Chapter 7
Experiments

The primary purpose of this chapter is to present an experimental evaluation of the theoretical and analytical results presented in the preceding chapters. It describes the experimental setup, presents the results, and provides a detailed analysis and interpretation of the results. Through the experimental data, the structural dynamics effects are observed on the MR damper performance and included in the MR damper model to better represent the seat suspension model. A simplified model is developed for the experimental seat suspension. After a reasonably accurate model is developed, the adaptive control algorithm is applied in its simplified formulation, in order to reduce the computational burden for the real time control. The algorithm is implemented on the dSPACE AutoBox. The experimental results show the effectiveness and feasibility of the adaptive control algorithm.

7.1 Experimental Setup
The experiment is set up at the Advanced Vehicle Dynamics Laboratory (AVDL) of Virginia Tech. The testing system includes the hydraulic actuation system that is used to vibrate the seat suspension, the dSPACE AutoBox that works as a real-time controller, the dSPACE TRACE that is used for data acquisition, and the dSPACE COCKPIT that can be used to adjust the controller parameters on-line. A detailed description of the hardware is included in Section 4.1 and Appendix A.

A schematic of the semiactive adaptive seat suspension system is shown in Fig. 7.1. The adaptive controller uses the seat acceleration and the suspension relative displacement to determine the current to the damper. A rate filter as discussed in Section 4.2.2 is employed to estimate the relative velocity for the control algorithm. The adaptation process occurs based on the MR damper model, suspension relative velocity, suspension relative displacement, seat acceleration, and the current supplied to the damper, as was described in Section 5.4.
The following sections provide the details of the excitation signal, noise filter for improving signal-to-noise ratio, as well as the seat suspension model used for experiments.

### 7.1.1 Excitation Signals

Different excitation signals were considered for testing the adaptive seat suspension including pure tone, swept sine or chirp, and different ISO-recommended signals, as shown in Figs. 7.2-7.4. Figures 7.2 and 7.4a are ‘strips’ plots with respect to time. One of the purposes is that dynamic characteristics can be demonstrated clearly through different excitations.

The pure tone sine excitation we considered included 1.5 and 3.5 Hz sine signals, as shown in Fig. 7.2. Recalling that the isolation frequency of the seat suspension is approximately 2 Hz, 1.5 Hz and 3.5 Hz pure tone signals were chosen in order to evaluate the system response at frequencies smaller and larger than the isolation frequency. As will be shown later, the adaptation of the electrical current exhibits opposite trends at these frequencies.

The swept-sine (chirp) signal was selected to have a time-varying amplitude in the frequency range of 0.6 to 6.0 Hz. It reaches the isolation frequency of 2 Hz in 12 to 16 seconds, as shown in Fig. 7.3. The swept sine was selected to evaluate the tracking ability of the adaptive control algorithm.
Figure 7.2 Time Trace of 1.5 and 3.5 Hz Sinusoidal Excitation

Figure 7.3 Swept Sine (Chirp) Signal; (a) Time Trace; (b) Sweep Rate
Figure 7.4 International Standard Organization (ISO) Excitation Signal;
(a) Time Trace; (b) Frequency Spectrum
Finally, a combination of the signals recommended by the International Standard Organization (ISO) is chosen (also used in Chapter 4), as shown in Fig. 7.4. The ISO signals include a series of random excitations, generally with large amplitudes at various lower frequency ranges (as shown in Fig. 7.4b), for evaluating vehicle suspension effects on human comfort. They are widely used throughout the industry for comparing different systems, such as the seat suspensions used in this study.

### 7.1.2 Noise Filter

Since the noise in the measurement signals affects the control system performance, it is important to design proper filters to eliminate the noise. As shown in Fig. 7.1, a noise filter is used before the measurement signals are supplied to the adaptive algorithm. It is assumed that the source vibration frequency for the seat suspension is below 6 Hz. Therefore, the break frequency of the noise filter can be set to 7 Hz, and the filter is 13th order with a stopband that is 20 dB down from the passband by using the MATLAB 5.2 analog filter function (see Section 6.2.2, too). The Bode plot of the noise filter is shown in Fig. 7.5, and its effect on the noise are presented in Fig. 7.6.

![Bode Plot of a 13th Order Cheby2 Noise Filter](image)

**Figure 7.5 Bode Plot of a 13th Order Cheby2 Noise Filter**
Figure 7.6 Measurement Noises in the Seat Suspension; (a) In Time Domain; (b) In Frequency Domain
It can be noted that below 7 Hz the gain is unity and there is obvious phase shift when frequency is near 7 Hz. The measurement noises of before- and after-filtering from a static seat are shown in Fig. 7.6. It shows that the signal amplitude below 7 Hz can be preserved while the higher frequency components are filtered out. Thus the noise filter can improve the signal-to-noise ratio for the adaptive algorithm, even though it can introduce phase errors (as shown in Fig. 7.5) into the filtered signals. As such the filter design should be trade-off with the control system performance.

In the following sections, we will present the experimental results and the implementation of the adaptive control in detail.

### 7.2 Experimental Parameters

This section will show how to experimentally determine such parameters as mass, stiffness, and resonant frequency. The analysis of kinematics and dynamics are presented. Finally, a simplified seat suspension model is built for the adaptive algorithm.

#### 7.2.1 Experimental Determination of Stiffness and Mass

If the volume of the air spring is constant, the air pressure inside the air spring determines the suspension stiffness. We can assume the air pressure remains relatively unchanged during the test because of the limited suspension travel. Therefore, the stiffness of the air spring can be assumed to be constant. In order to measure the stiffness, the MR damper was removed from the seat suspension. The difference between the static displacements of the seat with 45.34 Kg (100 lb) and 68.01 Kg (150 lb) weights placed on the seat was measured. The displacement difference was 5.396 cm (2.12 in) in response to the 22.67-Kg (50-lb) mass difference, therefore

\[
K = \frac{22.67 \text{ Kg}}{5.396 \text{ cm}} = \frac{222.4 \text{ N}}{5.396 \text{ cm}} = 41 \text{ N/cm} = 4100 \text{ N/m}, \quad (7.1)
\]

\[
K = \frac{50 \text{ lb}}{2.12 \text{ in}} = 24 \text{ lb/in}, \quad (7.2)
\]
Figure 7.7 Comparison of Simulation and Experimental Results without the MR Damper as a Linear System; (a) for 1.5 Hz Excitation; (b) for 3.5 Hz Excitation
In order to determine the effective mass of the seat, 1.5 Hz and 3.5 Hz pure tone signals were used to excite the seat suspension without the MR damper. Both suspension relative displacement and seat acceleration were measured. The data then was used to identify the seat mass \(M\) and the linear damping coefficient \(C\) by using the simplified model

\[
M\ddot{y} + C\dot{y}_{12} = -Ky_{12}
\]

(7.3)

where \(y_{12}\) is the vertical relative displacement of the vehicle seat suspension as shown in Fig. 4.4. This resulted in

\[
M = 77 \text{ Kg}
\]

\[
C = 82 \text{ N} \cdot \text{s/m}
\]

(7.4)

As shown in Fig. 7.7, the simplified model in Eq. (7.3) represents the system dynamics without the MR damper with sufficient accuracy. Using Eqs. (7.2) and (7.4), we can determine the undamped natural frequency of the suspension

\[
f_{\text{seat}} = \frac{1}{2\pi} \sqrt{\frac{K}{M}} = 1.16 \text{ Hz}
\]

(7.5)

### 7.2.2 Determination of the Damped Resonant Frequency

The resonant frequency is an important region for checking the effectiveness of the adaptive control algorithm. Equation (7.5) indicates the undamped resonant frequency. We determined the damped resonant frequency for the seat suspension with no electric current to the damper. This is an approximate method. A series of sinusoidal excitations from 1.1 to 1.8 Hz with the same amplitude were used to excite the suspension and pick out the peak-peak values. The frequency corresponding to the maximum value is considered to be the damped seat resonant frequency.
We used the testing data, listed in Table 7.1 and also presented in Fig. 7.8, to determine the resonant frequency of the seat suspension with 0 A to the MR damper. The experiment was performed while maintaining a fixed 1.27 cm (0.5 in) in displacement input at the hydraulic actuator. The fixed displacement was accomplished by setting the MTS controller in the displacement mode, which ensured that a preset displacement was maintained.

| Table 7.1 Testing Data of Peak-to-Peak Values of the Seat Relative Displacement |
|---------------------------------|---------------------------------|
| Frequency (Hz) | Peak-to-peak Relative Displacement (m) |
| 1.1              | 0.0028, 0.0088                  |
| 1.2              | -0.0233, 0.0430                 |
| 1.25             | -0.0232, 0.0448                 |
| **1.27**         | **-0.0263, 0.0481**             |
| 1.3              | -0.0215, 0.0448                 |
| 1.4              | -0.0215, 0.0430                 |
| 1.6              | -0.0214, 0.0358                 |
| 1.8              | -0.0150, 0.0310                 |

Figure 7.8 Peak-to-Peak Response of the Seat Suspension with 0 A to MR damper
As shown in Fig. 7.8, the damped resonant frequency of the seat suspension is approximately 1.27 Hz for 0-A current supplied to the MR damper. It should be noted that the two resonant frequencies of 1.27 Hz and 1.16 Hz (determined in the previous section) are different from the two different testing systems because the MR damper adds a slight amount of stiffness to the seat suspension.

7.2.3 Seat Kinematics
The seat kinematic mechanism is shown in Fig. 7.9. It should be noted that joints A and B can have rotation only, while joints C and D are allowed to have both rotation and linear sliding motion along the track that captures them. An angular displacement sensor is installed at the rotation joint E. The dimensions b and e remain unchanged at 7.25 in and 13 inch, respectively, while other dimensions vary with the suspension motion. The maximum of the dimensions $y_{12}$ and c are 8.75 inch and 6.30 inch, respectively. The velocity across the MR damper can be estimated if c can be calculated in real time based on the suspension displacement ($y_{12}$) and the geometric relationship. In the physical seat, the damper and the link AD almost overlap. Therefore, in the following discussion, we assume that they have the same angular motion represented by $\beta$.

![Figure 7.9 Geometric Configuration of the Experimental Seat](image-url)
The procedure is as follows

1. Determine the equilibrium point $\theta_e$ and the other geometric dimensions. It is worth noting that when the adaptive control algorithm is applied, the equilibrium point need not be known;
2. Read the angle $\theta$ from the angular sensor at E;
3. Calculate the relative displacements $y_{12}$ and $\Delta a$;
4. Calculate the MR damper length $c$ using

$$c = (a^2 + b^2 - 2ab \cos \theta)^{1/2} \quad (7.6)$$

5. Calculate the angle $\beta$ from the relationship

$$\beta = \sin^{-1}(b \sin \theta / c) \quad (7.7)$$

Based on the above geometric analysis, we can figure out the relationship between the suspension travel ($y_{12}$) and the damper length ($c$) variations around the equilibrium point of $\beta=25^\circ$ for our seat suspension system. Figure 7.10 shows that there exists a linear relationship. The geometric analysis will be further verified by the corresponding velocity relationship, Section 7.2.6.2.
Figure 7.10 Relationship of Suspension Distance and Dmaper Length Variations from the Mechanical Dimension Sizes

7.2.4 Seat Dynamics

The seat suspension can be considered as a suspended mass $M$ with a spring force $F_s$ and a vertical damping force $F_{vdf}$, shown in Fig. 7.11.

![Figure 7.11 Free Body Diagram of the Seat Suspension](image-url)
Furthermore, the MR damper can be considered to provide a vertical damping force to balance the seat dynamic load \( F \) by ignoring the frictional force and torque as

\[
F_{vd} = -F = -M\ddot{y}_1 - F_s
\]  

(7.8)

In order to simplify the force analysis, we can assume

1. \( F \) is at the middle of AC;
2. One end of the damper is at Joint D;
3. Frictional forces (i.e., \( F_{Cf} = F_{Df} = 0 \)) and torque at any joint are ignored, where \( F_{Cf} \) is the force acting between AC and BC as shown in Fig. 7.11. It is worth noting that the other forces used for the following derivation can also refer to Fig. 7.11.

Thus, for the link AC, we have

\[
F_{AX} = F_{CY} = 0
\]  

(7.9a)

\[
\sum M(H) = 0, F_{AY} AH = CH F_d, \text{ and since } H \text{ is the middle point of the link AC},
\]

\[
F_{AY} = (CH/AH)F_d < F_d
\]  

(7.9b)

where \( F_{AX}, F_{AY} \) and \( F_d \) are the forces from Joints A and C, respectively.

By noting that \( F_{AX}=F_{Df}=0 \), the link AD gives

\[
\sum F_X = 0, F_{EX} = -F_{mr} \cos\beta + F_{AX} + F_{Df} = -F_{mr} \cos\beta
\]  

(7.10a)

\[
\sum M(D) = 0, (\frac{1}{2} AD \cos\theta) F_{EY} + (\frac{1}{2} AD \sin\theta) F_{EX} = AD \sin\theta F_{AX} + AD \cos\theta F_{AY}
\]  

(7.10b)

Thus
Figure 7.12 Free Body Analysis of the Seat Suspension Linkage;  
(a) AC member; (b) AD member; (c) BC member

\[ F_{EY} = 2F_{AY} - F_{EX} \tan \theta = 2F_{AY} + F_{mr} \tan \theta \cos \beta = 2F_d + F_{mr} \tan \theta \cos \beta \quad (7.10c) \]

For BC:

\[ \sum M(B)=0, \]
\[ F_d (BC \cos \theta) + F_{EY} (\frac{1}{2} BC \cos \theta) \]
\[ = F_{EX} (\frac{1}{2} BC \sin \theta) + (F_{mr} \sin \beta) (c \cos \theta) + (F_{mr} \cos \beta) (c \sin \theta) \quad (7.11a) \]

Since BC = e (i.e., the length of BC shown in Fig. 7.9), simplifying Eq.(7.11a) gives

\[ F_d + \frac{1}{2} F_{EY} = \frac{1}{2} F_{EX} \tan \theta + c/e F_{mr} \sin(\theta+\beta) / \cos(\theta) \]
\[ F_d + F_d + \frac{1}{2} F_{mr} \tan \theta \cos \beta = - \frac{1}{2} F_{mr} \cos \beta \tan \theta + c/e F_{mr} \sin(\theta+\beta) / \cos(\theta) \]
Thus

\[ F_d = \{ - \cos \beta \tan \theta + c/e \sin(\theta + \beta)/\cos(\theta) \} F_{mr}/2 \quad (7.11b) \]

Thus, finally the relationship between the MR damping force and the vertical damping force is

\[ F_{vdf} = F_{AY} + F_d = (1+CH/AH)F_d \leq 2F_d \]

Therefore, substituting Eq. (7.11b) into the above equation gives out

\[ F_{vdf} \leq \{ - \cos \beta \tan \theta + c/e \sin(\theta + \beta)/\cos(\theta) \} F_{mr} \quad (7.12) \]

### 7.2.5 Seat MR Damper Model

The procedure used in Chapter 3 for developing non-parametric model of MR dampers was used to establish the damper model needed for our experimental work. As outlined in Fig. 3.2, we obtained the force-velocity data by testing our MR damper with sinusoidal excitation. The tests were conducted on an MTS load frame, capable of displacing the damper across its two ends at both a fixed frequency of 2.45 Hz and amplitude of 1 inch with varying currents supplied to the MR damper. The currents were 0, 0.25, 0.50, 0.75, and 1.0 A, respectively. The data was collected with an HP35665 Dynamic Signal Analyzer with a sampling frequency of 500 Hz.

Referring to Eqs. (3.3) to (3.7), the damper model is established as

\[ A_{mr}(I) = 262.82 + 0.97252I + 743.27I^2 - 1563.5I^3 + 964.3I^4 \quad (7.13) \]

\[ S_b(V_{mr}) = \tanh(0.4V_{mr}) \quad (7.14) \]

\[ F_s = A_{mr}(I)S_b(V_{mr}) \quad (7.15) \]
\[
\begin{aligned}
\dot{x} &= -160x + F_s \\
F_{mr} &= 160x 
\end{aligned}
\] (7.16)

where \(V_{mr}\) is the velocity across the MR damper, \(x\) is the state variable of the filter, and \(F_{mr}\) represents the damping force.

As described in Chapter 3, the variables in Eqs. (7.13)-(7.16) are

\[
\begin{align*}
\alpha_0 &= 262.82, \quad \alpha_1 = 0.97252, \quad \alpha_2 = 743.27, \quad \alpha_3 = -1563.5, \quad \alpha_4 = 964.3; \\
\beta_0 &= 2.7183, \quad \beta_1 = 0, \quad \beta_2 = 0.4, \quad \text{and} \quad \beta_0 = 0; \\
\gamma_0 &= 160, \quad \gamma_1 = \gamma_2 = \gamma_3 = 0, \quad \gamma_4 = 1 \quad \text{and} \quad \gamma_{bias} = 0
\end{align*}
\]

Figure 7.13 Comparison between Experimental Data and Optimal MR Damper Model
The above analytical model was compared with the experimental data, and as Fig. 7.13 shows the actual data and model predictions agree well. Therefore, we proceeded with using the model for our adaptive control experimentation.

7.2.6 Simplification of the Seat Suspension Model

The adaptive control algorithm is implemented based on a reasonable system model. The dynamic model can be built through kinematics and dynamics. But the authentic model may be unnecessarily complicated and may not work well by mechanically combining all of model equations together due to ignoring their interactions. Thus the model simplification from a system point of view is important. Here the seat suspension is considered as one DOF system. The values of the air spring stiffness and the effective seat mass can be determined experimentally as done in Section 7.2.1, and they can be used to evaluate the on-line estimates. The optimal MR damper model can function as a force sensor and be coupled into the adaptive algorithm (Section 7.3).

7.2.6.1 Observations

At the initial experimental testing the kinematic equations were used to measure dynamic motions of the seat suspension. But they were too complicated for a practical implementation. It was necessary to simplify the relationship between the vertical seat relative velocity \( \Delta \dot{y} \) and the relative velocity \( \Delta \dot{c} \) (i.e., \( V_{mr} \)) across the MR damper. The seat suspension with an MR damper was vibrated using 1.5 Hz and 3.5 Hz under different currents of 0, 0.25, 0.50, 0.75, and 1.0 A. The MTS machine was set to 1.2 inch and 0.8 inch span for 1.5 Hz and 3.5 Hz respectively. The collected data of the relative displacement for both frequency cases were organized to determine the relationship between two relative velocities, referring to the next section. Also based on the measured acceleration \( \Delta \ddot{y} \) and the vertical relative displacement \( \Delta y \), the vertical damping force can be determined as follows

\[
F_{vdf} = -M\ddot{y}_1 - Ky_{12}
\]  

where \( F_{vdf} \) is the vertical dynamic damping force.
But according to Eq. (7.12), since the maximum of \( c \) is 6.30 inch and \( e \) remains unchanged as 13 inch shown in Section 7.2.3, we can mathematically derive the following relationship

\[
\left| \frac{F_{vdf}}{F_{mr}} \right| < \left| - \cos \beta \tan \theta + \frac{c}{e} \sin(\theta + \beta) / \cos(\theta) \right| \\
< \left| - \cos \beta \tan \theta + \frac{1}{2} \sin(\theta + \beta) / \cos(\theta) \right| \\
\leq \left| - \frac{1}{2} \cos \beta \tan \theta + \frac{1}{2} \sin \beta \right| 
\]  
(7.18)

where \( \beta \) and \( \theta \) are shown in Fig. 7.9, and \( \beta \in [0^\circ, 90^\circ] \).

Since from the physical seat suspension the maximum relative displacement is 8.75 inch in Section 7.2.3, according to Fig. 7.9 we can find out that

\[
\theta = \arcsin \left[ \frac{y_{12}}{e} \right] \leq \arcsin \left[ \frac{\text{max}(y_{12})}{e} \right] = \arcsin (8.75/13) = 42.3^\circ 
\]  
(7.19)

which means that

\[
\tanh(\theta) \in [0, 1) 
\]  
(7.20)

because \( \theta \) is less than 45°. Since \( \theta \) and \( \beta \) satisfy Eqs. (7.19) and (7.20), Eq. (7.18) can become as

\[
\left| \frac{F_{vdf}}{F_{mr}} \right| < \frac{1}{2} \left| - \cos \beta \tanh \theta \right| + \left| \sin \beta \right| < 1 
\]  
(7.21)

i.e.,

\[
\left| \frac{F_{vdf}}{F_{mr}} \right| < 1 
\]  
(7.22)
Figure 7.14 Comparison of Predicted Values and Measurements about MR Damper Velocity and Damping Force for 1.5 Hz Excitation; (a) 0 Ampere Damper Current; (b) 0.25 Ampere Damper Current
Figure 7.15 Comparison of Predicted Values and Measurements about MR Damper Velocity and Damping Force for 1.5 Hz Excitation; (a) 0.51 Ampere Damper Current; (b) 0.75 Ampere Damper Current
Figure 7.16 Comparison of Predicted Values and Measurements about MR Damper Velocity and Damping Force; (a) 1.02 Ampere Damper Current for 1.5 Hz Excitation; (b) 0 Ampere Damper Current for 3.5 Hz Excitation
Figure 7.17 Comparison of Predicted Values and Measurements about MR Damper Velocity and Damping Force for 3.5 Hz Excitation; (a) 0.25 Ampere Damper Current; (b) 0.51 Ampere Damper Current
Figure 7.18 Comparison of Predicted Values and Measurements about MR Damper Velocity and Damping Force for 3.5 Hz Excitation; (a) 0.75 Ampere Damper Current; (b) 1.02 Ampere Damper Current
Equation (7.22) indicates that the MR damping force is always scaled down in the seat suspension. But from Figs. 7.14b-7.18b, it can be observed that the damping force $F_{mr}$ predicted from the MR damper model is always less than the vertical damping force $F_{vdf}$ that is calculated per Eq. (7.17) assuming mass and stiffness are known from Section 7.2.1. If the MR damper force were scaled down per Eq. (7.22), then the scaled force would be much less than the vertical damping force. Thus the damper model determined by using data from MTS machine could not be appropriately applied to the dynamic systems. Therefore, in order to implement the adaptive algorithm, the model had to be modified to reasonably represent the dynamics as will be shown in the following section.

7.2.6.2 Simplification of Seat Suspension Model

The model of the seat suspension is composed of three parts: kinematics equations, dynamic equations, and an MR damper model. The kinematics relationship is used to calculate the seat relative displacement (velocity) and the relative displacement (velocity) across the MR damper. The relative velocity can be obtained by differentiating the corresponding relative displacement. The geometry of the seat suspension is shown in Fig. 7.9. The clue here is that in order to use the MR damper model to predict the damping force it is necessary to determine the relative velocity across the MR damper. The procedure was first to calculate the relative displacement across the MR damper

$$\Delta c = c - c_e$$  \hspace{1cm} (7.23)

Then we used the rate filter to estimate the relative velocity $v_{mr}$ across the MR damper. In order to simplify the procedure described in Section 7.2.3, we assumed that there existed a linear relationship between $\Delta c$ and $y_{12}$ (as shown in Fig. 7.10) as follows

$$[\Delta c] = \rho [y_{12}]$$  \hspace{1cm} (7.24)

where $[\Delta c]$ and $[y_{12}]$ are nx1 vectors composed of the corresponding measured relative displacements according to Section 7.2.3, and $\rho$ is a scale factor. The least squares method could be applied to estimate $\rho$ as follows
\[ \rho = ([y_{12}]^T [y_{12}])^{-1} [y_{12}]^T [\Delta c] \]  

(7.25)

where \( T \) represents the matrix transpose. By solving Eq. (7.25), we have

\[ \rho = 0.13 \]  

(7.26)

If Eq. (7.24) holds, then we can say the velocities derived from the corresponding displacements also have the linear relationship as follows

\[ v_{mr} = 0.13 \dot{y}_{12} = \Delta \dot{c} \]  

(7.27)

where \( v_{mr} \) is the relative velocity across the MR damper, and \( \dot{y}_{12} \) is the velocity of the seat suspension.

It was determined that the linear relationship was sufficient as shown in Figs. 7.14a-7.18a. From the relative velocity curves of Figs. 7.14a-7.18a, it can be observed that there is a hysteretic relationship between the dynamic MR damper velocity and the velocity predicted by Eq. (7.23). Thus in order to better describe the velocity relationship including the effect from the suspension dynamics, it is necessary to include a first order transfer function

\[ T_v(s) = \frac{b}{s + a} \]  

(7.28)

where \( a \) and \( b \) are constants.

A more interesting observation is that the dynamic vertical damping force is different from the damping force predicted by the MR damper model of Eqs. (7.13) to (7.16) without geometric consideration, as shown in Figs. 7.14b-7.18b. The predicted force
level is lower than the corresponding vertical damping force $F_{vdf}$. The following equation is introduced into the suspension model for the force prediction modification

$$F_{vdf} = (f_1 + f_2 I)F_{mr}$$

(7.29)

where $f_1$ and $f_2$ are constants, $F_{mr}$ is the damping force predicted by the MR damper model. The process for obtaining the damper force applied to the seat from the relative displacement across the seat suspension is shown in Fig. 7.19.

As mentioned at the beginning of this section, the collected data is also used to optimize $T_v(s)$ and $(f_1 + f_2 I)$. The results are

$$a = 58.635$$

and

$$b = 200;$$

(7.30a)

$$f_1 = 0.338$$

and

$$f_2 = 0.959$$

(7.30b)

The comparison of the damping forces is shown in Fig. 7.20, which plots out both optimization results and experimental data. Figure 7.20 clearly shows that it is necessary to include the structural interaction functions $T_v(s)$ and $(f_1 + f_2 I)$ to correctly build a system model when the MR damper works dynamically in a structural system.
(a) Comparison of the Vertical Damping Force between Experimental Results and Suspension Model under 1.5 Hz with Different Currents

(b) Comparison of the Vertical Damping Force between Experimental Results and Suspension Model under 3.5 Hz with Different Currents

Figure 7.20 Evaluation of the Model from Vertical Relative Displacement to Vertical Damping Force in the Seat Suspension
Furthermore, the system model shown in Fig. 7.19 can be simplified as one effective unified MR damper model. Since the MR damper model has amplitude function and hysteresis equation, it is reasonable to use one model to include $T_v(s)$ and $(f_1+f_2I)$. Such simplification can facilitate the real-time implementation of adaptive control.

In our experiments, the seat suspension model from the relative displacement to the vertical damping force is shown in Fig. 7.21. Later, the semiactive adaptive algorithm was designed by using the simplified model as shown Eqs. (7.31)-(7.34).

Again, the seat damper model can be optimized by using the MATLAB function CONSTR. Both model results and the experimental data are shown in Fig. 7.22. The optimal simplified MR damper model is

$$A_{mr}(I) = 277.38 + 1671.3 I + 683.12 I^2 - 1443.9 I^3 + 491.53 I^4$$  \(7.31\)

$$S_b(\dot{y}_{12}) = \tanh(0.64\dot{y}_{12})$$  \(7.32\)

$$F_s = A_{mr}(I)S_b(\dot{y}_{12})$$  \(7.33\)

$$\begin{cases} \dot{x} = -(-37.4I + 94)x + F_s \\ F_{mr} = (-37.4I + 94)x \end{cases}$$  \(7.34\)
(a) Comparison of the Vertical Damping Force between Experimental Results and Modified MR Damper Model under 1.5 Hz with Different Currents

(b) Comparison of the Vertical Damping Force between Experimental Results and Modified MR Damper Model under 3.5 Hz with Different Currents

Figure 7.22 Evaluation of the Modified MR Damper Model by Using the Vertical Damping Force for the Simplified Seat Suspension
The comparison of the simplified model with the experimental data is shown in Fig. 7.22. It shows that the simplified model can capture the dynamics of the vehicle seat suspension well. Thus, this model will be used to implement the adaptive control algorithm.

7.3 Simplified Adaptive Algorithm

Based on the simplified model of Eqs. (7.31)-(7.34), the adaptive algorithm was programmed in Simulink and finally implemented in the dSPACE AutoBox for the hardware-in-the-loop testing. The implementation of adaptive control was based on the model derived from the previous section. The objective function was to minimize the seat acceleration, i.e., \( p=1 \) and \( q=0 \) are adopted for Eq. (5.9), through the adaptation of the current \( I \) to the MR damper according to Eq. (5.10).

Furthermore the update gradient for the adaptive algorithm can be solved according to

\[
\frac{\partial F_{mv}(k)}{\partial I(k)} = \tanh(0.64\dot{y}_{12}) \frac{\partial A_{mv}(I)}{\partial I(k)} + A_{mv}(I) \frac{\partial \tanh(0.64\dot{y}_{12})}{\partial I(k)}
\]

\[
M \frac{\partial \ddot{y}(k)}{\partial I(k)} + \frac{\partial F_{mv}(k)}{\partial I(k)} + K \frac{\partial y_{12}(k)}{\partial I(k)} = 0 \tag{7.35}
\]

\[
\frac{\partial J}{\partial I(k)} = 2\ddot{y}(k) \frac{\partial \ddot{y}(k)}{\partial I(k)}
\]

It should be noted that the hysteretic equation is ignored in the above equations, because the parametric study through simulation shows that it is feasible to ignore the hysteresis equation, referring to Section 6.3.3. The experimental results also demonstrate that such simplification is justified.

In order to apply the recursive least square method to estimate the unknown parameters, a filter \( G(s) \) used to reject DC components in the measured signals for Eq. (5.8)

\[
G(s) = \frac{s}{s + 0.2\pi} \tag{7.36}
\]
Then the recursive least mean square algorithm of Eqs. (5.5)-(5.7) is used for the parameter estimation.

Finally, the complete adaptive control algorithm is programmed in Simulink and then downloaded into the dSPACE AutoBox. The excitation signal is also loaded to Autobox and inputed to the MTS controller to command the hydraulic actuator to produce the expected excitations for the tested seat suspension. The Simulink program of adaptive control algorithm is shown in Fig. 7.23.
Figure 7.23 Implementation of Adaptive Control on the Vehicle Seat Suspension
7.4 Experimental Results

In order to demonstrate the adaptive control, different excitation signals are employed to test the adaptive seat suspension. For comparison purpose, soft (0 A) and hard (1 A) damping cases are also tested with the seat suspension.

Two cases were tested for adaptive control. One case assumed that mass and stiffness are known. The other was to test a black-box dynamic system, in which the system identification had to be used to estimate the unknown mass and stiffness. All of the testing results are presented from Figs. 7.24 to 7.26. Figure 7.27 shows that the root-mean-square (RMS) values of the seat accelerations vary with the current to the MR damper.

(a) Comparison of Accelerations with Sine Excitation

Figure 7.24 Testing Results with 1.5 and 3.5 Hz Sine Excitation
(b) Comparison of Gradient and Current with Sine Excitation

(c) Comparison of ID Values and Assumed True Values with Sine Excitation

Figure 7.24 Testing Results with 1.5 and 3.5 Hz Sine Excitation
Figure 7.25 Testing Results with Sweeping Excitation

(a) Comparison of Accelerations with Sweeping Excitation

(b) Comparison of Gradient and Currents with Sweeping Excitation
(c) Comparison of ID Values and Assumed True Values with Sweeping Excitation

Figure 7.25 Testing Results with Sweeping Excitation

(a) Comparison of Accelerations with Random Excitation

Figure 7.26 Testing Results with ISO Random Excitation
(b) Comparison of Gradient and Currents with Random Excitation

(c) Comparison of ID Values and Assumed True Values with Random Excitation

Figure 7.26 Testing Results with ISO Random Excitation
7.5 Observations

From Figs. 7.24a-26a, it can be observed that the soft and hard damping can minimize the seat acceleration for either lower or higher frequency range, but the adaptive control can maintain an optimal control in both frequency ranges to minimize the vibration. The adaptive control can work well under different frequency excitations.

Figures 7.24b-26b shows the control currents for different cases. The current from the adaptive algorithm is high for the lower excitation frequency range and becomes almost 0 A when the excitation frequency goes higher. The update gradient of the adaptive control is also presented and it shows that the update gradient converges to the values that minimize the seat acceleration. Furthermore, Fig. 7.27 shows the acceleration RMS value decreases for the low frequency excitation such as 1.5-Hz pure tone and ISO1 if the current increases, while if the system is exposed to high frequency excitations (for
example, 3.5-Hz pure tone and ISO4), then the acceleration RMS value becomes smaller with a smaller current applied to the MR damper. Such trend matches the gradient convergence per different situations as explained before.

In Figs. 7.24c-26c, it can be observed that the on-line system identification results match the predetermined true values. Furthermore it says that the modified MR damper can reflect the vertical damping force. The reasonable identification values guarantee that the adaptive control functions as expected. In the lower frequency range, the current increases to minimize the seat acceleration while the current is zero if the excitation frequency is higher than the seat suspension isolation frequency of about 2 Hz, as shown in Figs. 7.24b-26b and 7.27.

7.6 Summary
The effect from the seat suspension dynamics on the magneto-rheological damper is observed. The MR damper model has to include such effect. The modified model can be used as a force sensor to predict the dynamic damping force for the adaptive algorithm.

The implemented adaptive control shows reasonable results based on the simplified seat suspension model. The adaptive algorithm can also be simplified to reduce the DSP calculation. It can track the system variations and maintain an optimal control regardless of some unmodelled dynamics and noises. Further research shall be performed on the adaptation of skyhook control policy and its variations in order to obtain the advantages from both approaches.