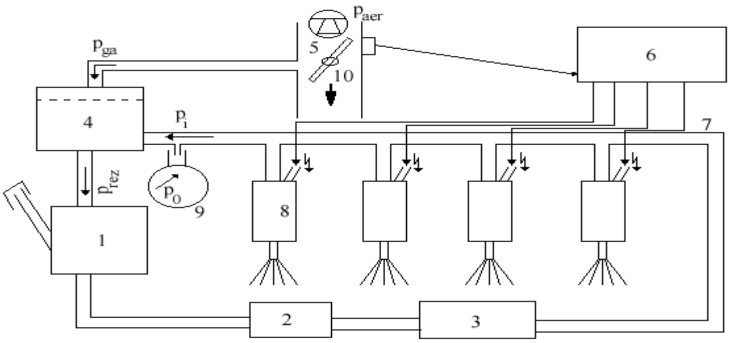

Figure 1 The fuel supply system: 1 – fuel tank; 2 – electric pump; 3 – fuel filter; 4 – pressure regulator; 5 – flowmeter to measure the intake air quantity; 6 – computer; 7 – injection manifold; 8 – electromagnetic injectors; 9 – valve to measure the pressure in the system; 10 – throttle; pair – the environmental pressure; peg – the pressure of the exhaust gases; pi – the injection pressure; ptank – the pressure in the fuel tank

The modelling of the SI engine cycle proposed by the authors is realized by running a computer program. The engine cycle proposed by the authors for the analytical calculus is given in figure 2.

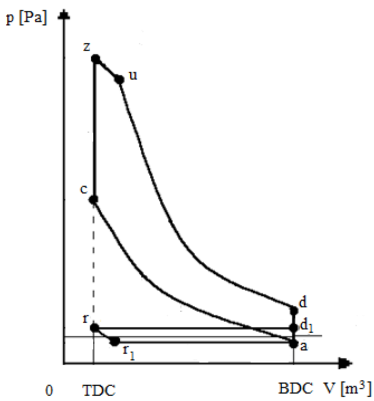

Figure 2 The engine cycle for the analytical calculus

The algorithm for calculating the working gas state parameters at the characteristic points of the engine cycle is presented: point (a) - the end of the intake stroke; point (c) - end of the compression process; point (z) - end of the isochoric combustion process; point (u) - end of the post-combustion process, point (d) - end of the expansion process; point (d1) - end of the free evacuation process; (r) - the end of the forced evacuation process.

The connection between the exhaust valve and the intake valve is made through the isentropic expansion ($r – r_1$). After the opening of the intake valve, the pressure drops from the value corresponding to the pressure of the residual burned gases (p_r) to the value ($p_{r1}=p_a$), equal to the intake pressure (considered to remain constant during the intake stroke). The pressure during the intake stroke (p_a) is lower than the environmental pressure (p_0) with a value (Δp_a), equal to the total pressure losses inside the intake manifold.

The initial data for the computation are the following: $D = 77 \cdot 10^{-3}$ m; – the cylinder bore; $S = 83{,}6 \cdot 10^{-3}$ m; the piston stroke; $V_s = 0{,}389 \cdot 10^{-3} m^3$; – the swept volume; R=290 J/kg – the working fluid constant; $Q_i = 44 \cdot 10^6$ J/kg; – the net calorific value; L_0=15 kg air/kg fuel – the stoichiometric air requirement; T_0=293 K – the environmental temperature; $p_0 = 1 \cdot 10^5$ Pa – the environmental pressure; λ=1 – the excess air factor; η_{ar}=0,9 – the burning process efficiency; ξ_0=0,8 – the heat release coefficient; ξ_{ga}=2 the gas-dynamic resistance factor of the intake manifold; ρ_0=1,177 kg/m^3 – the intake air density; $d_0 = 0{,}42 \cdot D$[m] – the inner diameter of the intake manifold at the valve port; n=500 … 5500 rot min (with a variation from 100 to 100 rot/min) – the crankshaft rotational speed; [m/s] – the mean piston speed; n_u=0,9 – the polytropic coefficient of the afterburning process; η_p=0,96 – the plenitude coefficient of the engine cycle; $p_r = 1{,}13 \cdot 10^5$ Pa – the residual exhaust gas pressure (the pressure in point r).

The logical scheme for the analytical calculation is presented in figure 4 [3] [4].

3. THE SIMULATE ON COMPUTER AND NUMERICAL RESULT

To can to describe a thermodynamics cycle that is give us for before equation (1 - 9), we must to know the value his nu, nc, nd, ke, α, δ. This size is estimate for normal conditions of temperature and pressure (To=293 K; po=102 kPa) and of revolution of n= 5500 rpm at λ=1, who show the medley character from gaze that are give to burn with pollution products minimum quantitative and qualitative [4].

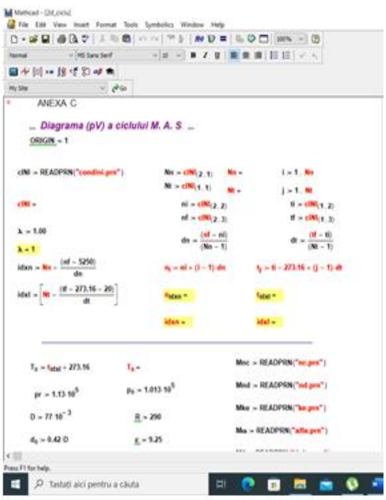

Figure 3 Part of the program with p-V diagram

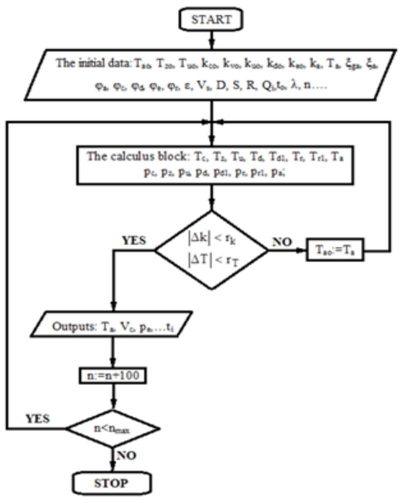

Figure 4 Part of the logic scheme

This the follow numerical for thermodynamics cycle for Engines with Spark Lightning in conditions than work above return it in table 1, figure 5 and 6.

***Table 1** The pressures and the volume in the characteristic points of the engine cyclepoints of the engine cycle*

The point	The pressure (10^5 Pa)	The volume (dm^3)
a	0,8023	0,43615
c	16,4727	0,04715
z	53,9902	0,04715
u	37,9328	0,06979
d	3,5918	0,43615
d_1	1,1300	0,43615
r	1,1300	0,04715
r_1	0,8023	0,06159

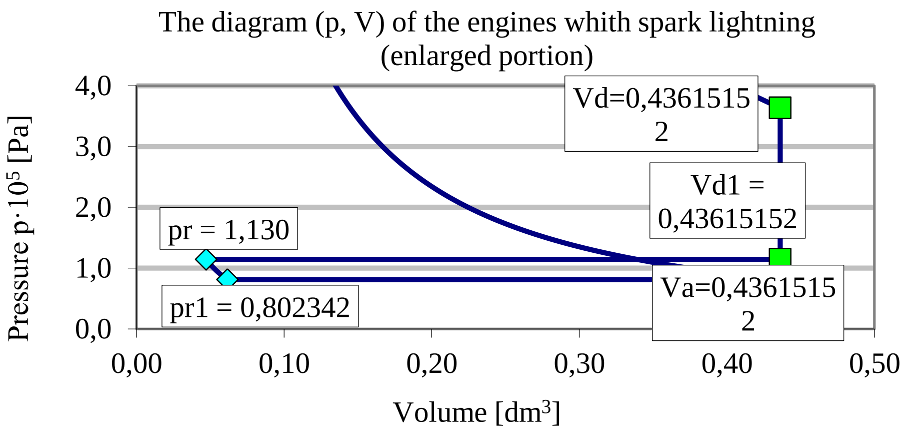

Figure 5 The diagram (p-V) to cycle Engines with Spark Lightning – Enlarged portion

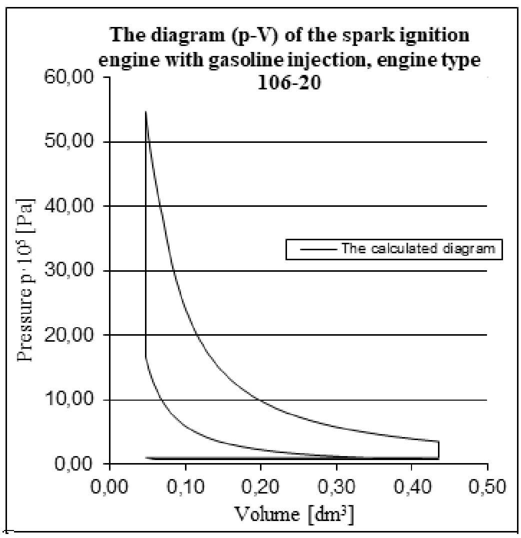

Figure 6 The diagram (p-V) to cycle Engines with Spark Lightning

4. THE MODEL CALCULATION FOR DIAGRAM P-φ

The isentropic expansion r-r1:

$$V_r = V_c \, ; V_{r1} - V_r = \frac{\pi \cdot D^2}{4} \cdot x_{r1} \text{, result: } x_{r1} = \frac{V_{r1} - V_r}{S_p}; \text{ where: } S_p = \frac{\pi \cdot D^2}{4}$$

$$\text{The equation of state shows: } p_r \cdot V_r^{k_e} = p_a \cdot V_{r_1}^{k_e}; \text{ result: } V_{r1} = V_r \cdot \left(\frac{p_r}{p_a}\right)^{\frac{1}{ke}} \tag{10}$$

It was noted $\Lambda = r/b$; r- crank length; b- connecting rod length. $\Lambda = 1/3{,}5$. The displacement of the piston is a periodic function of period 2π.

$$x_{r1} = S/2 \cdot [(1 - cos\varphi) + \Lambda/4\,(1 - \cos 2\varphi)];$$

$$\text{result: } 2x_1/S = (1 - cos\varphi) + (\Lambda/4) - (\Lambda/2) \cdot cos^2\varphi + (\Lambda/4) \tag{11}$$

$$C = (2x_1/S) - (\Lambda/2) - 1\, ; \; (\Lambda/2) \cdot cos^2\varphi + cos\varphi + C = 0$$

$$\text{result:} \cos\varphi\ \varepsilon\ [-1; 1] \quad \text{result } \varphi_{r1}$$

$$p = p_r \cdot \left(\frac{V_r}{V_{r1}}\right)^{k_e}; \quad x\varepsilon\ [0; x_{r1}]; \; \varphi\varepsilon\ [0; \varphi_{r1}] \tag{12}$$

The constant pressure inlet r1-a:

$$p = p_a\, ; x\varepsilon\ [x_{r1}\, ; S]; \; \varphi\varepsilon\ [\varphi_{r1};\ \pi] \tag{13}$$

The polytropic compression a-c:

$$pp_a V_a^{nc} = pV^{nc}; \quad p = p_a \left(\frac{V_a}{V}\right)^{nc}; \quad x \in [S; 0]; \quad \varphi \in [\pi; 2\pi] \tag{14}$$

V- The instant volume

$$V = V_c + (\pi D^2/4) x_{r1} \tag{15}$$

The isocor combustion c-z:

$$p_z = \alpha \cdot p_c\, ; \quad x = 0; \quad \varphi = 0 \tag{16}$$

The polytropic relaxation z-u:

$$V_u - V_c = \frac{\pi \cdot D^2}{4} \cdot x_u\, ; \quad x_u = \frac{V_c(\delta - 1)}{S_p}\, ; \quad S_p = \frac{\pi \cdot D^2}{4}\, ; V_u = \delta \cdot V_c \tag{17}$$

$$cos(\varphi_{1,2}) = \frac{-1 \pm \sqrt{1 - 2\Lambda C}}{\Lambda}\, ; \quad x_u = S/2[(1 - cos\varphi) + \Lambda/4(1 - cos2\varphi)];$$

$$\text{result: } 2x_u/S = (1 - cos\varphi) + \Lambda/4 - (\Lambda/2)cos^2\varphi + (\Lambda/4) \tag{18}$$

$$C_1 = (2x_u/S) - (\Lambda/2) - 1\, ; \; (\Lambda/2)cos^2\varphi + cos\varphi + C_1 = 0 \tag{19}$$

result: $\cos\varphi\varepsilon\ [-1; 1]$cosφ ε[-1;1]; result φ_u

$$p = p_z \cdot \left(\frac{V_z}{V}\right)^{nu}; \; x\varepsilon[0;\ x_u]; \quad \varphi\varepsilon[0; \varphi_u] \tag{20}$$

The polytropic destination u-d:

From the equation:

$$p_u \cdot V_u^{nd} = p \cdot V^{nd}$$

$$\text{Result: } p = p_u \cdot \left(\frac{V_u}{V}\right)^{nd};\ x \in [x_u; S];\ \varphi \in [\varphi_u; \pi] \tag{21}$$

The free evacuation d-d1:

$$pV_{d1} = V_a;\ \ p = p_{d1} \tag{22}$$

$$p_{d1} = \left(\frac{\varepsilon}{\delta}\right)^{nd} \cdot \delta^{nu};\ \ x = S;\ \ \varphi = \pi \tag{23}$$

The forced evacuation at constant pressure d1-r:

$$p_{d1} = p;\quad x\varepsilon[S;0];\quad \varphi\varepsilon\ [\pi; 2\pi]\ \ si\ \ p = p_r;\quad x = 0; \varphi = 0 \tag{24}$$

5. THE PROPOSED EXPERIMENTAL MODEL

The main task of the system is to establish the correlation between the engine's intake air mass and the injected fuel mass per cycle, forming a mixture of maximum economy for each engine operating regime. For electronic injection systems, the dependence between the amount of petrol injected per cycle at each engine operating mode and the injector opening time is determined in advance at the test stand, with manually operated control units, according to the criteria of the actual minimum specific fuel consumption, the maximum effective engine torque and the minimum pollutant emissions (carbon monoxide, hydrocarbons, nitrogen oxides), after which it is stored in the computer of the injection equipment, tabular or in the form of curves of variation of the opening time of the injector depending on the speed having as a variable parameter either the depression in the intake manifold or the position of the shutter damper, therefore the pressure transducer existing in these installations must be of special precision.

The adjustment of the amount of petrol injected per cycle for different engine operating modes is based on the amount of air drawn in by means of an air flow meter provided with a transducer which transmits information on the air flow to the computer. Replace the air flow meter with vane and transducer element with a Karman-Vartex type air flow meter based on ultrasonic waves, which improves engine performance. A study is made to choose the excess air coefficient λ; the pressure regulator is modeled; the pressure regulator constant is determined; the electromagnetic injector is modeled and the injection duration ratio is calculated.

Modeling the SI engine with gasoline injection proposed by the author is performed by running a computer program to determine the variation of three-dimensional and two-dimensional parameters in two cases: -dependence on engine speed and excess air coefficient at ambient temperature t_0 = -35… 45ºC and $p_0 = 1 \cdot 10^5$ Pa, in Annex A; [5] -dependence on engine speed and ambient temperature at λ = 1 and $p_0 = 1 \cdot 10^5$ Pa in Annex B. [5]; Logic of the program for modeling the SI engine with gasoline injection is presented in annex no. 37 [5]. It is proposed to use the model of the pressure regulator and the electromagnetic injector, the model proposed by the author, to determine the mass of fuel injected per cycle and the duration of the injection, with the engine speed and load. The three-dimensional cartographies were made on the computer according to the information provided by different translators and processed on the abscissa, the pressure in the intake manifold (throttle angle) and the orderly engine speed, the third dimension being the enrichment of the mixture. (injection time) for mapping. injection and ignition advance for the ignition mapping below. Renix multi-point electronic injection system that equips the adapted Dacia Logan Renault engines. The electronic unit is digital and combines two essential functions: ignition and petrol injection. Ignition points and injection time are calculated using the same information (engine speed and absolute pressure in the intake manifold), it makes sense to use a single computer to control the two functions. Among other things, ignition can benefit from additional corrections (water / air temperature) without the use of private translators. In this way, most transducers (speed, pressure, temperature) can be used together for 2 functions, which allows a high degree of reliability without cost compared to two separate computers.

The computer determines the ignition feed angle and the ignition coil contact time. These two pieces of information are emitted in the form of an electrical signal that is transmitted to an ignition power module (MPA), which conducts electricity to the primary coil to regulate the stored energy. A distributor distributes the spark energy to the cylinders. For injection, the central computer simultaneously controls the injectors (one per cylinder) at the same time as the crankshaft rotation, so that they supply each cylinder the same amount of gasoline depending on the amount of air drawn in. The amount of petrol injected depends on the absolute pressure in the intake manifold and the engine speed (crankshaft pressure-speed system). Ignition and injection are determined using information provided by different translators and processed by traversing two 3-dimensional maps, with abscissas (pressure), ordinates (speed), the third dimension is: enrichment of the mixture (or injection time) for injection mapping, figure 7; or the ignition advance for the ignition mapping, figure 8. For each mapping there are, for example, 9 inputs with regularly distributed pressure and 13 inputs with engine speed that does not need equal distribution. In total, therefore, 9 x 13 = 117 theoretical adjustment points are arranged. Writing (9-1) x (13-1) = 96 the elementary surface is made by computer with a double interpolation, with a weighting of each point a defined polygon. The theoretical values of the ignition advance and the duration of the injections are then modulated by corrective parameters: air pressure, air temperature, water temperature, battery voltage, acceleration, noise, etc.

The Renix system applies to multipoint injection or single point injection. It has a modular design, which allows easy and efficient adaptation to different types of 4 - or 6-cylinder engines. It has a self-diagnostic signal that facilitates control, adjustment, troubleshooting operations.

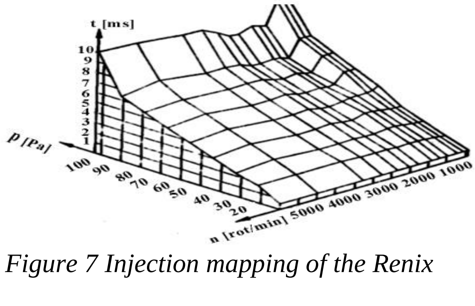

Figure 7 Injection mapping of the Renix

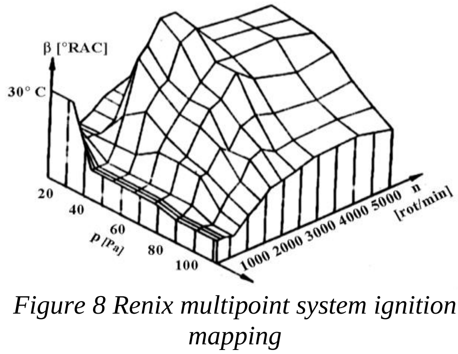

Figure 8 Renix multipoint system ignition mapping

Figure 9 shows the principle diagram of the Renix multipoint system that can be adapted to the Dacia Logan car.

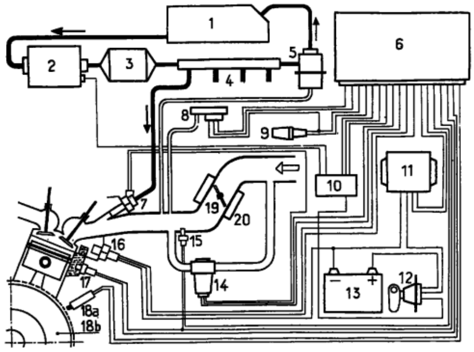

Figure 9 Principle diagram of the Renix multipoint system adapted on Dacia Logan: 1- fuel tank; 2- electric fuel pump; 3- fuel filter; 4- fuel distribution ramp at the injectors; 5- fuel pressure regulator; 6- computer; 7- injector; 8- absolute pressure transducer (air); 9- idle enrichment potentiometer; 10- control relay; 11- ignition power mode (MPI); 12- starting contactor; 13- battery; 14- electronic idle regulator; 15- air temperature transducer; 16- noise detector; 17- water temperature transducer; 18a- magnetic transducer (motor mode); 18b- solid sign of the flywheel; 19- flap; 20- box and flap contactor

They are proposed for determining the speed and the dead center of speed and angle reference transducers. They refer to the inductive transducer figure 10. [5] in which a magnetic field created by a permanent magnet 1 is partly wound on a steel core 2, is manifested in the air and in all neighboring media having a good magnetic conductivity. A gear wheel 3 with a steel pin 4 (reference mark) moving in front of the transducer changing the intensity and direction of the magnetic field. This variation of the magnetic field induces in the coil 5 an electric current of voltage U which is directed by the cable towards the electronic control unit.

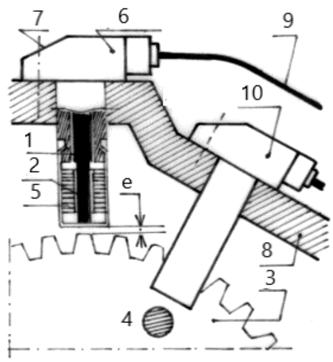

Figure 10 Speed and angular reference sensors: 1- permanent magnet; 2- steel core; 3- gear; 4- reference point; 5- coil; 6- speed sensor; 7- fixing device; 8- fixing support; 9- electric cable; 10- angular reference sensor; e - the interference between the crown and the sensor

The speed of the crankshaft is evaluated by an inductive speed transducer before which the teeth of the flywheel crown move. This transducer releases an output pulse on the tooth. The amplitude and shape of the electrical output signal transmitted by the computer depends on: the speed of rotation of the crown; the interference (e) between the crown and the sensor;

the profile of each tooth; the position (axial or radial) of the transducer in relation to the motor shaft; of the material in the sensor support.

The electronic control panel performs a signaling before being transmitted to the microcomputer. The angular position of the crankshaft is determined exactly by a position sensor or angular reference sensor which is also a pulse inductive sensor like the speed sensor. This sensor sends an output signal to the control panel with a pin or hole placed in front of it. There is an electrical output pulse for a crankshaft rotation. Without the crankshaft signal the engine does not start. For starting the minimum electric current is higher than 0.3 V. Resistance variation is 680 – 1200 Ω. When the speed signal is interrupted, the operation of the pump is cut off, the sensor performs the speed correction according to the speed. The sensor emits signals to synchronize the ignition with the injection. When oversaturated, it shuts off the injector.

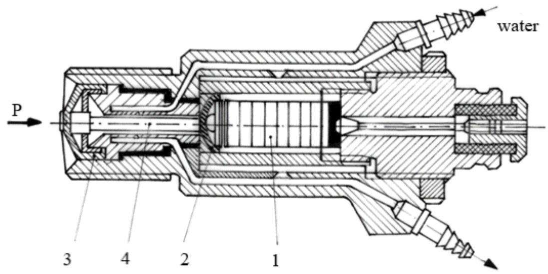

Figure 11 Piezoelectric pressure sensor

The diagram of the connection of the sensor in the measuring installation is shown in figure 12. The piezoelectric sensor imposes a typical structure of the adapter and auxiliary elements. The specific element of the adapter is the preamplifier PA, which has the role of adapting the high value output impedance of the sensor into one convenient for measurement and analysis. Obviously, this also amplifies the signal provided by the sensor. After preamplification, the signal passes through the I integrators, necessary to obtain signals proportional to the speed or displacement corresponding to the acceleration sensed by the sensor. The structure of the installation is completed with the module of the AM measuring amplifier, the block that provides the effective value BVE, and the filter block BF, which selects certain frequency ranges for the subsequent processing of the signal until the output.

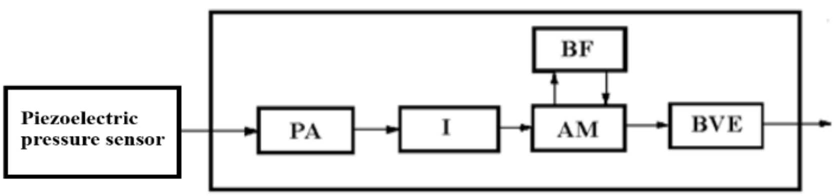

Figure 12 Sensor connection diagram in the measuring installation

6. THE REAL INDICATOR DIAGRAM

The general configuration of a test bed for the measurement of the parameters of a S.I. engine:

- The test bed on which the experimental data were acquired is provided with an electric machine with eddy currents of a W130 Shenk type.
- The setup of the test bed is presented in figure 13 [5].

Figure 13 The test bed used to obtain the indicator diagram

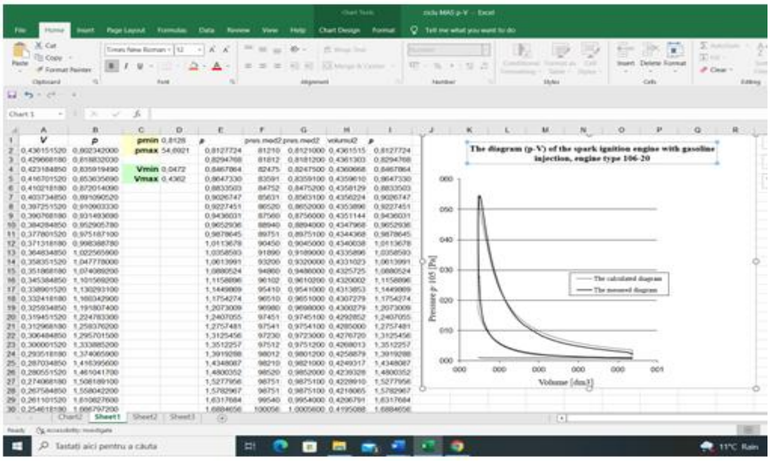

Figure 14 The capture with p-V cycle, the calculated and the measured chart the overlap

Specifications				
Measuring range			0...300 bar	
Overload			350 bar	
Lifetime	≥		10^8	load cycles
Sensitivity			19 pC/bar	nominal
Linearity	≤	±	0.3%	FSO
Natural frequency	~		170 kHz	
Acceleration sensitivity	≤		0.0005 bar/g	axial
Shock resistance	≥		2000 g	
Insulation resistance	≥		10^{13} Ω	at 20 °C
Capacitance			7.5 pF	
Operating temperature range			-40 ... 400 °C	
Thermal sensitivity change	≤		2 %	20 ... 400 °C
	≤	±	0.5 %	250 ± 100 °C
Load change drift			1.5 mbar/ms	max. gradient
Cyclic temperature drift *	≤	±	0.7 bar	
Thermo shock error **				
Δp	≤	±	0.4 bar	
Δp_{mi}	≤	±	2 %	
Δp_{max}	≤	±	1.5 %	
Thread diameter			M5x0.5	front sealed
Cable connection			M4x0.35	negative
Weight			2.2 grams	without cable
Mounting torque			1.5 Nm	

*) at 7 bar IMEP and 1300 rpm, diesel
**) at 9 bar IMEP and 1500 rpm, gasoline

Figure 15 The pressure sensor used for tests and its characteristics [5]

This numerical results for the cycle proposed by the author of Engines with Spark Lightning in corresponding conditions above it return in table 2 and figure 16.

Table 2 *The pressures and the rotation angle in the characteristic points of the engine cycle*

The point	The pressure (10^5 Pa)	The rotation angle [dgr]
r	1,13	0
r_1	0,8023	22
a	0,8023	180
c	16,4727	333
z	53,9902	365
u	37,9328	410
d	3,5918	537
d_1	1,13	540

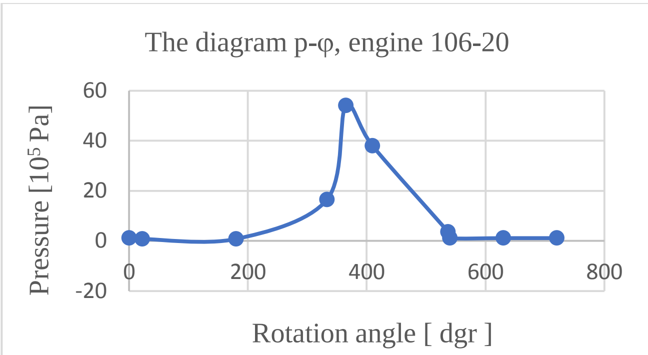

Figure 16 The diagram p-φ of cycle for Engines with Spark Lightning, whit gasoline injection, engine type 106-20

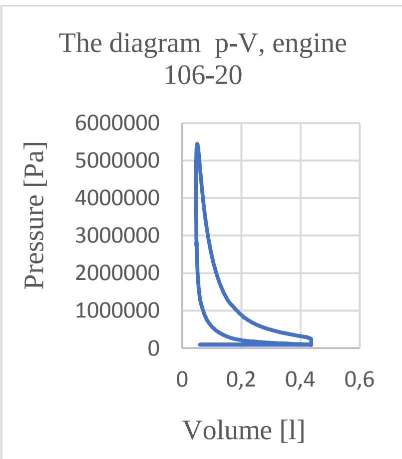

Figure 17 The measured diagram p-V

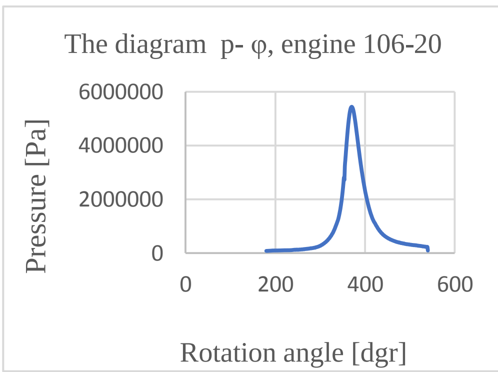

Figure 18 The measured diagram p-φ

7. CONCLUSIONS

The modelling of the SI engine cycle proposed by the authors consists in the presentation of the initial data of the computing program, the calculation and the correlation between the parameters of the engine for the realization of computing programs. The proposed model is a helping cycle to simulate the gasoline injection. The computer simulation permits the determination of the theoretical technical-economical parameters: the mechanical work corresponding to the rounded diagram, the indicated mean pressure, the indicated efficiency and the indicated specific fuel consumption. [1], [4]. After the calculation of the mechanical and the pumping losses, one can calculate the engine's effective technical-economical parameters. Using the model proposed in this paper, one can calculate the fuel mass injected per cycle, depending on the rotational speed at total load, the fuel flow through the electronic injector, reaching the purpose pursued, the determination of the injection duration depending on the rotational speed, at total load. It can be seen that the differences between the two data sets are in the range of 0.1-3.5%.

The calculated value of the pressure at the end of the intake process is 0.1% higher than the measured one. The pressure measured at the end of the compression stroke is 0.1% lower than the calculated one.

This small difference is due to the fact that the proposed model is almost similar to reality, and the pressure increase is higher than in the non-combustion cycle. However, the pressure difference at this point is small, because in this part (until the end of the compression stroke) only small amounts of fuel burn and, therefore, there is no significant amount of heat released.

Regarding the difference between the measured and the calculated value of the pressure at the end isochoric combustion, this is only 0.2%. However, the measured value is lower than the calculated one.

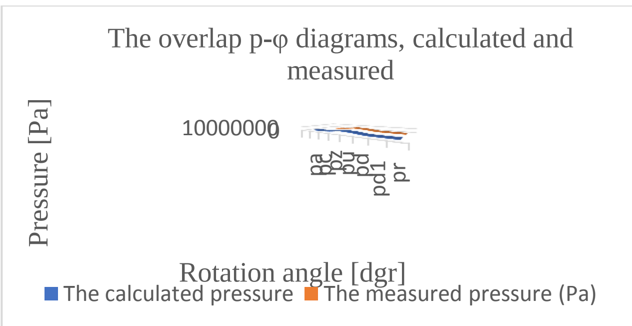

Figure 19 The overlap of the calculated and the measured indicator diagram p-φ

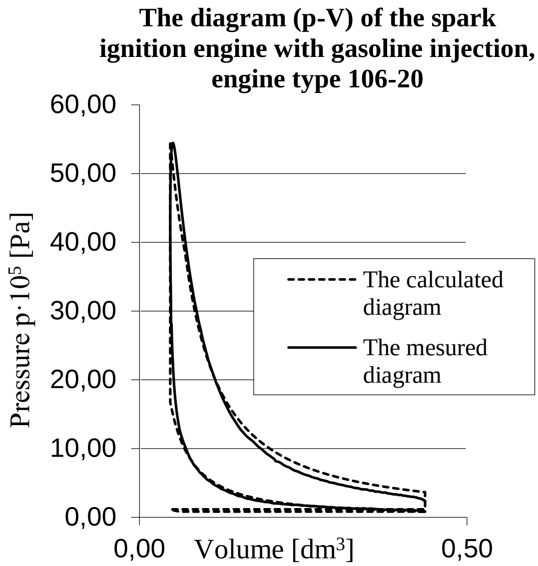

Figure 20 The overlap of the calculated and the measured indicator diagram p-V

Table 3 *The error betwenn the calculated and measured values in the characteristic points of the engine cycle*

The point	The calculated pressure [Pa]	The measured pressure [Pa]	The error [%]
a	81272	81210	0,1%
c	1668684	1667088	0,1%
z	5469207	5462756	0,2%
u	3842592	3841275	0,1%
d	363849	358818	1,4%
d1	114469	110466	3,5%
r	114469	110466	3,5%
r1	81272	81210	0.1%

At the end of post-combustion, the measurement is 0.1% lower than the calculated one. This can be explained by the fact that since the assumption that the combustion process begins in TDC, in theoretically the amount of fuel burned in the isochoric phase is less than the actual one. So, in the post-combustion, in the theoretical cycle, a smaller amount of fuel remains to be burned [4]. On the other hand, at the end of the expansion stroke, the measured value of the pressure is 3.5% lower than the calculated one, being the biggest difference, which is also visible on the overlapping diagrams. The difference in the exhaust pressure is also a maximum of 3.5%. This is probably the difference because the pressure is not measured in this region, but is estimated based on the minimum, maximum and average pressures at the characteristic points of the cycle. The values of the temperatures calculated at the characteristic points of the engine cycle were according to the statistical data. It was noticed that the calculated values are inside recommended intervals. If it is possible to develop a model for the study of ICE operation at low or medium load levels, it will be possible to determine the parameters that ensure a minimum fuel consumption and pollutant emissions at a certain operating regime. So, the model proposed in this paper, which proved to have a very good accuracy (and this can be improved) can be used to study the operation of SI motors in different conditions.

The program was used to study other normally aspirated SI gasoline injection engines.

The most conclusive gain for the Dacia Logan's petrol injection engine is the depollution of the engine. The pollutant emissions tests carried out in the Bosch laboratories at Schwieberdinger (under the control of UTAC-France representatives as the approval authority) have already confirmed that Dacia Logan complies with the depollution rules in force. The Renix multi-point injection system, which equips Renault engines with 4 cylinder Dacia cars, has been adapted to increase engine performance, reduce engine size, reduce fuel consumption, reduce fuel consumption and reduce pollution. Experienced petrol injection engines are superior to carburetor engines by reducing fuel consumption and pollutant emissions as well as dynamic construction and operating performance. The authors also intend to use this model for turbo engines.

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