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THE STUDY OF THE PROCESSES THAT TAKE PLACE IN GASOLINE INJECTION ENGINES

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RESEARCH ARTICLE

ABSTRACT: The fuel supply scheme used to model the system proposed by the authors is presented, the engine chosen, with the technical characteristics, the type of injection system adopted, the basic principle of electronic gasoline injection, the calculation cycle proposed for gasoline injection. Based on an own model, the authors realized an analytical calculation of the in diagram. The indicated diagram was also raised on an experimental stand in *p-V* coordinates. Using the equations from the characteristic points, the state parameters in the characteristic points of the engine cycle will be calculated. The diagram for the analytical calculation of the pressure in the intake manifold and of the intake pressure is presented; the variation of these parameters is represented depending on the speed and the ambient temperature. The logic diagram for the analytical calculation, determining the engine parameters are presented. The data obtained by calculations are compared with those obtained by measurements, and the results obtained show that the errors obtained are almost insignificant.

KEY WORDS: *total displacement, minimum cylinder volume, maximum cylinder volume, compression ratio, dosage, fuel mass, injection duration, indicated diagram.*

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PROUČAVANJE PROCESA KOJI SE ODVIJAJU U MOTORIMA SA UBRIZGAVANJEM BENZINA

REZIME: Prikazana je šema sistema za napajanje gorivom koja je korišćena za modeliranje sistema koji su predložili autori, izabrani motor, sa tehničkim karakteristikama, usvojenim tipom sistema ubrizgavanja, osnovnim principom elektronskog ubrizgavanja benzina, predloženim proračunskim ciklusom za ubrizgavanje benzina. Na osnovu samostalno rayvijenog modela, autori su realizovali analitički proračun indikatorskog dijagrama. Navedeni dijagram je određen na eksperimentalnom stolu u *p*-*V* koordinatama. Koristeći jednačine iz karakterističnih tačaka, izračunati su parametri stanja u karakterističnim tačkama ciklusa motora. Prikazan je dijagram za analitički proračun pritiska u usisnoj grani i usisnog pritiska; varijacija ovih parametara je predstavljena u zavisnosti od brzine i temperature okoline. Prikazan je algoritam za analitički proračun za određivanje parametara motora. Podaci dobijeni proračunom upoređeni su sa onima dobijenim merenjima, a dobijeni rezultati pokazuju da su dobijene greške gotovo beznačajne.

KLJUČNE REČI: *ukupna zapremina, minimalna zapremina cilindra, maksimalna zapremina cilindra, stepen kompresije, doziranje, masa goriva, trajanje ubrizgavanja, prikazani dijagram*

THE STUDY OF THE PROCESSES THAT TAKE PLACE IN GASOLINE INJECTION ENGINES

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INTRODUCTION

The mode of operation of the engine is defined by speed and load, but requires knowledge in addition to these and the thermal regime, ie parts temperature, coolant temperature, air temperature, exhaust temperature, altitude corrections, etc. [3]

It is most often preferred that the opening time of the electromagnetic injector be based on the depression in the intake manifold, as the amount of petrol injected per cycle is correlated with the amount of air drawn in per cycle, the speed dependence in this case being lower, and speed corrections will be made to operating modes that require such corrections.

Corrections to the fuel flow injected into the cylinder are required by a number of transient modes of engine operation, cold start, as well as coolant temperature, cylinder air temperature, lubricating oil temperature, atmospheric pressure. In view of all this, the basic principle expressed graphically in Figure 1 applies to the construction of the injection equipment.

A fuel pump extracts the fuel from the tank and discharges it to the electromagnetic injectors.

The gasoline pressure upstream of the injectors is kept constant by means of a pressure regulator which allows the return of excess gasoline discharged by the feed pump into the tank. [2]

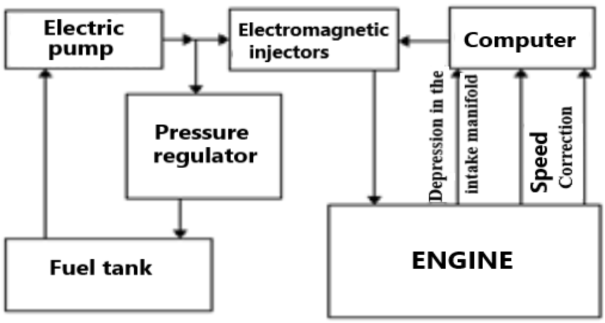

Figure 1. The basic principle of electronic fuel injection

The scheme of the fuel supply system used for the calculus is presented in figure 2.

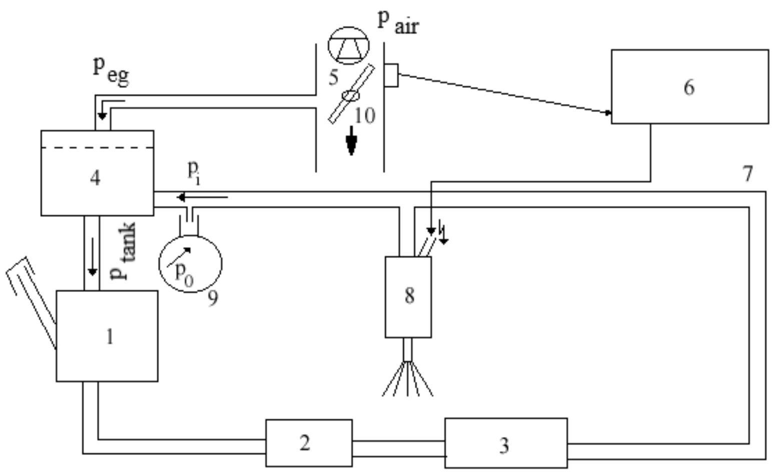

Figure 2. The fuel supply system: 1 – fuel tank; 2 – electric pump; 3 – fuel filter; 4 – pressure regulator; 5 – flow meter to measure the intake air quantity; 6 – computer; 7 – injection manifold; 8 – electromagnetic injectors; 9 – valve to measure the pressure in the system; 10 – throttle; pair – the environmental pressure; p_{eg} – the pressure of the exhaust gases; p_i – the injection pressure; p_{tank} – 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 presented in figure 3.[5]

The simplifying hypotheses that allow the definition of this cycle are the following:

- the intake process takes place at a constant pressure pa, permanently lower than the atmospheric pressure p_o with the value of the pressure losses Δp_a characteristic of the intake process. The intake process begins at point r_1 with the opening of the intake valve OIV and ends at the inner dead center TDC closing of the intake valve CIV;
- the d-d_1-r evacuation process takes place in two stages: after opening the exhaust valve OEV at BDC. the stage of free evacuation d-d_1 at constant volume of the gas takes place during which the pressure decreases from p_d to p_r higher than the atmospheric pressure with the value of the pressure losses Δp_r characteristic of the exhaust route. The evacuation at constant pressure p_r step $d_{1\text{-}r}$ follows, which ends with the closing of the exhaust valve CEV;
- the connection between the exhaust valve and the intake valve is made by means of the isentropic expansion r-r_1 of the waste gas.
- the a-c compression process of the fresh mixture is assimilated with a constant polytropic exponent $n_c < k_c$ during which the gas yields to the walls of the cylinder the heat H_{pc};
- the combustion process is schematized in two evolutions: the isochoric evolution c-z in which the heat input is H_v and the polytrophic evolution of exponent n_u <1 which defines the post-firing and in which the agent receives the heat H_u;

- the expansion process begins with the theoretical completion of the combustion at point u being assimilated with a polytrophic of exponent $n_{d1} > k_d$ which leads to the release of heat H_{pd} to the cylinder walls;
- the thermal agent is considered to behave as a perfect gas with specific temperature-dependent heat.

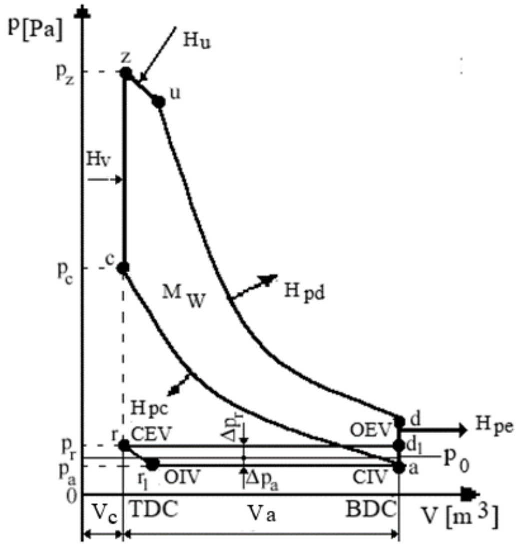

Figure 3. The engine cycle for the analytical calculus [1].

It is emphasized that the assimilation of the post-burning process with a non-subunit exponent polythrope is based on experimental data [5] which show that the final burning temperature T_u is higher than the final burning temperature T_z of constant volume burning c-z, due to an intense release heat on the first z-u portion of the *z-u-d* process of lowering the gas pressure in the cylinder.

By changing the exponent, the influence of the heat dissipation rate during post-firing cannot be highlighted, on the one hand, and on the other hand, experimental data can be used on the value of the polytropic exponent n_d which characterizes the general expansion of *z-u-d* gases. In other words, by combining the post-combustion *z-u* of the exponent $n_u < 1$ and the actual expansion *u-d* of the exponent $n_{d1} > k_d$, even the actual expansion *z-d* characterized as is known by a polytropic exponent $1 < n_d < k_d$ is obtained.

The main notations used at the modelling are: $\alpha = p_z/p_c$ – the pressure raise ratio in the isochoric phase of the burning process; $\delta = V_u/V_z$ – the volume raise ratio in the afterburning process; $\varepsilon_1 = V_u/V_z$ – the volume raise ratio during the expansion process; $\psi_0 = \Delta p_a/p_0$ – the relative pressure drop coefficient during the intake process; $\psi_1 = \Delta p_r/p_r$ – the relative pressure drop coefficient during the exhaust process; $\psi = p_a/p_r$ – the global coefficient of the pressure losses; $\varphi_r = T_{d1}/T_r$ – the ratio between the temperature at the end of the forced exhaust and the temperature at the end of the free exhaust.

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·D [m] – the inner diameter of the intake manifold at the valve port; n=500 … 5500 min^{-1} (with a variation from 100 to 100 min^{-1}) – 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=1,13·10^5 Pa – the residual exhaust gas pressure (the pressure in point *r*).

Below is presented the algorithm for the calculation of the state parameters of the working gas in the characteristic points of the engine cycle: point (*a*) – the end of the intake stroke; point (*c*) – the end of the compression process; point (*z*) – the end of the isochoric burning process; point (*u*) – the end of the afterburning process, point (*d*) – the end of the expansion process; point (d_1) – the end of the free exhaust process.

The logical scheme for the analytical calculation is presented in figure 8.[5]

The computation program is structured on 10 procedures and functions. In addition to the values of the constants declared at the beginning of the program, one also considers as initial data, assumed known arbitrarily chosen from statistic data of the SI. engine cycle: T_{a0}=322 K, T_{z0}=2530 K, T_{u0}=2660 K, k_{c0}=1,3, k_{v0}=1,3, k_{u0}=1,2, k_{d0}=1,3 and k_{e0}=1,3. Without these, it is not possible to calculate, generally speaking, all the other parameters that characterize the engine cycle. Thus, this initial data will play the role of parameters, the variables being explicitly the temperature between these thermal processes evolve. These temperatures are closely depending on the adiabatic coefficients which, in fact, are stabilized by the completion for several times of the engine cycle until these coefficients become constant, with an error of 0,000009. The decision for the exit from the cycle for a particular rotational speed is given by the decreasing of the constant error of the intake temperature (T_a) under the value 1,5 K. [5]

1. THE CALCULATION OF THE PRESSURE IN THE INTAKE MANIFOLD PGA AND THE INLET PRESSURE PA

Considering the intake process, the case when the engine fluid density is variable applies the Bernoulli gas flow equation, written for the inlet section 0-0 and section 2-2 (figure 4). [1]

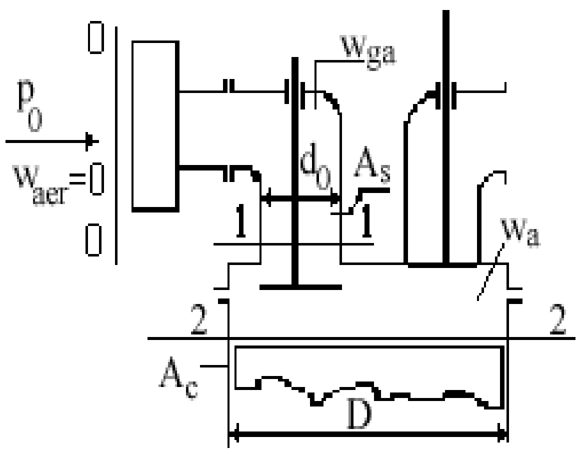

Figure 4. The sketch for the analytical calculation of p_{ga} and p_a

It is written Bernoulli's relation between sections 0-0 and 1-1 to determine the pressure in the intake manifold pga, and then between sections 0-0 and 2-2, in which case determine the inlet pressure p_a:

The pressure at the intake valve port [3] and [4]:

$$p_{ga} = p_0 - k_1\left(1+\xi_{ga}\right)n^2\left(\frac{V_s}{d_0^2}\right)^2$$

$$k_1 = \frac{1}{1+1+\xi_{ga}\dfrac{k_a}{2}\left(\dfrac{4}{30\pi}\right)^2\left(\dfrac{V_s^2}{D^2}\right)^2\left(\dfrac{n}{a_{sa}}\right)^2}, \qquad (1)$$

where:

- a_{sa} [m^2] – the effective flow area through the orifice controlled by the intake valve;
- k_a=1,4 – the adiabatic exponent of the intake process.

The pressure at the end of the intake stroke [1] and [4]:

$$p_a = p_{ga} - k_1 n^2\frac{V_s}{d_0^4}\left[\left(1+\xi_{ga}\right)\left(\frac{d_0}{D}\right)^4 - 1\right], \qquad (2)$$

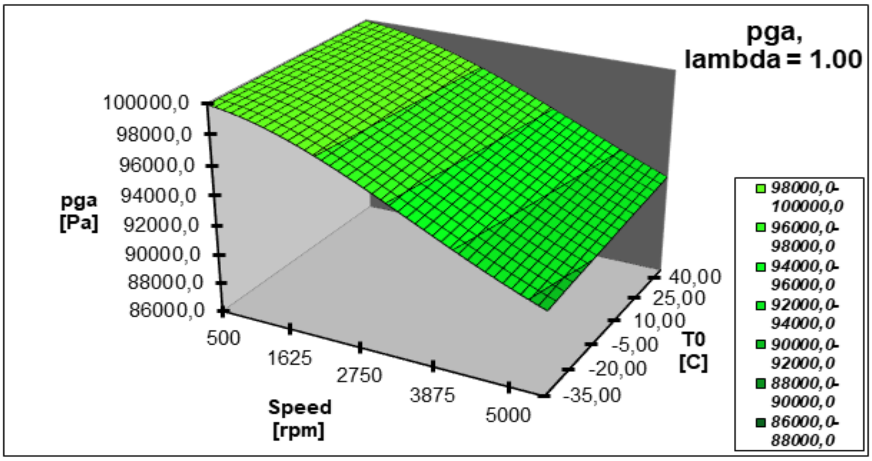

Figure 5. The variation of pressure in the intake manifold with speed and temperature

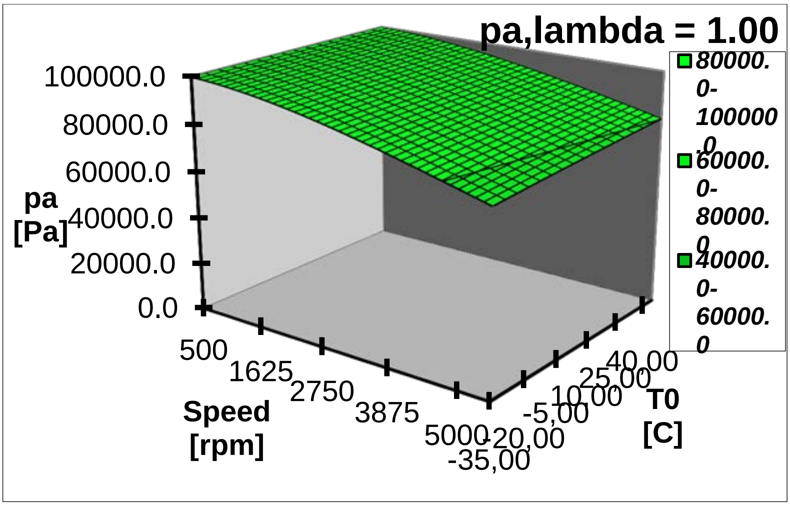

Figure 6. The variation of inlet pressure with speed and ambient temperature

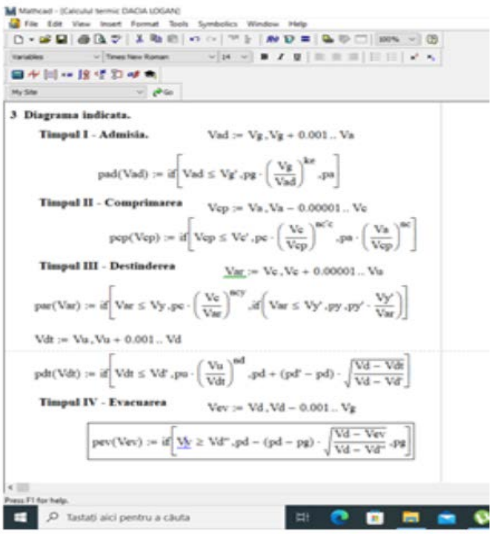

Figure 7. Part of the calculation program

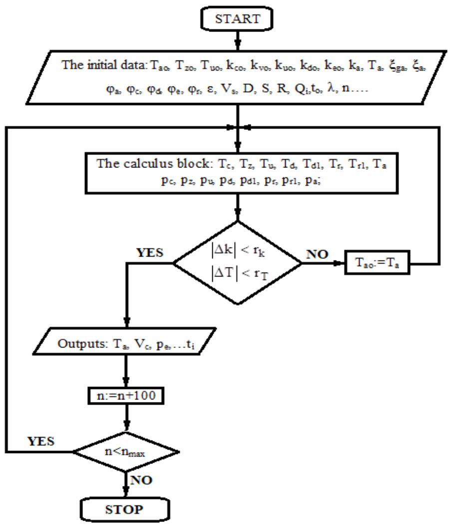

Figure 8. The logical scheme for the calculus

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

The mathematics equation for transformation what make the thermodynamics cycle after who run Engines with spark Lightning study are: [7];[8]

2.1 The polytrophic compression a-c:

$$p = p_a \left(\frac{V_a}{V} \right)^2 ; V \in [V_a ; V_c];$$
$$V_\mathrm{a} = V_\mathrm{c} + V_\mathrm{s} = \frac{V_s}{\varepsilon - 1} + V_s; \tag{3}$$

2.2. The isochors burn c-z:

$$p = \alpha p_c ; V = V_c = ct; \tag{5}$$

2.3. The polytrophic relaxation z-u:

$$p = p_z \left(\frac{V_z}{V} \right)^{n_u} ; V \in \left[V_c ; V_u \right] ; V_u = \delta V_c ; \tag{6}$$

2.4 The polytrophic relaxation u-d:

$$p = p_u \left(\frac{V_u}{V} \right)^{n_d} ; V \in \left[V_u ; V_d \right] ; V_{d1} = V_a , \tag{7}$$

2.5 The freely evacuation d-d₁:

$$p = p_{d1} ; V = V_a ; , \tag{8}$$

2.6 The forced evacuation d₁-r:

$$p = p_{d1} ; V = V_r = V_c ; , \tag{9}$$

2.7 The adiabatic extend to evacuation r-r₁:

$$p = \left(\frac{V_r}{V_{r1}} \right)^{k_e} ; V_{\mathrm{r}_1} = V_\mathrm{r} \left(\frac{p_\mathrm{r}}{p_a} \right)^{\frac{1}{k_e}} ; \tag{10}$$

2.8 Admissible at constant pressure r1-a:

$$p = p_a ; V = V_a ; , \tag{11}$$

Table 1. Representation of the entire the spark ignition engine cycle:

The point	pressure [10^5 Pa]	Volume [dm^3]
a	0,8127299	0,43615
c	16,6868451	0,04715
z	54,6920726	0,04715
u	38,4259264	0,06979
d	3,6384934	0,43615
d1	1,14469	0,43615
r	1,14469	0,04715
r1	0,8127299	0,06159

nciclu := n1 + n2 + n3 + n4 + n5 + n6 + n7 + n8

ic := 1 .. niclu

nciclu = 230

$p_{i1} := p1_{i1}$

$V_{i1} := V1_{i1}$

$p_{n1 + i2} := p2_{i2}$

$V_{n1 + i2} := V2_{i2}$

$p_{n1 + n2 + i3} := p3_{i3}$

$V_{n1 + n2 + i3} := V3_{i3}$

$p_{n1 + n2 + n3 + i4} := p4_{i4}$

$V_{n1 + n2 + n3 + i4} := V4_{i4}$

$p_{n1 + n2 + n3 + n4 + i5} := p5_{i5}$

$V_{n1 + n2 + n3 + n4 + i5} := V5_{i5}$

$p_{n1 + n2 + n3 + n4 + n5 + i6} := p6_{i6}$

$V_{n1 + n2 + n3 + n4 + n5 + i6} := V6_{i6}$

$p_{n1 + n2 + n3 + n4 + n5 + n6 + i7} := p7_{i7}$

$V_{n1 + n2 + n3 + n4 + n5 + n6 + i7} := V7_{i7}$

$p_{n1 + n2 + n3 + n4 + n5 + n6 + n7 + i8} := p8_{i8}$

$V_{n1 + n2 + n3 + n4 + n5 + n6 + n7 + i8} := V8_{i8}$

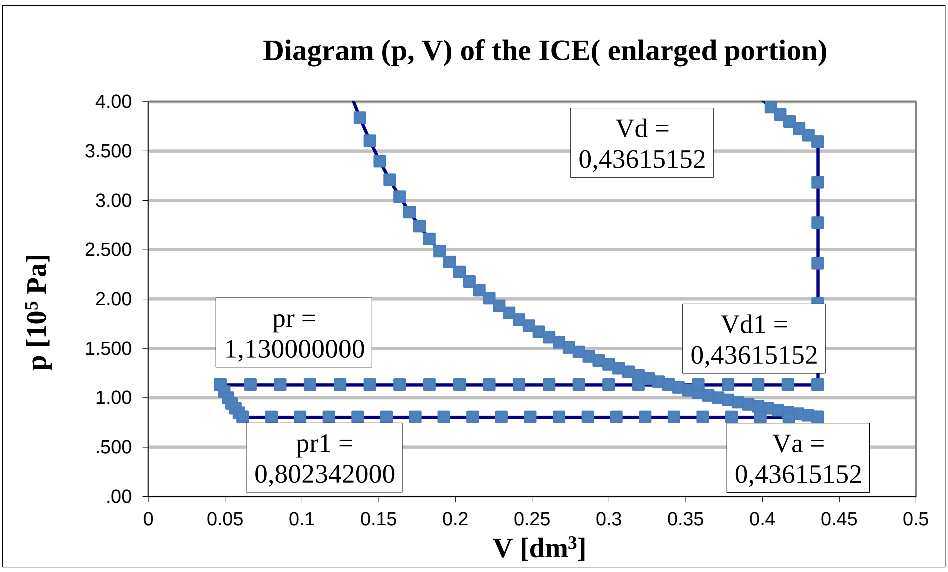

Figure 9. The gas exchange diagram

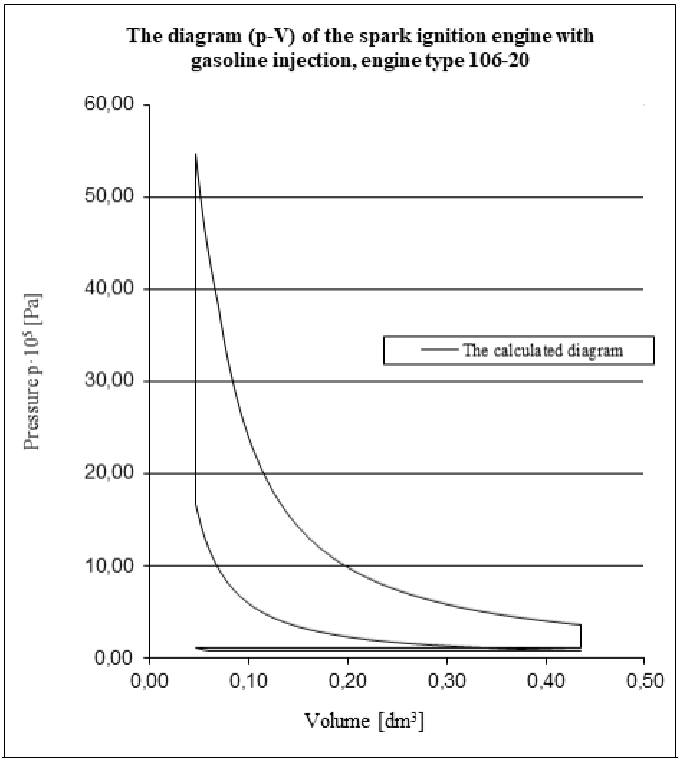

Figure 10. The calculated indicated diagram.

The computation program was realized in Mathcad and part of it is shown in figure 9.

The calculated indicated diagram, obtained with this computer program, is presented in figure 10.

3. THE REAL INDICATED 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. [9]

The data acquisition system sums 32 measurement lines to measure the pressure, the temperature, the rotational speed, the torque, the environmental parameters and so. The data acquisition and processing are automatically made by using a processes computer that is provided with a set of programs necessary for the execution of the set of especially programs to emit the commands necessary to execute the probs included in the testing procedures and for the automatic processing of the data recorded during the testing.

The transducers and the experimental devices are of:

- a resistive type – their verification was made with testers to verify the resistance depending on the cooling liquid temperature and on the intake air temperature;
- an inductive type, in which a magnetic field created by a permanent magnet has a good magnetic conductibility.

The variation of the magnetic field induces in the electromagnetic coil an electric current of voltage (V), that is directed through a cable to the electronic command unit. The setup of the test bed is presented in figure 11.

Figure 11. The test bed used to obtain the indicated diagram.

The pressure inside the cylinder was measured using a pressure sensor for combustion analysis AVL GH 14 DK. Its characteristics are presented in figure 12.

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 12. The pressure sensor used for tests and its characteristics

4. TEST SCHEDULE

The object Engine type: 106-20

Features: 1557 cm3 tranverse plasament motors ε= 9,25 supply/ignition with Bosch Mono-Motronic M.A. 1.7

The purpose:

Contributions to gasoline injection molding on 106-20 type engine. Definition of injection timing and ignition advance maps and correction charts function to ensure the dryveability of cars.[5]

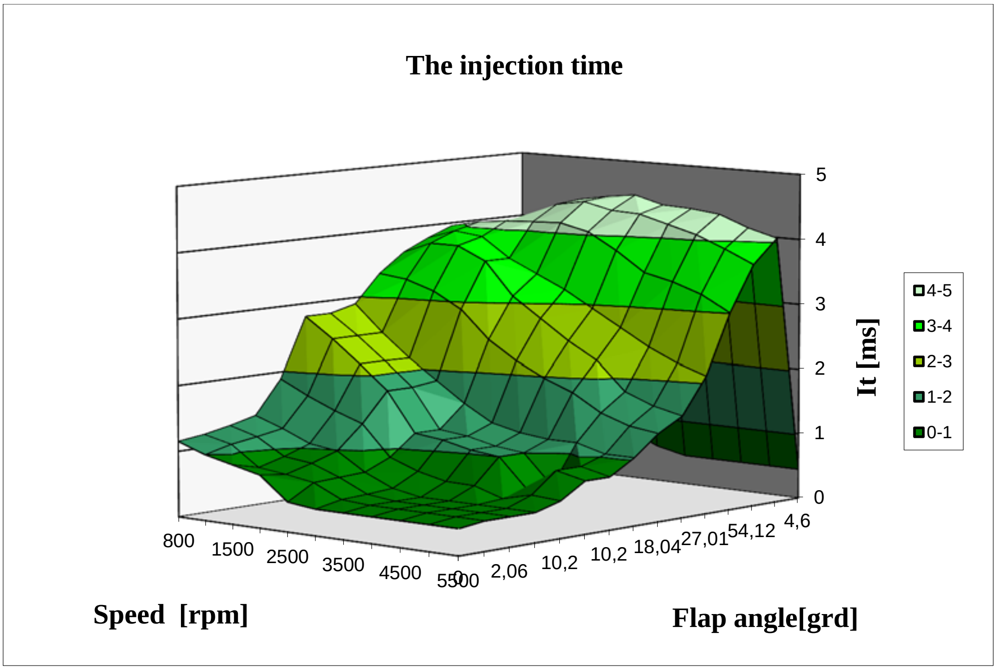

Figure 13. The injection duration cartography

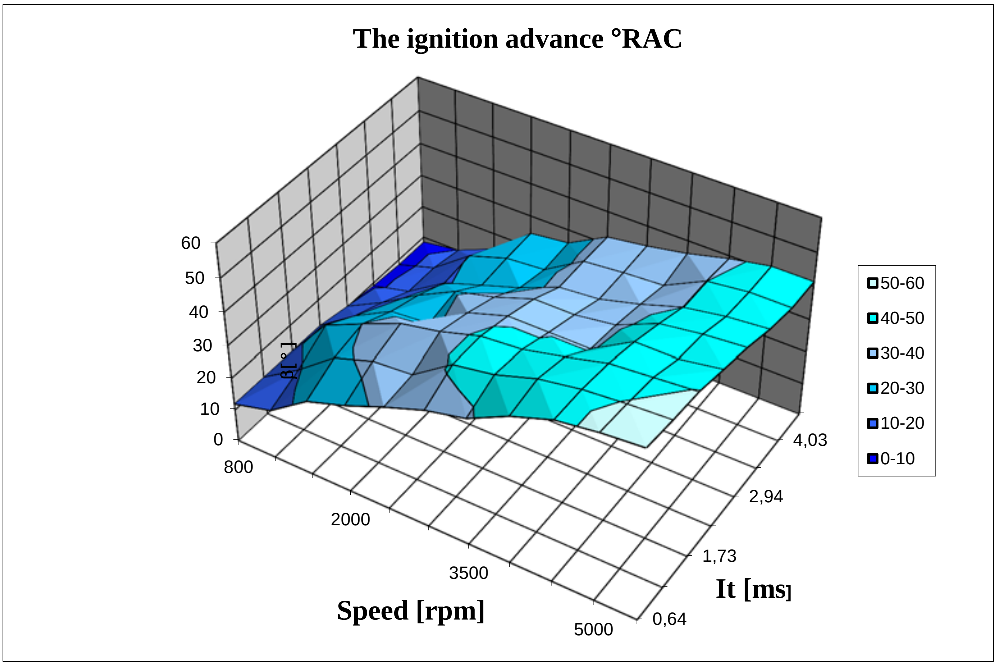

Figure 14. The ignition advance cartography

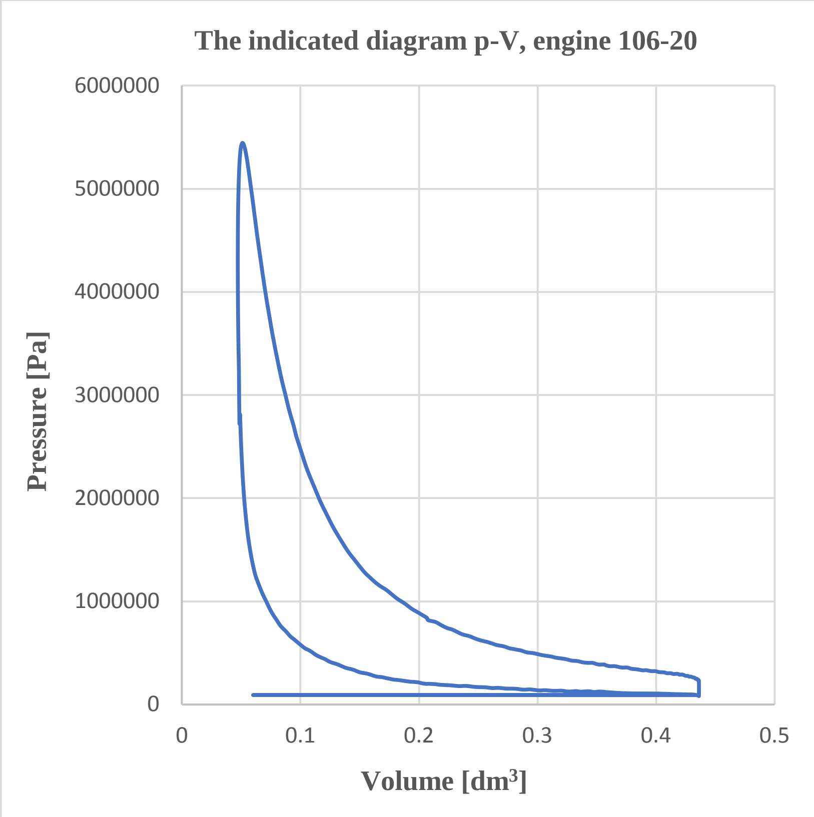

Figure 15. The real indicated diagram

5. CONCLUSIONS

The model for gasoline injection system was proposed, the author used the Bosch Mono-Motronic M.A.1.7 system, adapted to the engine of the Dacia Logan car. The study was made to choose the optimum excess air coefficient for the operation of gasoline injection engines. [5]

The calculation of the intake manifold pressure and the intake pressure in two cases was calculated; when the density of the motor fluid is constant and the case closer to reality when the density of the motor fluid is variable. In the program for calculating the parameters of the SI engine were the relationships in the second case. The three-dimensional variations of the pressure in the intake manifold pga and the inlet pressure pa were represented according to the engine speed and the ambient temperature, at the value of the excess air coefficient $\lambda = 1$ and po = 1 • 102 kPa, resulting from the program in the annex B. With the program in Annex A, the three-dimensional variations of the pressure in the intake manifold pga and the inlet pressure pa can be determined depending on the engine speed and the coefficient of excess air at ambient temperature to = -35ºC… 45ºC and po = 1 • 102 kPa. Temperatures in the range –35ºC and 45ºC, are denoted by 5 in 5ºC with To1, To2… To17. [5]

The modelling the SI engine with gasoline injection proposed by the author consists in presenting the initial data of the calculate on program; calculating and correlating the expressions of the parameters of the SI engine with the injection of gasoline for the implementation of the calculation programs in Annexes A and B. The calculation cycle of the proposed gasoline injection SI engine is an auxiliary cycle for the computer simulation of the gasoline injection. [5]

The computer simulation allows the determination of the proposed theoretical technical-economic parameters: the proposed theoretical mechanical work corresponding to the rounded diagram, the proposed theoretical pressure, the proposed theoretical efficiency, the proposed specific theoretical consumption. The pressures of mechanical and pumping losses are calculated, with the help of which the effective theoretical technical-economic parameters of the engine are calculated. [5]

By changing the conditions of the environment, the intake process is affected, there are changes in the combustion process, because the state conditions of the initial mixture change. As a result, the volumetric efficiency, the excess air factor, the indicated and efficient efficiency, ie all the factors that decide the level of power and specific fuel consumption will suffer deviations from their optimal values. In order to determine the optimal values for measurements, the atmospheric and altitude conditions under which these measurements are made and the net calorific value of the fuel must be taken into account.

Table 2. The comparison between the calculated and the real pressures in the characteristic points of the engine cycle.

The point	The calculated pressure [Pa]	The measured pressure [Pa]
a	81272	81200
c	1668684	1667000
z	5469207	5462000
u	3842592	3841000
d	363849	363000
d_1	114469	111300
r	114469	111400
r_1	81272	81200

It turns out that the differences between the two data sets are in the interval 0-2,68 %. The calculated value of the pressure at the end of the intake process is 0.09% higher than measured. The calculated pressure at the end of the compression stroke is 0.11% higher than measured. 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, no significant amount of heat is released.

The difference between the calculated and the measured value of the pressure at the end of the isochoric combustion, this is only 0.14%. However, the calculated value is higher than the measured value. At the end of post-combustion, the calculated pressure is 0.05% higher than the measured one.

These small differences explain the fact that in the engine during combustion there are other phenomena that could not be included in the calculation program, the amount of fuel burned in the isochoric phase is higher than the real one. In the post-combustion, in the theoretical cycle, a smaller amount of fuel remains to be burned. At the end of the expansion stroke, the calculated value of the pressure is higher than that measured by 0.24%.

The biggest difference is in the case of the exhaust pressure (2.68%). However, the calculated value is greater than the measured value. The values of the temperatures calculated in the characteristic points of the engine cycle were compared with those in the statistical data. It has been observed that the calculated values fall within the recommended ranges. For a better comparison between the calculated and the experimental data, the two diagrams were superimposed in Figure 16. [5]

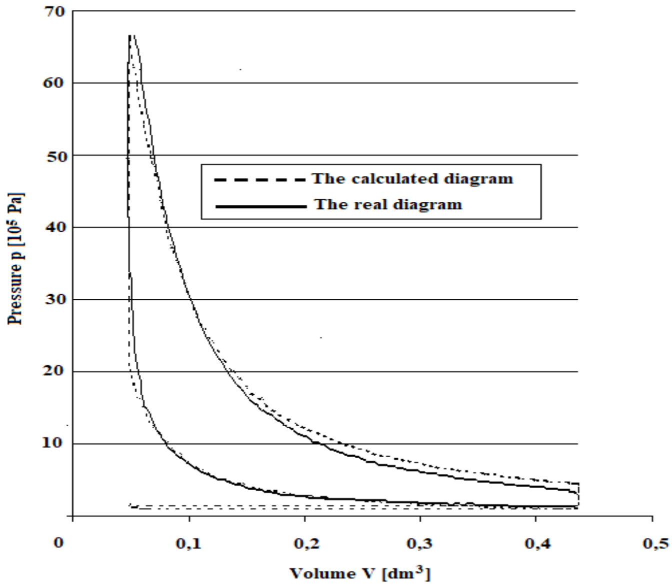

Figure 16. The overlap of the calculated and the measured indicated diagram.

Figure 16 shows that, at the end of the compression stroke, the difference between the calculated and the measured values is quite small (maximum 0.11%) and the area between the two curves is very small.

As mentioned earlier, this difference is due to the fact that the engine operating process is very complex and is influenced by many factors, which can influence the thermal processes in the engines.

A relatively large difference (maximum 2.5%) is also recorded at the beginning of the expansion race, corresponding to the delay in the theoretical cycle. This difference can be explained by the fact that the estimated amount of fuel burned in the actual cycle is higher than the theoretical one calculated are smaller than the real ones. There are some differences between the calculated and measured pressures in the second part of the expansion stroke, where the calculated values are higher than the measured ones (around 2.6%). The error can be reduced by changing the exponent of the expansion process.

The practical application of this calculus involves, on one hand, the improving of the physical-mathematical model, so that it will be as close as possible to the real development of the gasoline injection process. This means the reduction of the theoretical assumption and the calibration of some of the computation parameters, based on the experimental values. A theoretical model to compute with sufficient precision the thermal processes that take place in a SI engine can be useful for the development of new propulsion systems for road vehicles.

Many think that the transition to the full electric vehicles is a hybrid propulsion system. A variant is the inline hybrid propulsion system. In this case, the thermal engine should provide the energy to power the traction electric motor. The main advantage is the fact that the thermal engine can function at a single regime. This means that the adjustments that are necessary for the functioning at different regimes are no longer necessary. [10]

If it is possible the development of a model for the study of the ICE functioning at low or medium loads, it will be possible to determine the parameters that ensure minimum fuel consumption and polluting emissions at a specific functioning regime.

The model proposed in this paper, which proved to have a very good precision (and that can be improved) can be used to study the functioning of SI engines in different conditions.

The program was used for the study of other normally aspirated SI engines with gasoline injection.

The authors intend to use this model for turbocharged engines also.[10]

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