ON THE POSSIBILITY TO ASSESS POLLUTANT EMISSIONS OF A DIESEL ENGINE BY APPROXIMATION FUNCTIONS

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Abstract

Controlling the emissions is extremely important nowadays. In terms of engine operation, the actual engine control units allow the rapid adjustment of the values for many parameters governing the processes inside the cylinder.

The effective measurement of emissions is on the contrary a slow procedure, depending on many variables and featuring a limited precision. Therefore, a group of approximation functions based on the most sensitive factors could be more effective and preferable for combustion process control.

The proposed paper is offering an analysis of such functions, using experimental data and results obtained by advanced modeling programs. Finding certain methods to adjust the combustion for reaching desirable efficiency and emissions results represents also another goal of the presented work.

Introduction

Numerous research activities have strongly certified that modern Diesel engines are to accomplish complex and meanwhile opposite aims not only in stationary, but mainly in transient operating conditions, meaning that they are requested to rate high performance, to generate low emissions and featuring high operation efficiency. The allowable limits for NOx and soot Diesel emissions have drastically decreased from Euro 1 down to Euro 6b standards, despite the fact that representative driving cycles have not significantly changed related to the applied technology update, such as the use of Diesel Oxidation Catalyst, Cooled EGR, Diesel Particulate Filter or Lean NOx Trap [1]. In order that Diesel engines main simulation emissions comply with those of real drive measurement, engines calibration made in the area of steady-state speeds and loads must extend upon the driving transient behavior, driver disturbances and transient ambient conditions.

Corresponding to a first view, phenomenological models may predict the values for the indicated mean effective pressure (IMEP) according to the rated torque and to the NOx and smoke emissions on a cycle-by-cycle analysis. Results show a well prediction for the emissions resulted from stationary operating conditions and a slight deviation in case of smoke emissions for transient operation, explained by a low pass characteristics of the measurement apparatus [2,3]. Following these ideas, charting engine maps of fuel use according to imposed values for performance and emissions referring to transient driving cycles' conditions is a novel way to cover a range of various vehicles operation. Thus, the availability of accurate engine maps across a number of vehicles (DAF, Chevrolet Cruze and Volkswagen Golf) has been verified under specific conditions, like

those stipulated by the main Urban Dynamometer Driving Schedule (UDDS), by Federal Test Procedure (FTP), Highway Fuel Economy Test (HWFET) and by the high speed and aggressive US06 driving cycle [4]. These maps include: the insights into how control technologies can be tuned to reduce emissions under real driving conditions, the development of the in-use emissions factors, the identification of driving conditions corresponding to the most significant and incidental of fuel use and emissions events and the adjustment of the air-fuel mixture formation, using simulated per second emissions for vehicles operating in the urban environment or on certain road segments. An example of a new strategy calculates set points for the Engine Management System (EMS) for controllable fuel quantities with the purpose to minimize fuel consumption for a given engine speed and requested torque profile, while keeping accumulated emissions below given limits. There were followed three (EMS) evaluating approaches, using a complete Diesel engine vehicle system model and simulating the NEDC driving cycle: one is based on the steady-state methodology and two others use explicit transient compensations in terms of boost pressure and Oxygen fractions as feedback signals [5,6]; the conclusions were that large differences were noticed between the first and those last approaches while small differences were characterizing the results in-between the transient approaches.

For real-time applications, it is not reasonable to model the emissions only by phenomenological models, because of the large complexity of the physical and chemical reactions in the cylinder of an ECU-controlled engine. Therefore, beside phenomenological and map-based models there are semi-empirical models concerning NOx and soot estimation. These ones are based on polynomial functions, where the influence of each input parameter should be known from steady-state engine behavior [7-9]. There are also other modeling techniques called Artificial Neural Networks (ANN), developed as parallel processing algorithms, such as Multi-Layer Perceptron (MLP)[10], or Time-Delay Recurrent (TDNN) [11-14].

Emissions evaluation by empirical equations

The calculation of the cylinder inside emissions is a difficult task, characterized also by a high error level despite the great complexity of the calculating models. This is because of the in-cylinder thermal and chemical lack of uniformity, which is difficult to be established and is responsible for the conditions to generate nitrogen oxides and soot in certain zones, while global pressure, concentration and temperature values theoretically do not allow these combustion products to occur.

Another method marked by imprecision, but much faster considering the computation speed is that of introducing empirical equations in which the density of the emissions is given by a series of the engine's operating parameters. One paper [15], based on the analysis of a great number of pressure diagrams obtained from numerous engines and testing regimes is proposing the following calculating expressions for NO_x and smoke concentrations, starting from the indicated pressure diagram and from the inlet system parameters:

$$S = K_1 \times \tau_{aa}^{e1} \times \beta_1^{e2} \times (\frac{dp}{d\alpha})_{\max}^{e3} \times \beta_2^{e4} \times O_2^{e5} \times (\frac{m_{air}}{m_{total}})^{e6} \quad (1)$$

$$NO_x = K_2 \times \tau_{aa}^{f1} \times \beta_1^{f2} \times (\frac{dp}{d\alpha})_{\max}^{f3} \times \beta_2^{f4} \times O_2^{f5} \times (\frac{m_{air}}{m_{total}})^{f6} \quad (2)$$

Both equations depend on the shape of the rate of hate released, given by the two β angles, on the autoignition delay τ_{aa} and on the pressure diagram, characterized by the top value of the firing increasing rate (see Figure 1). The oxygen quantity from the exhaust gas delivers the value of the relative air-fuel ratio and subsequently the value of the engine coefficient load as that of the gas recirculation degree at part-loads regimes. The ratio between the air mass and the total aspirated mass gives the level of the recirculation fraction in the measuring point. The large number of the testing points analyzed in the paper [15] leads to some results marked by high dispersion and therefore, the authors of the present paper proposed two series of different exponents for different engine type and operating conditions.

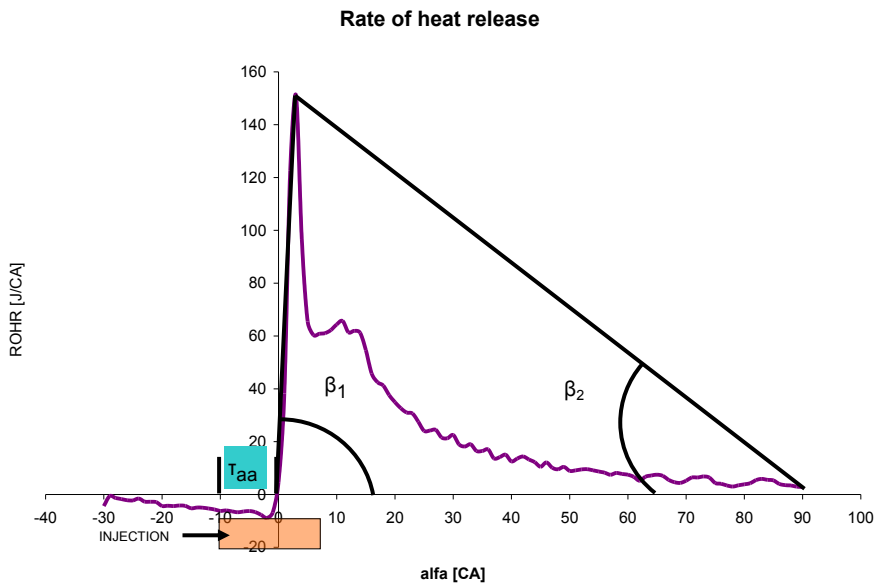

Figure 1 Graphical explanation for β angles

Experimental results and coefficients adjustment

The tested engine was a 4-cylinder, in-line, normally aspirated DI Diesel tractor engine, with a stroke/bore ratio of 115/102 mm/mm, rating a maximum power of 50 kW at 2400 rpm. For the analysis based on the proposed method, operating conditions corresponding to a partial power of 32 kW at 1400 rpm were chosen, together with the relative air-fuel ratio value of 1.34. The engine was operating without exhaust gas recirculation with the air as the single aspirated agent. The cylinder pressure and the rate of heat release (ROHR) characteristics are plotted in Figure 2.

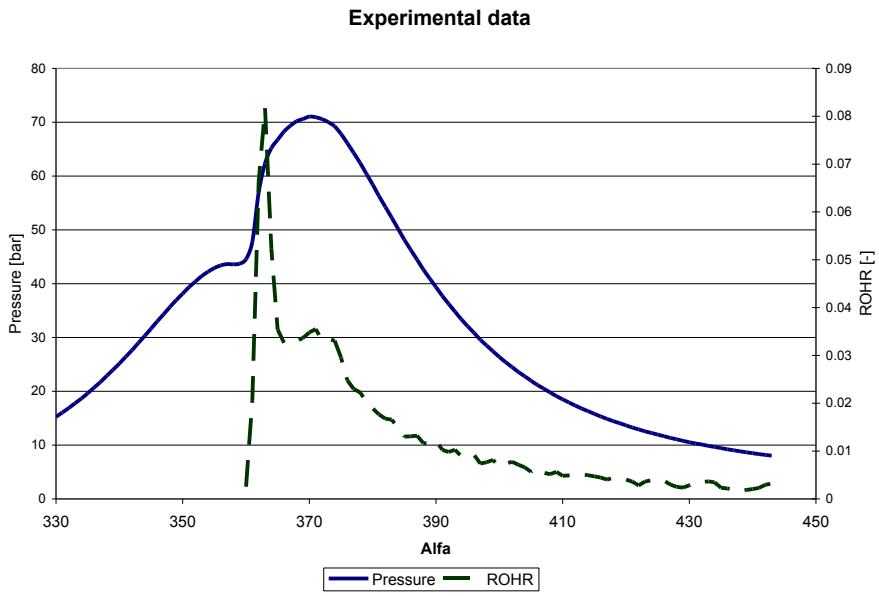

Figure 2 Pressure and rate of heat release characteristics

From these experimental data, the following results were obtained related to the emissions calculating expressions: τ_{aa} = 0.76 ms, β_1= 88.8 deg.CA, $(dp/d\alpha)_{max}$= 10.02 bar/deg.CA, β_1= 58 deg.CA, O_2=7.3%, m_{air}/m_{total} =1. The subsequent emissions values were S = 5 FSN and NO_x=1050 ppm. Considerring these, one series of values given in the paper [15] were chosed in order to fit as good as possible the experimental disposed data, as listed in Table 1:

Table 1 Coefficients values

	1	2	3	4	5	6
e	**0.7**	**0.77**	**-0.6**	**-0.93**	**-2.07**	**-0.14**
f	**0.23**	**1.16**	**1.86**	**-0.65**	**0.16**	**1.26**

The K coefficients from the front side of the equations were chosen with their values K_1=2040 and K_2=0.86, so that using the proposed equation for the experimental emissions values confirmed for smoke the same value of S = 5 FSN and for NO_x =1037 ppm, comparing to the measured value of 1060 ppm, meaning a relative error of about 2%.

The mathematical model

To evaluate the efficiency of such a formula it is necessary to apply it for a data series obtained by modeling the in-cylinder processes so that the best compromise between the emissions, the efficiency, the top firing pressure and eventually other engine parameters could be obtained. This paper would like to identify one favorable combination of split injections corresponding to a number of injection on-cycle advance values in terms of emissions reduction for the analyzed regime (output power of 32 kW at 1400 rpm, only aspirated air, relative air-fuel ratio of 1.34). To fulfill this goal, an AMESIM model was developed, with the combustion hypothesis kept as described in the paper [16]. Maintaining the engine features, a good approximation between calculated and experimental data highlighted, as seen in Figure 3.

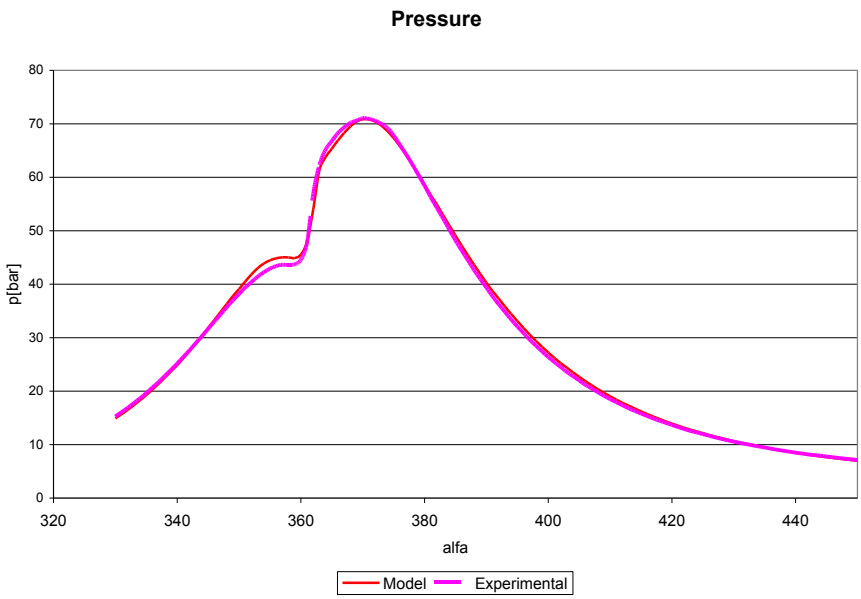

Figure 3 Cylinder pressure vs. crankshaft angle

In a first step, in the case of the undivided injection, the study focused on the injection advance in order to evaluate the way in which the emissions calculation formula reacts to the modifications listed in Table 2. Thus, the NO_x emissions are expected to grow when increasing the injection advance set values [17]. Analyzing the results show the fact that NO_x emissions increase from 1050 ppm to 4204 ppm while smoke drops at about its half value. Indicated mean pressure (IMEP) rises with approximately 11 % when the top firing pressure increases by 50%.

Tabel 2 Results obtained by single injection per cycle

No.	Injection adv. [deg.CA]	p_{max} [bar]	IMEP [bar]	$(dp/d\alpha)_{max}$ [bar/deg.CA]	NO_x [ppm]	S [FSN]
0	9	70	9.03	10	1050	5
1	12	81.38	9.35	12.63	1350	3.67
2	15	89.46	9.57	15.5	1999	3.207
3	18	98	9.78	18.79	2892	2.81
4	21	107	9.99	22.77	4204	2.49

In conclusion, using the given formula leads to results well comparable to those from the literature references. Following a second step, for the split injection 12 cases were followed, corresponding to injection advance values between 25 and 35 deg. CA BTDC; the percentage of the injected fuel mass during the initial phase varied from 11% to 20% and the advance of the main injection phase was delayed until 0 deg. CA BTDC (see Table 3). By consequence, the results are dispersed. Interesting cases appear to be those numbered with 5 and 8, for which NO_x emissions decrease without significant increase of the smoke ones. For both these cases, the relative heat release characteristics are traced in Figure 4.

Tabel 3 Results obtained by split injection per cycle

No.	Inj. adv.1 [deg.CA]	Inj. mass 1 [mg]	Inj. adv.2 [CA]	Inj. mass 2 [mg]	p_{max} [bar]	IMEP [bar]	$(dp/d\alpha)_{max}$ [bar/deg.CA]	NO_x [ppm]	S [FSN]
0	9	44	-	-	70	9.03	10	1050	5
5	25	9	10	35	87	9.31	9.06	876	5.34
6	25	9	5	35	74.56	8.93	8.44	880	6.67
7	25	9	0	35	65.47	8.51	8.42	898	6.92
8	25	5	10	39	82.29	9.21	8.99	861	5.32
9	25	5	5	39	69.6	8.81	6.02	439	7.85
10	25	5	0	39	58.5	8.35	6.06	444	8.02
11	30	5	15	39	95.68	9.57	13.82	1876	3.82
12	30	5	10	39	82.3	9.18	8.3	797	6.02
13	30	5	5	39	69.82	8.81	5.9	444	8.25
14	35	9	10	35	87.07	9.25	8.24	918	7.07
15	35	9	5	35	75.13	8.9	8.25	951	7.4
16	35	5	5	39	70	8.79	5.77	446	8.6
17	35	5	0	39	59	8.33	5.81	454	9

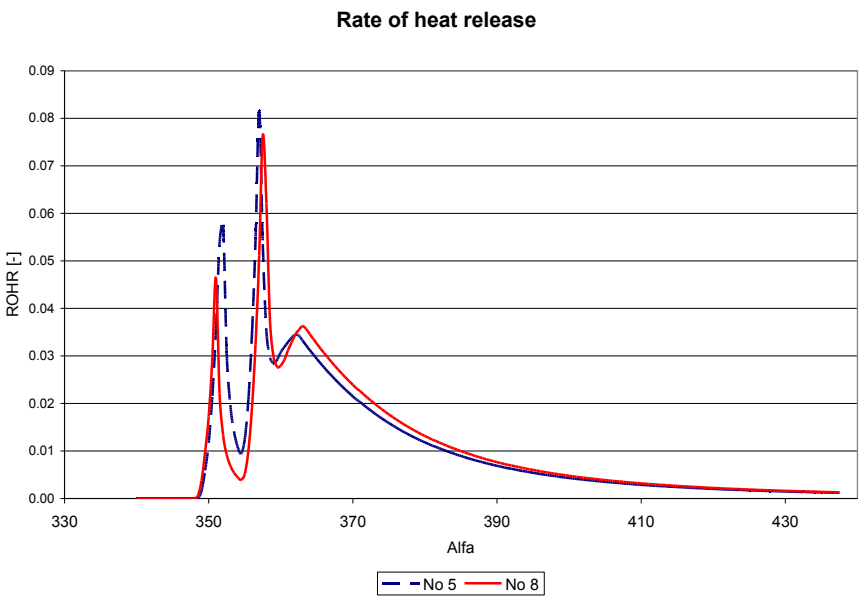

Figure 4 Rate of heat release for cases No.5 and No. 8

Conclusions

This work is proposing a relatively simple method to evaluate the emissions starting from the analysis of the pressure diagram. The functions used to establish NO_x and smoke concentrations use as variables the angle deducted from the shape of the relative heat release characteristic, the pressure increasing rate, the autoignition delay, the oxygen fraction from the exhaust gas and the exhaust gas recirculation degree.

The comparison between the simulated results and the experimental data revealed a good agreement. Base on this achievement a limited parametric study was performed by modifying the injection type and parameters showing this way a better strategy to control Diesel engine emissions.

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Nomenclature

S	Smoke, FSN
NO_x	nitrogen oxide, ppm
τ_{aa}	ignition delay, ms
β_1	the angle between the line joining the ign. point with the max. point of the ROHR and the axe, deg.
β_2	the angle between the line joining the max. point of the ROHR with the end of comb. and the axe, deg.
$(dp/d\alpha)_{max}$	the maximum for the pressure derivative, bar/deg.CA
O_2	oxygen concentration in exhaust gas, %
m_{air}/m_{total}	ratio between the air mass and total mass at intake
IMEP	indicated mean effective pressure
ROHR	rate of heat release
$e_1..e_{e6}$	coefficients
$f_1..f_{e6}$	coefficients
K_1,K_2	coefficients