Application of High-speed Solenoid Valve to the Semi-active Control of Landing Gear

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Abstract

To select or develop an appropriate actuator is one of the key and difficult issues in the study of semi-active controlled landing gear. Performance of the actuator may directly affect the effectiveness of semi-active control. In this article, parallel high-speed solenoid valves are chosen to be the actuators for the semi-active controlled landing gear and being studied. A nonlinear high-speed solenoid valve model is developed with the consideration of magnetic saturation characteristics and verified by test. According to the design rule of keeping the peak load as small as possible while absorbing the specified shock energy, a fuzzy PD control rule is designed. By the rule, controller parameters can be self-regulated. The simulation results indicate that the semi-active control based on high-speed solenoid valve can effectively improve the control performance and reduce impact load during landing.

Keywords: landing gear; shock absorber; semi-active control; high-speed solenoid valve

1 Introduction

Since the advent of semi-active control, it has gotten comprehensive researches and applications in the areas of vehicle suspension system, building shock absorption, bridge damping and so on[1-3].

In the early nineties of the 20th century, this control method was introduced into the domain of landing gear to improve the landing and taxiing performances of aircraft[4]. As for landing and taxiing, the latter is similar to vehicle traveling and attracts more researchers[4-5].

In contrast to vehicle traveling, the actuator for the semi-active control of landing gear should act as many times as possible in a short time during landing. Such requirement for actuator is difficult to be met and this problem has not been solved perfectly yet[5]. It has become the bottleneck of the study on the semi-active control of landing impact. Many articles did not take the actual actuator into account, and say nothing about the actual actuator’s performance[5-6]. Without the study on effective actuator, the semi-active control application to landing gear is only a mirage.

In the previous study, we did the drop test for the landing gear with fixed orifice and by means of the test, and verified the landing gear dynamic model we developed[7-8]. Additionally, we did some research on the semi-active control methods[9]. We chose high-speed solenoid valves as the actuators of semi-active control. To compensate for the smaller flux, we adopted parallel high-speed solenoid valves and developed a semi-active controlled landing gear simulation model. Then we studied the shock absorber performance of semi-active controlled landing gear[10]. In the studies mentioned above, we did not take into account the characteristics of the
valves or simplify it by linearizing the displacement of valve’s movable parts, and did not take the loads into consideration. These may bring about some inaccuracies.

In this article, considering the magnetic saturation, we have developed a nonlinear high-speed solenoid valve model and verified it by the test. The dynamic model of semi-active controlled landing gear based on the high-speed solenoid valve is set up; according to the design rule of keeping the peak load as small as possible while absorbing the specified shock energy, a fuzzy PD control rule is designed to realize the self-regulating parameters. Finally, we make a comparison between passive and semi-active controlled shock absorbers by simulation.

2 Mechanism of the Semi-active Control of Landing Gear

Fig.1 shows the mechanism of semi-active controlled shock absorber based on the parallel high-speed solenoid valves.

![Fig.1 Mechanism of a semi-active controlled shock absorber.](image)

This is an oleo-pneumatic shock absorber and different from that of the general types: apart from the basic orifice through which the oil flows from upper chamber into lower chamber or conversely, there is a bypass system which consists of two pipes and six parallel high-speed solenoid valves. Opening or closing these valves can control the flow which is passing through the valves. As a result, the damping of the shock absorber can be controlled.

During the landing and taxiing of aircraft, the control system receives signals from all kinds of sensors and gets information about airframe and shock absorbers, and then changes the opening of the valves to get good performances of landing and taxiing.

3 Mathematical Model of High-speed Solenoid Valve

In this article, a domestic two-way, normally closed high-speed solenoid valves are used. Fig.2 shows the structure of the high-speed solenoid valve. Table 1 shows the valve’s main structural parameters.

![Fig.2 High-speed solenoid valve’s structure.](image)

### Table 1 Valve’s main structural parameters

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Movable parts mass/g</td>
<td>5</td>
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<tr>
<td>Spring stiffness/(N·m⁻¹)</td>
<td>1 600</td>
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<tr>
<td>Valve port diameter/mm</td>
<td>1.32</td>
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<tr>
<td>Coils turns</td>
<td>385</td>
</tr>
<tr>
<td>Coil resistance/Ω</td>
<td>1.6</td>
</tr>
<tr>
<td>Armature maximum displacement/mm</td>
<td>0.45</td>
</tr>
<tr>
<td>Spring precompression/mm</td>
<td>0.7</td>
</tr>
<tr>
<td>Initial air gap/mm</td>
<td>0.5</td>
</tr>
<tr>
<td>Armature diameter/mm</td>
<td>8</td>
</tr>
<tr>
<td>Friction/N</td>
<td>&lt;0.55</td>
</tr>
</tbody>
</table>

When the solenoid is energized, after overcoming spring tension, friction and hydrodynamic force, the armature driven by magnetic force makes the ball move to right with the aid of actuator pin. Thus, the valve will open. When the solenoid is de-energized, the spring tension, friction and hydrodynamic force will make the ball move to left, as a result, the valve is closed.
Owing to the compact structure and light movable parts, the valve has a rapid response speed up to 200 Hz\cite{11}.

For general solenoid valve, the magnetic saturation property is usually neglected in many articles. However, to increase the opening speed of valve, high-speed solenoid valve is usually working in saturated region, and there will be a larger error if it is not taken into consideration or simply treated.

Fig.3 shows various simplified flux paths and the equivalent magnetic circuit of the high-speed solenoid valve.

![Simplified flux paths](image)

![Equivalent magnetic circuit](image)

Fig.3  Flux paths and equivalent circuit.

There are mostly three air gaps, they are respectively one main gap (working gap) and two secondary gaps (gap1 and gap2). The fringe effects of main gap is taken into account and those of the other gaps are neglected because their magnetic reluctance are much less than that of main gap. In Fig.3(a), path (1) is the magnetic circuit of main gap; paths (2)-(3) are the fringing flux paths; Paths (4)-(5) are the leakage flux paths\cite{12-14}. In the model, we assume the flux density is constant over the cross section of magnetic material.

Referring to Fig.3(b), the magnetomotive force is:

\[
\varepsilon_m = NI = \Phi R_m
\]

where \(N\) is coil turns, \(I\) current, \(R_m\) total magnetic reluctance, and \(\Phi\) total magnetic flux:

\[
\frac{1}{R_m - R_{\text{yoke}}} = \frac{1}{R_L} + \frac{1}{R_{\text{armature}} + R_{\text{maingap}} + R_{\text{gap1}} + R_{\text{gap2}} + R_{\text{polepiece}}}
\]

\[
\frac{1}{R_{\text{maingap}}} = \frac{1}{R_1} + \frac{1}{R_2} + \frac{1}{R_3} + \frac{1}{R_4}
\]

where \(R_1-R_4\) are respectively corresponding to the magnetic reluctance of magnetic flux paths (1)-(4) shown in Fig.3(a)\cite{12}, and \(R_1\) is corresponding to the magnetic reluctance of magnetic flux path (5)\cite{13}.

\[
R_1 = \frac{I}{\mu_0 R_1}
\]

\[
\frac{1}{R_2} = 1.63 \mu_0 \left( r_1 + \frac{I}{4} \right)
\]

\[
\frac{1}{R_3} = 2 \mu_0 r_1 \ln \frac{4(r_2 - r_1)}{\pi l}
\]

\[
\frac{1}{R_4} = \mu_0 \left[ \frac{\pi h (r_4 + r_1)}{8 (r_4 - r_1)} - (r_4 + r_1) + \frac{2(r_1^2 - r_2^2)}{\pi h} \right]
\]

where \(\mu_0\) is air permeability, and the others are shown in Fig.3(a).

The magnetic permeability \(\mu\) of kinds of magnetic material, such as armature, pole piece, yoke, etc., are treated as a variable. In this article, \(\mu\) can be determined by trial and error method and consulting the \(B-H\) curve\cite{15}.

For coil, there is \(\Phi N = LI\), and according to the Kirchhoff’s voltage rule and Eq.(1), we can get

\[
E_s = (\tilde{R}_s + \tilde{R}_L) I + N \frac{d\Phi}{dt}
\]

\[
(\tilde{R}_s + \tilde{R}_L) I + \frac{N^2}{R_m} \left( \frac{dI}{dt} - I \frac{dR_m}{dt} \right)
\]
where \( \tilde{R}_e \) and \( \tilde{R}_l \) are power supply resistance and coil resistance, respectively, and the expression can be transferred to

\[
\frac{dI}{dt} = \left[ E_s - (\tilde{R}_e + \tilde{R}_l)I \right] R_m + I \frac{dR_m}{dt}
\]

(10)

The electromagnetic force that acts on the armature of valve can be given by

\[
F_E = \frac{\Phi_{air}^2}{2\pi\mu_0r^2}\lambda^2
\]

(11)

where \( \Phi_{air} \) is magnetic flux passing through the working air gap, and \( \lambda \) the leakage coefficient of the main air gap is

\[
\Phi_{air} = \frac{(R_{armature} + R_{maingap} + R_{gap1} + R_{gap2} + R_{polepiece})\Phi}{R_{armature} + R_{maingap} + R_{gap1} + R_{gap2} + R_{polepiece} + R_l}
\]

(12)

\[
\lambda = \frac{R_l}{R_{maingap}}
\]

(13)

The dynamic equation of movable parts is

\[
m_v (\frac{d^2x_v}{dt^2} - \eta g) + C_s \frac{dx_v}{dt} + k(x_v + l_0) = F_E - f - f_c
\]

(14)

where \( m_v \) is the total mass of movable parts including armature, actuator pin, etc., \( g \) acceleration of gravity, \( C_s \) viscous damping coefficient, \( k \) spring stiffness, \( l_0 \) spring precompression, \( f_c \) coulomb friction, \( f_h \) hydraulic power, and \( x_v \) movable part displacement.

\[
x_v = l_{g0} - l_g
\]

(15)

where \( l_g \) is the length of main gap which is a variable during valve open and close and \( l_{g0} \) is the initial length of main gap.

In Eq.(14), \( \eta \) is the gravity influence factor. According to the valve’s normal fixed mode, \( \eta \) can be determined as follows

\[
\eta = \begin{cases} 
1 & \text{gravity is in the same direction with } F_E \\
0 & \text{gravity is perpendicularity to } F_E \\
-1 & \text{gravity is opposite to } F_E 
\end{cases}
\]

(16)

In many articles, the gravity’s influence is usually neglected. By means of test, however, we have found that under the situation of no-load, the gravity can greatly influence the performance of the valve, especially influence the valve’s opening time. Fig.4 shows the coil current during valve open and close.

4 Simulation and Verification

The developed valve’s model mentioned above should be verified by test. Under the existing condition, however, it is hard to measure the valve’s parameters such as armature displacement, electromagnetic force and so on. In this article, we adopted the method of measuring the coil current of the
high-speed solenoid valve to verify the model.

4.1 Test scheme

In Fig.5, a signal generator sends a square wave signal to control the switching tube, thereby to control the opening or closing of the high-speed solenoid valve.

![Valve test schematic diagram](image)

Fig.5 Valve test schematic diagram.

To learn the working state of the high-speed solenoid valve under the conditions of saturation and nonsaturation, 12 V and 2 V are loaded, respectively.

Because the data acquisition system adopted by us can only measure the voltage below 10 V, we chose two precision resistances in series to share the voltage that is loading on valves. Setting \( R_{p1} = R_{p2} \), then we can get the valve’s voltage \( V_{\text{valve}} \) through measuring the \( R_{p2} \)’s voltage:

\[
V_{\text{valve}} = 3V_{R_{p2}} \quad (17)
\]

By measuring the \( V_r \) of the precision resistance \( R_r \), we can get the current \( I_r \) that flows through the high-speed solenoid valve.

The data acquisition and control system adopts the method of interrupt mode and the sampling frequency is 10 kHz.

4.2 Simulation and verification

The measured voltage \( V_{\text{valve}} \) acts as the input of high-speed solenoid valve’s simulation model. Comparing the valve’s coil current acquired from the simulation with that acquired from the test, we attain the results as shown in Fig.6.

![Valve simulation and test](image)

Fig.6 Valve’s simulation and test.

As shown in Fig.6, the simulation curve and the test curve are fitted closely, and the simulation model can be thought to be accurate. We can use the model to find the way to improve valve’s performance and to study the semi-active control of landing gear based on the high-speed solenoid valve.

In addition, from Fig.6(c) we learn that valve will work at nonsaturation region when it is loaded with 2 V voltage, and will work at saturation region when it is loaded with 12 V voltage. To distinguish them easily, only the first half part of curve, when it is loaded with 12 V voltage, is shown in Fig.6(c).

The magnetization curve of the material used
in this article is static curve and we have ignored the hysteresis characteristic and assumed that the magnetic flux density is constant over the cross section of magnetic material. All of these may make some errors.

5 Control Strategy

5.1 Semi-active control objective

The semi-active control of landing gear aims to restrict the strut stroke within a given limits and to avoid over high peak load through controlling the valves according to system parameters.

As stated previously, following the design rule that keeps the peak load as small as possible while the specified shock energy is absorbed, we designed a control rule fitting the semi-active control drop test rig[7].

$$F = F_N = \frac{W_e}{S_{\text{max}}}$$

(18)

where $W_e$ is the energy needed to be absorbed by shock absorber in the first stroke, and $S_{\text{max}}$ the maximum stroke of the shock strut. Then, the lowest peak load is realized.

5.2 Parameter self-regulating fuzzy PD controller design

Based on the conventional PD controller, a parameter self-regulating module is introduced. The regulating law is determined by a series of fuzzy control rules so that the gain of the PD controller can be adjusted in real time. Then the system performance will be able to be improved further by the combination of the model characters and the expert knowledge.

(1) Control system framework

Fig.8 shows the fuzzy PD control strategy of the semi-active controlled landing gear based on the high-speed solenoid valves.

The error $\varepsilon$ is gotten by $F$ minus $F_N$ and its variation $d\varepsilon$ is taken as the input of the PD controller. The parameters $k_p$ and $k_d$ of PD controller are self-regulated on line by fuzzy reasoning. The acceleration of airframe is measured in real time. According to the following equation, the force $F$ is calculated:

$$F = M_1(g - a)$$

(19)
where \( a \) is the acceleration of airframe, and \( M_1 \) is the mass of airframe (sprung mass).

According to the magnitude of control signals and the parameters of valve, the controller will determine how many valves to be driven.

After many times of simulations and analyses, we have reached the conclusion: the valves should not be controlled until the loads of shock absorber exceed \( F_N \) for the first time (the \( S_0 \) point as shown in Fig. 7). The reason is that in the previous phase, the loads rise too quickly and it will cause large overshoot because of the delay and discreteness of the controller. So, in this specific phase all of the high-speed solenoid valves are open so as to get the maximal area of the orifice.

(2) Membership function and control rule

The initiate values of \( k_p \) and \( k_d \) are set respectively as 0.000 4 and 0.03.

The domain of input variables \( \varepsilon \) and \( d\varepsilon \) as well as output variables \( \Delta k_p \) and \( \Delta k_d \) are defined to be “PB, PM, PS, ZO, NS, NM and NB”, and these variables are quantified as follows

\[
\begin{align*}
\varepsilon & \rightarrow (-12,000, 3,000) \\
d\varepsilon & \rightarrow (-3,000, 3,000) \\
\Delta k_p & \rightarrow (-0.006, 0.006) \\
\Delta k_d & \rightarrow (-0.006, 0.006)
\end{align*}
\]

The membership function of input variables follows the trapezoid distribution and the membership function of output variables is composed of triangular, S and Z type membership functions.

The fuzzy control rules of \( \Delta k_p \) and \( \Delta k_d \) are showed respectively in Table 2 and Table 3.

### Table 2 Fuzzy control rule of \( \Delta k_p \)

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<tr>
<th>( \Delta k_p )</th>
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<th>NM</th>
<th>NS</th>
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### Table 3 Fuzzy control rule of \( \Delta k_d \)

<table>
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<tr>
<th>( \Delta k_d )</th>
<th>( d\varepsilon )</th>
<th>NB</th>
<th>NM</th>
<th>NS</th>
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The fuzzy controller is adopted to be Mamdani type and the mean of maximum method is used to defuzzify \( \Delta k_p \) and \( \Delta k_d \).

6 Results of the Semi-active Control Simulation

Based on the model of shock absorber with fixed orifice which has been verified by drop test, we conduct many times of simulations and find out the scheme which can reach the maximal efficiency. Then, we compare the performance of the optimal fixed orifice shock absorber with that of the semi-active controlled shock absorber based on high-speed solenoid valves.

Table 4 and Fig.9 show the simulation results of passive control and semi-active control.

### Table 4 Efficiency statistics

<table>
<thead>
<tr>
<th>Shock absorber efficiency</th>
<th>Total efficiency</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fixed orifice (selected)</td>
<td>0.88</td>
</tr>
<tr>
<td>Semi-active control</td>
<td>0.95</td>
</tr>
</tbody>
</table>

(a) Wheel load time history
Fig. 9(c) and Fig. 9(d) are respectively the efficiency diagrams of shock absorbing system (including shock absorber and wheel) and shock absorber. The former describes the relationship between wheel load and airframe displacement (consisting of shock absorber stroke and tire deflection). The efficiency diagram of shock absorber describes the relationship between shock absorber load and shock absorber stroke.

The results indicate that the efficiency of the semi-active controlled shock absorber is better than the fixed orifice shock absorber, and the peak values of acceleration or total load are lower and changed gently.

7 Conclusions

According to the simulation mentioned above, choosing high-speed solenoid valve as the semi-active control actuator is feasible, and the shock absorber’s performance is superior to that of the optimal fixed orifice.

To increase further the efficiency of the shock absorber, we should proceed from two respects: to increase responding speed of the valve, and to explore the more effective semi-active control rule.

Certainly, the more important worthiness of semi-active control of landing gear consists in changing the damping of shock absorber in real time according to practical landing or taxiing conditions, and being able to get better or best control effect in any conditions. On the other hand, the landing gear with fixed orifice or metering pin can get the best control only under the defined working condition. The control system should have better adaptability and robustness.

Limited by the objective conditions, tests done with the high-speed solenoid valve are not enough. In future research, we will deeply study the working characters of high-speed solenoid valve along with the further testing work, especially the characters of flow with load, and build more precise mathematical model.

At present, the landing gear drop test rig based on the parallel high-speed solenoid valves is about
to be completed, we will study the more effective control rule by simulation and test, and make a further research on the semi-active control of landing gear based on the parallel high-speed solenoid valves.

References


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