UNIVERSITY OF PITESTI SCIENTIFIC BULLETIN
FACULTY OF MECHANICS AND TECHNOLOGY AUTOMOTIVE series, year XVII, No.21 (2)

CALCULATION MODEL FOR OPTIMIZING THE SPARK IGNITION ENGINE COLLABORATION WITH THE DEPOLLUTING SYSTEMS

Daniel LIȚĂ*; Florian IVAN; Ion TABACU; Andrei BUŞOI
University of Pitesti, Romania

Abstract:*It is known that creating an economical and non-polluting engine within the EURO 5-6 standards supposes obligatorily complex research of working and optimization in general. In this category a great importance is given to optimization research of Spark Ignition Engine (S.I.E) collaboration with the depolluting systems, especially with the T.W.C. The aim of this optimization concerns the decreasing of the concentration of chemical emissions from the exhaust gases within a long-standing components of the depolluting system. Shortening the time of the optimization research represents a desideratum that cannot be reached without the development of some validated calculation models on engine testing at the stand in a variety of operational regimes. The paper presents such a model and the experimental results using for validation 12 conditions of load and rotation.*

Keywords: genesis depollution, S.I.E., T.W.C., Euro 5-6

INTRODUCTION

The aim of this pattern is to enable a correct estimate of the air mass admitted in the engine. In theory, at a given engine rotation, the filling volumetric efficiency , η_v, varies linearly according to the pressure in the exhaust main.
For each line, one estimates the collector pressure which gives a null air feed. This pressure has different values according to the rotation and it is called Zero Pressure (P0). At the same time, zero pressure can be defined as the related pressure of the gasdynamic losses for the whole intake section.
The theoretical filling principle is not respected in reality /1/ and /8/, because there are small differences between the real filling curve and the line that renders the theoretical filling (figure 1). For the determination of the filling volumetric coefficient one uses interdependence relations measuring the excess air coefficient, λ, with the help of the gas analyzer and the fuel consumption using the fuel balance at the engine stand. This is how the relation develops:

$$\eta_v = k \cdot \frac{C_h \cdot M_{as} \cdot \lambda}{V_t \cdot n} \qquad (1)$$

where: η_v – is the filling volumetric coefficient; k - constant = 28514.4; V_t - whole swept volume $\left[cm^3\right]$; n - engine rotation [rot/min]; C_h - hourly fuel consumption [kg/h]; M_{as} - stoichiometric fuel-air mass [kg]; λ - excess air coefficient.

Also, figure 1 shows the way of establishing the zero pressure as a result of the linear extrapolation for the values of the volumetric coefficient of filling and of the exhaust main pressure found at the engine stand. For establishing the useful pressure responsable for the engine filling one establishes the relation:

$$p_u = p_{ca} - p_0 \qquad (2)$$

where: p_{ca} – is the pressure in the exhaust main; p_0 - the appropriate pressure of the gasdynamic losses for the whole intake section.

* Corresponding author.

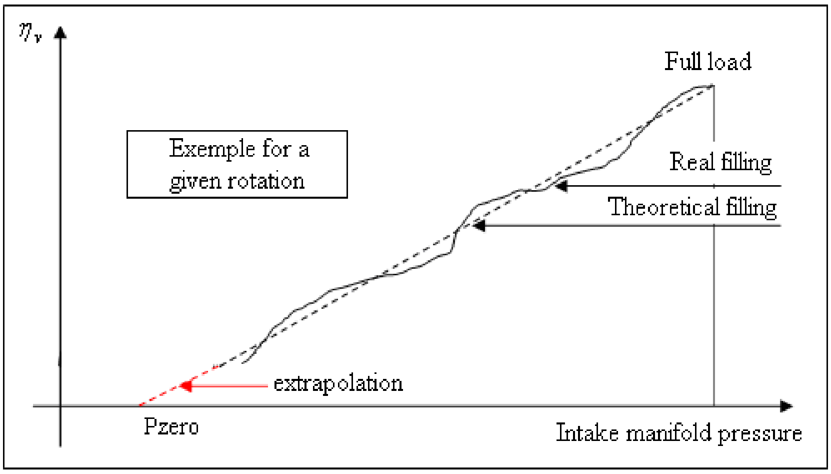

Figure 1. Highlighting the real filling curve and theoretical filling line

The air quantity that enters the cylinder is calculated with the perfect gas equation:

$$p \cdot V = m \cdot R \cdot T \Rightarrow m = \frac{p \cdot V}{R \cdot T} \tag{3}$$

where: p – is the useful pressure in the exhaust main [Pa]; V – cylinder volume [m^3]; T – air temperature in the exhaust main [K]; R - air ideal constant= 287,68 [J/kg·K]; m – the real air mass let in the cylinder which participates in the burning [mg].

Of course, the equation (3) is not reproducing perfectly the phenomenon of intaking the air in the engine, the flow is impermanent, turbulent, and influenced by thermogasdynamic losses. This losses are mainly caused by the architecture and the dimensions of the intake section. Therfore, the equation may be corrected with a β coefficient analytically found depending on rotation, pressure in the exhaust main and the engine building characteristics.

So, the equation (3) becomes:

$$m_{ar} = \frac{\beta \cdot p \cdot V}{R \cdot T} \cdot 10^6 \tag{4}$$

Having the air mass admitted to the cylinder, the volumetric filling coefficient can be calculated using the formula:

$$\eta_v = \frac{m_{ar}}{m_{at}} \tag{5}$$

where: η_v – is the volumetric filling coefficient [-]; m_{ar} - the real air mass admitted to the cylinder [mg]; m_{at} - the theoretical air mass which can fill the cylinder volume [mg].

To find the β coefficient one starts from estimating the pressure in the intake gallery, p_{ga}, relation that includes the global coefficient of resistance of the intake route, ς_a, which takes into account the losses due to friction, through the sudden variation of the section and through the change of the flow direction /2/. After determinating the loss coefficient for the whole intake section one can draw the map of the engine in question.

$$p_{ga} = p_{atm} - 0,5 \cdot 10^{-5} \cdot (1+\varsigma_a) \cdot \rho_{fp} \cdot \overline{W}_{ga} \tag{6}$$

The medium flow speed through the intake gallery, $\overline{W}_{ga}$, is determinated from the equation of the flow which passes through the free orifice:

$$\overline{W}_{ga} = (D^2 / d_a^2) \cdot \eta_v \cdot \overline{W}_p \cdot 180 / \Delta_{\alpha a} \tag{7}$$

(7)

where: p_{atm} - is the air pressure [bar]; ρ_{fp} - the fresh fluid density [kg/m^3]; D - the cylinder bore [cm]; d_a - the diameter of the intake system [cm]; η_v - the filling volumetric coefficient [-]; $\bar{W}_p$ - the piston's medium speed [m/s]; $\Delta_{\alpha a}$ - the intake valve opening time [ºRAC].
The relations (6) and (7) highlight the correlation between the air flow parameters, the engine building formula and the engine working conditions.
Therefore, the correction coefficient of the air mass admitted in the cylinder , β, may be written as:

$$\beta = p_{ga} / p_{gas} \tag{8}$$

where p_{gas} is the standard pressure technically accepted = 1 bar.

THE COLLABORATION OPTIMIZATION USING THE S.I.E. SIZING BY SELECTING THE AUTOMATIC CONTROL LOOP OF THE MIXTURE QUALITY LET IN THE CYLINDER

The aim of this modelling is to find the working way of the λ sound (open loop or shut loop), as well as to make the necessary preliminary calculations to determinate the excess air coefficient which will be used in the next strategies /4/.
In order to calculate the fuel quantity necessary for the stoichiometric burning one starts from the air mass admitted in the cylinder (The calculation of the volumetric coefficient).

$$m_{cs} = m_{ar} \cdot R_{st} \tag{9}$$

where: m_{cs} - is the fuel quantity necessary for the stoichiometric burning [mg]; m_{ar} - the real air mass in the cylinder [mg]; R_{st} - the stoichiometric ratio m_c / m_a (ex: $\frac{1}{14,3}$).
Normally λ value is "1" to give maximum efficiency to the catalytic converter, but in some cases λ can have other values as well (figure 2):
- the mixture enriching for stability in the case of idle conditions,
- the mixture enriching for maximum performances,
- the mixture enriching for heat protection of the catalytic reactor.

The calculation of the necessary fuel enrichment for high performances (ex: λ =0,892) or for heat protection of the catalytic converter uses in a first phase the formula (9) where the necessary fuel quantity for stoichiometric burning is found, thus the formula develops:

$$m_{ci} = \frac{m_{cs}}{\lambda_i} \tag{10}$$

where : m_{ci} - the fuel quantity nedded to allow the enriching value [mg]; λ_i - the imposed value by the enriching [-]; m_{cs} - the fuel quantity needed for the stoichiometric burning [mg].

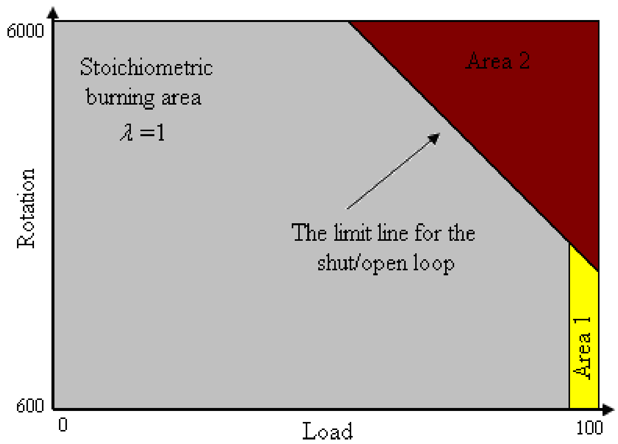

Figure 2. The engine working area

If the excess air coefficient equals 1 implies the best efficiency of the catalytic converter in treating the chemical emissions, the values between 0,909 and 0,869 (area 1 in figure 3) give high performances, with the observation that the efficiency of the catalytic convertor decreases leaving the setting range $1 \pm 0{,}03$.

As we can see in figure 3, the decrease of the excess air coefficient leads to the increase of Ce (specific consumption: the ratio between the injected fuel quantity and the obtained power), but the performances increase (the maximum torque was found for λ =0,892).

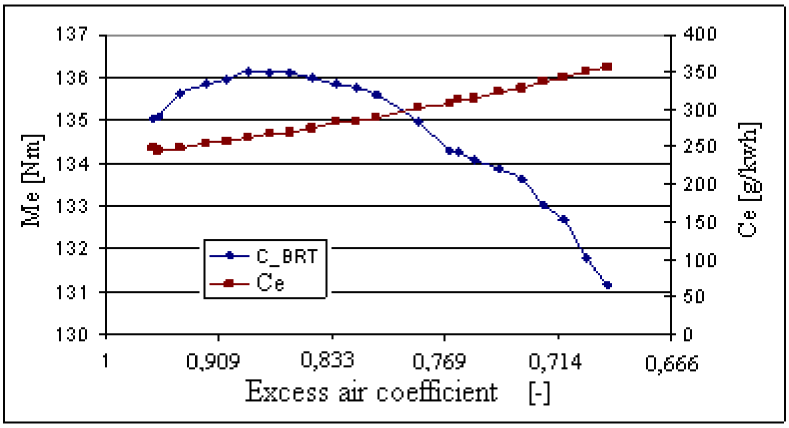

Figure 3. The influence of the mixture quality on the performances and consumption

In the case of high loads and/or rotations the exhaust gas temperature can be higher, exceeding the heat limits of the catalytic converter. In order to avoid the heat overcharging of it, the injected fuel quantity should be increased without respecting the area where λ=1 or the area for high performances. Experimentally, one had shown that at a drop of 0,01 of the excess air coefficient the exhaust gas temperature decreases with ~3 ºC when leaving the cylinder head and with ~4ºC at the global level of the exhaust line.

THE COLLABORATION OPTIMIZATION USING THE S.I.E. SIZING THROUNGH THE INJECTION TIME CALCULATION MODELS

This strategy connects the different modellings (the volumetric efficiency calculation, the calculation of the fuel quantity needed for the stoichiometric burning or the enrichment calculation) to get the final value of the applied injection time.

From the injection time point of view one should take into consideration several technical aspects which could lead to the injection of a wrong fuel quantity. One should know the static flow of this one when it is open (the needed time for the atomization of one fuel mg), the injector's dead time (the opening and shutting period of the injector), but as well the value of the tension applied to the injector (figure 4).

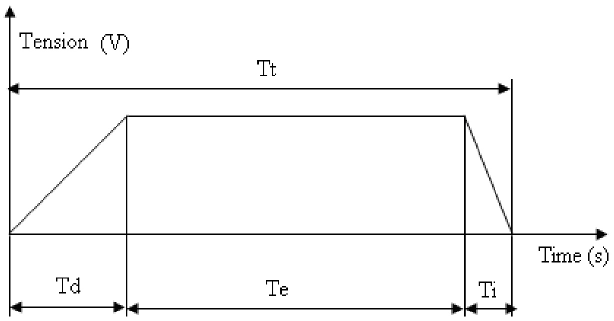

Figure 4. Injector working way

where: Tt – total time when the injector is under pressure; Te – the best time when the injector atomizes at maximum capacity; Td – opening injector time; Ti – shutting injector time.

In order to fiind the necessary injection time for injecting the optimal fuel quantity , within the best time when the atomization is maximum and constant (static flow) one adds up the time when the injector has a null flow (figure 5), this is approximately:

$$\frac{Td - Ti}{2} \tag{11}$$

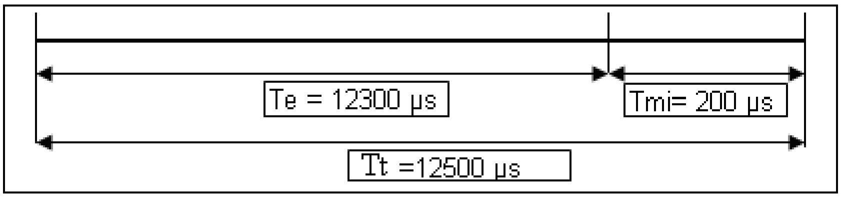

Figure 5. The injection time calculation

Figure 6 shows synthetically the influence of the injector dead time on the injected fuel quantity. If after modelling the filling volumetric coefficient and the fuel quantity that has to be injected for a given condition, the value of the injector dead time is not correctly chosen, then the total injection time can increase or decrease. In the present case the best injection time has been established at the value of 12300 µs where an injector dead time of 200 µs has been added, if the injector dead time would have been chosen at a value of 250 µs then in reality the injected fuel quantity would have been higher with 50 µs, leading to rich mixtures.

Starting from the value of the null flow (the injector dead time) and the static flow (the best time) given by the manufacturer or measured with specific equipment one can find the applied injection time value neccesary for the optimal dose :

$$T_t = m_{cn} \cdot D_{si} + T_{mi} \tag{12}$$

where: m_{cn} - the fuel quantity calculated for the given working conditions [mg]; D_{si} - injector static flow [µs/mg]; T_{mi} - injector dead time [µs].

EXPERIMENTAL RESEARCH CONCERNING THE COLLABORATION OPTIMIZATION OF THE S.I.E. WITH THE DEPOLLUTION SYSTEMS

For the experiments /4/, /5/ and /7/ one has chosen a series of 12 working conditions (rotation/load) which characterizes the engine working at low, medium and high loads (tab 1).

Table 1

Conditions	Rotation [rot/min]	Load [mbar]
R1	1900	440
		770
		Full load
R2	3000	440
		770
		Full load
R3	4200	440
		770
		Full load
R4	5000	440
		770
		Full load

In order to have a satisfactory optimized collaboration of the S.I.E. with the catalytic converter (T.W.C.) for the chemical emissions treatment, as well as to avoid its heat overcharging, one had in view that the difference between the filling volumetric coefficient calculated with the help of the shown model and the one measured at the engine stand should range between $\pm 1\%$ (table 2 and 3).

Taking into consideration the filling volumetric coefficient calculated with the help of η_{vc} model, it is refered to 25 ºC and 1,013bars air pressure, and the filling volumetric coefficient found at the engine stand η_{vd}, is referred to the same temperature but 1 bar air pressure, the relation to calculate the volumetric difference E_v is:

$$E_v = \frac{\eta_{vc} - (\eta_{vd}/1,013)}{(\eta_{vd}/1,013)} \cdot 100[\%] \qquad (13)$$

Table 2

Vs [m³]	ρfp [kg/m³]	Patm [bar]	ma ref [mg]	D [mm]	da [mm]	S [mm]	Δαa [º]	R [J/kg·K]	Dsi [µs]	Tmi [µs]
0,0004	1,1845	0,973	473,2	80,5	33	80	260	287	458	180

Table 3

Rotation [rot/min]	Pca [mbar]	ηv measured [-]	Pga [mbar]	Wga [m/s]	Wp [m/s]	Tca [K]	Pzero [mbar]	m [mg]	ηv calculated [-]	Ecart ηv [%]	ça [-]
	440	0,332	0,9670	6,93	5,07	297,6	100	153,81	0,325	-0,821	20
1900	770	0,624	0,9368	13,02	5,07	297,8	100	293,44	0,620	0,670	35
	968	0,794	0,9226	16,57	5,07	298,3	100	373,74	0,790	0,767	30
	440	0,323	0,9361	10,65	8,00	300,9	89	152,02	0,321	0,756	54
3000	770	0,622	0,9257	20,50	8,00	300,2	89	292,36	0,618	0,622	18
	953	0,834	0,9676	27,49	8,00	301,7	89	385,79	0,815	-0,973	0,2
	440	0,359	0,9567	16,56	11,20	303,4	55	169,02	0,357	0,791	9
4200	770	0,667	0,9562	30,78	11,20	305,5	55	311,51	0,658	-0,022	2
	944	0,82	0,9476	37,84	11,20	306,2	55	383,01	0,809	-0,009	2
	440	0,34	0,9627	18,68	13,33	305,8	75	159,97	0,338	0,722	4
5000	770	0,637	0,9367	34,99	13,33	305,8	75	296,40	0,626	-0,391	4
	938	0,788	0,9397	43,28	13,33	306	75	368,97	0,780	0,237	2

Therefore, by finding the air mass admitted in the cylinder one can establish the fuel quantity necessary for the stoichiometric burning, the enriching of the mixture for high performances (figure3, area 1) and the enriching of the fuel to avoid the heat overcharging of the catalytic converter (figure 3, area2). Table 4 shows synthetically the fuel quantity calculated for each given working condition, one had in view the finding of the fuel optimal quantity for stoichiometric burnings, m_{cs}, and the fuel quantity needed for the enriching of the mixture both for high performances and the protecting of the catalytic converter m_{ci}. In order to control the temperature within the accepted working limits of the catalytic converter /4/, by experimental findings at the engine stand an excess air coefficient is needed to meet these requirements. For example for the speed of 4200 rot/min and the load of the intake manifold of 770 mbar, in order to keep the monolith temperature of the catalytic converter at 950°C a value λ of 0,870 at the engine stand is needed. After obtaining the optimal fuel quantity needed to enrich the 25,03 mg mixture, one finds the total injection time, $T_t = 11643 \mu s$, based on the values of injector dead time and injector static load (table 2).

Table 4

Rotation [rot/min]	Pca [mbar]	mar [mg]	mcs [mg]	mci [mg]	Tt [µs]	λ impus [-]
	440	153.81	10.75	10.75	5104	1.000
1900	770	294.38	20.58	20.58	9604	1.000

Rotation [rot/min]	Pca [mbar]	mar [mg]	mcs [mg]	mci [mg]	Tt [μs]	λ impus [-]
	968	373.74	26.12	29.26	13581	0.893
	440	152.35	10.65	10.65	5057	1.000
3000	770	292.36	20.44	20.44	9540	1.000
	953	385.79	26.97	30.20	14013	0.893
	440	169.31	11.83	11.83	5600	1.000
4200	770	311.51	21.77	25.03	11643	0.870
	944	386.44	27.01	31.78	14735	0.850
	440	159.97	11.18	11.18	5301	1.000
5000	770	296.4	20.72	24.52	11410	0.845
	938	368.97	25.79	31.26	14498	0.825

CONCLUSIONS

After analyzing the experimental data the following real possibilities of improving the collaboration of S.I.E. with the depolluting system appear, namely:

- the rigurous control of the air mass admitted in the cylinder ensures both the possibility of stoichiometric burnings for the maximum efficiency in treating the chemical emissions, and the protection of the catalytic converter regarding heat requirements.
- the rigurous control of the injection system parameters leads to an economical and depolluted engine.
- in order to reach this desideratum one should find exactly the real filling coefficient, as well as the air mass aspirated in the case of a difference less than $\pm 1\%$. This way one can rigurously control the dose of the injected fuel as well as the corresponding injection time.

REFERENCES

[1] Cristea, D., Căi de optimizare a motoarelor cu ardere internă, Ed. Universității din Piteşti, 2009.

[2] Grunwald, B., Teoria, calculul si constructia motoarelor pentru autovehicule rutiere, Ed. Didactica si Pedagogica din Bucuresti, 1980.

[3] Ivan, F., Liță, D., *Contribution to the improving of the features of a quick cold start Diesel engine,* ESFA Bucureşti, Volumul 2, 2009.

[4] Ivan, F., Liță, D., Buşoi, A., Research on the construction and performances of the three way catalytic used in depolluting the S.I.E, Revista Ingineria Automobilului, Numarul 19, 2011.

[5] Liță, D., Ivan, F., *Construction, function and influence of exhaust gas recirculation upon pollutants,* Buletin Ştiințific Universitatea din Piteşti, Numarul 19, Volumul B, 2009.

[6] Liță, D., Ivan, F., *Ways of improving the particle filters' performances used in depolluting the quick Diesel engines,* CONAT Braşov, Volumul 2, 2010.

[7] Liță, D., Ivan, F., *Experimental research regarding the depollution of the formation and the improvement of top dynamic performances of cars through the optimization of the final moment of injection,* Buletin Ştiințific Universitatea din Piteşti, 2010.

[8] Mitrache, I., Ivan, Fl., Dumitrescu, V., Termotehnică şi maşini termice, Ed. Universității din Piteşti, 1995.

[8] Negrea, D., Sandu, V., Combaterea poluării mediului în transporturile rutiere, Ed. Tehnica Bucuresti, 2000.

Acknowledgement: This work was partially supported by the strategic grant POSDRU/88/1.5/S/52826, Project ID52826 (2009), co-financed by the European Social Fund – Investing in People, within the Sectoral Operational Programme Human Resources Development 2007-2013.