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ON THE POSSIBILITY TO ASSESS BY SIMULATION PERFORMANCE AND EFFICIENCY PARAMETERS OF A SI ENGINE OPERATING IN DIFFERENT CONDITIONS

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A spark ignition engine type Renault K7M710 fuelled with gasoline was tested on a test bed at light load and low speed, characteristic for the city traffic operation, for different ignition timings and different air-fuel ratios. A simulation model of the engine was developed in AVL BOOST software in order to obtain a predictive tool which can be further extrapolated for the use of alternative fuels. The model of this engine was explored and validated, by comparisons of simulation results with experimental data for performance and efficiency parameters, which finally shown very good fitting relative deviations over all considered operating conditions.

Keywords: SI engine, spark timing, zero-dimensional model, Wiebe function, two zone combustion model

1. Introduction

Models which are used in combustion process simulation can be classified as:

- thermodynamic models;
- phenomenological models;
- multi-dimensional models.

The thermodynamic modelling is the simplest method that can offer real cost cut-off solutions in the engine's development activity. First law of thermodynamics provides the key equation in the incremental procedure used in thermodynamic models. In these models the cylinder charge is divided into a number of zones with uniform pressure, temperature and composition [1].

Usually the number of zones is two (one dedicated to the fresh mixture and the other one dedicated to the burned gases), but sometimes the burned gas in its turn can be divided into several distinct zones for taking into account the temperature stratification during combustion.

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In [2] a thermodynamic combustion model with two zones was used to predict the performance characteristics (power and indicated specific fuel consumption) of a spark ignition (SI) engine. The simulation results have been evaluated against experimental data and has been found reasonable agreement. Parametric studies have been carried out to assess the effects of equivalence ratio, compression ratio and spark timing on the engine performance characteristics in order to show the capability of the model to predict engine performance and efficiency.

In paper [3] the authors have used a multi-zone thermo-fluid dynamic combustion model for the prediction of SI engine performances and emissions. It has been also evaluated the behaviour of the engine when fuelled either with gasoline, either with compressed natural gas. Despite the limitations imposed by the use of a quasi-dimensional approach, the results achieved are characterized by a good level of predictiveness and accuracy.

The researchers of paper [4] used a zero-dimensional simulation model to predict performance characteristics of a SI engine fuelled with biogas. The results of simulations and experimental test on the engine in similar conditions were compared and the model was validated (the relative difference between the simulation results and experimental data was lower than 5%). The model can be adapted for use of different raw or enriched biogas compositions and could prove to be a valuable tool to guide experimentation, reducing time and resources requested.

In [5] the authors evaluated experimentally and theoretically through a zero-dimensional combustion model the performances of a SI engine using n-butanol, hydrous ethanol and n-butanol/ethanol, gasoline/ethanol blends as fuels. The combustion model results have been considered quite satisfactory regarding the accuracy achieved, being validated for all fuels tested.

Paper [6] relates the experiments performed and developed through one zero-dimensional modelling study concerning the use of producer gas, a bio-derived fuel with low energy content in a SI engine. The standard Wiebe coefficients for conventional fuels and fuel-specific coefficients for producer gas were used to predict the heat release characteristics in order to evaluate the performance of a SI engine under naturally aspired and turbo-charged conditions. While simulation results with standard Wiebe coefficients lead in excessive deviations from the experimental results, excellent matching was observed when producer gas-specific coefficients were used.

In this work, a two-zone combustion model 0-D with Wiebe function was used for the estimation of the performance parameters of a SI engine using the AVL BOOST software in order to obtain a predictive tool which can be further extrapolated for the use of alternative fuels.

2. Experimental setup and test procedure

The test-cell is formed by a control room and an engine test bench (Fig.1).

Fig. 1. The engine test cell with the control room and the test bench

The engine under test was a passenger car spark ignited Dacia Logan 1.6L 8v K7M-710 with the technical specification presented in Table 1.

Table 1

Technical specification of Dacia Logan, engine type Renault K7M-710

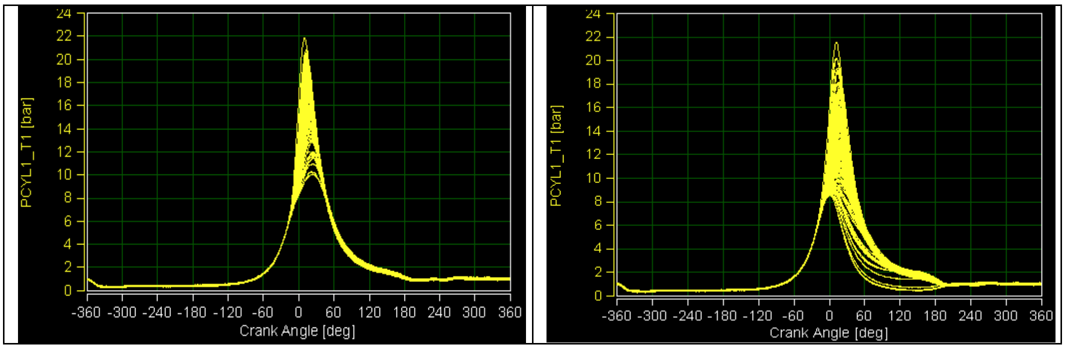
Power	Torque	Bore	Stroke	Compression ratio	Valves per cylinder	Fuel system
64 kW/ 5500 rpm	128 Nm/ 3000 rpm	79.5 mm	80.5 mm	9.5	2	Multi-point injection

Several determinations were made at constant speed 2000 rpm and 2 bar brake mean effective pressure - BMEP (constant load). The engine was tested in a spark timing characteristic looking for the optimum spark advance for two air-fuel ratios (AFR) respectively stoichiometric mixture $\lambda=1$ and lean mixture $\lambda=1.25$.

The ECU reference calibration maps for spark timings and for injection durations were changed using ETAS INCA v.7.2.1 software tool. The transition from $\lambda=1$ to $\lambda=1.25$ was made by lowering the pulse durations in order to reduce the fuel amount injected per cycle and cylinder. The shape and duration of injection pulses, the shape and duration of the spark voltage discharge, the variation of the cylinder pressure and the lambda sensor signal were visualized on four channel oscilloscope and registered on the test-bed computer.

For each of these determinations, global parameters were measured using the Froude Consine Texcel V6 software.

The AVL Indicom software was used for the pressure traces registration over 500 consecutive cycles, the mean pressure diagram and the main events specific for combustion characteristics as 5%, 10%, 50% and 90% of the heat release were also determined. The plotted of the pressure indicated traces at optimum spark timings for stoichiometric and lean mixture conditions are presented in Fig. 2. As it can be seen in this figure for lean mixture operation the cyclic variability is much more noticeable, with larger envelope covering from 6 to 22 bar.



a) λ=1 b) λ=1.25

Fig. 2. Pressure traces evolution over 500 consecutive cycles for optimum spark timings

3. Results obtained from the experimental data

One of the main important causes by which performance, efficiency and emissions of the petrol engines are negatively affected is the phenomenon of cycle-by-cycle variation. The cyclic variability is much more accentuated in lean or diluted mixtures in which also the ignition conditions are much more severe [7].

To characterize this harmful phenomenon the following parameters are usually used: the coefficient of variation in indicated mean effective pressure for high pressure loop (COV_{IMEPH}), as well as the coefficient of variation in peak fire pressure (COV_{pmax}). The results on the cyclic variability parameters that have been obtained in the tests performed are exposed in Fig. 3. Values lower than 2% respectively 13% are registered for COV_{IMEPH} and COV_{pmax} at stoichiometric mixture and optimum spark timing operation. For the lean mixture condition. both coefficients are much higher, achieving values of 13% respectively of 21% at optimum spark timing.

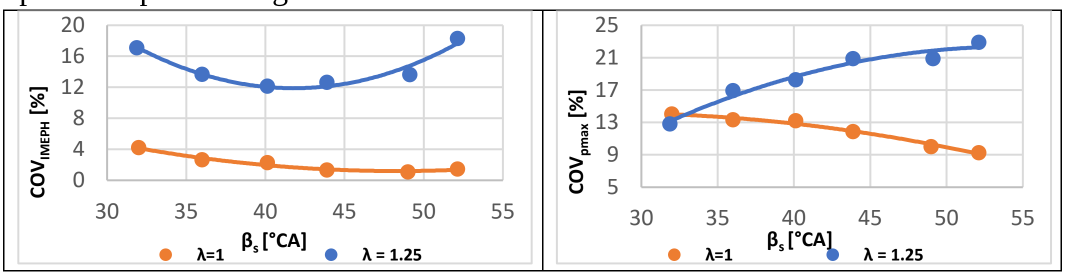


Fig. 3. Variation of COV_{IMEPH} and COV_{pmax} versus ignition timing (β_s) for engine speed of 2000 rpm and BMEP = 2 bar at λ=1 and λ=1.25

The performance, and efficiency parameters (effective power - EP and brake specific fuel consumption - BSFC) and the combustion characteristics too (based on the pressure diagrams) depend on the spark timing (β_s).

The spark advance (β_s) is reported as a positive value relative to the top dead center (TDC) position. Fig. 4 shows the performance and efficiency parameters variation versus the spark advance.

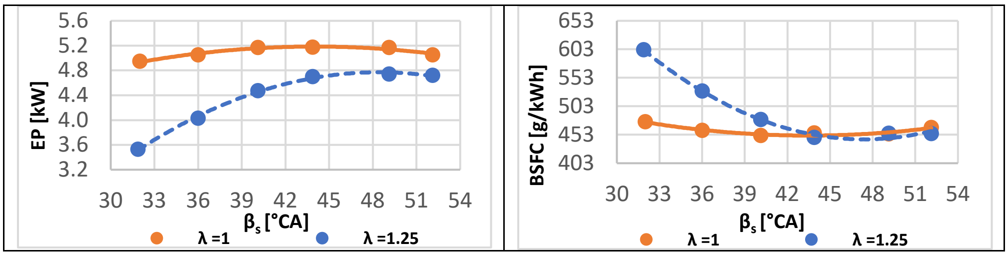
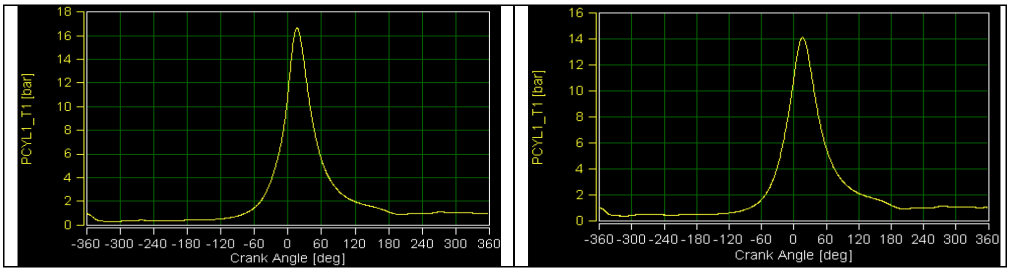


Fig. 4. Variation of the EP and BSFC with the spark advance (β_s)

The optimum spark timing in terms of EP and BSFC have been considered as being 40.12 °CA for λ=1 and 49.12 °CA for λ=1.25. At these timings there were registered the maximum performance and the minimum specific fuel consumption.

The average pressure traces over 500 consecutive cycles for the tested conditions previously mentioned are presented in Fig. 5. At optimum spark timings, the shape of mean pressure trace plotted against the crankshaft position are similar for both air-fuel ratios but, with smaller value by 2 bar for the peak fire pressure on lean mixture operation.



a) λ=1 b) λ=1.25

Fig. 5. The mean pressure diagrams for the optimal spark timings visualized with Indicom software

The combustion process has been divided into two stages:

- The initial combustion phase (from the spark timing at the moment when 5% of the heat was released);
- The main combustion phase (from the moment when 5% of the heat was released until the 90% of the heat was released).

The variations of these stages are shown in Fig. 6 for both air-fuel ratios. The duration of the initial stage of the combustion process increases with 21% at λ=1 and 30% at λ=1.25 because the pressures and temperatures in the cylinder decrease as the spark timing is moving in advance in compression stroke in respect to TDC end of compression. The duration of the main combustion stage decreases with 19% at λ=1 and 30% at λ=1.25 as the spark timing increases because the combustion takes place even further in the compression stroke where two effects overlap (temperature increase by compression made by the piston displacement and temperature rise by the heat release by combustion). This behaviour seems to be similar for both air-fuel ratios but with higher values for lean mixture operation.

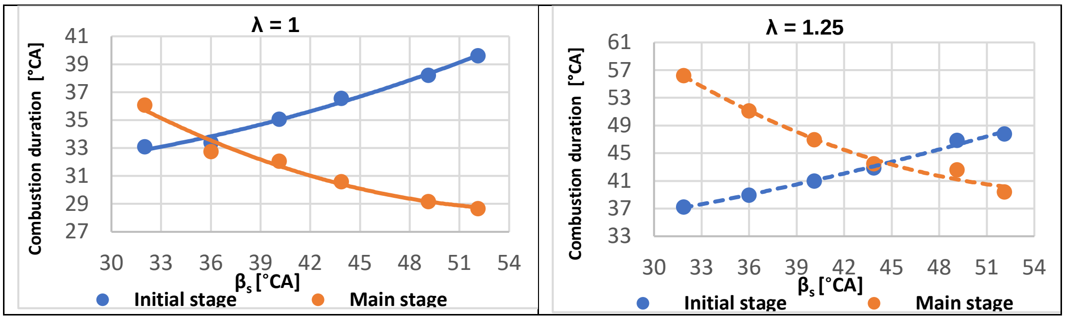


Fig. 6. The variation of combustion phases durations relative to the spark timing (β_s)

4. Simulation modelling

The AVL BOOST software was used to create a model for the K7M-710 engine. The model is made up of system boundaries (SB), pipes, air cleaner (CL), throttle (TH), junctions (J), injectors (I), cylinders (C), catalyst (CAT) and plenums (PL). Fig. 7 shows the symbolic model of the engine.

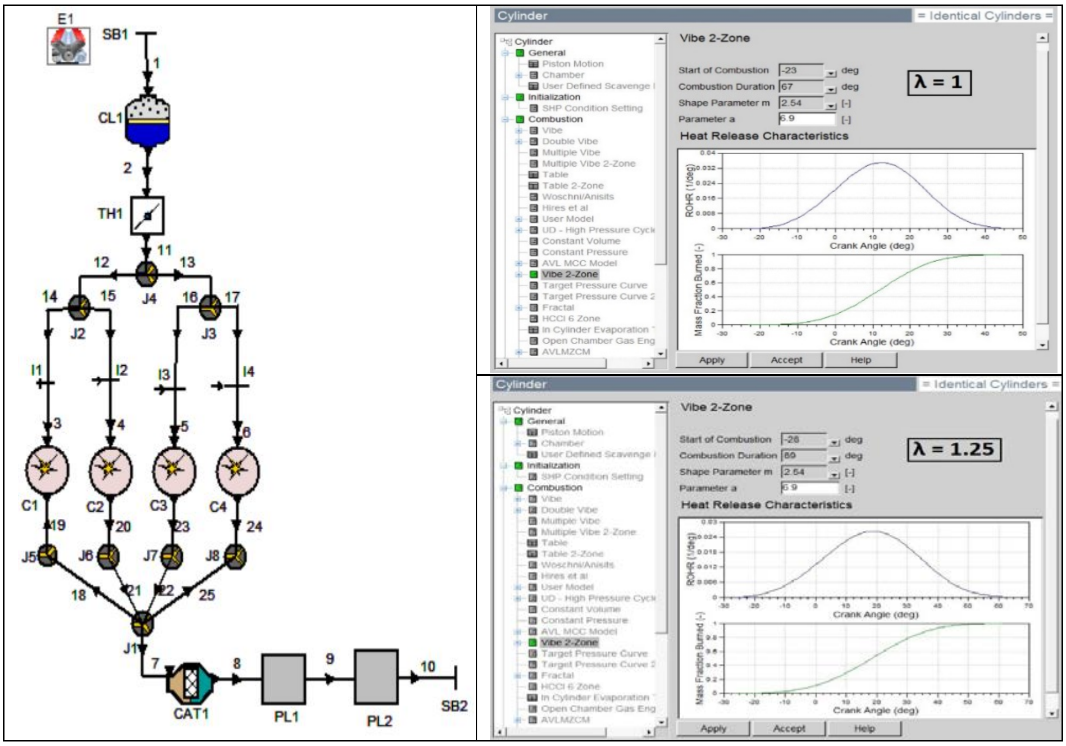


Fig. 7. K7M-710 engine symbolic model and ‘Vibe 2-Zone’ input data for optimum spark timings

In this simulation tool a set of equations are used to model the engine operation. Thus for:

- Fluid flow modelling according to [8]

The set of conservation equations to describe a one-dimensional pipe flow is given by the well-known ‘Euler Equation System’:

$$\frac{\partial U}{\partial t}+\frac{\partial F(U)}{\partial x}=S(U) \tag{1}$$

Where U represents the state vector:

$$U=\begin{pmatrix} \rho \\ \rho\cdot u \\ \rho\cdot\overline{c_v}\cdot T+\frac{1}{2}\cdot\rho\cdot u^2 \\ \rho\cdot w_j \end{pmatrix} \tag{2}$$

and F is the flux vector:

$$F=\begin{pmatrix} \rho\cdot u \\ \rho\cdot u^2+p \\ u\cdot(\mathrm{E}+p) \\ \rho\cdot u\cdot w_j \end{pmatrix} \tag{3}$$

$$E=\rho\cdot\overline{c_V}\cdot T+\frac{1}{2}\cdot\rho\cdot u^2 \tag{4}$$

In the above relations:

- ρ = density of the entire gas phase;
- u = mean mass weighed gas velocity;
- $\overline{c_v}$ = specific heat capacity of the gas (at constant volume);
- T = temperature;
- w_j = mass fraction of the species j in the gas phase;
- p = pressure;
- i = index of homogeneous chemical reactions;
- j = index of chemical species;
- S(U) = source term.

• Combustion ‘Vibe 2-Zone’ modelling according to [8]

‘Vibe 2-Zone’ considers the cylinder charge divided into two zones (one dedicated to the burned gas and the other dedicated to the unburned gas – the fresh mixture).

The first law of thermodynamics is applied for each zone:

$$\frac{dm_b\cdot u_b}{d\alpha}=-p_c\cdot\frac{dV_b}{d\alpha}+\frac{dQ_F}{d\alpha}-\sum\frac{dQ_{Wb}}{d\alpha}+h_u\cdot\frac{dm_b}{d\alpha}-h_{BB,b}\cdot\frac{dm_{BB,b}}{d\alpha} \tag{5}$$

$$\frac{dm_u\cdot u_u}{d\alpha}=-p_c\cdot\frac{dV_u}{d\alpha}-\sum\frac{dQ_{Wu}}{d\alpha}-h_u\cdot\frac{dm_b}{d\alpha}-h_{BB,u}\cdot\frac{dm_{BB,u}}{d\alpha} \tag{6}$$

- index ‘b’ = burned zone;
- index ‘u’ = unburned zone;
- m = mass;
- u = specific internal energy;
- p_c = in-cylinder pressure;
- V = volume;
- Q_F = fuel energy;
- Q_w = wall heat loss;
- α = crankshaft rotation angle;
- h_{BB} = enthalpy of blow-by;
- $\frac{dm_{BB}}{d\alpha}$ = blow-by mass flow;
- $h_u \cdot \frac{dm_b}{d\alpha}$ = enthalpy flow from the unburned to the burned zone due to the conversion of a fresh charge to combustion products.

Fig. 8 shows an image of the cylinder with two zones considered for the combustion model.

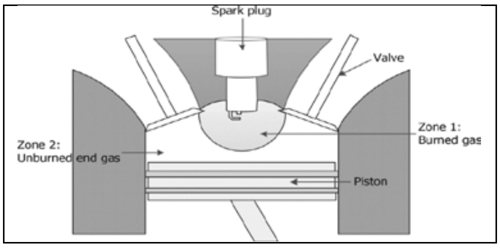


Fig. 8. The burned and unburned gas zones in ‘Vibe 2-Zone’ combustion model – [9]

The Wiebe function is commonly used to estimate the heat release characteristic due to its simplicity and versatility demonstrated by its forms and numerous applications. It is commonly used in internal combustion engine research as a singular, double or multiple function [10].

In [11] the authors present predictive simulations done by a 0-D single zone engine model of a SI engine fuelled by methane and methane-hydrogen blends. Single and double Wiebe function are used to model the combustion process. The results show better accuracy for the simulations performed with double Wiebe function.

In our case for the estimation of the heat release characteristic, the ‘Vibe 2-Zone’ combustion model uses single Wiebe function with the following expression rate of heat release:

$$\frac{dx}{d\alpha} = \frac{\alpha}{\Delta\alpha_c} \cdot (m+1) \cdot y^m \cdot e^{-a \cdot y^{m+1}} \tag{7}$$

$$dx = \frac{dQ}{Q} \tag{8}$$

$$y = \frac{\alpha - \alpha_0}{\Delta\alpha_c} \tag{9}$$

- α_0 = start of combustion;
- $\Delta\alpha_c$ = combustion duration;
- m = shape parameter;
- a = efficiency parameter, a=6.9 for complete combustion [8].

5. Simulation Results and comparison with experimental data

The ‘Vibe 2-Zone’ combustion model was used to fit the simulation results to the experimental operating data collected during tests. For this purpose, the combustion durations (CD) were considered based on pressure diagrams. It was used the value 2.54 for the shape parameter ‘m’ and the value 6.9 for the efficiency parameter ‘a’ (complete burn). In order to calibrate the calculation model using the experimental results, the start of combustion (SOC) for each spark timing was varied until optimal cases were found in terms of maximum pressure and effective power. The relative errors for the effective power and the maximum pressure between experimental data and simulation results were calculated and the optimal cases were considered where the minimum deviation values have been achieved.

Table 2 and 3 summarize the comparisons between experimental and simulation results.

Table 2

Comparison between experimental data and simulation results for λ=1

λ [-]	βs [°CA]	Experimental		AVL BOOST				Errors		
		EP [kW]	p_{max} [bar]	SOC [°CA]	CD [°CA]	EP [kW]	p_{max} [bar]	ε_{EP} [%]	ε_{pmax} [%]	ε_{rel} [%]
1	32	4.95	13.06	-16	69	4.93	13.43	0.40	2.83	2.86
				-15		4.86	13.03	1.82	-0.23	1.83
				-14		4.79	12.64	3.23	-3.22	4.56
	36	5.05	15.33	-20	66	5.11	15.77	1.19	2.87	3.11
				-19		5.07	15.32	0.40	-0.07	0.40
				-18		5.03	14.88	-0.40	-2.94	2.96
	40.12	5.17	16.81	-24	67	5.25	17.38	1.55	3.37	3.71
				-23		5.23	16.92	1.17	0.63	1.33
				-22		5.20	16.45	0.59	-2.16	2.24
	43.87	5.18	18.32	-27	67	5.38	18.97	-3.86	-3.55	5.24
				-26		5.38	18.49	-3.86	-0.93	3.97
				-25		5.37	18.01	-3.67	1.69	4.04
	49.12	5.17	19.92	-30	67	5.29	20.31	-2.32	-1.96	3.04
				-29		5.32	19.84	-2.90	0.40	2.93
				-28		5.33	19.37	-3.09	2.76	4.15
	52.12	5.05	21.34	-34	68	5.19	21.9	-2.77	-2.62	3.82
				-33		5.23	21.44	-3.56	-0.47	3.60
				-32		5.26	20.98	-4.16	1.69	4.49

Table 3

Comparison between experimental data and simulation results for λ=1.25

λ [-]	βs [°CA]	Experimental EP [kW]	Experimental p_{max} [bar]	AVL BOOST SOC [°CA]	AVL BOOST CD [°CA]	AVL BOOST EP [kW]	AVL BOOST p_{max} [bar]	Errors ε_{EP} [%]	Errors ε_{pmax} [%]	Errors ε_{rel} [%]
1.25	32	3.53	10.07	-18	93	3.78	9.81	-7.08	-2.58	7.54
				-17		3.69	9.51	-4.53	-5.56	7.17
				-16		3.59	9.21	-1.70	-8.54	8.71
	36	4.03	11.19	-21	90	4.22	11.34	4.71	1.34	4.90
				-20		4.14	10.99	2.73	-1.79	3.26
				-19		4.05	10.65	0.50	-4.83	4.85
	40.12	4.47	12.35	-24	88	4.57	12.88	2.24	4.29	4.84
				-23		4.50	12.50	0.67	1.21	1.39
				-22		4.43	12.12	-0.89	-1.86	2.07
	43.87	4.70	13.52	-27	86	4.62	14.28	1.70	-5.62	5.87
				-26		4.57	13.87	2.77	-2.59	3.79
				-25		4.51	13.47	4.04	0.37	4.06
	49.12	4.74	14.53	-29	89	4.88	14.78	-2.95	-1.72	3.42
				-28		4.83	14.36	-1.90	1.17	2.23
				-27		4.78	13.95	-0.84	3.99	4.08
	52.12	4.72	15.18	-30	87	4.92	15.59	-4.24	-2.70	5.02
				-29		4.88	15.16	-3.39	0.13	3.39
				-28		4.83	14.73	-2.33	2.96	3.77

In Fig. 9 are represented the heat release characteristics corresponding to the optimal spark timings for the two air-fuel ratios (λ=1 and λ=1.25) determined based on the experimental pressure diagrams and calculated by simulation with AVL BOOST.

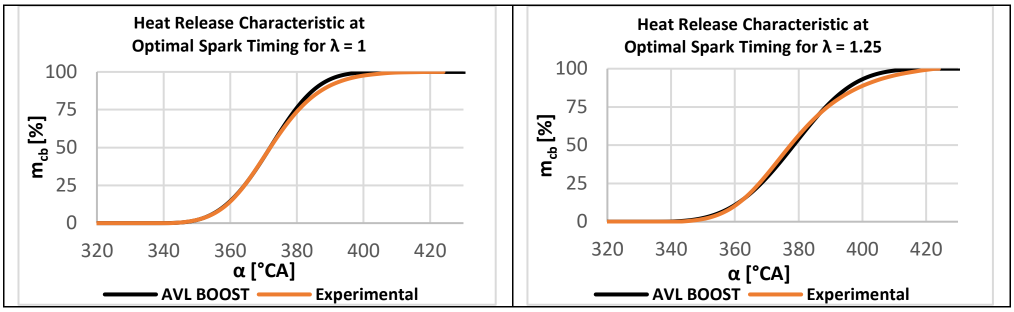


Fig. 9. Heat release characteristics corresponding to the optimal spark timings for λ=1 and λ=1.25

It can be noticed that there is good agreement between the experimental and simulation results for the heat release characteristics at optimal spark timings.

In Fig. 10 and Fig. 11 are offered the pressure traces variation in p-α and in p-V diagrams for the experimentally determined data (mean values for 500 consecutive cycles) and for the simulation results calculated using the AVL BOOST software for the optimal spark timings. It is noticeable that the matches between them are good enough but with some relative deviation for the lean mixture.

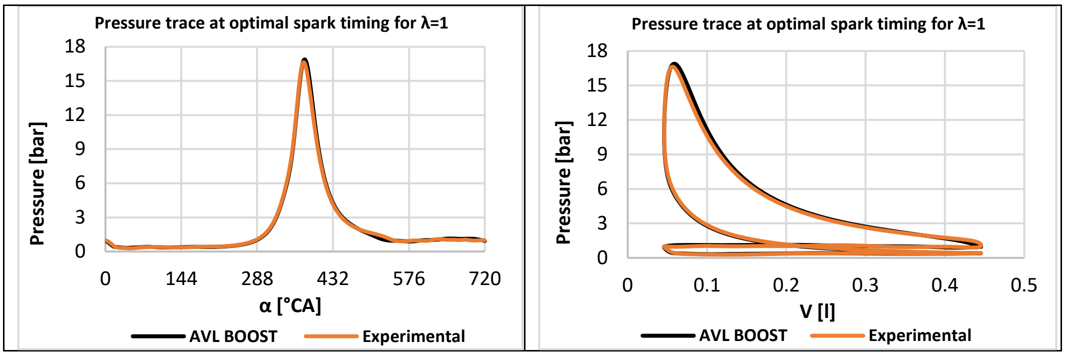


Fig. 10. Measured and calculated pressure charts for optimal spark timing at λ=1

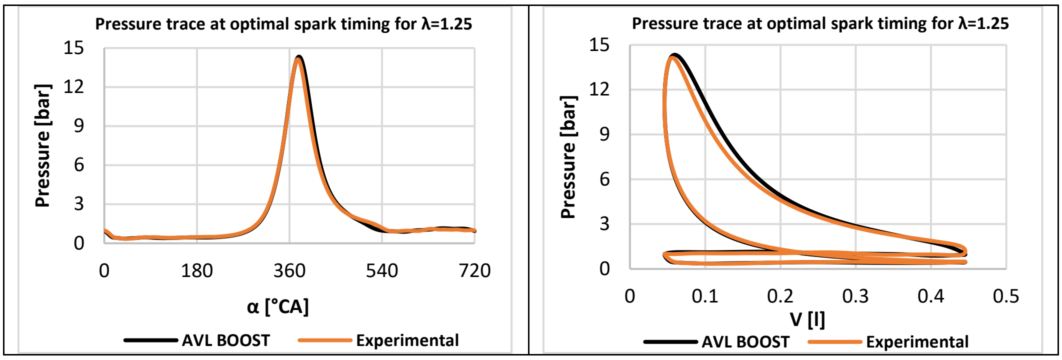


Fig. 11. Measured and calculated pressure traces for optimal spark timing at λ=1.25

6. Conclusions

The study showed that the zero-dimensional combustion models using the Wiebe heat release characteristics as 'Vibe 2-Zone' can accurately simulate the behaviour of a spark ignition engine at various operating conditions.

In this work, which is a new study for Renault K7M-710 engine, a two-zone combustion model developed in the AVL BOOST software as 0-D with single Wiebe function was used to simulate the performance parameters of this engine operating at constant speed 2000 rpm and 2 bar brake mean effective pressure (constant load) and different spark timings for two air-fuel ratios, stoichiometric mixture λ=1, and respectively lean mixture λ=1.25.

A widespread comparison was done between the experimental data recorded on the test bench research and the results obtained by theoretical investigation using simulation models. This comparison emphasized that by using the 'Vibe 2-Zone' combustion model the engine performance and efficiency were well fitted (relative deviations under 4% at λ=1 and 5% at λ=1.25 for effective power and relative deviations under 3% at λ=1 and 4% at λ=1.25 for brake specific fuel consumption). The peak fire pressures were also satisfactory estimated achieving relative deviations under 1% for λ=1 and under 6% for λ=1.25.

Correspondingly, it must be noticed that by such simple models can be obtained estimations of performance, efficiency and emissions parameters, determined by the evolution of the in-cylinder charge during the engine operating cycle, for different engine operating conditions and different fuelling strategies associated to alternative fuels usage. This will be a future step in the progress of our work where a refined fractal combustion model will be used.

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