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Vibration control of an energy regenerative seat suspension with variable external resistance

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Abstract

In this paper, an energy regenerative seat suspension with a variable external resistance is proposed and built, and a semi-active controller for its vibration control is also designed and validated. The energy regenerative seat suspension is built with a three-phase generator and a gear reducer, which are installed in the scissors structure centre of the seat suspension, and the vibration energy is directly harvested from the rotary movement of suspension's scissors structure. The electromagnetic torque of the semi-active seat suspension actuator is controlled by an external variable resistor. An integrated model including the seat suspension's kinematics and the generator is built and proven to match the test result very well. A simplified experimental phenomenon model is also built based on the test results for the controller design. A state feedback H_∞ controller is proposed for the regenerative seat suspension's semi-active vibration control. The proposed regenerative seat suspension and its controller are validated with both simulations and experiments. A well-tuned passive seat suspension is applied to evaluate the regenerative seat's performance. Based on ISO 2631-1, the frequency-weighted root mean square (FW-RMS) acceleration of the proposed seat suspension has a 22.84% reduction when compared with the passive one, which indicates the improvement of ride comfort. At the same time, the generated RMS power is 1.21 W. The proposed regenerative seat suspension can greatly improve the driver's ride comfort and has the potential to be developed to a self-powered semi-active system.

Keywords: semi-active control; energy harvesting; energy regenerative; vibration control; seat suspension.

1. Introduction

As the demands for the driver's ride comfort and health are increasing, the vehicle seat suspension design and control are widely studied in recent years. Semi-active and active seat suspensions are proposed to replace the conventional passive seat suspension [1-4]. There is no doubt that the active seat suspension has best performance in improving ride comfort, however, the high energy consumption is still the main issue for its practical application. On the contrary, the semi-active seat suspension consumes less energy and provides acceptable performance; but the conventional semi-active seat suspension, for example, the magnetorheological (MR) damper seat suspension, still needs

considerable extra energy to control its damping variation. So, an energy regenerative seat suspension will be a good option for future vehicles, especially for electrical vehicles.

The regenerative vehicle suspensions, which can harvest energy from the road vibration, are being studied recently [5-9]. Two kinds of motors or generators are applied, namely the rotary one and the linear one. Generally, the regenerative vehicle suspension with rotary motor needs a mechanism, such as rack and pinion, to transfer the linear suspension movement to rotary movement [10, 11]. On the contrary, the linear generator can directly harvest the vibration energy [12-14]; but, with a given space, the rotary generator is capable of generating more power [15]. A regenerative mechatronic damper is proposed for vehicular applications [9]; it applied a three-phase full-bridge boost converter which has been widely applied in motor applications to control the current [16]. It is believed that the regenerative suspensions should be combined with the energy harvesting and vibration control for their promising prospect [17]. Shi et al. [18] proposed a semi-active energy regenerative suspension and studied the ride comfort improvement with experiment; but an additional adjustable shock absorber is needed. For providing enough damping force, the MR and electrorheological (ER) fluid based regenerative vehicle suspensions are proposed [19, 20]. Because the seat suspension requires less damping force than the vehicle suspension, with a careful design, the electromagnetic force will be enough for vibration isolation [21].

In this paper, the vibration control of a regenerative seat suspension with variable resistance is studied. The contributions of this paper are listed as following:

- An energy regenerative seat suspension is designed and built with a three-phase AC motor (generator) and a gear reducer, which are installed in the centre of the scissors structure of a seat suspension. The rotary torque of the generator can be transformed to a vertical force with the original seat structure instead of using additional transmission devices.
- A semi-active control system prototype is built for the proposed regenerative seat suspension. The suspension's damping and stiffness are controllable by controlling a variable external resistance. Rotating a rheostat requires few energy consumption to overcome its very small friction torque and considering the energy regenerative characteristic, the proposed system has the potential to be a self-powered system.
- The regenerative seat suspension is comprehensively tested and the result is fitted with an integrated model including the seat suspension's kinematics and the generator.
- A simplified phenomenon model is built based on tests for controller design. Then, a semi-active state feedback H_∞ controller is proposed for the regenerative seat suspension, and it is validated with both numerical simulations and practical experiments.

The rest of the paper is organized as following: Section 2 presented the semi-active regenerative seat suspension system; Section 3 proposed a controller for the seat suspension; the simulation and experimental results are presented in Section 4; Finally, Section 5 presents the conclusions of this research.

2. Semi-active regenerative seat suspension system

2.1 System design and prototype building

A regenerative seat suspension system schematic diagram is shown in Fig. 1, where the vibration energy is stored in a battery by an energy storage circuit. The regenerative shock absorber produces electromagnetic force when it harvests energy from vibration. In this paper, the electromagnetic force can be controlled for isolating vibration and the regenerative shock absorber can be taken as a semi-active actuator. Generally, a rectifier is applied to transform the induced alternating current (AC) to direct current (DC) for the easy storage of energy. The principle of this semi-active system is shown in Fig. 2, where the regenerative shock absorber is working as an electromagnetic generator which has internal resistance and internal inductance. The electromagnetic force can be controlled by varying the external resistance of the circuit.

Fig. 1. Schematic of the regenerative seat suspension system.

Fig. 2. Semi-active electromagnetic generator model.

A regenerative seat suspension prototype is designed and built based on a modification of a normal passive seat suspension (GARPEN GSSC7) for heavy duty vehicles (see Fig. 3). The proposed regenerative seat suspension removed the original damper in Fig. 3(c), and installed an electromagnetic generator on the centre of left side scissors structure instead. With this modification, the rotary torque generated by the generator is transformed to a vertical force without additional transmission device (rotary movement to linear movement) [4]. A cam mechanism in the seat suspension is applied to transform the horizontal spring force to a vertical one. When the suspension is loaded, the cam, which is fixed with one of the scissors structure's bars, can push the follower to move along its guide, and then the spring is extended and the corresponding spring force is generated to support the external load. Considering that, with the same volume, the AC motor has higher power than DC motor; in this paper, the generator is composed of a 400 W three-phase motor

(MSMJ042G1U) and a gear reducer which can amplify the rotation of scissors structure and the electromagnetic torque output of the generator.

(a)

(b)

(c)

Fig. 3. Seat suspensions. (a) Semi-active regenerative seat suspension schematic. (b) Semi-active regenerative seat suspension prototype. (c) Passive seat suspension

The semi-active control system is built as Fig. 4. When seat suspension has relative movement under vibration, the three-phase AC will be generated and it is converted to DC by a three-phase rectifier. For easy implementation, a rotary rheostat is applied to vary the circuit external resistance in this paper, although there are circuits that can change the loop impedance. The resistance of the rotary rheostat is controlled by a motor which consumes very low energy to accurately rotate the rheostat. Comparing with other semi-active seat suspensions such as MR and ER damper seat suspensions, the energy consumption of the proposed seat suspension is much smaller. In [22, 23], for controlling a MR damper, the maximum current 1 A is required to energize a coil, while the proposed system just need energy to overcome a small friction torque of the rheostat. The energy storage components and circuit have not been included in this system, because this paper is focusing on investigating the vibration isolation potential of the proposed regenerative seat suspension.

Fig. 4. Semi-active vibration control system.

2.2 System model

The circuit model with three-phase diode bridge rectifier is complicated [24]; considering that this paper just aims to **emphasize** on vibration control, thus a simplified equivalent model, which is proven to be effective for describing the system, is applied (see Fig. 2). The system model can be fully built by combining the kinematics model of the seat suspension (see Fig. 5) and the semi-active electromagnetic generator model. The kinematics model of the seat suspension is defined as:

$$\dot{\theta}(t) = \frac{d}{dt} \left(2 \arcsin \left(\frac{H(t)}{L_0} \right) \right) = \frac{2\dot{h}(t)}{\sqrt{L_0^2 - H(t)^2}} \quad (1)$$

$$H(t) = h_0 + h(t) \quad (2)$$

$$\omega(t) = r_g \dot{\theta}(t) \quad (3)$$

where h_0 is the initial suspension height; $h(t)$ is the suspension relative movement displacement which can be measured in real time; L_0 is the bar length of scissors structure; $\dot{\theta}$ is the rotational rate of scissors structure centre; r_g is the gear reducer rate; ω is the rotational rate of the generator .

Fig. 5. Kinematics model of the seat suspension.

The BEM voltage U_{emf} is proportional to rotation speed ω with BEM voltage constant k_e :

$$U_{emf}(t) = k_e \omega(t). \quad (4)$$

Based on Kirchhoff's voltage laws, we can get:

$$U_{emf}(t) = L \frac{di}{dt} + i(t)(R_i + R_e) \quad (5)$$

where L is the equivalent internal inductance; R_i and R_e are the equivalent internal resistance and variable external resistance, respectively; i is the DC current through external resistance.

The current i in the motor will produce torque:

$$T_i(t) = k_T i(t) \quad (6)$$

where k_T is the torque constant. In the motor, we have $k_T = k_e$.

Considering the inertia of the motor rotor J_m , we have:

$$T_m(t) = T_i(t) + J_m \dot{\omega} \quad (7)$$

where T_m is the input mechanical torque on the motor.

The seat suspension upper platform is loaded with force F_t which follows the relationship:

$$-F_t(t) = \frac{2T_m(t)r_g}{\sqrt{L_0^2 - H(t)^2}} + f_r(t) + k_s h(t) \quad (8)$$

where k_s is the seat suspension spring stiffness; the inner friction of the suspension is simply defined as $f_r = f_0 \text{sgn}(\omega)$ where f_0 is the coulomb friction parameters.

2.3 System test and parameters identification

The dynamic properties of the regenerative seat suspension were tested by a MTS machine (Load Frame Model: 370.02, MTS Systems Corporation), and the voltage of the external resistance was measured with an NI myRIO as shown in Fig. 6. The seat suspension is fixed between the upper and lower grippers of the MTS machine; the upper gripper can force the suspension upper platform to move as desired sinusoidal routine. A load sensor is mounted on the lower gripper. The measured force can be defined as:

$$F = -F_t + f_b \quad (9)$$

where f_b is the test bias force.

(a)

(b)

Fig. 6. Model test system. (a) MTS system. (b) NI myRIO.

The external resistance-dependent, frequency-dependent and amplitude-dependent tests were implemented, respectively. Based on the test results, the parameters in the regenerative seat suspension model are identified as Table 1.

Table 1. Parameters identification.

Internal resistance R_i	5 Ohm
Voltage constant k_e	0.45 Vs/rad
Torque constant k_T	0.45 Nm/A
Internal inductance L	0.01 H
Rotor inertia J_m	0.26e-4 kg * m ²
Gear reducer ratio r_g	20
Coulomb friction constant f_0	60 N
Length of bar L_0	0.287 m
Initial suspension height h_0	0.01 m
Spring stiffness k_s	4600 N/m
Test bias force f_b	-952.87 N

The external resistance-dependent performance of the seat suspension is show in Fig. 7, when the external resistance was set with different values (0 Ohm, 3 Ohm, 5 Ohm, 10 Ohm and 50 Ohm) at constant amplitude (20 mm) and frequency (2 Hz) sinusoidal load. With the identified parameters, the experiment result (marking as ‘_exp’) matches the simulation result (marking as ‘_sim’) very well. As the area of the enclosed force-displacement loops indicate the system damping, the result shows that the seat suspension damping is decreasing when the external resistance is increased.

Fig. 7. External resistance-dependent test (amplitude 20 mm, frequency 2 Hz).

The frequency-dependent and amplitude-dependent test results further verify the accuracy of the regenerative seat suspension model (see Fig. 8 and Fig. 9). When higher frequency sinusoidal movement is loaded with same amplitude and external resistance, the measured force will increase. Similarly, with the same external resistance and same frequency sinusoidal movement, the area of the enclosed force-displacement loop is increasing with the movement amplitude. The results indicate that, with same external resistance, the BEM force will increase when the suspension relative velocity is increased.

Fig. 8. Frequency-dependant test (amplitude 20 mm, external resistance 4 Ohm).

Fig. 9. Amplitude-dependant test (external resistance 4 Ohm, frequency 2 Hz).

The voltage of the external resistance is defiend as:

$$U_{load}(t) = i(t)R_e \quad (10)$$

So, the **experimentally** measured U_{load} can also match the simulation result with the identified parameters, as shown in Fig. 10. The measured voltage magnitude is increasing with the external resistance value.

Fig. 10. Voltage of external resistance (amplitude 20 mm, frequency 2 Hz).

The test and simulation results indicate that the regenerative seat suspension model (Eq. 8) can effectively describe the regenerative seat suspension prototype, and the BEM force of the motor can be controlled by a variable external resistance.

3. Controller design

3.1 Model simplification

The fundamental principle of the semi-active control is to vary the suspension between ‘soft’ and

‘hard’ under different vibration situations. The proposed regenerative seat suspension can vary its electromagnetic force by its external resistance which is designed to vary from 0 Ohm to 50 Ohm; accordingly, the rheostat needs to be rotated from 0 degree to 100 degree. When the external resistance is set as 50 Ohm, the suspension is in the softest state; and when it is set as 0 Ohm, the suspension becomes hard. For simplifying the controller design, the regenerative seat suspension model is defined as Fig. 11 where m is the mass loaded on the suspension; k_0 and c_0 are the suspension stiffness and damping with 50 Ohm external resistance, respectively; Δk and Δc are the variable stiffness and damping, respectively; z_s is the seat displacement and z_v is cabin floor displacement.

Fig. 11. Semi-active seat suspension model.

The suspension equivalent stiffness and damping are identified with the external resistance-dependent test result as shown in Fig. 11 and Fig. 12. Thus, the relationship between suspension stiffness and damping with external resistance are defined as:

$$k = k_0 + \left(\frac{a_k}{b_k + R_e} + c_k \right) = k_0 + \Delta k \quad (11)$$

$$c = c_0 + \left(\frac{a_c}{b_c + R_e} + c_c \right) = c_0 + \Delta c \quad (12)$$

where a_k , b_k and c_k are parameters for stiffness fitting; a_c , b_c and c_c are parameters for damping fitting. Table 2 shows the identified parameters. The fitting results are shown in Fig. 12 and Fig. 13. When the external resistance is bigger than 5 Ohm, the stiffness variation is very small; by this reason, in some research [9], the stiffness variation caused by the electromagnetic force is ignored. The damping can be changed from 136.16 Nm/s to 801.57 Nm/s when the external resistance is varied from 50 Ohm to 0 Ohm.

Table 2. Parameters of sea suspension stiffness and damping.

k_0	4500	c_0	136.16
a_k	3592	a_c	4065
b_k	2.345	b_c	5.292
c_k	-99.94	c_c	-78.73

Fig. 12. Stiffness coefficients with different external resistance.

Fig. 13. Damping coefficients with different external resistance.

The seat suspension model is defined as:

$$m\ddot{z}_s = -k_0(z_s - z_v) - c_0(\dot{z}_s - \dot{z}_v) - F_d + u \quad (13)$$

$$F_d = f_r + \Delta u \quad (14)$$

where u is force output when the external resistance is controlled; F_d is the total disturbance; Δu is the uncertainty of force output which is caused by the model simplification.

3.2 State feedback H_∞ controller

Many reputable suspension control algorithms have been proposed [25, 26]; in this section, an H_∞ controller is designed for the simplified model. The state variables are chosen as $x_1 = z_s - z_v$, $x_2 = \dot{z}_s$. There are two disturbances, i.e., $d_1 = \dot{z}_v$, $d_2 = F_d$. Thus, the suspension model can be written as state-space form:

$$\dot{\mathbf{X}} = \mathbf{A}\mathbf{X} + \mathbf{B}_1\boldsymbol{\omega} + \mathbf{B}_2u \quad (15)$$

where $\mathbf{X} = [x_1 \ x_2]^T$, $\mathbf{A} = \begin{bmatrix} 0 & 1 \\ -\frac{k_0}{m} & -\frac{c_0}{m} \end{bmatrix}$, $\mathbf{B}_1 = \begin{bmatrix} -1 \\ \frac{c_0}{m} \end{bmatrix}$, $\boldsymbol{\omega} = [d_1 \ d_2]^T$, $\mathbf{B}_2 = [0 \ \frac{1}{m}]^T$.

For the seat suspension design, the seat acceleration should be controlled. At the same time, the seat

suspension deflection also needs to be controlled to ensure the driver can easily handle the operational devices because those devices, such as steering wheel and gear shift lever, are always fixed on the vehicle cabin floor. The controlled output is defined as:

$$\mathbf{Z} = \mathbf{C}_1 \mathbf{X} + \mathbf{D}_{11} \boldsymbol{\omega} + \mathbf{D}_{12} u \quad (16)$$

where $\mathbf{Z} = [z_s - z_v \ddot{z}_s]^T$, $\mathbf{C}_1 = \begin{bmatrix} \alpha & 0 \\ -\frac{k_0}{m} & -\frac{c_0}{m} \end{bmatrix}$, $\mathbf{D}_{11} = \begin{bmatrix} 0 & 0 \\ \frac{c_0}{m} & -\frac{1}{m} \end{bmatrix}$, $\mathbf{D}_{12} = [0 \ \frac{1}{m}]^T$, α is positive weighting constant.

Considering an H_∞ performance criterion under zero initial condition:

$$\int_0^\infty (\mathbf{Z}^T \mathbf{Z} - \gamma^2 \boldsymbol{\omega}^T \boldsymbol{\omega}) dt < 0 \quad (17)$$

where γ is the desired level of disturbance attenuation, a state feedback control is designed as:

$$u = \mathbf{K} \mathbf{X} \quad (18)$$

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

By referring to [27], if there exists a matrix $\mathbf{P} = \mathbf{P}^T > 0$, such that the following linear matrix inequality (LMI) is satisfied:

$$\begin{bmatrix} * + \mathbf{P}(\mathbf{A} + \mathbf{B}_2 \mathbf{K}) & * & * \\ \mathbf{B}_1^T \mathbf{P} & -\gamma^2 \mathbf{I} & * \\ \mathbf{C}_1 + \mathbf{D}_{12} \mathbf{K} & \mathbf{D}_{11} & -\mathbf{I} \end{bmatrix} < 0 \quad (19)$$

then the system (15) is stable with H_∞ disturbance attenuation $\gamma > 0$.

Pre- and post-multiplying (19) by $\text{diag}(\mathbf{P}^{-1}, \mathbf{I}, \mathbf{I})$ and its transpose, respectively, and defining $\mathbf{Q} = \mathbf{P}^{-1}$, $\mathbf{W} = \mathbf{K} \mathbf{Q}$, the following LMI is obtained:

$$\begin{bmatrix} * + (\mathbf{A} \mathbf{Q} + \mathbf{B}_2 \mathbf{W}) & * & * \\ \mathbf{B}_1^T & -\gamma^2 \mathbf{I} & * \\ \mathbf{C}_1 \mathbf{Q} + \mathbf{D}_{12} \mathbf{W} & \mathbf{D}_{11} & -\mathbf{I} \end{bmatrix} < 0 \quad (20)$$

By solving (20) with MATLAB LMI toolbox, the controller gain $\mathbf{K} = \mathbf{W} \mathbf{Q}^{-1}$ is obtained.

3.3 Controller implementation

The controller implementation is shown in Fig. 14 where a desired force is calculated out by the designed controller firstly, and then the desired external resistance is obtained with the simplified model.

Fig. 14. Controller implementation.

Because the external resistance can only be varied from 0 Ohm to 50 Ohm, Considering that Eqs. 11 and 12 are both monotonically decreasing, the two limitations of the generator's force output are defined as:

$$u_{lim1} = 0 \quad (21)$$

$$u_{lim2} = -\left(\frac{a_k}{b_k} + c_k\right)h(t) - \left(\frac{a_c}{b_c} + c_c\right)\dot{h}(t) \quad (22)$$

where when external resistance is 50 Ohm, the output force is u_{lim1} , and when external resistance is 0 Ohm, the output force is u_{lim2} .

The electromagnetic force of the seat suspension is generated passively. When force directions of u_{lim2} and u are different, the 50 Ohm external resistance should be applied. When force directions of u_{lim2} and u are identical and $|u| > |u_{lim2}|$, the 0 Ohm external resistance should be applied. So, the controlled external resistance is obtained:

$$R_e = \begin{cases} 50 & u_{lim2}u < 0 \\ 0 & u_{lim2}u > 0 \text{ and } |u| > |u_{lim2}| \\ R_{ed} & \text{others} \end{cases} \quad (23)$$

where the desired external resistance R_{ed} can be easily obtained by solving:

$$u = -\left(\frac{a_k}{b_k + R_{ed}} + c_k\right)h(t) - \left(\frac{a_c}{b_c + R_{ed}} + c_c\right)\dot{h}(t) \quad (24)$$

When the R_e is obtained, the rheostat will be rotated to the corresponding angle.

4 Evaluation

4.1 Numerical simulations

In this section, the proposed controller is validated with the identified regenerative seat suspension model in numerical simulation. The H_∞ controller gain is designed as $\mathbf{K} = [4600 \quad -600]$. The harmonic excitation test, which was a sweep frequency signal from 1 Hz to 3 Hz in 40 seconds with 30 mm amplitude, was implemented. Fig. 15 shows the seat absolute displacement comparison. When the external resistance is 50 Ohm, the suspension damping is small; big displacement appears around resonance frequency. When the external resistance is turned to 0 Ohm, the seat suspension has the high damping and stiffness; it can successfully suppress the resonance peak, but it has worse vibration isolation performance than the 'soft' one (50 Ohm) in high frequency. Obviously, the proposed semi-active H_∞ controller with simplified model has best performance. Because, for the conventional passive seat suspension, the damping has to compromise between the soft one which is comfortable and the hard one which can keep the suspension stable in resonance frequency, this simulation result indicates that the regenerative seat suspension with the proposed controller can improve the ride comfort greatly by combining the advantages of soft and hard seat suspensions.

Fig. 15. Seat absolute displacement.

The system natural frequency is defined as $\omega_n = \sqrt{k/m}$ and the damping ratio is $\zeta = c/2m\omega_n$. Thus, when the external resistance varies from 50 Ohm to 0 Ohm, the ω_n changes from 7.5 rad/s to 8.5 rad/s and ζ changes from 0.113 to 0.583. The results shows that the shift of the natural frequency is small (about 0.15 Hz), while the damping ratio varies 416%, which indicates that the damping variation makes the main contribution for the seat suspension to isolate vibration.

4.2 Experimental setup

The experimental system is shown in Fig. 16. The seat suspension is fixed on the top of a six-degree of freedom (6-DOF) vibration platform which is controlled by an NI CompactRio 9076 and can generate desired vibration in according to Computer 2's commands. The accelerations of the seat suspension base and top are acquired by two accelerometers (ACXL 203EB). A displacement sensor (Micro Epsilon ILD1302-100) is applied to measure suspension relative displacement. The displacement of vibration platform is also measured by a displacement sensor (Micro Epsilon optoNCDT 1700). By adding together the data of two displacement sensors, the seat absolute displacement can be acquired. The proposed controller is implemented on NI CompactRio 9074 which calculates out desired external resistance based on sensors data and then sends command to a motor to control the resistance of a rotary rheostat. The current though external resistance is measured out by a NI module (9227) on NI CompactRio 9074. And the voltage of external resistance is also acquired by the controller. Considering the rotary rate limitation of rheostat, the control frequency is set as 100 Hz.

The regenerative seat suspension is tested with different external resistances. Then the experiment results are compared with the semi-active control seat suspension. For further verifying the proposed semi-active seat suspension system, a well-tuned passive seat suspension (GARPEN GSSC7) is also tested.

Fig. 16. Experimental setup.

4.3 Experiment results

The sinusoidal excitations were applied to the seat suspensions with 80 kg load for testing their frequency performance. Fig. 17 shows the seat acceleration with 1.5 Hz vibration. When external resistance is varied from 50 Ohm to 0 Ohm, the suspension damping and stiffness are increasing accordingly; the suspension is becoming stiffer; therefore, the resonance vibration is suppressed. The semi-active control suspension has closely performance as the hardest one (0 Ohm external resistance). Fig. 18 shows their vibration transmissibility among the tested frequency range. When the vibration is around the resonance frequency, the softest suspension amplifies the vibration and the suspension is unstable. The semi-active control suspension can successfully suppress the resonance vibration; and its transmissibility value is just bigger than the hardest one a little. When the vibration is around 1.8 Hz, the seat suspensions is turning from amplifying vibration to isolating vibration; the semi-active seat suspension has the best performance. And in the higher frequencies, the performance of the semi-active control seat is close to the softest one which means the advantage of the soft suspension, namely ride comfort, is kept.

Fig. 17. Seat acceleration with 1.5 Hz vibration.

Fig. 18. Acceleration transmissibility.

The bump road test can indicate the controller's capacity to respond to excitation. Fig. 19 shows the seat acceleration of the well-tuned conventional passive seat suspension and the proposed semi-active one under bumpy road conditions. The peak acceleration magnitude drops from 1.652 m/s^2 to 1.265 m/s^2 ; there is a 23.4% reduction.

(a)

(b)

Fig. 19. Seat acceleration with bump road. (a) Time domain graph. (b) Zoom in.

The random excitation test is always applied to evaluate the seat suspension performance in time domain. Fig. 20 shows acceleration comparison of the semi-active control regenerative seat suspension with its soft (50 Ohm) and hard (0 Ohm) states; from Fig. 20 (b), the controlled seat suspension can suppress the resonance frequency and keep the vibration isolation ability in high frequency. Fig. 21 further displays that the resonance seat displacement with 50 Ohm external resistance is higher than the proposed system, while they have similar seat displacement at high frequency vibration. The performance comparison with conventional passive seat suspension is shown in Fig. 22 where the controlled regenerative seat suspension has better performance in all the test time.

(a)

(b)

Fig. 20. Regenerative seat suspension acceleration under random road. (a) Time domain graph. (b) Zoom in.

Fig. 21. Comparison of seat displacement.

(a)

(b)

Fig. 22. Comparison with conventional seat suspension acceleration under random road. (a) Time domain graph. (b) Zoom in.

For further analysing those suspensions' performance, the acceleration root mean square (RMS) is computed and ISO 2631-1 standard is applied to evaluate the ride comfort. The frequency weighted RMS (FW-RMS) acceleration is obtained based on the ISO 2631-1 recommended frequency-weighting curve which is related to ride comfort. The fourth power vibration dose value (VDV) is another evaluation method which is more sensitive to peaks than FW-RMS method. The seat effective amplitude transmissibility (SEAT) and VDV ratio are obtained as follows:

$$a_w = \left[\frac{1}{T} \int_0^T \{a_w(t)^2 dt\} \right]^{1/2}, \quad VDV = \left[\int_0^T \{a_w(t)^4 dt\} \right]^{1/4},$$

$$SEAT = \frac{a_{w,driver}}{a_{w,vibration}}, \quad VDV \text{ ratio} = \frac{VDV_{driver}}{VDV_{vibration}} \quad (25)$$

Table 3 shows the comparison of evaluation parameters for each seat suspensions. The RMS and FW-RMS acceleration of the passive one is between the uncontrolled soft and hard regenerative seat suspension; but its VDV value is smaller than both of them. This proves the passive seat suspension with nonlinear damper is well tuned. The semi-active controlled seat has best performance in all the

evaluation parameters. For clearly showing the performance improvement, Table 4 shows the vibration reduction percentage of the semi-active seat suspension comparing with other three. The 22.84% reduction of FW-RMS when comparing with the passive seat suspension validates the effectiveness of the proposed semi-active regenerative seat suspension.

Table 3. Seat vibration evaluation

	0 Ohm	50 Ohm	Passive	Semi-active
RMS (m/s^2)	0.9884	0.8496	0.9473	0.7694
FW-RMS (m/s^2)	0.7372	0.6056	0.6922	0.5341
VDV ($\text{m/s}^{1.75}$)	2.072	2.43	1.962	1.451
SEAT	0.6594	0.5417	0.6191	0.4777
VDV ratio	0.5351	0.6276	0.5067	0.3747

Table 4. Vibration reduction percentage of Semi-active seat suspension

	0 Ohm	50 Ohm	Passive
RMS (m/s^2)	-22.16%	-9.44%	-18.78%
FW-RMS (m/s^2)	-27.55%	-11.81%	-22.84%
VDV ($\text{m/s}^{1.75}$)	-29.97%	-40.29%	-26.04%
SEAT	-27.55%	-11.81%	-22.84%
VDV ratio	-29.97%	-40.29%	-26.04%

The current through the variable external resistance is shown in Fig. 23 where the maximum current is 1.267 A and the RMS current is 0.152 A. The generated power is defined as $P = U_e I_e$ where U_e is the voltage of external resistance and I_e is the current through it. Fig. 24 shows the generated power when the suspension is controlled; there is a maximum power of 13.88 W and the RMS power is about 1.21 W which shows the energy harvesting potential of the regenerative seat suspension. In view of that the system only consumes very few energy to overcome a small friction torque of the rheostat in order to vary the resistance, it has the capacity to be a self-powered one.

Fig. 23. Generated current of semi-active control.

Fig. 24. Generated power of semi-active control.

5. Conclusion

In this paper, an energy regenerative seat suspension with variable external resistance is proposed and built, and the designed semi-active controller is validated by both simulations and experiments. This energy regenerative seat suspension applied a three-phase rotary generator with a gear reducer to harvest the vibration energy from the seat's scissors structure. For controlling the electromagnetic force, a rotary rheostat is applied to vary circuit external resistance. The external resistance-dependent, frequency-dependent and amplitude-dependent tests were implemented, respectively; the integrated mathematical model including the seat suspension and generator can match the result very well. An experimental phenomenon model is also built for the controller design. A semi-active state feedback H_∞ controller is designed for this seat suspension, and it is validated with simulations and experiments. A well tuned passive seat suspension is applied to evaluate the performance improvement. Based on ISO 2631-1, the FW-RMS acceleration of the semi-active controlled

regenerative seat suspension has a 22.84% reduction when compared with the conventional passive seat under random vibration. This indicates a great improvement of ride comfort. At the same time, there are 1.21 W of RMS power can be harvested. The proposed regenerative seat suspension and controller can successfully improve the driver's ride comfort and harvest energy.

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