A MODEL BASED NEW METHOD FOR INJECTION RATE DETERMINATION

by

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This paper presents a detailed model of a common rail diesel injector and its validation using injection rate measurement. A new method is described for injector nozzle flowrate determination using simulation and measurement tools. The injector model contains fluid dynamic, mechanic and electro-magnetic systems, describing all-important internal processes and also includes the injection rate meter model. Injection rate measurements were made using the Bosch method, based on recording the pressure traces in a length of fuel during injections. Comparing the results of the simulated injection rate meter, simulated injector orifice flow and injection rate measurements, the simulated and measured injection rates showed good conformity. In addition to this, the difference between nozzle flow rate and the measured flow rate is pointed out in different operating points, proving, that the results of a Bosch type injection rate measurements cannot be directly used for model validation. However, combining injector, injection rate meter simulation and measurement data, the accurate nozzle flow rate can be determined, and the model validated.

Key words: *common rail, injector, injection rate, simulation, modelling, internal combustion engines*

Introduction

In recent decades, exhaust emissions of road vehicles have decreased considerably due to the strict emission standards introduced by different countries [1]. These standards urged researchers to develop cleaner operating vehicles and among others to focus on combustion development, as a key factor in reducing raw emissions. In Diesel combustion processes injection plays the most important role. This is why more emphasis was put on injection development compared to previous times. Many new high pressure systems have been designed, but it was realized that not only growing injection pressure can refine combustion. The shape of the rate of injection, combined with controlled in-cylinder flows, can influence combustion processes essentially and throughout this the efficiency, emissions and power output of the engine.

Introduction of the common rail (CR) injection systems was one of the most important steps in injection developments. The CR systems offer flexibility in injection pressure, timing and length under any engine operating point [2]. The most advantageous feature of CR systems is injection timing, and the orifice opening is separated from pressure genera-

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tion. Thus injection event is independent of the engine speed or load. Due to this feature, the pressure drop at low engine speeds and loads, particular to conventional injection systems, could be avoided [3]. This flexibility, coupled with the possibility of multiple pre, main and post-injections allowed more control over the fuel mixing process and through this over combustion [4]. Although this flexibility and independence can make Diesel engine operation cleaner and more efficient, it makes the set-up of the system time and resource consuming. Furthermore, complex operation, high speeds and small scales make injector processes more difficult and challenging to measure. The best answer to these difficulties is a detailed injector model, where all internal parts are represented, and the hydraulic, mechanic and electromagnetic systems are properly described. Through such a model, the internal processes can be analyzed, details can be described, and development can be made easier.

As CR systems became more popular, many different injector models appeared in the literature. Some models were very complex and contained the whole injection system, including the high pressure pump and the rail pipes [5], but most works concentrated only on the injector itself. Control oriented simplified models were created [6, 7], as well as detailed ones, the latter mostly contained mechanical and fluid dynamical calculations [8-12]. Electromagnetic circuits can be rarely found in these publications, but the work of Bianchi *et al.* [13-16] has to be highlighted, as a complete detailed and validated CR injector model, including electromagnetic simulations. If electromagnetic parts are neglected, solenoid coil force is substituted by interpolation of force data acting on the anchor, defined by measurements [17, 18].

An accurate fluid dynamic model is one of the most important factors in CR injector modelling. Most of the works contain cavitation models. Hence it has a crucial effect on the discharge coefficient of the orifices and through this on the mass-flow rate during injection events. This is why much emphasis was put on describing and determining this phenomenon in various orifice geometries [19].

Nearly all simulations contained mechanical models of the elastic axial deformation of the needle and control piston, along with the injector body [20]. This phenomenon is also very important because it changes effective needle displacement and the needle-seat passage area near the orifice holes.

Based on the described references, the CR injector employs three model types: an electro-mechanical, a hydraulic and a mechanical. If one's goal is to create a predictive injector model, it is needed to consider all three parts in a simulation and validate the results against measurement data.

There are several methods to measure injector operating parameters, but probably the most representative measurement type is injection rate measurement. Two measurement methods have spread in research and development to accurately specify the mass flow rate during injections. One was developed by Zeuch [21] and contained a relatively small device on which injectors could be changed quickly. Therefore it was more suitable for larger test numbers. As a disadvantage, it had a complicated operation and measurement method. The other was published by Bosch [22] and is used more widely, although it requires more space and injectors can not be so changed easily between measurements. Although it uses only a pressure sensor for measurement, therefore it is easier to use. Because the purpose of this work is not to improve or verify injectors, but to validate a model, therefore there is no need to change injectors, so the latter was chosen and built based on the parameters defined in [23].

The modelled injector

The model described below was based upon a Bosch CRIN1 type injector for commercial vehicles and its build-up follows the structure of the injector strictly. Figure 1 shows the cross-section of the injector, where the relevant parts are labelled by numbers.

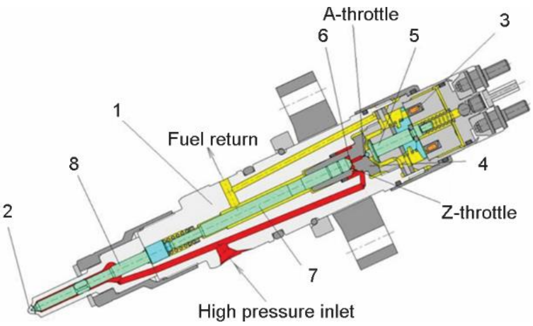

Figure 1. Cross-sectional view of the Bosch CRIN1 injector

The injector body – 1 holds together the multi-hole, valve covered orifice (VCO) type nozzle – 2 and the solenoid coil assembly – 3. It also contains the ball valve – 4, the control chamber – 6 with the control piston – 7 and the two calibrated orifices (A- and Z-throttles). These orifices control the pressure level in the control chamber – 6 during operation with the help of the ball valve, which opens and closes the hole of the A-throttle. The ball valve is operated by the armature of the solenoid assembly – 5. Control piston is in direct physical connection with the nozzle needle (8), which is pushed down by the needle spring to keep the nozzle holes closed. High-pressure lines are indicated with red color, while fuel return side is colored yellow. Detailed working principle of the injector is described in [2].

Simulation model

A hybrid model containing electromagnetic, hydraulic and mechanical parts was built up to maintain accuracy and predictivity and study the injectors internal processes. The structure was first presented in [23] and then validated against control piston lift measurements in [24], but injection rate validation was needed to make it predictive. The model also contains the simulation of the injection rate meter, so pressure traces from simulation and measurements could be directly compared, therefore the calculated injection rate from simulation and measurements could also be set together with the simulated nozzle flow.

The model was implemented in a commercial simulation software named GT-Suite, in the GT-Fuel submodule. It is capable of calculating problems in different disciplines of physics, *i. e*. electromechanical, thermal, fluid-dynamical and mechanical simulations [25].

Injector model

The model contained hydraulic, mechanical and electromagnetic parts. The build-up and the parameters will be thoroughly discussed further in the paper.

Hydraulic model

The hydraulic model strictly follows the injector layout. This means that all volumes downstream the high pressure tubes coming from the rail, up to the nozzles are modelled, containing all internal flow passages. The rail tube and high pressure pump are replaced by an unsteady pressure boundary condition, which was measured in the rail during the injection event. Thus the hydraulic system of the injector has been modelled as a network of pipes and chambers connected by orifices, higher-level components from the software model library were used to calculate hydraulic forces caused by fuel pressure and flow, fuel leakage is also modelled at the joint surfaces. The calculation is based on 1-D, unsteady, compressible flow and takes into consideration the dynamics of the attached mechanical components and structural heat transfer.

Figures 2 and 3 show the hydraulic and mechanical models of the injector, where fig. 2 presents the system of the injector body, while fig. 3 shows the solenoid assembly. All parameters needed in fluid-dynamic equations were measured, tab. 1 contains the most important orifice geometries.

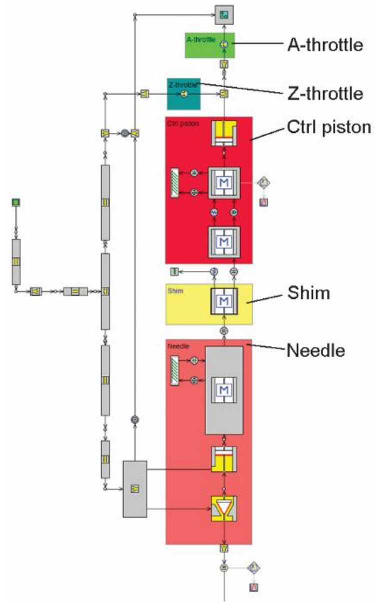

Figure 2. Hydraulic, mechanical, and electromagnetic model of the injector body

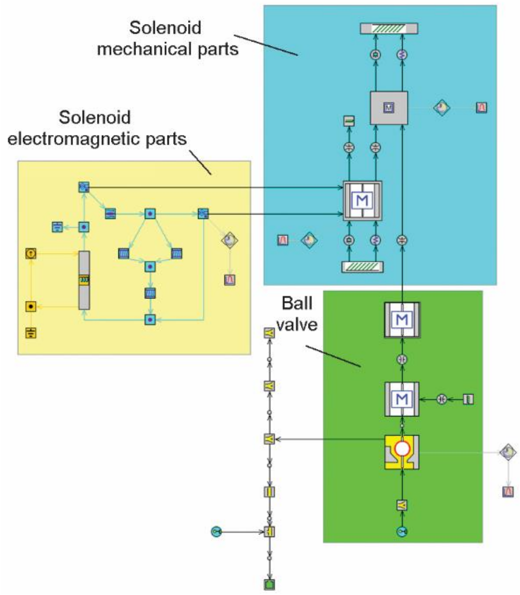

Figure 3. Hyfraulic, mechanical, and electromagnetic models of the solenoid assembly

Because temperature and pressure can remarkably vary in an injector during operation, it is very important to model fuel properties accurately [26]. The ISO4113 test oil was used as fuel in the simulations, the density, dynamic viscosity and bulk modulus of which was given as functions of pressure and temperature.

Table 1. Geometric parameters of orifices D_h *– hole diameter, L – hole length, and r – inlet corner radius*

	D_h [μm]	L [mm]	r [μm]
Injector orifice hole	152	1	20
Hole A	268	0.6	65
Hole Z	220	0.47	55

Mechanical model

Mass-spring-damper scheme is used to model moving mechanical parts of the injector. Masses in the simulation environment may translate in planar directions (x, y and angular velocity components), equations are based on Newton's second law and calculated according to the coordinate directions. Viscous, elastic and body forces are also taken into consideration, and external forces were also applied *e.g.* to the anchor of the solenoid.

As mentioned before, the working pressure range of a CR injector causes considerable deformation of the parts on the high pressure side. Therefore modelling of this phenomenon has critical importance on the accuracy of the simulation. Most important parts which experience such deformation are the needle, control piston and injector body. To simulate the change of the actual needle stroke due to deformation, material stiffness and damping was reduced into one element, namely the control piston. As fig. 2 shows, the injector needle is handled as a rigid body, but the control piston was split into two pieces with identical masses, and spring stiffness with a damping element was defined between them. These could be described as the resultant stiffness and damping of the system, which were defined using measurement results of the needle lift sensor and line pressure sensor. The needle lift sensor measures the displacement of the control piston while using pressure sensor value and knowing the diameter of the control piston, pressure force can be obtained. If deformation and force are known, resultant stiffness of the system can be calculated, this way, the laborious work of defining the stiffness of all different parts can be eliminated, but accurate results are provided.

Evaluation of the damping factor is far more difficult, considering that the damping factor shall include elements not only by the internal friction but also from fluid viscosity and friction between piston and liner. Experimental evidence shows that the friction component of the damping is more important, but it cannot be evaluated theoretically since machining tolerances affect it mostly. Therefore damping must be estimated during model tuning phase.

Electromagnetic model

Electromagnetic circuit of the model is responsible for transforming unsteady input current boundary condition to mechanical force. The output of the system is the magnetic force acting on the anchor of the solenoid coil, which is an input condition in the mechanic model part, as an imposed force on anchor mass. Dynamic behavior of the anchor is calculated based on the magnitude of this force, while the magnitude is calculated based on the reluctance of the circuit. When current begins to flow in the coil, magnetic flux is generated in the elements, and electromagnetic force is calculated between surfaces of the air gaps. Explanation and working principles are detailed in [27].

Injector rate meter model

Working principle of the Bosch method lies in recording pressure change in a length of compressible fluid during an injection event. The pressure wave is produced by the injected fluid, while the magnitude of the pressure change is proportional to the injected mass flow rate:

$$\frac{\mathrm{d}m}{\mathrm{d}t} = \frac{Ap}{c} \tag{1}$$

where dm/dt is the instantaneous mass-flow rate, A – the cross-sectional area of the measuring tube, p – the pressure of the fluid, and c – the speed of sound. The detailed explanation of the working principle and build-up, along with geometric parameters of the Bosch type injection rate meter was published in [23].

Figure 4 shows the injection rate meter model, which follows the structure of the existing device strictly. Fuel is injected into the injector mount – 1, which is connected to the measuring tube – 2. The measuring and following tubes – 4, are separated by an adjustable orifice – 3, while the check valve on the end of the following tube is substituted with an infinite volume environment – 5, to eliminate the dynamic model of a complicated valve-spring-damper mechanism with flow modelling. Pressure sensor – 6, is placed in the injector mount, while the rest of the model calculates mass flowrate according to eq. (1). This set-up allows to simulate the measurement accurately, so measured and simulated calculated mass flow rates can be compared to simulated injector flowrate.

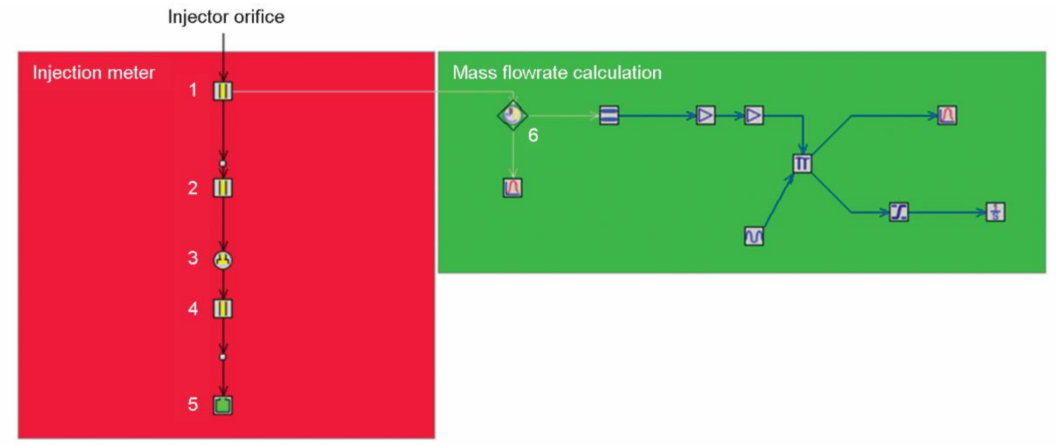

Figure 4. Model of the injection meter and mass-flow rate calculation

Measurements

Measurements were made using a turbocharged medium-duty Diesel engine installed on a test bench [28]. Due to the specific high pressure fuel connection of the CRIN1, an injector mount would be challenging to manufacture, so a cylinder head similar to the one on the engine was used to accommodate the injector, fig. 5. In the measurement set-up, the engine and its ECU was used to drive the measured injector through a flexible high pressure fuel hose, this way the engine ran with three cylinders, fourth being connected to the measured injector. The measured injector was equipped with a needle lift sensor, measuring control piston movement, and a current clamp was used to record the driving current of the injector. The measuring tube was mounted on the cylinder head with the help of an adapter unit, which contained the pressure transducer of the injection rate meter. Another pressure sensor was used to measure rail pressure, so all boundary conditions of the test cases could be recorded. An adjustable orifice was mounted on the measuring tube to separate it from the following tube. A check valve with adjustable backpressure was closing the measurement line.

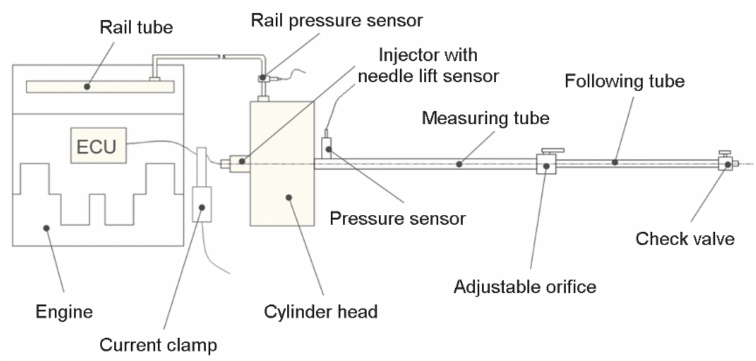

Figure 5. Measurement layout

Four test cases were defined to cover the widest possible range of injector operation from partial loads to full load. Table 2 shows that rail pressure varied from 450 to 900 bar, while excitation time from 1 to nearly 3 ms. Due to the measurement set-up, it was not possible to change line pressure or excitation time directly, just through engine operation point changes. Thus the injection parameters were defined by the predefined ECU tables.

Table 2. Test case parameters

Test case	Engine speed [rpm]	Engine torque [Nm]	Injection pressure [bar]	Excitation time [ms]
1	1400	400	890	2.7
2	1500	300	680	2.55
3	1500	200	540	1.9
4	1500	100	450	1.15

Results and discussion

In previous works, the model had been validated against needle lift measurements with accurate results [23, 24, 27], but injection rate measurements were still needed to make the complete model predictive. The four test cases cover an adequate range of operating points with different rail pressures and opening times from partial to full loads in the critical operating range of an engine. Measured and simulated injection rates can be followed in figs. 6-9. On each of the figures, three traces are depicted: the injection rate calculated based on the measured pressure fluctuation denoted by blue, the same calculation done from simulation results denoted by red and the simulated injector nozzle flowrate with green color.

It is worthwhile to mention that measurement data contained a considerable amount of noise. In a small scale, high-speed flow like fuel injection, it is crucial to avoid time and phase shifts made by different filters, so noise was eliminated averaging injection event pressure traces. At least a hundred injections were recorded for every test case and fit on each other to calculate an average for every time step. This method gave satisfying results, so simulation and measurement results could be compared.

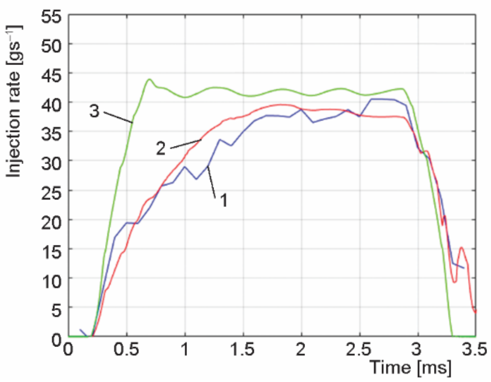

Figure 6. Measured and simulated injection rate traces, Test case 1 (1400 rpm 400 Nm);
1 – injection rate – measurement, 2 – injection rate – simulation, 3 – nozzle injection rate – simulation

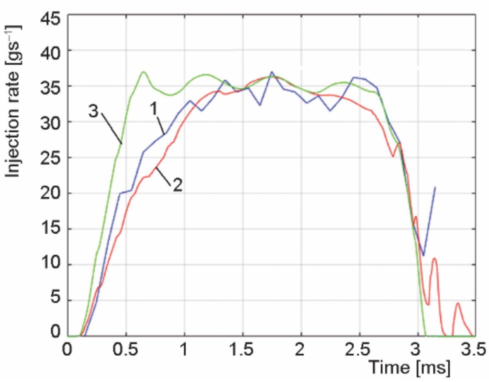

Figure 7. Measured and simulated injection rate traces, Test case 2 (1500 rpm 300 Nm);
1 – injection rate – measurement, 2 – injection rate – simulation, 3 – nozzle injection rate – simulation

Based on figs. 6-9, two main conclusions can be pointed out. From the model validation point of view, it is very important that measured and simulated calculated injection rate traces show good conformity. According to tab. 3, root mean square (RMS) errors of all test cases stay below 9%. Thus, simulation results are accurate and describe injector and injection rate meter processes precisely. The RMS errors were calculated according to:

$$\varepsilon_i = \sqrt{\frac{1}{T}\int_0^T \left(\frac{m_{i,\text{meas}} - m_{i,\text{sim}}}{m_{i,\text{meas}}}\right)^2} \tag{2}$$

where ε_i is the RMS error in the i^{th} test case, T – the time range, $m_{i,\text{meas}}$ – the measured injection flowrate, and $m_{i,\text{sim}}$ –the simulated flowrate in the i^{th} test case, based on pressure change calculation.

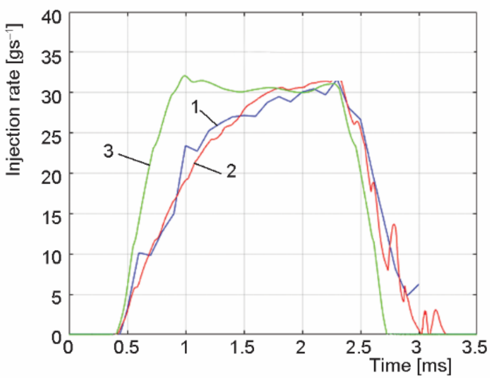

Figure 8. Measured and simulated injection rate traces, Test case 3 (1500 rpm 200 Nm) ;
1 – injection rate – measurement, 2 – injection rate – simulation, 3 – nozzle injection rate – simulation

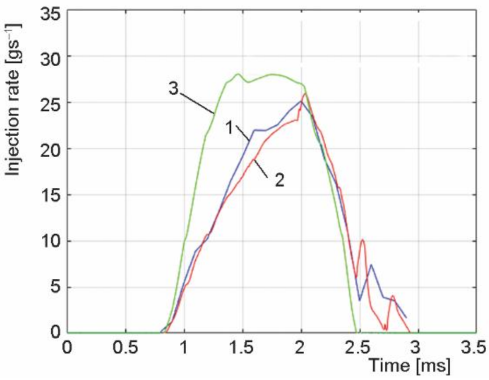

Figure 9. Measured and simulated injection rate traces, Test case 4 (1500 rpm 100 Nm);
1 – injection rate – measurement, 2 – injection rate – simulation, 3 – nozzle injection rate – simulation

Table 3. The RMS errors of the test cases

Test case	1	2	3	4
The RMS error	0.087	0.093	0.078	0.080

The second main conclusion is that the causes and effects remarkably differ in such a setup. The Bosch type injection meter was considered to indicate the exact injection rate, but the results show that the *cause*, the injector nozzle flow rate, and the *effect*, the subsequent pressure variation is different in every test case. This is mainly due to the bulk modulus and damping of the system, including fuel compressibility, pipe deformation, *etc.* If one examines the curves closely, it can be concluded, that neither dynamics nor the quasi-steady state values match the nozzle flow rate. When injection flowrate is increasing, the measurable injection rate lag behind considerably, and when the flowrate reached its steady-state value, the calculated value is still changing. In Test cases 2 and 3, the calculated value reaches the nozzle flowrate before the end of the injection, but in cases 1 and 4, they do not match. In Test case 4 the reason for this might be slower pressure dynamics, meaning that the calculated injection rate does not have time to reach a steady-state value, but Test case 1 is different. Here calculated injection rate reaches steady-state, but it shows more than 10% difference compared to nozzle flow. This might be because of the non-linearity of fluid dynamics and pipe deformation, and this is why Bosch type injection rate meter gave accurate values only for an interval of dosage and for different injectors different meters had to be constructed.

Finally, it can be stated, that Bosch type injection rate meter is not adequate to directly validate a simulation, but if the simulation is tuned according to the injection rate meter results, the actual nozzle flowrate which causes the pressure rise can be determined. According to the authors' knowledge, there is no other method that could give satisfactory results using the Bosch type meter.

Conclusion

The model of a CR injector and a Bosch type injection rate meter was presented and validated against injection flowrate measurements. The simulated and measured flowrate traces based on calculation showed a good fit, but a remarkable difference was pointed out between the injection rate measurement and the injector nozzle flow. This non-linear deviation makes the Bosch method inadequate for direct model validation, but if the rate meter is also part of the simulation, the injector model can be tuned according to the calculated mass-flow rate. Thus, if the measured and calculated flowrates agree, the nozzle flow in the simulation will be accurate as well.

Nomenclature

A – cross sectional area, [m^2]
c – speed of sound, [m/s]
D_h – hole diameter, [μm]
L – hole length, [mm]
m – mass, [kg]
$m_{i,\mathrm{meas}}$ – measured injection flowrate in the i^{th} test case, [gs^{-1}]
$m_{i,\mathrm{sim}}$ – simulated injection flowrate in the i^{th} test case, [gs^{-1}]
p – pressure, [Pa]
r – inlet corner radius, [μm]
T – time range in the i$^{\mathrm{th}}$ test case, [s]
t – time, [s]

Greek symbol

ε_i – root mean square error in the i$^{\mathrm{th}}$ test case, [–]

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