

Dynamic Modelling of Common Rail Diesel Engines Combined with Amesim and Simulink

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ABSTRACT

The fuel injector of the diesel engine is the core component of the high-pressure common rail system of the diesel engine. The ECU performs flexible control on the fuel injector to realize the precise adjustment of the fuel injection rate and fuel injection quantity of the engine under different working conditions, and achieve the purpose of reducing fuel consumption and noise. The internal structural parameters of the fuel injector have a significant impact on the fuel injection rate, and it is often necessary to make a large number of modifications to the structural parameters of the fuel injector in order to meet the increasingly stringent emission regulations. In this paper, in order to analyze the effect of various structural parameters of fuel injectors on fuel injection rate, increase fuel injection rate, and obtain the best effect, AMESim and simulink software are used to build a simulation model of a common rail fuel injection device and a diesel engine. This model provides a theoretical basis for the structure design of the fuel injector.

Keywords : Mathematical Modelling, Common Rail, Diesel Engine, Amesim, fuel injector

I. INTRODUCTION

The characteristics of fuel injection of diesel engine play an important role in the performance and exhaust emission of diesel engine. In common rail fuel system, the characteristics of fuel injection is concerned both the structure parameters of injector and control strategy of control system (or electronic control unit-ECU). Meanwhile, the speed of the engine and the

pressure in the rail pipe are feed to ECU, and the ECU issues commands to adjust fuel injection time and fuel injection duration according to the signal received and interior control strategy. The sub-models, which represent high pressure fuel system and engine performance in common rail diesel engine, have been described with AMESim code, and fuel injection flow rate and cylinder pressure obtained by measurement are used to validate the sub-models[1]. A mathematical

common rail injection system model has been developed using the Simulink/Matlab code to execute simulation of the high-pressure Common Rail (HPCR) injection system[2]. A methodology to mathematically model a common-rail injection system by using a Bond Graph approach and AMESim commercial software is described in the literature[3]. In literature [4], modelling and common-rail pressure control of high pressure common rail injection system (HPCRIS) is presented the mathematical model of high pressure common rail injection system which contains three sub-systems: The high-pressure part of a research type Heavy Duty Common Rail (CR) fuel injection system reports on the observed dynamics of the injection system and the capability of the model to capture this[5]. In literature[6], a piezo-driven diesel injector, as a new method driven by piezoelectric energy, has been applied with a purpose to develop the analysis model of the piezo actuator to predict the dynamics characteristics of the hydraulic component (injector) by using the AMESim code and to evaluate the effect of this control capability on spray formation processes. It details how to model the most important part of an injection system, with an emphasis on the most important component of the system, the injector itself. The performance of the high-pressure fuel injection system and its effect on diesel engine combustion are greatly influenced by the injector characteristics and the settings of the overall equipment control system [7]. In literature [8], a high-pressure system based on a common rail structure was analyzed using AMESim. A dynamic modelling, comprehensive experimental and numerical investigation has been performed to determine the influence of operating parameters and critical injector design parameters on the dynamic performance of advanced high-pressure electronically controlled diesel injection systems[9]. A one-dimensional model of a solenoid diesel injector was developed with AMESim to numerically investigate the individual effects of fuel viscosity and density as a function of fuel temperature on the injection rate of

the solenoid diesel injector [10]. Reliable design, optimization, and analysis of the entire mechatronic system, including the actual process and control system of Common Rail Injection Systems, were conducted in the AMESim environment [11]. In this way, in previous literatures, various assumptions, mathematical models, and programs were used to create a dynamic model of common rail injection in diesel engines. In this study, a mathematical modelling method for analyzing the dynamic characteristics of Common Rail Injection of a diesel engine by combining Amesim and Simulink was specifically mentioned. At first the sub-models, which represent high-pressure fuel injection system and engine performance in common rail diesel engine, have been described with AMESim code. Then the sub-models, together with the sub-model of control system which is made with Matlab/Simulink, have been combined and a powerful complete co-simulation model is established.

II. Mathematical modelling

The scheme of common rail system in diesel engine is shown in Fig. 1. The fuel is supplied from fuel tank to rail pipe by fuel supply pump. The rail pressure sensor in the rail pipe measures pressure and feeds back to ECU, and ECU controls the open and close of solenoid valve in fuel supply pump and fuel injector according to receiving signal of rail pressure, speed of engine, intake pressure etc. The main components of fuel supply system are fuel injector, fuel supply pump and rail pipe.

2.1 Mathematical modelling of common rail fuel injection system

2.1.1 Mathematical modelling of injector

The injector is a complex system composed of machine, electricity and hydraulics. The model of injector are divided into the solenoid valve control model, the volume chamber model and the needle valve model, which is as following.

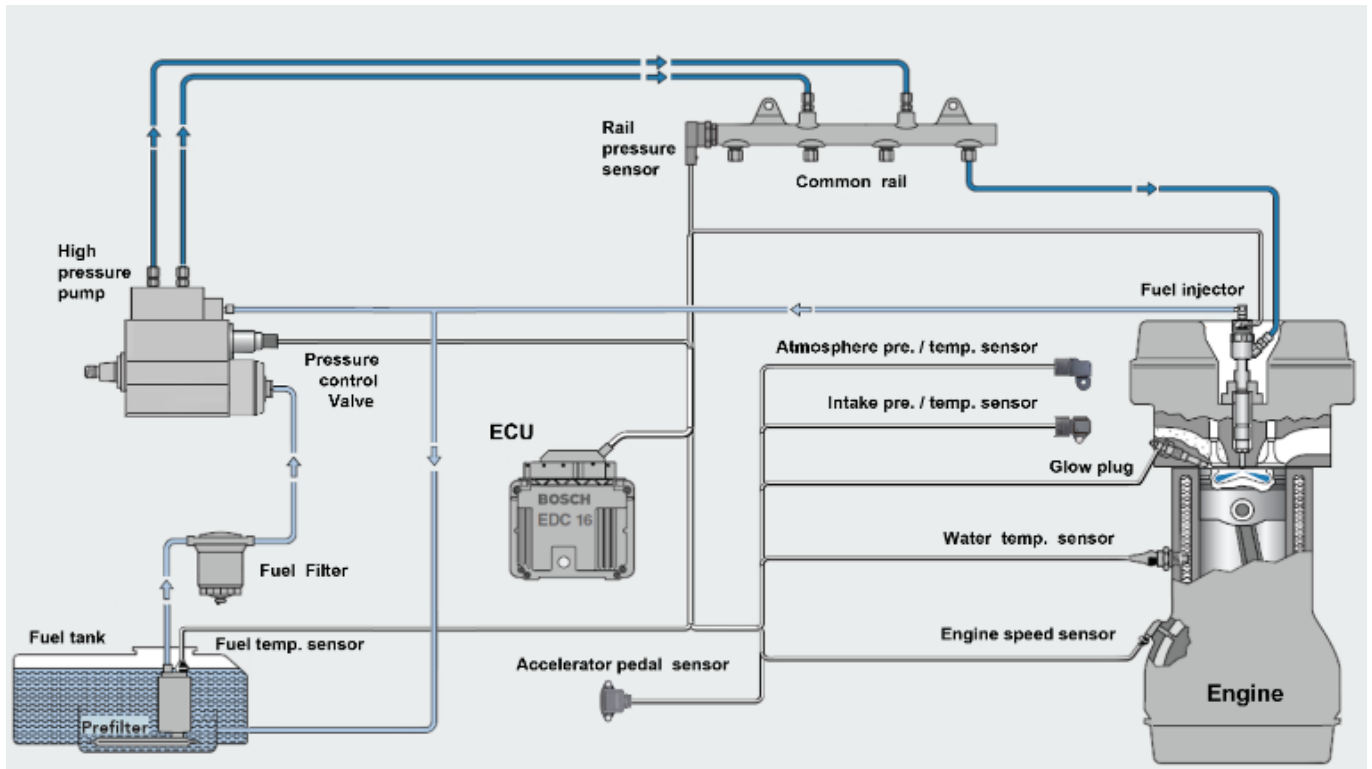


Fig. 1 Scheme of common rail system in diesel engine

2.1.1.1 Solenoid valve control modelling

The mathematical model of the solenoid valve includes circuit equations, electromagnetic force equations and armature motion equation.

~ Circuit equation

$$u = i(t) \cdot r + \varphi \frac{dN}{dt} \tag{1}$$

Where u-driving voltage of electromagnet

i- current in the coil

r- internal resistance of the coil

N- number of turns in the coil

φ- magnetic flux

During the working process of the electromagnet, electromagnetic eddy current will be generated due to the phenomenon of electromagnetic induction, which will not be considered here for simplicity

~ Electromagnetic force equation

$$F_m = \frac{1}{2} \cdot u_\delta^2 \cdot \frac{dG}{d\delta} \tag{2}$$

Where F_m - electromagnetic force

u_δ - air gap magnetic drop

G- air gap magnetic permeability

δ- air gap.

When the air gap in the system is small, the formula can be simplified to:

$$F_m = \frac{1}{2} \cdot \frac{B^2 \cdot S}{u_0} \tag{3}$$

Where B- air gap magnetic induction

u_0 - air magnetic permeability

s - air gap magnetic permeability area

~ Motion equation of armature

$$F_m + F_p - F_s = m \frac{d^2x}{dt^2} \tag{4}$$

Where F_p - effective fuel force which the armature is subjected in the pressure relief chamber

F_s - spring force

m- quality of the armature

x- displacement of the armature.

2.1.1.2 Volume chamber modelling

The volume modulus of elasticity E which expresses the compressibility of the fuel:

$$E = - \frac{dP}{dV/V} = \frac{dP}{d\rho/\rho} \tag{5}$$

Where V - instantaneous volume of the chamber

dV/dt - volume changes caused by mechanical parts piston, the intake and the outtake flows.

The increase of dP causes the volume decrease of a unit volume of liquid(dV). dV/V is dimensionless. From Eq. (5), the time derivative of the fuel pressure can be obtained:

$$\frac{dP}{dt} = -\frac{E}{V} \cdot \frac{dV}{dt} \tag{6}$$

Assumption:

- ① The control chamber, the oil storage chamber, the needle valve seat and the pressure chamber of nozzle in the common rail injector can all be regarded as lumped volumes.
- ② Only the volume is considered, the shape is not considered, and the internal pressure and density of the volume chamber are considered to be the same everywhere.

Using the lumped volume method, the mathematical model of the volume chamber is established by flow conservation equation.

$$\frac{V}{E} \cdot \frac{dP}{dt} + \frac{dV}{dx} = \sum Q_i \tag{7}$$

Where V- volume of the volume chamber

E- the modulus of elasticity of the fuel

P- internal pressure of the volume chamber

t- time

Q_i – the sum of fuel flow rate of inflowing or outflowing in the *i*th volume chamber per unit time
 Considering the factors affecting the fuel volume change, Eq. (7) can be changed as follows:

$$\frac{dP}{dt} = -\frac{E}{V} \cdot \left(\frac{dV_0}{dt} - Q_{in} + Q_{out} \right) \tag{8}$$

This is the basic pressure dynamics equation in each control volume.

Where Q_{in} - intake flow

Q_{out} - outtake flow

dV₀/dt is the volume change rate caused by the mechanical piston, which will be specified for the high pressure pump, the common rail pipe and the injector, respectively.

Except the high pressure pump, all the elements in HP CRIS have a constant volume.

Based on the energy conservation law, intake and outtake flows Q_{in} and Q_{out} can be expressed as follows:

$$Q = \text{sign}(\Delta P) \cdot \mu \cdot S_0 \cdot \sqrt{\frac{2|\Delta P|}{\rho}} \tag{9}$$

Where sign(ΔP)-sign function affecting the flow direction

$$\text{sign}(\Delta P) = \begin{cases} 0 & \text{if } P_p \leq P_{cr} \\ 1 & \text{if } P_p > P_{cr} \end{cases} \tag{10}$$

μ - discharge coefficient

S₀ - orifice section area

ρ - dual density

ΔP - fuel pressure difference across the orifice

From Eq. (7), the fuel pressure dynamics of injector is shown as

$$\frac{dP_{injk}}{dt} = \frac{E}{V_{injk}} \cdot (Q_{crik} - Q_{cylk}) \tag{11}$$

Where P_{injk} -pressure on the injector

V_{injk} -volume of the kth injector

Q_{crik} -injection flow of kth injector

Q_{cylk} -injection fuel flow from the kth injector

to cylinder, which is shown as follows:

$$Q_{cylk} = \text{sign}(P_{injk} - P_{cylk}) \cdot E_{Tk} \cdot \mu_{ik} \cdot S_{injk} \cdot \sqrt{\frac{2(P_{injk} - P_{cylk})}{\rho}} \tag{12}$$

Where P_{cylk} - cylinder pressure ,

μ_{ik} - discharge coefficient

E_{Tk} - square signal that equals to 1 during injections , Anything else is 0.

S_{injk} - orifice area of the injector nozzle.

2.1.1.3 Needle valve modelling

According to Newton's second law, the motion equation of the needle valve parts is as followings:

$$m_z \cdot \frac{d^2h}{dx^2} = F_z + F_y + F_m + F_k - F_c - C \cdot \frac{dh}{dt} \tag{13}$$

Where F_z - fuel pressure on the needle seat

F_y - fuel pressure on the needle sac volume

F_m - fuel pressure on the accumulation volume

F_k - spring force on the needle valve

F_c - fuel pressure on the control chamber

c - damping coefficient of the needle valve

h - lifting displacement of the needle valve

m_z - mass of needle valve parts

2.1.2 Mathematical modelling of high pressure pump (HPP)

Since the pump is driven by the camshaft, it varies according to engine speed. It is connected by a small orifice to the low pressure circuit and by delivery valve

with a conical seat to the high pressure circuit. According to Eq. (7), the fuel pressure in the high pressure pump can be expressed as followings:

$$\frac{dP_p}{dt} = -\frac{E}{V_p} \cdot \left(\frac{dV_p}{dt} - Q_p - Q_{pcr} + Q_{pl} \right) \quad (14)$$

Where Q_p -inlet fuel flow

Q_{pcr} -outlet fuel flow to the common rail

Q_{pl} -fuel leakage flow

P_p -fuel pressure of high pressure pump.

The volume of the high pressure pump V_p changes according to the camshaft motion can be expressed as followings:

$$\frac{dV_p}{dt} = -S_p \omega \frac{dh_p}{d\theta} \quad (15)$$

Where S_p -sectional area of plunger, $S_p = \pi d_p^2 / 4$

d_p -diameter of plunger

h_p -plunger lift

ω -camshaft rotational speed

θ -camshaft angle.

Q_{pcr} is the flow from high pressure pump to common rail, which is given as followings:

$$Q_{pcr} = \text{sign}(P_p - P_{cr}) \cdot \mu_p \cdot S_{pcr} \cdot \sqrt{\frac{2}{\rho} |P_p - P_{cr}|} \quad (16)$$

Where P_{cr} - fuel pressure on the common rail

μ_p - discharge coefficient that changes according to the pressure ratio in steady working condition of diesel engine.

S_{pcr} - high pressure pump cross-sectional area of outlet port

P_{cr} - common rail fuel pressure

The sign function determines the flow direction, when $P_p \leq P_{cr}$, the check valve (one-way valve) between the high pressure pump closes and prevents the fuel in the rail flow back to high pressure pump. The fuel leakage flow Q_{pl} can be regarded as a constant.

2.1.3 Pressure change modelling of common rail

The pressure change modelling of common rail is obtained by considering the balance between the high pressure pump inlet flow and injectors' outlet flows. Based on Eq. (6), the common-rail fuel pressure dynamics can be modeled as followings:

$$\frac{dP_{cr}}{dt} = \frac{E}{V_{cr}} \cdot (Q_{pcr} - Q_{cri} - Q_{crb}) \quad (17)$$

Where V_{cr} - constant volume of common rail pipe

Q_{cri} - outlet fuel flow of the common rail pipe which is equal to the sum of inlet fuel flow of injectors.

Q_{crb} -return flow of common rail which is considered as constant.

$$Q_{cri} = \sum_{k=1}^4 Q_{crik} \quad (18)$$

The injection flow(Q_{crik}) of k-th injector can be expressed as followings:

$$Q_{crik} = \text{sign}(P_{cr} - P_{injk}) \cdot \mu_{injk} \cdot S_{crik} \cdot \sqrt{\frac{2}{\rho} |P_{cr} - P_{injk}|} \quad (19)$$

Where μ_{injk} - discharge coefficient

S_{crik} - cross-sectional area of the inlet port in the k th injector.

2.2. Mathematical modelling of diesel engine

In the engine performance simulation, the most important sub-model is combustion model and turbocharger model.

2.2.1 Combustion model

In combustion model, the combustion heat release is calculated using a model by Barba[2]. After the beginning of the fuel injection, an auto-ignition delay is computed using a model based on an Arrhenius law. When the auto-ignition delay is reached, the combustion starts. The injected fuel in pre-mixed zone burns with a premixing model, whereas the remaining fuel burns with a diffusion model. The model could compute up to 3 different injections. For each injection an auto-ignition delay is defined and the injected fuel could burn either in a pre-mixed or a diffusion zone. The effects of the residual burned gases are taken into account in the calculations of the auto-ignition delay. The equation for the auto-ignition delay is presented below:

$$\theta = (1 + 10X_{res}) A_t p^{-n} e^{-\frac{T_a}{T}} \quad (20)$$

Where T- temperature in the combustion chamber

P- pressure in the combustion chamber

X_{res} - molar fraction of the burned gases in the combustion chamber before the combustion

A_t - gain for auto-ignition delay

n -power gain for auto-ignition delay

T_a -activation temperature for ignition delay
 The pre-mixed combustion is divided into two mode. The first burning mode is defined as the propagation of a flame in a homogeneous zone. This zone is supposed to be spherical and the propagation of the flame is computed using a standard flame propagation model [3]. Using the hypothesis of a spherical shape for the flame, the combustion heat release rate $\frac{dQ_{pre1}}{dt}$ produced by this burning mode is:

$$\frac{dQ_{pre1}}{dt} = H_u \bar{\rho}_f^u U_T S_L \quad (21)$$

Where H_u - fuel low heating value

$\bar{\rho}_f^u$ - averaged density of the fuel in the unburned part of the pre-mixed zone.

U_T -turbulent flame speed, computing using the method of Metghalchi and Keck correlation.

S_L -laminar flame surface area

With the local evolution of the Fuel/Air equivalence ratio in the pre-mixed zone, the evolution of the mass of the gas in the burned and unburned zones has been computed. In addition to this phenomenon, Barba defines a second burning mode, which is representative of the multiplication of the ignition location in the pre-mixed zone. This multiplication induces a reduction of the total flame surface. The combustion heat release rate for this burning zone $\frac{dQ_{pre2}}{dt}$ is defined as followings:

$$\frac{dQ_{pre2}}{dt} = \frac{1}{3} H_u \frac{(m_f^{ug})}{m_{fg} + m_{rg}} \frac{U_L}{R_z} \quad (22)$$

Where m_f^{ug} - available mass of fuel in the unburned zone

m_{fg} - mass of entrained fresh air in the zone

m_{rg} - mass of entrained residual burned gas in the zone

R_z -radius of the pre-mixed zone (with the hypothesis of a spherical shape).

The total combustion heat release rate for the pre-mixing combustion is the minimum of the combustion heat release rate for the burning mode 1 and 2, as:

$$\frac{dQ_{pre}}{dt} = \text{Min} \left(\frac{dQ_{pre1}}{dt}, \frac{dQ_{pre2}}{dt} \right) \quad (23)$$

The Barba turbulent combustion model [2] is connected to a mixing frequency defined with the

averaged piston velocity and the spray turbulent kinetic energy.

Using a frequency model to define the mixing controlled combustion, the combustion heat release rate $\frac{dQ_{diff}}{dt}$ becomes:

$$\frac{dQ_{diff}}{dt} = H_u f_m m_{diff} \quad (24)$$

Where f_m - mixing frequency

m_{diff} - available mass of fuel in the diffusion combustion zone.

A first approach is to consider the mixing frequency as the quotient of a characteristic mixing velocity and a characteristic mixing length. The characteristic mixing length L can be defined as followings:

$$L = \sqrt[3]{\frac{\Phi V_{cyl}}{n_N}} \quad (25)$$

Where Φ -averaged Fuel/Air equivalence ratio

V_{cyl} -current cylinder volume

n_N -number of holes of the injector

L -proportional to the cubic root of the current cylinder volume.

The actual Fuel/Air equivalence ratio takes into account that the characteristic mixing length is not only given by the actual cylinder volume but also by the concentration of oxygen. Finally, the number of nozzle holes influences also the mixing length. The characteristic mixing velocity is proportional to the turbulent kinetic energy in the chamber and to the piston averaged speed. The turbulence is mainly generated by the injected spray and defined with a creation and a dissipation term. The characteristic mixing velocity U_{mix} is the sum of the following two factors:

$$U_{mix} = \sqrt{C_G u_{pis}^2 + C_k k} \quad (26)$$

Where u_{pis} - averaged piston velocity

k -spray turbulent kinetic energy.

The two parameters C_G and C_k are important parameters for this model.

Finally, the combustion heat release rate for the mixing controlled combustion is defined as:

$$\frac{dQ_{diff}}{dt} = H_u \frac{\sqrt{C_G u_{pis}^2 + C_k k}}{\sqrt[3]{(\Phi V_{cy})/n_N}} m_{diff} \quad (27)$$

2.2.2 Characteristic definition of the turbocharger model

The turbocharger model is composed of sub-models of turbo and compressor and an inertia element. The compressor sub-model is ICECOMP02 model in AMESIM, which is a compressor model for multi-gas applications. The main input data for this sub-model is a performance map (Fig. 2) which follows the standard SAE format. This map usually does not cover the whole operating range of the compressor, so pre-processing functions are integrated in the sub-model to extend the range of the map to the critical regions (low and high speeds, surge, choke).

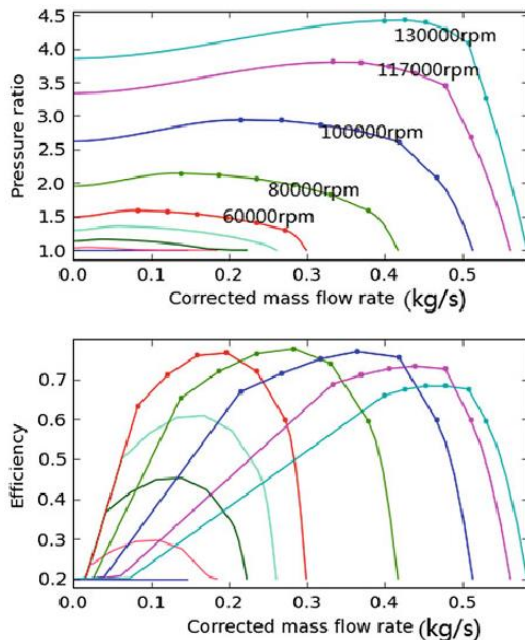


Fig. 2 Performance map of compressor

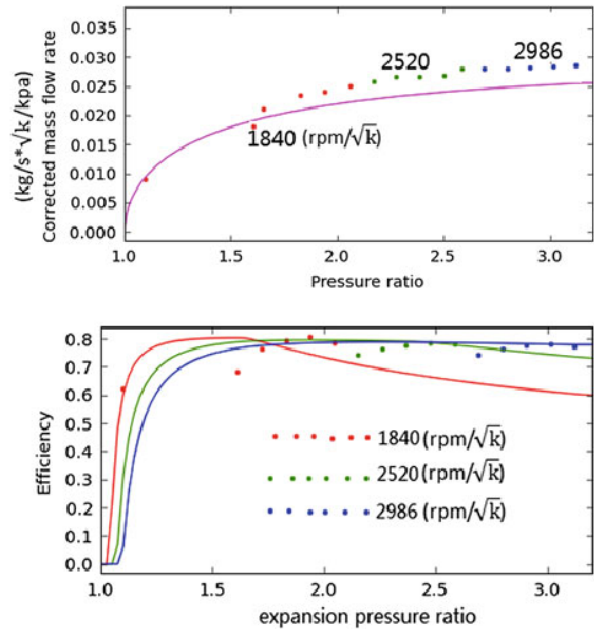


Fig. 3 Performance map of turbine

The turbo sub-model is CETRBSS02 model, which is a single-scroll turbine model for multi-gas applications. The main input data for this sub-model is a performance map (Fig. 3) which also follows the standard SAE format. This map usually does not cover the whole operating range of the turbine, so pre-processing functions are also integrated in the sub-model to extend the range of the map to a wider operating range.

2.3 Injection phase model

The Electronic Control Unit (ECU) activates each fuel injector 4 times per cycle. The fresh charge is internally prepared by injecting fuel twice, the first just before and the second just after the Top Dead Center at the beginning of the suction stroke. This early injection using the same high-pressure CR injection system allows for fuel heating and distribution and gives more time for fuel vaporization and mixing with air. The other two injections are the pilot and main injections that take place before and after the TDC at the start of expansion stroke. The following subscripts are used to substitute the subscript *i* in $\theta_{inj, i}$ or $dt_{inj, i}$

Where p- pilot injection

M- main injection

1_{pre}-first preparation injection

2_{pre} -second preparation injection

There are two preparations injections ($\theta_{inj, 1pre}$ or $dt_{inj, 1pre}$) and ($\theta_{inj, 2pre}$ or $dt_{inj, 2pre}$), pilot injection ($\theta_{inj, p}$ or $dt_{inj, p}$) and main injection ($\theta_{inj, M}$ or $dt_{inj, M}$).

The start and the duration of each injection are obtained by data regression of the experimental data concerning the measurement of the triggering signals of the injector. The pilot injection is just before the Top Dead Centre (TDC) at the end of compression stroke. Nearly 2ms later, the main injection takes place.

The start of pilot injection TS_p is calculated from the TDC at the end of compression stroke related to the engine speed and load by the equation.

$$TS_p [\text{ms}] = 0.001017 \times n + 0.00505 \times \text{load} - 0.32 \quad (28)$$

The pilot injection duration $dt_{inj, p}$ is calculated as a function of engine speed and load from the following equation:

$$dt_{inj, p} [\text{ms}] = (5.88 \times 10^{-5} \times n + 0.003222 \times \text{Load} + 0.223048) \times (0.00015 \times n + 0.002194 \times \text{Load} + 0.75495) \quad (29)$$

The start of main injection TS_M is 2ms later from the end of pilot injection. The main injection duration $dt_{inj, M}$ is calculated as a function of engine speed and load from the following equation.

$$dt_{inj, M} [\text{ms}] = ((0.00004 \times n + 0.002 \times \text{Load} + 0.58) \times (0.00015 \times n + 0.002194 \times \text{Load} + 0.75495)) \quad (30)$$

Two more short injections are timed just before the end of the exhaust stroke $\theta_{inj, 1pre}$ and just after the start of the suction stroke $\theta_{inj, 2pre}$. These to injection control the

preparation of the fresh homogenous charge for the next cycle. The Start of first preparation injection at the end of exhaust stroke of the previous engine cycle TS_{1pre} is adjusted 1.5ms before the TDC. The first preparation injection duration $dt_{inj, 1pre}$ equal 1ms at all speeds and loads. So the end of first preparation injection can be determined as 0.5 before TDC.

$$TS_{1pre}[\text{ms}] = 1.5\text{ms} \quad (31)$$

The start of 2nd preparation injection TS_{2pre} is 1ms later from the end of first preparation injection i.e 0.5 after TDC at the start of suction stroke. The second preparation injection duration $dt_{inj, 2pre}$ equal 1ms at all speeds and loads. The trigger signals before and after the TDC before the suction stroke, (1st and 2nd preparation injections) consist of multiples of very short successive triggers. To account for this discontinuity of the triggered signal a correction coefficient for the area of the injection nozzles during 1st and 2nd preparation injections is established as function of engine speed and load as

$$ACorr = -0.0000495 \times n + 0.001457 \times \text{Load} + 19624 \quad (32)$$

All time durations or starting times for injection can be converted to crank angle domain.

$$OS_i[\text{degree}] = 6 \times n \times TS_i [\text{ms}] \quad (33)$$

$$\theta_{inj, i} [\text{degree}] = 6 \times n \times dt_{inj, i} [\text{ms}] \quad (34)$$

In the above equations engine speed is in [rpm] and engine load is in [N.m]

III. Injection characteristic modelling of common rail diesel engine combined with Amesim and Simulink

3.1 Injection characteristic modelling of common rail diesel engine using Amesim

3.1.1 Injection characteristic modelling of common rail injection system using Amesim

The structure sketch of high pressure pump is shown in Figs. 4 is its AMESim model scheme.

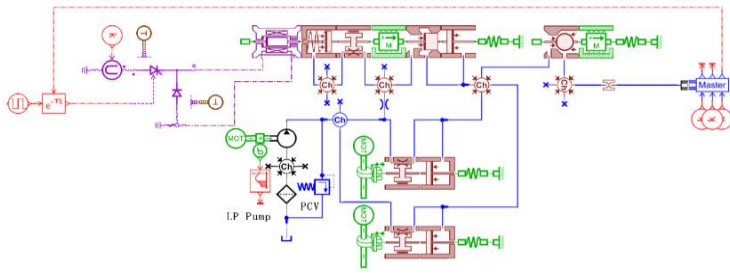


Fig. 4. AMESim model scheme on the high pressure pump

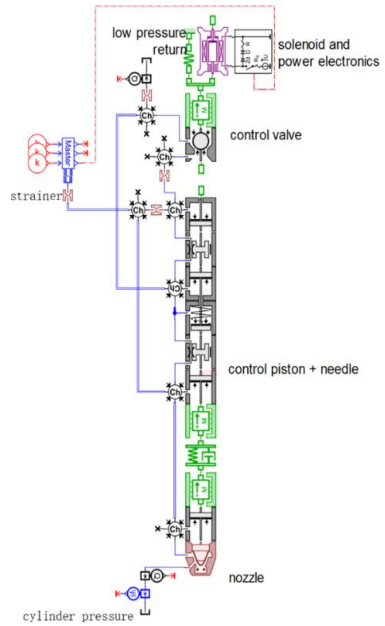


Fig 6. AMESim model scheme on the injection

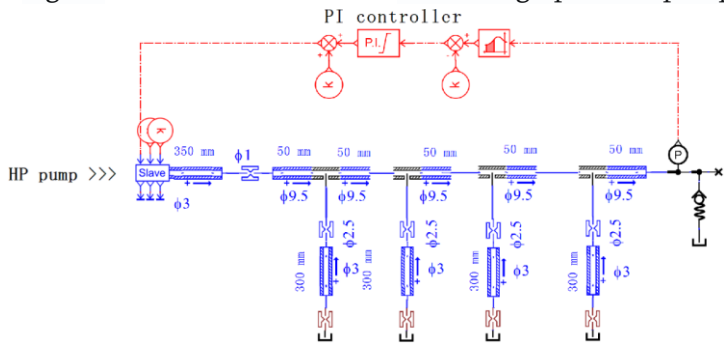


Fig 5 AMESim model scheme on the common rail

The structure sketch of common rail is shown in Figs. 5 is its AMESim model scheme.

The structure sketch of fuel injector is shown in Figs. 6 is its AMESim model scheme.

Fig7 is shown Amesim model of common rail injection system which consists of high pressure pump, common rail and four injectors.

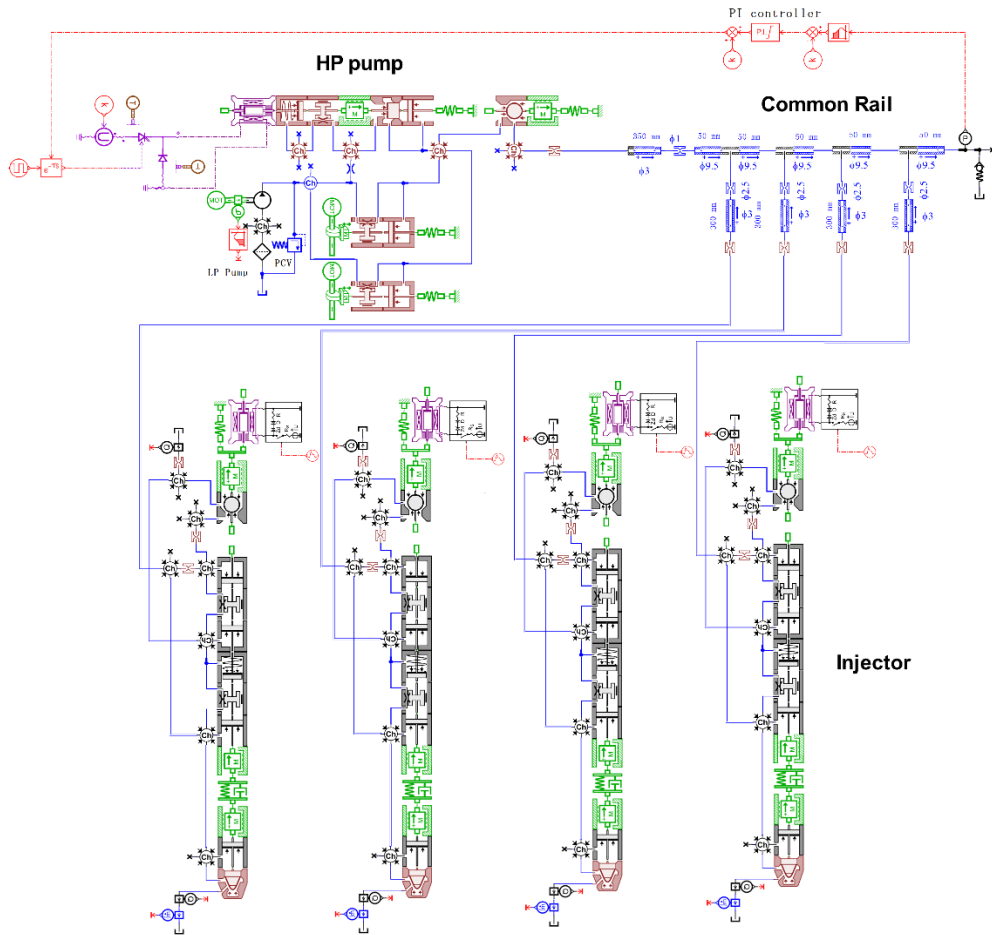


Fig 7. AMESim model scheme on the common rail injection system

3.1.2 Modelling of diesel engine using Amesim

Figure 8 is the scheme of the Amesim model on the diesel engine, in which the intake and exhaust system, and the compressor and turbo system are included.

3.2 Diesel engine injection control system modelling using Simulink

The control system processes the engine operative conditions and, according to the speed and load values, it sets the optimal injection time, quantity and quality of the injection, aimed at controlling combustion noise, fuel consumption, operation roughness and exhaust pollutant emissions. The control system model which is simulated with Matlab/simulink, shown in Fig. 9, consists of different sections: the fuel injection quantity, the objective rail pressure, fuel supply time, fuel injection time, fuel injection duration etc. In the control system model, the processing of engine speed signal and the rail pressure signal, and the time of sampling of the rail pressure and the fuel injection pulse duration are consistent with ECU in real engine. The signal of engine speed, opening degree of accelerator pedal, intake air pressure, intake air temperature, atmosphere pressure, atmosphere temperature, cooling water temperature and fuel temperature, is input into control system, and control signal of solenoid valve of fuel supply pump and solenoid valve of fuel injector is output from control system.

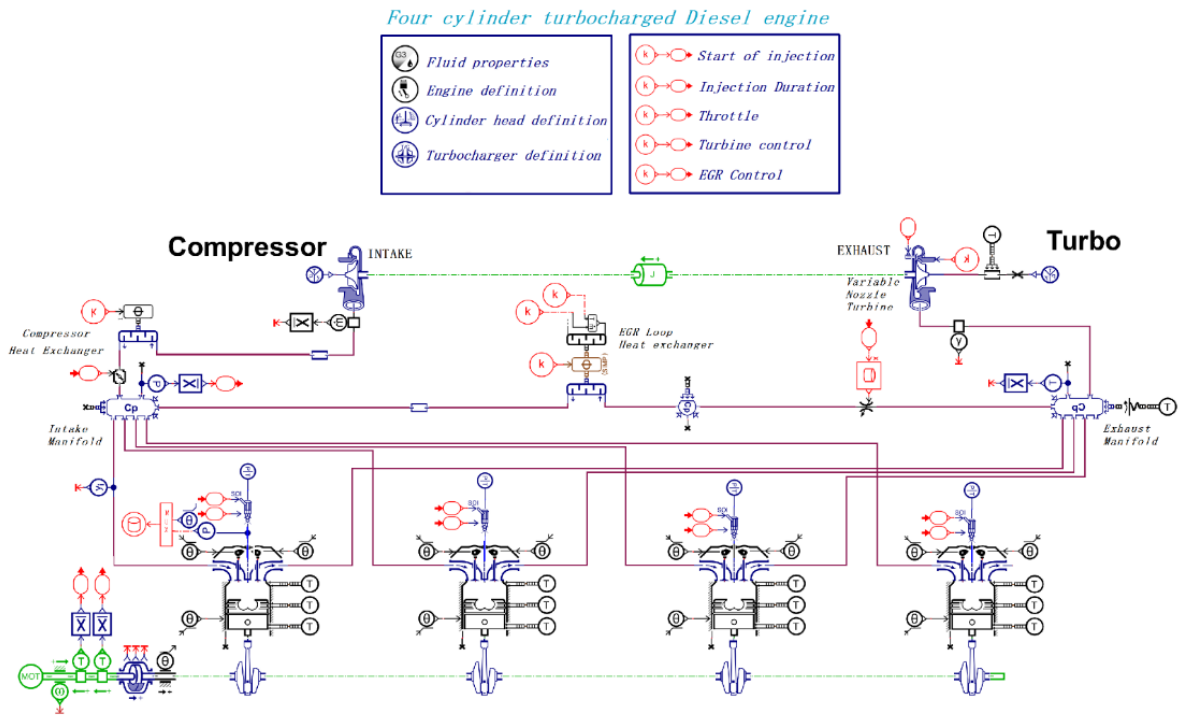


Fig 8. Modelling of diesel engine

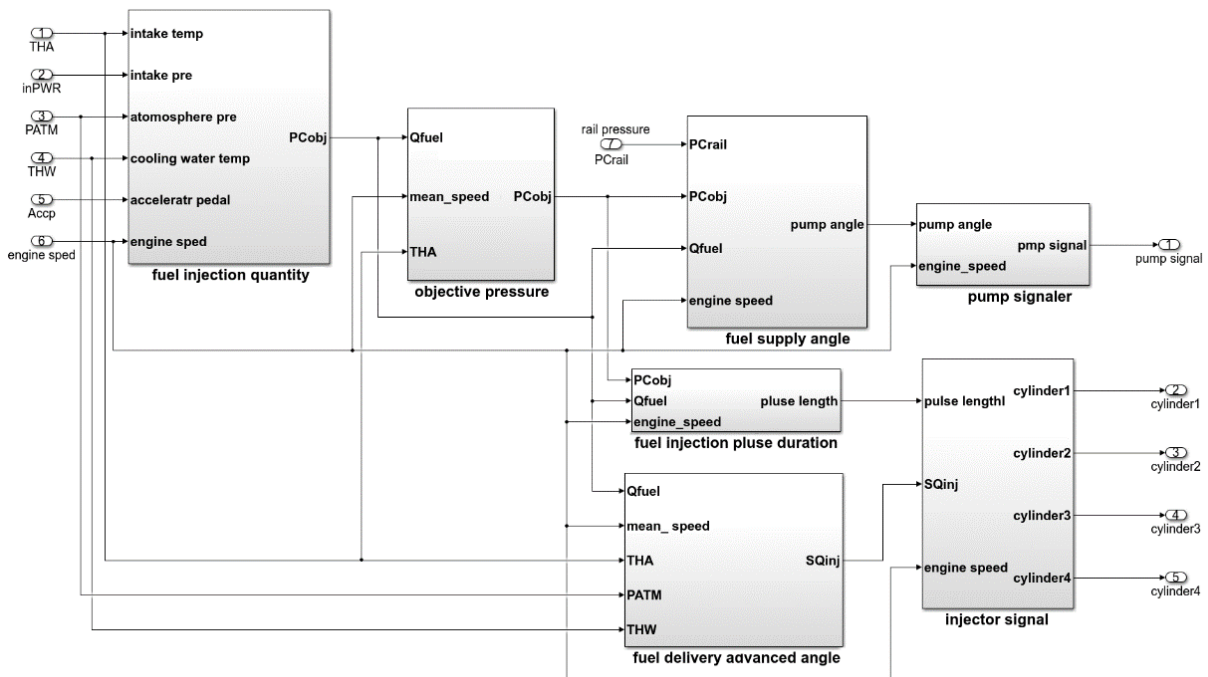


Fig 9. Control model

3.3 Integrated model

Fig 10 is co-simulation model of high-pressure fuel system and engine performance system and control system in common rail diesel engine, in which the sub-models of fuel supply pump, rail, fuel injector, engine, turbocharger and ECU are included.

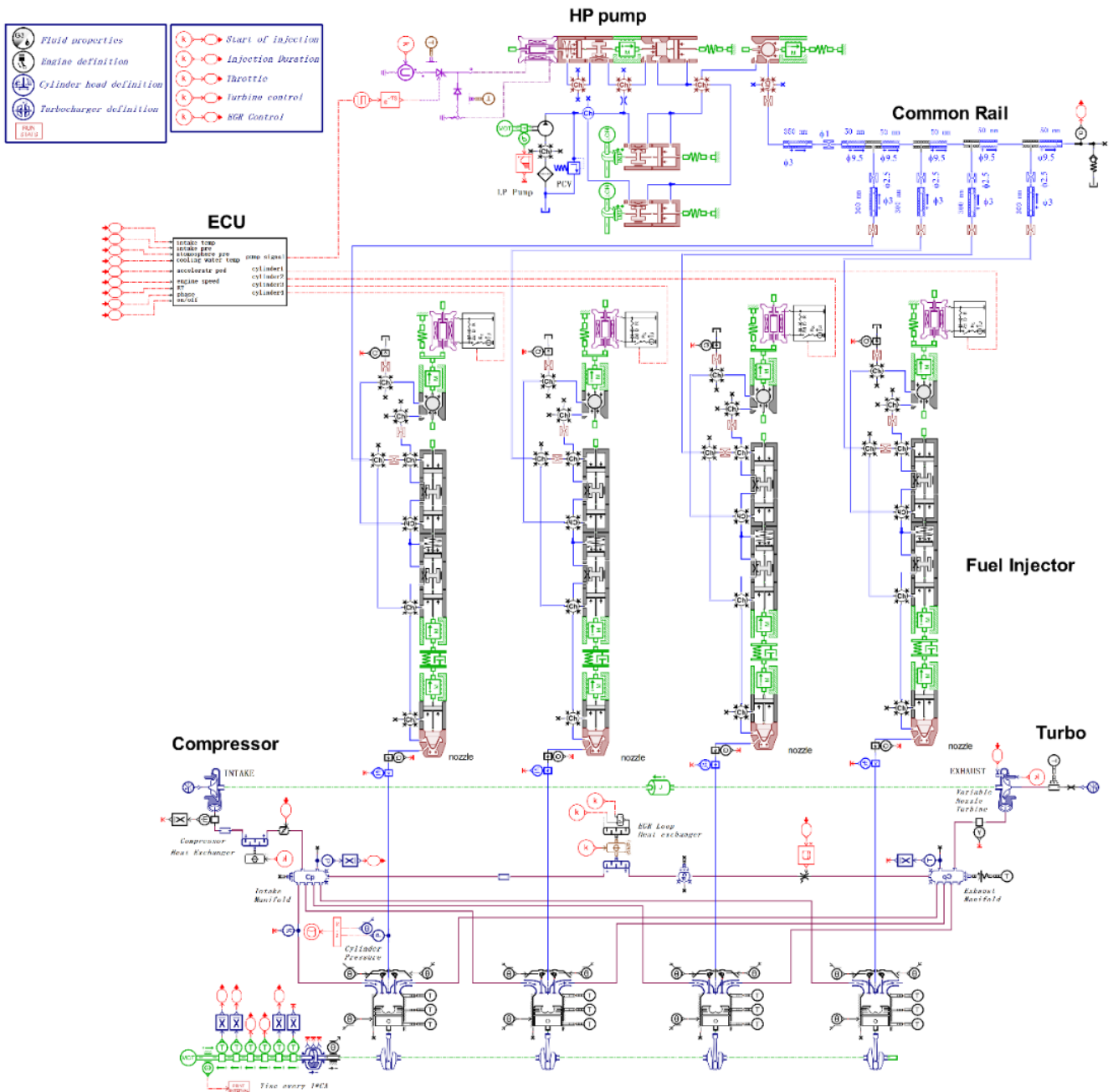


Fig 10. Integrated model

IV. Discussion

In order to study the influence of electronically controlled fuel injector parameters on the fluctuation of circulating fuel injection volume in high-pressure common rail system, a numerical simulation model of electronically controlled high-pressure common rail fuel injection system was established by using the AMESim and Simulink. Firstly, the mathematical modelling of the common rail fuel injection system was created, and here, the mathematical model was speci-

fically created with a solenoid valve control model, a volume chamber model, and a needle valve model. Next, Mathematical modelling of diesel engine was created for performance simulation of diesel engine including combustion model and turbocharger model. Finally, after modelling the injection characteristics of the common rail diesel engine using Amesim and modelling the diesel engine injection control system using Simulink, an integrated model was created.

V. REFERENCES

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