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Abstract

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Integrated active and semi-active control for seat suspension of a heavy duty vehicle

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Abstract

In this paper, an integrated active and semi-active seat suspension for heavy duty vehicles is proposed, and its prototype is built; an integrated control algorithm applied measurable variables (suspension relative displacement and seat acceleration) is designed for the proposed seat prototype. In this seat prototype, an active actuator with low maximum force output (70 N), which is insufficient for an active seat suspension to control the resonance vibration, is applied together with a rotary magnetorheological (MR) damper. The MR damper can suppress the high vibration energy in resonance frequency, and then a small active force can further improve the seat suspension performance greatly. The suspension's dynamic property is tested with a MTS system, and its model is identified based on the testing data. A modified on-off controller is applied for the rotary MR damper. A H_∞ controller with the compensation of a disturbance observer is used for the active actuator. Considering the energy saving, the control strategy is designed as that only when the MR damper is in the off state (0 A current), the active actuator will have active force output; or the active actuator is off. Both simulation and experiment are implemented to verify the proposed seat suspension and controller. In the sinusoidal excitations experiment, the acceleration transmissibility of integrated control seat has lowest value in resonance frequency and frequencies above the resonance, when compared with power on (0.7 A current), power off (0 A current) and semi-active control seat. In the random vibration experiment, the root mean square (RMS) acceleration of integrated control seat suspension has 47.7%, 33.1% and 26.5% reductions when compared with above-mentioned three kinds of seat suspension. The power spectral density (PSD) comparison indicates that the integrated

seat suspension will have good performance in practical application. The integrated active and semi-active seat suspension can fill energy consumption gap between active and semi-active control seat suspension.

Keywords: active control; semi-active control; seat suspension; vibration control.

1. Introduction

Seat suspension plays an important role in improving drivers' ride comfort and protecting drivers' health from the vibration caused by rough road (Metered and Šika, 2014; Paddan and Griffin, 2002). Especially, heavy duty vehicles work in severe condition where the vibration magnitude is always higher than normal passenger vehicles. Much research is being done on the heavy duty vehicle seat suspension. Generally, the highest vibration magnitude of heavy duty vehicle seat is in the frequency range 2 to 4 Hz, and heavy vehicle drivers usually experience vibration around 3 Hz which increases fatigue and drowsiness (Mabbott et al., 2001). The resonance frequency of seat suspension, which is depended on the spring stiffness and the loaded mass, is often approximate 2 Hz. The vibration is amplified at resonance frequency, and attenuated at frequencies above resonance; the amplification and attenuation are controlled by a damper. The semi-active seat suspension has been widely studied with the electrorheological (ER) fluid damper and the MR fluid (MRF) damper (Choi and Han, 2007; Choi et al., 2000; Hiemenz et al., 2008; Zhu et al., 2012). Because the semi-active dampers can change damping in real time, large damping is applied at resonance frequency and small damping is applied in higher frequency. The vibration magnitude at resonance can be easily suppressed with semi-active damper. There is no doubt that the active seat suspension has better performance in terms of ride comfort. Several kinds of active actuators has be applied for active seat suspensions; e.g. a hydraulic absorber and a controlled air-spring (Maciejewski et al., 2010), electric servomotor with a ball screw mechanism (Kawana and Shimogo, 1998), electromagnetic linear actuators (Gan et al., 2015) and rotary motors (Ning et al., 2016).

In recent years, some research into active and semi-active seat suspension control has been conducted; e.g., H_∞ control (Zhao et al., 2010), linear quadratic gauss (LQG) (Türkay and Akçay, 2008; He and McPhee, 2005), and adaptive control (Darling et al., 2015). A new adaptive fuzzy sliding mode controller is proposed for MR damper (Choi et al., 2015). An active vibration control strategy is proposed based on a primary controller and actuator's reverse dynamics (Maciejewski, 2012). In practical application, the relative seat suspension displacement and acceleration can be easily measured. When the controller of a seat suspension is designed, the measurable variables should be considered as feedbacks.

The active and semi-active seat suspensions have their own advantages and disadvantages. There is no doubt that the active seat suspension has better performance in vibration control. However, the cost and energy consumption, which are closely related to the active actuator's maximum continuous force output, are much higher. In authors' previous study (Ning et al., 2016; Sun et al., 2016), large power output of the actuator is required when the vibration is around the resonance frequency of the seat suspension. Generally, a 150 to 200 N maximum force output is required by active control. By contrast, the semi-active seat suspension has its best performance around resonance frequency with small energy consumption. In addition, the semi-active suspension is a fail-safe system, because it can only passively generate force. On the other hand, the effectiveness frequency of semi-active seat suspension control is around resonance frequency; when vibration frequency is high than resonance frequency, a small damping is desired which means no current input to the MR damper all the time.

In this paper, an integrated active and semi-active seat suspension for heavy duty vehicles is investigated. A low continuous force output actuator, which is insufficient to work as an active actuator, is applied together with a semi-active actuator. The integrated active and semi-active seat suspension has many advantages, which will fill performance gap between active and semi-active seat suspension. Firstly, the energy consumption will be lower than barely active seat suspension. The low force output actuator means low power consumption. Secondly, the benefits of active and semi-active seat suspension will be integrated. The large acceleration around resonance frequency will be controlled by semi-active actuator with small energy consumption, and in the higher frequency, the

active seat suspension can perform better than a soft passive seat suspension with relatively small force. Thirdly, it becomes a fail-safe system because of the semi-active actuator. Additionally, the low power active actuator will have small size and small current requirement which have benefits to the direct installation and application in a semi-active seat suspension. The proposed integrated controller only applies measurable variable in practical application. Considering the energy saving, the semi-active controller is the main controller; the active actuator only works as the MR damper is at off state.

The remainder of the paper is organized as follows: Section 2 presents the proposed integrated seat suspension prototype; Section 3 discusses the prototype model; Section 4 proposes an integrated controller; the simulation and experimental results are presented in Section 5; Finally, Section 6 presents the conclusions of this research.

Notation: I is used to denote the identity matrix of appropriate dimensions. The symbol * indicates the transposed element in the symmetric position of a matrix.

2. Integrated active and semi-active seat suspension prototype

An integrated active and semi-active seat suspension prototype is designed and built (see Figure 1). This prototype is a modification of a normal commercial vehicle seat (GARPEN GSSC7) where the original damper has been removed. An active rotary motor (Panasonic MSMJ042G1U) and a semi-active rotary MR damper are installed on scissors structures centre of the suspension's two sides, respectively. The rated torque output of the motor is 1.3 Nm and its power is 400 W. Gear reducers, with ratio 20:1 and 8:1, are applied to amplify torque output of the active motor and semi-active MR damper. The active rotary motor with its gear reducer is the active actuator; similarly, the rotary MR damper with its gear reducer is the semi-active actuator.

Figure 1. The integrated active and semi-active seat prototype.

Figure 2 shows the structural design of the applied rotary MR damper. This rotary MR damper mainly consists of a cylinder, a coil, a shaft, a rotor and the enclosed reservoir that is filled with MRF. Unlike the linear MR damper, there is no spring or gas chamber to provide the initial pressure in the rotary MR damper. For the consideration of magnetic circuit design, the rotor is made of low-carbon steel and the shaft is made of aluminium. When the coil, which is composed by winding the copper wire around the rotor, is energized, the magnetic circuit is formed as shown in Figure 2. This design can guarantee that the magnetic flux passes through the MRF.

Figure 2. Rotary MR damper schematic.

3. The seat suspension prototype test and model identification

3.1 Testing method

The dynamic properties of the seat suspension system were tested and evaluated with a MTS machine (Load Frame Model: 370.02, MST Systems Corporation) as shown in Figure 3. The seat suspension is

fixed between the upper and lower grippers of the MTS machine. When the testing started, the upper gripper will force the suspension to move in accordance with a preprogramed sinusoidal routine. The response force of the seat suspension can be measured and saved with the movement routine together.

Figure 3. Test system.

3.2 Test results

The field, amplitude, and frequency-dependent performance of the seat suspension are evaluated with above described test setup. For field-dependent test, different currents were chosen to energize the magnetic field with same sinusoidal movement routine (with 10 mm amplitude and 1 Hz frequency). Figure 4 shows the force-displacement relationships of the seat suspension with different currents. As area of the enclosed force-displacement loops can indicate the system damping, it can be seen that the damping of this seat suspension can be controlled by the applied current.

Figure 4. Test results with different currents.

The amplitude-dependant performance of the seat suspension is shown in Figure 5, when the seat suspension was loaded with sinusoidal signals with different amplitudes (3 mm, 5 mm and 10 mm) at constant frequency (1 Hz) and current (0.5 A). The response force of seat suspension is nearly equal in 0 mm where the spring force of seat suspension is 0 N. It indicates that with the same current, the saturation force of the MR damper is equal in different velocity. Similarly, the effects of changing frequencies are presented in Figure 6 where the amplitude and current are constants with 10 mm and 0.5 A, respectively. It can be seen that the maximum force has slight variations with different frequencies.

Figure 5. Test results with different amplitudes.

Figure 6. Test results with different frequencies.

The above experiments show that the semi-active actuator of the integrated seat suspension is controllable and has stable properties in different velocities. On the other hand, the torque output of the active actuator can be accurately controlled by the motor drive. As the transformation of active actuator from torque output to vertical force output can be easily implemented (Ning et al., 2016), the whole seat suspension system is controllable.

3.3 Model identification

As two gear reducers are applied in the system, the friction force in the system cannot be ignored. The seat suspension model is shown in Figure 7 where z_s and z_v are the displacement of upper and lower platform of the seat suspension, k_s is the spring stiffness, f_r is the inner friction. The Bouc-Wen model is applied to describe the MR damper. Considering that the friction has similar hysteresis phenomenon as MR damper, the inner friction is modelled into MR damper force together.

Figure 7. Seat suspension model

The seat suspension model is defined as:

$$x_d = z_s - z_v \quad (1)$$

$$F_t = k_s x_d + f_r + F_{mr} \quad (2)$$

$$F_d = f_r + F_{mr} = c_1 \dot{y}_d \quad (3)$$

$$\dot{y}_d = \frac{1}{c_0 + c_1} [\alpha z_d + c_0 \dot{x}_d + k_0 (x_d - y_d)] \quad (4)$$

$$\dot{z}_d = -\gamma_d |\dot{x}_d - \dot{y}_d| |z_d| |z_d|^{n-1} - \beta_d (\dot{x}_d - \dot{y}_d) |z_d|^n + A_d (\dot{x}_d - \dot{y}_d) \quad (5)$$

$$\alpha = \alpha_1 + \alpha_2 I + \alpha_3 I^2 \quad (6)$$

$$c_0 = c_{01} + c_{02} I + c_{03} I^2 \quad (7)$$

$$c_1 = c_{11} + c_{12} I + c_{13} I^2 \quad (8)$$

where I is the current applied to the MR damper. The parameters $\gamma_d, \beta_d, A_d, n, k_0, \alpha_1, \alpha_2, \alpha_3, c_{01}, c_{02}, c_{03}, c_{11}, c_{12}$ and c_{13} are used to characterize the MR damper (Yang et al., 2013; Du et al., 2013). The optimized values of those parameters are determined by fitting the model with experimental data using MATLAB parameter estimation toolbox. Table 1 shows the identified model parameters. The fitting result in Figure 8 shows the simulation data matches the experimental data very well.

Table 1. The identified model parameters

γ_d	7.263e+05		α_3	-1.005e+05
β_d	7.359e+05		c_{01}	95.873
A_d	1.860		c_{02}	9.428e+02
n	2		c_{03}	-5.249e+02
k_s	4600		c_{11}	1.338e+04
k_0	3.519e+02		c_{12}	-5.981e+04
α_1	7.553e+04		c_{13}	3.078e+05
α_2	1.972e+05

Figure 8. The fitting result of model

4. Control algorithm

This section proposes a control algorithm for the integrated active and semi-active seat suspension. This algorithm is based on a modified semi-active on-off control method with the compensation of small active force on the off state of the MR damper. The relative displacement and seat acceleration, which can be easily obtained in practical application, are used as control feedback.

4.1 Integrated seat suspension model

The integrated seat model is shown in Figure 9 where m is the mass loaded on the suspension, F_d includes inner friction and variable damping force, and u is the output of active force. The dynamic model of the system can be described as:

$$m\ddot{z}_s = -k_s(z_s - z_v) - F_d + u \quad (9)$$

Figure 9. Integrated active and semi-active seat model

In the practical application, the suspension relative displacement and seat acceleration can be measured; the suspension relative velocity can be obtained from the differential of relative displacement. Therefore, the state variables are chosen as $\mathbf{X} = [z_s - z_v \quad \dot{z}_s - \dot{z}_v]^T$, the vibration disturbance is $d = \ddot{z}_v$, and the measurement variables are $\mathbf{Y}_1 = [z_s - z_v \quad \dot{z}_s - \dot{z}_v]^T$ and $Y_2 = \ddot{z}_s$. Thus, combining with Eq. (9) the system model is defined as:

$$\dot{\mathbf{X}} = \mathbf{A}\mathbf{X} + \mathbf{B}_1(u - F_d) + \mathbf{B}_2d \quad (10)$$

$$\mathbf{Y}_1 = \mathbf{C}_1\mathbf{X} \quad (11)$$

$$Y_2 = \mathbf{C}_2\mathbf{X} + D_2(u - F_d) \quad (12)$$

where $\mathbf{A} = \begin{bmatrix} 0 & 1 \\ -\frac{k_s}{m} & 0 \end{bmatrix}$, $\mathbf{B}_1 = \begin{bmatrix} 0 \\ \frac{1}{m} \end{bmatrix}$, $\mathbf{B}_2 = \begin{bmatrix} 0 \\ -1 \end{bmatrix}$, $\mathbf{C}_1 = \begin{bmatrix} 1 & 0 \\ 0 & 1 \end{bmatrix}$, $\mathbf{C}_2 = \begin{bmatrix} -\frac{k_s}{m} & 0 \end{bmatrix}$, $D_2 = \frac{1}{m}$.

The seat acceleration is the main optimization objective in the controller design. Therefore, the controlled output is defined as:

$$Z_1 = \mathbf{C}_3 \mathbf{X} + D_3(u - F_d) \quad (13)$$

where $\mathbf{C}_3 = \alpha \mathbf{C}_2$, $D_3 = \alpha D_2$, α is a constant.

4.2 Controller design

Firstly, the controllers for the two actuators are designed, respectively.

Controller 1:

The modified on-off control method is a very efficient way for the semi-active actuator. A modified on-off controller for the MR damper is designed as $u_I = \text{sat}(I)$, where $\text{sat}(I)$ is a saturation function defined as:

$$I = k_i \ddot{z}_s (\dot{z}_s - \dot{z}_v) \quad (14)$$

$$\text{sat}(I) = \begin{cases} I_{max}, & I > I_{max} \\ I, & I_{max} > I > 0 \\ 0, & \text{Others} \end{cases} \quad (15)$$

where k_i is positive constant, I_{max} is the maximum input current for MR damper. This controller is very intuitive. When the direction of the generated force of the MR damper is opposite to the inertia force $M\ddot{z}_s$, it can impede the increase of the acceleration, thus a large damping is required by loading current into the MR damper; otherwise it will keep its smallest state (smallest damping). For instant, if the direction of $\ddot{z}_s$ and $\dot{z}_s - \dot{z}_v$ are both positive, thus the force of the MR damper is opposite with the inertia force. In this case, the acceleration can be reduced when the force of the MR damper is increased by loading current. On the contrary, when $\ddot{z}_s$ and $\dot{z}_s - \dot{z}_v$ have different directions, there should have no current input applied to the MR damper to keep the MR damper to be soft.

In the practical application, the measurement noise of sensors is inevitable. This modified on-off controller with saturation can attenuate the frequent variation of MR damper between maximum current and no current caused by measurement noise.

Controller 2:

The inner friction cannot be measured; and the force output of MR damper, which varies based on suspension relative velocity and the output of Controller 1, is also unmeasured. A disturbance observer is applied to estimate the unmeasurable force F_d .

$$\dot{\hat{F}}_d = L[Y_2 - (\mathbf{C}_2\mathbf{X} + D_2(u - \hat{F}_d))] \quad (16)$$

where the observer gain L is a constant to be designed.

A H_∞ controller with disturbance compensation is constructed as:

$$u = \mathbf{K}\mathbf{X} + \hat{F}_d \quad (17)$$

where $\mathbf{K}$ is the state feedback gain to be designed.

The disturbance observer gain L and H_∞ controller gain $\mathbf{K}$ can be obtained by solving following linear matrix inequality (LMI) with Matlab LMI toolbox:

$$\begin{bmatrix} \mathbf{A}\mathbf{Q} + \mathbf{B}_1\mathbf{R} + (\mathbf{A}\mathbf{Q} + \mathbf{B}_1\mathbf{R})^T & * & * & * \\ -\mathbf{B}_1^T & (\mathbf{G}D_2)^T + \mathbf{G}D_2 & * & * \\ \mathbf{B}_2^T & 0 & -\lambda^2 & * \\ \mathbf{C}_3\mathbf{Q} + \mathbf{D}_3\mathbf{R} & -D_3 & 0 & -\mathbf{I} \end{bmatrix} < 0 \quad (18)$$

where $\mathbf{Q} = \mathbf{Q}^T$, $\mathbf{Q} > 0$, and λ is a given performance index. After solving Eq. 18 for matrices $\mathbf{Q}$, $\mathbf{R}$ and $\mathbf{G}$, the controller gain is obtained as $\mathbf{K} = \mathbf{R}\mathbf{Q}^{-1}$, and the observer gain is $L = \mathbf{P}_2^{-1}\mathbf{G}$. The computation procedure is similar as that used in (Ning et al., 2016) and omitted here for brevity. In this study, L is chosen as -2000; $\mathbf{K}$ is chosen as $[3000 \quad -20]$.

Secondly, the two controllers are integrated together. Apparently, the MR damper can only do negative work which means the direction of the force generated by the MR damper should be opposite to the direction of suspension relative movement. When the negative work can dissipate the vibration energy and thus suppress the seat vibration, the output of Controller 1 is not equal to zero. On the contrary, when the seat suspension needs positive work to reduce seat vibration, the MR damper should keep its softest state to do least negative work due to its semi-active control characteristic. In the proposed integrated seat suspension, when the MR damper keeps its softest state, the output of

Controller 1 is equal to zero, but the motor will do positive work to reduce seat acceleration. Figure 10 shows the control strategy. The Controller 1 is the main controller which controls the current for rotary MR damper. The Controller 2's output will send to the active actuator when Controller 1's output is equal to zero. This control strategy can save the active actuator's energy consumption.

Figure 10. Integrated controller.

5. Evaluation

5.1 Numerical simulation

In this test, the performance of the integrated active and semi-active controller applied to the seat suspension model in section 3 is evaluated. The cushion model is included in the simulation with a stiffness 90000 N/m and damping 2000 Ns/m. The suspension and driver body mass are 28 Kg and 70 Kg, respectively. The saturated active force is chosen as 70 N which is insufficient to control the resonance vibration independently. The saturated semi-active control current is chosen as 0.7 A, and $k_i = 100$ is applied.

The harmonic excitation test, which was a sweep frequency signal from 1 Hz to 3 Hz in 20 seconds with 30 mm amplitude, was firstly implemented. Figure 11 shows the seat absolute displacements of semi-active control compared with seat suspension with different current (0 A, 0.1 A and 0.7 A). The soft suspension (0 A current) has resonance around the 5 second and performs best in higher frequency. When 0.1 A current is loaded, the resonance amplitude has been greatly suppressed.

Although the hard suspension (0.7 A current) has best performance around resonance frequency, it works worst in high frequency. The semi-active control suspension can successfully suppress the resonance vibration. At the same time, it performs close to a soft suspension (0 A current) at high frequency. This result indicates that the semi-active control can greatly improve the seat suspension performance. Figure 12 shows the result comparison of semi-active and integrated control. The seat displacement of integrated control is further reduced in the whole test time range. Especially around the resonance frequency, the MR damper can suppress the resonance vibration, and then a small active force can further isolate the vibration.

Figure 11. Semi-active control result comparison.

Figure 12. Integrated control result comparison.

The random vibration test was implemented. Table 2 shows the RMS acceleration comparison, where the integrated control has the lowest value, and has a 15.5% reduction compared with semi-active control. The time domain and frequency domain acceleration in Figure 13 further validates the control algorithm. The integrated control has lowest PDS value around resonance frequency, and has low value in higher frequencies vibration.

Table 2. RMS acceleration of random vibration (m/s^2)

Integrated	Semi-active	Power on	Power off
1.09	1.29	1.40	1.46

Figure 13. Acceleration with random vibration

5.2 Experimental setup

The experimental setup is shown in Figure 14. One six degrees of freedom (6-DOF) vibration platform was applied to generate the vertical vibration excitation based on preprogramed routine. The seat suspension was fixed on the upper platform of the 6-DOF vibration platform. A NI CompactRio 9074 was applied to acquire data and implement designed controller; the control frequency is 500 Hz. Two accelerometers (ADXL203EB) were used to measure the vertical seat acceleration and excitation acceleration, respectively. The suspension relative displacement was measured by a laser displacement sensor (Micro Epsilon ILD1302-100). The desired current signal is amplified by a power amplifier and then used to control the rotary MR damper. The active actuator applied in this prototype can output a maximum force of 150 N. For verify the proposed controller, the active force was limited as ± 70 N.

There are three kinds of experiments. The passive experiment tests the seat suspension's passive response when the MR damper is loaded with different current (0 A, 0.1 A and 0.7 A). In the semi-active control experiment, the semi-active controller is applied. In the integrated control experiment, the proposed integrated controller is implemented. It should be emphasised that the active actuator only work when the MR damper is in off state (0 A current). Disturbance may be introduced because the active actuator is turned on and off suddenly. For example, when MR damper turns off, and a 50 N active force is required, the motor output needs to change from 0 N to 50 N. So, a one-order low pass point by point filter module in Libview (a software for developing program in NI CompactRio 9074) was applied to the active control output to avoid its sudden change.

Figure 14. Experimental setup

5.3 Experimental results

The sinusoidal excitations were applied to the seat suspensions with 75 Kg load. **Figure 15 shows the seat acceleration comparison in both time and frequency domain with resonance frequency (1.6 Hz).** When the MR damper is power off (0 A), the seat has maximum peak acceleration. The integrated control has the best performance. When the vibration frequency is higher than resonance frequency range, a soft seat suspension should perform better than a hard suspension. Figure 16 shows the seat acceleration with 2.4 Hz vibration when the soft seat suspension (0 A) can successfully isolate vibration. When the MR damper is loaded with 0.1 A current, the peak acceleration will greatly increase. The performance of semi-active control is between when the MR damper has 0 A and 0.1 A current. The integrated control can greatly reduce the acceleration; its acceleration is even lower than the 0 A current case.

(a)

(b)

Figure 15. Seat acceleration at 1.6 Hz vibration. (a) Time domain; (b) Frequency domain.

(a)

(b)

Figure 16. Seat acceleration at 2.4 Hz vibration. (a) Time domain; (b) Frequency domain.

The transmissibility of seat suspension is shown in Figure 17. When the MR damper is powered with 0.7 A current, the vibration transmissibility is around 1.05 in the whole testing frequency range. On the contrary, when the MR damper is power off (0 A), the vibration transmissibility is highest in 1.6 Hz, and then going down. In other words, the soft seat suspension has resonance around 1.6 Hz, and starts to isolate vibration when the vibration frequency is above resonance frequency. The semi-active control seat suspension can suppress the resonance vibration and even performs better than a hard (0.7 A) suspension in 1.6 Hz. The integrated control has lowest transmissibility in 1.6 Hz, and performs better than the soft seat suspension (0 A) in higher frequency. The experiments result verified the seat suspension's design idea and validated the proposed control algorithm. When the vibration is around resonance frequency, the semi-active control MR damper can dissipate much vibration energy, and then a small active force, which is insufficient to control vibration independently, can successfully further isolate the vibration to a low magnitude. In the higher frequency vibration, the required active force is relative small, so the applied active force can reduce the vibration transmissibility to values lower than soft seat suspension (0 A). Generally, the semi-active control is hard to perform as good as a passive soft seat suspension in practical application.

Figure 17. Transmissibility

The random vibration is often used to evaluate seat suspension performance in the time domain. Figure 18 shows the seat acceleration with random vibration. When the MR damper is power off (0 A), there is resonance around the 8 second. On the other hand, when the MR damper is power on (0.7 A), its acceleration is obviously larger than other cases except in the resonance range. Table 3 shows the RMS acceleration comparison. The acceleration of integrated control seat suspension has 47.7% and 33.1% reductions when compared with power on and power off seat suspension. Furthermore, there is 26.5% reduction of RMS acceleration when compared with semi-active control seat suspension. Figure 19 shows the PSD value of acceleration, which further indicates that the integrated seat suspension with small active force can improve the performance of a semi-active control seat suspension greatly.

Figure 18. Seat acceleration with random vibration

Table 3. RMS acceleration of random vibration (m/s^2)

Power off	Power on	Semi-active	Integrated
1.3054	1.0195	0.9296	0.6825

Figure 19. PSD of acceleration with random vibration

6 Conclusion

In this paper, an integrated active and semi-active seat suspension for heavy duty vehicles has been proposed, built and tested. The proposed control algorithm for the seat suspension prototype applied measureable variables in practical, namely, relative displacement and seat acceleration. The integrated seat suspension has an active motor and a semi-active MR damper; gearboxes are applied to amplify the actuators torque output. The field, amplitude, and frequency-dependent performance of the seat suspension have been tested with MTS system. The tests results show that the semi-active actuator of the integrated seat suspension is controllable. The simulation and experiment were implemented and both validated the proposed integrated seat suspension and controller. In the random vibration experiment, the RMS acceleration of integrated control seat suspension has 47.7%, 33.1% and 26.5% reductions when compared with power on, power off and semi-active control seat suspension, respectively. The acceleration PSD value further indicates that the integrated seat suspension has best performance with random vibration. With the proposed seat suspension, the MR damper can suppress resonance vibration with low power consumption, and then a small active force can further isolate the resonance vibration; when vibration frequencies are above resonance frequency, the MR damper is difficult to further improve the ride comfort, then the small active force can also improve the vibration isolation performance. This integrated seat suspension has a practical value in engineering application.

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