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ECU Oriented Simplified Diesel Engine Model

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Abstract

In this study, a four-cylinder simplified diesel engine model has been developed with a model-based design approach. In the modeling of the diesel engine, the signals coming from and going to the ECU are taken as basis. The injection time coming from the ECU and the duty cycle of the high pressure fuel pump are the main inputs. The outputs of the diesel engine model to the ECU are the rail pressure, crank and cam signals generated according to engine speed. Measurement-based simplifications were used in creating the model. The model was created in the Simulink environment. It has been observed that the outputs obtained by simulation of the model created are similar to the results obtained in real engines and other experimental studies.

Keywords: Diesel engine model, model based design, idle speed control, common rail model, rail pressure control

1. INTRODUCTION

In the development of engine control units in the automotive sector, model-based design has replaced classical code-based development due to the advantages it brings [1]. It is widely used for modeling of diesel engines, engine development, calibration and Engine Control Unit (ECU) development activities. While detailed models that best represent the real engine are needed for engine and calibration development, more simplified control-oriented models are sufficient for ECU development.

Interfaces between ECU and the engine consist of sensors and actuators. Injectors and high-pressure fuel pump control valve are the two most important actuators. The injection time together with the rail pressure determines the fuel quantity injected into the cylinder. High-pressure fuel pump controls the fuel flow to the rail and ensures the rail pressure to be at the desired level. On the sensors side, the most important sensors are speed sensors that give the crank angle and engine speed, and the rail pressure sensor. Other temperature and pressure sensors are used to make corrections for optimization and to detect fault conditions.

Although many parameters such as in-cylinder pressures, fuel pump fluctuations depending on crank angle are modeled in detailed engine models, these parameters do not have a direct input to the ECU [2]. In the studies on common rail modeling, very detailed injector and fuel pump models are based on [3]. Control oriented models have been simplified [4]. At the datasheet of fuel pumps It can be seen easily that the fuel pump flow rate is linearly proportional to the engine speed [5]. Again, many detailed studies have been carried out on injector simulation. A simple relationship is observed in the small number of models based on measurement where the fuel quantity discharged from the injector is proportional to the injection duration and the rail pressure [6].

Detailed engine models for ECU development are commercially available [7]. The Hardware-in-Loop (HIL) systems on which the engine model is running can be used as an ECU development environment [8]. Model-based commercial ECU development kits are also available on the market [9-10]. Commercial HIL systems are not widely used for educational purposes or by students due to their high cost.

In this study, an engine model with sufficient input output for idle speed control, rail pressure control and engine speed limitation, which are the basic functions of ECU, has been developed. ECU controls the duty cycle of the fuel pump with the proportional integral derivative (PID) output for the control of the rail pressure. Rail pressure sensor is sufficient as ECU input. Idle speed control and engine speed limiter controls are realized by controlling the torque demand with again PID. The translation of engine torque request from the ECU is performed using injection duration. The input that ECU needs for idle speed and engine speed limiter controllers is the engine speed. Crank-Cam signals and injection signals are also generated in accordance with the model.

2. MODEL DEVELOPMENT

ECU model and engine are modeled as separate blocks. Figure 1 shows overall view of the model. The engine model takes the injection time and fuel pump duty cycle inputs calculated by the ECU. The engine produces torque according to the amount of fuel quantity taken and rotates the crankshaft. The high fuel pump in the engine model is also controlled by duty cycle input. The only input from the user to the ECU is the accelerator pedal.

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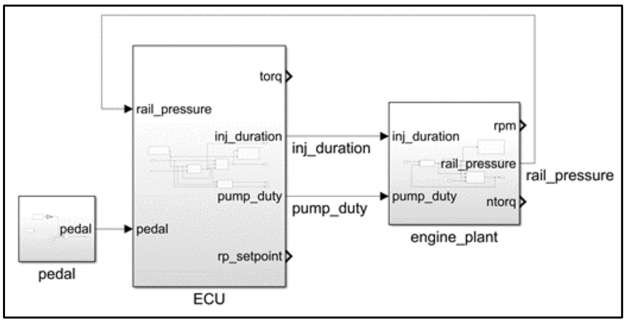

Figure 1. Overall view of the model

2.1. Engine Model

In this study, a 4-cylinder diesel engine is modeled. In the engine model, a torque proportional to the injection quantity is produced [11]. Engine speed is realized depending on engine inertia with the produced torque. Crank and cam signals are generated by a written s-function according to engine speed. In the common rail model, it is modeled to give the rail pressure output. Engine model is shown in Figure 2.

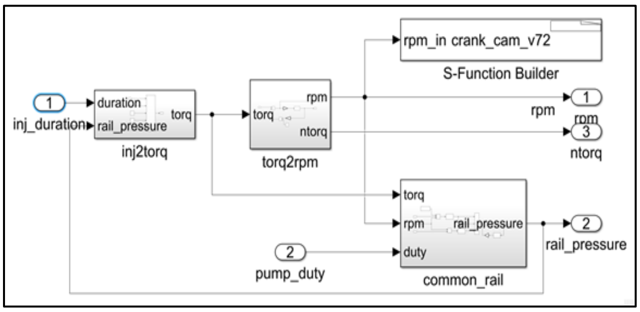

Figure 2. Engine model

In diesel engines, the brake specific fuel consumption (BSFC) value usually varies between 200-260 g/kWh depending on the engine speed. BSFC is defined as the amount of fuel consumed per power. BSFC can be formulated by writing the product of torque and engine speed instead of power and transforming radian to rpm. If the BSFC value is known, the fuel consumption can be calculated.

The brake specific fuel consumption (BSFC) value can be calculated by Eq. 1.

$$BSFC = \frac{Q_i}{\mathrm{T}_e \times N_e} \times \frac{2 \times \pi}{60 \times 60000} \qquad (\mathrm{g/kWh}) \qquad (1)$$

where, Q_i, is amount of fuel consumed per minute, T_e is engine torque and N_e is engine speed.

The fuel consumed per minute (Q_i) can be calculated by Eq. 2.

$$Q_i = \frac{2 \times \pi \times BSFC}{60 \times 60000} \times T_e \times N_e \qquad (\mathrm{g/min}) \qquad (2)$$

The relationship between fuel consumed per engine cycle (Q_c) and fuel consumed per minute (Q_i) depends on engine speed. Here the ratio between torque and fuel injected per cycle can be reached. Since the BSFC value changes within a narrow range, it can be assumed constant. In this case, it will be obtained that the engine torque (T_e) is proportional to the amount of fuel quantity consumed per cycle ($\mathrm{Q_c}$). The amount of fuel quantity consumed per cycle ($\mathrm{Q_c}$)is calculated by Eq.3.

$$Q_c = \frac{1000 \times Q_i}{2 \times N_e} = \frac{BSFC \times T_e \times \pi}{60 \times 60} \quad \left(\frac{mg}{cycle}\right) \qquad (3)$$

The engine torque (T_e) is calculated by Eq.4.

$$T_e = \frac{3600}{\pi \times BSFC} \times Q_c = \mathrm{K_q} \times \mathrm{Q_c} \qquad (Nm) \qquad (4)$$

The amount of fuel consumed per cycle $\mathrm{Q_c}$ is proportional to the rail pressure ($\mathrm{P_r}$) and injection duration ($\mathrm{t_i}$) [6, 11]. Number of cylinders (C_y) has to be multiplied, too. The K_i coefficient depends on the fuel injection capacity of the injector used and is provided by the manufacturer. The amount of fuel consumed per cycle $\mathrm{Q_c}$ can be calculated by Eq. 5.

$$\mathrm{Q_c} = \mathrm{C}_y \times \mathrm{K_i} \times \mathrm{P_r} \times \mathrm{t_i} \qquad (mg/cycle) \qquad (5)$$

In this way, engine torque (T_e) can be calculated based on rail pressure and injection time (Eq. 6).

$$\mathrm{T}_e = \mathrm{K_q} \times \mathrm{C}_y \times \mathrm{K_i} \times \mathrm{P_r} \times \mathrm{t_i} \qquad (Nm) \qquad (6)$$

Injection duration to torque conversion is shown in Fig. 3.

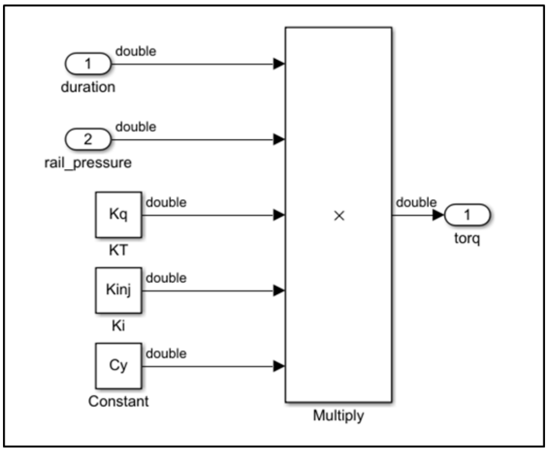

Figure 3. Injection duration to torque conversion

In this study, it is assumed that engine has a fixed geometry turbocharger and the turbocharger is not controlled. Since the engine cannot get enough air at low rpm, it will not be able to return all the calculated injection quantity to torque. For this reason, the injection duration before 1200 rpm was reduced by the ECU in proportion to rpm. In this way, the rpm torque curve obtained as a result of the simulation reflects a real engine better. Torque vs engine speed curve of the simulated engine is shown in Fig 4.

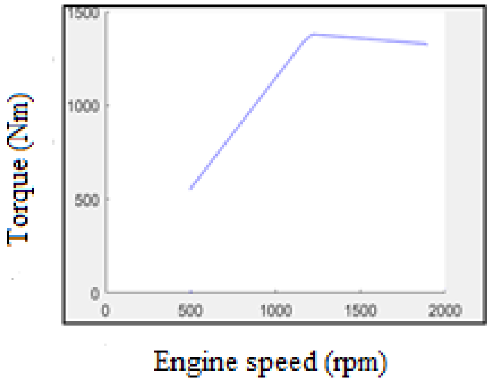

Figure 4. Torque vs engine speed curve

The engine speed N_e can be calculated by integrating the angular velocity obtained by dividing the torque generated by the engine inertia I_e. Radian rpm conversion should be done. The friction force F_e that increases in proportion to the speed of the engine is subtracted from the torque generated by combustion [12].

$$N_e = \frac{60}{2\pi} \times \int \frac{T_e - F_e}{I_e}\, dt \qquad (rpm) \qquad (7)$$

$$F_e = F_o + K_e \times N_e \qquad (Nm) \qquad (8)$$

Torque to engine speed conversion is shown in Fig. 5.

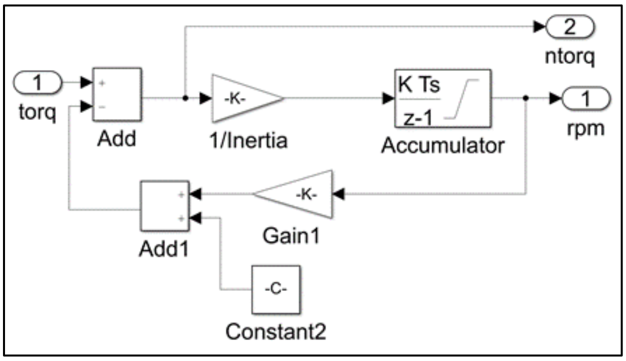

Figure 5. Torque to engine speed conversion

2.2. Common Rail Model

Rail pressure P_r is proportional to the integral of the fuel pumped from the high pressure pump Q_p minus the fuel discharged at the injector Q_i and the fuel leaking Q_l [4]. The amount of fuel provided by the pump must be higher than the maximum value of the fuel injected by the injectors. Fuel leaking back from the injector system can be taken as K_l percentage (typically 10%) of the fuel supplied from the pump. The amount of fuel coming out of the pump is controlled by changing the duty cycle D_u of the fuel metering valve. The bulk modulus K_f value of diesel fuel is expressed depending on the rail pressure [4]. Rail volume V_r is constant in cm^3. K_u is the coefficient required to convert fuel flows from g/min to m^3/s.

$$P_r = K_u \times \frac{K_f}{V_r} \times \int (Q_p - Q_i - Q_l) dt \qquad \text{(bar)} \qquad (9)$$

$$K_f = 12000 \times (1 + 12 \times P_r) \qquad \text{(bar)} \qquad (10)$$

$$Q_i = \frac{2 \times \pi \times BSFC}{60 \times 60000} \times T_e \times N_e \qquad \text{(g/min)} \qquad (11)$$

$$Q_p = D_u \times \frac{2 \times \pi \times BSFC}{60 \times 60000} \times T_{emax} \times N_e \qquad \text{(g/min)} \qquad (12)$$

$$Q_l = K_l \times Q_i \qquad \text{(g/min)} \qquad (13)$$

Common rail model is shown in Fig.6.

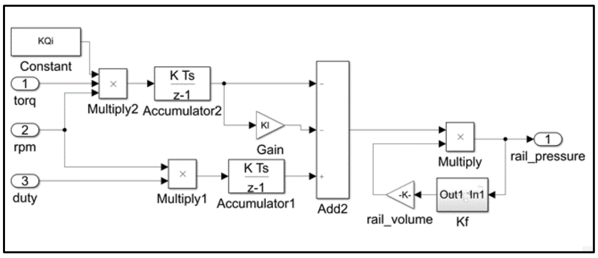

Figure 6. Common rail model

2.6 Simulation Results

In the developed model, a diesel engine with 1300 Nm torque is simulated. Engine inertia is assumed to be 2.5 kg/m^2. The BSFC value has been assumed constant as 240 g/ kWh.

Diesel engine control can be basically divided into three regions as idle speed control, torque control and engine speed limiter. As soon as the engine starts, the idle speed control is activated and keeps the engine at the specified idle speed if the accelerator pedal is not pressed. In this area, the PID idle speed controller sets the torque demand. By pressing the accelerator pedal, the idle speed control is released and the torque control zone is reached. In the torque control zone, the torque demand is entirely dependent on the accelerator pedal. Engine speed should not exceed the determined upper limit. This limitation is provided by the PID controller. Figure 7 shows engine control regions.

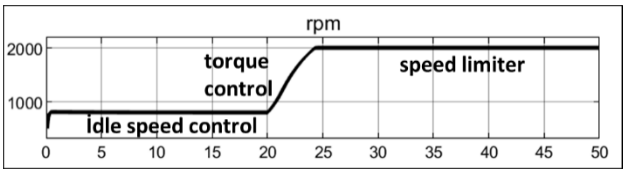

Figure 7. Engine control regions

In the developed model, the engine starts to operate at 500 rpm. It is determined as the idle speed zone between 0-1200 rpm. Idle speed set point is determined as 800 rpm. The engine maximum speed is selected as 2000 rpm. When the engine exceeds 1900 rpm, the PID controller is activated and keeps the engine at 2000 rpm.

Rail pressure control is very important for a stable injection. Rail pressure can be followed by PID controller to set points determined with a map. Two configurations tested in a similar study were simulated [13]. In the first configuration, when the engine is in the idle speed control zone, it has been shown that the set points are successfully followed at the end of the step test simulation, which turns from 900 bar to 1000 bar and again to 900 bar. The variation of the pump's duty cycle has the same pattern as in the comparison study. [13]. Figure 8 shows step change simulation at 800 rpm (rail pressure (straight) and set points (dotted) at idle speed).

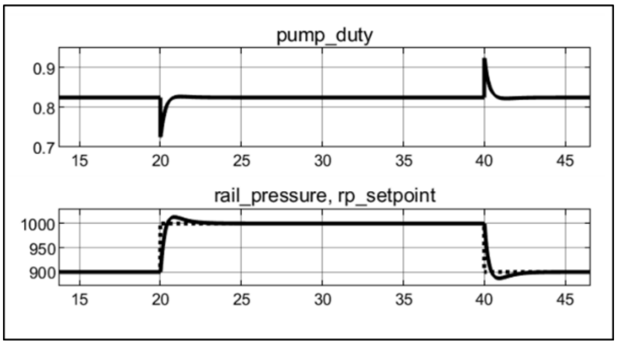

Figure 8. Step change simulation at 800 rpm (rail pressure (straight) and set points (dotted) at idle speed)

In the second configuration included in the study of Hong et al. is the rail pressure change due to the increased pressure demand. In this configuration the engine is at 800 rpm in idle speed mode. Pressure set points are changed to 900, 1000 and 1100 bar. It is seen that the rail pressure control algorithm can follow the demand by changing the duty cycle according to the demand. Fig. 9 shows step follower simulation at 800 rpm (rail pressure (straight) and set points (dotted)).

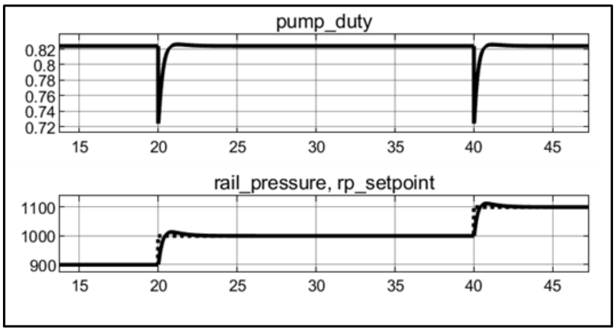

Figure 9. Step follower simulation at 800 rpm (rail pressure (straight) and set points (dotted)).

Crank and cam signals are generated according to the rpm value formed in the engine model. Authors had similar study for cam and crank signal generation [14]. Injection signals are also created according to the injection duration value calculated in the model (see Fig. 10).

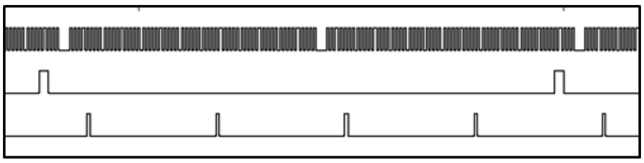

Figure 10. Crank (top), cam (middle) and injection (bottom) signals

3. CONCLUSIONS

In this study, a diesel engine model has been developed that can provide the necessary inputs and outputs for the basic functions of the ECU. The developed model is sufficient to develop and test the ECU's idle speed control, rail pressure control and engine speed limiter algorithms. There is essentially no need for a very detailed engine model in developing engine control software and testing algorithms. However, the model must be able to simulate real engine behavior. It has been observed that the outputs obtained by the simulation of the developed model are similar to the results obtained in real engines and other empirical model studies.

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