

Identification and Robust Control of LPG Fuel Supply System

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ABSTRACT

This paper proposes a new returnless LPG fuel supply system designed to increase the efficiency of current LPG engines. With a conventional engine fuel supply system, the fuel pump is driven at a certain speed to pressurize the fuel to an excessive level, and excess fuel that is discharged from the fuel pump but not injected from the injector is returned to the fuel tank via a pressure regulator and a return line. This arrangement keeps the pressure in the fuel supply line at a constant level. Accordingly, during engine idling, fuel cut-off or other times when very little or no fuel is injected from the injector, nearly all the fuel discharged from the fuel pump is returned to the fuel tank via the pressure regulator and return line. Therefore, the energy (electric power) applied to drive the fuel pump is wastefully consumed. Moreover, returning a large amount of excess fuel to the fuel tank can raise the fuel temperature in the tank, causing the fuel to evaporate. To avoid that problem, it is necessary to raise the pressure in the tank to an excessive level, especially when using fuel that evaporates easily. In addition, if the fuel pressure could be controlled at a lower pump speed, it would cause less component wear and have a good effect on equipment durability compared with the situation where the fuel pump is operated at high speed all the time. As a solution to this issue, this paper proposes a returnless fuel supply control system that does not use a pressure regulator or a return line. This system maintains a constant pressure in the fuel supply line by controlling the fuel pump so that it supplies only the quantity of fuel

needed to meet the injection quantity that the engine demands from the injector. The modeling of the fuel supply system is described first along with the system identification based on the use of a genetic algorithm (GA). The system was configured by designing a robust sliding mode controller in relation to the identified nonlinear system. Simulations and experiments conducted with the control system showed that it provides excellent control performance as planned, with good agreement seen between the simulation and experimental data. That agreement confirmed the validity of the modeling method. The results also verified that this fuel supply control system saves needless energy consumption, especially under conditions of low engine output such as when idling. As a result, it works to improve the efficiency of the overall engine system.

INTRODUCTION

In recent years, automotive engines have been required to provide high levels of power and torque, while at the same time incorporating measures to deal with environmental pollution. Engines fueled by compressed natural gas (CNG) or liquefied petroleum gas (LPG) have attracted interest as low-emission engines capable of satisfying these requirements.

Like other industrial fields, the automotive sector is also being called upon to take further measures to reduce pollution accompanying the worsening of the global environment. Among the low-emission vehicles now being researched and developed, alternative-fuel

vehicles powered by gaseous fuels such as compressed natural gas (CNG) and liquefied petroleum gas (LPG) are thought to be the most promising candidates for implementation in the relatively near future because they only require minor modification of existing mass-produced engines. Accordingly, demands have been rising for more highly accurate fuel supply control for LPG vehicles as well as for lower fuel consumption. One technology thought to be capable of meeting these demands is a liquid LPG injection system of the multi-point injection (MPI) type. The fuel supply control system has an especially close relationship with engine controllability and reliability. For that reason, assurance of high system reliability is required along with meeting demands for higher speed, higher performance and greater energy savings.

With a conventional engine fuel supply control system, the fuel pump is driven by a drive motor to pressurize the fuel and a pressure regulator is used to maintain a constant fuel pressure. Excess fuel that is discharged from the fuel pump but is not injected from the injector is generally returned to the fuel tank via the pressure regulator and a return line. In this case, the discharge flow rate from the fuel pump is set higher than the maximum fuel flow rate emitted from the injector in order to meet all sorts of engine operating conditions. Accordingly, the fuel pump constantly operates at a high speed. During engine idling, fuel cut-off or other times when little or no fuel is injected from the injector, nearly all the fuel discharged from the fuel pump is returned to the fuel tank via the return line and pressure regulator. For that reason, the energy (electric power) applied to drive the fuel pump is wastefully consumed. Moreover, returning a large quantity of excess fuel to the tank can raise the fuel temperature in the tank, causing the fuel to evaporate. An excessive rise in the tank pressure is apt to become a problem especially when using fuels that evaporate easily. In addition, operating the fuel pump at high speed all the time can also give rise to another problem in that the durability of the pump may decline due to component friction, among other factors.

Taking these problems into account, this paper proposes a returnless fuel supply control system that eliminates the return of excess fuel to the tank via a return line from the engine fuel supply system. This control system is configured so that it supplies only the amount of fuel needed to meet the injection flow rate that the engine demands from the injector.

This paper first describes the modeling of the fuel supply system and the use of a genetic algorithm (GA) for system identification. It then describes the application of robust sliding mode control to the identified nonlinear system. The results of simulations and experiments are then presented to show that the proposed method achieves a control system capable of supplying the

amount of fuel matching the injection flow rate demanded from the injector by the engine.

CONFIGURATION AND MODELING OF LPG FUEL SUPPLY SYSTEM

Figure 1 shows the configuration of the LPG fuel supply system that is the target of control in this study.

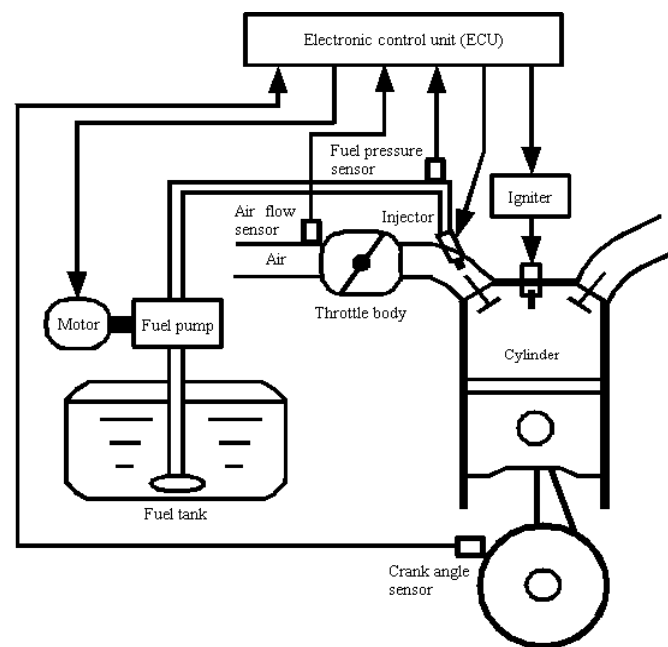


Fig.1 Engine control system

This system mainly consists of a drive motor, a fuel pump that is driven by the motor, a fuel tank, a fuel line and a fuel injector. A fuel injection pressure sensor is also provided that produces the signal for feedback control. That signal is sent to the electronic control unit (ECU), which controls the drive motor according to the control algorithm. The following discussion describes the modeling of these system components.

MODELING OF FUEL PUMP SYSTEM

The fuel pump used in this fuel supply system is a three-cylinder plunger pump with a three-lobed cam. The total load torque and the total discharge flow rate of the pump are equal to the respective sum of the load torque and the discharge flow rate of the individual cylinders. In this case, the load torque and discharge flow rate per cylinder can be found at the time of lift acceleration and lift deceleration with the following equations using the plunger load, injection pressure, plunger stroke, plunger area and cam rotational angle[1]-[3].

At the time of lift acceleration when $0 \leq \theta \leq \delta$, the plunger stroke and load torque are given as follows:

$$h_a = r_k \frac{(1 - \cos(\theta)) \sin(\gamma - \delta)}{\sin(\gamma) - \sin(\gamma - \delta)} \quad (1)$$

$$T_a = (PA + f_s + k_s h_a)(a_k + r_k) \tan(\alpha) \quad (2)$$

$$Q_{ia} = Ah_a \dot{\theta} \quad (3)$$

where

h_a : plunger stroke at the time of lift acceleration

r_k : cam base radius

θ : cam rotational angle

$\dot{\theta}$: cam (pump shaft) rotational speed

γ : cam working angle

δ : positive velocity region angle

T_a : load torque at the time of lift acceleration

P : pump discharge pressure (fuel injection pressure)

A : plunger area

f_s : spring set load

k_s : spring constant

a_k : nose radius

α : power point centripetal angle

Q_{ia} : pump discharge flow rate at the time of lift acceleration

At the time of lift deceleration when $\delta \leq \theta \leq \gamma$, the plunger stroke and load torque are given as follows:

$$h_r = r_k \frac{\cos(\gamma - \theta) \sin(\delta)}{\sin(\gamma) - \sin(\gamma - \delta)} - r_k \quad (4)$$

$$T_r = (PA + f_s + k_s h_r)(a_k + \sqrt{(s_k + r_k)^2 - \Psi^2}) \tan(\beta) \quad (5)$$

$$\Psi = (R_k - r_k) \sin(\theta) - R_k \cos(\delta + \frac{\pi}{2} - \theta) \quad (6)$$

$$Q_{ir} = Ah_r \dot{\theta} \quad (7)$$

where

h_r : plunger stroke at the time of lift deceleration

T_r : load torque at the time of lift deceleration

s_k : maximum plunger stroke

β : power point centripetal angle

R_k : radius of curvature

Q_{ir} : pump discharge flow rate at the time of lift deceleration

In the case of this three-cylinder plunger pump with a three-lobed cam, the total load torque T_L is the sum of the load torque expressed by the position of the cam of each cylinder. The load torque per cylinder and the sum of the load torque of all the cylinders are shown in Fig. 2. The total pump discharge flow rate Q_i is the sum of the pump discharge flow rate expressed by the position of the cam of each cylinder. Figure 3 shows the pump discharge flow rate per cylinder and the sum of the pump discharge flow rate for all the cylinders. In calculating the total load torque T_L and the total pump discharge flow rate Q_i , the pressure was set at $P = 3[\text{MPa}]$ and the pump rotational speed at $\dot{\theta} = 2600[\text{rpm}]$.

However, in order to calculate the load torque with the above-mentioned equation, it is necessary to know the pump discharge pressure and the fuel injection flow rate, which are given by the following equations[4],[5].

$$\frac{dP}{dt} = (\rho Q_i - \rho Q_j - V_p \frac{d\rho}{dt}) \frac{1}{V_p K_v} \quad (8)$$

$$Q_j = C_n A_n \sqrt{2g \frac{(P - P_a)}{\rho}} \quad (9)$$

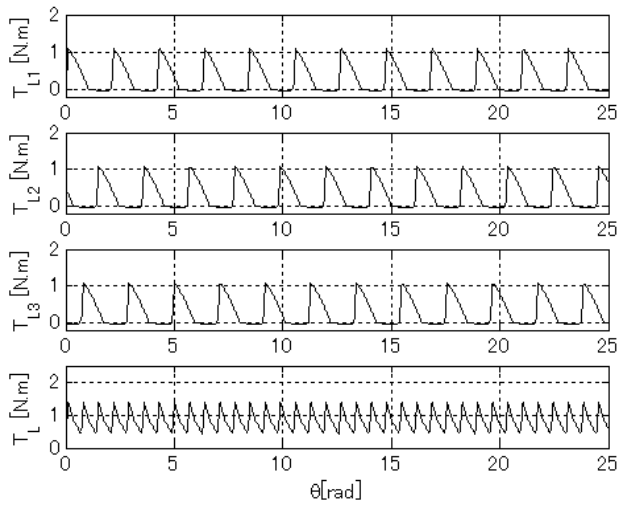


Fig.2 Load torque of fuel pump

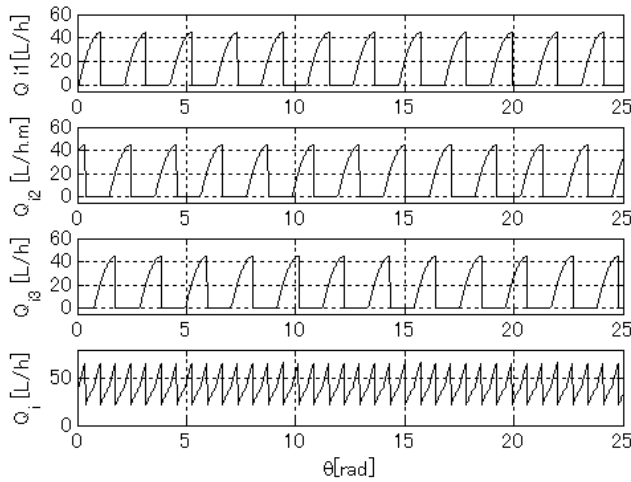


Fig.3 Pump discharge flow rate

$$\rho = k_a P + k_b \quad (10)$$

where

ρ : fuel density in the fuel line

Q_j : fuel injection flow rate

V_p : fuel line volume from pump exit to fuel injector

K_v : volumetric elasticity coefficient

C_n : discharge flow rate coefficient

A_n : injector hole area

g : gravitational velocity

P_a : atmospheric pressure

k_a : pressure coefficient

k_b : pressure coefficient

MATHEMATICAL MODEL OF DRIVE MOTOR

The following mathematical model can be considered for the motor control system of a pump drive motor including the gears [6],[7]. As is well known, the electrical characteristics of the DC motor that actually drives the fuel pump can be given by the following equation.

$$L \frac{di_a}{dt} + R_a i_a + K_e N \frac{d\theta}{dt} = U_a \quad (11)$$

where i_a is the armature current, U_a is the input voltage applied to both armature terminals, L and R_a are the armature inductance and resistance, K_e is the induction voltage constant, N is the gear ratio and θ is the cam rotational angle. Letting $T = NK_t i_a$ represent the electromagnetic torque applied to the armature, the equation of motion of the fuel pump system can be expressed as:

$$J \frac{d^2\theta}{dt^2} + D \frac{d\theta}{dt} + d_k \operatorname{sgn}\left(\frac{d\theta}{dt}\right) = NK_t i_a - T_L \quad (12)$$

where J is the total moment of inertia equivalent to that of the camshaft of the system, D is the viscous friction coefficient, d_k is Coulomb friction, K_t is the motor torque constant and T_L is the load torque. Assuming that the armature inductance component L can be ignored, Eq. (11) is substituted into Eq. (12) to obtain the following equation.

$$\ddot{\theta} = -\frac{1}{J} \left\{ \left(D + \frac{N^2 K_t K_e}{R_a} \right) \dot{\theta} - d_k \operatorname{sgn}(\dot{\theta}) - T_L \right\} + \frac{NK_t}{R_a J} U_a \quad (13)$$

From Eqs. (13), (8) and (10), the following equation of state and output equation of the system can be given by applying the state variables $x_1 = \theta$, $x_2 = \dot{\theta}$ and $x_3 = P$.

$$\begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \\ \dot{x}_3 \end{bmatrix} = \begin{bmatrix} x_2 \\ a_1 x_2 + a_2 \operatorname{sgn}(x_2) + a_3 T_L \\ \rho Q_i - \rho Q_j - V_p \dot{\rho} \frac{1}{V_p K_v} \end{bmatrix} + \begin{bmatrix} b_1 \\ b_2 \\ b_3 \end{bmatrix} U_a \quad (14)$$

$$y = x_3 \quad (15)$$