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ABSTRACT

The performances of high pressure fuel-injection systems and their effects on diesel engine combustion are strongly influenced by the injector characteristics and the set up of the whole equipment control system. High-pressure system based on the common-rail architecture allows a multi-stage injection, which is of paramount importance in controlling combustion noise, fuel consumption, operation roughness and exhaust pollutant emissions.
Common rail fuel injection equipment for automotive diesel engine, together with its control system have been analysed by using AMESim environment; both standard library elements and self-developed sub-models have been adopted.
At first the different components have been considered one by one; in this way the behaviour of high pressure pump (radial-jet), pressure regulator, rail, injectors, system control (e. c. u.) has been investigated; the results have been compared with experimental measurements. Then all components have been collected and a simulation tool has been set up, able to perform the parametric analysis of the geometry of some components. The performance of the complete system has been investigated and the response of the system to a typical driving cycle has been evaluated.

INTRODUCTION

Electronically controlled high pressure fuel injection system holds an important role concerning both the emission control strategy and the improvement of internal combustion engine performance [1, 2, 3].
High pressure injection allows to finely atomize the fuel spray and to promote fuel and air mixing, resulting in significant combustion improvements [4, 5, 6].
Following the previous considerations, the prediction of common rail equipment behaviour is of practical relevance in design concerning automotive engine.
Realistic investigation can not set aside the analysis of the complete system, in both its hydraulic components (injector, rail and pressure control valve) and control device.
At present different kinds of numerical approaches are adopted to model and simulate the injection system [7].
The geometrical complexity of the system, together with the unfavourable surface to volume ratio and the high impulsive feature that characterizes the phenomenon, give rise to a not very profitable simulation when CFD codes are employed.
Moreover, uncertainty on the real small-scale behaviour of the fluid and on impulsive compression and expansion cycles exists, and experimental data are not easily available.
Phenomenological models, based on simple schemes, such as lumped parameters or one-dimensional models [8, 9, 10], seem to present the best ratio between benefits and computational requirements since, in author’s opinion, they are able to catch the fundamental aspects of the phenomenology, taking full advantage of the experimental measurements that, usually, are expressed by global quantities.
Although self-developed codes are available, the authors have privileged the use of a commercial code, since it is able to guarantee a tested architecture in which new original components can be included, allowing the optimisation of the computational resource, by focusing only on fundamental and critical elements.
The complete common rail fuel injection equipment of an automotive diesel engine together with its control system, have been modelled by means of AMESim environment [11]; each component has been modelled by using both standard and new sub-models.
Since the control system has to guarantee the optimal setting of time, quantity and quality of the injection for any changing of engine speed and load, the model response to a typical driving cycle has been investigated, under transient and steady state conditions.
In the procedure devoted to select the most effective variables able to optimise the injection process, a parametric analysis of the geometry of injector (control volume, inlet and outlet hole section, injector nozzle), pressure control valve, rail, high pressure pump, is required. The developed model is well suitable to evaluate the effect of both geometrical and operational variations on the complete system.

SYSTEM MODELLING

The simulation of the common rail fuel injection equipment and of its control system has been carried out in AMESim (Adaptive Modelling Environment for Simulation) environment. The code is a design package which allows the investigation of single components and/or complete systems.

The investigated common-rail system, depicted in figure 1, has been divided into components and each one has been investigated by means of sub-models belonging to existing libraries or self-developed. By creating new components and by modifying the mathematical model which underlies existing ones, the overall system has been customized. All ducts connecting the different submodels have been analysed by means of a lumped parameter scheme, accounting for the compressibility and the pipe friction.

Figure 1 –Common-rail system.

A simplification of the complete scheme has been obtained by using the super component facility that allows to collect more sub-models into a single one.

The differential and algebraic equations describing the behaviour of the complete system have been solved by using Gear's and Adams-Multon's methods; the code selects automatically the most appropriate algorithm based on the system stiffness.

The sketch of the investigated common-rail system in AMESim environment is shown in figure 2.

From figure 2 some components can be identified: the control system (e. c. u.), the power unit (e. p. u.), the pressure control valve, the rail, the high pressure pump, the injectors.

The control system processes the engine operative conditions and, according to the speed and load values, it sets the optimal time, quantity and quality of the injection, aimed at controlling combustion noise, fuel consumption, operation roughness and exhaust pollutant emissions.

Such a control system, shown in figure 3, consists of different sections: the main injection control (m. i. c.), the pilot injection control (p. i. c.), the engine mapping (e. m.) and the pressure rail control block (p. c. b.).

The m. i. c. realizes, in standard conditions, an injection of 1 ms timing across the TDC. Since the engine operative conditions vary, such a block produces different lengths of both injection period and phase.

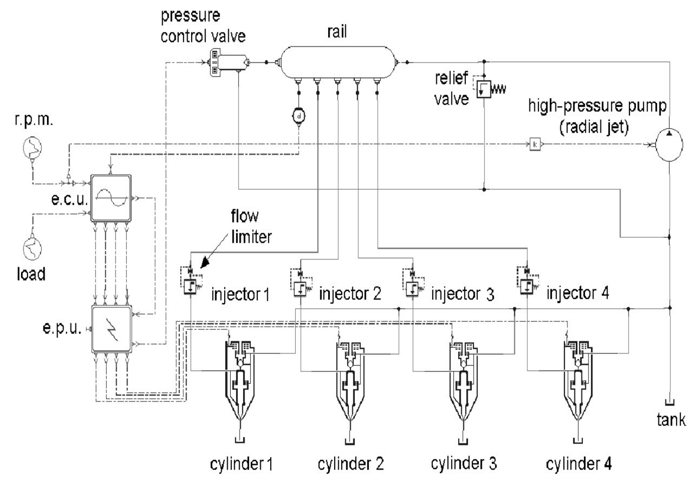

Figure 2 – Scheme of the complete system in AMESim environment.

The p. i. c. acts in the same way of the m. i. c. Due to the reduction of the positive effect of the pilot injection as the engine running increases, the p. i. c. block realizes a pilot injection characterized by a shorter length and a greater advance as regards to the main one, according to the engine speed.

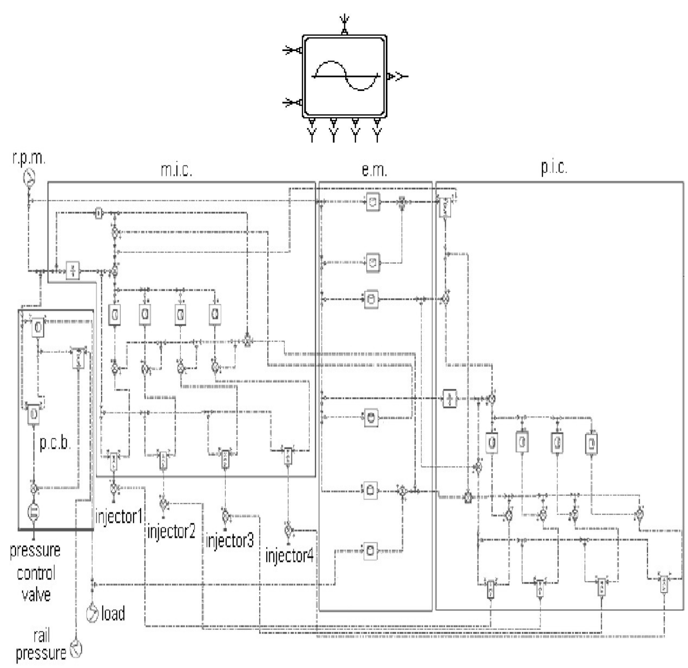

Figure 3 – Scheme of the super-component representing the control system.

The pressure control valve, shown in figure 4, acts on the pressure regulator in order to modulate the rail pressure according to the engine operative conditions; its operation is controlled by the e. c. u..

Such a component is composed of basic (spring, inertia and ball poppet with conical seat) and self-developed elements (solenoid).

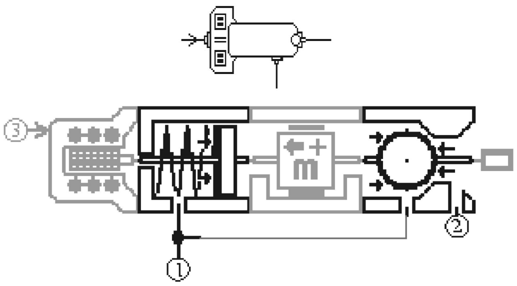

Figure 4 – Pressure control valve super-component.

The sub-model of the high pressure tank (rail) has been developed, aimed at taking into account for the damping of the flow fluctuations mainly due to the high pressure pump, but also to the periodic injector opening and to the pressure regulator valve operation. In order to realize the control system feedback, the capacity element has been connected to the pressure sensor.

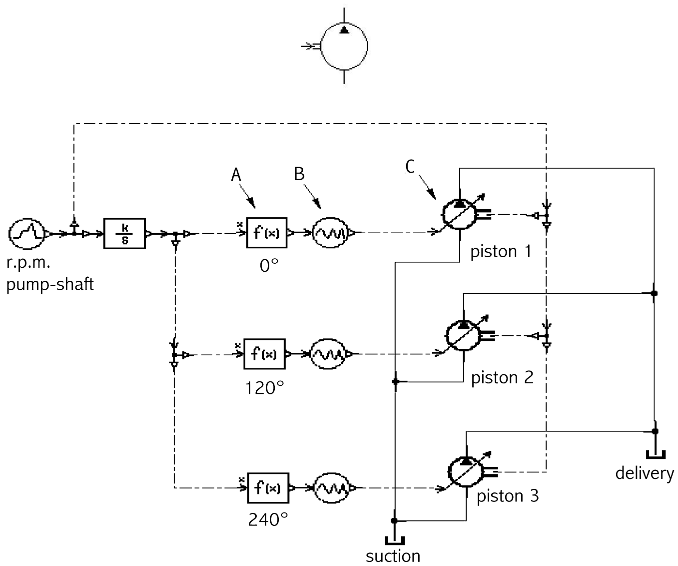

Figure 5 –Radial jet super-component.

The pump is a radial three piston pump able to guarantee high pressure and low flow fluctuations.
The operation of each piston has been simulated by means of three elements representing the phase angle (A) among the three pistons, the piston movement (B) and the proper pumping element (C). In figure 5 all components are shown.
The injector has been modelled by using two different mechanical components, the electro-valve block, controlled by the e. p. u., and the nozzle block. An hydraulic volume with a fixed orifice (A hole) allows their connection, as shown in figure 6. The former block is composed of basic (spring, inertia and ball poppet with conical seat) and self-developed elements (solenoid); in the latter one a spring, an inertia, a flapper nozzle valve and a conical poppet valve are included. The ball poppet of the electro-valve block is connected to the flapper of the nozzle block through the A hole and the hydraulic control volume (c. v.). The nozzle block is connected to the rail by means of an hydraulic pipe. Such a pipe supplies both the conical poppet valve and, by means of the Z hole, the flapper. The conical poppet valve is connected to an hydraulic volume (sac) supplying the cylinder by means of the nozzle holes.
The behaviour of all hydraulic volumes has been simulated by means of developed submodels.

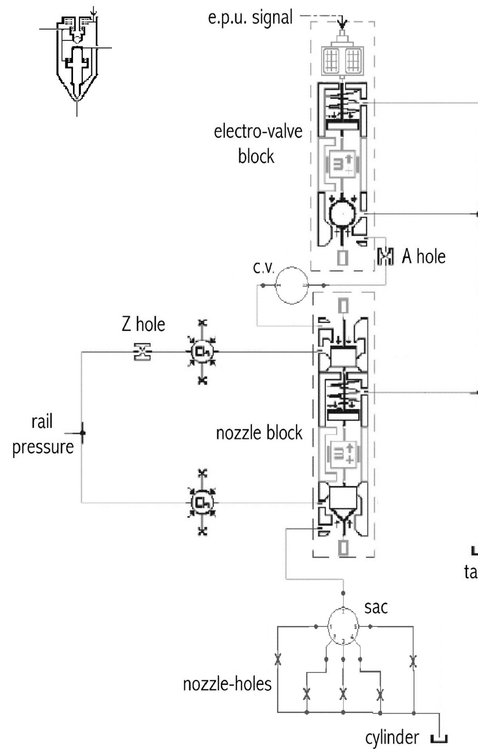

Figure 6 – Injector super-component.

SIMULATION AND RESULTS

The fuel properties have been set according to ISO 4113.
At first the behaviour of each developed component has been investigated and some results for key components (the rail, the pressure control valve and the injector) are herewith presented.

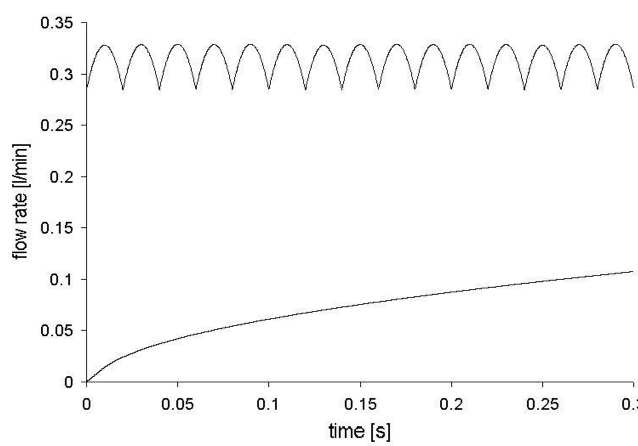

Figure 7 – Rail inlet (upper trace) and outlet flow rate (lower trace).

Figure 7 shows the rail inlet and outlet flow rate traces. The graph has been obtained by considering a simple

system, composed of the pump and the rail; the injector has been substituted by a fixed orifice.
In the lower trace trend, the time constant strictly depends on the rail geometrical characteristics; any modification of such parameters affects the rail dumping effect.
A pressure control valve has been added to such a simple circuit, together with the p. c. b. system. In figure 8 the effect of different engine speeds and loads on rail pressure is highlighted.

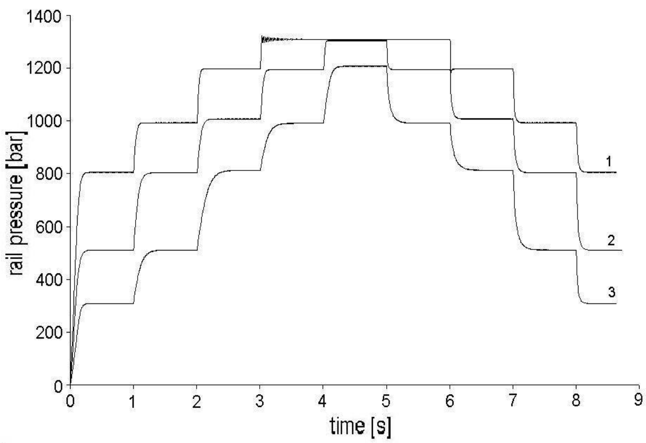

Figure 8 – Rail pressure for different engine speeds: trace 1 refers to 3000 rpm, trace 2 refers to 2000 rpm and trace 3 refers to 1000 rpm.

In the simulations the load conditions have been varied according to an increased and then decreased trend composed of 5 steps (0%, 25%, 50%, 75% and full load). As the load increases, the rail pressure values become higher, exhibiting a little delay in following the optimal rail pressure value.

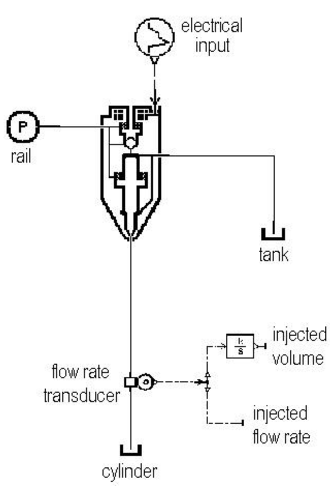

Figure 9 – Scheme of circuit used to investigate the injector behaviour.

Aimed at simulating the injector behaviour, the circuit shown in figure 9 has been realized, in which the rail and a volume, representing the cylinder, are included. Some tests have been carried out and in figures 10–15 the obtained results are compared with the experimental measurements.

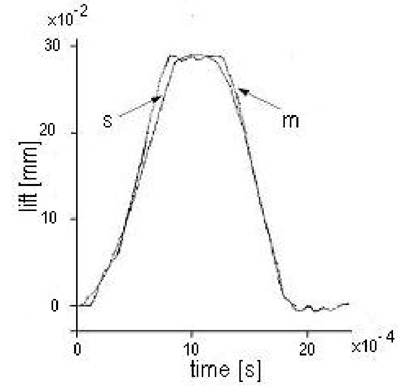

Figure 10 – Nozzle lift traces (800 bar, 1 ms); s stands for simulated, m for measured.

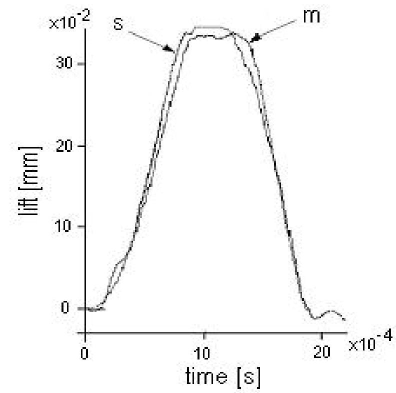

Figure 11 – Nozzle lift traces (1200 bar, 1 ms); s stands for simulated, m for measured.

The simulations have been performed by considering one injection of 1 ms length, and by imposing different rail pressure values (800 bar and 1200 bar). Figures 10 and 11 show the injector lift histories.
In figures 12, 13 and 14, 15 the injected flow rates and the injected volume are respectively depicted. All graphs show the close agreement between the predictions and the experimental values. It is necessary to specify that, at lower pressure, a small delay in the simulated time of injection end (elastic release) has been found, as regards the measured one. Such effect is due to the non direct proportionality existing between stress and deformation in empirical data, whereas the AMESim model refers to Hook law.

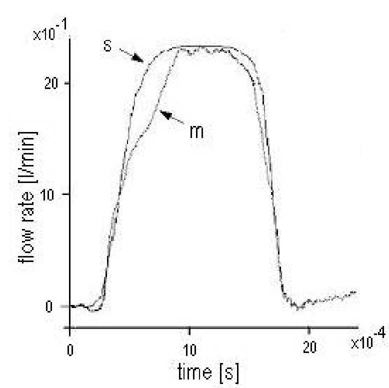

Figure 12 – Injected flow rate traces (800 bar, 1 ms); s stands for simulated, m for measured.

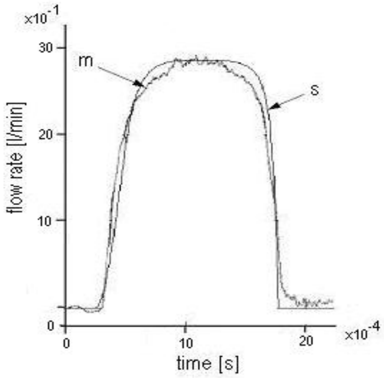

Figure 13 – Injected flow rate traces (1200 bar, 1 ms); s stands for simulated, m for measured.

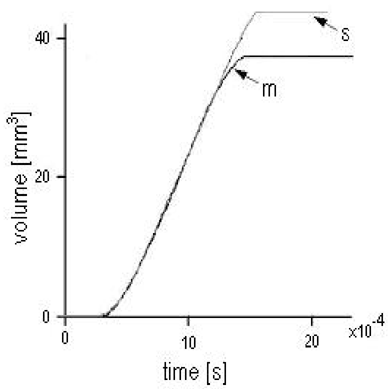

Figure 14 – Injected volume traces (800 bar, 1 ms); s stands for simulated, m for measured.

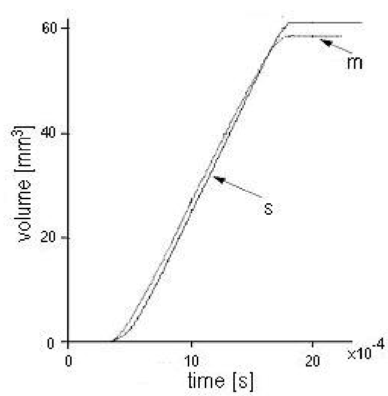

Figure 15 – Injected volume traces (1200 bar, 1 ms); s stands for simulated, m for measured.

As consequence of the delay, a greater volume is injected in the simulated case, especially at lower pressure, as shown in figures 14 and 15.
In order to analyse the complete common rail system, all components have been collected, and the scheme of figure 2 has been realized.
The control system behaviour has been investigated, and the effect of different parameters has been analysed: ET (energizing time of main injection), ET_{pil} (energizing time of pilot injection), DT_{pil} (dwell time of pilot injection). The reduction of the combustion noise, especially at low engine speed, requires both high ET_{pil} and DT_{pil} values. Since the positive effect of the pilot injection on the combustion noise reduces itself as the rpm increases, a decrease of the ET_{pil} value, together with an increase of the ET_{pil} value must occur, until the pilot injection becomes unnecessary when higher engine speed are reached.
A submodel for the e. c. u. has been developed, able to realize, in a satisfactory way, the ready variation of the injection control parameters according to any change of engine speed and load.
Aimed at highlighting the capability of the complete model to simulate the response of the electro-hydraulic system to a generic set of control values, the effect of a driving cycle, whose main characteristics are reported in table 1, has been investigated.

	engine speed (rpm)	load
1	1000	25%
2	1000	50%
3	2000	25%
4	2000	75%
5	3000	25%
6	3000	100%

Table 1 – Driving cycle main characteristics.

In table 2 some results of the simulation are shown, characterizing both the control system and the rail outputs.

	ET (ms)	ET_{pil} (ms)	DT_{pil} (ms)	rail pressure (bar)
1	0.50	0.12	2.4	500
2	0.60	0.1	2.3	800
3	0.50	0.06	3.15	800
4	0.65	0.045	3.15	1200
5	0.45	/	/	1000
6	0.58	/	/	1350

Table 2 – E. c. u. parameters and rail pressure.

It can be highlighted how the pilot injection retains less importance as the rpm increases. The rail pressure trend is close to that one of figure 8; the required value for the optimal injection is reached with a longer delay, due to the presence of all components in the circuit and to the flow rate tapped by the injectors.
In figures 16-18 the injected flow rate time histories are shown. Each trend refers to a particular couple of speed and load (table 1).

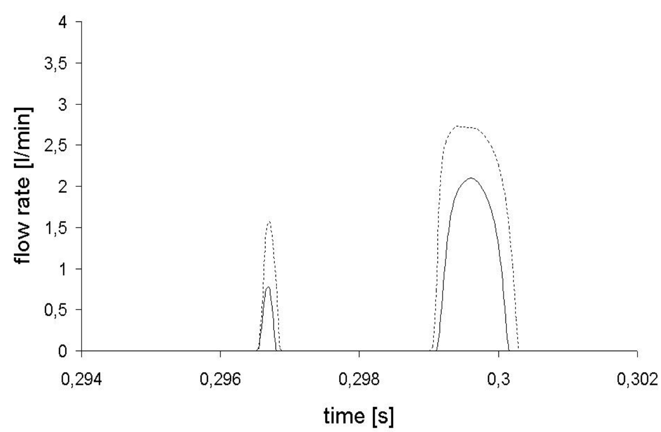

Figure 16 – Injected flow rate; 1000 rpm: 25% load ——, 50% load -----.

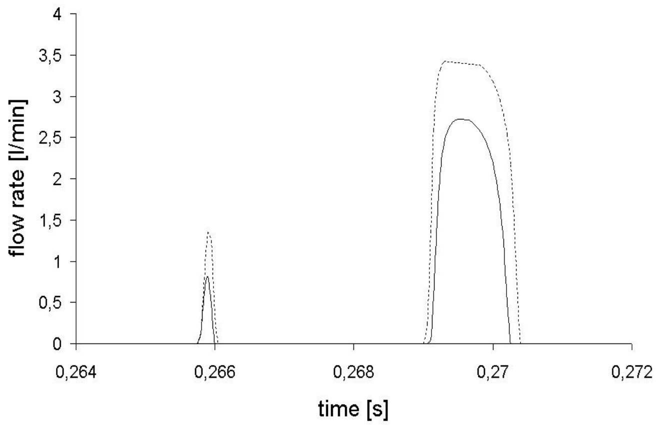

Figure 17 – Injected flow rate; 2000 rpm: 25% load ——, 75% load -----.

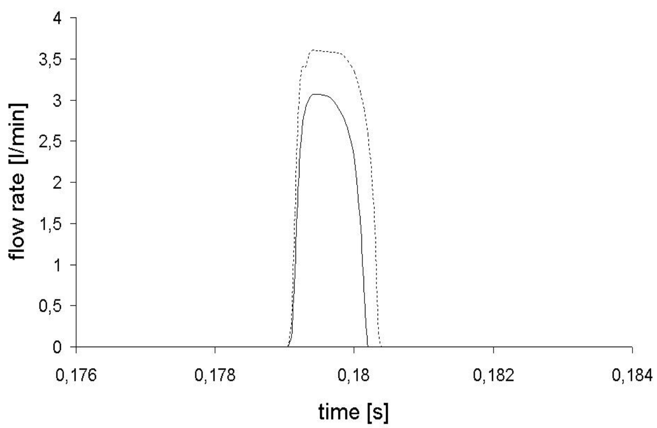

Figure 18 – Injected flow rate; 3000 rpm: 25% load ——, 100% load -----.

In order to summarize and highlight the main aspects of such figures, the table 3 is shown. Each line refers to one figure, and the length of both main and pilot injections, together with the injected mass and the pick value of the flow rate are reported.

	injection length (ms)		flow rate pick value (l/min)		injected mass (mg)	
	main	pilot	main	pilot	main	pilot
1	1.1	0.3	2.1	0.77	22.1	1.7
2	1.3	0.4	2.7	1.5	35.7	3.65
3	1.2	0.2	2.73	0.81	34	1.36
4	1.35	0.25	3.4	1.3	44.2	2.72
5	1.15	/	3.07	/	37.4	/
6	1.4	/	3.6	/	51.85	/

Table 3 – Results of the main and pilot injections.

By comparing these results with those of table 2, in which the parameters imposed by the control unit (e. c. u.) are shown, some differences arise. Such differences are due to the delay introduced by both the electro-valve block and the nozzle block in the injection stroke. Among them, the difference between energizing time and actual injection time retains particular importance in the setting of e. c. u. parameters.

From the table, a very important parameter can be obtained, consisting in the ratio between the mass injected by the main injector and that one due to the pilot injector, during the different operative conditions. As required, such a ratio is characterized by a decreased trend as the engine speed increases (under the same load), and by an increased trend as the load increases (under the same engine speed).

Figures 19-21 show the time histories of the control volume pressure (c. v. of figure 6). Different values for the engine speed and the load have been fixed, according to table 1.

Such trends are strictly connected with dynamic of opening and closing of both the ball poppet valve and the nozzle flapper; the capacity of the pipe and of the c.v retains an important rule, too.

The pressure curves are quite similar since are characterized by the presence of two zones of high variation, due to the pilot and main injections. During such injections, the pressure trend in the control volume highlights the delay with which the flapper movement follows that one of the ball.

The results of the analysis of some details of the common rail system point out the effect of the geometry of the control volumes, the holes and the supplying ducts on the fluid dynamic quantities characterizing the behaviour of the different components.

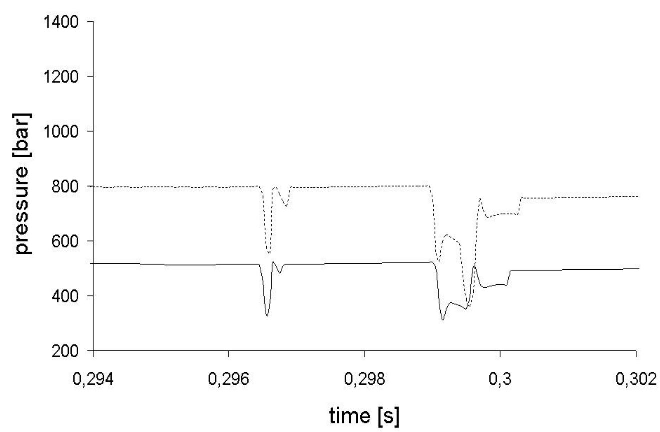

Figure 19 – Control volume pressure; 1000 rpm: 25% load ——, 50% load -----.

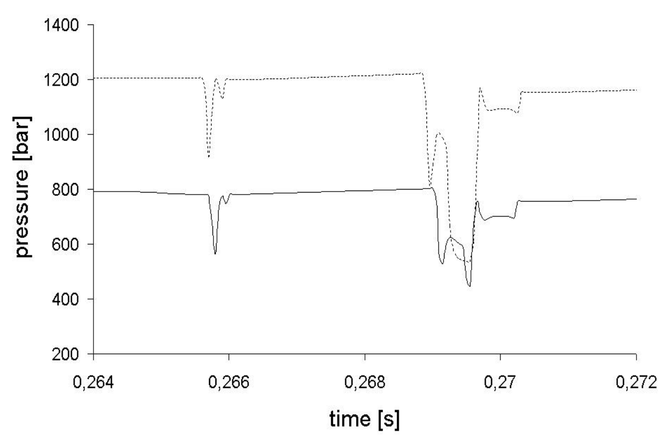

Figure 20 – Control volume pressure; 2000 rpm: 25% load ——, 75% load -----.

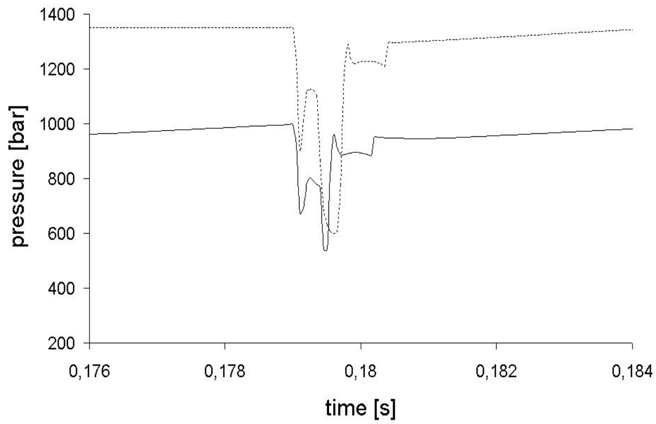

Figure 21 – Control volume pressure; 3000 rpm: 25% load ——, 100% load -----.

CONCLUSIONS

In this paper, the performance of a high pressure fuel injection system of an automotive diesel engine has been investigated. Since a realistic analysis can not set aside the simulation of the complete system, both the hydraulic components (injector, rail and pressure control valve) and the control device have been considered.

The AMESim environment has been used; each component has been modelled by means of both standard library elements and self-developed submodels. Such a code retains the feature to guarantee a tested architecture, in this way the attention has been focused on the development of the most critical and fundamental components.

At first the submodels, representing the different elements, have been described, and the results of some preliminary tests have been presented, together with the comparison with experimental measurements.

Then all components have been collected and the complete system has been analysed. The response of the model to a typical driving cycle has been investigated and the results have been presented, highlighting the potentiality of the developed model to realize a powerful simulation tool, able to perform a parametric analysis of some components, aimed at selecting the most effective variables to guarantee the optimisation of the injection process, and to make the engine set up easier.

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