DRC-1021

The 4th TSME International Conference on Mechanical Engineering

16-18 October 2013, Pattaya, Chonburi

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2013

Common-Rail Pressure Control in a Diesel-Dual-Fuel Truck Using Sliding Mode Controller

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Abstract

Diesel-dual-fuel engine is an engine that uses mixed fuels between diesel and compressed natural gas (CNG). The engine experiences abnormal vibration during operating at high speed, low load due to inaccuracy of diesel injection pressure. At present, diesel injection system is common-rail system. When the desired pressure command and diesel injection command are changed frequently, there is wave in the common rail, which is disturbance of the system. Moreover, the common-rail diesel injection system is an uncertain system. These result in deviation between actual diesel injection pressure and desired pressure. Normally, control of the common-rail diesel injection pressure uses proportional-integral-derivative (PID) controller, which is a controller that cannot handle disturbance and system uncertainty well, resulting in the inability to control the pressure accurately. This paper proposed a method to control the diesel injection pressure using sliding mode controller. The proposed controller is robust and does not require accurate system model. From experimenting with a Toyota truck, running a diesel-dual-fuel engine, modified from a Toyota diesel engine number 2KD-FTV 2.5L with intercooler and variable nozzle turbo-charger, we found that the proposed controller delivers much more accurate diesel injection pressure, when compared to the PID controller. Especially, during transient operations, the injection pressure has shorter settling time.

Keywords: Sliding Mode Control, Diesel-Dual-Fuel Engine, Common-Rail Pressure.

Introduction

Diesel-dual-fuel engine is a modified diesel engine that uses mixed fuel between diesel and compressed natural gas (CNG). CNG injectors are equipped at each intake manifold, and CNG is injected in it. The engine uses cheaper fuel as CNG as the main fuel and has low modification cost. Thus, it is an interesting alternative engine.

Common-rail injection system is an up-to-date injection system developed to use in diesel engine. The main advantage of the system is its ability to vary injection pressure and timing in wide scale. As a result, diesel engines are powerful, economical and clean [1].

The common-rail system is more complicated. When the desired pressure command and diesel injection command are changed frequently, there is fluctuation in the common rail, which is a

disturbance to the system. Moreover, the common-rail diesel injection system is an uncertain system.

Injections in the diesel engine and DDF engine are different. The DDF engine uses mixture of two fuels, so diesel is usually injected in small amount. In addition, the injection strategy in the DDF engine is unlike that of the diesel engine. The desired pressure varies in wide range. Thus, the controller design must support this different injection effect.

Due to its complexity, common-rail pressure control is a big challenge. More precise injection pressure brings better engine performance, lowers fuel consumption and reduces emission.

There are some researchers improve the common-rail system model. [2], [3] discuss pressure fluctuations and formulate a detailed model. [4] proposes the model based on energy principle. [5] presents a vey detail model. Both [4] and [5] are compared with experiments, and show that, the model track the experiments quite well.

Some papers propose a common-rail pressure control technique. [6] presents a model from physical law, and propose sliding mode controller. The controller contains signum function, cause an extensive control chattering. [7] proposes common-rail pressure control using single neuron adaptive PID. [8] presents the system control using feed forward fuzzy PID controller.

[9] and [10] present control technique on diesel-dual-fuel engine. [9] proposes controller design base on quantitative feedback theory. The experiment conduced on modified Ricardo Hydra engine. [10] proposes gain-scheduling and integrator-augmented sliding mode control

technique which experiments on truck and engine testbed. This research implemented with high performance platform. Both of papers have good tracking performance.

This paper proposes a common-rail pressure control using sliding-mode controller which is robust and does not require accurate system model. This proposed technique includes a sliding-mode controller with an augmented integrator and mean pressure compensator. The experiment is conducted on truck. The engine is modified to operate with diesel-dual-fuel and implement with a commercial scale electronic control unit.

Common-rail Injection System

Fig. 1 depicts typical common-rail injection system. A low-pressure pump, which connects to the fuel tank, supplies fuel to a high-pressure pump. The high-pressure pump generates high-pressure fuel and delivers it to the common rail. A metering unit, controlled by the electronic control unit and flanged on the high-pressure pump, regulates fuel to the fuel rail. The common rail acts as an accumulator, stores fuel at high pressure, and distributes high-pressure fuel to the injectors. A pressure sensor, installed at one end of the common rail, senses the pressure inside the common rail and sends data to the electronic control unit

The experiment is conducted on Toyota Vigo Champ 2.5E, whose injection system is common-rail. The engine model is Toyota 2KD-FTV 2.5 Liters equipped with variable nozzle turbocharger and intercooler as shown in Fig. 2. The engine specification is shown in Table. 1. For the

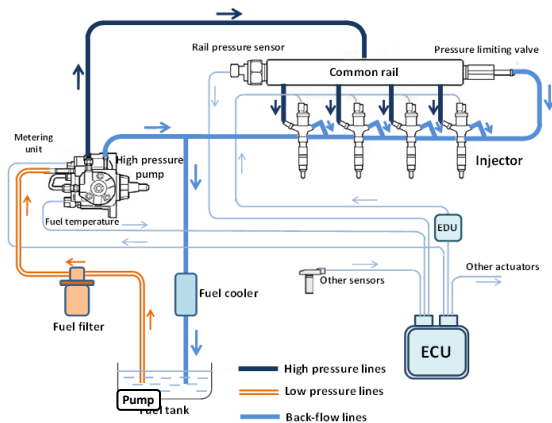experiments, the engine is modified to run with diesel-dual-fuel.

Fig. 1 The schematic of a common-rail injection system

Fig. 2 Toyota 2KD-FTV 2.5 Liters Engine

Table. 1 Experimental truck engine specification

Model:	Toyota 2KD-FTV, diesel Engine
Number of cylinders:	4 (Inline)
Number of valves:	16 (DOHC)
Manifold:	Cross-flow with VN turbocharger and intercooler
Fuel system:	Common-rail direct injection
Displacement:	2,494 cc
Bore:	92.0 mm
Stroke:	93.8 mm
Compression ratio:	17.4:1
Max power:	106 kW at 3,400 rpm
Max torque:	343 Nm at 1,600 - 2,800 rpm

Sliding Mode Controller

Sliding mode control is robust control under conditional uncertainty. The control signal consists of two phase, reaching phase and sliding phase. During the reaching phase, the control law used to bring the error trajectory to the sliding phase and maintain it there. And on the sliding phase, the error trajectory move toward to the origin.

The control scheme is shown in Fig. [3]. The controller consists of two control actions, one is of the sliding mode controller with an augmented integrator and the other is of the mean pressure compensator. A fuel injector has a hydraulic solenoid behavior. The actuator is insensitive to small input. The proposed mean pressure compensator is to compensate this nonlinearity effect.
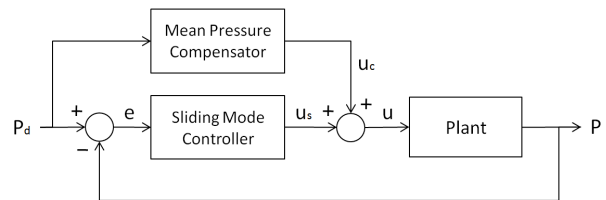

Fig. 3 The control scheme

The objective of this paper is to design control law that to drive error, $e = p - p_d$, to zeros. That is, actual common-rail pressure p will track desired pressure p_d closely.

Consider the second-order system

$$\dot{x}_1 = x_2$$
$$\dot{x}_2 = h(x) + g(x)u$$

where h and g are unknown nonlinear function and $g(x) \geq g_0 > 0$ for all x.

In the sliding phase, suppose we can design a control law that constrains the motion of the system to the surface

$$s = a_1 x_1 + x_2 = 0$$

Define $e_1 = p - p_d$ and $e_2 = \dot{p} - \dot{p}_d$. We can get the state that drive the trajectory of e_1 and e_2 to the surface as

$$s = a_1 e_1 + e_2 = 0 \qquad (1)$$

On this surface, the motion is governed by $\dot{e}_1 = -a_1 e_1$. Choosing $a_1 > 0$ ensures that $e_1(t)$ and $e_2(t)$ tend to zeros as time t tends to infinity, and rate of convergence can be controlled by choice of a_1. The motion on the surface $s = 0$ and maintain it there. Since

$$\dot{s} = a_1 \dot{e}_1 + \dot{e}_2$$

Suppose the desired pressure p and its derivatives satisfy the inequality

$$\left| \frac{a_1 e_2 + h(e_1, e_2)}{g(e_1, e_2)} \right| \le \rho(e_1, e_2), \quad \forall e_1, e_2 \in R \qquad (2)$$

for some known function $\rho(e_1, e_2)$, with $V_1 = (1/2)e_1^2$ as a Lyapunov function candidate, we have

$$\dot{V}_1 = s_1 \dot{s}_1 = s[a_1 e_2 + h(e_1, e_2)] + g(e_1, e_2)s u_s$$
$$\le g(e_1, e_2)|s|\rho(e_1, e_2) + g(e_1, e_2)s u_s$$

Taking

$$u_s = -\beta(e_1, e_2)\operatorname{sgn}(s) \qquad (3)$$

where $\beta(e_1, e_2) \ge \rho(e_1, e_2) + \beta_0, \beta_0 \ge 0$, and $\operatorname{sgn}(s)$ is signum function yield

$$\dot{V} \le |s|\rho(e_1, e_2) - s[\rho(e_1, e_2) + \beta_0]\operatorname{sgn}(s) = \beta_0|s|.$$

Thus, $W = \sqrt{V} = |s|/\sqrt{2}$ satisfies the differential inequality $D^+ W \le -\beta_0/\sqrt{2}$ and from the comparison lemma [11] we have

$$W(s(t)) \le W(s(0)) - \beta_0 t/\sqrt{2}.$$

Therefore, the trajectories of e_1 and e_2 reach the surface $s = 0$ in finite time and, once on the manifold, they cannot leave it, as seen from the inequality $\dot{V} \le -\beta_0|s|$.

The control law (3) brings the trajectories of e_1 and e_2 to the surface $s = 0$ and maintains it there.

The region of attraction can be estimated. Suppose (2) becomes

$$\left| \frac{a_1 e_2 + h(e_1, e_2)}{g(e_1, e_2)} \right| \le k_1, \quad \forall e_1, e_2 \in \Omega,$$

where Ω is the region of attraction, for some known nonnegative constant k_1. We can take $u_s = -k \operatorname{sgn}(s)$, $k > k_1$. The condition $s\dot{s} \le 0$ in the set $\{|s| \le c\}$ makes it positively invariant. From $\dot{e}_1 = e_2 = -a_1 e_1 + s$ and the function $V_1 = (1/2)e_1^2$, we have

$$\dot{V}_1 = e_1 \dot{e}_1 = -a_1 e_1^2 + e_1 s \le -a_1 e_1^2 + |e_1|c \le 0,$$
$$\forall |e_1| \ge c/a_1$$

Thus, $|e_1(0)| \le c/a_1 \Rightarrow |e_1(t)| \le c/a_1$, $\forall t \ge 0$ and the set $\Omega = \{|x_1| \le c/a, |s| \le c\}$ is positively invariant.

Since the control law (3) contains the discontinuous signum function, control chattering will occur. To avoid the chattering problem, we replace the discontinuous function with a continuous arctan function

$$u_s = -k(2/\pi)\arctan(s/\varepsilon) \qquad (4)$$

To obtain zero steady-state errors and integrator can be augmented to the system. Let $e_0 = \int e_1 dt$. Then, we get the sliding surface as

$$s = a_0 e_0 + a_1 e_1 + e_2 = 0 \qquad (5)$$

In summary, our control law is Eq. (4), where s is given by Eq. (5). There are four design parameters: k, ε, a_0 and a_1

Experimental Results

This paper experiments on Toyota Vigo, whose engine specification is shown in the past section. The engine is modified to operate with the diesel-dual-fuel mode by installing CNG injectors on each of four intake ports. The distance from the CNG injector to the port is kept

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as small as possible. CNG injectors, fuel injectors, high pressure pump and other actuators are controlled by an electronic control unit. The electronic control unit is Mototron ECM-0563-048-0704-C/-F, which is the commercial scale control unit, instead of OEM electronic control unit. The microprocessor unit is Freescale MPC563, 56MHz, and the memories are 512KB flash memory, 32KB RAM and 128KB parallel EEPROM. Injection strategy and control program are implemented with MATLAB/Simulink software.

The proposed controller is compared with fine-tuned PID controller with mean pressure compensator.

For the experiments, the pick-up truck runs with diesel-dual-fuel operation. The truck uses three gear levels in 200-meter distance. So, the result plot contains three-pulse shapes. The gear level is changed each time when the engine speed is about 2500 rpm. The experimental results are described in Fig. 4, which consists of four sub-figures as Fig. 4(a) to Fig. 4(d)

Fig. 4(a) shows the plot of common-rail pressure using PID controller with mean pressure compensator. The black line represents the actual common-rail pressure, and the red line represents the desired pressure. Both are plotted versus time. The controller is operated with fine-tuning gain, k_p = 0.4, k_i = 0.1 and k_d = 0. The plot shows that the actual pressure does not track the desired pressure well. It has high overshoot and tracking error, especially during transient states. Fig. 4(b) shows the plot of common-rail pressure using sliding mode controller. The black line and the red line represent the same quantities as those in Fig. 4(a). The parameter is tuned well, that is, a_0 = 12, a_1 = 22, ε = 160 and k = 7. The plot shows that

the actual pressure is able to follow the desire pressure well. The results show little overshoot and just small tracking error, especially in transient state.

From Fig. 4(a) and Fig. 4(b), the root-mean-square tracking errors are 5.9739 and 3.9290 respectively. The quantities show that, the sliding mode controller has less error than PID and ensure of the better tracking performance.
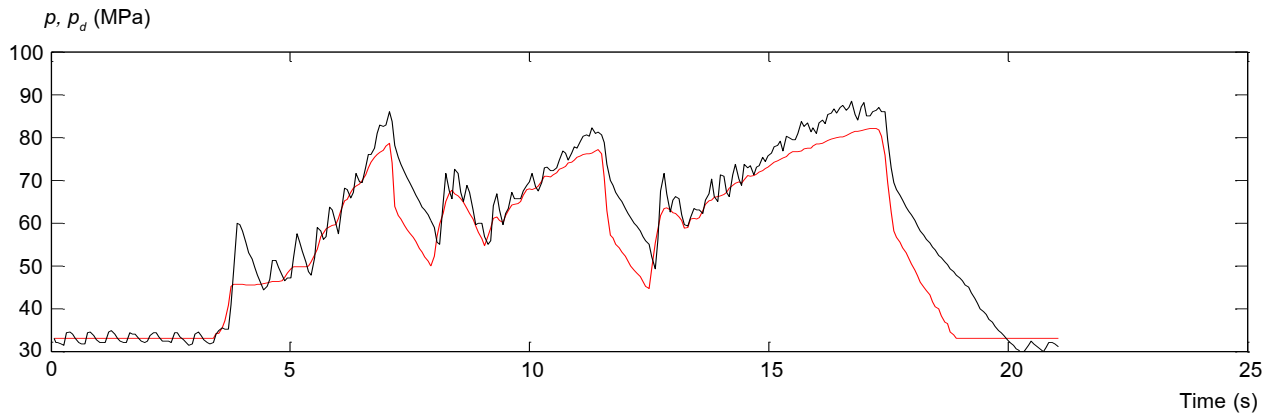
The control signals are plotted in Fig. 4(c) and Fig. 4(d), which are of PID controller and of sliding mode controller, respectively. Fig. 4(d) can be seen as more control chattering than Fig. 4(c), however, this is the result of having more active control.
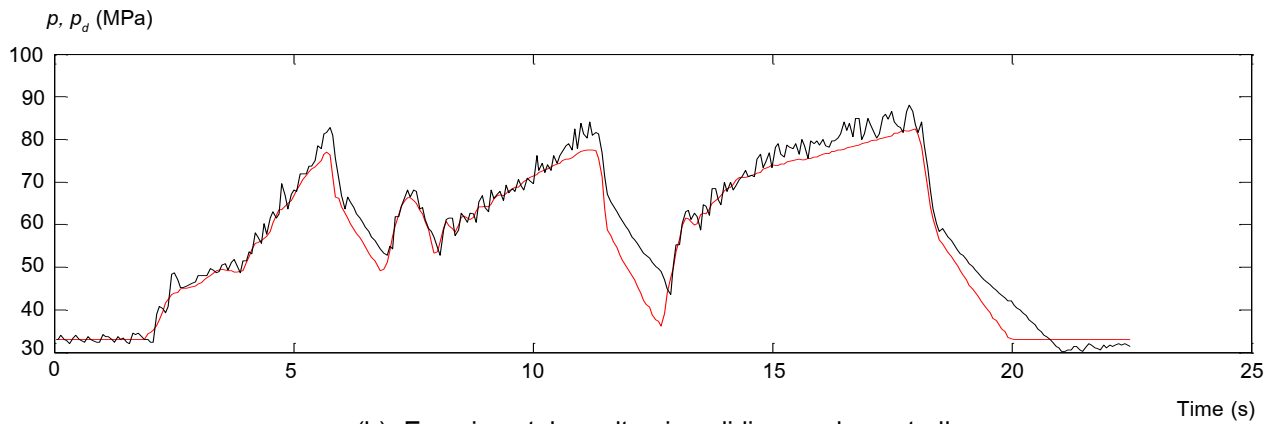
Conclusions

This paper proposes common-rail pressure control method in diesel-dual-fuel engine using sliding mode controller. This proposed technique consists of sliding mode controller with an augmented integrator and mean pressure compensator. The controller is experimented on a truck and is implemented with a commercial scale electronic control unit. The experimental result is compared with a traditional PID controller. The results show that the proposed controller has improved control performance, especially it has better tracking performance in transient state.
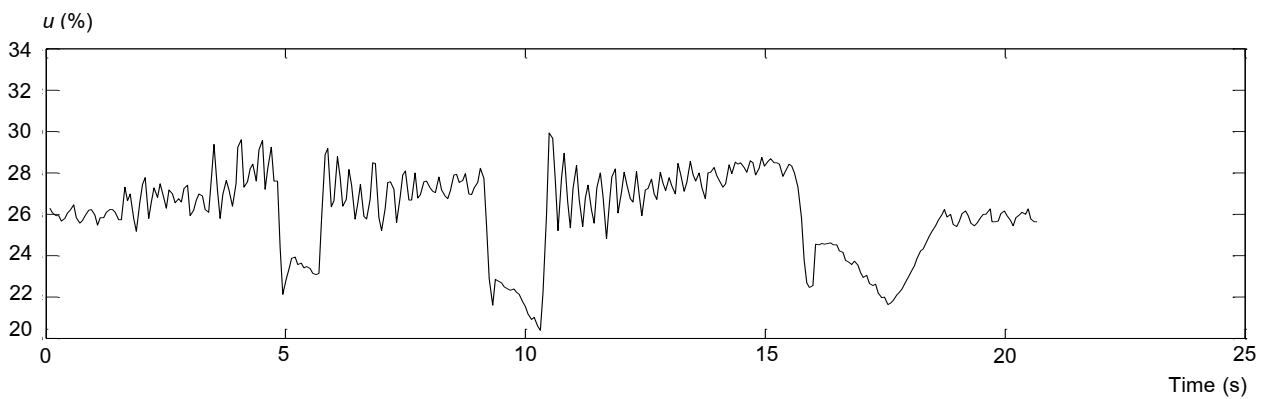
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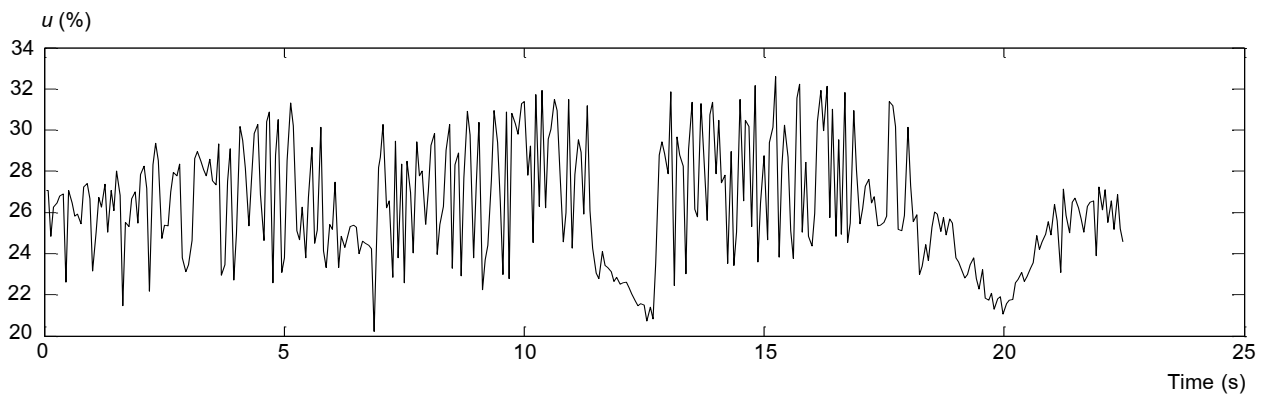
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p, p$_d$ (MPa)

(a) Experimental result using PID controller

p, p$_d$ (MPa)

(b) Experimental result using sliding mode controller

u (%)

(c) The control signal: *u* of PID controller

u (%)

(d) The control signal: *u* of sliding mode controller

Fig. 4 Experimental results

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