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THE CALCULUS OF THE FUEL CONSUMPTION AND OF THE INJECTION DURATION FOR A SPARK IGNITION ENGINE

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Abstract**—** In this paper is presented the calculus, designed by the authors, of the fuel supply scheme for a multipoint injection system. The calculated outputs are the fuel consumption and the injection duration depending on the rotational speed and on the environmental temperature. A computer program was developed to determine the parameters of a SI engine in two cases: 1. the calculation of the injection parameters depending on the rotational speed (n) and on the air-fuel ratio (λ) for environmental temperatures t_0=-35...+45 ^{0}C and environmental pressures $p_0=1\cdot10^5$ Pa; 2. the calculation of the injection parameters depending on the rotational speed (n) and on the environmental temperature t_0 for a air-fuel ratio $\lambda=1$ and environmental pressures $p_0=1\cdot10^5$ Pa.

Keywords**—** electric pump, electronic injection, flow meter, air-fuel ratio

I. INTRODUCTION

THE paper presents a mathematical model to calculate the parameters of the electronic injection for SI engine for $\lambda=1$. The computational program developed by the authors determines the variable parameters of the SI engines with electronic injection. The calculated data were compared with the experimental ones.

The mathematical algorithm includes the equations that describe the phenomena occurring in the electric fuel supply pump, in the pressure regulator and in the injector.

The model was developed based on a Bosch Motronic in port fuel injection system. A data acquisition plate was used. The temperature, the pressure, the rotational speed and the crankshaft offset were determined using electric sensors. To compare the calculated data with the experimental ones it was necessary, based on the acquisitioned data, to draw the engine's indicated diagram. The mathematical model proposed by the authors can be used to improve the dynamic parameters of the vehicle and to reduce fuel consumption and pollutant emissions.

II. COMPARATIVE STUDY BETWEEN FUEL SUPPLY SYSTEMS WITH CARBURETOR AND GASOLINE INJECTION SYSTEMS

Over time, the constructive form of the carburetors was improved. Dual body and four body carburetors were built. In order to improve the air-fuel mixture, multiple carburetors were also used, one for each cylinder or for a group of cylinders. For easier fuel vaporization, at the start of the engine, the fresh charge is heated by the exhaust gases (Fig. 1a). This device is placed at the exit of the carburetor and has an adjustment flap commanded by a rheostat. When reaching the operating temperature, the rheostat commands the closure of the loop circuit of the exhaust gases and the heating of the fresh charge stops (Fig. 1b).

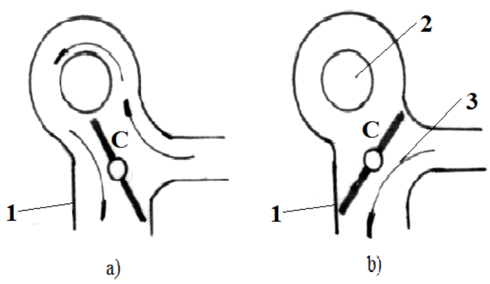

Fig.1 The mixture heating scheme: 1 – the exhaust pipe; 2 – the intake pipe; 3 – exhaust gases; C the adjustment flap

The introduction of gasoline injection for SI engines aimed mainly to improve power and fuel consumption performances, performances capped by the way the fuel-air mixture is formed and the less volumetric efficiency of the engines with carburetor. Subsequently, the restriction regarding the environmental pollution of the SI engines exhaust gases imposed a new development direction for the gasoline injection systems, that is to reduce pollutant emissions [1].

The carburetors with electronic command have considerable advantages compared with the classic ones such as: simplified mechanical construction, reduced fuel consumption, the reduction of pollutant emission, automated driving system and stable operation at idle. The most improved carburetors provide a homogenous fuel-air mixture accurately dosed for all operating regimes. Appreciable accelerations are obtained and the fuel consumption is acceptable. Meantime, the carburetors adapted for high loads operate with difficulty at lower loads.

The main advantages of the gasoline injection are the following: reduces the inconvenient of the carburetor for all operating regimes, increases the power and reduces the fuel consumption and the polluting emissions.

The advantages of the gasoline injection, compared with carburetor, are: fine gasoline spraying for all operating regimes, greater uniformity of the injected mass especially when there's an injector for each cylinder, the raise of the volumetric efficiency, a precise fuel-air ratio, the increasing of the effective engine's power with 10-15%, a decrease of the fuel consumption with 12-15% and the reduction of the pollutant emissions.

III. THE DEVELOPMENT OF THE CALCULUS ALGORITHM

The Bosch-Motronic is a highly complex in port fuel injection system. It combines the electronic injection with the electronic ignition. The command and the control of the fuel mass injected in one engine cycle and the start of the injection for all operating regimes are performed by a computer.

The theme of this paper is a part of a program developed for the calculus of the parameters of a SI engine with gasoline injection. With the aid of this program one can raise also the indicated diagram of the engine.

The general program was developed based on a simplified engine cycle proposed by the authors. The program calculates the heat input of the cylinder's unit volume (q_{cb}) and the volumetric efficiency (η_V). It is determined the pressure increasing ratio during the constant volume burning phase (α) and the equation of the volume increasing ratio during the afterburning (δ). The program includes the calculus of: the mean adiabatic exponent of the intake process (k_a), the mean adiabatic exponent of the compression process (k_c), the mean adiabatic exponent of the heat input during the fast burning phase (k_V), the mean adiabatic exponent of the heat input during the afterburning (k_u) the mean adiabatic exponent of the expansion process, (k_d), the mean adiabatic exponent of the exhaust process (k_a), the polytropic exponent of the intake process (n_a), the polytropic exponent of the compression process (n_c), the polytropic exponent of the expansion process (n_d) and the polytropic exponent of the exhaust process (n_a).

A sensitive issue in terms of computer programming for the relations of the adiabatic exponents is that the initial values of the temperatures in the characteristic points of the indicated diagram are not known. So, these initial values were chosen based on a preliminary calculus [2].

The temperature at the end of the intake process was considered known as an initial data, and the calculated expression was used as verification equation at the end of the computing cycle. All the expressions used for developing the program are correlated with each other in order to be introduced in the algorithm. These relations are correlated only depending on the temperatures in the characteristic points of the engine cycle. The other parameters in these points were substituted with the values from the initial data.

By recalculating the adiabatic exponents with the values of the determined temperatures the engine cycle was browsed again. So, after some computing cycles the error can be reduced under a value appreciated as been admissible. Another simplifying assumption is that the composition of the gases inside the cylinder instantly changes into the final one (the burning is instantaneous), corresponding to the air-fuel ratio at which the fuel burning takes place.

The program also calculates the effective mean pressure (p_e), the effective efficiency (η_e), the power (P_e), the torque (M_e), the fuel mass injected in an engine cycle (m_{cb}) and the specific fuel consumption (c_t). And finally, the indicated diagram of the proposed engine cycle was raised.

The calculus of the fuel mass injected in an engine cycle and of the injection duration was developed for a Mono-Motronic injector with electronic command.

The calculus for the electromagnetic injector was developed based on the scheme presented in Fig.2.

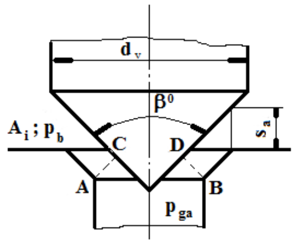

Fig. 2 The scheme for the calculus of the fuel flow through an electromagnetic injector

The fuel mass injected during an engine cycle is proportionally to the injection pressure and the injection duration [3], [4].

The flow section of the spray orifice is determined using the following equation [3]:

$$A_i = \pi \cdot s_a \cdot \sin\left(\frac{\beta}{2}\right) \cdot \left(\frac{d_V - 1}{s_a \cdot \sin\beta}\right); \tag{1}$$

where: A_i (m^2) – the instantaneous area of the flow section;

d_v (m) – the needle's diameter in the tip zone;
β (^{0}) –the seat angle;
s_a (m) – the needle's stroke (it is considered to be constant;

The volumetric flow that crosses through the injector is calculated using the formula:

$$Q_b = \mu_i \cdot A_i \cdot \sqrt{\frac{2 \cdot (p_b - p_{ga})}{\rho_b}} \tag{2}$$

where: Q_b (m^3/s) – the volumetric flow;
μ_i – the mean flow coefficient of the flow area; μ_i=0,8 … 0,93;
p_b (Pa) – the gasoline pressure at the entrance in the injector;
p_{ga} (Pa) – the fresh charge pressure in the valve port;

$$p_b = k_r + p_{ga} \tag{3}$$

k_r=1 .. 4 – the regulator's constant;
and, it results:

$$Q_b = \mu_i \cdot A_i \sqrt{\frac{2 \cdot K_r}{\rho_b}} \tag{4}$$

Using the definition of the volumetric flow, results:

$$Q_b = 10^3 \cdot \frac{m_b}{\rho_b \cdot \tau_i} \tag{5}$$

where: V_b (dm^3) – the volume of fuel that passes through the injector in one engine cycle;
τ_i (ms) – the duration of the injection;
t_o (^{0}C) – the environmental temperature;
m_b (kg) – the mass of fuel that passes through the injector;
ρ_b (kg/dm^3) – the fuel's density;

The fuel mass injected in one engine cycle is:

$$m_{cb} = 10^3 \cdot \xi \cdot (1/\lambda) \cdot \eta_V \cdot \rho_0 \cdot V_s \tag{6}$$

where: m_{cb} (g/cycle) – the fuel mass injected in a cycle;
ξ – the ratio between the intake air mass and the intake fresh charge mass;
ρ_0 (kg/dm^3) – the fresh charge density;
V_s (dm^3) – the swept volume;

The variation of the fuel mass injected in one cycle depending on the rotational speed (n) and on the environmental temperature (t_0) for a fuel-air ratio λ=1 is presented in Fig. 3.

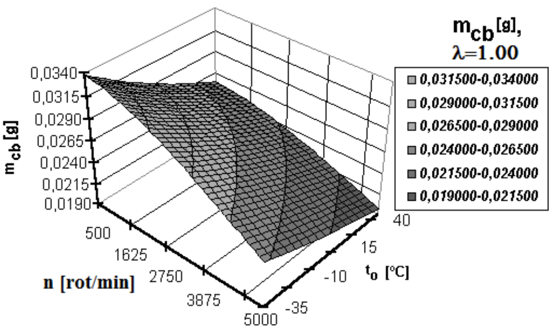

Fig. 3 The variation of the fuel mass injected in one cycle depending on the rotational speed (n) and the environmental temperature (t_0)

From (4) and (5) results that the duration of the injection is:

$$\tau_i = \frac{m_{cb}}{\mu_i \cdot A_i \cdot \sqrt{2 \cdot K_r \cdot \rho_b}} \tag{7}$$

The variation of the injection duration depending on the rotational speed and on the environmental temperature, for an air-fuel ratio λ=1, is presented in Fig. 4.

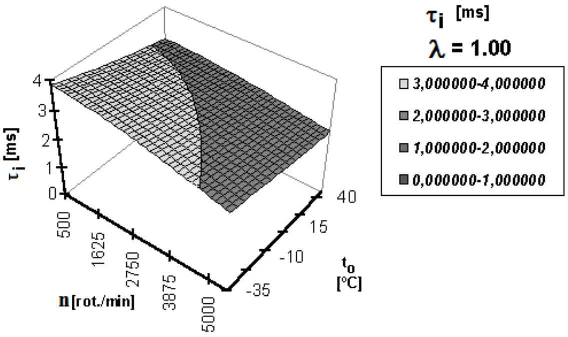

Fig. 4 The variation of the injection duration depending on the rotational speed (n) and the environmental temperature (t_0)

IV. Comparison Between Experimental and Calculated Data

The experimental data were obtained on a test bed that includes an eddy currents electrical brake.

The comparison between experimental and calculated data is presented in Fig. 5.

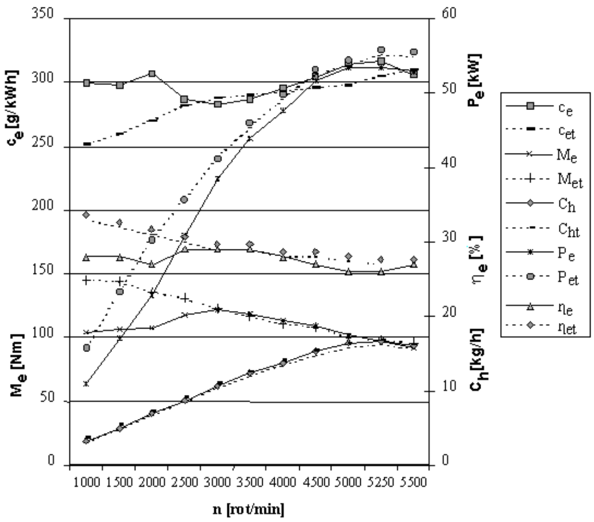

Fig. 5 Comparison between some experimental and calculated parameters

The meaning of the parameters shown in Fig. 5 is:

c_e, c_{et} (g/kWh) – the experimental and the calculated specific fuel consumption;

M_e, M_{et} (Nm) – the experimental and the calculated torque;

C_h, C_{ht} (kg/h) – experimental and calculated hourly fuel consumption;

P_e, P_{et} (kW) – experimental and calculated power;

η_e, η_{et} – experimental and calculated engine effective efficiency;

As seen in Fig. 5, the errors between experimental and calculated data are acceptable.

In this paper only the fuel consumption will be analyzed.

One can observe that for the hourly fuel consumption there are practically no differences.

The calculus must be adjusted for the specific fuel consumption. For rotational speeds under n=2300 rot/min the difference between the experimental and the calculated data are around 20%, but in rest the differences are no bigger than 10%.

The main reason why these differences occur is the fact that in the mathematical algorithm the flow coefficient (μ_i) was considered to be constant. In reality it's variable because also the area of the flow section is also variable. For small flow section the difference between the mean value and the instant one of the flow coefficient can be 50-70%.

In Fig. 6 is presented the comparison between experimental and calculated data for the injection duration.

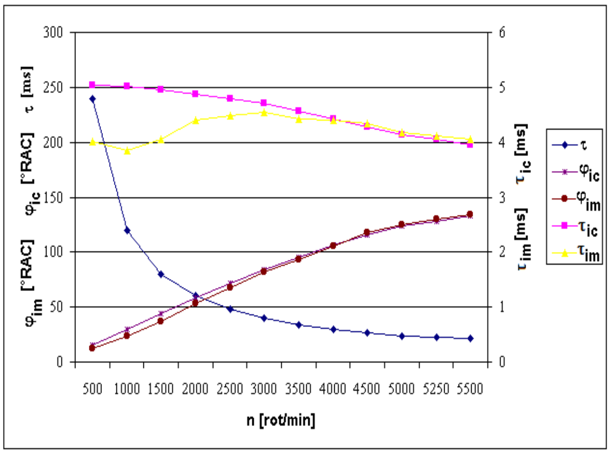

Fig. 6 Comparison between experimental and calculated data for the injection duration

The meaning of the parameters shown in Fig. 6 is:

τ (ms) – the duration of the engine cycle;

φ_{im}, φ_{ic} – the start of the injection (expressed in crankshaft offset - ^{0}RAC);

τ_{im}, τ_{ic} (ms) – the injection duration;

Regarding, the injection duration one can observe the similarity with the curves representing the specific fuel consumption in Fig. 5.

This is normal because the duration of the injection is proportional to the injected fuel quantity. So, the calculated injection duration for rotational speeds under 3500 rot/min are smaller than one measured.

V. Conclusion

As was stated above, the mathematical algorithm presented in this paper is only a part of a larger one, developed for the calculus of the SI engines with gasoline injection.

Besides the calculation of the engine cycle, some computational models were developed for the elements of the injection system, for the fuel supply electrical pump, and for the gasoline pressure regulator.

The model proposed for the determination of the injection duration as function of the rotational speed and of the load proposed in this paper is an original one.

Using the general computation program, an analysis was conducted on the influence of the air-fuel ratio on the burning process at different environmental temperatures.

It was determined the theoretical optimal start of the injection at different rotational speeds.

After some slight adjustments, that include calibration with experimental data, the proposed model can be used for future research activities such as:

1) *the determination of mechanical losses due to internal frictions between the moving parts of the crank gear's parts, especially at low rotational speeds [5];*
2) *the influence of the air-fuel mixture quality on the burning process and on the engine's parameters;*
3) *the reduction of fuel consumption and polluting emissions;*
4) *the study of the fuel flow through the injector in order to determine with more accuracy the flow coefficient (μ_i);*

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