

Injection Quantity Range Enhancement by Using Current Waveform Control Technique for DI Gasoline Injector

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ABSTRACT

We have achieved injection quantity range enhancement by using the current waveform control technique for direct injection (DI) gasoline injectors. In this study, we developed an injection quantity simulator to find out the mechanism of non-linear characteristics. We clarified the non-linear production mechanism by using the simulator. This simulator is a one-dimensional simulator that incorporates calculation results from both unsteady electromagnetic field analysis and hydraulic flow analysis into the motion equation of this simulation code. We investigated the relation between armature and the injection quantity by using the simulator. As a result, we clarified that the non-linearity was produced by the bounce of the armature in the opening action. Thus, we found that it is effective to reduce the armature bounce to improve the linearity of the injection quantity characteristics.

To reduce the bounce of the valve, we devised a current waveform control technique that includes a current cut-off part just after the armature launching. This current waveform decelerates the armature just before the armature collides into the magnetic core. As a result, the armature bounce can be reduced.

This paper discloses the details of this current waveform and the armature motion by using the developed simulator. Furthermore we discuss whether the developed waveform can improve the injection quantity linearity by using both the simulator and experimental results. Finally, we conclude that the waveform can achieve the minimum injection quantity reduction.

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INTRODUCTION

Recently, the DI gasoline engine has been downsized to reduce fuel consumption by using turbo-charger [1]. One of the issues for the DI engines is particulate matter (PM) reduction in exhaust gas.

PM reduction can be achieved by reducing the fuel adhesion onto a cylinder wall and piston surface. An effective method to reduce the fuel adhesion is splitting the fuel injection multiple times in one combustion cycle [2]. For multiple injections, it is necessary to control the small injection quantity for each single injection. On the other hand, the downsized engine requires large maximum injection quantity because of its high maximum power due to the turbo-charger.

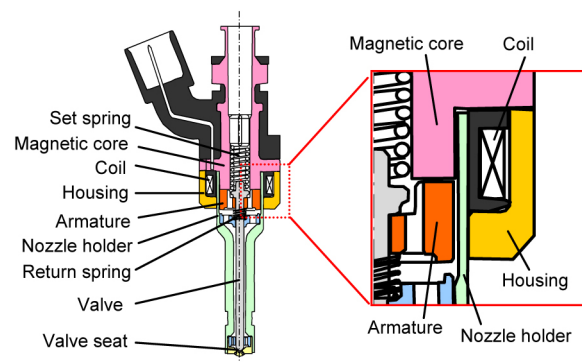


Figure 1. Injector schematic

For this reason, the enhancement of the injection quantity range is a key feature for the injector. To enhance the injection quantity range, reduction of the minimum injection quantity is important. However, there are few reported cases about the simulator for the DI gasoline injector [3].

The minimum injection quantity is defined as the smallest value of the injection quantity range in which the relation of the injection quantity and input pulse width becomes linear. Therefore, to reduce the minimum injection quantity, the linearity of the injection quantity characteristics must be improved.

A typical DI gasoline injector (Fig.1) is based on a solenoid driven valve. The solenoid magnetic force lifts the armature and valve up and the injector starts to eject fuel from its nozzle. The magnetic circuit that causes the magnetic force consists of a solenoid coil, housing, nozzle holder, armature and magnetic core. In addition, the injector used in this report is structured to separate the armature and the valve needle (which is moving part). This injector restricts unexpected post injection caused by valve bounce by reducing movable mass in collision with the valve seat in the valve needle [4].

Fig.2 (a), (b), (c) shows the opening operation of the injector. The closing state is drawn in Fig 2-(a). The closed valve without current supply is loaded in the closing direction by the set spring load and fuel pressure. In the closed state, the solenoid coil is not energized, so the valve remains in contact with the valve seat.

The Fig.2-(b) shows a state when electric current is supplied to the coil. The coil generates magnetic force on the armature. When the magnetic force becomes greater than the spring set load and fuel pressure force, the valve begins to lift. After having arrived at the target lift position when the armature and the magnetic core are in contact, the valve separates from the armature and exceeds the target lift (overshoot). Finally, the valve moves to the closing direction due to the set spring load, and is in the stable opening state on the armature.

Fig.3 (a), (b), (c) shows the valve closing behavior. When the current stops, the magnetic force disappears, and the armature and valve start to close when the opening force becomes less than the closing force. After the valve comes into contact with the valve seat. The armature can continue to move independently. The kinetic energy of the armature can dissipate into the fuel around the armature. Therefore, the valve bounce is reduced, and the valve quickly reaches the closing state. In this design, the inlet of the passage is blocked by the opposing surface of the magnetic core. This is the bounce prevention mechanism.

Fig.4 describes a injection quantity variance caused by piece to piece variation of the injectors. The energization duration time T_i determines the injection quantity. A long injection pulse T_i produces a linear relation between the T_i and the injection quantity Q . On the other hand, the relation becomes non-linear when the injection pulse T_i shortens [5].

Therefore, to reduce the minimum injection quantity, the linearity of the injection quantity characteristics must be improved, and a mechanism analysis of the non-linearity is also necessary.

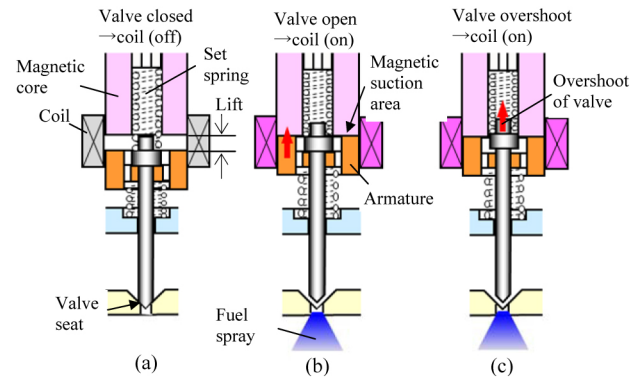


Figure 2. Valve opening behavior

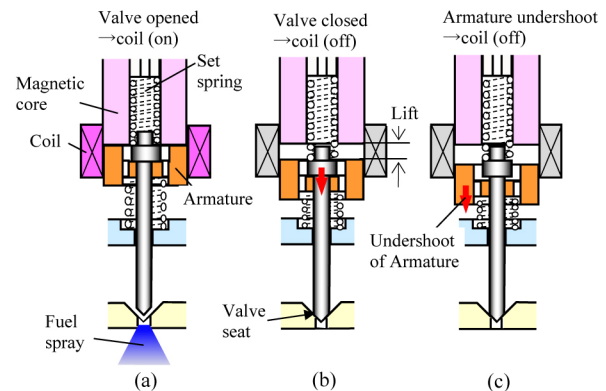


Figure 3. Valve closing behavior

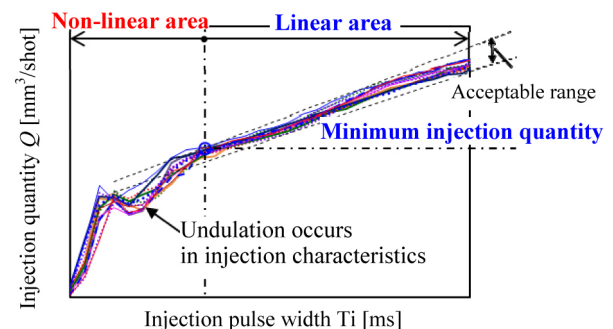


Figure 4. Injection characteristics variance

INJECTION VALVE ACTUATION MODEL

We developed a simulator that can calculate the valve behavior to elucidate non-linearity mechanism of the injection quantity characteristics. This chapter describes the simulator development, we discuss about actuator modeling and result accuracy.

Motion Equations

The moving parts of the injector are the armature and the valve. The simulator includes motion equations for the moving parts. Equations (1) and (2) show the motion equation of the

armature and the valve. When the valve is in contact with the valve seat armature displacement $x_{anc}=0$ and valve displacement $x_{pr}=0$.

The forces acting on the armature are magnetic force F_{mag} , Force F_{ap} by pressure difference between the upper and lower surface of the armature (defined as pressure differential force), Force F_{sq1} by squeeze effect that acts in thin gap between the armature and the magnetic core (defined as squeeze force), Squeeze force F_{sq2} that acts between the armature and valve, Force F_{ab} by the static pressure drop caused by the Bernoulli theorem (defined as Bernoulli-effect force), Spring load F_{zsp} by the return spring, impact force F_{core} by the core, and impact force F_{int} by the valve.

The forces that act on the valve are the resultant force F_p which is caused by the pressure differential force, and the Bernoulli-effect force, set spring load F_{sp} , squeeze force that acts between the valve and armature, impact force F_{int} of the armature, and impact force F_{seat} of the valve seat.

$$M_{anc}\ddot{x}_{anc} = F_{mag} - F_{ap} + F_{sq1} + F_{sq2} - F_{ab} - F_{zsp} - F_{core} - F_{int} \quad (1)$$

(x_{anc} : Displacement of the armature, M_{anc} : Armature mass)

$$M_{pr}\ddot{x}_{pr} = -F_p - F_{sp} + F_{sq2} + F_{int} + F_{seat} \quad (2)$$

(x_{pr} : Displacement of the valve, M_{pr} : Valve mass)

Fig.5 shows the model overview of the injection quantity simulator structured in commercial 1-D solver. Table 1 is a list of the acting forces in equations (1), and (2).

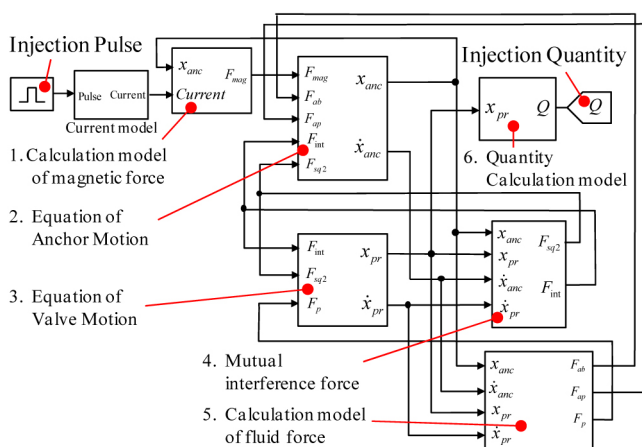


Figure 5. Model overview of simulator

Table.1. List of Acting force

Symbol	Explanation
F_{mag}	Magnetic force(Armature)
F_{ap}	Pressure differential force (Armature)
F_{sq1}	Squeeze force (Armature · Magnetic core)
F_{sq2}	Squeeze force (Armature · Valve)
F_{ab}	Bernoulli-effect force (Armature)
F_{zsp}	Return spring force (Armature)
F_{core}	Impact force (Armature · Magnetic core)
F_p	Resultant of pressure differential force and Bernoulli-effect force (Valve)
F_{sp}	Set-spring force (Valve)
F_{int}	Impact force (Armature · Valve)
F_{seat}	Impact force (Valve, Valve seat)

Acting Forces

This section describes calculation methods for each acting force.

First, we explain the calculation method of magnetic force F_{mag} . Fig.6 shows the magnetic circuit of the injector solenoid. The magnetic force that acts on the armature is determined by magnetic flux ϕ_s in equation (3).

$$F_{mag} = \frac{B_s^2 S}{2\mu_0} = \frac{\phi_s^2}{2\mu_0 S} \quad (3)$$

(μ_0 : Magnetic permeability in vacuum, B_s : Magnetic flux density of the suction side [Wb], S :suction area[m²])

We can calculate the magnetic flux ϕ_s from magnetic flux leakage ϕ_{md} and total magnetic flux ϕ_{total} by using equation (4). The total magnetic flux ϕ_{total} occurs in the whole magnetic circuit, and the ϕ_{md} is magnetic flux leakage which travels through a pipe shaped part (nozzle holder) surrounding the armature and the magnetic core.

$$\phi_s = \phi_{total} - \phi_{md} \quad (4)$$

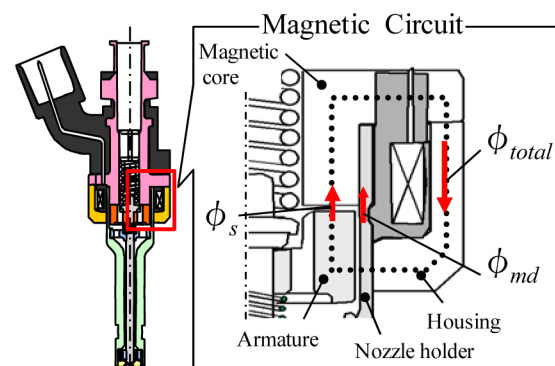


Figure 6. Magnetic circuit of injector