

Advanced Mathematical Modeling of Electronic Unit-Injector Systems for Heavy Duty Diesel Engine Application

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ABSTRACT

A rather complete mathematical model to simulate HD-diesel-engine EUI-system dynamics was developed and applied as a complementary tool of experimentation, for supporting design and performance optimization.

The thermo-fluid dynamics of the hydraulic components, including plunger cavity, internal injector pipes and nozzle, was modeled with the solenoid-circuit electromagnetics and the mechanics of rocker arm and follower subsystem. Onedimensional flow equations in conservation form were used to simulate wave propagation phenomena through the injector high-pressure drilled passages. To calculate the temperature variations due to the compressibility of the liquid fuel, the energy equation was used in addition to mass conservation and momentum balance equations. Furthermore, in order to determine the value of the electromagnetic force acting on the spill-valve, the application of a practical procedure was made using easily available experimental current and voltage data.

An advanced mechanical model of the rocker arm and follower subsystem was developed, taking the mechanical drive deformability into account. It was capable of accurately simulating the effects of the pressure wave dynamics on the plunger kinematics as well as on the Hertzian contact forces at the plunger-follower and cam-rocker-arm interfaces.

The numerical code was validated at different working conditions through the comparison of predicted results and experimental data in terms of pressure in the plunger cavity, injected flow-rate, needle- and spill-valve lift time-histories.

The developed EUI model was then applied for better understanding the cause and effect relationships in the analysis of system transients. Furthermore, a parametric study was carried out to assess the effects of important geometrical and mechanical system parameters on EUI operation. Finally, design keys were provided to improve the trade off between the hydraulic injector performance, the engine-out emission levels as well as the mechanical noise reduction.

INTRODUCTION

Heavy-duty diesel engines need to reduce the emission levels of NO_x and PM over the appropriate test cycles to comply with future emission-standards ([1]). Furthermore, there are continuous demands to increase engine torque, rated power, and to decrease fuel consumption ([2]). All these issues pose further requirement for more advanced combustion processes even in the presence of aftertreatment systems.

The optimization of the fuel injection system is essential to achieve improved combustion. EGR can be applied to reduce engine-out NO_x ([3]), but as its level is increased, the residence time of fuel-rich mixtures and consequently the soot formation rise. Therefore, a primary need is to improve the rate of air-fuel mixing to compensate for the effect of EGR on PM. Injection pressures approaching 2000 bar in combination with small orifice holes improve air-fuel mixing, reducing soot emissions and combustion noise ([4]). Further improvements in combustion can be expected when the high injection-pressures are adjusted at individual load and speed working conditions ([5-7]). The combined application of square injection-rate shapes with very high injection-pressure minimizes the injection temporal length and thus maximizes the amount of fuel delivered and burned close to top dead center ([8]). This results in reduced fuel consumption. The combination of shorter injection duration and enhanced air-fuel mixing makes it possible to delay injection timing with respect to systems that employ increasing or triangular rate shapes. Such a possibility allows for engine operation at low NO_x and PM especially at full load conditions ([8]).

The common rail (C.R.) and the electronic unit injector apparatuses are commonly employed in modern Diesel engines. The EUI systems can inject fuel at a very high pressure because of the absence of high-pressure pipelines ([9]). Another advantage of EUI with respect to C.R. is represented by the higher system hydraulic efficiency, in terms of ratio between the kinetic energy of the fuel flowing out of the nozzle and the effective pumping work ([10]). In C.R. systems this latter is much higher because the pump continuously produces fuel at the rail pressure level, whereas in the EUI apparatus the high pressure is

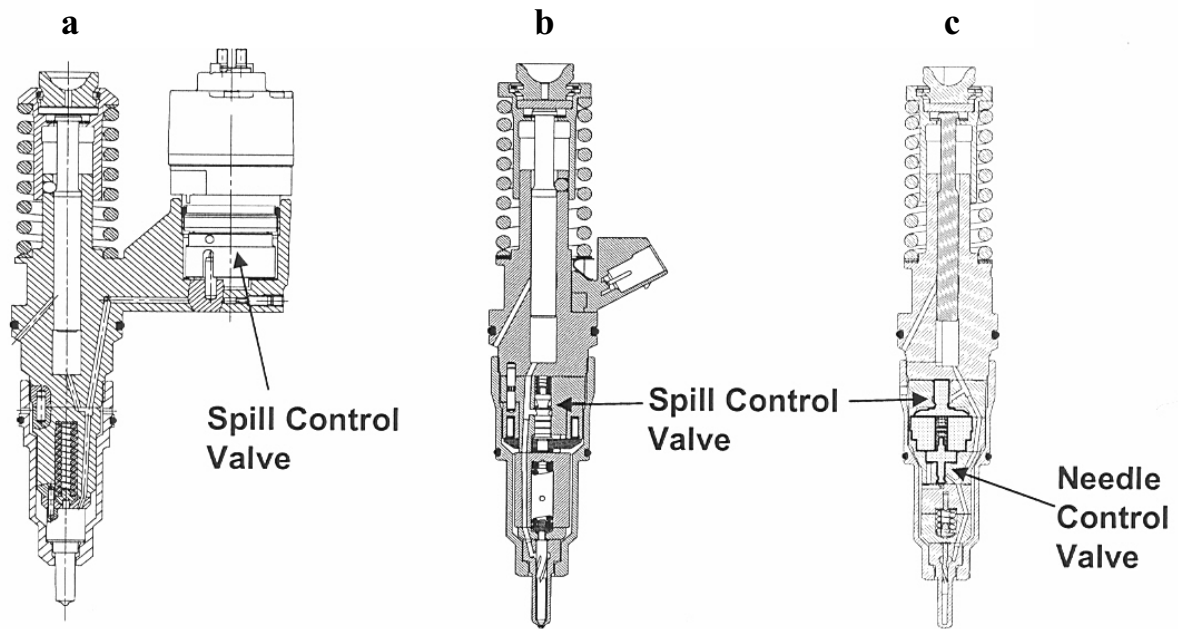


Figure 1 – EUI evolution ([16])

purposely generated for the injection event. On the other hand, C.R. systems, currently used in automotive diesel-engines, make it possible to have multiple injections with more than five shots per engine cycle ([11]). Recently, a new diesel UI, powered by hydraulics and provided with electronic control has been developed, so as to combine advantages of both EUI and C.R. systems ([12]). Such a hydraulic electronic-unit-injector (HEUI) is made up of a high-pressure rotary pump, a common rail equipped with a pressure control valve and an electronically controlled intensifier injector ([13]). It is capable of variable injection rate shaping and more flexible multiple injections ([14]) than EUI, because the intensifier valve can be opened and closed independently of engine timing ([15]). In any case, it is worth pointing out that flexible multiple injection performance are not an issue for trucks and actually no more than double injections are considered in the engine maps for heavy-duty application. This is a reason for the persistent interest of HD diesel engine manufacturers in EUI systems.

Current EUIs with a single electronically controlled spill-valve already provide a very efficient way of generating very high injection-pressure with low engine-out emission levels, good durability and reasonable costs for heavy-duty diesel applications ([16]). The Euro 3 type EUI was equipped with a side-mounted electronically controlled spill-valve (Fig. 1a) ([17]). The Euro 4 injector standard setup also has a single actuator, but with the spill control valve mounted in a central position within the body of the injector (Fig. 1b). This setup gives a much more compact arrangement with shorter drillings, smaller high-pressure volume and a substantial weight reduction. The single-actuator EUI increases injection-pressure proportionally to speed ([6]). An alternative setup of Euro 4 EUI (Fig. 1c), which has recently been cast in production, incorporates both a spill-valve and a needle control-valve within the same standard injector layout as in Fig. 1b. The needle

control-valve is aimed at injection-pressure control independent of speed and load ([16]). Improvements in the performance from Euro 3 to Euro 4 EUI setup are: faster solenoid response, reduced fuel leakage, higher injection pressure. However, the Euro 4 EUI layout led to steeper pressure gradients in the hydraulic circuit, which are the main source of mechanical noise. The study of the forces acting on the mechanical drive system due to the plunger-cavity pressure should help to better understand such a problem and provide keys for its solution.

In order to enhance the high potential of the Euro 4 EUI, a deeper knowledge of the mechanical-hydraulic behavior of the injector is required. This can be obtained using an integrated theoretical-experimental approach by means of refined numerical models and high-performance test rigs. In the current literature accurate 1D thermohydraulic models are present for C.R. systems ([18, 19]). However, the key points of EUI modeling are different from those for C.R. systems. Main differences consist in the intrinsic properties of these two apparatuses: the C.R. systems require a refined simulation of pressure wave oscillations along high-pressure pipelines because of these pressure-transient sensible impact on injected volume fluctuations when the dwell time between consecutive injections is varied ([20, 21]). On the other hand, in EUI systems the pressure changes of several orders during engine cycle and therefore it is very important to accurately evaluate thermal effects due to the fuel compressibility in order to correctly predicting pressure distributions ([22]). Besides, the EUI presents difficulties in the simulation on the drive side, with rocker arm and follower subsystem, as well as in the accurate prediction of the crank angle dependent plunger-cavity pressure ([7]).

In the present work a complete onedimensional model of Euro 4 EUI was developed and validated by comparing the predicted time distributions of plunger-cavity pressure

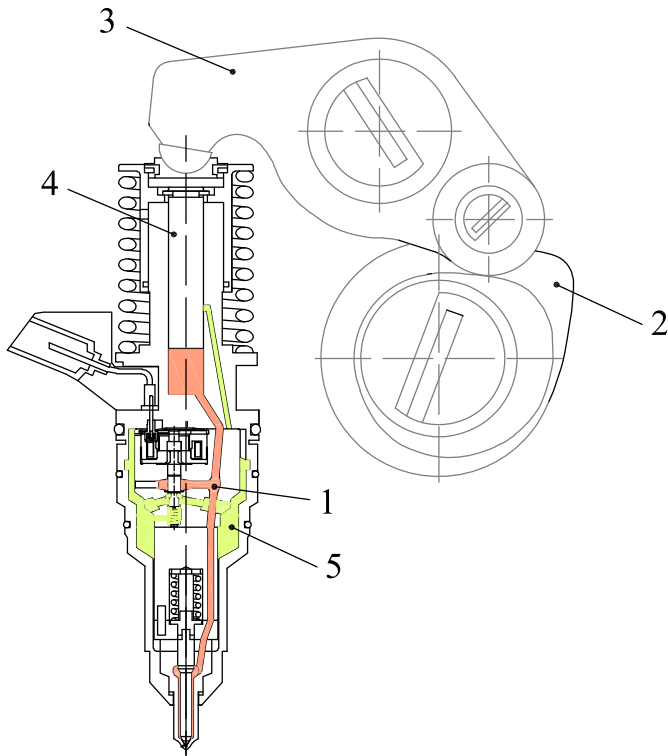


Figure 2 – EURO 4 EUI

injected flow-rate, needle as well as spill-valve lifts, with experimental data. The electric and magnetic processes in the electromagnet were simulated in conjunction with the pressure wave dynamics and kinematics of the driving mechanical components. The developed model was thus applied to the HD diesel EUI for better understanding the cause-effect relationships in its dynamics and to evaluate the influence of the system main design parameters on hydraulic performance, engine-out emissions, mechanic and combustion noise. Design aspects for their trade-off optimization were analyzed.

EXPERIMENTAL FACILITY

The experimental activity was carried out using the high performance test bench Moehwald-Bosch MEP 2000 - CA 4000 installed at the 'IC Engines Advanced Laboratory' (ICEAL) of Politecnico di Torino. The bench, rating 35 kW as maximum power and 6100 rpm as maximum shaft speed, can simulate actual engine working conditions and injection-system transient operations ([21]). It is equipped with the following main measuring instrumentation: the injection-rate indicator EVI, of the Bosch type ([23]), piezoresistive sensors to record the pressure time-history in the plunger-cavity, as well as in the delivery chamber, displacement Hall-effect magnetic transducers to detect both the spill-valve and the needle lifts, and acoustic sensors to monitor noise. The following quantities were kept under control: camshaft angular speed, temperature and pressure of the oil feeding the injectors.

The ISO 4113 fluid was used for all the experiments to simulate the Diesel fuel. A homemade software, for data

acquisition in a LabVIEW environment, was applied to monitor and control the experiments.

Figure 2 shows a drawing of the considered Euro 4 EUI and of its mechanical drive system. The high-pressure oil circuit is marked as 1. The engine driven cam (2) operates the pumping plunger (4) by the rocker arm (3). A gallery (5) is drilled into the injector cylinder head, in which the fuel circulates at low pressure. Spill and filling of injector high pressure line take place through the same gallery. Hydraulic control valves in the low-pressure circuit of the test rig allowed for the management of the pressure and temperature conditions in the gallery.

MATHEMATICAL MODEL

HYDRAULIC SUBMODEL

Figure 3 represents the hydraulic submodel of the EUI shown in Fig. 2. The schematic of the hydraulic system is basically made up of chamber and pipe elements. The valves were simulated as harmonic oscillators. Thermo-fluid dynamic properties in the chambers are taken with uniform values that can depend only on time, according to a lumped parameter mass model. The injector inner pipes, which connect the plunger cavity (1, in Fig. 3) to the junction (2) and this latter to the spill-valve upstream chamber (5) and to the delivery chamber (3), were treated as slender transmission lines, for which a 1D flow model is suitable.

The pipe Euler's equation model was formulated in terms of conservative variables as follows:

$$\frac{\partial}{\partial t} \begin{bmatrix} \rho \\ \rho u \end{bmatrix} + \frac{\partial}{\partial x} \begin{bmatrix} \rho u \\ p + \rho u^2 \end{bmatrix} = \begin{bmatrix} 0 \\ -4 \tau_w / d \end{bmatrix} \quad (1)$$

In Eq. (1) ρ , u and p are, respectively, the average pipe-cross-sectional density, velocity and pressure of the fluid;

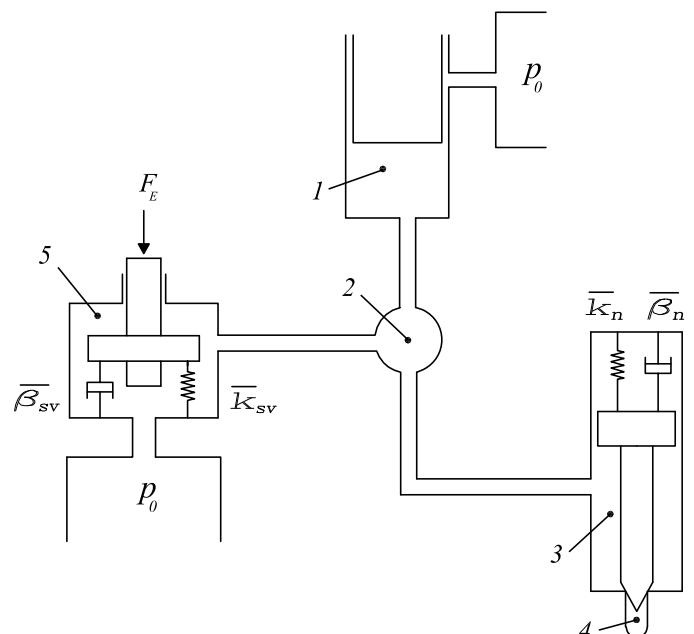


Figure 3 – EUI Hydraulic circuit

τ_w is the wall shear stress and d is the pipe diameter. The wall shear stress was evaluated using the following expression ([18]):

$$\tau_w = \frac{1}{8} f \rho |u| u + \delta \frac{16}{d^2} \mu \int_0^t \frac{\partial u(x, \lambda)}{\partial t} W(t - \lambda) d\lambda \quad (2)$$

The first term on the right hand-side of Eq. (2) is due to the stationary friction component, while the second term represents the frequency dependent component of τ_w , where the coefficient δ takes the effect of turbulence into account. It is worth pointing out that the wall shear stress models, which are commonly applied to take the friction frequency dependence into account, were obtained for laminar flows ($\delta = 1$) ([24, 25]). The way of estimating δ for the case of turbulent flow is reported in [18].

The only mass and momentum conservation equations are generally adequate for mathematically modeling of a compressible liquid isothermal flow. In such a case, the additionally required equation is the fluid state one, which reduces to $p = p(\rho)$, being the fluid temperature specified. However, in the presence of high pressure-gradients, it is essential to take thermal effects into account, due to liquid fuel compressibility, for an accurate simulation of the pipe fluid dynamics ([22]). By assuming negligible or reciprocally compensating effects of wall heat-transfer and viscous flow dissipation, an isentropic fuel evolution can be taken. Hence, the following equation for the fluid transformation is obtained from the energy equation:

$$dh - \frac{dp}{\rho} = 0 \quad (3)$$

The liquid enthalpy fundamental equation in differential form can be easily verified to be ([26]):

$$dh = c_p dT + (1 - \beta T) \frac{dp}{\rho} \quad (4)$$

Thus, from eqs. (3) and (4), taking $a_T = \sqrt{E_T / \rho}$ in mind, the following nondimensioned equation is derived linking pressure and temperature variations:

$$\frac{dp}{E_T} = \frac{c_p}{\beta a_T^2} \frac{dT}{T} \quad (5)$$

With reference to any chamber j in the mathematical model of Fig. 3, the continuity equation can be written as:

$$Q_{in,j} - Q_{out,j} = \frac{1}{E_j} \frac{dp_j}{dt} V_j + \frac{dV_j}{dt} \quad (6)$$

where $Q_{in,j}$ and $Q_{out,j}$ are the mass flow rates entering and leaving the chamber j , respectively; V_j is the volume of the chamber j , p_j is the pressure of the fluid in such a chamber and E_j is the elasticity bulk-modulus of the fluid, which depends on p_j and T_j . With reference to Fig. 3, the volume V_1 is a function of the plunger displacement; the volumes V_3 and V_4 depend on the needle-valve lift; the volume V_5 depends on the spill-valve lift. If the chamber j

is connected to a pipe, the flow rate at the chamber-pipe interface is worked out by solving the pipe-flow model PDE, i.e., Eq. (1), and then it is applied to the chamber as a boundary condition. The chamber mathematical model was completed by Eq. (5) of isentropic evolution and by the state equation of the liquid ([22]).

The fluid velocities through the nozzle holes, as well as through the needle and spill-valve seat passages, were modeled using the Torricelli's formula. More specifically, when a restriction of cross section A is located between two chambers, j and $j + 1$, the volumetric flow rate Q through it (positive when $\Delta p > 0$) is expressed by:

$$Q = \frac{\Delta p}{|\Delta p|} \mu A \sqrt{\frac{2}{\bar{\rho}} |\Delta p|} \quad (7)$$

where: $\Delta p = p_j - p_{j+1}$, μ is the discharge coefficient and $\bar{\rho}$ is an average density value obtained by considering mean temperature and pressure between the values in the chamber j (p_j, T_j) and $j + 1$ (p_{j+1}, T_{j+1}). The values of enthalpies upstream and downstream from a restriction were taken to be equal, consistently with the occurrence of throttling. Therefore, the link between temperature and pressure across any restriction was given by integration of the following differential relation:

$$c_p dT = -(1 - \beta T) \frac{dp}{\rho} \quad (8)$$

From eq. (8) it results that a pressure decrease leads to a temperature increase in agreement with the fact that the Joule-Thompson coefficient of the ISO 4113 oil, which is similar to the fuel, is negative ([22, 26]). The discharge coefficients of the injection holes and the needle-seat passage were taken as functions of needle lift ([27]). This approach allowed to consider the effects of unsteadiness, of flow-compressibility and of vena contracta on the flow-coefficient μ . The discharge coefficient for the spill valve flow passage was evaluated with the procedure reported in [28, 29].

The mechanical-valve motion results from the action of fuel pressure, inertia, viscous resistance, jet forces and spring tension. In the case of the spill valve (1 in Fig. 4), which is normally open, the electromagnet closing action exerted by the solenoid on the armature (2) must also be considered. The dynamics of both the spill- and needle-valve was simulated according to the following second-order mass-spring-damper model:

$$m_m \ddot{x}_m + \bar{\beta}_m \dot{x}_m + \bar{k}_m x_m + \bar{F}_{0,m} = F_{J,m} + \varepsilon_m F_E + (-1)^{\varepsilon_m} \sum_k p_k S_k \quad (9)$$

being

$$\begin{aligned} x_m < 0 & \quad \bar{k}_m = k_m + k_{bs} & \bar{\beta}_m = \beta_m + \beta_{bs} & \bar{F}_{0,m} = F_{0,m} \\ 0 \leq x_m \leq X_m & \quad \bar{k}_m = k_m & \bar{\beta}_m = \beta_m & \bar{F}_{0,m} = F_{0,m} \\ x_m \geq X_m & \quad \bar{k}_m = k_m + k_{bs} & \bar{\beta}_m = \beta_m + \beta_{bs} & \bar{F}_{0,m} = F_{0,m} - k_{bs} X_m \end{aligned}$$

with

$$\beta_m = 2\zeta_m \sqrt{k_m m_m} \quad (10)$$

In Eqs. (9, 10), the subscript m indicates either the spill valve ($m = sv$) or the needle valve ($m = n$). The symbol m_m represents the mass of the valve, k_m the spring rate, β_m the damping, x_m the valve displacement ($x_{sv} > 0$ when the spill valve moves downward, $x_n > 0$ when the needle moves upward) and X_m the valve stroke-end; $F_{0,m}$ is the spring preload, $\varepsilon_m F_E$ the electromagnetic force exerted by the solenoid, ($\varepsilon_m = 1$ for $m = sv$ and $\varepsilon_m = 0$ for $m = n$) and $F_{j,m}$ the steady hydraulic jet-force. This last was included in the model, because of the relatively high flow-rates through both the nozzle and the spill valve discharge areas, and was expressed as:

$$F_{j,m} = \frac{2\rho_m Q_m^2}{A_m} \left(\frac{1}{2} - \varepsilon_m \right) \quad (11)$$

where A_m is the instantaneous value of the restricted flow area between mobile element and seat and ρ_m , Q_m are the values of density and volumetric flow-rate at such location, respectively. In Eq. (9) the stiffness k_{bs} and the damping β_{bs} of the basement were also simulated in order to take the needle bouncing effects into account at nozzle closure and engender the nozzle opening delay at injection start.

ELECTROMAGNETIC SUBMODEL

The electrical behavior of the solenoid (3 in Fig. 4) was modeled by a lumped parameter approach as follows.

$$V - N \frac{d\Phi}{dt} - RI = 0 \quad (12)$$

expresses the generalized Ohm equation, in which R is the electrical resistance of the solenoid, I is the electrical current through the solenoid, V is the voltage applied to it, N and Φ are the solenoid turns and the magnetic flux through each turn, respectively. By applying the energy equation to the valve-coil system, one obtains:

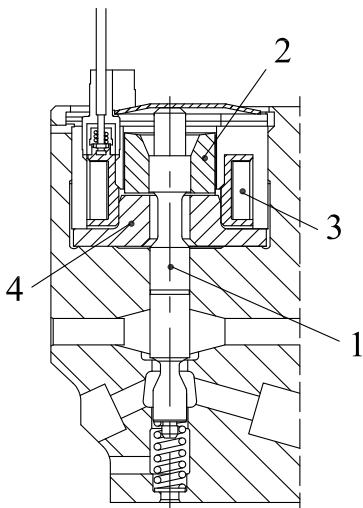


Figure 4 – Spill valve and electromagnet

$$VI dt = RI^2 dt + F_E dx_{sv} + dU_m \quad (13)$$

where VI is the electric power to the system, $F_E dx_{sv}$ is the mechanical work done by the electromagnetic force on the spill-valve armature (2 in Fig. 4); dU_m is the change of the system magnetic energy. By combining Eqs. (12) and (13) with the following general expression of the magnetic energy U_m , which is accumulated in the solenoid:

$$U_m = \frac{1}{2} NI \Phi \quad (14)$$

one obtains:

$$F_E dx_{sv} = \frac{1}{2} (NI d\Phi - N\Phi dI) \quad (15)$$

The Hopkinson's law for the equivalent magnetic circuit of the control system is given by the following equation:

$$NI = \Re \Phi \quad (16)$$

where $\Re = \Re(x_{sv})$ is the total reluctance of the system, which decreases linearly with x_{sv} . Differentiating Eq. (16) and multiplying both sides of the resultant equation by Φ , after the application again of the Hopkinson's law, one obtains:

$$\Phi^2 d\Re = \Phi N dI - NI d\Phi \quad (17)$$

Equations (15) and (17) yield the following expression for the magnetic force acting on the spill valve:

$$F_E = -\frac{1}{2} \Phi^2 \frac{d\Re}{dx_{sv}} \quad (18)$$

The only part of the circuit reluctance that changes with x_{sv} is the gap between the magnet (4) and the armature (2), so that the derivative of $\Re$ with respect to the variable x_{sv} reduces to:

$$\frac{d\Re}{dx_{sv}} = -\frac{1}{\mu_0} \frac{1}{\Sigma} \quad (19)$$

μ_0 being the magnetic constant and Σ the cross section of the magnetic circuit delimited by the internal and the external diameters of the armature, whose values were easily measured. In order to calculate the electromagnetic force from Eq. (18), the magnetic flux of the circuit has to be determined. By recording the current and voltage time histories, the following expression of the magnetic flux as a function of time can be derived from Eq. (12):

$$\Phi(t) = \frac{1}{N} \int_0^t [V(t) - RI(t)] dt \quad (20)$$

where the instant $t = 0$ corresponds to the initial time of solenoid excitation. The values of the resistance R and of the turn number N of the solenoid were experimentally determined. Equations (18-20) were implemented in the numerical code for evaluating the time distribution of the electromagnetic force F_E acting on the armature, so as to simulate the spill-valve dynamics.

ROCKER ARM AND FOLLOWER SUBMODEL

The rocker arm and follower deformability are of great importance for the generation of pressure in the plunger cavity, because they contribute to the system elasticity in a large measure. Actually the total mechanical stiffness of the drive system was shown to be in the same range of the hydraulic stiffness of the plunger-cavity and hence cannot be neglected ([7]).

Stiffer drive systems produce higher injection pressures. Furthermore, the mechanical vibrations occurring in a too compliant drive system can give rise to secondary injections ([17]). Therefore, in order to control such vibrations too weak individual pump-drives should be avoided ([9]).

Figure 5 reports the rocker arm and follower subsystem scheme. The follower (1 in Fig. 5) pushes the plunger (2) by an elastic contact whose stiffness k_{ct} (≈ 30 KN/mm) is equivalent to the driving-system total stiffness, including the plunger axial deformability. The displacement of the follower is determined by the cam profile (3), through the rocker arm (4) according to a rigid kinematic coupling. The tappet lift (y) and velocity ($\dot{y}$) can be determined once the cam geometry is assigned and therefore must be taken as input data for the rocker-arm and follower submodel. The rotation angle of the rocker arm (ϑ) and its circular frequency ($\dot{\vartheta}$) are linked to y and $\dot{y}$, respectively, by the following expressions:

$$\begin{cases} y = R_1 \sin \vartheta \\ \dot{y} = R_1 \cos \vartheta \dot{\vartheta} \end{cases} \quad (21)$$

The rocker-arm rotational equilibrium can be written as:

$$I_{ra} \ddot{\vartheta} = F_C R_1 - F_P R_2 + W_{ra} b \cos \vartheta \quad (22)$$

where I_{ra} is the moment of inertia of the rocker arm, W_{ra} is its weight and b is the distance of the rocker-arm center of mass (point G) from the rotation axis; F_C is the normal force exerted by the cam on the tappet and F_P is the axial force applied by the follower to the plunger. The force F_P can be expressed as a function of the driving-system stiffness k_{ct} in the following way:

$$F_P = k_{ct} (R_2 \sin \vartheta - z) \quad (23)$$

The mechanical model of the drive system is completed by the Lagrange's equation for the plunger:

$$\frac{d\ell}{dt} - \frac{\partial \ell}{\partial z} = -p_1 S \quad (24)$$

where ℓ is the Lagrange's function, equal to the sum of the kinetic energy and of the elastic potential energy:

$$\ell = \frac{1}{2} m_{pl} \dot{z}^2 - \frac{1}{2} k_s z^2 - \frac{1}{2} k_{ct} (R_2 \sin \vartheta - z)^2 - F_0 z \quad (25)$$

By introducing Eq. (25) into Eq. (24) and developing the derivatives one obtains:

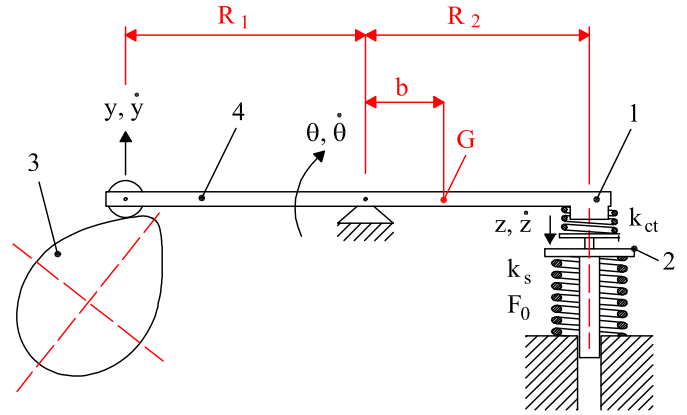


Figure 5 – Rocker arm and follower subsystem

$$m_{pl} \ddot{z} = -p_1 S - F_0 - k_s z + k_{ct} (R_2 \sin \vartheta - z) \quad (26)$$

From a theoretical point of view, the contact between plunger and follower exists if $F_P \geq 0$, that is, according to Eq. (23), if $R_2 \sin \vartheta \geq z$. This condition is always satisfied because the injector-spring stiffness and preload (k_s, F_0) and the oil pressure (p_1 in Fig. 3)) thrust the plunger against the follower. Thus, since the contribution of the inertia force $m_{pl} \ddot{z}$ is marginal, Eq. (26) holds only if F_P is higher than zero.

Substitution of Eqs. (21), (23) into Eqs. (22), (26) gives rise to a differential system of two equations with three unknown variables, namely F_C , p_1 and z . The matching of the rocker and follower submodel to the hydraulic one closes the resulting equation set.

NUMERICAL ALGORITHM

The mass and momentum conservation equations for the pipe flow were discretized through the high-resolution finite-volume scheme SLIC (Slope Limiter Centered) with a *minmod* slope limiter ([18]). In fact, a slope-limited linear spatial reconstruction of the conservative variables was performed at time t and then a time upgrading step was carried out to obtain the values of same variables at time $(t + \Delta t/2)$, which were necessary for the SLIC scheme to calculate the numerical flux functions. Besides, the multi-step Adams-Moulton method of the fourth order accuracy was applied for the discretization of the continuity and energy equations modeling the chambers, and also for discretizing the dynamics equations of the injector mobile elements and of the electromagnet ([18]). The LabVIEW CVI software package for C language was used to create and run the model.

MODEL VALIDATION

Figures 6-9 show the comparison between the model prediction (solid line) and experimental results (symbols) for an engine speed (n) of 2000 rpm and energizing time (ET) of $2600 \mu s$ corresponding to full load conditions. The experimental data refer to an EURO 4 heavy-duty engine