

Study on a Versatile Liquid Dosing Device for IC Engine After-Treatment System

Yanxiang Yang, Bingqian Tan, and Changwen Liu
Tianjin Univ.

Ping Zhang
Zhejiang Univ.

Daguang Xi
Zhejiang FAI Electronics Co. Ltd.

ABSTRACT

A versatile liquid dosing device along with its metering theory, which can be applied to both SCR dosing system and DPF regeneration system of IC engine after-treatment system, is presented in this paper. The device is composed of a solenoid driven plunger pump, a nozzle and a pressure tube, and is pump-end controlled by PWM signals. Both electrically resistive and conductive liquids including DEF for SCR system, fuel for DPF regeneration, and gasoline for spark ignition engine, can be dispensed quantitatively with this device.

A metering theory determining the liquid discharged per injection is developed by studying the system using a physical-mathematical model. The study shows that the liquid discharge can be well correlated with a measurable variable T_3 , which is associated with the net output energy. Experimental investigations verified that the metering results were independent of the state changes. The developed device was able to work at a frequency from 0 to 80Hz with largest dosing range of 0 to 14 L/h at 24 volt.

The liquid dosing device was applied and tested on a SCR system as a verification example. The pump was so compact that it could be located at the bottom inside the DEF tank, which greatly simplified the anti-freeze design consideration and the SCR system. Tested results on a diesel engine with the SCR system showed good capability to reduce the engine NOx emission with high maximal NOx conversion rate and good controllability to achieve a NOx emission target with high reliability and high DEF efficiency.

CITATION: Yang, Y., Tan, B., Liu, C., Zhang, P. et al., "Study on a Versatile Liquid Dosing Device for IC Engine After-Treatment System," *SAE Int. J. Engines* 8(3):2015, doi:10.4271/2015-01-1035.

INTRODUCTION

Modern diesel and gasoline engines are using after-treatment systems to fulfill more stringent regulations of pollution matter emissions around the World. Selective Catalytic Reduction (SCR) system for catalytic conversion of NOx in engine exhaust pipe is getting popular to meet regulations such as China Stage IV/V, Euro IV/V and EPA standards [1, 2, 3]. Diesel Particulate Filter (DPF) technology is also considered as a necessary means to meet China Stage V/VI, Euro V/VI and EPA standards [4, 5, 7].

In the above mentioned engine after treatment system, liquid dosing and injection are fundamentals for supplying a liquid reagent or fuel to the exhaust gases of the engine. In SCR system, usually an aqueous urea solution (Diesel Exhaust Fluid=DEF) must be dosed and atomized when supplied to the engine exhaust pipe [1,2]. In a DPF system, an active regeneration system is mostly needed to burn out the particulate matters (soot) loaded on the DPF filter frequently to avoid DPF clogging which will negatively influence engine normal

operation. The best active regeneration system is to supply fuel in the exhaust gas to increase the exhaust temperature and to initiate the burning of collected soot on the DPF by an exothermic reaction of the supplied hydrocarbon and oxygen over a catalyst [4, 5, 6].

Dosing and atomization of liquid usually are carried out by a pump and nozzle system [7, 8]. But because DEF is electrically conductive, general purpose fuel pumps can not be used as a DEF pump. Existing SCR systems are using mostly new developed diaphragm type pumps which dose the DEF supply by nozzle-open time control or by positive pump revolution counting. But none of the dosing systems is simple enough to install on the DEF tank or to avoid the problematic freezing and clogging issues inside the supply tube at cold temperatures or soak conditions. These characteristics make the SCR system more expensive and less reliable. A new approach is required [9, 10].

In this paper, a versatile liquid dosing device along with its metering theory, which can be applied to both SCR dosing system and DPF regeneration system of IC engine after-treatment system, is presented.

The versatile liquid dosing device is composed of a solenoid driven plunger pump, an auto-open nozzle and a pressure tube long or short connecting them as indicated in Figure 1. Both electrically resistive and conductive liquid can be dispensed quantitatively with this device. The liquid can be fuel or other chemical solutions such as DEF of SCR system, hydrocarbon for DPF regeneration, etc. The solenoid pump is propelled by PWM (pulse width modulation) power from an ECU (electronic control unit) and so the liquid dosing device is a pump-end controlled pump-nozzle liquid supply unit.

In comparison with the conventional injection system with a liquid rail and open-time controlled nozzle, the pump-end controlled liquid dosing device as in Figure 1 is compact in that there is no need of mechanical driving or pressure regulator, and it is able to offer higher injection pressure, higher reliability, and higher energy efficiency. However, unlike the conventional system, the output of pump-end controlled liquid dosing device is no longer proportional to the input pulse width. The determination of liquid discharge rate for such a liquid dosing device is far more sophisticated than that for a conventional injection system with a liquid rail and open-time controlled nozzle [7, 9].

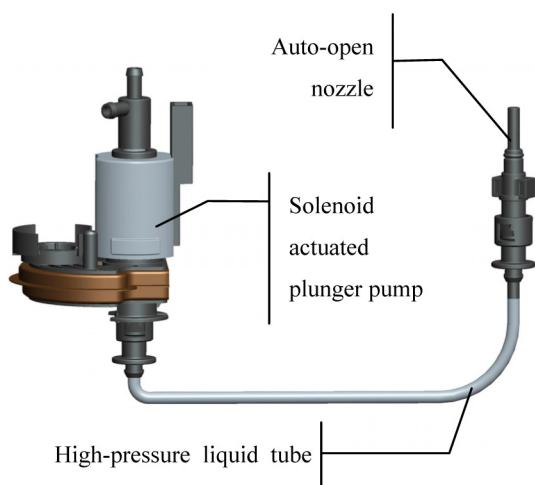


Figure 1. A pump-end controlled pump-nozzle liquid dosing device

In comparison with the conventional injection system with a liquid rail and open-time controlled nozzle, the pump-end controlled liquid dosing device as in Figure 1 is compact in that there is no need of mechanical driving or pressure regulator, and it is able to offer higher injection pressure, higher reliability, and higher energy efficiency. However, unlike the conventional system, the output of pump-end controlled liquid dosing device is no longer proportional to the input pulse width. The determination of liquid discharge rate for such a liquid dosing device is far more sophisticated than that for a conventional injection system with a liquid rail and open-time controlled nozzle [7, 9].

For such a liquid supply unit, the PWM pulse time T_i actuated on the solenoid can only be correlated to the liquid injected amount with large errors because too many factors may affect the result. To achieve a higher metering accuracy, a metering theory predicting the liquid discharging rate is needed.

A schematic illustration of the solenoid driven plunger pump of the pump-end controlled pump-nozzle liquid dosing device is shown in Figure 2.

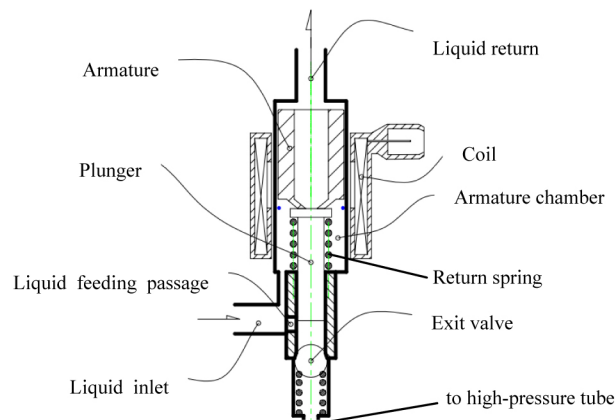


Figure 2. A schematic illustration of the pump structure

The actuator can be considered as an electro-mechanical system, which consists of electro-mechanical energy conversion part and working part. The energy for the forward motion of the armature is from a modulated power. The backward motion is finished by the energy stored in the return spring.

The liquid dosed and injected from the nozzle per pulse equals that supplied by the pump to the high-pressure liquid tube as long as the residual pressure inside the high-pressure tube is kept unchanged. This is true because a check valve is used as the exit valve for the liquid supply to the tube and the nozzle open pressure is constant, and the tube has a limited length.

To achieve a higher metering accuracy of such a liquid dosing system, a metering theory predicting the liquid discharging rate is developed by studying the system using a physical-mathematical model. Experimental investigations verifying the theory are also shown. As a result, the discharging rate is successfully related to a single valued function of $T3$, and so the theory is named as $T3$ metering theory.

The frequency characteristics of the liquid dosing device were also tested, to verify whether the dosed liquid flow could be controlled linearly by PWM frequencies and the constant $T3$ with high precision and a wide appreciation range.

The liquid dosing device was applied and tested on a SCR system set on a 6.5L turbo-charged diesel engine as a verification example. Test results of NOx conversion rate and minimum ESC test NOx emission achievable with the SCR system will be shown, and targeted NOx emission control strategies will be discussed.

FUNDAMENTALS ON THE PUMP

The Process Studied

The process that the pump responds to a power pulse can be characterized by a number of parameters. For a relatively long power pulse, the current passed through the wire of the solenoid starts showing

after the coil is energized, and stops growing when $t=t_{ld}$, then the current keeps unchanged, or drop a little bit in some cases, as time increases, which is considered as a result of the armature moving speed and position [9]. The current goes up again with a relatively small slope after a period of time. Once the solenoid coil is de-energized, the current reaches its maximal, i_p , and then drops to zero rapidly. The armature does not stop going forward after the coil is de-energized until the time $t=t_e$. In the process above, a number of time and position related parameters are involved, some of which will be used as dimension scales in the later analysis. For a given power level, the minimum pulse width resulting in mass discharge onsets is defined as drop-out pulse width. It is found that the drop-out pulse width is almost equal to t_{ld} , the time at which the current passed through the coil wire stops growing. The maximal current for a given power pulse is equal to i_p , which is here defined as end-up current. The maximal distance that the armature can travel for a given power pulse is denoted by x_e , and the time taken is denoted by t_e , the end-up current corresponding to the drop-out pulse width is defined as drop-out current i_d . The maximal distance that the armature can travel under the drop-out pulse width is defined as drop-out displacement, which is denoted by x_d .

Figure 3 shows the physical model of the pump, in which x is the displacement of the plunger with an initial value of x_0 . The operation of the system is related to several different forces, which include spring force characterized by spring coefficient K ; pressure force f_p producing net work for liquid delivery; damper force characterized by damper constant D opposing armature motion; inertia force characterized by total mass M of the moving parts. The sum of the 4 kinds of forces is equal to the electric field force provided by the energy conversion part.

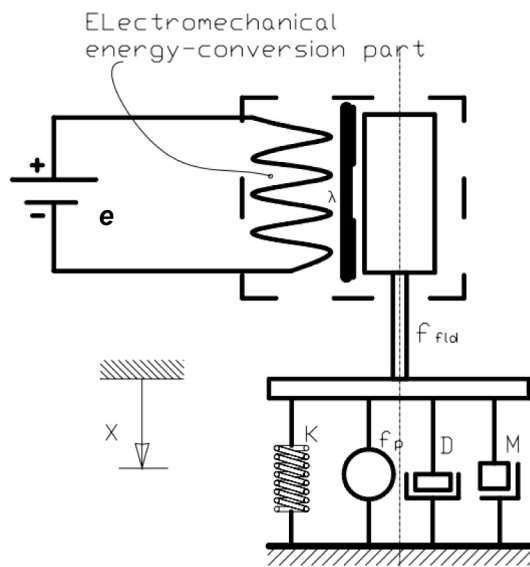


Figure 3. Physical model of the pump

For such an electro-mechanical energy conversion system, the difficult in controlling the liquid delivery amount is that the output of the system is inherently subject to variation due to state changes. The change of the state could be caused by: 1) the electric resistance change

of the wire; 2) the viscosity change of the liquid; 3) phase change of the liquid in the armature chamber [11]; 4) fluctuation of power levels (supply voltage); and 5) mechanical wear of moving parts.

Therefore, the theoretical investigation of liquid measurement model free from the state variation is fairly significant for a pump-end controlled liquid dosing unit. In the following sections, the physical process of such pump's work is systematically studied from a new point of view, which recognizes the close linkage between the energy balance, mass discharge and other relevant electric variables. The study reveals that an electrically characterized variable, $T3$, which is associated with the net output energy, can directly result in a mass discharge. Study of dimensionless variable $T3$ laid a foundation of a whole-state liquid measurement model, which is free from state variation.

Theoretic Description

The system shown in Figure 3 can be described mathematically in terms of an energy conservation equation and an electric equation [12]. The equations vary with the difference in driving circuitry. A typical electric circuit that is used for driving the electro-mechanical system is shown in Figure 4, in which, the MOSFET switch is employed to produce the PWM power pulse, and R is a resistor for protecting the electric switch, r is the electric resistance in the coil wire, and λ is the flow linkage, and ECU is the electronic control unit.

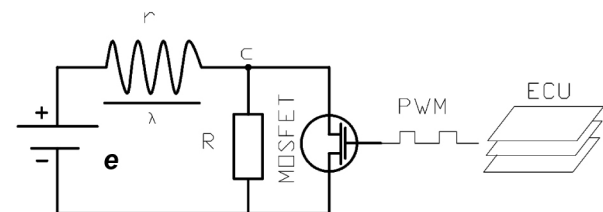


Figure 4. A typical driving circuitry for the pump

Governing Equations

Both energy conservation and electric circuit principles are employed in setting up the governing equations. The total energy output ($ei - r i^2$) dt , of the electromagnetic system within the time increment of dt , is composed of the following items,

- Work done by the spring force, $K(x+x_0)dx$;
- Mechanical energy output by the plunger, $f_p(t)dx$;
- Work done by the damper force, $D(dx/dt)dx$;
- Void electric field energy stored in the system, w_{fld} .

The energy balance equation is then given by,

$$\int_0^{t_i} (ei - ri^2) dt = \int_0^{x_i} D \left(\frac{dx}{dt} \right) dx + \int_0^{x_i} K(x + x_0) dx + \int_0^{x_i} f_p dx + w_{fld} \quad (1)$$

The electric equations are given by

$$\frac{d\lambda}{dt} = e - ir \quad (t < t_i)$$

$$\frac{d\lambda}{dt} = e - iR \quad (t > t_1) \quad (3)$$

Where, i is the electric current in the coil, and e is the power source voltage.

Obviously, it is difficult to solve the governing equations (1), 2, (3) analytically. Besides the unknown plunger force f_p , the damper constant D needs to be determined before the equation is solved, while the damper constant D is function of the thermal state. Nevertheless, the governing equations are too complicated to be solved by a microprocessor in real time during the process of engine operation. Consequently, simplifications of energy balance equation (1) have to be made in practice. One example of simplifications is that only the current integral term on the left side of equation (1) is considered, which is supposed to be proportional to the liquid discharge rate as proposed in reference [13].

Equations in Dimensionless Form

The energy balance equation (1) can be rewritten in a dimensionless form as,

$$\int_0^{T_1} (EI - \bar{r}I^2) d\tau = \int_0^{x_d} \bar{D} \left(\frac{dX}{d\tau} \right) dX + \int_0^{x_d} \bar{K} (X + X_0) dX + \int_1^{x_d} F_p dX + W_{fld} \quad (4)$$

The dimensionless electric equations (2) and (3) are,

$$\frac{d\Lambda}{d\tau} = E - I\bar{r} \quad (\tau < T_1) \quad (5)$$

$$\frac{d\Lambda}{d\tau} = E - I\bar{R} \quad (\tau > T_1) \quad (6)$$

The scales for the no dimensional change are listed as followings.

- Time: t_{1d} , the drop-out pulse width
- Length: x_d , the drop-out displacement
- Mass: Q_0 , the rated maximal mass discharge of the liquid.
- Voltage: v_{ref} the reference voltage for measuring T_3 (see following sections)

The dimensionless variables and parameters are defined by

- $\tau = t/t_{1d}$
- $X = x/x_d$
- $T_1 = t_1/t_{1d}$
- $X_e = x_e/x_d$
- $X_0 = x_0/x_d$
- $F_p = f_p/(Q_0 x_d / t_{1d}^2)$
- $W_{fld} = w_{ld}/(Q_0 x_d^2 / t_{1d}^2)$
- $E = e/v_{ref}$

- $I = i/(Q_0 x_d^2 / t_{1d}^3 v_{ref})$
- $\Lambda = \lambda/(v_{ref} t_{1d})$
- $\bar{r} = r/(v_{ref}^2 t_{1d}^3 / Q_0 x_d)$
- $\bar{R} = R/(v_{ref}^2 t_{1d}^3 / Q_0 x_d)$
- $\bar{D} = D/(Q_0 / t_{1d}^3)$
- $\bar{K} = K/(Q_0 / t_{1d}^2)$

In principle, the mass discharge rate of liquid is associated with the mechanical energy output by the plunger to the liquid.

ENERGY-MASS DISCHARGE RELATIONSHIP

Pressure Model

Before finding the relationship between the mechanical energy output by the piston and the mass discharge of liquid, we need to know the force model that the plunger acts on the fluid. A pressure model is proposed as following.

The pressure within the front chamber of the plunger, P , can be generally expressed as,

$$P = \sum_{j=0}^n \beta_j (X_e) X^j \quad (1 < X < X_e) \quad (7)$$

Where, P satisfies following boundary conditions,

$$P(X=1) = P(X=X_e) = P_0 \quad (8)$$

$$\left. \frac{\partial}{\partial X_e} \left(\frac{\partial^{n-1} P}{\partial X^{n-1}} \right) \right|_{X=1} = 0 \quad (9)$$

Where, P_0 is the sum of exit valve opening pressure and nozzle valve opening pressure, a design parameter with a specific value.

By substituting equation (7) into the boundary conditions (8) and (9), a second order approach to P is obtained, which is,

$$P = \frac{\Omega}{X_e - 1} (X - 1)(X_e - X) + P_0 \quad (10)$$

Where, Ω is constant and is more easily determined by following relation,

$$\Omega = \frac{4P_{\max}}{X_e - 1} \quad (11)$$

Where, $P_{\max}$ is the dimensionless initial pressure in the plunger pump, and which can be measured experimentally.