Engine Valvetrain Lift Prediction Using a Physic-based Model for The Electronic Control Unit Calibration

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Abstract. The electronic control has an increasingly important role in the evolution of the internal combustion engine (ICE) and the vehicle. Research in the automotive sector, in this historical period, is dictated by three main guidelines: reducing polluting emissions and fuel consumption while maintaining high performance. The Electronic Control Unit (ECU) has made it possible, complicating the engine both in terms of architecture and in terms of strategies, controlling, through simplified functions, physical phenomena in an ever more precise way. The ECU functions are experimentally calibrated, reducing the error between the quantity estimated by the function and the experimental quantity over the entire operating range of the engine, developing extensive experimental campaigns. The calibration process of the ECU functions is one of the longest and most expensive processes in the development of a new vehicle. Some lines of research have been explored to reduce the experimental tests to be carried out on the test bench. The use of neural networks (NN) has proven to be effective, leading to a reduction in experimental tests from 40 to 60%. Another methodology consists in the use of 1D/0D Thermo-fluid dynamic models of the ICE. These models are used as virtual test benches and through them it is possible to carry out the experimental campaigns necessary for the calibration of the control unit functions. At the real test bench, only the few experimental tests necessary for the validation of the model must be carried out. One of the simplifications that is usually made in the 1D/0D ICE models consists in assigning a single intake and exhaust valve lift, without taking into account the effect of the engine speed on the valve lift in early intake valve closure (EIVC) mode for engines equipped with VVA. This phenomenon has a not negligible effect on engine performance, especially at high engine speeds. In the case of engine models equipped with VVA, the valve lift cannot be imposed, since it is unique for each closing angle at each engine speed. Indeed, in order to assign the correct valve lift for a given engine speed and EIVC, numerous experimental tests should be carried out, making vain the beneficial effects of the method. In this work, the authors propose the use of a 0D/1D CFD model of the entire electro-hydraulic valvetrain VVA module, coupled with 1D lumped mass for reproducing the linear displacements of the intake valve, and for simulating the interactions between flow and mechanical systems of the solenoid hydro-mechanical valve. Thus, model simulations allow to predict the valve lift in all the necessary conditions in the

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experimental campaigns for the calibration of the control unit functions. Starting from geometric valvetrain data, the model has been validated with a parametric analysis of some variables on which there was greater uncertainty, by comparing the valve lift obtained by the model with the experimental ones in certain engine speeds. Subsequently, the authors have obtained the valve lifts in conditions not used for model validation, comparing them with their respective experimental lifts. The model has proven to be sensitive to the effect of the variation of the engine speed, reproducing the valve lift with a low error. In this way it is possible to reduce the experimental effort aimed to the calibration process considering that the virtual experimental campaign has proven to be reliable.

**Keywords:** Thermo-fluid dynamic analysis, Experimental tests, 1D engine model calibration, ECU base calibration.

## 1 Introduction

The on-board electronic control unit (ECU) has allowed to develop the engine and vehicle with new strategies and architectures, by controlling complex physical phenomena in a more precise way.

Fig. 1 shows the operating principle of the control units. The objective algorithms must estimate the quantities required by the driver, while, at the same time, the estimation algorithms must estimate the quantities supplied by the engine. If these two quantities do not coincide, the engine control varies the actuation (e.g., throttle valve, spark advance, injection etc.) in order to reduce the difference between the required quantity and the estimated one.

![Logical Scheme of the Engine ECU.](image)

The ECU features are:
- Thousands of algorithms or functions,
- Limited computing power,
- Providing of the required signal in real time.

In order to achieve the combination of these features the ECU functions must necessarily be simplified. An example of a really used function is the following:

\[ VOL_{EFF} = \frac{M_{AIR}}{M_{REF}} \] (1)

Where \( VOL_{EFF} \) is the volumetric efficiency calculated as the ratio between the mass trapped in the cylinder \( M_{AIR} \) and a reference mass \( M_{REF} \). \( M_{AIR} \) is estimated as:

\[ M_{AIR} = \frac{(P_{man} \cdot V_c \cdot a - V_{cc} \cdot P_{eg} \cdot b)}{R \cdot T_{man}} \cdot c \] (2)
Where $P_{man}$ is the manifold pressure, $P_{eg}$ is the exhaust gas pressure, $V_c$ is the engine displacement, $V_{cc}$ is the volume of the combustion chamber, $R$ is the gas constant and $T_{man}$ is the manifold temperature. It derives from the Ideal gas law: a very complex phenomenon is described by a simple empirical equation. Obviously, this equation is not able to recreate the value of volumetric efficiency over the entire operating range of the engine, but only in a few points. For this reason, and not to increase the computational costs deriving from an increase in the complexity of the functions, the functions are equipped with corrective parameters that can be maps, vectors or scalars, which, according to the operating point, correct the values of the function to ensure that it is as close as possible to the experimental quantity. In the case of function (2), 3 corrective maps are used, which, at each operating point of the engine, provide the value of the corrective (or calibration) parameter.

The choice of the values of the calibration parameters is part of a much longer process named engine base calibration process [1-8], and is carried out with suitably developed software [1,4], which require an intense experimental campaign as described in [2-3]. In fact, at each single operating point, the software calibrates the functions, by comparing the experimental quantity and the quantity estimated by the function, reducing the error by changing the calibration parameters. The more the points where the function is validated, the better the engine behaviour will be. Generally, during the engine base calibration process, experimental campaigns are carried out, thousands of operating conditions are analyzed, and in each operating condition hundreds of experimental quantities are obtained. The purpose of carrying out the experimental campaigns is to provide the calibration software all the quantities necessary, collected in datasheets, to compare them with the quantities estimated by the ECU functions in certain operating conditions. In the past the authors have used two different approaches to try to decrease the experimental tests: the use of neural networks (NN) to extend the experimental campaigns has proven effective, leading to a reduction from 40 to 60% of the experimental tests without affecting the calibration performance [2-3]. The second approach is the use of 1D simulation models of the engine to be used as a virtual test bench in order to obtain all the experimental quantities necessary for the calibration of the functions [2,5]. This approach has shown good results, but has also highlighted some simplifications made to the model, which, if implemented, can lead to an improvement of the process.

2 Background and Motivation

Fig. 2 shows the schematization of the engine model used in the second approach. It is a two-cylinder gasoline engine, 900 cm$^2$ of total displacement, with indirect injection, equipped with turbocharger and VVA distribution system; more details are reported in [2,5,9].
The model was validated [2,5] with a multi-variable multi-objective validation process [10-13], choosing 5 calibration parameters (3 relating to turbulent combustion, 1 relating to the heat exchange of the cylinder walls and 1 relating to pressure drops in the intake system) and 6 quantities evaluated by the model to reduce the error compared to the same experimental quantities in 16 different experimental conditions. Furthermore, the throttle model and the friction model were validated using other 15 experimental conditions, for a total of 31 operating conditions [5]. The model demonstrated an average error of 3.3% for the 6 quantities used as an objective function in the 16 experimental conditions. Subsequently, the model, used as a virtual test bench, produced an experimental campaign which can be described as follows:

- The engine speed is fixed and increased by steps. The chosen engine speeds are: 1500, 1800, 2100, 2300, 2500, 2700, 3000, 3300, 3600, 4000, 4400, 4800, 5100, 5500 rpm.
- The wastegate valve is completely opened.

For each engine speed the following operations are performed:

- An intake valve closing sweep (EIVC) is made, from the maximum to the minimum allowed value (i.e. 630, 625, 620… 430 CA degrees)
- The throttle valve is regulated to reach a downstream/upstream pressure ratio of 0.88
- The spark advance is modified in order to obtain the maximum torque or incipient detonation.

In Fig. 3 (a) part of the valve lifts of the intake valves imposed in the various operating conditions are represented.

With this strategy 285 different simulated operating conditions have been obtained and they represent all the experimental quantities necessary for the ECU base calibration process.

The datasheet thus obtained has been used as input in the calibration software to calibrate the volumetric efficiency function. Fig. 4 shows the difference in performance obtained by calibrating the function using the real datasheet (a) and the virtual datasheet (b). In the first case it is noted that the calibration performance reaches 94.5%, or 94.5% of the errors evaluated as:

$$\text{err}_\% = \frac{Q_{\text{EXP}} - Q_{\text{EST}}}{Q_{\text{EXP}}} \cdot 100$$

falls within the acceptable range of ±5%. In the second case the performance reaches the value of 89.5%.

This difference in performance could be due to some simplifications made in the 1D / 0D model, including the use of an empirical friction model [14] and the use of a simplified model of intake valves. Indeed, as shown in Fig. 3 (a), the same lifts of the intake valves were used.
when the engine speed changed, without taking into account the effect of the change in the engine speed on the valve lift. As the engine speed increases, with the same closing angle of the intake valves in VVA mode, the valve lift is gradually lower, as shown in Fig. 3 (b). This effect is due to the fact that, for the same $\Delta \theta$, the available $\Delta t$ for valvetrain to make the lift decreases. For this reason, the electric closing signal must be sent earlier, resulting in a reduced lift at the same closing angle. This phenomenon has a not negligible effect on the filling of the cylinder and consequently on the behaviour of the engine.

![Fig. 4 - difference in performance obtained by calibrating the function using the real datasheet (a) and the virtual datasheet (b)](image)

For this reason, in order to increase the calibration performance, the research team decided to develop an engine valvetrain lift prediction physic-based model and friction models [8,15,16]. The obtained lift is then inserted in the simplified model used in the 1D / 0D model of the engine.

### 3 Engine valvetrain model

Fig. 5 shows the one-dimensional flow schematization of the tested VVA system. The used approach is the traditional one, applied to the flows in the injectors [17-18] and in the hydraulic valvetrain systems [19]. For the chambers $V_1$-$V_7$ of the Fig. 5, pressure and temperature are assumed constant in the space and the velocity is taken to be virtually nil. So, the governing equations of the phenomena are only the mass conservation and the total energy balance, that can be written in this form:

$$\begin{align*}
V \frac{d\rho}{dt} + \rho \frac{dV}{dt} = m_{in} - m_{ex} \\
\rho \cdot V \cdot c_{oil} \frac{dT}{dt} + c_{oil} \cdot T \cdot \frac{d\rho}{dt} + c_{oil} \cdot T \cdot \rho \frac{dV}{dt} = -p \frac{dV}{dt} - \frac{dQ_w}{dt} + m_{in}h_{in} - m_{ex}h_{ex}
\end{align*} \tag{4}$$

where $m_{in}$ is the input mass flow rate coming into the chamber, $m_{ex}$ is the output mass flow rate going out of the chamber, $V$ the chamber volume, $\rho$ the density of the fluid in the chamber, $t$ the time, $c_{oil}$ the specific heat of the oil, $T$ the temperature, $p$ the pressure, $Q_w$ the heat exchanged with the wall, $h_{in}$ enthalpy per unit mass of the fluid coming into the chamber and $h_{ex}$ enthalpy per unit mass of the fluid going out of the chamber.

The pipes $L_1$-$L_7$ of the Fig. 5, due to the importance of wave phenomena, has been simulated by means a one dimensional simulation approach. Therefore, the mass and momentum conservation equations assume the form [17]:

5
\[
\begin{align*}
\frac{\partial \rho}{\partial t} + \frac{\partial (\rho u)}{\partial x} &= 0 \\
\rho \frac{\partial u}{\partial t} + \rho \cdot u \frac{\partial (u)}{\partial x} + \frac{\partial (p)}{\partial x} + \rho \cdot R \cdot u &= 0
\end{align*}
\]

(5)

The terms \( u \) and \( R \) respectively represent the speed and the resistance factor.

The system (Fig. 5) consists of a cam moved by a camshaft, which pushes a piston inside a cylinder. Also, there is a solenoid valve. A current signal determines the position of the piston in the solenoid valve. If the piston is in position 1, the movement of the cam causes a decrease in pressure in the oil circuit, since the passage to the accumulator is open: in this case the intake valve remains closed. If the piston is in position 2, the movement of the cam causes an increase in pressure in the circuit causing the intake valve opening.

![Fig. 5 – 1D-0D scheme of the Hydraulic Valvetrain System.](image)

When the solenoid valve is in position 2, the pressure exerted by the cam on the piston \( M_1 \) is transmitted to the piston \( M_2 \), causing the valve to open. In this work, the Early Intake Valve Closing (EIVC) strategy was considered. Initially, the solenoid valve piston is in position 2 and the active profile of the cam determines the opening of the valve due to the increase in pressure in the circuit. When an early closure is required, during the active profile of the cam, the current signal is interrupted and the solenoid valve piston passes to position 1, causing a reduction in the pressure in the oil circuit and consequently the closing of the valve.

The movement of the piston and the masses \( M_1 \) and \( M_2 \) are determined by the equation:

\[
m \ddot{x} + \sigma \dot{x} + SM \cdot kx = F_e + \sum p_i S_i
\]

(6)

where \( m, x, \sigma, k, SM, F_e, p_i \) and \( S_i \) are respectively the mass, the position, the viscosity, the spring constant, the spring stiffness multiplier, the external force, the pressure and the surface. The differential equation (6) is solved with the constraint:

\[
x \geq x_c
\]

(7)

where \( x_c \) is the cam position.
Fig. 6 – Experimental and calculated valve lift and cam profile.

Fig. 6 (a) shows the difference between valve lift and cam profile evaluated experimentally. As can be observed, the valve lift is higher than the cam profile. This is due to the difference in area between the piston \( M_1 \) and the piston \( M_2 \). Fig. 6 (b) shows the difference in the lift of the experimental valve with that obtained by the model at three different engine speeds. The valve lift calculated by the model is below the experimental lift, but the start and end positions of the lift are well coordinated. This is due to an inaccuracy on the areas of the two pistons \( M_1 \) and \( M_2 \). Indeed, the shape of the active areas that determines the pressure in the oil circuit is known with an uncertain.

4 Parametric analysis: results and discussion

A parametric analysis was performed by varying the area of the piston \( M_1 \) ranging from its initial diameter \( \Phi_{M1} \) value of 9 mm to the value of 11 mm.

Fig. 7 – Parametric analysis of the valve lift varying the diameter \( \Phi_{M1} \).

Fig. 7 (a) shows the trend of the valve lift calculated by the model as the diameter \( \Phi_{M1} \) changes, for different engine speeds. It should be noted that as the area of the piston \( M_1 \) increases, the valve lift increases. The optimal value of the diameter \( \Phi_{M1} \) that produced the minor error is equal to 10.55 mm. Fig. 7 (b) shows the difference between the lift of the experimental valve and that calculated by the model using \( \Phi_{M1} = 10.55 \text{ mm} \). As can be
observed, the model faithfully reproduces the lift of the experimental valve in Full Lift condition.

Table 1 - Relative error percentage of the 3 operating conditions

| RPM | 1500 | 3000 | 6000 |
|-----|------|------|------|
| Max lift error% | 1.1  | 0.2  | 1.7  |
| Relative error % | 1.2  | 1.9  | 4.3  |

Table 1 shows the relative error percentage of the 3 operating conditions evaluated as:

\[ e_{i}^{\%} = \left( \frac{1}{N} \sum_{j=x_{1}}^{x_{2}} \frac{Y_{\text{exp},j} - Y_{\text{calc},j}}{Y_{\text{exp},j}} \right) \times 100 \]  

Where \( x_{1} \) and \( x_{2} \) are the opening and the closing angle respectively, \( Y_{\text{exp},j} \) and \( Y_{\text{calc},j} \) are the experimental lift and that calculated by the model at the j-th crank angle. The error was estimated in steps of 1 degree of crank angle. It can be noted that the relative average error is very low in all 3 operating conditions, with a maximum error of 4.3% relating to the condition of 6000 rpm.

Consequently, the model was used to create valve lifts in EIVC mode as required by the experimental campaign described above. In the model only the current signal of the solenoid valve is modified in order to control the pressure in the system and achieve early closing. Each closing angle corresponds to its current profile.

Fig. 8 shows only part of the early closing lifts obtained by the model compared to the experimental ones, at 1500, 3000 and 6000 rpm.

Table 2 – Max and relative error percentage for early closing lifts.

| RPM | 1500 | 3000 | 6000 |
|-----|------|------|------|
| EIVC | 380° | 430° | 484° | 548° | 602° | 396° | 440° | 492° | 544° | 596° | 522° | 533° | 554° | 579° | 608° |
| Max lift error% | 1.1  | 0.2  | 1.1  | 1.1  | 8    | 7.3 | 0.1  | 0.2  | 0.2  | 0.2  | 1.1  | 5.7  | 1.4  | 1.5  |
| Relative error % | 2.1  | 2.6  | 1.5  | 1.4  | 1.4  | 6.8 | 2.9  | 5.6  | 7.1  | 1.8  | 16.3 | 16.3 | 17.1 | 15   | 4.8  |

As can be observed from the Fig. 8 and Table 2 the model is able to recreate the valve lift correctly. The error increases as the engine speed increases. In particular, at 1500 rpm the model faithfully reproduces the experimental lifts, and the average relative error is less than
2.6% while the error on the maximum lift is less than 1.1%. At 3000 rpm the error increases as the closing angle decreases. Although the maximum relative error is greater than 7%, it is acceptable, as very small measures are involved with absolute errors of a few tenths of a millimeter. At 6000 rpm the highest errors occur, and in particular the error increases as the closing angle decreases, having a minimum error of 4.8% with a closing angle at 608° and a maximum error of 16.3% with an angle of closing of 522°.

The calibration proved to be very promising, as the model is able to reproduce the experimental lifts with a reduced error. A further parametric analysis has been carried out in order to reduce the error where it is higher, i.e. at 6000 rpm in EIVC mode. As can be seen, in this case the source of error is due to the descent phase of the valve. This discrepancy could be caused due to an underestimation of the stiffness of the intake valve spring. This element affects the force that the M₂ mass exerts on the oil during the descent phase.

The second parametric analysis concerned the spring stiffness multiplier ($SM$ of equation 6), varying it from 1 to 1.6. The best results were obtained with a $SM$ value of 1.5, as shown in Fig. 9. At 6000 rpm condition, this parameter greatly influences the valve descent phase, and its effect is greater with the reduction of the valve closing angle. The results in terms of error are shown in Table 3. It can be noted a high reduction in error at earlier valve closing angles. The effect of the variation of the $SM$ is less marked in conditions close to full lift.

![Fig. 9 – Parametric analysis at 6000 rpm varying the spring stiffness multiplier.](image)

The same parametric analysis was also carried out in the conditions of 1500 and 3000 rpm, to analyze the effect even at low rpm. In this case, the effect of the change of SM is less marked. As can be observed in Fig. 10 (a) and Table 3, it does not have a consistent effect at
1500 rpm condition. The relative error is very similar to the case of SM = 1, except in the condition at closing angle of 484°, in which there is a slight increase from 1.5 to 5%.

Table 3 - Max and relative error percentage for early closing lifts after the parametric analysis

| RPM | EIVC | Max lift error % | Var | Relative error % | Var |
|-----|------|------------------|-----|------------------|-----|
|     | 1500 |                  |     |                  |     |
|     | 1500 | 380°             | 1   | 0.2              | 0.2 |
|     | 3000 | 430°             | 0.2  | 1.7              | 1.7 |
|     | 6000 | 484°             | 0.3  | 0.8              | 0.8 |
|     | 1500 | 544°             | 1.1  | 3.5              | 3.5 |
|     | 3000 | 596°             | 0.1  | -0.6             | -0.6 |
|     | 6000 | 522°             | 0.2  | 2                | 2   |
|     | 1500 | 554°             | 0.2  | 0.1              | 0.1 |
|     | 3000 | 579°             | 1.1  | 0.2              | 0.2 |
|     | 6000 | 596°             | 1.1  | 0.2              | 0.2 |

Fig. 10 - Parametric analysis at 1500 and 3000 rpm varying the spring stiffness multiplier.

Analyzing Fig. 10 (b) relating to the condition at 3000 rpm, it is noted that SM has a greater influence on the valve lift than in the case of 1500 rpm. As also shown in Table 3, there is an increase in the error, in particular in the condition with a closing angle of 440°, going from 2.9 to 10.6%. This increase is partly due to the increase in the error of the valve lift peak.

5 Conclusion

This work addresses the possibility of applying a 0D-1D physic-based valvetrain model for the reduction of the experimental tests of the basic calibration process with the procedure described in [5]. In this procedure the use of 1D-0D engine models requires the imposition of valve lifts. The authors propose the use of 0D-1D physic-based valvetrain model to predict the valve lifts considering the effect of the variation of engine speed. The model has been validated using a preliminary parametric analysis of 2 parameters on which there was greater uncertainty, by reducing the error between the lift predicted by the model and the experimental one at 3 different engine speeds. The results highlight the model's capability to recreate the valve lift over the entire operating range of the engine with a low error compared to the experimental lifts. This model can be used as an additional useful tool to recreate the virtual experimental campaigns necessary for the ECU base calibration. Moreover, further improvements in the model accuracy can be reached by using a multivariable-multiobjective
validation process during the validation phase of the valvetrain model, considering all the variables on which there is greater uncertainty, without increasing the experimental costs.

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References

1. de Nola F. et al., “A Model-Based Computer Aided Calibration Methodology Enhancing Accuracy, Time and Experimental Effort Savings Through Regression Techniques and Neural Networks”, SAE Technical Paper # 2017-24-0054, doi: 10.4271/2017-24-0054. (2017).
2. de Nola F. et al., “Reduction of the experimental effort in engine calibration by using neural networks and 1D engine simulation”, Energy Procedia, Volume 148, August 2018, Pages 344-351, doi: 10.1016/j.egypro.2018.08.087. (2018)
3. de Nola F. et al., “Volumetric efficiency estimation based on neural networks to reduce the experimental effort in engine base calibration”, Fuel, Volume 244, 15 May 2019, Pages 31-39, doi: 10.1016/j.fuel.2019.01.182. (2018)
4. de Nola F. et al., “Enhancing the accuracy of engine calibration through a computer aided calibration algorithm”, Energy Procedia, Volume 148, August 2018, Pages 916-923, doi: 10.1016/j.egypro.2018.08.094. (2018)
5. de Nola F. et al., “Definition of a Methodology Promoting the Use of 1D Thermo-Fluid Dynamic Analysis for the Reduction of the Experimental Effort in Engine Base Calibration”, SAE paper 2019-24-0013, doi: 10.4271/2019-24-0013. (2019).
6. Gimelli A. et al., “Preliminary Analysis of a Hydraulic Variable Valve Actuation Loss Model for the Control-Oriented Base Engine Calibration”, AIP Conference Proceedings, Volume 2191, 17 December 2019, doi: 10.1063/1.5138823. (2019)
7. de Nola F. et al., “A control-oriented and physics-based model of the engine crank mechanism friction for the base calibration: Parametric analysis”, AIP Conference Proceedings, Volume 2191, 17 December 2019, Article number 020060, doi: 10.1063/1.5138793. (2019)
8. Brancati R. et al., “Crank Mechanism Friction Modeling for Control-Oriented Applications”, Mechanisms and Machine Science, Volume 91, 20 August 2020, doi: 10.1007/978-3-030-55807-9_81. (2020)
9. Bozza F. et al., "Strategies for Improving Fuel Consumption at Part-Load in a Downsized Turbocharged SI Engine: a Comparative Study", SAE Int. J. Engines 7(1):2014, doi:10.4271/2014-01-1064. (2014)
10. Poloni C. et al., “Hybridization of a multi-objective genetic algorithm, a neural network and a classical optimizer for a complex design problem in fluid dynamics”, Comput. Methods Appl. Mech. Engrg. 186 (2000) 403-420.
11. Gimelli A. et al., “Efficiency and Cost Optimization of a Regenerative Organic Rankine Cycle Power Plant through the Multi-Objective Approach”, Applied Thermal Engineering, 114, 601–610, doi: 10.1016/j.applthermaleng.2016.12.009. (2017)
12. Gimelli A. et al., “Optimal Design of Modular cogeneration Plants for Hospital Facilities and Robustness Evaluation of the Results”, Energy Conversion and Management, 134, 20–31, doi: 10.1016/j.enconman.2016.12.027. (2017).
13. Gimelli A. et al., “The Key Role of the Vector Optimization Algorithm and Robust Design Approach for the Design of Polygeneration Systems”, Energies 2018, 11(4), 821; doi:10.3390/en11040821. (2018)
14. Chen S.K. et al., "Development of a Single Cylinder Compression Ignition Research Engine", SAE Paper 650733. (1965)
15. Gimelli A. et al., Study of a New Mechanical VVA System. Part I: Valve Train Design and Friction Modeling, International Journal of Research Engines, SEP 2015, Volume: 16 Issue: 6 Pages: 750-761. DOI: 10.1177/1468087414548773.
16. Gimelli A. et al., Study of a New Mechanical VVA System. Part II: Estimation of the Actual Fuel Consumption Improvement through 1D Fluid Dynamic Analysis and Valve Train Friction Estimation, International Journal of Engine Research, SEP 2015, Volume: 16, Issue: 6, Pages: 762-772. DOI: 10.1177/1468087414548773.
17. Augugliaro G. et al., “A Simulation Model for a High Pressure Injection Systems”, SAE paper 971595, (1997).
18. Catania A. E. et al., "Development and Application of a Complete Multijet Common-Rail Injection-System Mathematical Model for Hydrodynamic Analysis and Diagnostics", SAE Paper 780232.
19. Wong P. K. et al., "Modeling and Simulation of a Dual-Mode Electrohydraulic Fully Variable Valve Train for Four-Stroke Engines", International Journal of Automotive Technology, Vol. 9, No. 5, pp. 509–521, doi: 10.1007/s12239-008-0061-2. (2008)