A Control-Oriented Model of a Common Rail Injection System for Diesel Engines

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Abstract

This paper presents a model of a Common Rail injection system for diesel engines. The model is derived by considering the components of the system as control volumes and applying elementary fluid dynamics and mechanics laws. Suitable simplifications are introduced, to make the model adequate for control purposes, trading off between computational effort and accuracy. The model obtained is a fifth order nonlinear one, in state-space representation, and relies on simple, well defined, geometric parameters of the system. The results obtained are compared with data collected on an experimental setup, and with those obtained through the fluid dynamic simulation AMESIM®.

1. Introduction

Using the Common Rail injection system allows Diesel engines to improve their performance and reduce pollutant emissions, noise and fuel consumption [13]. The injected fuel amount, however, must be precisely metered by the regulation of the injection pressure and timings, in order to get the proper air/fuel mixture demanded by the engine speed and load. Obviously, a mathematical model of the injection system is necessary to design a control strategy for the rail pressure.

Simulation models help in simplifying the controllers design process, and in saving time as it is possible to tune its parameters directly from model equations. However, high non-linearities due to complex fluid-dynamic phenomena in the injection system make it difficult to build an accurate model which is valid in every working condition. On the other side, a model which accurately describes the system dynamics could be not appropriate for designing control laws. Basically, a control-oriented model has to simplify the control strategy design process, but it must still guarantee a good prediction of dynamics of the most effective variables in the injection process.

Many diesel injection system models have been previously proposed, based on the equations of the physics underlying the process or alternatively developed through simulation packages. The form of these models is suitable for mechanical design but is often too complex to be appropriate for control purposes. For example in [1] and [8] an accurate model considers the fuel injection system of diesel engines with a distributor-type pump and properly represents the unsteady flow in the pipeline connecting the jerk pump and the injector. Then the resultant hyperbolic equations are solved by using an explicit scheme of the predictor-corrector type. Ficarella *et al.* in [5] consider the same complex fluid-dynamic phenomena and study the instability in a Common Rail Injection system for high speed diesel engines. The good accuracy of all these models is counterpoised from the complexity of the resulting equations.

Simulation models are also applied to the study of fuel injection systems. For example [2] and [10] simulate the common rail injection system using the AMESIM (Advanced Modeling Environment for Simulation) environment [6], a design package which allows the investigation of systems by means of models of single elements belongings to existing or self-made component libraries; for each component a mathematical model is provided, including the effect of frictions and leakages. The aim of models of this kind is to characterize system dynamics, to perform components parametric analysis and geometric optimization, to evaluate and validate the effects of operational conditions on the injection process. Nevertheless, despite of good prediction capabilities, they are not appropriate for control as they do not provide any mathematical representation of the process dynamics.

Finally, different classes of models, both linear and nonlinear, can also be generated by identification based on experimental data. Automotive applications of this approach regard mainly the whole engine modeling, while it is still not extensively used to model the injection system alone. For example, [3] presents an integrated approach to identify a NARX (Nonlinear AutoRegressive Exogenous) model of an internal combustion engine and to design a nonlinear idle speed regulator. Recent results, see [9], show that nonlinear identification is a promising approach to model the injection system too.

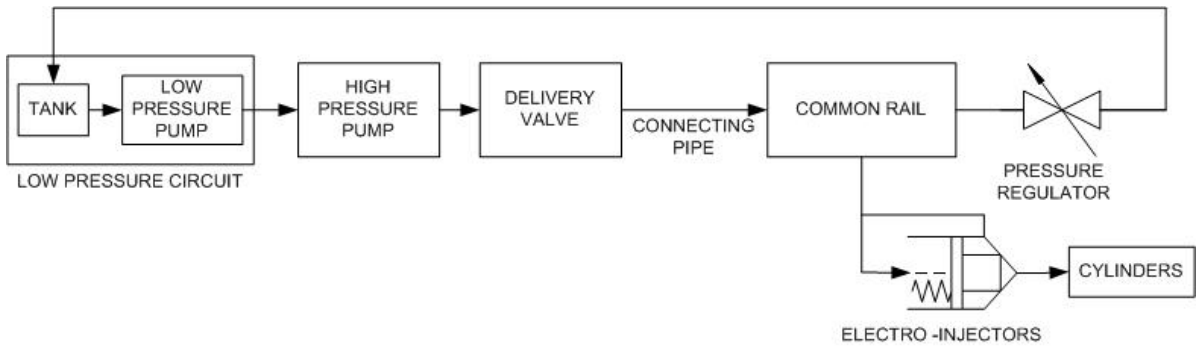

Figure 1: A block scheme of the Common Rail injection system for diesel engines.

In this paper we propose a model of a Diesel Common Rail injection system, suitable for control purposes. It does not require a high computational effort as it is based on a lumped parameters representation, and can be used to design a controller for the rail pressure. The model can also be adapted to different injection systems with the same architecture, only by setting adequate geometrical parameters. Finally, it is accurate enough to predict the system behavior to test the controller performances.

2. The Common Rail injection system

The main elements of a Common Rail Diesel injection system are a low pressure circuit, including the tank and a low pressure pump, a high pressure pump, including a delivery valve, a common rail and the electro-injectors (Fig. 1) [2],[5], [13].

The injection system operates according to the following steps. The low pressure pump sends the fuel coming from the tank to the high pressure pump, which raises its pressure depending on the engine speed. When the pump pressure exceeds a threshold, the delivery valve opens so that the fuel can reach the common rail, which supplies the electro-injectors. An electro hydraulic valve placed on the common rail and driven by the Electronic Control Unit (ECU), drains a proper fuel amount to regulate the fuel pressure to a reference value.

The high pressure pump is a reciprocating pump with radial pistons driven by the camshaft. Piston motion is due to the eccentric profile of the camshaft (Fig. 2). The pump is connected to the low pressure circuit through a small orifice and to the high pressure circuit through a delivery valve with a shutter pushed over a conical seat by a preloaded spring. When the piston of the pump is at the lower dead centre the intake orifice is open and the fuel fills the cylinder, while the forces acting on the shutter of the downstream delivery valve make it to close. Then, due to the camshaft rotation, the intake orifice closes and establishes the compression of the fuel inside the pump chamber. When the resultant of valve and pump pressures overcomes a threshold which depends on the spring preload and stiffness, the fuel flows from the pump to the delivery valve and then to the common rail.

The common rail is a steel pipe providing high resistance to high working pressures. Its volume is designed large enough to appropriately damp the pressure oscillations due to pump and injectors operations. However, it still guarantees a fast response during transients and the starting phase, when it is still empty.

The regulator, i.e. the electro-hydraulic valve, encompasses an electromagnet with a moving anchor, and a spherical shutter, integral with the anchor. In a non-energized condition, the shutter and the anchor are held closed by a preloaded spring against the hydraulic force, and block the fuel flow towards the low pressure circuit. The spring preload is chosen to let the valve open when the rail pressure overcomes a fixed safety threshold. When the electromagnetic circuit is activated the magnetic force overcomes the spring preload. Hence the anchor and the poles come together and the pressure force shuttles the sphere to open the drain port. When the solenoid circuit is turned-off, the anchor is forced down and the shutter is pushed against the seat. In this way, the rail pressure is regulated by varying the duty cycle of the input voltage, making the valve to be partially opened. The regulator acts only during a specific camshaft angular interval, and its action is synchronized with the pump motion.

The electro-injectors can be considered as valves driven by the ECU through an electro-magnetic circuit [2], [4]. When the electro-magnetic circuit is excited the injector opens, allowing the injection of the fuel in the cylinder. The *Energizing Time* (ET) depends on the fuel amount to be injected. In the system under study, a complete injection cycle takes place in a 360° camshaft angular interval and consists of two injections starting every 180°.

3. The injection system model

To model the injection system each of the main elements, excluding the low pressure circuit and cylinders are represented as control volumes, in which

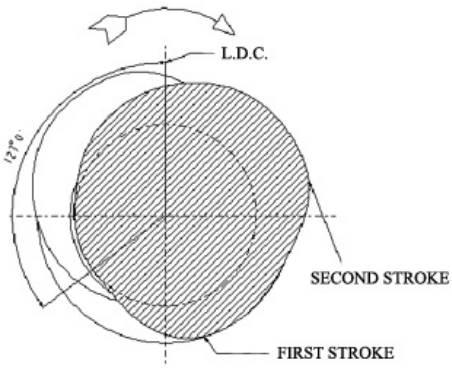

Figure 2. Camshaft section profile [4].

the fuel pressure dynamics can be calculated by suitably combining the continuity equation, the momentum equation and Newton's motion law. In our model we assume that the fuel temperature is constant during operations, consequently the system state can be represented by the pressure in each control volume. The low pressure pump and the cylinders are considered as infinite volumes of constant pressure. We also neglect fluid dynamic phenomena connected to flows through pipes. In particular, the connecting pipe between delivery valve and common rail is short and therefore introduces a negligible delay in the pressure disturbances transmission. Hence the volume of this pipe can be included in that of the delivery valve. Moreover, the phenomena are further simplified by considering a stiff pipe, thus neglecting the pressure variations related to the pressure wave propagation. On the other hand, to consider deformable pipes, an extremely accurate representation of fluid dynamic phenomena due to valve, to injectors opening and closing transients and to interruptions of pump delivery is necessary. However, this analysis is much more complex and, in the end, not suitable for control purposes. Finally, the electro hydraulic valve is considered just as a variable section, neglecting the shutter inertia.

In our model we consider compressibility of the fuel be expressed by the bulk modulus of elasticity [12]:

$$K_f = -\frac{dP}{dV/V} = \frac{dP}{d\rho/\rho} \tag{1}$$

where $-dV$ is the volume decrease of a unit volume of liquid due to a dP pressure increase. Since dV/V is dimensionless, the units of K_f and P are the same. In normal operating conditions K_f is set equal to 12000 bar and can be related to the fuel pressure P [bar] by the following expression [7]:

$$K_f = 1.2 \cdot 10^4 \cdot \left[1 + 0.6\frac{P}{600}\right] \tag{2}$$

From equation (1) it is possible to obtain the time derivative of the fuel pressure:

$$\frac{dP}{dt} = -\frac{K_f}{V} \cdot \frac{dV}{dt} \tag{3}$$

where V is the chamber instantaneous volume, dV/dt takes into account the intake and the outtake flows Q_{in} and Q_{out}, while the term dV/dt expresses volume changes due to mechanical parts motion. Equation (3) can also be rewritten as:

$$\frac{dP}{dt} = -\frac{K_f}{V} \cdot \left(\frac{dV}{dt} - Q_{in} + Q_{out}\right) \tag{4}$$

that gives, by integration, the pressure dynamics in each control volume. All the elements in the injection system, except the high pressure pump, have a constant volume V, therefore the derivative term at the second member of (4) can be cancelled.

Intake and outtake flows Q_{in} and Q_{out} can be calculated by applying the energy conservation law, which gives:

$$Q = \mathrm{sgn}(\Delta P) \cdot c_d \cdot A_0 \cdot \sqrt{\frac{2|\Delta P|}{\rho}} \tag{5}$$

where sgn(ΔP) is the sign function affecting the flow direction, c_d is a discharge coefficient, defined as the rate of actual and ideal flows, taking also into account three dimensional flow effects and the flow process non-isentropicity; A_0 is the orifice section; ρ is the fuel density and ΔP is the fuel pressure difference across the section A_0. In presence of small pressure differences among cross sections, numerical instabilities can occur during flow computation. To overcome this problem, the computed flows are multiplied by a corrective relaxing coefficient [7]:

$$Q_{rel} = \left[1 - e^{-3 \cdot \left|\left(\frac{P_{in}}{P_{out}}\right)^5 - 1\right|}\right] \cdot Q \tag{6}$$

where P_{in} and P_{out} are the fuel pressures across the section.

Concerning the high pressure pump, the actual volume changes due to piston motion, whose position with respect to the camshaft angle is known, can be computed as:

$$V_{pump} = V_{init} - h_{pump}(\theta) \cdot A_{pump} \tag{7}$$

where A_{pump} is the cylinder bore, h_{pump} is the piston instantaneous axial displacement, θ is the camshaft angle, calculated by integrating camshaft speed; V_{init} is cylinder total volume. Hence the volume change due to piston motion is:

$$\frac{dV_{pump}}{dt} = A_{pump} \cdot \frac{dh_{pump}}{dt} \tag{8}$$

where the time derivative of h_{pump} is a nonlinear function of the camshaft angular position and speed.

The instantaneous pump inlet section is a function of the piston axial displacement and the system geometry, while the intake pressure is constant.

In this model, we neglect opening and closing transients of the delivery valve. Therefore the valve is considered to switch between two conditions: closed or opened with a flow section A_{valve}. This simplification does not introduce a considerable error, while reduces noticeably the computation time.

To compute the electro hydraulic valve outflow with equation (4) we set the outflow section equal to $A_{max} \cdot duty \cdot W_{reg}$, where A_{max} is the maximum outflow section, corresponding to a valve completely opened; *duty* is the driving current *duty cycle*; W_{reg} is a square signal of values 0 and 1 representing the valve activation window. Using the previous expression is equivalent to considering the relationship between exciting current duty cycle and flow section to be linear. This choice is justified by the fact that the shutter lift is small enough to consider the flow section proportional to shutter axial position.

By modeling the injectors as control volumes it is possible to predict irregular injection behaviors and get useful information for injection control. Moreover, in comparison to system dynamics, opening and closing transients are negligible. Thus the outflow section can be only 0 or be equal to the maximum value. For each injector, equation (5) can be rewritten in the following form:

$$Q_{cyl} = \operatorname{sgn}(\Delta P) \cdot c_{d,cyl} \cdot A_{cyl} \cdot ET \cdot \sqrt{\frac{|\Delta P|}{\rho}} \tag{9}$$

where A_{cyl} is the total outflow section, depending on the number of injection nozzles, *ET* is a square signal equal to 1 during injections; the pressure inside cylinders, which is necessary to compute Q_{cyl}, is assumed to be constant.

Equations (1) to (9) can be rewritten in a state space form, assuming the pump, delivery valve, common rail and injector pressures as state variables and the camshaft angular position and speed, the regulator exciting signal and the injectors driving signal as inputs. With the following positions:

$$\begin{aligned} x(t) &= \begin{bmatrix} P_{pump} & P_{valve} & P_{rail} & P_{inj,1} & P_{inj,2} \end{bmatrix}^T \\ u(t) &= \begin{bmatrix} \theta & rpm & duty \cdot W_{reg} & ET_1 & ET_2 \end{bmatrix}^T \end{aligned} \tag{10}$$

and assuming without loss of generality that no reversal flows occur, we get the state space representation:

$$\begin{aligned} \dot{x}_1 &= \frac{K_{f_{pump}}}{V_{init} - A_{pump} \cdot f_{1,1}(u_1)} \cdot \Big(-a_{1,1}\sqrt{x_1 - P_{tank}} + \\ &\quad - a_{1,2}\sqrt{x_2 - x_1} + A_{pump} \cdot f_{1,2}(u_1, u_2) \Big) \\ \dot{x}_2 &= \frac{K_{f_{valve}}}{V_{valve}} \cdot \Big(a_{1,2}\sqrt{x_2 - x_1} - a_{2,1}\sqrt{x_2 - x_3} \Big) \\ \dot{x}_3 &= \frac{K_{f_{rail}}}{V_{rail}} \cdot \Big(a_{2,1}\sqrt{x_2 - x_3} - a_{3,1}\sqrt{x_3 - x_4} + \\ &\quad - a_{3,1}\sqrt{x_3 - x_5} - a_{3,2} \cdot u_3 \cdot \sqrt{x_3 - P_{tank}} \Big) \\ \dot{x}_4 &= \frac{K_{f_{inj}}}{V_{inj}} \cdot \Big(a_{3,1}\sqrt{x_3 - x_4} - a_{4,1} \cdot u_4 \cdot \sqrt{x_4 - P_{cyl}} \Big) \\ \dot{x}_5 &= \frac{K_{f_{inj}}}{V_{inj}} \cdot \Big(a_{3,1}\sqrt{x_3 - x_5} - a_{4,1} \cdot u_5 \cdot \sqrt{x_5 - P_{cyl}} \Big) \end{aligned} \tag{11}$$

where $a_{i,j}$ are constant parameters.

4. Simulation Results

The model has been implemented and simulated in the MATLAB/Simulink® environment. The integration method adopted is a fourth-order Runge-Kutta.

To assess the model performance, simulation results have been compared with experimental data obtained on a common rail injection system. The parameters of the model have been tuned on the basis of the geometric specifications of the mechanical parts and experimental tests carried out in different operational conditions. The experimental setup consists of a reciprocating pump, acted by a camshaft with a double lift profile, a common rail, an electro hydraulic valve and two electro-injectors. The available experimental data are the angular position and speed of the camshaft, the power required by the driver (expressed by the position of the gas pedal), the air intake at each phase, the rail pressure, the atmospheric pressure, the temperature of the air intake (used to adjust the air intake flow).

Some simulation results, which highlight the modeling capability, are illustrated in Figs. 3 and 4. The curves refer to a whole revolution of the camshaft, with a speed of 1300 rpm, and a 500 μs excitation time for the electro-injectors. From Fig. 3, it can be noticed that the pump pressure increases because of the piston motion, until the delivery valve opens. From this moment on, the pressure decreases because of the outflow towards the rail. Subsequently, the pressure increases again because of the camshaft profile. The rail pressure is constant in the angular interval in which the delivery valve is closed, and injections do not occur. The opening of the delivery valve causes a pressure increase, which is immediately compensated by the injection fluxes and by the intervention of the regulation valve (see Fig. 4). Injections occur at 180° angular intervals, whereas the electro-hydraulic valve is active in the angular interval [28°, 185°].

The model performance in absence of injections, in different operational conditions, can be evaluated in Figure 5. The figure compares experimental [4] and

simulated rail pressures, for different duty-cycle of the regulator driving current, and different rotational speeds of the camshaft. The voltage duty-cycle affects the inlet section of the electro-hydraulic valve: as long as the duty-cycle increases, the valve outflow increases, and consequently the rail pressure decreases. On the other hand, when the pump speed increases the rail inflow increases and consequently, considering the same duty-cycle, the rail pressure increases. The model accurately approximates the rail pressure for an 800 rpm speed. A larger error is committed with high rotation regimes (1500 rpm) and high duty-cycle values. The modeling error is mainly due to the simplifications introduced in the electro-hydraulic valve model. In particular assuming a section which varies proportionally with the duty-cycle, and neglecting the force acting on the shutter which regulates the opening of the valve seem to have the major influence. In the real system, in fact, the rail pressure acts on the shutter axial displacement, and consequently on the valve outflow section.

To assess the capability of the model of predicting the rail pressure dynamics for each injection cycle, the curves obtained with the AMESIM® software for fluid dynamic simulation (Fig. 6) with those obtained with the proposed model (Fig. 7) are compared. We consider the AMESIM model presented in [4] due to the lack of experimental data; it is a reliable representation of the real system as it gives an accurate prediction of the system dynamics. The curves are plotted for different values of the electro-hydraulic valve duty-cycle (18%, 28%, 64%), in absence of injections, and with a rotational speed of the camshaft of 1800 rpm. A comparison between the two figures allows us to conclude that the pressure peaks generated by the pump are well modeled, both in amplitude and in timing. The difference in the steady values is due, as previously mentioned, to the approximation introduced in the characterization of the electro-hydraulic valve.

5. Conclusions

This paper presents a mathematical model of a common rail injection system for diesel engines. The model equations are obtained by resorting to physical laws regulating the main fluid-dynamic and mechanical phenomena; its parameters are derived directly from a minimal set of known geometrical data, so it can be easily tuned or eventually adapted to different system configurations. The proposed model is validated by comparing simulation results with both experiments and the outputs of an accurate fluid-dynamic model. The comparison shows that the model can accurately predict the system behavior in every working condition, in terms of injected flows and control volume pressures. Finally, our model is simple enough to derive model-based control laws for the rail pressure controller, thus reducing the controllers development cost and time.

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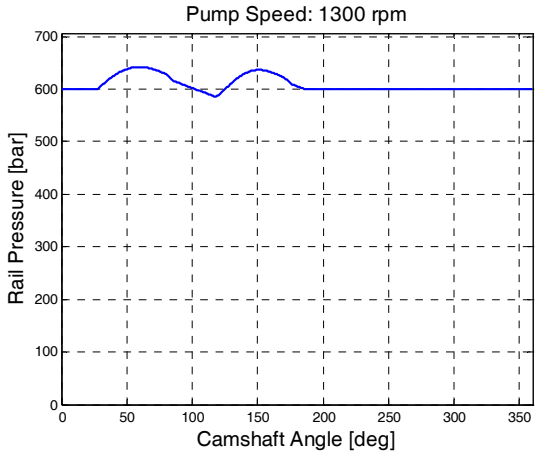

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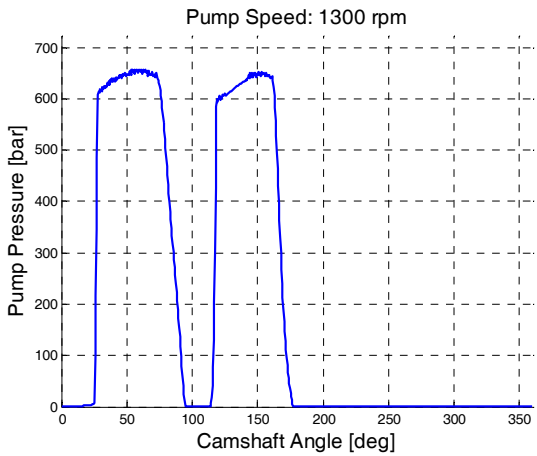

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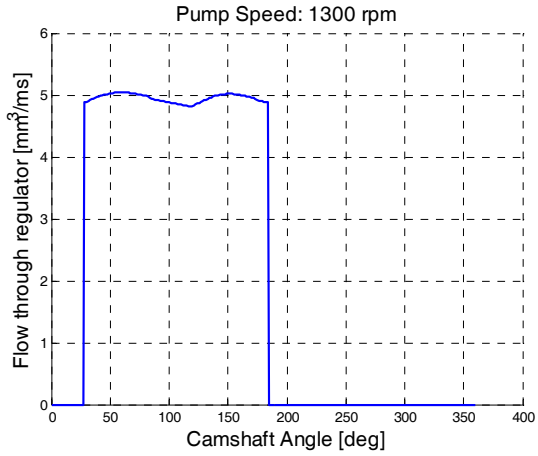
Figure 3. Simulation results for a 1300rpm camshaft speed, a 15% regulator duty cycle, and a 500 μs *ET*; (a) rail pressure; (b) pump pressure.



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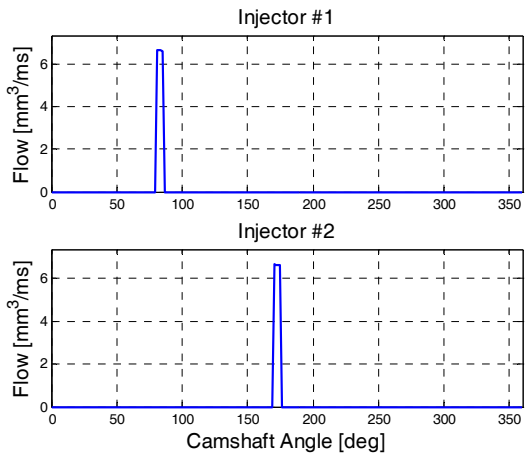

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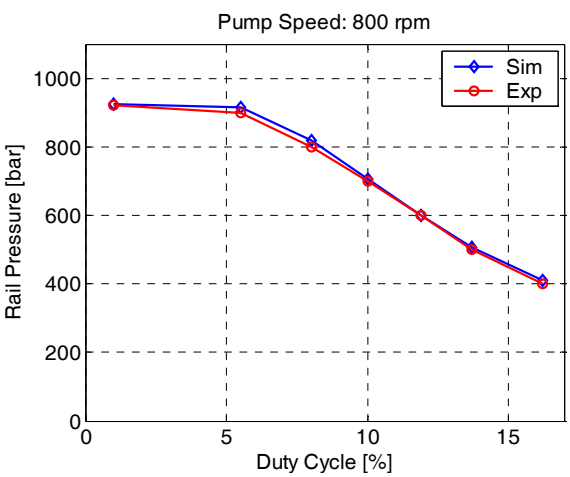
Figure 4. Simulation results for a 1300rpm camshaft speed, a 15% regulator duty cycle, and a 500 μs *ET*; (a) regulator flow; (b) injection flow.



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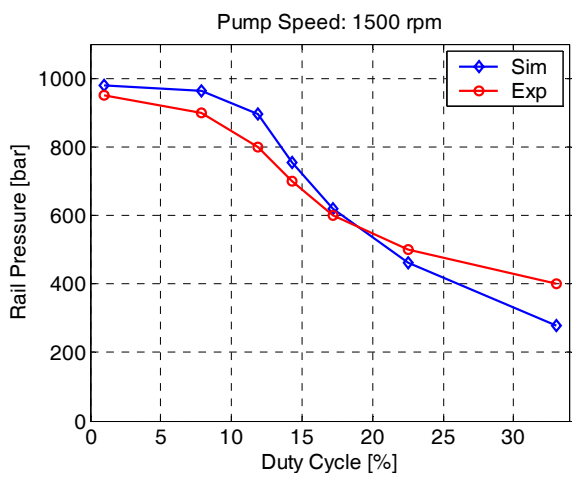

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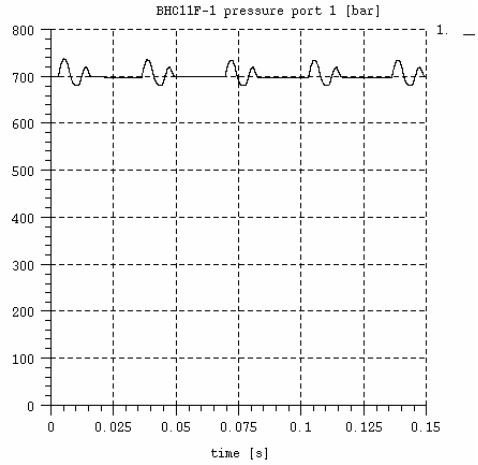
Figure 5. Experimental [4] and simulation results for different camshaft speeds.



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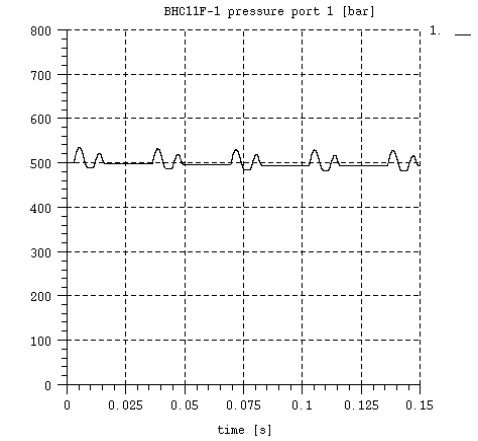

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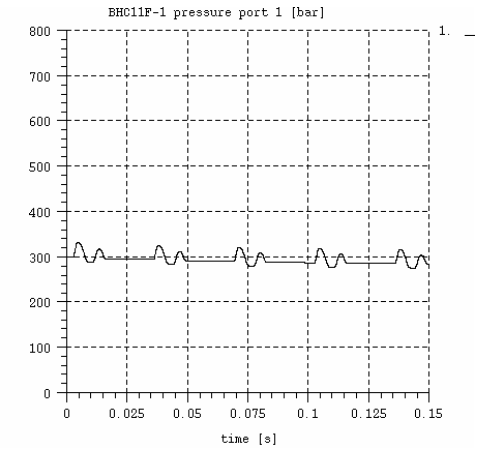
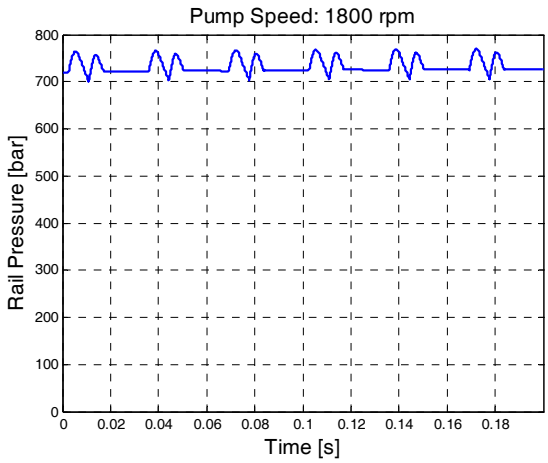


(c)

Figure 6. AMESIM simulation rail pressure for a 1800rpm camshaft speed and different values of regulator duty cycle [Dinoi]; (a) *duty cycle*= 18%; (b) *duty cycle* = 28%; (c) *duty cycle* = 64%.



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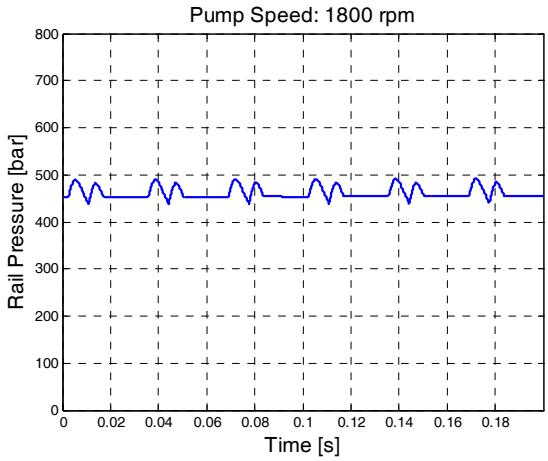

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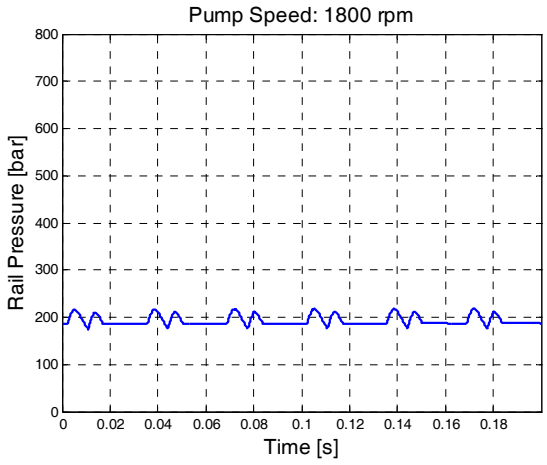

(c)

Figure 7. Simulated rail pressure for a 1800rpm camshaft speed and different values of regulator duty cycle; (a) *duty cycle* = 18%; (b) *duty cycle* = 28%; (c) *duty cycle* = 64%.